5th International Supercritical CO_2 Power Cycles Symposium
March 29 - 31, 2016, San Antonio, Texas

"**Challenges in Designing Fuel-Fired sCO_2 Heaters for Closed sCO_2 Brayton Cycle Power Plants**"

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Abstract

The closed Brayton Power cycles using supercritical CO_2 (sCO_2) as the working fluid is driven by heat added to the sCO_2 at high temperature. For fuel-fired implementations of the closed cycle technology, this means a fired sCO_2 heater similar in concept to the steam generator in a steam-Rankine cycle power plant. There are, however, significant differences between the fired heaters necessary for the sCO_2 Brayton power cycle and steam generators commonly deployed for steam-Rankine cycle power plants. These include: greater recuperation in the sCO_2 Brayton power cycle leading to higher fired heater inlet temperatures (accompanied by lower enthalpy rise in the fired heater), much higher sCO_2 mass flow (per unit power output), and lower tolerance for pressure drop (sCO_2 compression vs. feedwater pumping). These differences will significantly affect the design of the fired heater, and are also likely to impose constraints on the design of the sCO_2 power cycle in order to maximize overall plant efficiency.

A preliminary screening study was conducted to thermally integrate a coal-fired sCO_2 heater with a 750 MWe recompression sCO_2 Brayton power cycle with 700°C (1292°F) turbine inlet temperature; comparable to advanced, ultra-supercritical steam cycle power plants currently being contemplated. The study identified at least two design approaches and identified the challenges in the respective

designs. This paper will summarize the fired sCO_2 heater scoping study and suggest design development needs to meet the challenges.

Introduction

Recent study of closed Brayton power cycles using sCO_2 as the working fluid have focused largely on nuclear and solar thermal resources to provide the high temperature heat. The most notable characteristic of these thermal resources is that the heat can be provided to the power cycle working fluid at a relatively high, relatively constant temperature characteristic of the thermal resources. In contrast, sCO_2 Brayton power cycles that are thermally integrated with largely sensible thermal resources such as combustion gases and “waste” heat will need to make efficient use of the thermal resource over a wide range of temperatures.

A coal-fired deployment of the closed Brayton power cycle will require a coal-fired sCO_2 heater whose function is much like the coal-fired steam generator serving a steam-Rankine power cycle. However, the differences between the sCO_2 Brayton cycle and the steam-Rankine cycle for a given electrical output present a number of design challenges. First, the working fluid inlet temperature to the fired heater is likely to be significantly higher, which complicates the cooling of the furnace enclosure and results in a furnace exit gas temperature well in excess of that used in conventional steam boiler levels. Second, the allowable pressure drop for the working fluid is much lower. This, combined with a much higher working fluid flow rate, complicates the hydraulic design of heat transfer surfaces. The low-pressure drop requirement combined with the higher flow rate also drives up the cost of the expensive high energy piping delivering sCO_2 from the fired heater to the turbine.

To give more definition to the technical challenges associated with coal-fired heaters for a sCO_2 closed Brayton cycle power plant, this scoping study was undertaken by EPRI and The Babcock & Wilcox Company (B&W). The primary objective of the scoping study was to conduct a preliminary design of a coal-fired sCO_2 heater/power cycle to be compared with a similar-sized A-USC steam power cycle to identify those features of the fired sCO_2 heater design that will require additional development.

Scoping Parameters

A previous EPRI study compared a full-scale advanced ultra-supercritical (A-USC) steam power cycle with several recompression sCO_2 Brayton power cycle configurations, all with the same turbine inlet temperature of 704°C (1300°F)[1]. The cycle schematic is shown in Figure 1 along with the cycle state points. A notable feature of this cycle is the temperature of the sCO_2 inlet to the fired heater: 530°C (987°F). This entering sCO_2 will be able to cool the combustion products leaving the fired heater to no less than about 565°C (1050°F). The comparable combustion products temperature leaving an A-USC steam generator will be close to 370°C (700°F) due to a lower feedwater temperature entering the steam generator. For the purposes of this study, a second cascading sCO_2 Brayton power cycle was assumed to cool the combustion products from 530°C (987°F) to 370°C (700°F) at which point a conventional air heater could be employed to recover residual flue gas heat to the combustion air, a common practice in coal-fired steam generators. This second, cascading sCO_2 Brayton power cycle is shown in Figure 2 along with cycle state points.

