The 6th International Supercritical CO_2 Power Cycles Symposium
March 27 - 29, 2018, Pittsburgh, Pennsylvania

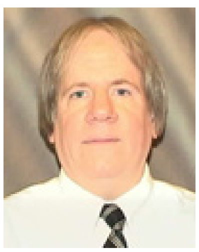
sCO_2 Cycle as an Efficiency Improvement Opportunity for Air-fired Coal Combustion

Charles W. White
Principal Engineer
KeyLogic
Morgantown, WV USA

Walter W. Shelton
Engineer
National Energy Technology Laboratory
Morgantown, WV, USA

Nathan T. Weiland
Engineer
National Energy Technology Laboratory
Pittsburgh, PA USA

Travis R. Shultz
Supervisor- Energy Process Analysis Team
National Energy Technology Laboratory
Morgantown, WV, USA



Charles White is a principal engineer at KeyLogic, 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 has received a BS, MS (University of Virginia) and PhD (University of Pennsylvania) in Chemical Engineering.

Walter Shelton is an engineer in the Systems Engineering & Analysis group at the National Energy Technology Laboratory (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 (University of Minnesota) in Chemical Engineering.

Dr. Nathan Weiland is an engineer in the Systems Engineering & Analysis group at the National Energy Technology Laboratory (NETL), 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.

Mr. Shultz serves as the supervisor of the Energy Process Analysis Team (EPAT), within the National Energy Technology Laboratory's Systems Engineering & Analysis Directorate. EPAT conducts plant-level process and cost engineering analyses to assess the potential thermodynamic, environmental, and economic performance of commercial and advanced technologies, and compare the latter with existing and competing technologies. These analyses also serve to help identify appropriate R&D targets for cost and performance. He possesses BS and MS degrees in Mechanical Engineering from West Virginia University.

ABSTRACT

For most of the last century, coal has been the fuel of choice to satisfy the United States' need for baseload electricity generation. Low natural gas prices and increased market penetration of low marginal cost, non-dispatchable renewable power sources are forcing the electric power industry to scrutinize the long-term viability of both existing and new coal-fueled assets. Advanced technologies that offer high efficiency, low emissions, and improved flexibility may permit coal to remain a competitive component of the Nation's energy portfolio.

The current study objective is the techno-economic assessment of greenfield indirect supercritical carbon dioxide (sCO_2) power plants - based on an air-fired coal combustion process - that demonstrate improvement in efficiency, cost of electricity (COE), and emissions over state-of-the-art (SOA) commercial technologies. The approach develops initial process configurations from reference cases selected from existing NETL studies on oxy-fired coal indirect sCO_2 systems. Configurations are evaluated at a turbine inlet temperature of 760 °C and pressure of 34.5 MPa. Multiple process configurations are examined, including the use of reheat (up to two stages) and intercooling (up to two stages), among other improvements. Sensitivity analyses are performed, including the minimum temperature approach in recuperators, cycle pressure ratio, and assumed cycle pressure drops. These sensitivity analyses provide insight into system behavior and provide a quantitative basis for setting design targets.

INTRODUCTION

Coal-Fired Power Generation

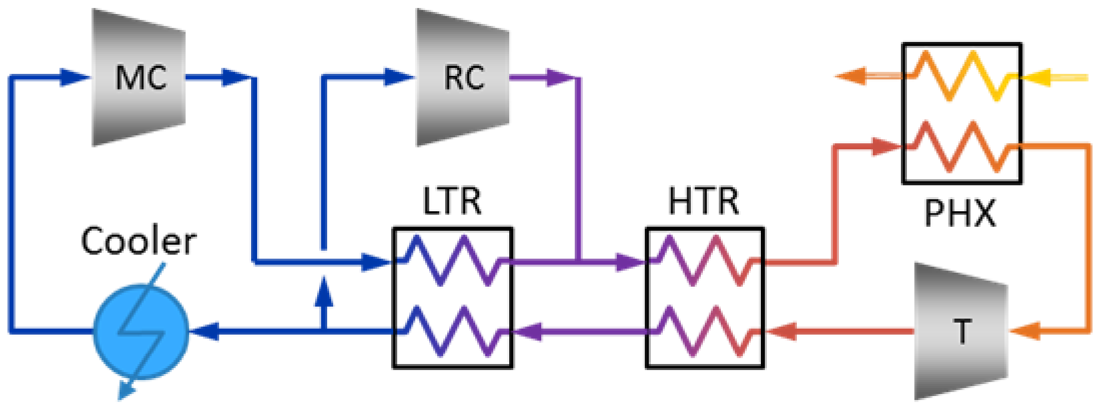
For most of the last century, coal has been the fuel of choice to satisfy the United States' need for baseload electricity generation. Low natural gas prices and increased market penetration of low marginal cost, non-dispatchable renewable power sources are forcing the electric power industry to scrutinize the long-term viability of both existing and new coal-fueled assets. Advanced technologies that offer high efficiency, low emissions, and improved flexibility may permit coal to remain a competitive component of the Nation's energy portfolio.

Coal-based power generation systems incorporating advanced power cycles offer numerous advantages over state-of-the-art supercritical/ultrasupercritical Rankine cycles. These include the likelihood of significantly higher efficiencies, lower cost of electricity (COE), and the possibility of more cost-effective compliance with increasingly-stringent emissions requirements. Finally, some advanced technology cycles in various phases of research and development offer the possibility of dramatic reductions in water consumption and cost-effective, efficient operation in arid regions without water cooling. One such advanced cycle receiving increased attention is the closed (indirect) Brayton cycle using supercritical CO_2 (sCO_2) as the working fluid. Beyond coal, the indirect sCO_2 cycle holds promise for other electric power generation applications, including nuclear and concentrated solar. In addition to utility scale applications, the indirect cycle may be used at smaller scales, such as waste heat recovery from industrial-scale combustion turbines.

Overview of Indirect sCO_2 Power Cycles

Figure 1 is a simplified block flow diagram depicting the recompression closed Brayton sCO_2 power cycle. This is one of the most well-studied and highest performing of the indirect sCO_2 power cycles, particularly for heat sources that operate over a relatively narrow temperature range. In this cycle a primary heat exchanger (PHX) is used to heat the high-pressure CO_2 working fluid to the turbine inlet temperature. The CO_2 expands and partially cools through the turbine. However, most of the enthalpy in the working fluid remains in the turbine effluent - this energy is exchanged with the recycled high-pressure CO_2 leaving the compressors via recuperative heat exchangers. This recuperation step is a significant contributor to high cycle efficiency. Because of a mismatch in thermal capacitance between the high-pressure and low-pressure CO_2, a high recuperator effectiveness cannot be achieved unless a portion of the low-pressure working fluid bypasses the main CO_2 cooling step. This bypass compression step is the key feature in the recompression sCO_2 Brayton cycle.

Figure 1 Recompression sCO_2 power cycle



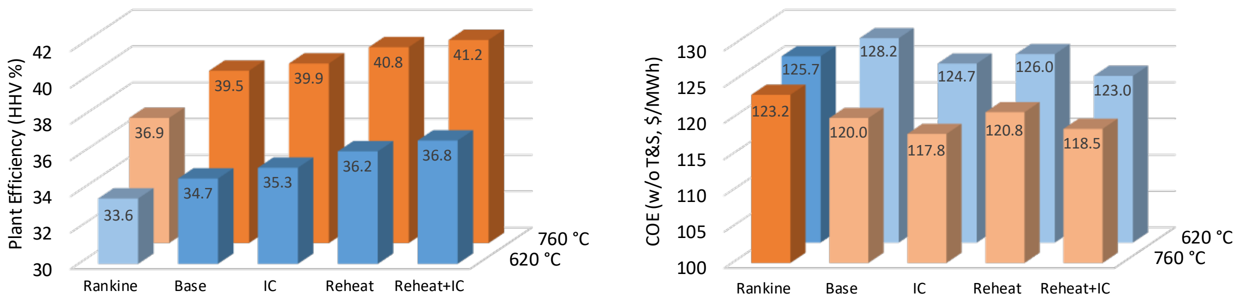
With potential applications using such diverse fuel sources as nuclear, solar, waste or cultivated renewables, or fossil fuels, this cycle has received extensive attention in the literature. Many facets of NETL's R&D portfolio have been focused on this cycle.

NETL Coal-fired Oxy-CFB Indirect sCO_2 Power Cycle Study

A recent NETL report examined the cost and performance of a baseline coal-fired oxy-CFB power plant with carbon capture, incorporating the recompression closed Brayton sCO_2 power cycle [1]. Two different turbine inlet temperatures (TIT) were used: one at 620 °C (similar to ultrasupercritical (USC) Rankine cycle conditions) and the other at 760 °C (approximating advanced ultra-supercritical (AUSC) Rankine cycle conditions). For both TITs, the corresponding sCO_2 plant offered a significantly higher efficiency and lower COE than plants employing Rankine cycles at operating conditions similar to the sCO_2 plants. The results were consistent with prior sensitivity analyses on indirect sCO_2 cycle sensitivity to turbine inlet temperature [2].

The study further showed that the sCO_2 plant performance could be improved by using a single turbine reheat stage and/or a single compressor intercooler stage. A summary of the cost and performance results is shown in Figure 2. The blue shaded bars correspond to a TIT of 620 °C while the red shaded bars correspond to a TIT of 760 °C. The various plant configurations are labeled as Base (Baseline Case), IC (uses main compressor intercooling), Reheat (uses a single turbine reheat stage) and Reheat+IC (uses both turbine reheat and main compressor intercooling). Note that at the highest TIT, the results suggest that adding reheat to a configuration with main compressor intercooling may not be cost effective. Study results when compared with a reference supercritical (near ultrasupercritical) Rankine pulverized coal (PC) plant shown in Figure 3, indicate (for TIT of 620 °C) 1.1 – 3.2 percentage points increase in plant efficiency and 3.2 - 5.4 $/MWh reduction in the COE. For the higher TIT (760 °C), the changes are 2.6 – 4.3 percentage points increase in plant efficiency and 2.5 – 2.7 $/MWh reduction in the COE.

Figure 2 Cost and performance summary for oxy-CFB indirect sCO_2 power plants [1]



Literature Review

A number of prior studies have examined the performance and in some cases the cost of non-capture coal-fired power plants based on various configurations of the sCO_2 power cycle. Results for some of these cases are included below, with the comparison to the reference cases used. These results are only qualitatively comparable due to the varying assumptions employed by the studies.

The study by Mecheri and Moullec [3] evaluates the thermodynamic performance of multiple sCO_2 power cycle configurations in conjunction with an air-fired PC combustor heat source. The key conclusions were that a recompression Brayton cycle is necessary for the sCO_2 power cycle to attain a higher net plant efficiency than can be obtained from a power plant based on the Rankine cycle, that single reheat offers a further 1.5 percentage point efficiency increase, and that more complex cycle variations such as double reheat and double recompression offer relatively small improvements in efficiency. The final conceptual design had an overall efficiency of 47.8% (LHV basis), 2.4 percentage points higher than the state-of-the-art reference power plant based on the Rankine cycle.

The study by Park et. al. [4] compared the cost and performance of three coal-fired power plants based on differing sCO_2 Brayton power cycle configurations. This study found a 6.2–7.4% increase in plant efficiency compared to a power plant based on a conventional steam Rankine cycle, along with a reduction in levelized COE (LCOE) of about 7.8–13.6%.

Bai et. al. [5] describe a coal-fired power plant based on a novel sCO_2 Brayton cycle with a parallel HTR/economizer arrangement. This cycle consists of a recompression Brayton cycle with an intercooled main CO_2 compressor and a take-off of high pressure recycle CO_2 from the highest temperature portion of the HTR to a low temperature boiler economizer (termed a bleeding anabranch). Their results show that the proposed power plant can achieve a gross cycle efficiency of 52.3% and a net efficiency of 49.5% (LHV basis) using turbine inlet conditions of 650 °C and 29.6 MPa. Increasing these conditions to 700 °C and 37 MPa resulted in a 2.14 percentage point increase in gross cycle efficiency.

EPRI [6] presented a cycle performance only comparison of multiple candidate sCO_2 power cycles with similar Rankine cycles. The study results showed that a recompression Brayton cycle using sCO_2 can achieve approximately 3.5 percentage points higher cycle efficiency than a comparable Rankine cycle operating at turbine inlet temperatures near 700°C and that a further 1 percentage point increase in efficiency might be achieved by employing a reheat turbine in the sCO_2 power cycle. Results for a combined cycle configuration (sCO_2 topping/Rankine bottoming) compared to an AUSC PC plant showed only modest improvement.

