The 7th International Supercritical CO_2 Power Cycles Symposium
March 31 – April 2, 2020, San Antonio, Texas
Paper #112, NETL-PUB-22739

A PERFORMANCE AND ECONOMIC COMPARISON OF PARTIAL COOLING AND RECOMPRESSION SCO$_2$ CYCLES FOR COAL-FUELED POWER GENERATION

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ABSTRACT

Recent studies have highlighted the significant challenge of designing a coal-fired primary heater for indirect supercritical CO_2 (sCO_2) power cycles that has an acceptable sCO_2 pressure drop and a reasonable capital cost. Designing plants for high efficiency rather than low cost has led to extensive use of the recompression cycle, though this leads to a narrow temperature window for heat addition to the cycle. This in turn results in high thermal recuperation rates for both the combustion system and the sCO_2 cycle to maintain high heat addition temperatures, further increasing system costs. The present study investigates the use of the partial cooling cycle as a potentially lower cost alternative to the recompression cycle for fossil-fueled power applications.

To fully explore the design tradeoffs between the cycles, a simplified circulating fluidized bed (CFB) design tool is developed and calibrated against steam plant boiler design data from the STEAMPRO software package. The CFB design tool calculates the sCO_2 pressure drop and overall CFB cost given some basic CFB design parameters, which are used as inputs to techno-economic models of partial cooling and recompression cycles with and without reheat. The results show that the partial cooling cycle yields lower recuperation and CFB costs relative to recompression cycles, which more than offsets the capital costs of the additional cooler and a high volumetric flow compressor required in the cycle. These effects are due to the higher pressure ratio of the partial cooling cycle, which generally leads to lower sCO_2 flow rates, lower recuperation duties, and a larger CFB temperature window for heat addition to the cycle. The net result is a lower cost of electricity (COE) for partial cooling cycles relative to recompression cycles in this application. Reheating is shown to significantly increase sCO_2 cycle pressure drop for both cycle types, increasing the preferred CFB tubing diameter, while significant reductions in COE can be attained for minimal loss in efficiency by considering smaller CFB tubing sizes in non-reheated cycles. While these results require validation with more detailed CFB designs, they highlight the importance of focusing R&D efforts on development of the primary heater, which is the main contributor to plant cost, sCO_2 cycle pressure drops, and hence the COE of the plant.

INTRODUCTION

Indirect-fired supercritical CO_2 (sCO_2) power cycles are being pursued as an attractive alternative to steam Rankine cycles for a wide range of thermal heat sources (e.g., nuclear, concentrated solar, fossil, biomass, waste heat). Due to its high thermal efficiency, the recompression cycle (RC) is presently the most widely adopted indirect sCO_2 power cycle configuration in the sCO_2 research community, particularly for heat sources that operate over a relatively narrow temperature window. In this cycle, a primary heat exchanger (PHX) is used to heat the high-pressure CO_2 to the turbine inlet temperature. The CO_2 expands and partially cools through the turbine. The remaining thermal energy from turbine exhaust is recuperated to heat the high-pressure CO_2 from compressors. Because of the mismatch in thermal capacitance between the high-pressure and low-pressure CO_2, a higher recuperator effectiveness cannot be achieved unless a portion of the low-pressure working fluid bypasses the cooler. This bypass compression step is the key feature in the recompression sCO_2 Brayton cycle [1]. The main compressor (MC) takes advantage of high working fluid density near the CO_2 critical point (31 °C, 73.7 bar) to reduce the power required for compression and hence, increase the power output of the cycle. An added benefit of indirect sCO_2 cycles is that the high operating pressure and low cycle pressure ratios allow for use of compact turbomachinery and heat exchangers.

System analysis studies by several research organizations including NETL have projected efficiency improvements of 2–6 percentage points for recompression sCO_2 cycles compared to steam Rankine cycles operating at similar process conditions [1, 2, 3, 4]. Recently, NETL performed a techno-economic analysis of a coal-fired power plant based on the recompression sCO_2 power cycle, using an oxy-combustion circulating fluidized bed (CFB) as the primary heater [2]. The study showed that incorporating main compressor intercooling improves the plant efficiency and lowers the cost of electricity (COE), while the addition of a single stage of reheat always improves efficiency, but only improves COE for lower turbine inlet temperatures.

Prior NETL studies assumed CO_2 pressure drops in the range of 68 – 136 kPa for the CFB [2]. Recent detailed design study conducted by EPRI showed that the CO_2 pressure drops are in the range of 2000 – 3000 kPa for the primary heater [3, 5], though in EPRI's study the primary heat source is a pulverized coal combustor. Due to lower pressure ratios and the higher specific compression power for sCO_2 relative to water, the sCO_2 power cycles are more sensitive to cycle pressure drops than steam Rankine cycles. This problem is exacerbated by high sCO_2 cycle mass flows relative to steam, due to the high degree of recirculation. The combined need for low CO_2 pressure drop and high CO_2 flow rate presents a significant design challenge for coal-fired primary heaters for sCO_2 power cycles [3].

CFB cost estimates used in prior NETL studies utilize a heat duty-based cost scaling algorithm derived using STEAMPRO data for steam Rankine plants [2]. Owing to internal heat recuperation, the heat transfer driving forces within the CFB are significantly different for sCO_2 power cycles compared to steam cycles. As noted in the prior studies, using steam as the surrogate fluid introduces significant uncertainty in the CFB cost estimates for sCO_2 applications. Typically, CFB capital cost is significantly higher than the capital cost of entire sCO_2 power block. Therefore, it is important to improve the accuracy of CFB cost estimates for more accurate calculation of the plant's COE and a better assessment of the technology.

Though the recompression cycle is widely investigated, it may not be an optimum cycle configuration for all heat sources. For heat sources that require a larger window of heat addition (e.g., concentrated solar, coal combustion), the partial cooling cycle (PCC) might be an attractive alternative. The PCC has a higher pressure ratio than the recompression cycle (RC) and therefore, increases the specific work of the turbine and lowers the recirculating CO_2 flow rate [6, 7]. Due to the larger window of heat addition, the driving forces with the CFB are expected to be higher, and CFB costs are expected to be lower, for the PCC compared to the RC. In studies utilizing nuclear [6] and concentrated solar [7] heat sources, the PCC is shown to offer slightly lower efficiency and lower recuperation duties compared to the RC equivalent. However, these studies focused only on the efficiency/performance benefits, and economic analyses of the PCC are not reported in open literature. Moreover, to the authors' best knowledge, application of a partial cooling cycle for coal-fired power plants is not reported in open literature.

Objectives of Current Study

In light of the above discussion, the main objective of the current study is to conduct techno-economic analyses to compare the performance and COE of the recompression and partial cooling cycles for coal-fired power plants. The paper will first provide details about the plant configurations, plant modeling details and assumptions, and economic analysis methodology. Next, a simplified CFB design tool that is developed to capture the impact of design variables on CO_2 pressure drop and CFB cost will be discussed. The goal of this design tool is to improve the accuracy of CFB cost estimates and the corresponding sCO_2 pressure drops. The paper will then be followed by discussion of results and plant sensitivity analyses on CFB CO_2 pressure drop and turbine inlet temperature.

METHODOLOGY

Plant Configuration

Figure 1 shows a block flow diagram for the oxy-fired indirect sCO_2 power plants in this study. Illinois No.6 coal is fed to the atmospheric pressure CFB heat source together with limestone that is added for sulfur capture. Oxygen is provided by a low-pressure air separation unit (ASU). The bed temperature is assumed to be 871.1 °C (1600 °F) with ~99% carbon conversion and 1 percent heat loss. These assumptions are consistent with reference steam Rankine plants utilizing CFB boilers [8].

Heat from the CFB is transferred to the power cycle via tube banks within the combustor. Flue gas then passes through a series of heat exchangers to maximize the heat recovery and improve plant efficiency. The economizer recovers high-quality heat from combustion flue gas in the convective back pass of the CFB and heats a portion of the high-pressure CO_2 in parallel to the high temperature recuperator (HTR). After the economizer, flue gas pre-heats the O_2/CO_2 oxidant mixture and passes through a bag house for removal of particulate matter. The flue gas then heats nitrogen which is used to regenerate the adsorbent in the ASU air pre-purifier. Finally, a flue gas cooler transfers additional low-quality heat to a portion of the cold CO_2 exiting the main compressor, in parallel with the low temperature recuperator (LTR). Condensate is removed from the cooled flue gas in a water knockout tank. A portion of the flue gas, which is primarily CO_2, is recycled to the CFB as fluidizing gas. The remainder of the dried flue gas is fed to the cryogenic CO_2 processing unit (CPU) for further purification and compression.