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[1] *Program on Technology Innovation: Modified Brayton Power Cycle for Use in Coal-Fired Power Plants.* EPRI, Palo Alto, CA: 2013. 1026811.

It should be noted that the combination of these two sCO_2 power cycles is not to be interpreted as an "optimal" closed sCO_2 Brayton power cycle for the coal-fired application. The optimal power cycle flow sheet has not yet been identified at this point in the development of technology. Indeed, it is unlikely that the power cycle flow sheet can be optimized without consideration of practical features in the fired heater design. While not an optimal power cycle flow sheet, the combination of these recompression and cascaded power cycles does provide for working fluid heat sinks whose temperatures match flue gas temperatures within commonly accepted steam boiler practice and, thus, can serve to scope fired sCO_2 heater design in sufficient detail to give visibility to design challenges not commonly encountered in steam boiler designs.

Scoping Study Approach

A concept and preliminary design for a coal-fired sCO_2 heater providing the heat required for a nominal 750 MWe (net) closed Brayton power cycle was developed to compare with similar scope for a coal-fired A-USC steam-Rankine power plant. The coal-fired sCO_2-1 heater design and its impact on the total plant is compared against the A-USC design previously published by EPRI.[2]

The overall approach taken in this study was to size the fired sCO_2 heater to meet the heat duty required by the 750 MWe (net) recompression power cycle. The cascaded power cycle was then sized to cool the flue gas temperature to near 370°C (700°F). The resulting design has coal flow approximately 14% greater than the baseline 750 MWe (net) A-USC steam cycle plant. (The net generation increase is 73 MWe, a 10% increase.) The overall power process parameters for the A-USC steam-Rankine and recompression/cascaded sCO_2 Brayton power cycle plants are listed in Table 1.

Conceptual Solution

The design concept adopted in this study is generally that of the type of coal-fired compressed air heater supporting a 2.3 MWe (net) closed air-Brayton cycle power plant in Ravensburg, Germany in the 1950s.[3]

The inverted downdraft firing concept to turn over a tower type steam generator was already being considered for A-USC boilers at B&W (EU patent 14187421.4–1610, 05/01/15). A side elevation and plan view of the design concept applied to sCO_2 heating for this study is shown in Figure 3. The primary goal of the design is to shorten the lead distance from the fired heater outlets to the turbine to minimize the amount of high-cost, high-energy pipe material to deliver sCO_2 from the final heater to the turbine.

Plan and Side Elevation

The coal-fired sCO_2 heater design developed for this study is a B&W downdraft inverted tower configuration (patent pending) developed for 700°C-class (1300°F) A-USC steam-Rankine cycle power plants. The cycle mass flow of sCO_2 is about 12 times the water mass flow of the baseline A-USC steam-Rankine cycle power plant. The major design differences between the steam-Rankine and sCO_2-Brayton cycle fired heaters include:

- The sCO_2-side pressure drop must be much lower for components of the sCO_2 Brayton cycle than is commonly allowed in steam generators.

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[2] *Engineering and Economic Evaluation of 1300°F Series Ultra-Supercritical Pulverized Coal Power Plants: Phase 1.* EPRI, Palo Alto, CA: 2008. 1015699.

[3] *Closed Cycle Gas Turbines: Operating Experience and Future Potential.* H. U. Frutschi. ASME Press, New York, NY. 2005. ISBN 0-7918-0226-4.

- The entering sCO_2 temperature is much higher requiring a change to the enclosure wall cooling technique.

The fired heater design uses externally cased, refractory lined tangent tube walls for the furnace and convection pass enclosures due to the lack of sufficient sCO_2 at low enough temperatures to cool an enclosure wall.

It is envisioned that conventional support rods and external structural steel will be used to hang the furnace and convection pass enclosures and heat transfer surfaces. To protect the furnace, it is internally covered by the final superheater as "tangent tube curtain" walls. Curtain walls have been employed in the past, primarily for reheaters, but this design presents some challenges in regard to support and restraint while accommodating a large temperature difference between the cased wall operating at around 370°C (700°F) and the final superheater tubes operating at around 760°C (1400°F) and attendant differential expansions. Since the final superheater is high in nickel content, its coefficient of expansion is not as different from carbon steel casing as stainless steel so these issues are believed to be manageable. The final superheater extends downward in the furnace with the outlet pipes at the bottom to minimize the distance to the turbine, reducing the lengths and costs of the high nickel outlet piping.

