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Performance and Economic Evaluation of sCO_2 Bottoming Cycles for Natural Gas Combined Cycle Plants with Capture

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ABSTRACT

Natural gas combined cycles (NGCCs) with carbon capture are expected to play a significant role in the decarbonization of the power generation sector. NGCC plants generally use triple-pressure reheat steam Rankine power cycles for the bottoming cycle. Some studies in the literature have investigated the application of recompression and cascade-style supercritical CO_2 (sCO_2) cycles for NGCC bottoming cycle applications, but these studies have focused on power plants without carbon capture. However, NGCC plants fitted with post-combustion solvent-based CO_2 capture systems will require a significant amount of steam for solvent regeneration, and this can have a major impact on the optimal sCO_2 bottoming cycle design. This study investigates the performance and economic potential of sCO_2 bottoming cycles for H-class gas turbine-based NGCC plants with a post-combustion capture system. A portion of the gas turbine exhaust heat is used for the generation of steam required for a solvent-based capture system while the rest of the waste heat is utilized in an sCO_2 bottoming cycle for power generation. Overall, the performance and LCOE of investigated sCO_2 bottoming cycles are similar to those of a state-of-the-art triple-pressure reheat steam Rankine cycle. As the gas turbine exhaust temperature increases (beyond 630°C), sCO_2 bottoming cycles begin to show greater performance and economic benefits compared to a steam Rankine cycle with ~0.7 percentage point higher plant efficiency and 1.4% lower LCOE.

INTRODUCTION

Indirect supercritical CO_2 (sCO_2) power cycles are of interest for a wide range of applications such as nuclear, concentrated solar, fossil, or biomass-based power generation. In general, sCO_2 power cycles are shown to have higher efficiency than steam Rankine cycles operating under comparable conditions, and sCO_2 cycle configurations such as recompression Brayton, partial cooling cycles have been identified that lead to best performance and reduced cost of electricity (COE) compared to steam plants. [1–4] Another area of application that is garnering commercial interest is the use of sCO_2 power cycles for the conversion of waste heat to power. Echogen is a small business that has developed a 7-8 MWe sCO_2 bottoming cycle for use in generating power from industrial waste heat sources or the exhaust streams of small simple cycle gas turbines. Peregrine Turbine Technologies also has small-scale sCO_2 bottoming cycle systems under development that will be commercially available soon, and General Electric (GE) has similarly begun process and hardware development to target small-scale waste heat recovery systems. For waste heat recovery systems, the sCO_2 power cycle employed is different from those used for primary cycle power generation. Recompression Brayton and partial cooling cycles are best suited to heat sources that provide a narrow temperature window for heat addition such as those for nuclear or concentrated solar power generation. In contrast, cascade

sCO_2 cycles are proposed in the literature for waste heat recovery because they maximize the amount of heat transferred from flue gas or some other hot waste stream, albeit at potentially lower power conversion efficiency. [5–8] An economic analysis performed by Echogen has shown that sCO_2 bottoming cycles are economically viable in small-scale applications (around 8 MW) with a lower COE than steam. [6] The National Energy Technology Laboratory (NETL) conducted a techno-economic analysis of sCO_2 power cycles for utility-scale F-class gas turbine NGCC plants. The study showed that the cascade cycles proposed in literature achieve efficiency and COE parity with a reference NGCC plant with a triple-pressure reheat steam bottoming cycle. [8] Most of the studies in the literature are focused on the application of sCO_2 power cycles for utility-scale NGCC plants or small-scale gas turbines without carbon capture. However, to achieve net zero targets by 2050, NGCC plants with carbon capture will likely play a more significant role in the decarbonization of the power sector. Amongst the available carbon capture technologies, a post-combustion solvent-based CO_2 capture system (such as Shell Cansolv) has the highest Technology Readiness Level and is likely to be deployed as a retrofit or in future capacity additions by electric utilities. Solvent-based CO_2 capture technology requires a significant amount of steam for solvent regeneration, and this steam is typically generated within the heat recovery steam generator (HRSG) of NGCC plants, which impacts the heat recovery to the bottoming cycle. Therefore, the application of sCO_2 power cycles for NGCC plants with capture will likely have an impact on the design of the bottoming cycle (optimal configuration, design variables, etc.). In light of the above discussion, the primary objective of this study is to investigate the performance and economic potential of sCO_2 bottoming cycles for utility-scale NGCC plants with capture, which is not reported in the open literature to the author's best knowledge.

Reference NGCC Plant

The reference NGCC plant under investigation in this study is based on Case B32B.95 from the NETL Rev4a baseline study. [9] The block flow diagram (BFD) of the reference NGCC plant is shown in Figure 1.

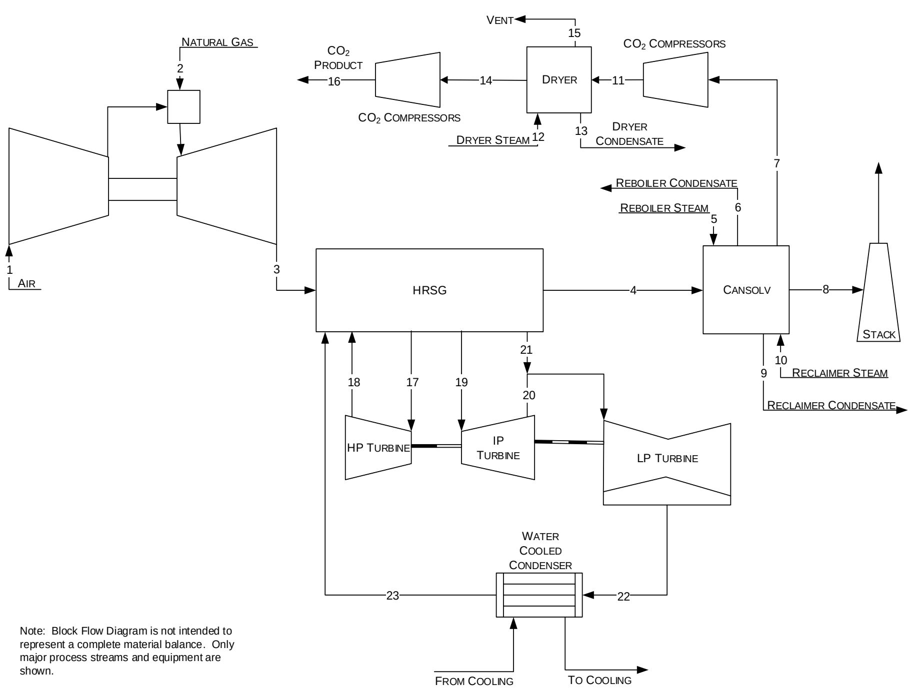


Figure 1. Reference natural gas combined cycle (NGCC) plant with triple-pressure reheat steam bottoming cycle and post-combustion solvent-based capture system.