As described above, sCO_2 power cycles of interest in the present study are the RC and PCC, both of which are shown in Figure 1. CO_2 heated in the CFB is expanded in the turbine for power generation, and its exhaust is cooled in the HTR and LTR. In the case of recompression cycle, the low-pressure CO_2 stream leaving the LTR is split into two portions; one is cooled (in the Main Cooler) and passes to the main CO_2 compressor, and the other bypasses the main cooler directly to the bypass CO_2 compressor. The cold high-pressure CO_2 leaving the compressors is heated in the flue gas cooler, LTR, HTR and economizer before final heating to the turbine inlet temperature in the CFB.

For the PCC, the addition of a pre-cooler and boost compressor (highlighted in green in Figure 1) are needed to handle the larger pressure ratio. After exiting the LTR, the entire low-pressure CO_2 stream is further cooled in the pre-cooler and compressed to cycle intermediate pressure in the boost CO_2 compressor. After the boost compressor, the operation of the PCC is similar to that of the RC. It should be noted the RC and PCC configurations shown here are modified from cycles known by these names in the literature to include two additional heat exchangers (economizer, flue gas cooler) in parallel to the cycle recuperators (HTR, LTR) to maximize heat recovery from flue gas. These modifications may not be necessary for nuclear, concentrated solar, or other heat sources.

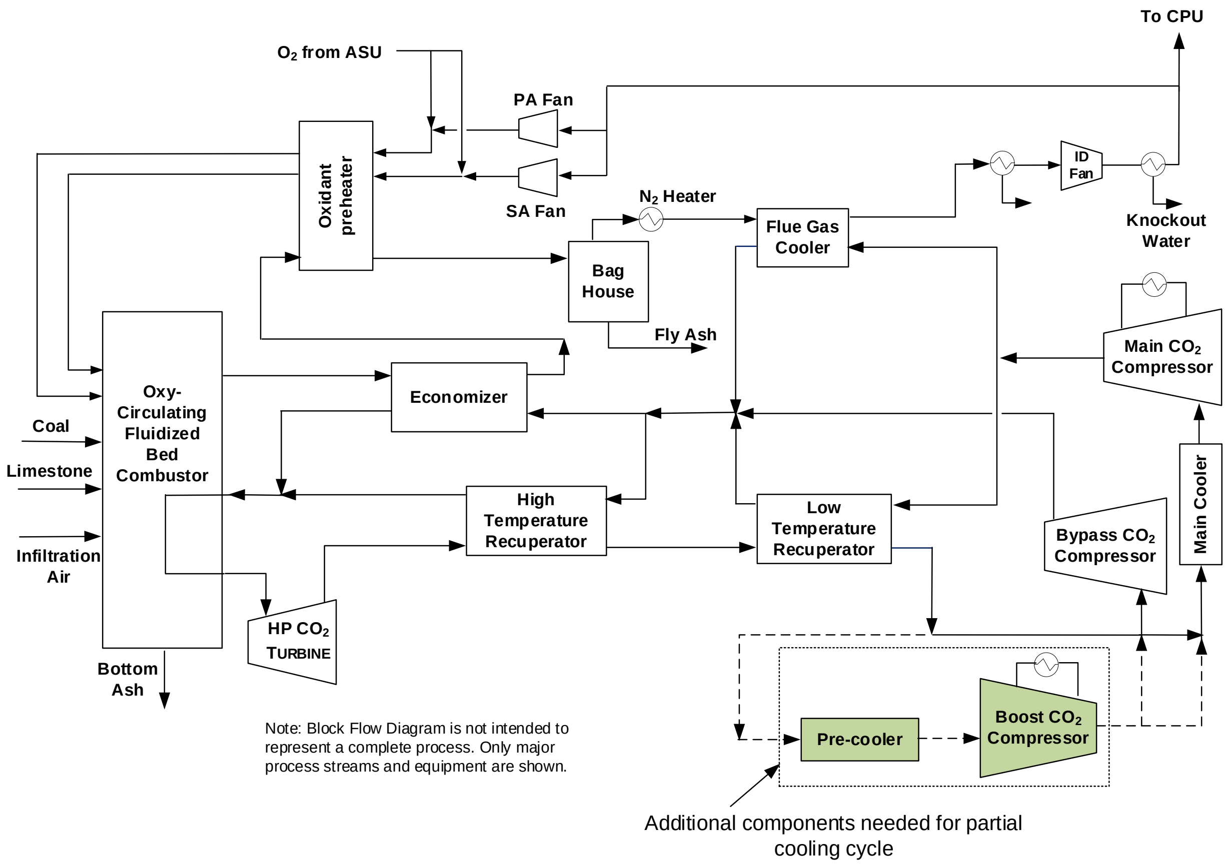

Figure 1: Block flow diagram for the recompression and partial cooling power cycles

Modeling Approach and Assumptions

A steady-state model of the plants described above is developed using Aspen Plus® (Aspen). The individual unit operation models are the same as those used in the previous NETL oxy-coal indirect sCO_2 cycle study [2], which included a CFB heater with carbon capture enabled by oxy-combustion and a net power output of roughly 550 MWe. Details on the design basis, assumed feed composition, and state point tables can be found in Ref. [2].

Modeling of sCO_2 power cycles requires accurate calculation of CO_2 thermophysical properties, particularly near the critical point. The Span-Wagner equation of state [9], the most accurate property method for processes consisting of pure CO_2, is incorporated into the REFPROP physical property method developed by National Institute of Standards and Technology (NIST) and is implemented in Aspen as the physical property method REFPROP [10]. This method is used for all physical property estimates within the sCO_2 power cycle. Leakage and make-up flows are not modeled.

The sCO_2 cycle recuperators and coolers are modeled as adiabatic counterflow heat exchangers with perfect thermal mixing in the transverse direction and negligible heat transfer in the axial direction. A minimum temperature approach of 5.6 °C (10 °F) is enforced within the recuperators and within the flue gas cooler. Due to a mismatch in the specific heat capacities, the minimum temperature approach typically occurs at the cold end of the HTR and/or within the LTR. The overall conductance (UA) of the recuperators and coolers is calculated using discretized heat exchanger model that includes 100 segments (of equal heat duty) in the direction of flow, allowing for more accurate estimation of UA and detection of internal pinch points.

All of the coolers are assumed to be sCO_2-water heat exchangers with an evaporated forced draft wet cooling tower used to reject the heat absorbed by water to the atmosphere. The cooling tower water inlet and outlet temperatures assumed to be 26.7 °C (80 °F) and 15.6 °C (60 °F), respectively, and the ambient dry and wet bulb temperatures are assumed to be 15 °C and 10.8 °C, respectively, consistent with other NETL techno-economic analyses [11].

Table 1 summaries the sCO_2 power plant design conditions for the RC and PCC cases. Four plant designs are selected for investigation in the present study. Two of the plants are based on the RC without reheat (RC760) and with reheat (RhtRC760). Two of the plants are based on PCC without reheat (PCC760) and with reheat (RhtPCC760). These four plants will be described using their assigned nomenclature in the rest of the paper.

Table 1: sCO_2 power plant design conditions for recompression and partial cooling cycles

Parameter	Recompression cycle	Partial cooling cycle
Coal flow rate (kg/h)	193,230	
Limestone flow rate (kg/h)	46,486	
Oxygen flow rate (kg/h)	412,962	
Turbine inlet temperature (°C)	760	
Cooler outlet temperature (°C)	35	
Cycle minimum pressure (MPa)	9.1	5.3
Cycle maximum pressure (MPa)	34.6	
Cycle intermediate pressure (MPa)	-	8.27
Number of compressor intercoolers (Main/Boost compressor)	1	2
Recuperator and flue gas cooler minimum approach temperature (°C)	5.6	
Recuperator, cooler pressure drops (kPa)	68.9	
Intercooler pressure drop (kPa)	13.8	
Turbine isentropic efficiency (%)	92.7	
Compressors isentropic efficiency (%)	85	

The main compressor for the RC760 and RhtRC760 plants is a two-stage compressor with a single stage of intercooling. The main and boost compressors for the PCC760 and RhtPCC760 plants are both three-stage compressors with two stages of intercooling. For cases with turbine reheat (RhtRC760, RhtPCC760), a single stage of reheat is used and the pressure ratio of the main and reheat turbines is assumed to be equal. The cycle minimum and intermediate pressures have been selected to optimize plant efficiency in this study, but will be used to optimize the plant's COE in future work.