The platen superheater is next in the gas flow path from the furnace. A traditional platen arrangement does not provide enough tube flow paths to keep the pressure drop low enough. The smaller inlet and outlet headers are longitudinal with the flue gas flow. The orientation of the platen surface is non-conventional although based on the premise that the dominant radiation heat transfer will provide even absorption per tube and the low convection heat transfer will not produce wide variations in tube-to-tube absorption.

The gas flow path is downward, horizontal, upward into a platen superheater, and then horizontal through the remaining heat transfer surfaces. As the gas makes the horizontal turn at the bottom of the furnace and, after a short horizontal path, flows upward again, perhaps 30% of the ash will drop out. At the rear of the horizontally oriented convection pass is another hopper to collect some additional ash before the gas flows upward and out to an ammonia injection grid and into the selective catalytic reduction (SCR) system for nitrogen oxides (NO_x) removal. (SCR would not be required for oxy-coal fired heaters).

The configuration and auxiliary equipment following the boiler, including SCR, air heater, and primary and forced draft fans, is the same as used in conventional pulverized coal-fired plants.

sCO2–side Flow Schematic and Heat Transfer Surface Performance

The sCO_2 flow schematic is shown in Figure 4. The corresponding temperatures of the flue gas and sCO_2 in each bank are shown in Figure 5.

The mass flow of sCO_2 through the furnace tubes is a factor of 12 higher than the mass flow (and 10 times the volume flow) of steam/water through the furnace tubes of a comparable A-USC steam generator. The furnace tube pressure drop with the required tube thickness is impractically high and would require eight times the number of tubes making up the wall enclosure and still results in 0.66 MPa (94 psi) tube pressure drop.

Key Issues for Further Development

The following are design issues which emerged by this preliminary coal-fired sCO_2 heater scoping study that must be addressed and resolved for commercial-scale deployment.

Process

1. The most challenging feature of a fired heater design to support sCO_2 Brayton power cycles is the issue of fired heater pressure drop. The pressure drop challenge is compounded by a much higher sCO_2 mass flow than is normally encountered in steam generators. In addition, the lower sCO_2 velocities required to limit pressure drop reduce tube side heat transfer coefficient and increase tube metal temperatures increasing tube thickness.
2. The flow of sCO_2 working fluid is about 12 times the mass flow (10 times the volume flow) of a Rankine cycle steam generator near this capacity. While this moderates the rapid enthalpy increase of the sCO_2, which is only 218 kJ/kg (94 Btu/lb) from inlet to outlet, the fluid handling design expense and higher pressure drop adversely impact the cycle efficiency and cost. The heating process for the sCO_2 is extremely rapid compared to steam that has about a 2326 kJ/kg (1000 Btu/lb) enthalpy differential across the fired heater and has a flatter heat capacity (kJ/kg-°C, Btu/lb-°F) over the temperature range of interest.
3. The overall process flow sheet, which integrates the sCO_2 power generation equipment and the fired heater has not been optimized. Such flow sheet optimization is unlikely to be accomplished separately for the power cycle and fired heater; they will need to be designed with full thermal integration.
4. An air heater, which could accept flue gas temperatures up to 565°C (1050°F) and deliver commensurately higher combustion air temperatures to the burners would greatly simplify the power cycle flow sheet. Design and fabrication of such an air heater is likely to require significant development. Significantly higher combustion air temperatures would also require redesign of air ducting, wind box, and the burners.

Furnace Enclosure and Curtain Walls

5. Casing and refractory enclosure wall design (lagging, insulation, stiffened casing, and refractory) including structural support of the wall dead loads (top supports) may require considerable effort to develop.
6. Burner openings in the curtain walls need to be developed and demonstrated.
7. Support of the curtain walls vertically and horizontally needs to be developed and demonstrated.
8. Design concepts that reduce the temperature of the tubes nearest the wall to near the temperature of steam boiler membrane tubes would allow use of more conventional, less expensive furnace insulation systems. However, this would also likely impact the overall power cycle.