The reference NGCC plant consists of two state-of-the-art H-class gas turbines, two HRSGs, and one steam turbine in a multi-shaft 2x2x1 configuration. Each gas turbine has a power rating of 343 MW when firing natural gas under ISO conditions with a gas turbine exhaust temperature of 596°C. The HRSG is configured with HP, IP, LP steam drums, and superheaters, reheater, evaporator, and economizer sections. The HP drum is supplied with feedwater by the HP feed pump to generate HP steam, which passes to the superheater section for heating to 585°C (HP steam pressure = 18.4 MPa). The IP drum is supplied with feedwater by an interstage bleed from the HP feed pump. The IP steam is mixed with the HP turbine exhaust before being reheated to 562°C (IP steam pressure = 3.2 MPa). The combined flows are admitted into the IP section of the steam turbine. The LP drum provides steam to the LP turbine. A carbon dioxide recovery (CDR) facility is used to remove CO_2 in the flue gas exiting the HRSG, purify it, and compress it to desired pipeline conditions. It is assumed that all the carbon in the natural gas is converted to CO_2 and 95% of the CO_2 is removed and captured in the CDR facility. The CO_2 absorption/stripping/solvent reclamation process is based on the CANSOLV system. The reboiler requires low-pressure steam for solvent regeneration and the steam requirement is approximately 2.4 MJ/kg CO_2. The rest of the modeling details and detailed stream tables can be found in the NETL report. [9]

sCO_2 Bottoming Cycle Descriptions

The steam bottoming cycle in the reference NGCC plant described above is replaced with sCO_2-based bottoming cycles. Three different sCO_2 bottoming cycle options are considered in this study. Figure 2 shows the BFD of a modified sCO_2 Brayton cycle with a low-temperature (LT) economizer. In this configuration, the gas turbine exhaust (F1) heats the high-pressure sCO_2 (stream 6) to the turbine inlet temperature within the primary heat exchanger (PHX). The flue gas (F2) then generates the necessary LP steam for the CANSOLV system within the LP steam generator. The rest of the heat recovery from flue gas occurs within the Economizer, which operates in parallel to the low-temperature recuperator (LTR). The low-pressure hot sCO_2 turbine exhaust (stream 8) preheats the high-pressure cold stream from the compressors in the recuperators (LTR and HTR). Following the recuperators, the turbine exhaust stream (stream 12) is cooled to the desired temperature followed by compression to cycle maximum pressure.

Figure 3 shows the BFD of a modified sCO_2 Brayton cycle with LT- and HT-Economizer. The high-temperature (HT) economizer operates in parallel to the high-temperature recuperator (HTR). Adding a second economizer (HT-Economizer) offers a greater degree of control over the heat recovery process from the flue gas. This version of the sCO_2 power cycle allows for a portion of the heat duty to be shifted from the PHX to the HT-Economizer, thereby allowing for independent heat recovery from gas turbine exhaust (LT- and HT- Economizers) and sCO_2 turbine exhaust (LTR and HTR).

Figure 4 presents the BFD of the Cascade cycle that has been proposed and reported in the literature for waste heat recovery applications. [5 – 8] The cascade cycle is similar to the modified Brayton cycle with LT-Economizer and HT-Economizer with the addition of an LT turbine that bypasses the PHX. Adjusting the flow to LT-turbine and HT-turbine independently enables maximum control of the heat recovery process from the gas turbine and sCO_2 turbine exhaust streams.

A common feature among all the sCO_2 cycle configurations considered in this study is the exclusion of a bypass compressor used in the recompression Brayton cycle proposed in the literature. [1 – 3] In the recompression Brayton cycle, a bypass compressor is needed to overcome the specific heat mismatch between high- and low-pressure streams in the LTR. However, for waste heat recovery applications, the high-pressure stream exiting the compressor is split into two streams for heat recovery from flue gas, which negates the need for a bypass compressor.