In addition to the design parameters, the following constraints are imposed for plant modeling: 1) The split flow between compressors is adjusted such that the high-pressure CO_2 temperature exiting LTR is equal to the bypass compressor outlet temperature. 2) The split flow between LTR and flue gas cooler is adjusted such that the flue gas cooler CO_2 outlet temperature is equal to the high-pressure CO_2 temperature exiting LTR. 3) The split flow fraction between HTR and economizer is adjusted such that the economizer CO_2 outlet temperature is equal to the high-pressure CO_2 temperature exiting HTR. Note that these constraints commonly maximize the efficiency of the plant, though relaxing these constraints may yield better economic performance.

Economic Analysis Methodology

Based on the steady-state modeling results, pertinent operating conditions of the cycle and balance of plant (BOP) components are used to estimate costs for equipment, installation, contractor fees, and contingencies. Following a standard cost estimating methodology [12, 13], these are summed to arrive at total plant cost (TPC), expressed in June 2011 dollars for consistency with other NETL studies [14, 15]. Using a consistent methodology and set of assumptions [13], owner's costs are also calculated as a fraction of the operations and maintenance (O&M) costs and TPC. Owner's costs are those arising from plant pre-production and inventories as well as land purchase and other items. These are added to TPC to arrive at the total overnight cost (TOC), which is used to calculate the capital component of the cost of electricity (COE).

Capital Cost Estimates for sCO_2 Cycle Components

No oxy-CFB cost estimates are available in public domain for the sCO_2 power cycles. The approach taken in the present study is to develop a simplified CFB design tool that can be used to estimate the capital cost of CFB with reasonable accuracy and at the same time capture the impact of varying sCO_2 pressure drop, turbine inlet pressure and temperature on the CFB capital cost. Details of the simplified CFB design tool are described in the subsequent section. It should be noted that the economizer and oxidant preheater from Figure 1 are also part of the CFB capital cost estimates.

The rest of the sCO_2 cycle component capital costs are calculated using component cost algorithms developed under prior work for indirect fired sCO_2 power cycle applications [16]. The sCO_2 compressors are assumed to be barrel-type centrifugal compressors which are scaled to inlet volumetric flow rates. The recuperator and cooler (excluding the flue gas cooler) capital costs are calculated using a UA-based cost scaling algorithm developed from vendor quotes, which includes a temperature correction factor for maximum temperatures greater than 550 °C [16]. To reduce capital cost, the HTR is split into two units when the hot side fluid temperature is above 550 °C and the two units are assumed to be installed in series. The turbines are assumed to be axial turbines and their costs are calculated using shaft power-based cost scaling algorithm developed from vendor quotes [16]. The flue gas cooler cost is scaled based on an analogous syngas cooler from IGCC pathway studies. The reference syngas cooler used for scaling has a heat duty of 78.2 MW_{th} and a bare erected cost (BEC) of $18,614,000. The CO_2 cycle piping costs are dominated by high temperature and pressure piping between the CFB and turbine. For this study, the pipe lengths between the HTR and CFB and between the CFB and turbine are assumed to be 150 ft. The pipe inner diameter is calculated assuming fluid velocity of 150 ft/sec. The selection of pipe material and thickness is based on a NETL report [17]. The material capable of meeting the service temperature and pressure with lowest estimated pipe cost is selected. A 30 percent installation factor is assumed for the piping.

Other Capital Cost Estimates

Aside from the power cycle, the BOP unit operations and equipment items are analogous to those found in a reference steam Rankine plant [8], and are scaled using a consistent methodology used in all NETL studies [12]. The cost estimates for these items are based on a combination of vendor data, estimates from Worley-Parsons, power law scaling and correlations that are fit to historical cost estimates published in previous NETL reports [2, 13]. The same BOP cost accounts as for the reference steam Rankine plant are used.

Cost of Electricity Calculation

The COE consists of five components representing normalized contributions from capital costs, fixed O&M costs, variable O&M costs, fuel costs, and the transportation & storage (T&S) costs for captured CO_2. These are calculated following a consistent methodology used in all NETL studies [13]. The financial assumptions used in the COE calculation for the sCO_2 plant are detailed in Ref. [2].

Simplified CFB Design Tool

As noted above, a simplified CFB model is developed as part of this study to capture the impact of sCO_2 pressure drop, turbine inlet temperature & pressure, and choice of reheat/non-reheat on the CFB cost, at least in a qualitative sense. The model includes a bottoms-up CFB cost estimate derived using the STEAMPRO software package. This section summarizes details about the CFB model and several assumptions used to develop the model.

First, several simulations are conducted in STEAMPRO for an air-fired CFB steam Rankine plant to aid in the development of a CFB model for sCO_2 power cycles. The breakdown of the CFB cost in STEAMPRO includes several cost sub-accounts. To be consistent with STEAMPRO, the CFB cost breakdown is redefined for sCO_2 power cycles as shown in Table 2. Sub-account SA1 includes the cost of radiative tube banks (waterwall, wingwall, and radiative superheaters and reheaters), while sub-account SA2 includes the cost of all the convective tube banks, which are located within the convective backpass of CFB. Sub-account SA3 includes the cost of interconnecting piping, cyclones, refractory and other miscellaneous items. Sub-account SA4 covers the cost of tubular oxidant pre-heater and sub-account SA5 includes soot blowers, ducts, feeders, fans and other structural element costs. It should be noted that the interconnecting piping and cyclones included in SA3 are part of the sub-accounts SA1 and SA2 within STEAMPRO, however, a separate cost sub-account (SA3) is created for these components due to lack sufficient information to calculate their cost scaling.

Table 2: CFB cost breakdown for sCO_2 power cycles

CFB cost sub-account	Description
SA1	Furnace radiative tube banks
SA2	Convective tube banks
SA3	Interconnecting piping, cyclones, refractory etc.
SA4	Tubular oxidant pre-heater
SA5	Rest of the CFB (Soot blowers, ducts, feeders, fans, structural)

Out of all the sub-accounts from Table 2, SA3 through SA5 are assumed to be independent of the working fluid (same for sCO_2 and steam cycles) and the maximum CFB temperature. The tubular oxidant pre-heater (SA4) cost is calculated using a UA-based scaling algorithm derived from STEAMPRO data. Likewise, the rest of the CFB (SA5) cost is calculated using a CFB heat duty-based scaling algorithm derived from STEAMPRO data.

CFB Tube Bank Performance Model

Sub-account SA1 and SA2 costs from Table 2 are expected to be different for steam and sCO_2 power cycles due to the difference in working fluid operating conditions (mass flow, temperature, pressure etc.) and properties. Likewise, the working fluid pressure drops within the CFB are expected to be different for steam and sCO_2 power cycles. Therefore, a tube bank sizing and cost model is developed as part of the simplified CFB design tool to calculate the sCO_2 pressure drop and tube bank costs. The model inputs and outputs are listed in Table 3.