Heat Transfer Banks

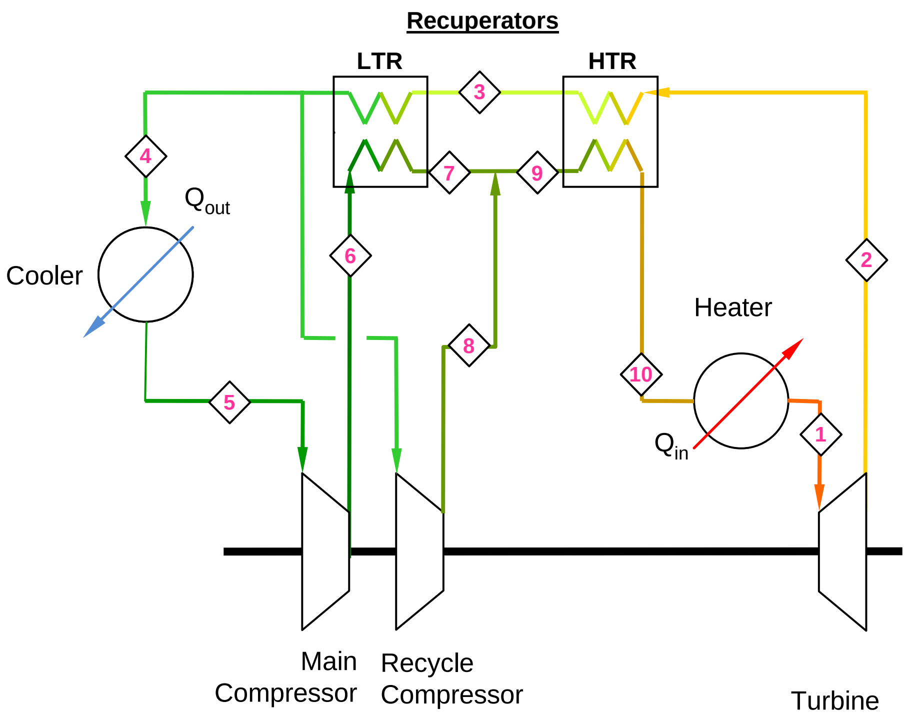
9. The orientations of the platen headers is non-conventional. The tube sections in the gas path are more conventional, although with many tube rows deeper than normal.
10. The design employs in-line tube arrangements for convection banks without using extended surface. The inverted arrangement where the hopper could be credited with higher ash/slag removal rates may allow the use of wider spaced fins on the last heating surface banks.

Operation, Start Up and Shutdown Potential Challenges

11. Steam thermal units are commonly designed using variable pressure to maintain the turbine throttle temperature at full set point for as much of the upper load range as possible. Whether the variable pressure operation is suitable for the heater design was outside the scope of this

study. The control of fluid temperature by firing rate will be slow to respond and may have severe temperature overshoot if rapid load response is needed.

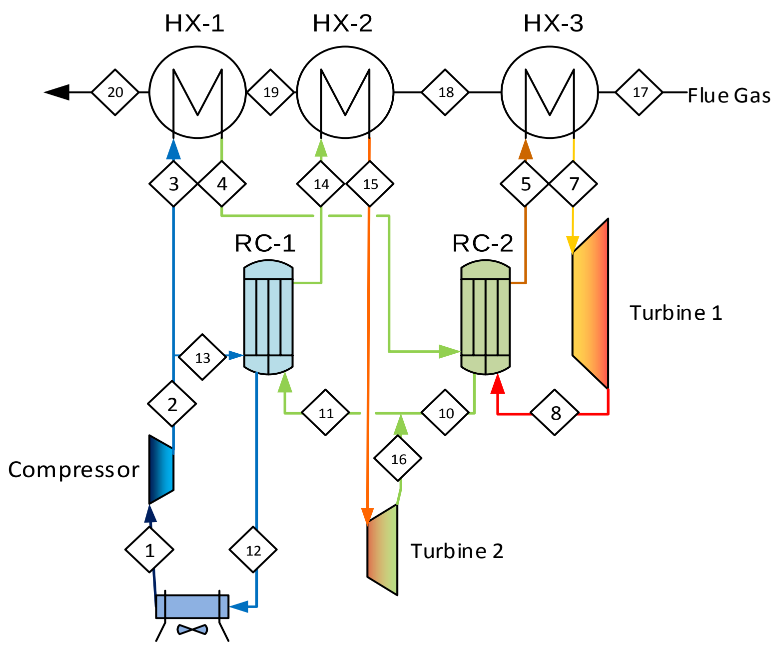
12. Startup is likely to require the motoring of the generator or other means to circulate CO_2 prior to firing.
13. Normal shutdown is likely to require the reduction of firing to a level where net output power is zero. On a sudden loss of load, there is a need to continue circulating sCO_2 until the heater tubes are cooled to a safe level. On a black plant trip, the ability to natural draft the furnace setting to cool the tubes must also be considered.