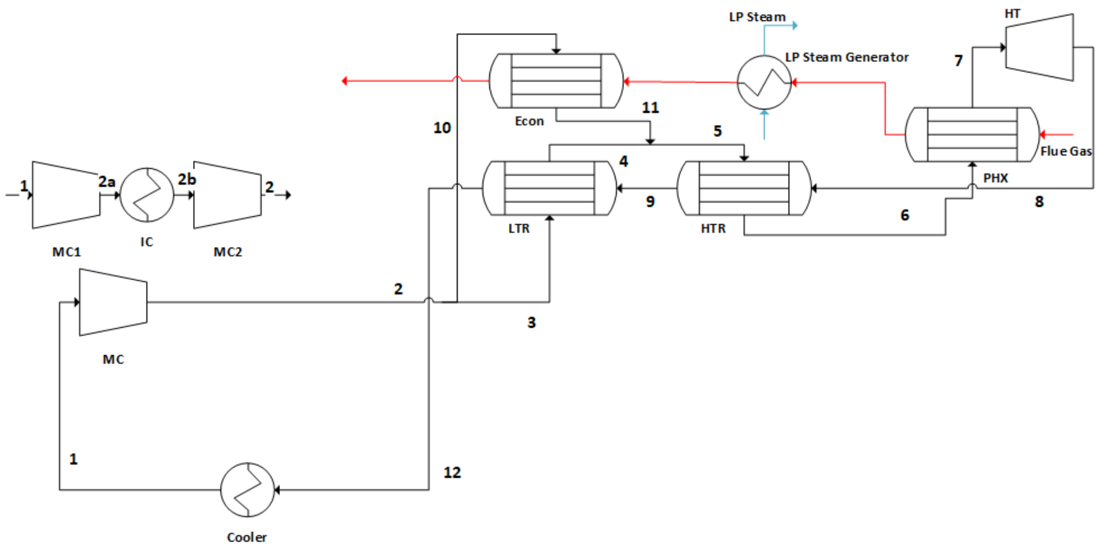


Figure 2. Modified sCO_2 Brayton Cycle with LT economizer

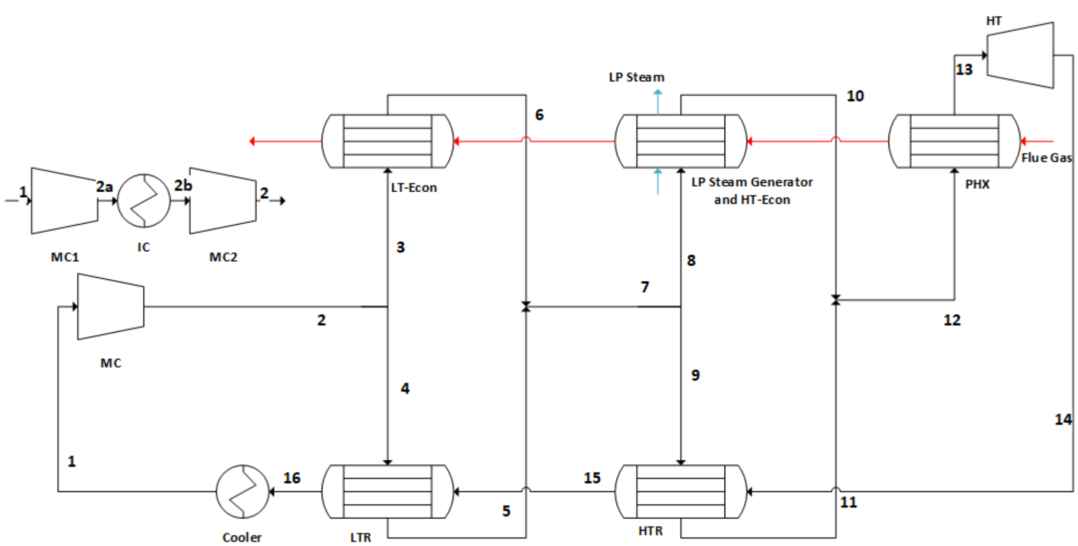


Figure 3. Modified sCO_2 Brayton Cycle with low LT and HT economizers

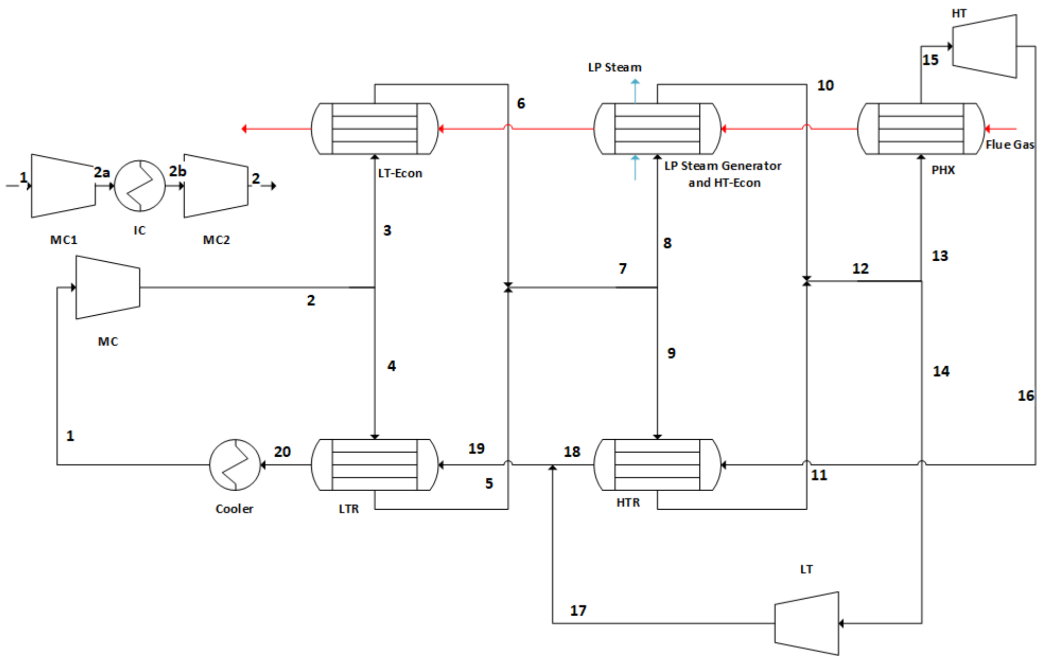


Figure 4. Cascade sCO_2 cycle

MODELING APPROACH

The design bases from NETL's Baseline study and Quality Guidelines for Energy System Studies (QGESS) series [10] were adopted so that the results from this study would be consistent and comparable to the reference NGCC plant. All the plants are assumed to be located at a generic plant site in the midwestern United States at sea level with an ambient dry bulb temperature of 15°C and 60% relative humidity. All the plants are assumed to have an 85% capacity factor, and the natural gas properties used in this study are taken from the 2019 revision of the NETL QGESS document, "Specification for Selected Feedstocks."

Performance Modeling Methodology

The thermodynamic performance of all the plants is based on the output from a steady-state model developed using the Aspen Plus® (Aspen) modeling program. In addition to the Aspen model, sub-system models for recuperators and coolers are used for estimating their capital costs as described in subsequent sections. Accurate modeling of sCO_2 power cycles requires appropriate calculations of the CO_2 thermophysical properties, particularly near the critical point (31°C, 7.37 MPa). The Span-Wagner equation of state (EOS) is the most accurate property method for processes containing pure CO_2. [11, 12] The power cycle working fluid is assumed to be pure CO_2, and Span-Wagner EOS is used for all the property estimates within the sCO_2 power cycle. Leakage and make-up flows are not modeled. For the flue gas components of the plant, Peng-Robinson EOS is used.

sCO_2 Power Cycle

Table 1 summarizes the sCO_2 power cycle design conditions used for all the cases in this study. The main compressor is a two-stage compressor with one stage of intercooling. The pressure ratio for all the stages is assumed to be equal. The CO_2 outlet temperatures for all the coolers and intercoolers are assumed to be the same. The split flow between recuperators (LTR and HTR) and economizers (LT-Economizer and HT-Economizer) is calculated to meet the design temperature approach specifications. For the Cascade sCO_2 cycle, the split flow between LT- and HT-turbines sets the PHX temperature approach in addition to other temperature approach specifications for recuperators and economizers.

Table 1. sCO_2 power cycle design conditions

Section	Parameter	Value
Turbines	Isentropic efficiency	92.7%
Main compressor	Stages	2
	Intercooling stages	1
	Isentropic efficiency	85%

Recuperators

The sCO_2 cycle recuperators (LTR and HTR) are envisioned to be compact diffusion-bonded heat exchangers, commercially known as printed circuit heat exchangers (PCHEs). A 1-D PCHE model is developed in Aspen Custom Modeler platform for the design of the cycle recuperators (HTR and LTR). The following assumptions are made for the model [13]:

- Fully developed turbulent flow
- Ideal counter-current flow without entrance and exit effects
- Negligible heat loss and axial dispersion effects
- 1D temperature distribution along the channel axis
- Uniform temperature, pressure, and velocity at the entrance of each channel

As shown in Figure 5(a), the hot and cold plates with etched channels in PCHE are arranged alternately and assembled by diffusion bonding. For this study, the cold and hot fluids flow mainly counter-current. The geometry parameters related to plate arrangement are the number of plates (N_p), the number of channels per plate (N_c), and the ratio between the number of hot and cold plates (R_p). The core dimension is characterized by L_x x L_y x L_z. For the present study, R_p is set to 2 for uniform distribution of pressure drop on hot and cold sides. To capture the sharp variation in thermo-physical properties near the critical point, the number of nodes along the z direction is set to 50.