Table 3: Tube bank sizing and cost model inputs and outputs

Inputs	Outputs
Steam/sCO_2 flow rate	Tube bank heat duty
Steam/sCO_2 inlet pressure and temperature	Driving force ($LMTD$)
Steam/sCO_2 outlet temperature	Steam/sCO_2 pressure drop
Flue gas inlet temperature	Required heat transfer area
Flue gas outlet temperature	Tube wall thickness
Tube material	Maximum wall temperature
Tube outer diameter	Tube bank cost
Tube length	

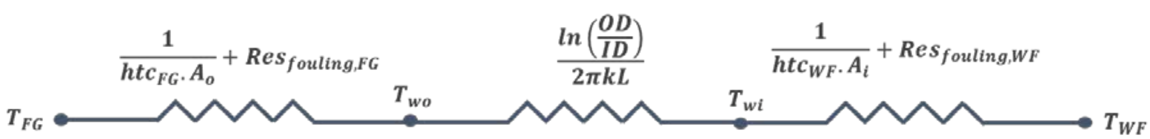

Figure 2: Heat transfer resistance network used for the CFB tube bank

The heat transfer resistance network used in the model is shown in Figure 2. To calculate the total heat transfer resistance (and overall conductance), the working fluid heat transfer coefficient (htc_{WF}) is calculated using Dittus-Boelter correlation. The flue gas heat transfer coefficients (htc_{FG}) are assumed to be same as those for the steam Rankine plants from STEAMPRO; 200 W/m^2-K and 75 W/m^2-K for the radiative and convective tube banks respectively. It should be noted that htc_{FG} depends on several factors such as CFB operating pressure, fluidization velocity, etc., and a more detailed model is needed to improve accuracy. The flue gas and working fluid fouling resistances per unit area, $Res_{fouling}$, are assumed to be 0.00088 W/m^2-K and 0.000176 W/m^2-K respectively, based on STEAMPRO data. Using these assumptions, the tube bank heat duty, Q, can be correlated to the driving force and heat transfer resistance as follows:

$$Q = \left(\frac{1}{Res_{total}}\right).N_{tubes}.LMTD$$

$$LMTD = \frac{(T_{FG,in} - T_{WF,out}) - (T_{FG,out} - T_{WF,in})}{LN\left(\frac{T_{FG,in} - T_{WF,out}}{T_{FG,out} - T_{WF,in}}\right)}$$

where Res_{total} is the total heat transfer resistance in K/W and is equal to the sum of all heat transfer resistances from Figure 2. $T_{FG,in}$ and $T_{FG,out}$ are the inlet and outlet temperatures of flue gas, and $T_{WF,in}$ and $T_{WF,out}$ are the inlet and outlet temperatures of working fluid. N_{tubes} is the total number of tubes needed to meet the tube bank heat duty.

The tube wall thickness, t_{wall}, for a known tubing outer diameter (OD) and design condition (temperature, pressure) is calculated according to the ASME A-317 equations as follows [18]:

$$t_{wall} = \frac{OD\left(1 - e^{-\frac{P}{SE}}\right)}{2} + C + F$$

where P is the design pressure in MPa, S is the ASME code allowable stress in MPa, E is the joint efficiency, assumed to be 1 here for seamless tubes. C and F are the allowances for threading and expansion and are assumed to be zero. The design pressure is calculated by multiplying the maximum pressure of working fluid by a safety factor of 1.2. The ASME code allowable stress is a function of the tube material and design temperature. The design temperature is taken as the maximum wall temperature (T_{wo}) experienced by the tube and is calculated as follows (according to heat transfer resistance network from Figure 2):

$$T_{wo} = T_{FG} - (T_{FG} - T_{WF})\left(\frac{\frac{1}{htc_{FG}.A_o} + Res_{fouling,FG}}{Res_{total}}\right)$$

It should be noted that calculation of the heat transfer area (number of tubes), maximum tube wall temperature and tube wall thickness is an iterative process. Therefore, the above-mentioned equations are solved iteratively within specified tolerance using initial guess values for these variables. The tube lengths and number of passes for the tube banks are kept same as the reference STEAMPRO cases. For example, length of the waterwall in STEAMPRO reference cases is 65.6 m and the same value is used for sCO_2 power cycle CFB waterwall.

Once the tube thickness and tube bank geometry are determined, the working fluid pressure drop (ΔP_{WF}) is calculated iteratively using the Colebrook friction factor correlation:

$$f = \left(\frac{1}{-2\log\left(\frac{K}{3.7} + \frac{2.51}{Re_{WF}\sqrt{f}}\right)}\right)^2$$

$$\Delta P_{WF} = 4f.\frac{G^2}{2\rho_{WF}}.\frac{L}{ID}.N_{pass}$$

where f is the Fanning friction factor, Re_{WF} is the working fluid Reynolds number, K is the normalized roughness factor, set to 0.00015, G is the mass flux in kg/m^{2}s, ρ_{WF} is the working fluid density in kg/m^3, L is the tube length in meters, N_{pass} is the number of passes.

CFB Tube Bank Cost Model

Once the tube wall thickness and heat transfer area are known, the total weight of tubes (M_{tubes}) and cost of tube bank ($Cost_{tubes}$) are calculated as follows:

$$M_{tubes} = \frac{\pi}{4}(OD^2 - ID^2) \times L \times N_{tubes} \times \rho_{alloy}$$

$$Cost_{tubes} = M_{tubes} \times SC_{alloy} \times 1.5$$

where ρ_{alloy} is the tube material density, SC_{alloy} is the specific tube material cost per unit weight, and the 1.5 multiplication factor accounts for the fabrication costs.

CFB Tube Bank Model Validation

The simplified CFB design tool is calibrated/validated with STEAMPRO data for two cases, as shown in Table 4. Both these cases have a net power output of ~550 MWe, turbine inlet temperature and pressure of 760 °C and 29.6 MPa respectively with a single stage of reheat, but have different tube bank arrangements within the CFB. The sub-account SA3 cost is assumed to be $85M, which minimizes the difference between calculated and STEAMPRO costs to <±4% for both the cases, and is subsequently applied to the sCO_2 power cycles. Note however that the interconnecting piping cost is a function of the working fluid pressure, temperature and tube diameter. For example, smaller diameter tubes would require more flow distributions to the tube bank and the cost of interconnecting piping is expected to be higher in such cases. Therefore, better cost data is needed for SA3 for a more accurate estimation of the CFB cost.

Table 4: Comparison of CFB cost estimates using developed CFB model and STEAMPRO

CFB cost sub-account	Case1	Case2
SA1	$188,427,401	$118,442,386
SA2	$66,891,332	$18,731,595
SA3	$85,000,000	$85,000,000
SA4	$31,572,263	$37,298,332
SA5	$25,882,768	$25,881,783
Total calculated CFB cost	**$397,773,764**	**$285,354,096**
CFB cost from STEAMPRO	**$407,829,000**	**$275,026,300**
% Difference	**-2.5%**	**+3.8%**

The calculated steam pressure drops and maximum tube wall temperatures are also compared to STEAMPRO data for several tube banks. The CFB tube bank model is able to calculate the pressure drop for tube banks such as economizer, convective superheaters and reheaters with good accuracy. Maximum deviation occurred for reheaters likely due to different surface roughness assumption compared to STEAMPRO.

The working fluid pressure drop depends on the tube outer diameter, tube material (impacts the wall thickness), and arrangement of heat transfer surfaces/tube banks. All of these design variables also impact the CFB cost. Sensitivity analyses relating the CFB cost to its sCO_2 pressure drop are derived by varying the CFB tube diameter in the present study.

RESULTS AND DISCUSSION

The performance and economic analysis results of the RC and PCC configurations with and without reheat (cases RC760, RhtRC760, PCC760 and RhtPCC760) are summarized below, including a summary of the CFB design results. These are followed by sensitivity analyses on CFB tubing diameter and turbine inlet temperature.

Performance Results

Table 5 shows the performance summary results for all four cases, each run with the same coal flow rate, and each yielding 92.7% carbon capture. Out of the four plants, RhtPCC760 case has the highest net power output and plant efficiency followed by RC760, RhtRC760 and PCC760. The net plant efficiency of RhtPCC760 case is 2.55 percentage points higher than a reference oxy-CFB AUSC Rankine plant (B24F) which has a net plant efficiency of 36.9% [8].