		Flow	T	P	Enthalpy	Entropy	Efficiency	Power
Tag	Stream	klb/hr	°F	psia	Btu/lbm	Btu/lbm-R	%	MW
1	Heater 1 Out	52,320	1300	3000	528.59	0.70503		
2	Turbine Out	52,320	1070	1143	463.21	0.70982	90%	1002.5
3	HTR Hot Out	52,320	332	1134	259.07	0.52876		
4	LTR Hot Out	31,915	154	1122	204.34	0.45028		
5	Cooler Out	31,915	89	1110	130.74	0.32000		
6	Main Compressor Out	31,915	144	3050	140.62	0.32246	85%	-92.5
7	LTR Cold Out	31,915	320	3035	230.34	0.45449		
8	Recycle Compressor Out	20,405	322	3035	231.09	0.45545	85%	-160.0
9	HTR Cold In	52,320	321	3035	230.63	0.45486		
10	HTR Cold Out	52,320	987	3020	434.77	0.64604		

Gross Power	**1002 MW**
Net Power	**750 MW**
Heat Added	**1438 MWth**
Cycle Efficiency	**52.1%**

Figure 1
Recompression sCO_2 Brayton Power Cycle Configuration and State Points
(HTR = high-temperature recuperator, LTR = low-temperature recuperator)



		Flow	T	P	Enthalpy	Entropy	Efficiency	Power
Tag	Stream	klb/hr	°F	psia	Btu/lbm	Btu/lbm-R	%	MW
1	Compressor In	6144	90	1117	131.7	0.3216		
2	Compressor Discharge	6144	146	3092	141.8	0.3241	85%	-18
3	HX-1 CO_2 In	3963	146	3092	141.8	0.3241		
4	HX-1 CO_2 Out	3963	392	3062	255.5	0.4849		
5	HX-3 CO_2 In	3963	707	3032	325.1	0.5822		
7	Turbine 1 In	3963	986	3002	434.2	0.6457		
8	Turbine 1 Out	3963	783	1150	381.7	0.6504	90%	61
10	RC-2 Low-Pressure Out	3963	428	1139	285.0	0.5593		
11	RC-2 Low-Pressure In	6144	506	1139	306.0	0.5821		
12	Compressor Inlet Cooler In	6144	320	1128	255.7	0.5246		
13	RC-2 High-Pressure In	2181	146	3092	141.8	0.3241		
14	HX-2 CO_2 In	2181	479	3062	283.5	0.5162		
15	Turbine 2 In	2181	842	3031	391.6	0.6142		
16	Turbine 2 Out	2181	648	1139	344.3	0.6190	90%	30

Gross Power	**91 MW**
Net Power	**73 MW**
Heat Added	**293 MWth**
Cycle Efficiency	**25 %**

Figure 2
Cascaded sCO_2 Brayton Power Cycle Configuration and State Points
(HX = heat exchanger, RC = recuperator)

Table 1
Comparison of Power Cycle Parameters

Parameter	Baseline AUSC Rankine Cycle	Recompression	Cascaded
Net Output (MW_e)	750	750	73
Working Fluid Mass Flow Rate (klb/hr)	4,620	52,320	6,144
Throttle Pressure (psi)	5085	3000	3000
Throttle Temp (°F)	1256	1300	986
Working Fluid Specific Volume (ft^3/lb)	0.1782	0.1497	0.1214
Working Fluid Volume Flow Rate (ft^3/min)	13,720	130,540	12,430
Reheat Temp (°F)	1292	n/a	n/a
Feed Temp (°F)	649	987	147
Inlet Pressure (psia)	5670	3020	3020
Specified Heater Pressure Drop (psid)	585	20 (not achieved)	
Heat Input (million Btu/hr)	5,930	6,759	
Fired Heater Efficiency (HHV)	87.1%	87.3%	