As shown in Figure 5(b), the cross-section of the etched channels is mostly semi-circular with a channel width (D_c) varying from 0.2 mm to 5 mm. The wall thickness (t_2) and the ridge width (t_3) are determined based on the operating conditions, channel width, design stress, and corrosion allowance of the selected material using the ASME 13-9 code. [14]

As shown in Figure 5(c), a zigzag pattern is commonly used in the design of PCHEs. In addition to D_c, wave angle (α) and length-to-width ratio(L_w/D_c) are two other important channel design parameters. In this study, a high-angle channel design is adopted using thermal-hydraulic correlations developed based on experimental data available in the open literature. [14] The actual channel length (L_c) is correlated to the length of the PCHE core (L_z) as follows:

$$L_z = L_c \cos(\alpha)$$

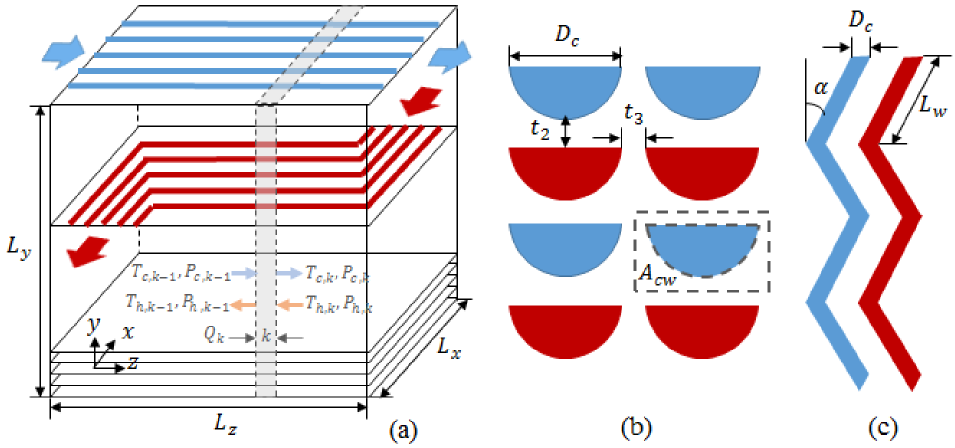


Figure 5. Geometry of PCHEs: (a) plate arrangement, (b) cross-sectional view, (c) channel design

Further details of the PCHE model and validation can be found in Jiang et al. [13]
Table 2 summarizes the design assumption used for modeling the sCO_2 cycle recuperators.

Table 2. sCO_2 power cycle recuperator design assumptions

Parameter	Value
Channel shape	Zigzag semi-circle
Channel design	High-angle channel from Heatric [14]
Channel width, D_c (mm)	2
Number of hot plates per cold plate, R_p	2
Number of discrete points along the length	50

Coolers and Intercoolers

The sCO_2 power cycle coolers and intercoolers are made up of modular adiabatic cooler bays. Adiabatic coolers are used in the CO_2 refrigeration industry to enhance the performance of CO_2 coolers during hot ambient conditions. The schematic of an adiabatic cooler bay is shown in Figure 6.

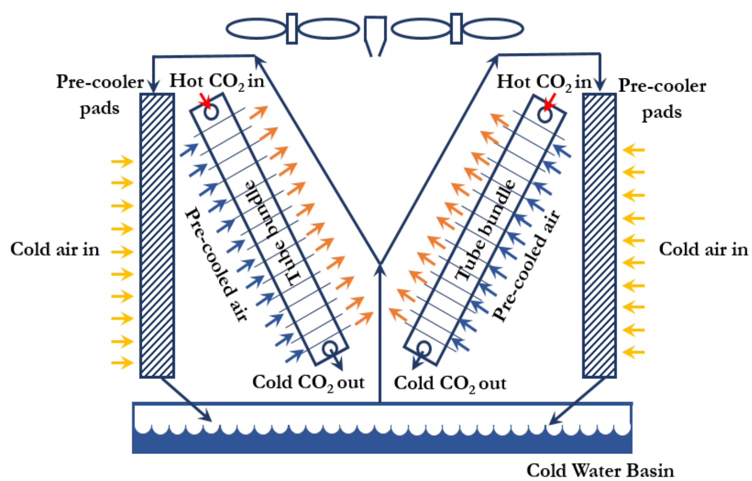
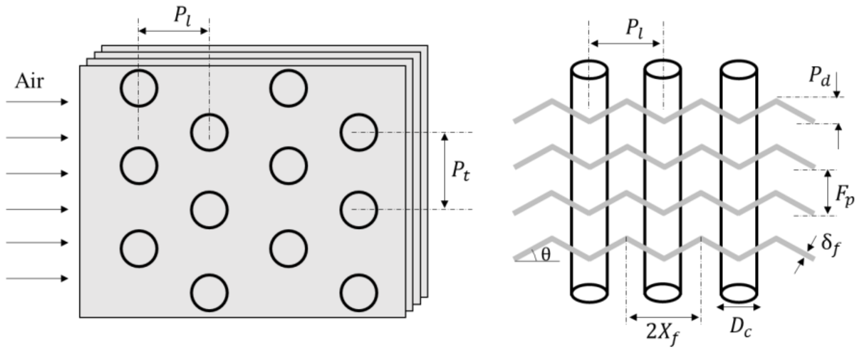


Figure 6. Schematic of adiabatic cooler bay

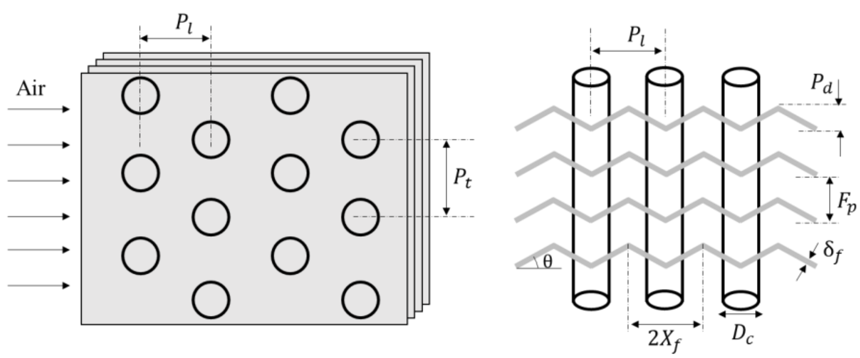
As depicted in the schematic, CO_2 flows through the finned tube heat exchanger bundles with multiple rows of tubes and multiple passes. The induced draft fans located at the top of the cooler draw cold air over the tube bundles in a crossflow arrangement to the CO_2 flow. Pre-cooler pads are installed prior to the tube bundles. These pre-cooler pads are wetted with water and as the air is drawn over the wet cooling pads, water evaporation humidifies the air, cooling it to approach the ambient wet bulb temperature. To meet the required cooling duty several such bays are employed. The geometrical parameters of the modeled tube bundles are provided in



along with the associated nomenclature.

Table 3. Geometric dimensions of modeled plate-fin-and-tube heat exchanger

Parameter	Value
Tube outer diameter, D_o(mm)	12
Tube wall thickness, t_w (mm)	0.7
Finned tube length, L (m)	11.385
Tube arrangement pattern	Staggered
Fin thickness, δ_f (mm)	0.15
Tube collar diameter, $D_c = D_o + 2\delta_f$ (mm)	12.3
Longitudinal fin pitch, P_l (mm)	25
Transverse fin pitch, P_t (mm)	50
Fin pitch, F_p (mm)	2
Fin corrugation angle, θ (degrees)	8
Number of tube bundles, $N_{bundles}$	2
Number of tubes per row, N_{tubes}	64
Number of tube passes, N_{passes}	3
Number of tube rows per pass, N_{rows}	2



An Excel-based performance model of an adiabatic cooler bay is developed. The heat exchanger tube bundles are discretized into multiple sub-sections to account for the non-linear variation in thermo-physical properties of CO_2 near the critical point. The model was validated to the data provided by the vendor. The adjustable model inputs include CO_2 operating conditions (Inlet pressure and temperature, flow rate, outlet temperature, and pressure drop); ambient air dry and wet bulb temperatures; and the number of discretization points (N) along the tube bundle length.