Miller et. al. [7] performed a comparison of CO_2 capture PC plants using USC and AUSC Rankine cycles to plants using multiple sCO_2 power cycle variations, including cascade and recompression cycles, as well as a hybrid concept for a recompression cycle with low grade heat recovery. Other sCO_2 cycle variations considered included multi-stage reheat/compression, air pre-heater temperature, and various cooling technologies for the CO_2 cooler. Using a multivariate nonlinear optimization design process, optimized designs for each configuration were selected and power cycle cost versus net plant efficiency plots were generated. Comparisons between sCO_2 power cycles showed that recompression cycles outperform cascade cycles and that the highest net plant efficiencies can be attained with the low-grade heat recovery option with multistage compression used in conjunction with turbine reheat. Compared to the Rankine cycles, the plants using recompression sCO_2 cycles attained a 3.3 percentage point higher efficiency at USC conditions and 4 percentage point higher efficiency at AUSC conditions.

Objectives of Current Study

The purpose of this study is to explore the potential for indirect sCO_2 power cycles to significantly increase process efficiency and lower the cost of electricity compared to conventional technology as well as other candidate advanced technologies. Increasing power plant efficiency and maintaining or improving cost competitiveness is fundamental to the research carried out and sponsored by NETL. In the near term and within the time horizon for the commercial development of power plants based on the sCO_2 power cycle, non-capture options must be explored. In addition to the obvious goals of improving power plant efficiency and reducing COE, an advanced power plant based on the sCO_2 power cycle may be able to offer a better emissions profile than other advanced non-capture processes.

Current Study Reference Cases and Study Approach

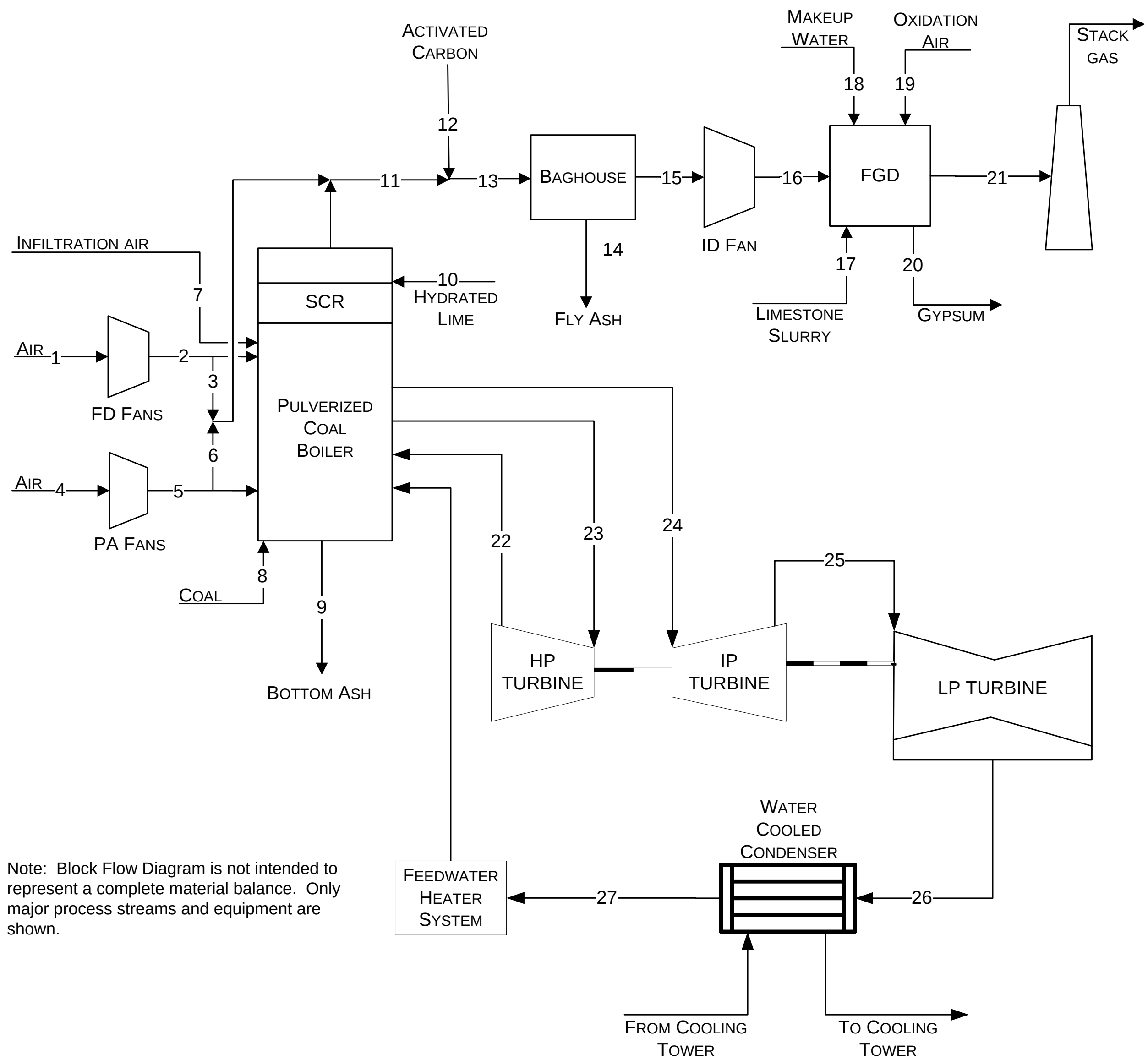
Reference Case B12A

This reference case is a supercritical (SC) PC plant which represents a commercially available plant [8]. It has a nominal output of 550 MWe and uses a single reheat 24.1 MPa/593°C/593°C Rankine cycle. Lower main steam pressure aside, these conditions result in performance near that of a state-of-the-art ultrasupercritical Rankine cycle. Figure 3 shows the block flow diagram for this process.

Coal (stream 8) and primary air (PA) (stream 4) are introduced into the boiler through the wall-fired burners. Additional combustion air, including the over-fired air is provided by the forced draft (FD) fans (stream 1). The boiler operates at a slight negative pressure so air leakage is into the boiler, and the infiltration air is accounted for in stream 7. Streams 3 and 6 show Ljungstrom air preheater leakages from the FD and PA fan outlet streams to the boiler exhaust.

Flue gas exits the boiler through the selective catalytic reduction (SCR) reactor where hydrated lime is injected (stream 10) for the reduction of SO_3. It then passes through the combustion air preheater (where the air preheater leakages are introduced) and is cooled to 143°C (289°F) (stream 11) before powdered activated carbon is injected (stream 12) for mercury reduction. The flue gas then passes through a fabric filter for particulate removal (stream 15). An induction fan (ID) fan increases the flue gas temperature to 153°C (308°F) and provides the motive force for the flue gas (stream 16) to pass through the flue gas desulfurization (FGD) unit. FGD inputs and outputs include makeup water (stream 18), oxidation air (stream 19), limestone slurry (stream 17) and product gypsum (stream 20). The clean, saturated flue gas exiting the FGD unit (stream 21) passes to the plant stack and is discharged to the atmosphere.

Figure 3 Reference Case B12A block flow diagram, supercritical unit without CO_2 capture



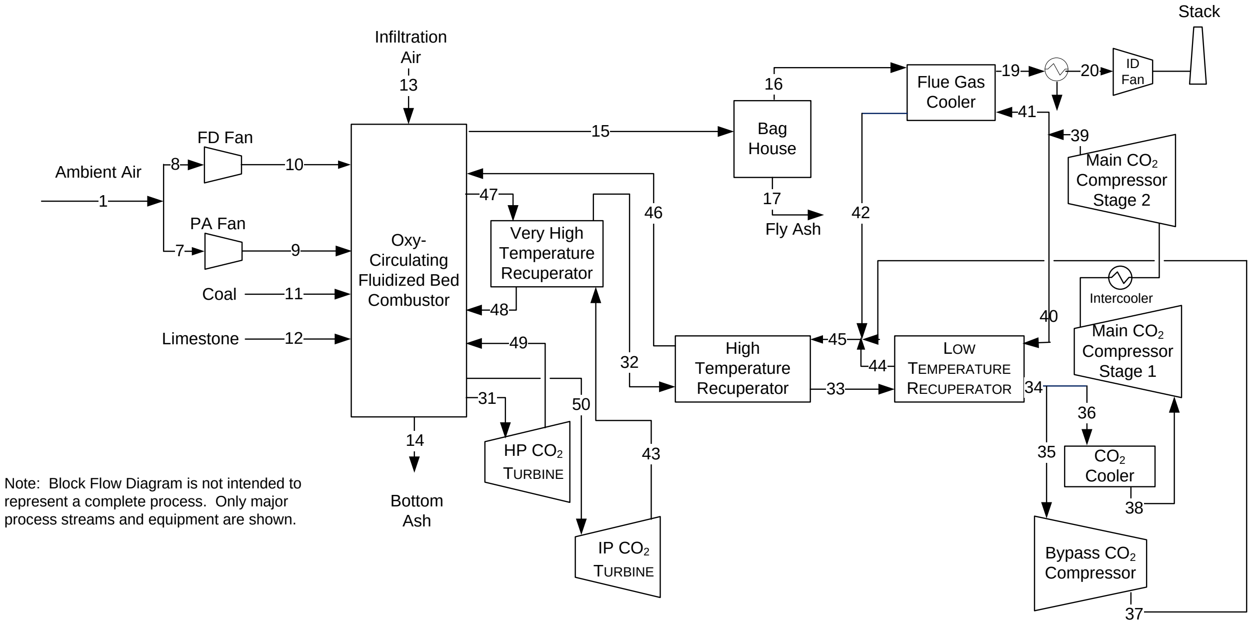
Reference AUSC Case 5

This reference case is an advanced ultra-supercritical (AUSC) PC plant which represents an even more advanced, developmental system than Case B12A [9]. The overall process is the same as depicted in Figure 3 except that the Rankine cycle operating conditions are 34.5 MPa/732°C/760°C. This case also incorporates a conceptual Downdraft Inverted Tower Boiler design developed by Babcock & Wilcox (B&W), which uses a non-conventional boiler arrangement in order to minimize the length and cost associated with the main steam and reheat piping leads.

Reference Case RhtIC760A

This case is a simple variation of the baseline indirect sCO_2 cycle conceptual design for a CO_2 capture plant with an oxy-CFB heat source utilizing a 2-stage main CO_2 compressor with intercooling and a reheat turbine having an inlet temperature of 760 °C, Case RhtIC760 [1]. In this variation, the air separation unit (ASU) and CO_2 purification unit (CPU) are removed along with the flue gas recycle. The block flow diagram for this process is shown in Figure 4.

Figure 4 Reference Case RhtIC760A block flow diagram, CFB with indirect sCO_2 power cycle



This process is similar in many ways to the PC reference cases depicted in Figure 3. The major differences are the replacement of the PC/SCR heat source with a CFB with in-bed sulfur capture, the replacement of the single reheat Rankine cycle with a recompression sCO_2 Brayton cycle with a 2-stage intercooled main CO_2 compressor and a reheat turbine, and the inclusion of a flue gas cooler (economizer) to recover low grade heat from the flue gas into the low temperature end of the sCO_2 power cycle. In addition, the power cycle includes additional high temperature heat recovery not included in a simple recompression Brayton cycle. A high temperature economizer within the CFB heats the high pressure sCO_2 (stream 46) exiting the high temperature recuperator (HTR) and this stream is heated further in a very high temperature recuperator (VTR) using the effluent from the reheat turbine stage.

Table 1 lists the CFB and power cycle operating conditions used for this case. For the final cycle configurations developed for this study and described below, the same cycle conditions were used unless explicitly stated otherwise.

Table 1 CFB and power cycle operating conditions for reference Case RhtIC760A

Section	Parameter	Reference Case RhtIC760A
CFB	Primary air fraction	0.235
	Secondary air fraction	0.765
	Bed temperature	871 °C
	Pressure drop	6.6 kPa
	Excess air	3.1%
	Infiltration air	2%
	Lime molar feed rate	2.4 times sulfur feed rate
Expander	Inlet temp	760 °C
	PR, P_{exit}	4.05, 8.51 MPa
	Isentropic efficiency	0.927
Recuperator	P_{drop} HP side	0.2%
	P_{drop} LP side	0.8%
	LTR Avg T_{app}	5.6 °C
	HTR Min Tapp	5.6 °C
CO_2 cooler	Non-cond cooler	35 °C
	P_{drop} CO_2 side	0.8%
	Cooling source	Process cooling water/Cooling tower
CO_2 main compressor intercoolers	Non-cond cooler	35 °C
	P_{drop} CO_2 side	13.8 kPa per stage
	Stages	1
	Cooling source	Process cooling water/Cooling tower
Recompression	CO_2 bypass	22.4%
CO_2 main compressor	P_{inlet}	7.75 MPa
	P_{exit}	35.10 MPa
	Isentropic efficiency	0.85
	Stages	2
	Intercooling stages	1
CO_2 bypass compressor	Exit pressure	35.03 MPa
	Isentropic efficiency	0.85
	Stages	2
	Intercooling stages	0

The following sections describe the methodology used to develop the baseline plant configuration and to estimate the thermodynamic performance and cost for the plant.