Table 5: Performance comparison between recompression and partial cooling cycle cases

Parameter	RC760	RhtRC760	PCC760	RhtPCC760
Net plant efficiency (HHV %)	38.6%	38.4%	38.0%	39.5%
Cycle efficiency (%)	52.0%	52.6%	51.8%	53.6%
Cycle specific power (kJ/kg)	160.3	164.8	216.6	233.1
sCO_2 flow rate (kg/s)	4,353.5	4,304.5	3,221.7	3,224.8
Power Generation Summary (MW_e)				
Turbines gross power	937.3	933.9	942.6	955.1
Main compressor power	-116.1	-116.4	-81.2	-78.5
Boost compressor power	--	--	-62.6	-60.4
Bypass compressor power	-97.0	-97.2	-86.3	-82.9
Generator losses	-10.9	-10.8	-10.7	-11.0
Other auxiliaries	-150.6	-150.6	-148.0	-147.8
Net power plant output	562.7	558.8	553.9	574.6
Power Cycle Heat Duties (MW_{th})				
CFB thermal input	1,106.2	1,106.2	1,111.3	1,111.3
HTR duty	1,953.9	2,464.0	1,263.0	1,677.9
LTR duty	730.8	732.3	406.2	391.8
Main Cooler duty	489.6	490.6	198.2	191.6
MC Intercooler duty	138.9	137.2	165.0	159.5
Pre-Cooler duty	0.0	0.0	94.8	91.6
BC Intercooler duty	0.0	0.0	177.6	171.3

The sCO_2 mass flow rate of partial cooling cycles (PCC760, RhtPCC760) is roughly 25% lower than the recompression cycles (RC760, RhtRC760). Due to lower cycle flow rate, the main compressor's power consumption is reduced for partial cooling cycles. The boost compressor operates in the superheated vapor region where the working fluid density is low and therefore, the boost compressor power consumption is quite significant even with two stages of intercooling. The bypass compressor power consumption is comparable for the recompression and the partial cooling cycles. Due to lower CO_2 flow rate, one would have expected lower power consumption for the bypass compressor in partial cooling cycles. However, in partial cooling cycles the bypass compressor pressure ratio is higher and also, a larger fraction of CO_2 exiting the boost compressor flows to the bypass compressor.

Due to lower CO_2 flow rate, the recuperator heat duties are significantly lower for the partial cooling cycles compared to the recompression cycles. This is illustrated in the difference in specific enthalpies for recuperation processes on the P-h diagram in Figure 3. Factoring in the differences in mass flow rate as well, the total recuperation heat duty for the PCC760 case is about 38% lower than the RC760 case, per Table 5. Also of note in Figure 3 is the larger temperature (~enthalpy) window for heat addition in the CFB for the PCC760 case. For the same turbine inlet temperature, the PCC turbine's larger pressure ratio relative to the RC case reduces the turbine outlet temperature, equal to the low pressure HTR inlet temperature. This in turn reduces the high pressure HTR outlet and CFB inlet temperature.

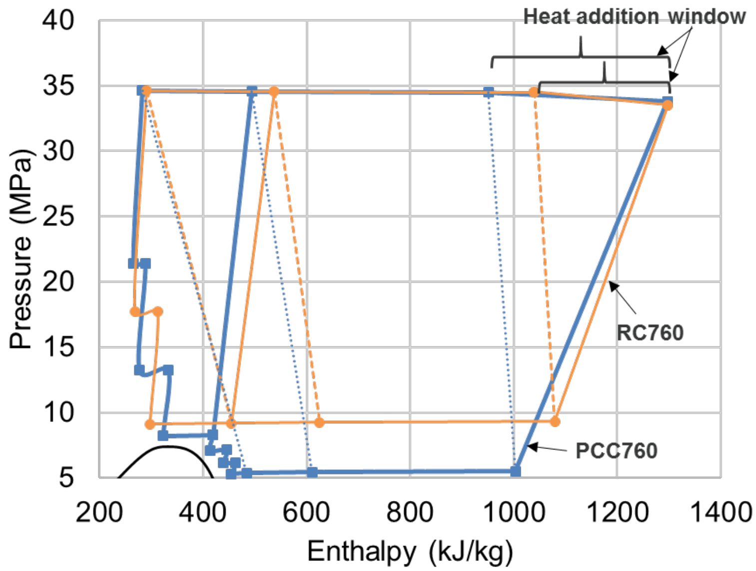

Figure 3: P-h diagrams of the partial cooling cycle (PCC760) and recompression cycle (RC760)

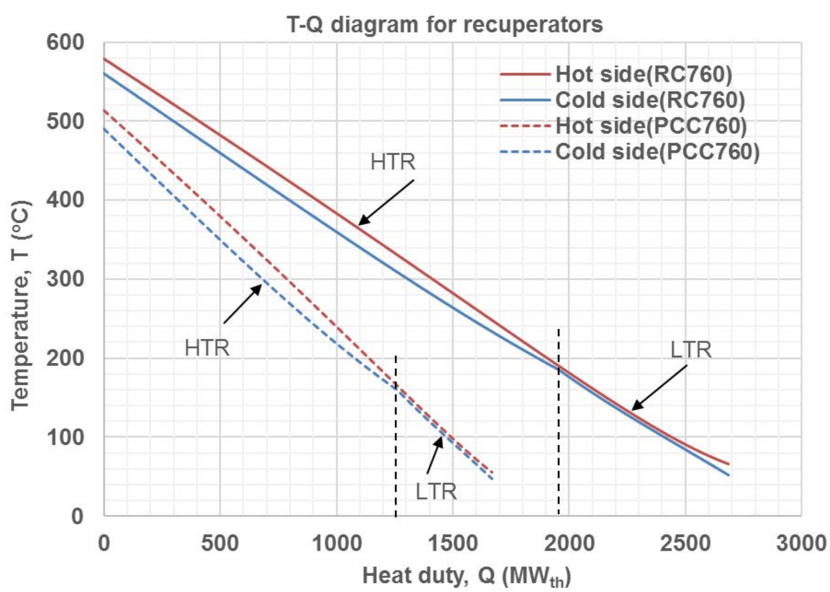

Figure 4: T-Q diagram for the recuperators in the partial cooling cycle (PCC760) and recompression cycle (RC760)

These changes in recuperation temperatures are shown in the T-Q diagram in Figure 4, where the demarcation between the HTR and LTR is shown as a dashed vertical line. It can be seen that the temperature of high-pressure CO_2 (cold side) exiting the HTR is significantly lower for the PCC760 case. Under identical CFB operating conditions, this leads to the larger temperature window of heat addition noted above, as well as larger driving forces (LMTD) that can reduce the required heat transfer surface area for PCC cases. A similar set of conclusions can be drawn for the reheat cases as well. However, reheat increases the turbine exhaust temperature and consequently, the temperature of high-pressure CO_2 exiting the HTR is also higher for reheat cases. As described earlier, the minimum temperature approach typically occurs at the cold end of HTR and/or inside the LTR, as shown in Figure 4.

CFB Design Summary

The simplified CFB design tool described above is used to calculate basic CFB design parameters, as well as the accompanying sCO_2 pressure drop and CFB cost. The primary heater and the reheat heater tube banks experience wall temperatures greater than the turbine inlet temperature (760 °C) and hence, require high Ni-alloy tubes. The material for these tube banks is assumed to be Inconel alloy 740H (IN740H). However, the economizer tube bank experiences significantly lower wall temperatures and the tube material is assumed to be TP347 stainless steel. These materials are selected by STEAMPRO for the high temperature steam CFB tube banks, and are maintained here for consistency. For all the tube banks, the tubing outer diameter (OD) is assumed to be 2.0”, which is again consistent with the steam CFB design.