N is assumed to be 10 for all the cases. The model iteratively calculates the number of bays (N_{bays}), total auxiliary fan power consumption, and total water consumption rate to meet the desired operating conditions. Further modeling details, CO_2-side and air-side heat transfer, and pressure drop correlations can be found in Pidaparti et al. [15, 16] The balance of plant (BOP) cooling duty is met using a mechanical draft cooling tower.

Economic Analysis Methodology

Plant capital costs in this study are estimated according to NETL's QGESS document. [17] The capital costs are defined at two levels: bare erected cost (BEC) and total plant cost (TPC), which are overnight costs expressed in 2018 base-year dollars. Process and project contingencies are included in cost estimates to account for unknown costs that are omitted or unforeseen due to a lack of complete project definition and engineering. Process contingencies compensate for uncertainty in cost estimates caused by performance uncertainties associated with the development status of a technology. Lower process contingency costs are used for sCO_2-specific components, which is more reflective of a NOAK cost estimate.

Operation and maintenance (O&M) costs are divided into two categories: fixed O&M costs that are independent of plant operation hours (e.g., labor, overhead, etc.), and variable O&M costs that are proportional to the power generation (e.g., consumables, waste disposal, maintenance materials). The variable O&M and fuel costs are multiplied by an assumed capacity factor of 85% to arrive at the actual annual expenditure. The captured CO_2 transportation and storage (T&S) costs are estimated at $10/tonne. [18] and the natural gas fuel cost is assumed to be $4.42/MMBtu. [19]

The Levelized COE (LCOE) is reported on a $/MWh basis and consists of contributions from the O&M costs (fixed, variable, and fuel), CO_2 T&S costs, and the annualized capital cost over the assumed 30-year lifetime of the plant. Additional details on the cost estimating methodology and other economic assumptions are provided in Ref [9].

Aside from the sCO_2 power cycle components, the BOP unit operations and equipment are analogous to those found in the reference NGCC plant [9], and these costs are scaled using a consistent methodology used in all NETL studies. The cost estimates for these BOP 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. [9] All the capital costs are escalated to 2018 dollars using the Chemical Engineering Plant Cost Index.

sCO_2 Power Cycle Components Cost Estimates

All the sCO_2 power cycle component costs follow a general power law form:

$$C = aSP^b \times f_T$$

where SP is the scaling parameter, a and b are the scaling coefficients, and f_T is a temperature correction factor of the following form:

$$f_T = \begin{cases} 1 & if\ T_{max} < T_{bp} \\ 1 + c(T_{max} - T_{bp}) + d(T_{max} - T_{bp})^2 \end{cases}$$

where T_{bp} is the temperature breakpoint of 550°C and T_{max} is the maximum temperature rating of the component. The scaling parameters and coefficients for all the sCO_2 power cycle components are listed in **Error! Reference source not found.**. Except for recuperators and coolers, these values are taken from Weiland et al. [20] For the recuperators, Weiland et al. [20] developed cost correlation based on overall conductance (UA). However, this cost correlation cannot capture all the impacts of design variables considered in this study such as maximum pressure, recuperator pressure drops, etc. So, a cost correlation using recuperator mass (M_{recup}) as the scaling parameter was developed using the same vendor quotes database. M_{recup} for these quotes is calculated using the recuperator model described above. The sCO_2 compressors are assumed to be barrel-type centrifugal compressors, and the costs are scaled with respect to the inlet volumetric flow rate ($\dot{V}_{in}$). All the sCO_2 turbines are assumed to be axial turbines, and the costs are scaled with respect to shaft power ($\dot{W}_{sh}$). The equipment costs of coolers and intercoolers are scaled linearly with the number of calculated adiabatic cooler bays (N_{bays}) calculated by the cooler model described above. The coefficient, a, for the cooler represents the cost per bay quoted by the vendor.

Table 4. Cost scaling parameters and coefficients for the sCO_2 power cycle components

Component	Scaling Parameter (Units)	Coefficients				Uncertainty range
		a	b	c	d	
Recuperators	M_{recup} (kg)	1,371	0.78	0	0	-28% to +33%
sCO_2 turbines	$\dot{W}_{sh}$ (MW_e)	182,600	0.56	0	1.106e-4	-25% to +30%
sCO_2 compressors	$\dot{V}_{in}$ (m^3/s)	6,220,000	0.11	0	0	-30% to +50%
Generator	$\dot{W}_e$ (MW_e)	108,900	0.55	0	0	-19% to +23%
Compressor motor	$\dot{W}_e$ (MW_e)	399,400	0.61	0	0	-15% to +20%
Coolers and Intercoolers	N_{bays}	141,387	1.00	0	0	-25% to +28%

HRSG/Primary Heater Cost Estimate

For the primary heater (or HRSG), a cost correlation based on UA was developed using data generated from GT-PRO software. The correlation is shown in Figure 7. The cost correlation is also compared with two sCO_2-specific vendor quotes and is within ±20% of the vendor quotes. Note that the GTPRO software uses different materials of construction than the vendor-provided data. The GT-PRO software will only use T22 and T91 materials regardless of the UA values or temperatures provided. This cost correlation is used for the economizers (LT- and HT-Economizer) and PHX. The UA of these heat exchangers is calculated using multi-stream heat exchanger blocks in Aspen and combined UA is used in the cost correlation.

OPTIMIZATION APPROACH

Table 5 shows the selected optimization design variables and the associated minimum/maximum limits. All these design variables impact both plant efficiency and LCOE to varying degrees. There is usually a trade-off between plant efficiency and LCOE with respect to all the design variables. The lower and upper limits are established based on prior studies and tuning during the optimization process. The number of design variables available for optimization is different for each sCO_2 power cycle configuration. For example, modified Brayton cycle with LT-Economizer does not have an HT-economizer or an LT-turbine, so the design variables $Q_{HT-Econ}$ and X_{LT} are not applicable for the modified Brayton cycle with LT-Economizer, whereas all the design variables are available for optimization for the Cascade cycle configuration.

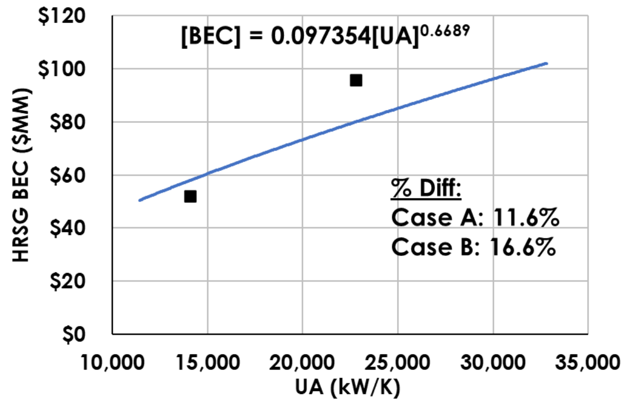


Figure 7. HRSG/primary heater cost correlation developed using GT-PRO

Table 5. Optimization design variables and associated lower/upper limits.