METHODOLOGY

Aspen Plus Modeling Approach

The thermodynamic performance of the plant concepts described in this paper were based primarily on the output from a steady-state system model developed using Aspen Plus® (Aspen). The individual unit operations models were the same as those used in the NETL oxy-CFB indirect sCO_2 cycle study [1]. Details on the design basis, assumed feed compositions, and state point tables can be found in the NETL oxy-CFB indirect sCO_2 cycle study [1]. Specific assumptions used in the process models are described below.

Physical Property Methods

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). The Span-Wagner equation of state (EOS) is the most accurate property method available for processes consisting of pure CO_2 [10]. The Span-Wagner EOS is incorporated into the REFPROP (Reference Fluid Thermodynamic and Transport Properties Database) physical property method developed by the National Institute of Standards and Technology (NIST) and is implemented in Aspen as the Physical Property Method REFPROP. This method was used for all physical property estimates within the sCO_2 power cycle. The power cycle working fluid was assumed to be pure CO_2. Leakage and make-up flows were not modeled.

For the boiler and flue gas components of the process, the Physical Property Method PENG-ROB was used. This is consistent with the property methods used in other NETL systems studies of power plants with CFB or PC heat sources, including Reference Case B12A and Reference AUSC Case 5.

Modeling Assumptions

A number of assumptions regarding unit operation and cycle performance were incorporated in the Aspen model. These are described in the following sections.

Boiler and Flue Gas Section

CFB

The CFB was modeled using three lumped parameter reactor blocks in series. The first stage was a yield reactor (RYield) which decomposes the feed coal into conventional molecular species and ash. The second stage applies the combustion chemistry model using user-supplied conversions for the assumed reactions. The combustion chemistry model consisted of the following three reactions:

$$C + O_2 \rightarrow CO_2$$

$$2\,H_2 + O_2 \rightarrow 2\,H_2O$$

$$S + O_2 \rightarrow SO_2$$

The assumed conversions per pass were 50 percent for C and 100 percent for H_2 and S. NOx formation is neglected. The third reactor stage applies the in-bed sulfur capture chemistry. The assumed reaction is:

$$CaCO_3 + SO_2 + 2\,H_2O + 0.5\,O_2 \rightarrow \text{Gypsum} + CO_2$$

The assumed SO_2 conversion per pass is 94 percent. The molar lime feed rate is calculated as

2.4 times the molar sulfur feed to the in-bed capture reactor. The flue gas exiting the bed passes through a solids disengager (cyclone) which captures all but 0.03 percent of the solids. 99.05 percent of these solids are recirculated back to the CFB with the remainder taken off as the bottom ash.

The air feed to the CFB is calculated to give 3.1 percent excess oxygen giving a flue gas oxygen concentration of 0.9 percent leaving the CFB. The feed air is split into a primary air stream (23.5 percent) and a secondary or over-fired air stream (76.5 percent). A small infiltration air stream is assumed to enter the CFB and is calculated as 2 percent of the total feed air. The total CFB pressure drop is assumed to be 6.6 kPa.

Heat recovery takes place in the bed (where the sCO_2 entering the two turbine stages is heated to the TIT) and downstream from the solids disengager (high temperature economizer and air pre-heater). A 1 percent heat loss from the bed and the economizer is assumed while no heat loss is assumed for the air pre-heater.

Baghouse

The flue gas exiting the CFB passes through the baghouse to remove the remaining solids (fly ash) entrained from the solids disengager. The baghouse is assumed to have a 100 percent solids removal efficiency with no heat loss except for the sensible heat contained in the removed solids. The pressure drop is assumed to be 1.4 kPa.

Flue gas cooler

Before exiting to the stack, the flue gas is further cooled using a slip stream from the recycle sCO_2 exiting the main CO_2 compressor. The slip stream flow rate is set to give a 5.6 °C temperature approach at the cold end of the cooler. The hot end temperature approach is also set to 5.6 °C and controlled by adjusting the air pre-heater duty. The cooler is modeled in Aspen as an adiabatic countercurrent heat exchanger. The flue gas side pressure drop is assumed to be 0.7 kPa. The sCO_2 side pressure drop was set equal to the cold side pressure drop in the LTR.

sCO_2 Power Cycle

sCO_2 Turbine

The sCO_2 turbine was modeled in Aspen using the isentropic efficiency method with an assumed isentropic efficiency of 0.927. The TIT for all cases other than the Rankine reference cases was 760 °C and at this temperature, it was assumed that turbine blade cooling would not be needed. The turbine inlet pressure was capped at 34.5 MPa and the exit pressure was set to give a main compressor inlet pressure of 7.75 MPa. For the reheat turbine, the pressure ratios of the two stages were set approximately equal. The reheat exchanger heated the sCO_2 to 760 °C. A pressure drop of 1 percent was assumed for the sCO_2 side of this exchanger.

Recuperators

All of the recuperators in the power cycle were modeled as adiabatic countercurrent heat exchangers. The pressure drops on the high-pressure side were assumed to be 0.2 percent and the pressure drops on the low-pressure side were assumed to be 0.8 percent.

CO_2 Cooler and Main Compressor Intercoolers

The CO_2 Cooler and the main compressor intercoolers were modeled as adiabatic water-cooled countercurrent heat exchangers. The temperature was the same in both sets of coolers and its value was a design parameter for the Baseline Case. The CO_2 cooler pressure drop on the CO_2 side was assumed to be 0.8 percent. The main compressor intercooler pressure drop on the

CO_2 side was assumed to be 13.8 kPa per stage.

Main and Bypass CO_2 Compressors

The CO_2 compressors were modeled in Aspen using the isentropic efficiency method with an assumed isentropic efficiency of 0.85. Multistage compression was only used for the main CO_2 compressor. The compressor discharge pressure was controlled to achieve a turbine inlet pressure of 34.5 MPa after accounting for the recuperator and economizer pressure drops. The turbine exit pressure was controlled to achieve a main compressor inlet pressure of 7.75 MPa after accounting for the recuperator and cooler pressure drops.

Methodology for Capital Cost Estimation and Economic Analysis

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. These are summed to arrive at a total plant cost (TPC), expressed in June 2011 dollars for consistency with other NETL studies. [8] [11]

The total overnight cost (TOC) is equal to the sum of the TPC and the owner's costs. The owner's costs consist of costs arising from plant pre-production and inventories as well as land purchase and other items. The owner's costs are calculated as a fraction of the O&M costs and the TPC. The TOC is used to calculate the capital component of the COE.

From the standpoint of capital cost estimation, there are two primary components of the plant: the CFB and flue gas train, which is considered to be mature technology, and the sCO_2 power cycle components which are considered advanced or emerging technologies. The cost estimates for the mature technology 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 [1] [8] [9] [12] [13] [14]. This methodology is considered sufficiently accurate for a Class 4 estimate [15].

Cost scaling algorithms with reference costs and reference process variable data were taken from Reference Case RhtIC760 [1]. Exponents and ranges were based on QGESS - Capital Cost Scaling Methodology, January 2013 [14]. The same balance of plant (BOP) cost accounts as for Case RhtIC760 were used except that there was no ASU or CPU [1].

Capital Cost Estimates for sCO_2 Power Cycle Components

In performing the economic analysis of the sCO_2 power cycle cases, one of the biggest challenges is in estimating the capital costs for the unit operations in this emerging technology. Unlike the boiler and flue gas train, vendor cost data for commercial scale units is unavailable resulting in a relatively large uncertainty in the estimated costs. The following sections describe in detail the data and approaches used to develop capital cost estimates for the power cycle. Most of the cost algorithms are identical to those used for the oxy-CFB indirect sCO_2 cycle study [1].

CFB Boiler Costs

The reference CFB boiler cost was taken from a study based on a SC Rankine power cycle, Case B22F in the NETL CFB study [16] which made use of data from a report by Alstom [17]. A cost adjustment for AUSC conditions was derived from simulations using the Thermoflow products STEAM PRO and PEACE. Simulations were prepared for a 550 MW air-fired CFB plant with a SC Rankine cycle and another plant of the same size with an AUSC Rankine cycle. The ratio of the estimated cost for the AUSC plant to the SC plant was used to scale the reference SC Rankine case cost to AUSC conditions.

For the CFB cost estimates for the sCO_2 cases, a cost algorithm was derived that consisted of two components: the oxidant preheater and the remainder of the CFB and auxiliaries. The total heat transfer area and the capital cost of the oxidant preheater were determined from STEAM PRO and PEACE from which a cost per unit area could be derived.

The cost algorithm for the balance of the CFB and auxiliaries is comprised of two components: a heat duty dependent component and a driving force dependent component. A series of sensitivity analyses were performed in STEAM PRO and PEACE varying the plant capacity and steam TIT (driving force). Correlations of the results provided the necessary scaling coefficients for heat duty and driving force. The same CFB installation factor as used in the reference SC case was used for the other cases.

Note that detailed designs of the oxy-fired CFBs for the sCO_2 cases were not performed and, hence, direct estimates of the driving force in the CFB, including driving force adjustments due to reheat, were not possible. Using the steam TIT as a surrogate variable for driving force introduces a significant uncertainty in the cost estimate.

A sensitivity analysis was performed using STEAM PRO and PEACE on an air-fired CFB without a reheat steam turbine. The results of that sensitivity analysis showed that the use of reheat in the steam turbine increase the CFB cost by only 1.3 percent at a turbine inlet temperature of 760 °C. Hence, neglecting the effect of reheat on the CFB cost in this study is not expected to introduce a significant error.

Flue Gas and CO_2 Coolers

Power law cost scaling algorithms for the flue gas and main CO_2 coolers were derived based on reference, analogous coolers from the IGCC pathway studies. The reference cooler used for scaling the flue gas cooler had a heat duty of 78 MW and a bare erected cost (BEC) of $18,614,000 while the reference cooler used for scaling the CO_2 cooler had a heat duty of approximately 600 MW.

sCO_2 Cycle Main and Bypass CO_2 Compressors

The cost algorithms for the main sCO_2 and bypass compressor were derived from a proprietary vendor quote for two recompression cycles of different but comparable scale. To reflect expected changes in the compressor cost with intercooling, the scaling algorithm was divided into three components with one component dependent on required power (60 percent), another component dependent on volumetric flow rate to the compressor (20 percent), and the final component dependent on the temperature of the stream entering the main compressor stage (20 percent).

Recuperators

Equipment costs of the CO_2 recuperators are based on a report by Aerojet Rocketdyne [18]. This report contained quotes from two vendors. For the vendor designated Vendor A, the recuperators were a microchannel heat exchanger based on a shell and tube design that used a small number (17) of extremely large units. The HTR was broken up into two HTRs made of different materials. The high temperature (HT) HTR (hot side temperature ≥ 260 °C) consisted of 5 units while the low temperature (LT) HTR consisted of 4 units. The unit duties for the three recuperator types were 463 MW, 170 MW, and 94 MW for the HT HTR, LT HTR, and LTR, respectively. The estimated unit costs were $44,531,000 (2011$) for HT HTR units and $7,422,000 for LTR and LT HTR units. The cost was based on material and manufacturing only (BEC). The cost algorithm derived for this study adjusts the BEC for change in duty and log mean temperature difference (LMTD) driving force. The cost estimate for the VTR uses the cost algorithm for the HT HTR.

For the vendor designated Vendor B, the recuperator design was based on a counterflow

“Extended-Surface Unit-Cell Design Concept” [18]. This design uses a large number (629) of relatively small units. A single material was used for the HTR. The unit duties were 8 MW and 3 MW for the HTR and LTR, respectively. The unit costs were $97,625 (2011$) for HTR units and $152,000 for LTR units. The cost was based on material and manufacturing only (BEC). The cost algorithm derived for this study adjusts the BEC for change in duty and log mean temperature difference (LMTD) driving force. The cost estimate for the VTR used in the reference Case RhtIC760A uses the cost algorithm for the HTR.

All economic analysis results for the sCO_2 power cycle cases in this study used the cost algorithm based on the Vendor B design.