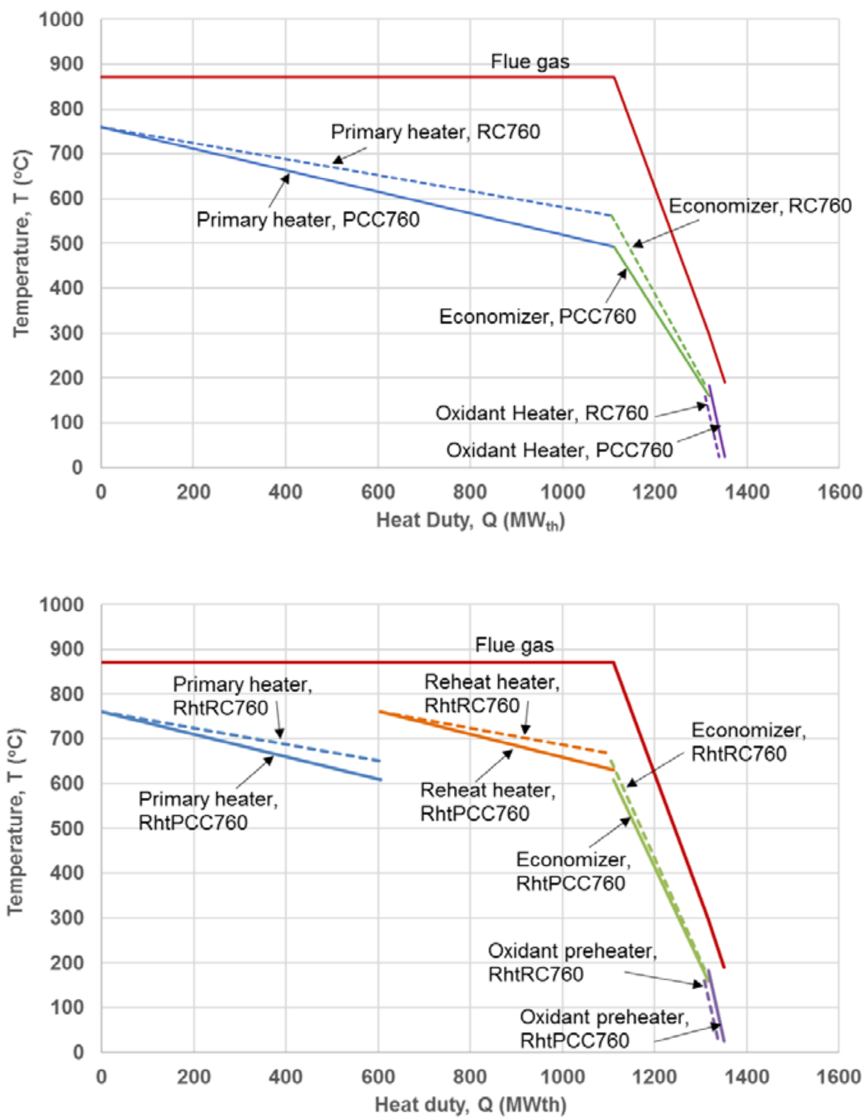

Figure 5: T-Q diagram for recompression and partial cooling cycles; Top plot is for cycles without reheat (RC760, PCC760) and bottom plot is for cycles with reheat (RhtRC760, RhtPCC760)

The CFB temperature vs heat duty (T-Q) diagram for all cases is presented in Figure 5. The primary heater and reheat heater tube banks are assumed to be located in the fluidized bed region as part of the waterwall, while the economizer tube bank is located in the convective backpass of CFB. These are very crude assumptions that must be revisited in a more detailed CFB design effort, but allow for a relative assessment of the impacts on CFB cost. From Figure 5, it is clear that the heat transfer driving forces within the CFB are higher for the partial cooling cycles (PCC760, RhtPCC760) compared to the recompression cycles (RC760, RhtRC760).

Table 6 shows the CO_2 pressure drop and CFB cost breakdown for all four cases under these design conditions. Following observations can be made from Table 6: 1) Due to lower CO_2 flow rate, the sCO_2 pressure drop is lower for partial cooling cycles than their recompression counterparts. For example, the total pressure drop for the RhtPCC760 case is ~38% lower than that for the RhtRC760 case. 2) The sCO_2 pressure drop is significantly higher for the reheat cases compared to the non-reheat cases. This is primarily due to increased fluid velocities within the tube banks for reheat cases. 3) Owing to the larger driving force, the CFB cost is lower for the partial cooling cycles than their recompression counterparts. 4) The CFB cost is lower for the reheat cases compared to the non-reheat cases, due to the lower design/operating pressures for the reheat tube bank, which reduces the required tube wall thickness and material cost.

Table 6: CFB pressure drop and cost breakdown for the recompression and partial cooling cycles with and without turbine reheat

	RC760	RhtRC760	PCC760	RhtPCC760
CO_2 pressure drop breakdown				
Primary heater (kPa)	999	2,092	719	1,367
Reheat heater (kPa)	-	678	-	376
Economizer (kPa)	49	132	79	55
Total pressure drop (kPa)	1,048	2,902	798	1,798
Cost breakdown (x$1000)				
SA1	$505,267	$467,863	$460,723	$407,875
SA2	$21,561	$38,868	$17,134	$25,624
SA3	$85,000	$85,000	$85,000	$85,000
SA4	$3,101	$3,101	$4,351	$4,351
SA5	$30,034	$30,004	$30,277	$30,277
Total CFB cost	$644,963	$624,865	$597,485	$553,126

It is important to note that the CFB design summary presented in Table 6 is based on a simplified model and therefore, significant uncertainty exists in the cost and/or pressure drop estimates. For example, the sub-account SA3 cost (includes interconnecting piping) is assumed to be a constant value of $85M. Also, no attention is paid to the arrangement of the tube banks within the furnace section (fluidized bed region). Depending on the dimensions of the furnace section, the arrangement of tube banks might have to be reconsidered to achieve a feasible design. Nevertheless, the model can qualitatively capture the impact of several design variables (like tube diameter, turbine inlet temperature and pressure, reheat/non-reheat etc.) on CO_2 pressure drop and CFB cost.

Economic Analysis Summary

Table 7 shows the economic analysis summary for all four cases. The table includes capital costs for the entire plant, sCO_2 power block components and a COE breakdown. For these cases, CFB capital costs in Table 6 constitute the largest portion of the CFB & Accessories capital cost account, with the balance mostly consisting of the ASU capital cost.

Table 7: Capital cost and COE comparison between recompression and partial cooling cycle plants with and without reheat

Cost Account Description	RC760	RhtRC760	PCC760	RhtPCC760
Capital Costs (TPC, x$1000)				
Coal & Sorbent Handling	53,718	53,718	53,718	53,718
Coal & Sorbent Prep and Feed	28,782	28,782	28,782	28,782
Feedwater & Miscellaneous BOP	28,143	28,166	25,902	25,636
CFB & Accessories	1,153,474	1,129,007	1,095,709	1,042,365
Flue Gas Cleanup & Piping	21,871	21,871	20,965	20,965
CO_2 Removal & Compression	204,281	204,281	204,281	204,281
FG Recycle, Ductwork & Stack	30,105	30,105	30,105	30,105
sCO_2 Power Cycle	436,954	548,516	395,824	469,929
Cooling Water System	57,525	57,581	57,885	56,930
Ash & Spent Sorbent Handling	34,372	34,372	34,372	34,372
Accessory Electric Plant	110,851	110,833	110,143	110,210
Instrumentation & Control	32,241	32,241	32,166	32,161
Improvement to Site	19,053	19,200	18,868	18,899
Buildings & Structure	72,866	73,051	72,653	72,645
Total Plant Cost (TPC)	**2,284,236**	**2,371,724**	**2,181,374**	**2,200,999**
Total Plant Cost ($/kWe)	**4,059**	**4,244**	**3,938**	**3,831**
sCO_2 Power Cycle Cost Breakdown (TPC, x$1000)				
Main Compressor	58,274	58,307	43,362	42,974
Bypass Compressor	46,011	46,042	42,113	41,647
Boost Compressor	-	-	35,480	47,865
High Temperature Recuperator	88,306	161,122	48,358	82,294
Low Temperature Recuperator	78,912	79,040	53,733	52,183
Coolers (Including intercoolers)	30,489	30,535	46,971	45,762
Turbines	81,952	81,786	82,208	82,812
System Piping	46,291	84,978	37,047	67,814
System Foundations	6,718	6,706	6,552	6,579
COE Breakdown ($/MWh)				
Capital	82.69	86.40	80.23	78.04
Fixed O&M	17.51	18.22	17.08	16.60
Variable O&M	13.50	13.80	13.49	13.02
Fuel	25.97	26.14	26.38	25.43
COE (without T&S)	**139.67**	**144.56**	**137.18**	**133.09**
T&S	7.47	7.52	7.59	7.32
COE (with T&S)	**147.14**	**152.08**	**144.77**	**140.11**

Out of the four plants, RhtPCC760 case has the lowest COE followed by PC760, RC760 and RhtRC760. Although the RC760 case offers higher net plant efficiency, the COE of the PCC760 case is 2.5 $/MWh lower than the RC760 case. The COE without transportation and storage (w/o T&S) of RhtPCC760 case is 7.37 $/MWh higher than reference AUSC steam Rankine plant

(B24F) which has a COE w/o T&S of 125.72 $/MWh [8]. The higher COE for RhtPCC760 case is primarily due to significant increase in the CFB cost relative to the reference AUSC Rankine plant. The TPC of CFB & Accessories for RhtPCC760 case is $1,042.3M compared to $885.1M for reference AUSC steam Rankine plant [8], indicating that the turbine inlet temperature of 760 °C is likely uneconomical for sCO_2 power cycles with oxy-coal CFB heat sources.