Design Variable	Lower limit	Upper limit
sCO_2 cooler outlet temperature, T_{Cooler} (°C)	20.0	35.0
Primary turbine inlet temperature, TIT (°C)	520.0	595.0
Cycle maximum pressure, P_{Max} (MPa)	20.0	35.0
Turbine outlet pressure, $P_{Turb,out}$ (MPa)	3.45	6.55
HTR cold end approach, $T_{App,HTR}$ (°C)	5.6	65.0
PHX approach temperature, $T_{App,PHX}$ (°C)	1.0	20.0
LTR cold end approach, $T_{App,LTR}$ (°C)	5.6	65.0
LT Economizer cold end approach, $T_{App,LT-Econ}$ (°C)	1.0	20.0
HT Economizer heat duty, $Q_{HT-Econ}$ (MW)	100.0	250.0
PHX heat duty, Q_{PHX} (MW)	250.0	500.0
HTR total pressure drop, ΔP_{HTR} (kPa)	68.9	344.7
LTR total pressure drop, ΔP_{LTR} (kPa)	68.9	344.7
Main Cooler pressure drop, ΔP_{MC} (kPa)	1.7	137.9
Compressor intercooler pressure drop, ΔP_{MCIC} (kPa)	34.5	206.8
Flow split fraction to LT turbine, X_{LT}	10%	25%

Prior NETL analysis investigated the relationship among plant efficiency, compressor inlet pressure, and sCO_2 cooler outlet temperature (T_{cooler}). For given T_{cooler}, the optimum compressor inlet pressure ($CIP_{optimum}$) is slightly higher than saturation pressure or pseudo-critical pressure to take advantage of high fluid density during compression. [16] Therefore, the compressor inlet pressure is set using the following correlation for $CIP_{optimum}$ vs. T_{cooler} derived using data from prior analysis [16]:

$$CIP_{optimum} = 0.08953\, T_{cooler}^2 - 1.15314\, T_{cooler} + 512.75589$$

Using this approach for setting the main compressor inlet pressure resulted in a significant reduction in optimization run time. The compressor inlet pressure is set during optimization by setting the turbine outlet pressure ($P_{Turb,out}$) equal to $CIP_{optimum}$ plus the sum of CO_2 pressure drops across the hot side of HTR, LTR, and main cooler.

Optimization Platform

For this study, the NETL in-house developed Framework for Optimization and Quantification of Uncertainty and Sensitivity (FOQUS) software is used for optimization. FOQUS allows for the integration of commonly used chemical engineering process modeling software like Aspen Plus, Aspen Custom Modeler, Excel, etc. [21]. A FOQUS model is developed to integrate the plant Aspen model, sub-systems models for recuperators, and coolers for simulation-based optimization. The resulting FOQUS model flowsheet and associated data transfers are shown in Figure 8. The “PCHE_Recuperators” node calculates HTR and LTR mass. The “CoolersModel” node calculates the auxiliary fan power consumption, number of cooler bays, and water consumption rate. The node includes calculations for all the cycle coolers and intercoolers. Finally, the “ExcelTemplate” node calculates the plant efficiency and LCOE.

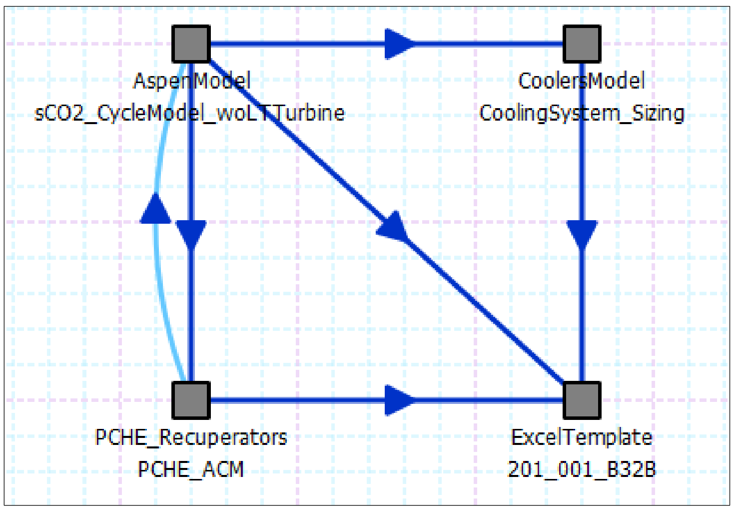


Figure 8. FOQUS flowsheet with integrated models

Several derivative-free optimization (DFO) solvers are available under the FOQUS platform that can be selected for simulation-based multi-variable design optimization. [21] One of the challenges associated with the optimization of sCO_2 power cycles is dealing with pinch-point issues in the recuperators. Depending on the cycle operating conditions, temperature cross-overs could occur within the recuperators, and the solution is considered as an infeasible design. Therefore, any selected optimization solver should be able to navigate search spaces, which results in infeasible designs to find the global optimum. One such DFO optimization solver available under the FOQUS platform is the Covariance Matrix Adaption Evolution Strategy (CMA-ES) solver [22], which is selected for the optimization in the current study. CMA-ES belongs to the class of evolutionary algorithms. In each iteration, new candidate designs are generated by variation of the current design variables, usually in a stochastic way. After each iteration, some candidate designs are selected to become the parents in the next iteration based on their objective function value. Thus, over the iteration sequence, candidate designs with progressively better objective function values are generated.

RESULTS AND DISCUSSION

Sample Optimization Results

Figure 9 presents the LCOE vs. plant efficiency of all the data points generated during optimization for the modified Brayton cycle with LT-Economizer. The objective function for optimization is to minimize the LCOE without any additional constraints on the design variables or plant efficiency. For this particular case, nearly 1,000 samples were generated by the optimization solver. As seen in Figure 9, plant efficiencies (HHV basis) of >49.0% are possible; however, these design solutions do not necessarily lead to the lowest possible LCOE. For plant efficiencies >49.5%, the LCOE starts to increase exponentially due to increased capital costs (CAPEX) of sCO_2 power cycle components.

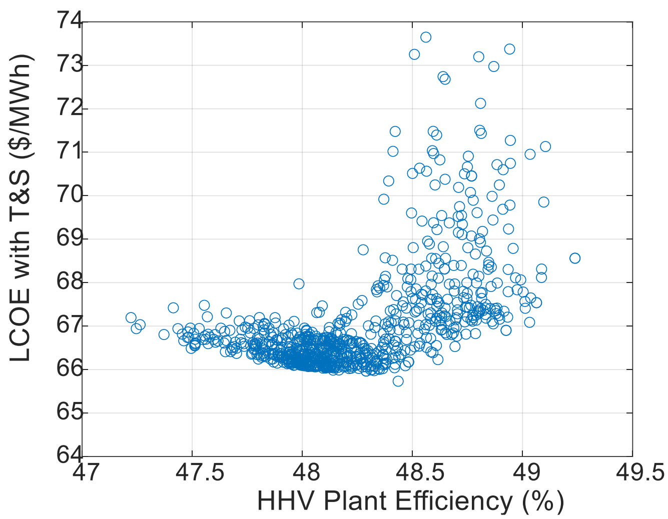


Figure 9. LCOE vs. Plant Efficiency for Modified Brayton cycle with LT-Economizer; samples generated by CMA-ES algorithm under FOQUS optimization platform.