CO_2 Turbine

The BEC of the CO_2 turbine is based on a paper by Le Moullec [19] that cites a turbine cost of 102 €2010/kW including civil works, piping, etc. for a turbine in the 1 GWe range. This cost was assumed to include installation. Indexing that cost to June 2011 dollars provided a BEC cost of $109.3/kW for each turbine. The sCO_2 plant has two turbines. One (235,205 kW) is tied to the two CO_2 compressors, and the other (692,810 kW) is the power turbine.

The algorithm was further adjusted to cost variation based on the expected impacts of reheat. The algorithm base was broken into three components: one based on the gross turbine output (30 percent), another based on the ratio of cross sectional areas needed (37 percent), and the last based on the average turbine gas temperature (33 percent). The weighting factors were adjusted such that the calculated BEC for SC and AUSC conditions would be the same based on a rough order of magnitude (ROM) estimate from General Electric (GE).

CO_2 System Piping

The pipes that transport CO_2 working fluid between the heat source and turbine are very expensive due to the large flowrate of CO_2 and the elevated temperature and pressure of the fluid. For the purposes of this economic assessment, the pipe lengths between the HTR and heat source and between the heat source and turbine are assumed to be 150 ft. The pipe inside diameter was calculated based on the actual working fluid volumetric flow rate and an assumed fluid velocity of 150 ft/sec. The selection of pipe material and thickness was based on a NETL report [20]. The material capable of meeting the service temperature and pressure with the lowest estimated pipe cost was selected. A 30 percent installation factor was assumed.

Approach and Methodology for Case Permutations

The dual objectives of this study were to identify process configurations that would maximize process efficiency without leading to a larger COE than for reference Case B12A. Two approaches were used to achieve these objectives. A preliminary techno-economic analysis of reference Case RhtIC760A indicated that compared to reference Case B12A, its overall efficiency was significantly higher with a COE that was close to but slightly higher. Based on this result, a set of cycle state point changes and minor cycle configuration changes was developed and applied individually to reference Case RhtIC760A. The modifications that increased process efficiency and either reduced COE or had a minimally higher COE than reference Case RhtIC760A were retained for further consideration.

The second approach used was based on the pinch analysis of the power cycle and particularly on the temperature-enthalpy (T-Q) diagram of the recuperator train. Using this data, adjustments were proposed to the heat integration scheme that were likely to increase the power cycle efficiency without having a significantly adverse impact on the COE. The combination of these two approaches led to the identification of two candidate process configurations; a Baseline Case and an Alternate Configuration Case.

Selection of Cycle Configurations

The Baseline Case was developed by applying the promising cycle state point changes and configuration changes identified in Approach 1 to reference Case RhtIC760A sequentially. The order of introduction of the changes was based on the results obtained from the one-off analysis. Those changes that both increased process efficiency and lowered COE were applied first followed by the changes that increased efficiency with a neutral or slightly negative impact on COE. Changes that would adversely impact efficiency but lead to a large drop in COE were only to be considered if the COE goal could not otherwise be achieved.

The Alternate Configuration Case was developed by first identifying promising changes in the heat integration scheme. If a preliminary techno-economic analysis yielded a higher efficiency or lower COE than for reference Case RhtIC760A, then the configuration was retained for further study and the same methodology for sequentially applying process changes used to generate the Baseline Case was used to develop the final configuration for the candidate alternate configuration. Due to time and resource limitations, only the most promising of these alternatives were investigated and developed into the Alternate Configuration Case.

BASELINE CASE AND ALTERNATE CONFIGURATION

Figure 5 shows a simplified block flow diagram for the Baseline Case. To simplify the representation, only the power cycle components are depicted. The representations of the CFB and flue gas train are unchanged from Figure 4 except for the representation of the economizers. Econ1 in Figure 5 corresponds to the high temperature CFB economizer in Figure 4 that heats stream 46 to stream 47. Econ2 is the low temperature economizer (flue gas cooler) represented in Figure 4. The temperatures and heat exchanger duties shown in Figure 5 are the results from the converged Aspen model.

Figure 5 Simplified block flow diagram for Baseline Case

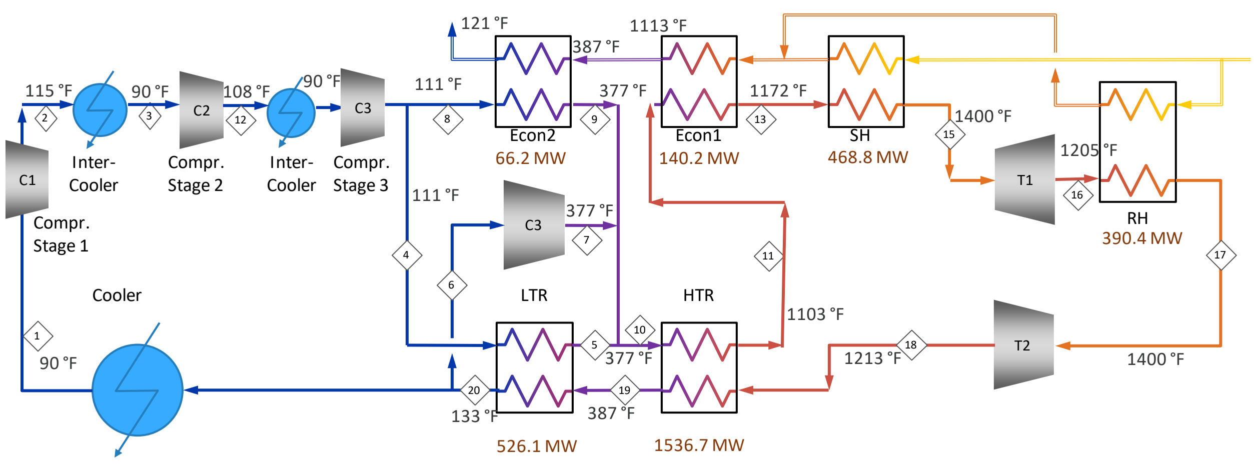


Table 2 compares the power cycle configuration and operating point for the Baseline Case with reference Case RhtIC760A. The first change incorporated in the Baseline Case was the elimination of the VTR. In the oxy-fired indirect sCO_2 cycle study [1], it was found that the use of the VTR improved the cycle and process efficiency so much that the COE was reduced as well, despite the capital cost of the VTR. For the air-fired Baseline Case, this benefit was not observed. This may be the result of changes in the heat distribution between the economizers, air pre-heater, and recuperators, or to differences in the flue gas heat capacity for an oxy-fired boiler versus an air-fired boiler. Regardless, the VTR was found to offer no benefit for the Baseline Configuration and was eliminated.

The second process modification was to lower the CO_2 cooler temperature. Prior sensitivity studies have shown that the cycle and process efficiencies increase monotonically with decreasing CO_2 cooler temperature [1] [2]. The Baseline Case cooling temperature was selected as 32 °C as this is the lowest cooling temperature that remains above the CO_2 critical temperature. Future analysis will explore the impact of using even lower cooler temperatures in a condensing power cycle configuration.

The third process modification was to increase the number of main compressor stages and intercooler stages. Increasing the number of intercoolers from 1 to 2 increased the process efficiency enough to offset any capital cost impact on the main CO_2 compressor. Increasing the number of compressor and intercooler stages further offered no benefit to the cycle.

The final configuration change adopted in the Baseline Case was to increase the CFB operating pressure slightly. This eliminated the need for an induction fan to propel the flue gas through the baghouse and flue gas cooler. By eliminating the capital cost and auxiliary power requirement of the induction fan, the process efficiency increased, and the COE decreased. Note that no secondary impacts on the CFB from the increase in pressure were accounted for including the carbon and sulfur conversions and the amount of infiltration air.

The final entry in Table 2, the CO_2 bypass fraction, is not a design decision variable but rather is set by the minimum temperature approach specifications applied in the heat integration scheme. The results illustrate that the cycle configuration changes implemented in the Baseline Case had some negative impact on the cycle efficiency although the impact on the overall process efficiency was positive.

Table 2 Baseline Case power cycle configuration changes

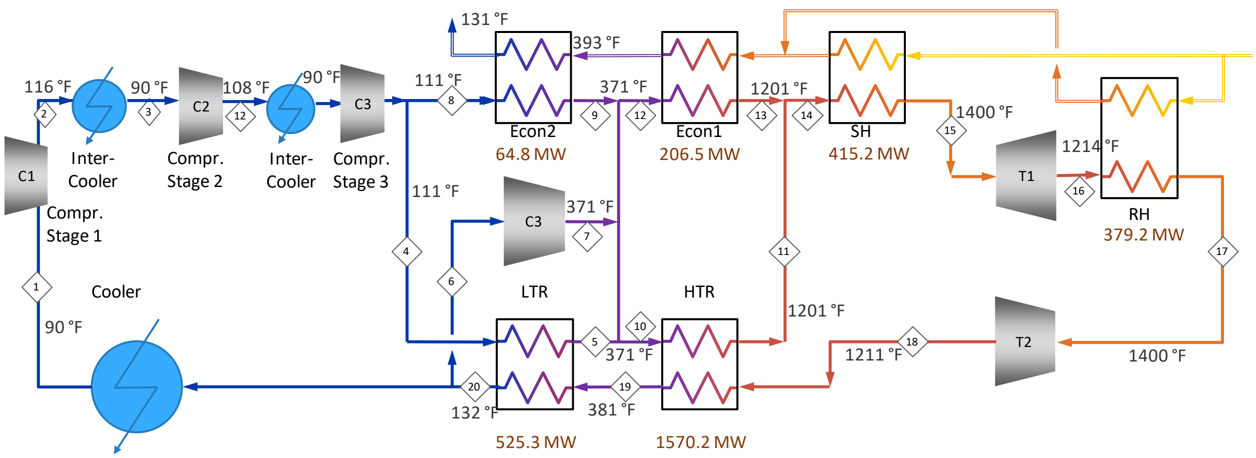
Parameter	Baseline Case	Reference Case RhtIC760A
Recuperators	LTR, HTR	LTR, HTR, VTR
CO_2 cooler temperature	32 °C	35 °C
Main CO_2 comp IC stages	2	1
CFB pressure	0.107 MPa	0.101 MPa
CO_2 bypass fraction	0.233	0.224

In addition to the cycle configuration changes identified in Table 2, numerous other potential configuration changes were considered including the use of an aftercooler on the main CO_2 compressor, eliminating the reheat turbine, introducing a double reheat turbine, and altering the air pre-heater duty. None of these possible configuration changes were found to be beneficial in meeting the study objectives.

Figure 6 shows a simplified block flow diagram for the Alternate Configuration Case. The major change in the heat integration scheme for this case compared to either reference Case RhtIC760A or the Baseline Case relates to the high temperature economizer (Econ1). In the Baseline Case, this economizer occurs in series with the HTR but in the Alternate Configuration Case, it occurs in parallel with the HTR. This creates a more balanced thermal capacitance between the hot and cold sides of the recuperator train, reducing the approach temperatures in the recuperators and increasing their effectiveness. The unavoidable consequence of this is that the driving force for heat transfer in the recuperators is reduced, increasing their required surface area and cost.

The additional cycle configuration changes shown in Table 2 were also applied to the Alternate Configuration Case. The only difference was that for the Alternate Configuration Case, the CO_2 bypass fraction was 0.242, approximately 4 percent greater than for the Baseline Case. As with the Baseline Case, no other cycle configuration changes considered offered additional improvement to the performance of the Alternate Configuration Case.

Figure 6 Simplified block flow diagram for Alternate Configuration Case



Performance Results Summary

Table 3 shows the performance summary results for the Baseline Case and the Alternate Configuration Case. Also shown in Table 3 are the performance summary results for the 3 reference cases.