From Tables 6 and 7, it is important to note that the capital cost of the CFB is 14-51% higher than that of the entire sCO_2 power block. This highlights the need to focus research and development (R&D) efforts on development of the CFB or other coal-fired heat sources.

The breakdown of sCO_2 power block costs show that the recuperators (HTR and LTR) are the dominant portion of the total power block cost for RC cases. The capital cost of recuperators is significantly lower for the PCC cases due to lower heat duties, as explained in the previous section. However, the boost compressor is a high volumetric flow machine (due to lower CO_2 density) and contributes significantly to the total power block cost. Likewise, the capital cost of coolers for partial cooling cycles is higher than the recompression cycles since they require pre-coolers and boost compressor intercoolers. Overall, the additional capital cost for compressors and coolers in the PCC cases is more than offset by the decrease in CFB and recuperator capital costs, thus the COE for PCC cases is lower than their RC counterparts.

It's worthwhile to note that the sCO_2 turbine capital costs in Table 7 are calculated using a shaft power-based cost scaling algorithm [16], thus these costs are similar for recompression and partial cooling cycles. The PCC turbine is expected to have more stages to accommodate larger pressure ratios relative to a RC turbine, while the reduced mass flow rate should yield a lower annular flow area through the turbine. The PCC turbine may therefore be longer and narrower than the RC turbine, though these impacts are not accounted for in the present study.

Sensitivity Analysis to CFB Tube Diameter

This section presents sensitivity analyses with the goal of understanding the impact of sCO_2 tube diameter in the CFB tube banks on sCO_2 pressure drop, plant efficiency, CFB cost, and COE. Figure 6 presents sCO_2 pressure drop and CFB cost as a function of tube outer diameter for the four cases. The tube diameter is assumed to be the same for all tube banks, with the main heat and reheat tube banks made of IN740H and economizer tube banks made of TP347 stainless steel. Detailed pressure drop and CFB cost breakdowns for a 2" tube OD are found in Table 6.

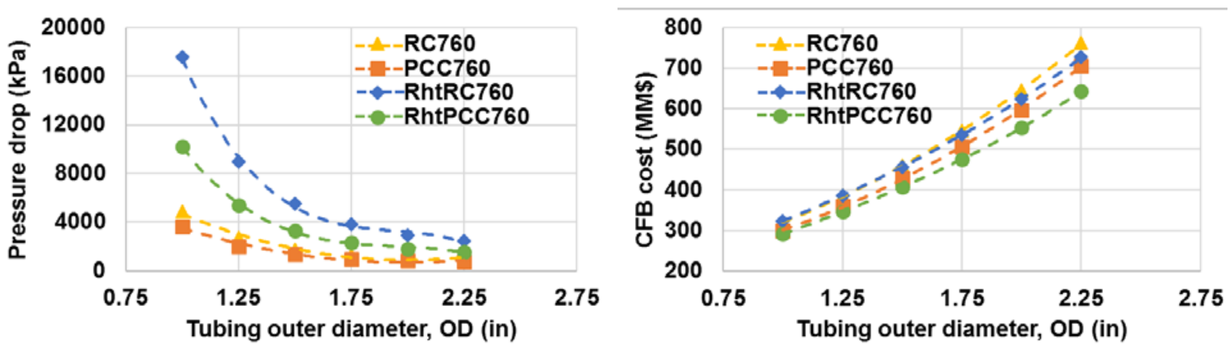

Figure 6: CFB sCO_2 pressure drop and CFB cost as a function of tubing outer diameter

For all cases, decreasing the tube outer diameter increases the sCO_2 pressure drop due to higher fluid velocities in the tube banks. On the contrary, smaller tubes require thinner walls which results in lower material and total CFB cost. Due to these opposing trends, one would expect to see an optimum tube diameter that minimizes the COE. Further, as the tube diameter decreases, sCO_2 pressure drop for cases with reheat (RhtRC760, RhtPCC760) increases significantly, due primarily to the significant increase in reheater pressure drop. Figure 7 shows the sCO_2 pressure drop vs CFB cost dependency for all four cases, as derived from Figure 6. Following observations can be made from Figure 7: 1) Designing the CFB to have low sCO_2 pressure drop can significantly increase the CFB cost. 2) For the same pressure drop, CFB cost is lowest for the PCC760 case followed by RC760, RhtPCC760, RhtRC760.

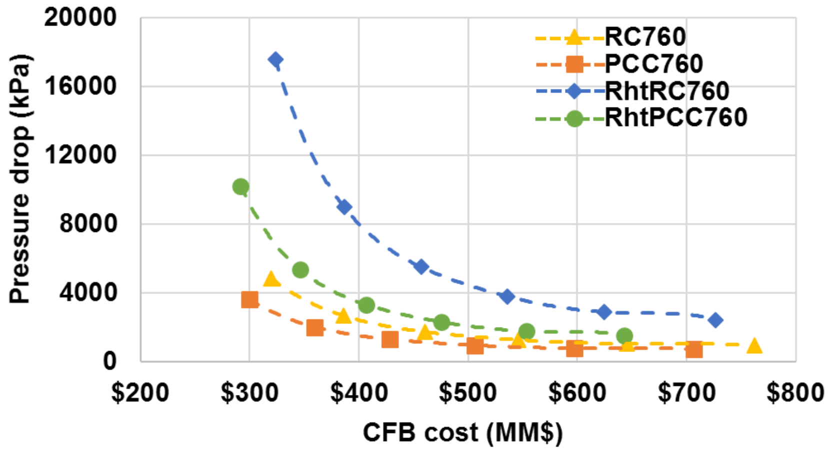

Figure 7: sCO_2 pressure drop vs CFB cost dependency

The sCO_2 pressure drop vs CFB cost dependency data presented in Figure 7 is used to understand the impact on net plant efficiency and COE in Figure 8. As expected, all the cycles are extremely sensitive to the sCO_2 pressure drop and plant efficiency decreases significantly with increasing sCO_2 pressure drop. It is interesting to note that the plant efficiencies of cases without reheat (RC760, PCC760) are less sensitive to sCO_2 pressure drop than their corresponding cases with reheat (RhtRC760, RhtPCC760). Out of the four cases, the RhtPCC760 case offers highest plant efficiency for wide range of sCO_2 pressure drops. Since coal flow rate is same for all the cases, the trends of net plant power output are similar to the net plant efficiency.

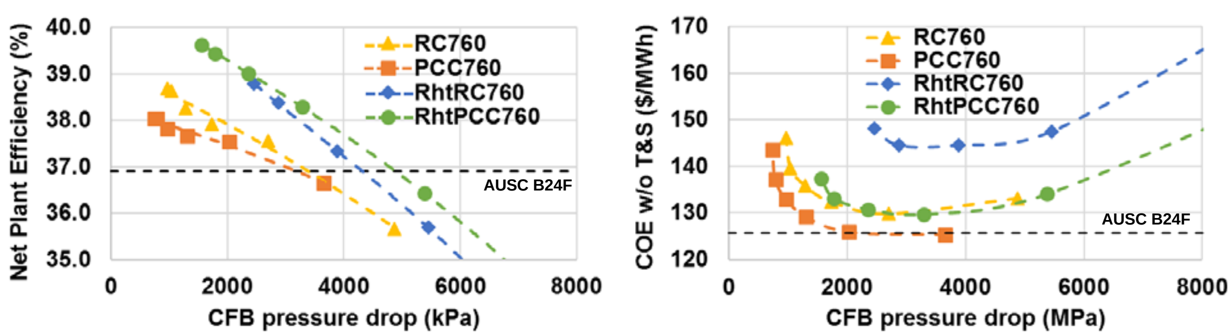

Figure 8: Net plant efficiency and COE w/o T&S as a function of sCO_2 pressure drop within CFB

Per Figure 7, the plant TPC decreases with an increase in CO_2 pressure drop due to a decrease in the CFB cost, thus each case attains an optimum COE with respect to pressure drop in Figure

8. An interesting point to note is that cases without reheat (RC760, PCC760) are able to achieve significant reduction in COE for less than 1% percentage points drop in the net plant efficiency. It is also notable that the RC case with reheat, while a favorite in the literature due to its high cycle efficiency, is the least economic of the cases studied, yielding a minimum COE that is at least 10% greater than COE minima for the other cycles studied. Overall, partial cooling cycles have lower COEs than their recompression cycle counterparts and appear to be better suited for coal-fired CFB heat sources.