Table 6 presents the optimized design variables for all the optimized sCO_2 bottoming cycle options investigated in this study. For the selected ambient design conditions (Midwest ISO conditions), T_{cooler} for all the plants are in the range of 18 – 22°C and the turbine inlet temperatures are in the range of 535 – 590°C (Gas turbine exhaust temperature = 596°C). For the modified Brayton cycle with LT-Economizer, the approach temperatures within the recuperator are higher than that of the other two configurations due to a lack of independent control of the heat recovery process from the flue gas and sCO_2 turbine exhaust simultaneously. This lack of control of the heat recovery process also results in significantly lower turbine inlet temperatures compared to the other two cycle configurations. To compensate for the lower turbine inlet temperature and higher cold-end recuperator approach temperatures, the optimization process leads to lower cooler temperatures and higher maximum cycle pressure for the modified Brayton cycle with LT-Economizer. On the other end of the spectrum, the Cascade cycle offers maximum control of the heat recovery process leading to higher turbine inlet temperatures and lower recuperator approach temperatures.

Table 6. Optimized design variables for different sCO_2 bottoming cycle options.

Design Variable	Modified Brayton (with LT-Economizer)	Modified Brayton (with LT- and HT-Economizer)	Cascade Cycle
T_{Cooler} (°C)	18.8	20.6	21.6
TIT (°C)	537.0	563.5	587.1
P_{Max} (MPa)	30.1	30.6	27.7
$T_{App,HTR}$ (°C)	60.0	15.0	13.0
$T_{App,PHX}$ (°C)	7.3	9.0	15.0
$T_{App,LTR}$ (°C)	6.4	6.8	7.5
$T_{App,LT-Econ}$ (°C)	2.8	6.2	4.0

$Q_{HT-Econ}$ (MW)	N/A	129.1	215.6
Q_{PHX} (MW)	470.1	362.9	276.4
ΔP_{HTR} (kPa)	320.0	290.0	190.0
ΔP_{LTR} (kPa)	120.0	120.0	220.0
ΔP_{MC} (kPa)	4.9	4.5	4.9
ΔP_{MCIC} (kPa)	50.0	120.0	110.0
X_{LT}	N/A	N/A	20.6%

Table 7 and Table 8 show the overall performance summary and account level TPC, respectively, for all the optimized sCO_2 bottoming cycle options investigated in this study along with the reference B32B.95 case. The natural gas flow rate and gas turbine efficiency are the same for all the cases. None of the sCO_2 bottoming cycle options have higher plant efficiency than the reference NGCC plant when minimizing for the LCOE. The water consumption of all the sCO_2 power cycles is lower than the reference plant due to differences in the cooling technology for the bottoming cycle (wet cooling vs. adiabatic cooling). Despite having a lower turbine inlet temperature and higher recuperator approach temperatures, the modified Brayton cycle with LT-Economizer offered the highest plant efficiency of all the sCO_2 bottoming cycle options, and the plant efficiency is 0.3 percentage points lower than the reference NGCC plant.

Table 7. Performance summary for optimized sCO_2 bottoming cycle options.

Parameter	B32B.95	Modified Brayton (with LT-Economizer)	Modified Brayton (with LT- and HT-Economizer)	Cascade Cycle
Natural gas Flow Rate (kg/hr)	124,605	124,605	124,605	124,605
HHV Combustion Turbine Efficiency, %	38.0%	38.0%	38.0%	38.0%
HHV Net Plant Efficiency, %	48.7%	48.4%	48.2%	48.2%
Water Consumption (gpm/MW_{net})	4.3	3.9	3.8	3.8
Power Generation Summary				
Combustion Turbine Power (MWe)	686.0	686.0	686.0	686.0
sCO_2/steam Power Cycle (MWe)	256.0	245.0	242.0	241.0
Total Gross Power (MWe)	**942.0**	**931.0**	**927.0**	**926.0**
Auxiliary Breakdown (kWe)				
Circulating Water Pumps	5,570	3,620	3,620	3,620
Combustion Turbine Auxiliaries	1,320	1,320	1,320	1,320
Condensate Pumps	200	-	-	-
Cooling Tower Fans	2,880	1,870	1,870	1,870
Adiabatic Cooling System	-	2,496	2,501	1,950
CO_2 Capture/Removal Auxiliaries	19,200	19,200	19,200	19,200
CO_2 Compression	25,130	25,130	25,130	25,130
Feedwater Pumps	5,760	-	-	-

Ground Water Pumps	520	430	430	430
Miscellaneous Balance of Plant	710	710	710	710
SCR	3	3	3	3
sCO_2/Steam Turbine Auxiliaries	230	230	230	230
Transformer Losses	3,020	2,970	2,960	2,950
Total Auxiliaries, MWe	**65**	**58**	**58**	**57**
Net Power, MWe	**877.0**	**873.0**	**869.0**	**868.0**

The TPC of the sCO_2 bottoming cycles is slightly lower or similar to that of the reference B32B.95 case on a $/kWe basis. HRSG or primary heater costs are lower for the sCO_2 bottoming cycles due to higher approach temperatures and overall conductance (UA). Likewise, the feedwater and cooling water system costs are lower for the sCO_2 bottoming cycles, however, these are compensated by the higher sCO_2 power cycle costs. sCO_2 power cycle costs are nearly twice that of steam Rankine cycle on a TPC basis. These differences arise from the need for additional heat exchangers (recuperators, coolers, and intercoolers) for sCO_2 power cycles. As seen in Table 9, coolers and intercoolers make up nearly 43 – 48 percent of the total sCO_2 power cycle costs. Recuperators make up an additional 26–30 percent. Therefore, combined, the heat exchangers make up ~75 percent of the total sCO_2 power cycle costs. Despite having higher sCO_2 power cycle costs and lower plant efficiency, the modified Brayton cycle with LT-Economizer has slightly lower overall TPC ($/kWe basis) than the reference NGCC plant.

Table 8. Capital costs (TPC/1,000) breakdown for optimized sCO_2 bottoming cycle options.

Cost Account Description	B32B.95	Modified Brayton (with LT-Economizer)	Modified Brayton (with LT- and HT-Economizer)	Cascade Cycle
Feedwater & Miscellaneous BOP	$139,816	$117,385	$117,229	$116,921
Flue Gas Cleanup & Piping	$588,429	$571,598	$571,598	$571,598
Combustion Turbine & Accessories	$220,813	$220,813	$220,813	$220,813
HRSG, Ductwork, & Stack	$168,537	$129,104	$159,527	$163,872
Steam/sCO_2 Turbine & Accessories	$87,607	$160,351	$152,039	$167,315
Cooling Water System	$59,145	$45,436	$45,413	$45,407
Accessory Electric Plant	$86,659	$82,146	$82,122	$81,718
Instrumentation & Control	$25,072	$24,672	$24,671	$24,635
Improvement & Site	$33,192	$33,009	$32,951	$32,927
Buildings & Structure	$20,691	$20,157	$20,051	$19,998
Total	**$1,429,961**	**$1,404,649**	**$1,426,415**	**$1,445,204**
Total, $/kWe	**1,630**	**1,610**	**1,641**	**1,665**

Table 9. sCO_2 power cycle capital cost (TPC/1,000) breakdown for optimized sCO_2 bottoming cycle options.