Table 3 Overall performance summary

Performance Summary	Reference Case B12A	AUSC Case 5	Case RhtIC760A	Baseline Case	Alternate Config
Total Gross Power, MWe	**580**	**578**	**571**	**584**	**584**
CO_2 Capture/Removal Auxiliaries, kWe	0	0	0	0	0
CO_2 Compression, kWe	0	0	0	0	0
Balance of Plant, kWe	29,688	27,190	17,966	16,990	16,990
Total Auxiliaries, MWe	**30**	**27**	**18**	**17**	**17**
Net Power, MWe	**550**	**550**	**553**	**567**	**567**
HHV Net Plant Efficiency (%)	40.7%	44.1%	48.2%	49.5%	49.5%
HHV Net Plant Heat Rate, kJ/kWh (Btu/kWh)	8,841 (8,379)	8,158 (7,732)	7,462 (7,073)	7,280 (6,900)	7,280 (6,900)
LHV Net Plant Efficiency (%)	42.2%	45.8%	50.0%	51.3%	51.3%
LHV Net Plant Heat Rate, kJ/kWh (Btu/kWh)	8,527 (8,082)	7,869 (7,458)	7,197 (6,822)	7,021 (6,655)	7,022 (6,655)
HHV Boiler Efficiency, %	89.1%	89.1%	92.4%	92.9%	92.9%
LHV Boiler Efficiency, %	92.4%	92.4%	95.8%	96.3%	96.3%
Power Cycle Efficiency, %	48.2%	52.0%	53.9%	54.8%	54.8%
Power Cycle Heat Rate, kJ/kWh (Btu/kWh)	7,473 (7,083)	6,926 (6,564)	6,675 (6,326)	6,567 (6,224)	6,568 (6,225)
CO_2 Cycle Cooling Duty/Condensor Duty, GJ/hr (MMBtu/hr)	2,192 (2,078)	1,873 (1,776)	1,725 (1,635)	1,701 (1,612)	1,701 (1,613)
As-Received Coal Feed, kg/hr (lb/hr)	179,193 (395,053)	165,482 (364,825)	152,162 (335,460)	152,162 (335,460)	152,162 (335,460)
Limestone Sorbent Feed, kg/hr (lb/hr)	17,707 (39,037)	16,352 (36,050)	35,618 (78,525)	35,618 (78,525)	35,618 (78,525)
HHV Thermal Input, kWt	1,350,652	1,247,323	1,146,927	1,146,927	1,146,927
LHV Thermal Input, kWt	1,302,740	1,203,058	1,106,225	1,106,225	1,106,225
Raw Water Withdrawal, $(m^3/min)/MW_{net}$ (gpm/MW_{net})	0.035 (9.3)	0.030 (8.0)	0.025 (6.6)	0.024 (6.2)	0.024 (6.2)
Raw Water Consumption, $(m^3/min)/MW_{net}$ (gpm/MW_{net})	0.028 (7.4)	0.024 (6.4)	0.019 (5.0)	0.018 (4.7)	0.018 (4.7)
O_2 Mole Percent in Boiler Exit, %	3.4%	3.4%	1.0%	1.0%	1.0%
CO_2 Emissions (lb CO_2/MWh-gross)	1,617	1,490	1,383	1,353	1,353

The results show that the thermodynamic performance of the Baseline Case and the Alternate Configuration Case are virtually identical. Although the parallel economizer arrangement in the Alternate Configuration Case had the desired effect of lowering the temperature approach through the HTR, this did not lead to an increase in net cycle power output or efficiency.

Compared to reference Case RhtIC760A, the two modified sCO_2 power cycle cases had a 1.3 percentage point higher process efficiency and a 1.0 percentage point higher cycle efficiency. Most of the remaining differences between these cases in Table 3 are a direct result of the difference in efficiency.

Compared to reference AUSC Case 5, the two modified sCO_2 power cycle cases had a 5.4 percentage point higher process efficiency and a 2.8 percentage point higher cycle efficiency. Some of the net plant efficiency improvement results from the difference in the heat source with the CFB having a lower auxiliary power requirement due to in-bed sulfur capture. However, most of the net plant efficiency improvement results from the higher power cycle efficiency for the sCO_2 cases and the increased fractional heat recovery (boiler efficiency) in the cycle configurations.

For the plants listed in Table 3, the CO_2 emissions are inversely proportional to the plant efficiency. The CO_2 emissions for the reference Case B12A are well above the EPA limit of 1400 lbs CO_2/MWh gross [21] and the CO_2 emissions for the reference AUSC Case 5 exceed the EPA limit by 6.4%. In contrast, the CO_2 emissions for all of the plants based on the sCO_2 power cycle are below the EPA limit with the CO_2 emissions for the Baseline Case and Alternate Configuration 3.4% lower than the limit. However, the emissions reported in Table 3 assume full load and steady-state operation whereas EPA's standard is based on average annual emissions. Additional analyses are required to assess system performance under realistic annual operating profiles, including part-load conditions.

An additional finding from the results shown in Table 3 is the significantly lower water consumption for the plants based on the sCO_2 power cycle compared to the reference Rankine cases. Compared to the reference Case B12A, the water consumption for the Baseline Case and Alternate Configuration is a third lower and compared to the reference AUSC Case 5 the water consumption for these plants is over 22% lower. This reduction in water consumption results from the higher plant thermal efficiencies of the sCO_2 plants as well as the elimination of intrinsic water losses arising from the Rankine cycle such as from blowdown.

Table 4 shows the power summary results for the 5 cases. As with the overall performance summary, there is no significant difference in the total generated power and total auxiliary power for the Baseline Case and the Alternate Configuration Case. There is a variation in the turbine and compressor power values for these two cases, but the differences cancel each other. Compared to the reference Case RhtIC760A, the modified sCO_2 power cycle cases show an approximately 1 MW decrease in auxiliary power resulting from the elimination of the ID fan.

Table 4 Overall plant power summary

Power Summary	Reference Case B12A	AUSC Case 5	Case RhtIC760A	Baseline Case	Alternate Config
POWER GENERATION SUMMARY					
sCO_2/Steam Turbine Gross Power	588,516	586,305	735,537	741,272	740,316
sCO_2 Main Compressor	---	---	-93,844	-85,654	-83,172
sCO_2 Bypass Compressor	---	---	-61,695	-62,558	-64,096
Generator Loss	-8,828	-8,795	-8,700	-8,896	-8,896
TOTAL POWER GENERATED	**579,688**	**577,510**	**571,298**	**584,164**	**584,151**
AUXILIARY LOAD SUMMARY					
Coal Handling	-430	-420	-399	-399	-399
Sorbent Prep/Injection	-958	-890	-157	-157	-157
Pulverizers	-2,690	-2,480	-72	-72	-72
Ash Handling and Dewatering	-620	-580	-1,756	-1,756	-1,756
Baghouse	-90	-80	-7	-7	-7
Turbine Auxiliaries	-400	-400	-400	-400	-400
Wet FGD	-2,830	-2,610	---	---	---
Condensate Pumps	-800	-640	---	---	---
SCR	-40	-40	---	---	---
Miscellaneous Balance of Plant	-2,000	-2,000	-2,000	-2,000	-2,000
Circulating Water Pump	-4,980	-4,290	-3,734	-3,636	-3,636
Cooling Tower Fans	-2,340	-8,950	-2,180	-2,122	-2,122
Air & Flue Gas Fan Power	-9,690	-2,010	-5,503	-4,647	-4,647
Transformer Losses	-1,820	-1,800	-1,758	-1,794	-1,794
TOTAL AUXILIARIES	**-29,688**	**-27,190**	**-17,966**	**-16,990**	**-16,990**
NET POWER	**550,000**	**550,320**	**553,332**	**567,174**	**567,162**

Figure 7 shows the T-Q diagram for the recuperators in the Baseline Case while Figure 8 shows the T-Q diagram for the economizers. The vertical dashed lines show the demarcation between the low and high temperature exchangers. For the recuperators, note that the LTR has an

Figure 7 Recuperator T-Q diagram for the Baseline Case

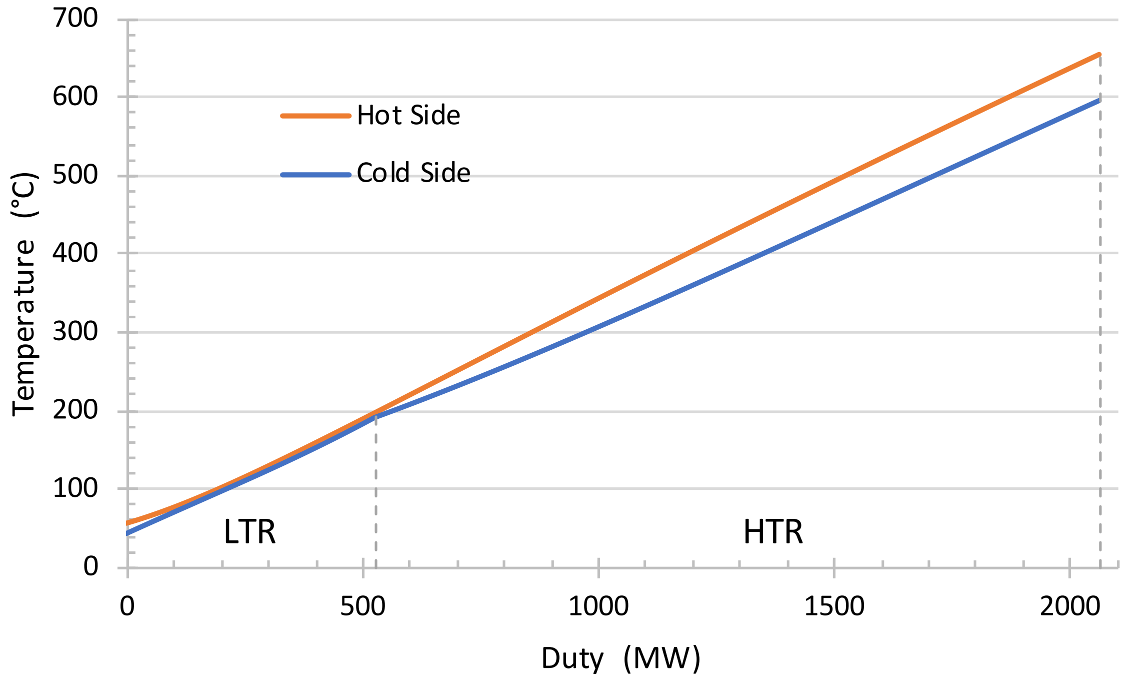
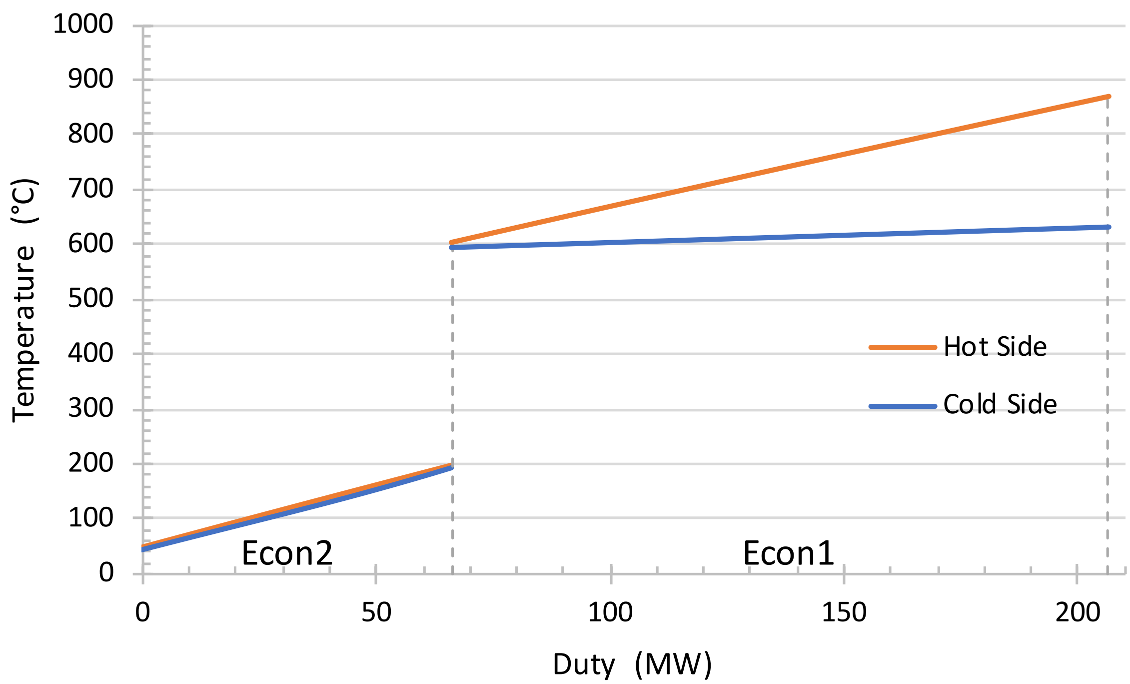


Figure 8 Economizer T-Q diagram for the Baseline Case



internal pinch point with a minimum temperature approach of 3.7 °C. The cold end temperature approach was increased 11.9 °C to attain an average temperature approach of 5.6 °C throughout the LTR. The temperature approaches in the HTR are considerably larger ranging from 5.6 °C at the cold end to 61.2 °C at the hot end. Reducing this large temperature approach was the primary objective in developing the parallel economizer heat integration scheme in the Alternate Configuration Case. For the economizers, a minimum temperature approach of 5.6 °C was specified on both ends of Econ2 but could only be achieved at the cold end of Econ1.