Preliminary Sensitivity Analysis to Turbine Inlet Temperature

As mentioned in the previous sections, the turbine inlet temperature (TIT) of 760 °C might not be an economical design choice for sCO_2 power cycles with a coal-fired CFB as the heat source. A preliminary sensitivity analysis to the TIT is conducted for the PCC cases. Figure 9 shows the net plant efficiency and COE w/o T&S as a function of the TIT for PCC cases with and without reheat. The plant design conditions used for this analysis are same as those listed in Table 1, except for the TIT, which is varied. For each of the data points in Figure 9, the sCO_2 pressure drop within CFB and CFB capital cost are calculated using the simplified CFB design tool, assuming a tube diameter of 2.0" for all the tube banks and IN740H as the main heat and reheat tube material. Results for other tubing materials will be developed in future studies.

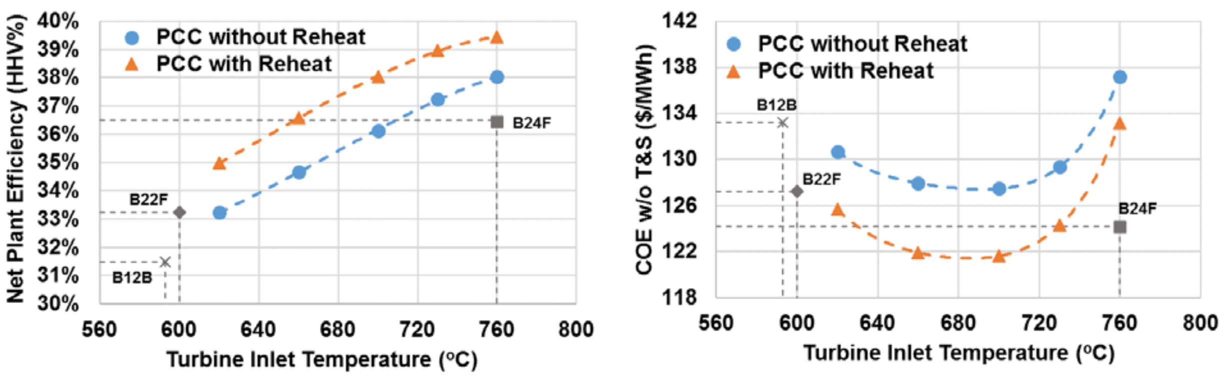

Figure 9: Net plant efficiency and COE w/o T&S as a function of turbine inlet temperature for partial cooling cycles with and without reheat

The plant efficiency is shown to increase almost linearly with the TIT. Increasing the TIT from 620 °C to 760 °C improves the net plant efficiency by almost 4.5 percentage points for the PCC with reheat. However, it is interesting to note that the COE w/o T&S for PCC with reheat at TIT of 620 °C is 7.5 $/MWh lower than at a TIT of 760 °C. Both of the partial cooling cycles exhibited a minimum COE for TIT around 700 °C if IN740H is the main CFB tubing material. At this temperature, the net plant efficiency of a PCC with reheat is 1.5 and 4.8 percentage points higher than reference oxy-CFB steam Rankine plants B24F and B22F respectively [8]. At the same TIT of 700 °C, the PCC plant with reheat has a COE w/o T&S that is 2.6 $/MWh and 5.6 $/MWh lower than the reference steam Rankine plants B24F and B22F, respectively [8].

Another important point to note is that the PCC with reheat has higher plant efficiency and lower COE than PCC without reheat at all turbine inlet temperatures. However, based on results from the previous section, there could be further room for improvement if CO_2 tube diameter vs. pressure drop and CFB cost tradeoffs are considered. Moreover, using this tool, the COE could be further reduced by considering tubing materials and the turbine inlet pressure as optimization variables.

CONCLUSIONS & FUTURE WORK

This paper presents the techno-economic analysis results of partial cooling and recompression Brayton cycles with and without reheat using an oxy-coal CFB as the heat source. A simplified CFB design tool was developed to improve the accuracy of CFB sCO_2 pressure drop and capital cost estimates, as well as to capture the impact of turbine inlet temperature & pressure, CFB sCO_2 pressure drop, choice of reheat/non-reheat, etc., on the CFB capital cost.

The simplified CFB design tool was calibrated/validated using data from the STEAMPRO software package. Using this design tool, sCO_2 pressure drop and cost estimates are developed for partial cooling and recompression cycles with and without reheat. The results show that the CO_2 pressure drop is significantly lower for the partial cooling cycles relative to their recompression counterparts due to lower cycle flow rate. Likewise, due to larger driving forces, the CFB cost is significantly lower for partial cooling cycles. Also, using reheat is shown to increase CO_2 pressure drop, but lower the CFB cost, for both partial cooling and recompression cycles.

Though development of the CFB design tool is certainly a step in the right direction, the model made several simplifying assumptions which introduce significant uncertainty in CO_2 pressure drop and/or CFB cost estimates. For example, cost of interconnecting piping is assumed to be a constant value derived from STEAMPRO, which introduces uncertainty in CFB cost estimates for other design conditions. Also, no attention was paid to the arrangement of the tube banks within the CFB. Depending on the dimensions of the CFB furnace section, the arrangement of tube banks might have to be reconsidered to achieve feasible and acceptable designs. In spite of these deficiencies, such a design tool is valuable in understanding the impact of design variables on CFB, overall plant efficiency and COE.

For a given CFB tube diameter, the performance and economic analysis shows that the partial cooling cycle with reheat (RhtPCC760) has the highest plant efficiency and lowest COE out of the four cases investigated. Partial cooling cycles have significantly lower mass flow rate, lower recuperator costs and CFB costs than the recompression cycles. However, in order to accommodate larger cycle pressure ratios, partial cooling cycles require additional coolers and a boost compressor which contribute to the total capital cost, though these are generally offset by lower recuperator and CFB costs. Considering all these capital cost impacts, the COE of partial cooling cycles is lower than their recompression cycle counterparts in this application.

Given the large contribution of the CFB to the overall plant sCO2 pressure drop, efficiency, cost and COE, a sensitivity analysis with respect to the CFB tubing diameter was conducted to explore potential improvements. The partial cooling cycle without reheat (PCC760) offers the lowest COE when CFB pressure drop vs. cost impacts are considered using smaller tubing diameters. However, given the large uncertainty in the developed CFB model, a more detailed CFB design study should be undertaken to validate these conclusions. Nevertheless, these results highlight the importance of focusing R&D efforts on development of the coal-fired CFB and possibly other coal-fired heat sources. Due to the difference in operating conditions between steam Rankine cycles and indirect sCO_2 power cycles, traditional heat source design methods used for steam Rankine plants may need to be reconsidered for the sCO_2 power cycles.

Finally, a preliminary sensitivity analysis was conducted with respect to the turbine inlet temperature for partial cooling cycles, where the optimum turbine inlet temperature that minimizes plant COE appears to be around 700 °C, without considering CFB tubing material changes. At this turbine inlet temperature, the net plant efficiency of a partial cooling cycle with reheat is 1.5 percentage points higher, and the COE is 2% lower, than a reference oxy-CFB steam Rankine plant (B24F) with a turbine inlet temperature of 760 °C. There is further room for COE reduction by considering CFB pressure drop vs cost tradeoffs and by considering turbine inlet pressure and tubing material as optimization design variables. These impacts are currently being evaluated under NETL's on-going projects.

ACKNOWLEDGMENTS

The authors would like to thank Richard Dennis (NETL) for his project support. The authors also wish to thank Wally Shelton, Travis Shutlz (NETL), Mark Woods (Keylogic) for their technical assistance and input. The work was completed under DOE NETL Contract Number DE-FE000401.

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