	Modified Brayton (with LT-Economizer)	Modified Brayton (with LT- and HT-Economizer)	Cascade Cycle
Main CO_2 Compressor	$11,716	$11,699	$11,817

High Temperature Recuperator	$16,047	$19,206	$29,929
Low Temperature Recuperator	$26,719	$21,061	$20,261
Adiabatic Coolers	$77,212	$70,885	$73,296
CO_2 Turbine	$11,926	$11,989	$14,834
Piping System	$12,703	$12,703	$12,703
System Foundations	$4,543	$4,496	$4,475
Total	**$160,351**	**$152,039**	**$167,315**
Total, $/kWe	**$184**	**$175**	**$193**

Figure 10 shows the LCOE breakdown of the sCO_2 bottoming cycles along with the reference NGCC plant (B32B.95 case). LCOE of the sCO_2 bottoming cycles is slightly lower or similar to that of the reference NGCC plant. Of the sCO_2 bottoming cycle options, the modified Brayton cycle with LT-Economizer offered the lowest LCOE and the LCOE is $0.4/MWh lower than the reference NGCC plant.

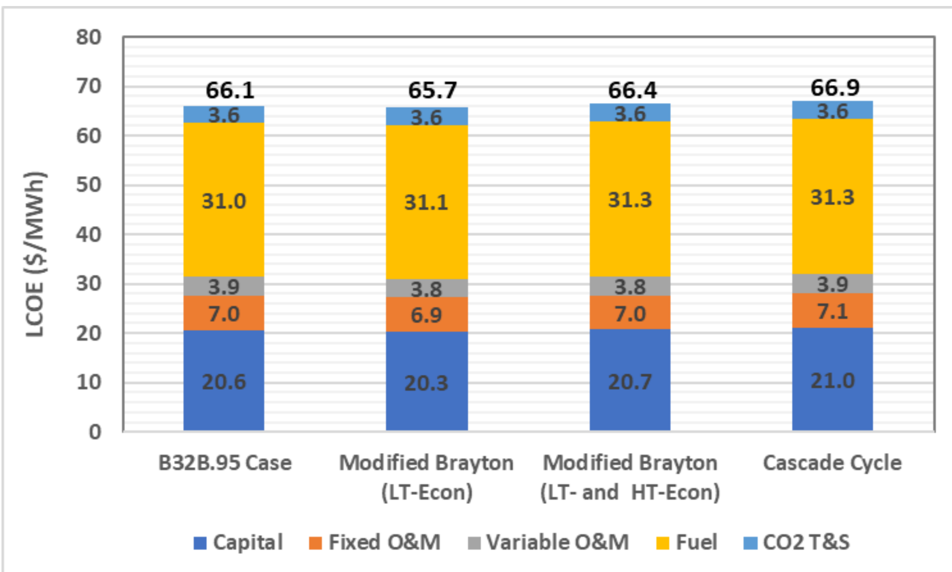


Figure 10. LCOE breakdown for optimized sCO_2 bottoming cycle options.

Impact of Exhaust Gas Temperature

The gas turbine temperature for the reference case selected in this study is 596°C. However, there is some uncertainty in this value based on the literature review and values from GT-PRO. According to GT-PRO and other literature data, the exhaust gas temperature is 629.0°C for the GE 7HA.02 gas turbine. Increasing the gas turbine exhaust temperature to 629.0°C and re-optimizing the modified Brayton cycle with LT-economizer increases the bottoming cycle power from 245 MW to 256 MW. This translates to a ~1 percentage point increase in the plant efficiency (HHV basis) and a $0.5/MWh decrease in LCOE. Table 10 presents the performance and economic breakdown of modified Brayton cycle (with LT-Economizer) for exhaust gas temperatures of 596°C and 629°C. This indicates that as the gas turbine exhaust temperature increases, the sCO_2 bottoming cycles get more efficient due to higher turbine inlet temperatures and could offer a more economic benefit compared to a steam bottoming cycle for NGCC applications with capture. It should also be noted that for all the cases in this study assume that the gas turbine design point is unchanged. However, for sCO_2 bottoming cycles the optimal gas turbine design point might be different from that of steam bottoming cycles as shown in study conducted by Thanganadar et al. [23]. Hence, optimization of NGCC plants with sCO_2 bottoming cycles should consider modifications to gas turbine design which requires use of an accurate gas turbine model to determine several design parameters such as coolant flows, pressure ratios etc.

Table 10. Performance and Economic breakdown for modified Brayton cycle with different gas turbine exhaust temperatures

Parameter	B32B.95	Modified Brayton (with LT-Economizer; EGT=596°C)	Modified Brayton (with LT-Economizer; EGT=629°C)
Natural gas Flow Rate (kg/hr)	124,605	124,605	126,432
HHV Combustion Turbine Efficiency, %	38.0%	38.0%	38.4%
HHV Net Plant Efficiency, %	48.7%	48.4%	49.4%
Water Consumption (gpm/MW_{net})	4.3	3.9	4.4
Combustion Turbine Power (MWe)	686.0	686.0	692.0
sCO_2/steam Power Cycle (MWe)	256.0	245.0	256.0
Total Gross Power (MWe)	**942.0**	**931.0**	**948.0**
Total Auxiliaries, MWe	**65**	**58**	**59**
Net Power, MWe	**877.0**	**873.0**	**889.0**
LCOE Breakdown ($/MWh)			
Capital	20.6	20.3	20.0
Fixed O&M	7.0	6.9	6.8
Variable O&M	3.9	3.8	3.8
Fuel	31.0	31.1	31.2
CO_2 T&S	3.6	3.6	3.5
Total	**66.1**	**65.7**	**65.2**

CONCLUSIONS

This study presented the techno-economic optimization results of NGCC power plants with carbon capture based on sCO_2 bottoming cycles. As part of the optimization, three sCO_2 bottoming cycle configurations were examined – modified Brayton cycle with LT-Economizer, modified Brayton cycle with LT- and HT-Economizer, and Cascade cycle. Multi-variable automated design optimization is conducted for each option using the CMA-ES solver available under NETL's FOQUS platform. With the objective function to minimize LCOE, sCO_2 bottoming cycles have similar plant efficiency and LCOE compared to state-of-the-art NGCC plants with triple-pressure reheat steam bottoming cycles. Of the sCO_2 bottoming cycle options, the simplest cycle configuration (with only LT-Economizer) is more efficient and cost-effective for NGCC plants with capture. This configuration also has the lowest turbine inlet temperature. As the gas turbine exhaust temperature increases beyond 600°C, the sCO_2 bottoming cycles will become more competitive compared to steam Rankine bottoming cycles.

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