Figure 9 shows the T-Q diagram for the recuperators in the Alternate Configuration Case while Figure 10 shows the T-Q diagram for the economizers. For the recuperators, the hot and cold

Figure 9 Recuperator T-Q diagram for the Alternate Configuration Case

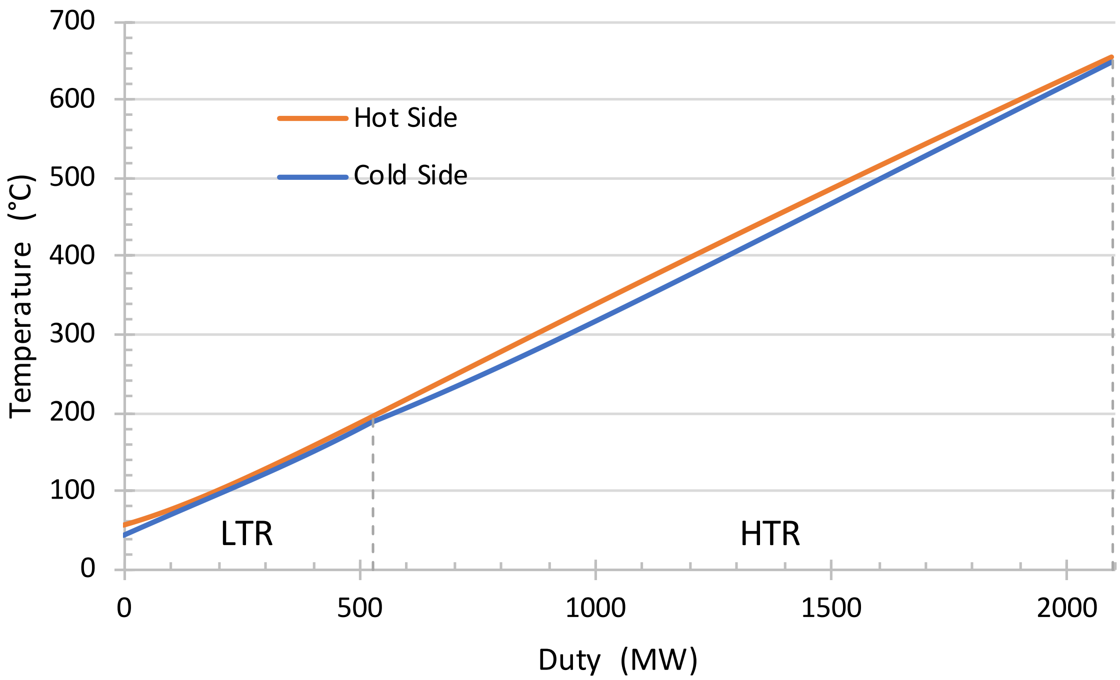
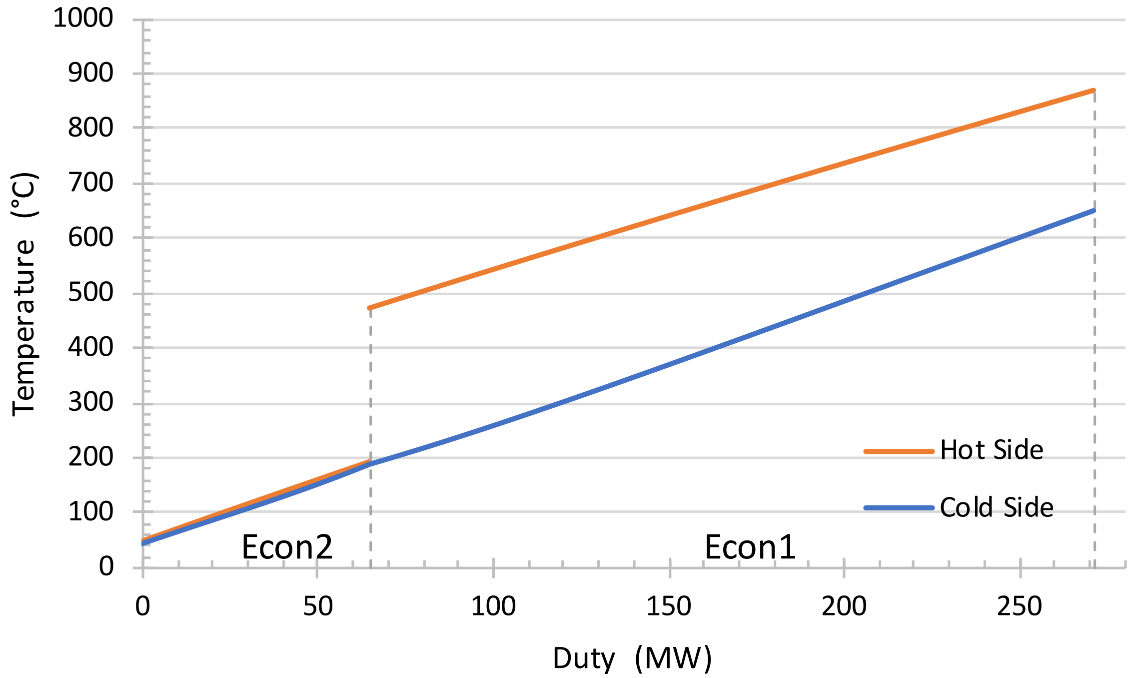


Figure 10 Economizer T-Q diagram for the Alternate Configuration Case



Composite curves are much closer than in the Baseline Case indicating a lower average driving force for heat transfer but a higher recuperator effectiveness. However, for the economizers, this situation is reversed which acts to negate the benefit of a higher recuperator effectiveness.

Economic Analysis Results Summary

Table 5 shows the capital cost summary for the 5 cases. The reported costs are in units of $1,000 and in 2011 dollars. Both the total overnight cost (TOC, for plant level comparisons) and the total plant cost (TPC, for account level comparisons) are provided.

Table 5 Capital Cost Summary (1,000 2011$)

Performance Summary	Reference Case B12A	AUSC Case 5	Case RhtIC760A	Baseline Case	Alternate Config
Account	TPC ($1,000)				
Coal & Sorbent Handling	45,398	43,201	46,035	46,035	46,035
Coal & Sorbent Prep & Feed	21,531	20,438	24,393	24,393	24,393
Feedwater & Miscellaneous BOP Systems	93,644	80,319	21,312	21,025	21,025
Boiler & Accessories	341,722	336,100	361,614	380,576	361,308
Gas Cleanup & Piping	167,272	157,731	34,059	32,386	32,204
HRSG, Ducting, & Stack	45,629	45,014	48,845	48,845	48,845
Steam/sCO_2 Power Cycle	166,934	179,487	382,882	340,547	364,790
Cooling Water System	44,037	39,814	41,270	40,542	40,542
Ash & Spent Sorbent Handling Systems	16,778	16,047	29,717	29,718	29,717
Accessory Electric Plant	61,735	60,145	57,879	57,068	57,068
Instrumentation & Control	26,316	26,021	24,129	23,928	23,928
Improvements to Site	16,394	15,655	16,554	16,473	16,486
Buildings & Structures	66,971	65,241	68,803	68,655	68,672
Total Plant Costs	**1,114,361**	**1,085,214**	**1,157,494**	**1,130,193**	**1,135,013**
	Owner's Costs & TOC ($1,000)				
Owner's Costs	264,273	256,122	282,518	277,003	277,977
Total Overnight Cost (TOC)	**1,378,634**	**1,341,336**	**1,440,013**	**1,407,196**	**1,412,991**

Comparisons of the costs among the 5 cases is complicated by the fact that the Rankine and sCO_2 plants use different heat source technologies as well as different power cycle technologies. Also, the sCO_2 power cycle costs are better encapsulated than the Rankine costs since it has more interaction with the other major cost accounts. Finally, the Rankine plants were both designed for a net output of 550 MW whereas the sCO_2 all use a constant coal feed rate. To help sort out these complicating factors, Table 6 shows the normalized costs results on a $/kW basis, also in 2011 dollars.

Table 6 Capital Cost Summary (2011$/kW)

Performance Summary	Reference Case B12A	AUSC Case 5	Case RhtIC760A	Baseline Case	Alternate Config
Account	TPC ($/kW)				
Coal & Sorbent Handling	83	78	83	81	81
Coal & Sorbent Prep & Feed	39	37	44	43	43
Feedwater & Miscellaneous BOP Systems	170	146	38	37	37
Boiler & Accessories	621	611	651	669	635
Gas Cleanup & Piping	304	287	61	57	57
HRSG, Ducting, & Stack	83	82	88	86	86
Steam/sCO_2 Power Cycle	304	326	690	599	641
Cooling Water System	80	72	74	71	71
Ash & Spent Sorbent Handling Systems	31	29	54	52	52
Accessory Electric Plant	112	109	104	100	100
Instrumentation & Control	48	47	43	42	42
Improvements to Site	30	28	30	29	29
Buildings & Structures	122	119	124	121	121
Total Plant Costs	**2,026**	**1,972**	**2,085**	**1,986**	**1,995**
	Owner's Costs & TOC ($/kW)				
Owner's Costs	481	465	509	487	489
Total Overnight Cost (TOC)	**2,507**	**2,437**	**2,594**	**2,473**	**2,483**

Compared to the reference Rankine AUSC Case 5, the Baseline Case had a very slightly greater TOC (1.5 percent) on a $/kW basis. Compared to the reference Rankine SC Case B12A, the Baseline Case had a 1.4 percent lower TOC on a $/kW basis.

In comparing the account level costs for the Baseline Case with the reference Case RhtIC760A, the most significant differences in the costs are in the Boiler & Accessories account and the Power Cycle account. The differences in the other accounts are either very small or in inverse proportion to the plant efficiency ratio between the two cases. The process modifications incorporated in the Baseline Case resulted in a $42MM reduction in the power cycle costs although this was partially offset by a $19MM increase in the CFB cost. Comparing the Baseline Case to the Alternate Configuration Case, the TOC was nearly the same in the two cases with the Alternate Configuration Case having an approximately $6MM increase compared to the Baseline Case. Again, most of the individual account differences were confined to the Boiler & Accessories account and the Power Cycle account. These cost differences were primarily due to differences in the relative distribution of the recuperator, economizer, and air pre-heater duties as well as to the lower driving force for heat transfer in the HTR for the Alternate Configuration Case.

Table 7 shows the estimated COE for the 5 cases, broken down by the major contributions.

Table 7 COE Summary (2011$/MWh)

COE Summary	Reference Case B12A	AUSC Case 5	Case RhtIC760A	Baseline Case	Alternate Config
	$/MWh	$/MWh	$/MWh	$/MWh	$/MWh
Capital	39.0	38.0	43.2	41.2	41.4
Fixed O&M	9.6	9.5	10.1	9.7	9.7
Variable O&M	9.1	8.5	9.1	8.8	8.8
Fuel	24.6	22.7	20.7	20.2	20.2
Total (with T&S)	**82.3**	**78.6**	**83.1**	**79.9**	**80.1**

Compared to reference Case RhtIC760A, the COE for the Baseline Case is 4 percent lower. This is all due to the cumulative impact of the process changes shown in Table 2. The COE for the Alternate Configuration Case is only very slightly larger than the COE for the Baseline Case ($0.2/MWh) with all of this difference being contributed by a higher capital cost. Although this result would suggest that the flattening of the composite curves in the T-Q profile was not cost effective, the capital cost difference is well within the uncertainty range of the capital cost algorithms suggesting that there is no significant difference between the two cases from the perspective of COE.

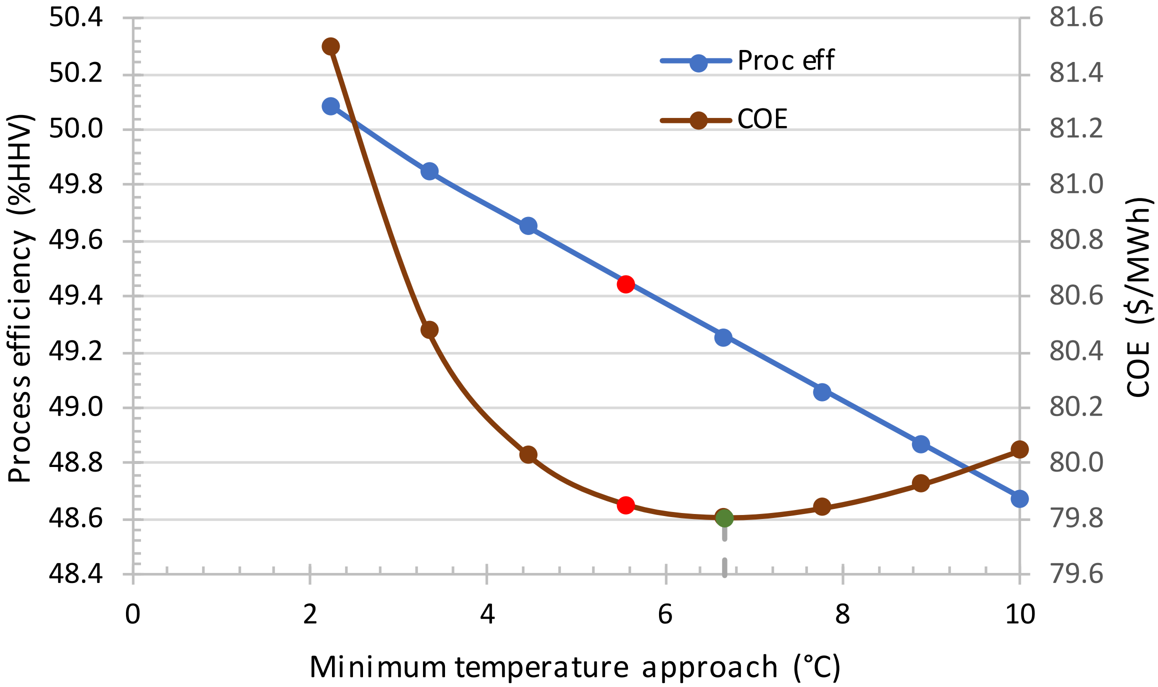
Compared to reference Case AUSC Case 5, the Baseline Case had a 1.6% higher COE. Examining the components of the COE, it would appear that the increase in efficiency achieved by the Baseline Case was not sufficient to overcome the higher capital cost of the CFB/sCO_2 power cycle plant compared to the PC/Rankine plant. If the accounts associated with the heat source (Coal & Sorbent Handling, Coal & Sorbent Prep & Feed, Boiler & Accessories, and Ash & Spent Sorbent Handling Systems) and power generation (Feedwater & Miscellaneous BOP Systems, Gas Cleanup & Piping, HRSG, Ducting, and Stack, and Steam/sCO_2 Power Cycle) are arrogated separately, the heat source and power generation costs for reference AUSC Case 5 are $416MM and $463MM, respectively while these same two items for the Baseline Case are $481MM and $443MM, respectively. It would appear that the higher capital cost of the sCO_2 cases is the result of a higher capital cost for the heat source. This is consistent with the results from the oxy-CFB indirect sCO_2 cycle study [1] where the heat source technology was the same for both the Rankine and sCO_2 power cycle plants and where the sCO_2 plants at AUSC conditions had a lower capital cost.

SENSITIVITY ANALYSES

Minimum Temperature Approach in Recuperators

The target temperature approach used in this study (5.6 °C) was somewhat arbitrary and based on the value used in the oxy-CFB indirect sCO_2 cycle study [1]. A sensitivity analysis was performed on the Baseline Case to examine the impact of the minimum or target temperature approach in the recuperators on the cost and performance of the plant. The results are shown in Figure 11. The results for the Baseline Case are depicted with red markers in this plot.

Figure 11 Sensitivity of Baseline Case performance to minimum temperature approach

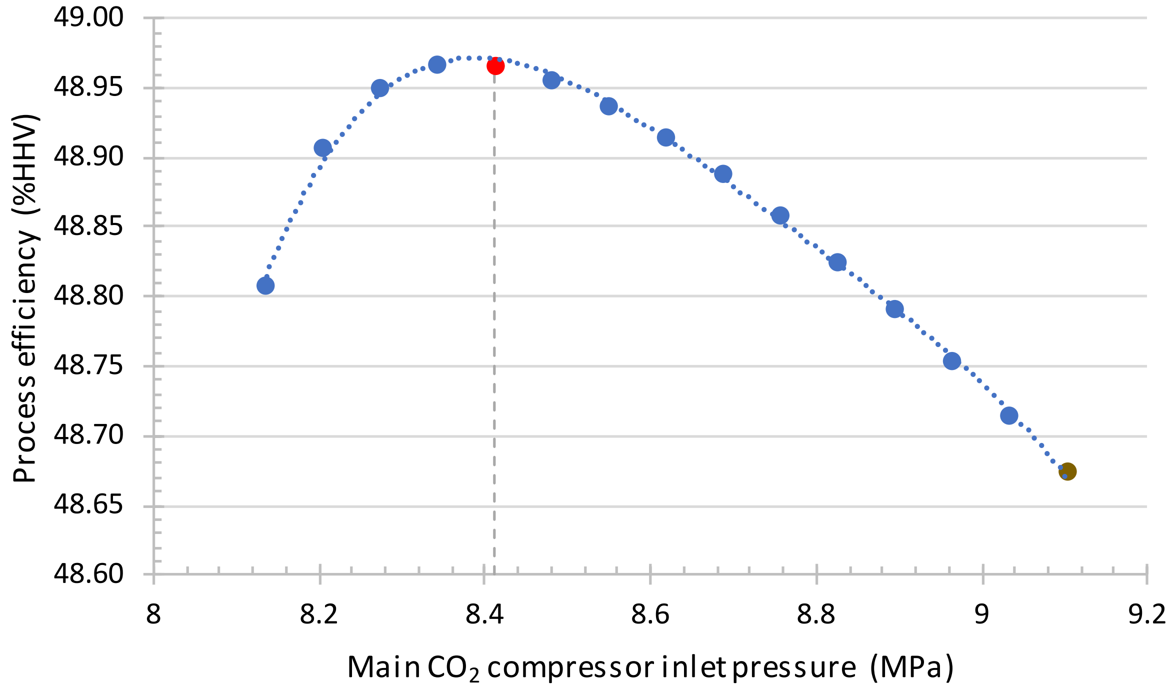


The results show that the process efficiency decreases monotonically and almost linearly with increasing minimum temperature approach whereas the COE passes through a minimum at a temperature approach of 6.7 °C (gray dashed vertical line and green marker). While this result suggests that a lower COE could be attained with a higher minimum temperature approach of 6.7 °C, the 0.2 percentage point drop in process efficiency was deemed more significant than the negligible drop in COE.

Cycle Pressure Ratio

For any sCO_2 power cycle there is an optimum turbine inlet pressure and an optimum main CO_2 compressor inlet pressure. A sensitivity analysis was performed on reference Case RhtIC760A to determine the optimum turbine inlet pressure, but it was found to exceed the maximum turbine inlet pressure constraint of 34.5 MPa adopted for this study. The sensitivity analysis to determine the optimum main CO_2 compressor inlet pressure is shown in Figure 12.

Figure 12 Sensitivity of reference Case RhtIC760A to main CO_2 compressor inlet pressure



This performance only sensitivity analysis showed that the maximum process efficiency was attained using a compressor inlet pressure of 8.41 MPa. This yielded an optimum cycle pressure ratio of 4.1.

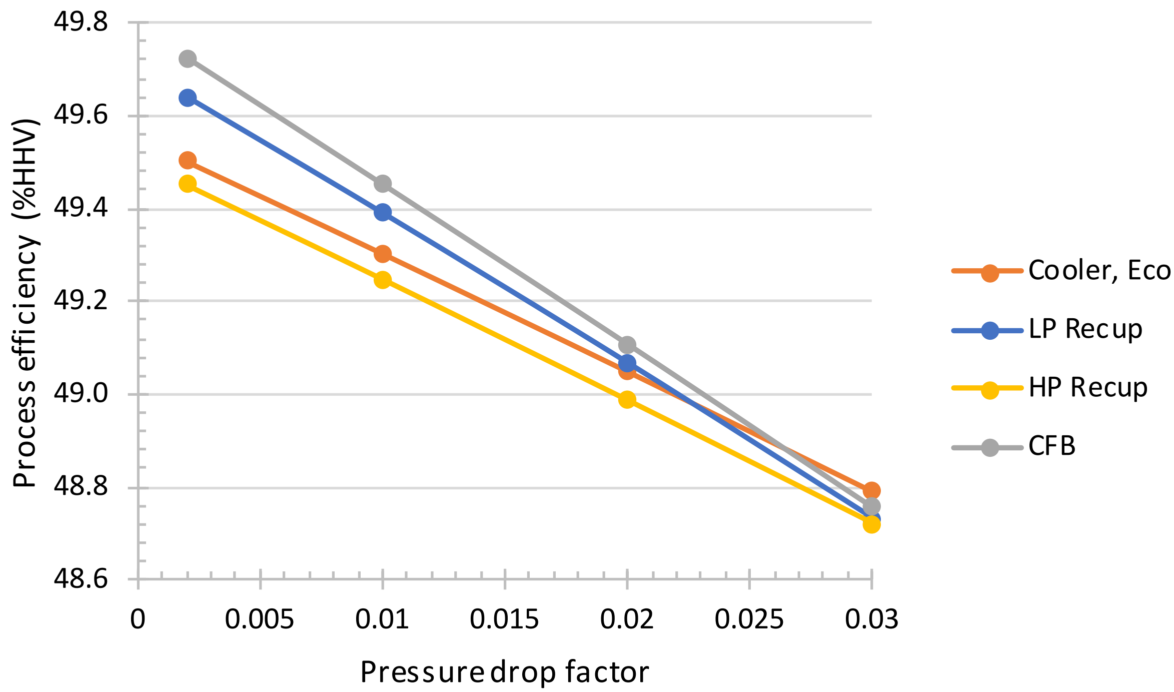
Unit Operation Pressure Drops

The pressure drops assumed for the sCO_2 power cycle unit operations are design variables. Since detailed designs were not undertaken for the power cycle components, the pressure drops used should be considered optimistic targets that were based on the desire to maximize process efficiency. For some unit operations, such as the CFB, a minimum pressure drop is necessary from an operational standpoint but determining this requires a detailed design. For the recuperators, there is an indirect relationship between pressure drop and capital cost but there was insufficient data available to incorporate such a relationship in the cost algorithms.

A performance only sensitivity analysis was performed on reference Case RhtIC760A to quantify the impact of pressure drop on process efficiency. The results are shown in Figure 13. Since the pressure drops were calculated as a percentage of the inlet pressure to the unit, the sensitivity analysis independent variables were the pressure drop factors. For the CO_2 cooler and CO_2 side of the economizers as well as the low-pressure side of the recuperators, the values used were 0.008. For the high-pressure side of the recuperators the value used was 0.002 and for the CO_2 side of the CFB, the value used was 0.01.

The results show that the process efficiency for the sCO_2 power cycle is moderately sensitive to the pressure drop, particularly on the low-pressure side of the cycle. If more conservative pressure drop factors had been used, such as triple the optimistic values, the efficiency benefit of the Baseline Case compared to reference AUSC Case 5 would have been cut by 30%.

Figure 13 Sensitivity of process efficiency to unit operation pressure drop

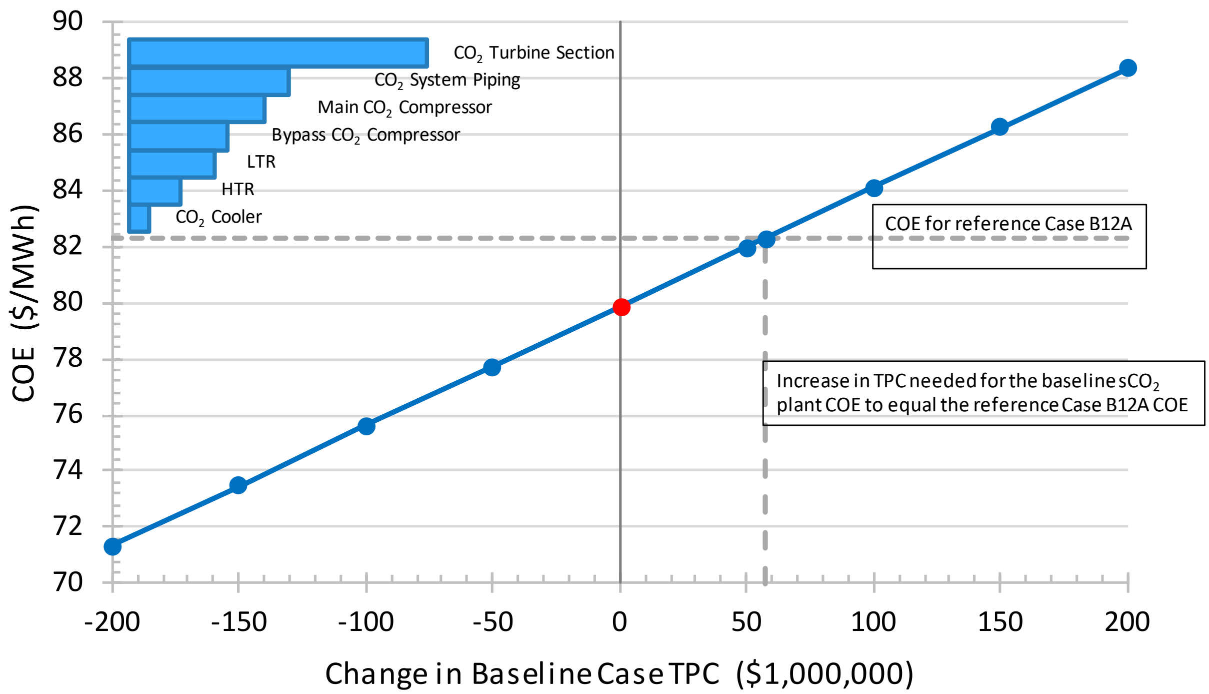


Capital Cost

As noted previously, the uncertainty in the sCO_2 power cycle capital costs is relatively high. To quantify this uncertainty, a sensitivity analysis was performed on the TPC for the Baseline Case. The results are shown in Figure 14. The plot shows the calculated COE as a function of changes (in $MM) to the estimated TPC. The red dot corresponds to the results for the Baseline Case. The horizontal dashed line corresponds to the COE for reference Case B12A. It is the goal that the COE for the advanced technology power plants not exceed this value. The vertical dashed line shows the actual increase in TPC that would have to occur for the COE to reach the target limit. For the Baseline Case, this value is $57MM.

Also shown on Figure 14 are a set of seven horizontal bars that denote the TPC for the major unit operations in the sCO_2 power cycle. The horizontal bars are drawn to scale with respect to the horizontal axis. The results indicate that it would take a 5 percent increase in the aggregate TPC for the Baseline Case COE to reach the same COE as reference Case B12A. If this uncertainty were all confined to the power cycle account, where the uncertainty is relatively large, it would take a 17 percent increase to have the same effect on COE. Within the power cycle, it is generally regarded that the greatest cost uncertainty is in the turbine and recuperators. Based on the results of this sensitivity analysis, it would take a 49 percent increase in the estimated turbine cost or a 107 percent increase in the recuperator costs to increase the COE of the Baseline Case to the level of reference Case B12A.

Figure 14 Sensitivity of Baseline Case COE to change in capital cost



CONCLUSIONS & PLANS FOR FUTURE WORK

This paper has presented the results of a preliminary examination of the potential benefits of the indirect sCO_2 power cycle for improving the efficiency and cost of non-capture coal-fired power plants. The results have shown that the sCO_2 power cycle can achieve much higher efficiencies than SOA PC/Rankine systems with no increase in COE. Compared to prior NETL systems studies on advanced power generation technologies, such as the PC power plant with an AUSC Rankine cycle, the sCO_2 power cycle offers a significant increase in overall efficiency of greater than 5 percentage points.

With full-load, steady-state carbon dioxide (CO_2) emissions of 1353 lbs CO_2/MWh gross determined for examined sCO_2 system configurations, this result nominally meets the current EPA's 1400 lbs CO_2/MWh gross for new coal plants [21]. However, the EPA's standard is based on average annual emissions – additional analyses are required to assess system performance under realistic annual operating profiles, including part-load.

The study has also shown that plants based on the sCO_2 power cycle have significantly lower (22-33%) water consumption than comparable reference Rankine cycle plants. This results from the higher thermal efficiencies of the sCO_2 plants along with the elimination of intrinsic water losses arising from the Rankine cycle such as from blowdown.

Future work at NETL will continue to explore the indirect sCO_2 power cycle with the goal of expanding its range of application, further optimizing its performance and cost, reducing the current level of uncertainty in the performance and cost models, and exploring more complex aspects of the cycle development related to system dynamics.

One study currently underway is exploring in greater detail the impacts of various cooling technology options on the cycle and overall plant performance with the goal of optimizing the cooling technology choice for any given ambient condition or site location. Other concepts planned for near term examination are based on the results of the sensitivity analyses performed in this study and include investigations of condensing cycles as well as performing a more thorough optimization of the cycle parameters including individual minimum temperature approaches for each end of every recuperator, economizer, and intercooler, as well as better defining the trade-off between process efficiency gains and capital cost results from pressure drops in the cycle unit operations.

NOMENCLATURE

Aspen	=	Aspen Plus®
ASU	=	Air separation unit
atm	=	Atmosphere (14.696 psi)
AUSC	=	Advanced ultra-supercritical
B&W	=	Babcock and Wilcox
BEC	=	Bare erected cost
BFD	=	Block flow diagram
BOP	=	Balance of plant
Btu	=	British thermal unit
Btu/hr	=	British thermal units per hour
Btu/kWh	=	British thermal units per kilowatt hour
CCCMP	=	Clean Coal and Carbon Management Program
CCS	=	Carbon capture and storage
CFB	=	Circulating fluid bed
CO_2	=	Carbon dioxide
COE	=	Cost of electricity
CPU	=	CO_2 purification unit
DOE	=	Department of Energy
EERE	=	Energy Efficiency and Renewable Energy
EOR	=	Enhanced oil recovery
EOS	=	Equation of state
EPA	=	Environmental Protection Agency
EPRI	=	Electric Power Research Institute
FE	=	Fossil energy
FG	=	Flue gas
gpm	=	Gallons per minute
h, hr	=	Hour
H_2O	=	Water
HHV	=	Higher heating value
HP	=	High pressure
HTR	=	High temperature recuperator
IC	=	Intercooler
ID	=	Induced draft
IP	=	Intermediate pressure
ISO	=	International Organization for Standardization
kW	=	Kilowatt
lb	=	Pound
lb/hr	=	Pounds per hour
lbmol	=	Pound mole
lbmole	=	Pound mole
LP	=	Low pressure
LTR	=	Low temperature recuperator
MAC	=	Main air compressor
MM	=	Million
MMBtu	=	Million British thermal units
MPa	=	Mega Pascal
MW	=	Megawatt
MWh	=	Megawatt-hour
N_2	=	Nitrogen
NE	=	Nuclear energy
NETL	=	National Energy Technology Laboratory
NIST	=	National Institute of Standards and Technology
O_2	=	Oxygen
PC	=	Pulverized coal
ppmv	=	Parts per million volume
PR	=	Pressure ratio
psi	=	Pound per square inch
psia	=	Pound per square inch absolute
psig	=	Pound per square inch gauge
R&D	=	Research and development
RD&D	=	Research, development, and demonstration
SC	=	Supercritical
SCR	=	Selective catalytic reduction
sCO_2	=	Supercritical carbon dioxide
SOA	=	State-of-the-art
T	=	Temperature
TIT	=	Turbine inlet temperature
TOC	=	Total overnight cost
T-Q	=	Temperature-enthalpy
TRL	=	Technical readiness level
U.S.	=	United States
USC	=	Ultra-supercritical
VTR	=	Very high temperature recuperator
°C	=	Degrees Celsius
°F	=	Degrees Fahrenheit

REFERENCES

[1] National Energy Technology Laboratory (NETL), "Techno-economic Evaluation of Utility-Scale Power Plants Based on the Indirect sCO2 Brayton Cycle," DOE/NETL- 2017/1836, Pittsburgh, PA, September 2017.

[2] National Energy Technology Laboratory (NETL), "An Assessment of Supercritical CO2 Power Cycles Integrated with Generic Heat Sources," in *4th International Symposium – Supercritical CO2 Power Cycles*, September 9-10, 2014.

[3] M. Mecheri and Y. L. Moullec, "Supercritical CO2 Brayton cycles for coal-fired power plants," *Energy,* vol. 103, pp. 758-771, Elsevier, 2016.

[4] S. Park, J. Kim, M. Yoon, D. Rhim and C. Yeom, "Thermodynamic and Economic Investigation of Coal-fired Power Plant combined with Various Supercritical CO2 Brayton Power Cycle," *Applied Thermal Engineering,* vol. 130, pp. 611-623, Elsevier, 2018.

[5] Z. Bai, G. Zhang, Y. Li, G. Xu and Y. Yang, "A Supercritical CO2 Brayton Cycle with a Bleeding Anabranch used in Coal-fired Power Plants," *Energy,* DOI: 10.1016/j.energy.2017.09.121, 2017.

[6] D. Thimsen, "Program on Technology Innovation: Modified Brayton Cycle for Use in Coal-Fired Power Plants," EPRI, 1026811, 2013.

[7] J. Miller, D. Buckmaster, K. H. T. Hart, D. Thimsen, A. Maxson, J. Phillips and S. Hume, "Comparison of Supercritical CO2 Power Cycles to Steam Rankine Cycles In Coal-Fired Applications," in *Proceedings of ASME Turbo Expo 2017: Turbomachinery Technical Conference and Exposition*, GT2017-64933, Charlotte, NC, June 26-30, 2017.

[8] National Energy Technology Laboratory (NETL), "Cost and Performance Baseline for Fossil Energy Plants Volume 1: Bituminous Coal and Natural Gas to Electricity, Revision 3," NETL, Pittsburgh, PA, June 2015.

[9] National Energy Technology Laboratory (NETL), "Development of Advanced Ultra-Supercritical (AUSC) Pulverized Coal (PC) Plants," NETL – PUB – 21022, Pittsburgh, PA, December 2016.

[10] Q. Zhao, M. Mecheri, T. Neveux, R. Privat and J.-N. and Jaubert, "Thermodynamic model investigation for supercritical CO2 Brayton cycle for coal-fired power plant application," in *5th International Supercritical CO2 Power Cycles Symposium*, San Antonio, Texas, March 29-31, 2016.

[11] National Energy Technology Laboratory (NETL), "Performance and Cost Assessment of a Coal Gasification Power Plant Integrated with a Direct-Fired sCO2 Brayton Cycle," DOE/NETL-341/063016, Pittsburgh, PA, 2017.

[12] National Energy Technology Laboratory (NETL), "Quality Guidelines for Energy System Studies, Cost Estimation Methodology for NETL Assessments of Power Plant Performance," NETL, Pittsburgh, January 2013.

[13] National Energy Technology Laboratory (NETL), "Quality Guidelines for Energy System Studies -- Fuel Prices for Selected Feedstocks in NETL Studies," NETL, Pittsburgh, PA, November 2012.

[14] National Energy Technology Laboratory (NETL), "Quality Guidelines for Energy System Studies -- Capital Cost Scaling Methodology," NETL, Pittsburgh, January 2013.

[15] AACE International Recommended Practice Number 16R-90, "Conducting Technical and Economic Evaluations -- As Applied for the Process and Utility Industries; TCM Framework 7.3 -- Cost Estimating and Budgeting," AACE, 2003.

[16] National Energy Technology Laboratory (NETL), "Techno-economics of Bituminous Coal Atmospheric Air and Oxy-Combustion Plants with Circulating Fluidized Bed Technology," NETL, Pittsburgh, June 2015.

[17] Alstom Power Inc., "Commercialization Development of Oxygen Fired CFB for Greenhouse Gas Control Volume 1: Pilot Scale Testing and Design Study of an Existing CFB Retrofit to Oxygen Firing and CO2 Capture Final Report," August 2007.

[18] Aerojet Rocketdyne, "Turbine Technology Program Phase III - Task 2(b): Supercritical CO2 Turbomachinery (SCOT) System Studies Recuperator Development," DOE Contract No. DE-FE0004002, August 13, 2014.

[19] Y. Le Moullec, "Conceptual study of a high efficiency coal-fired power plant with CO2 capture using a supercritical CO2 Brayton cycle," *Energy,* vol. 49, 2013.

[20] National Energy Technology Laboratory (NETL), "Report on Newly Developed A-USC Materials," NETL, Pittsburgh, PA, July, 2016.

[21] EPA, *Standards of Performance for Greenhouse Gas Emissions From New, Modified, and Reconstructed Stationary Sources: Electric Utility Generating Units,* Federal Register, October 23, 2015.

[22] National Energy Technology Laboratory (NETL), "Cost and Performance Baseline for Fossil Energy Plants Volume 1a: Bituminous Coal (PC) and Natural Gas to Electricity, Revision 3," NETL, Pittsburgh, July 2015.

ACKNOWLEDGEMENTS

The authors wish to thank Richard Dennis (NETL) for his support and input on this project. This work was completed under DOE NETL Contract Number DE-FE0004001. The authors also wish to thank Andrew O’Connell and Mark Woods (KeyLogic) and Dale Keairns (Deloitte Consulting, LLP) for their technical assistance and input.