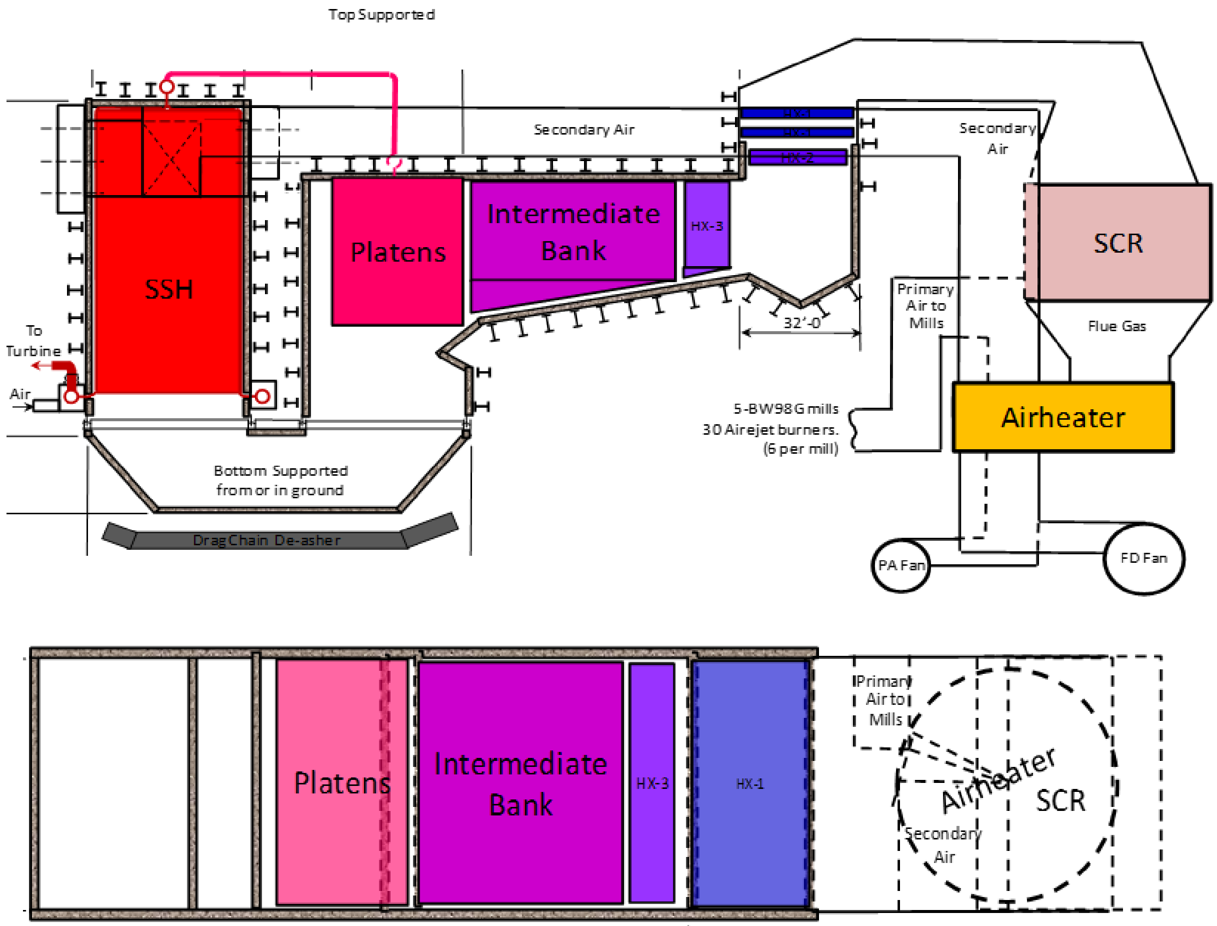


Figure 3
Elevation and Plan Views of the Coal-Fired sCO_2 Heater

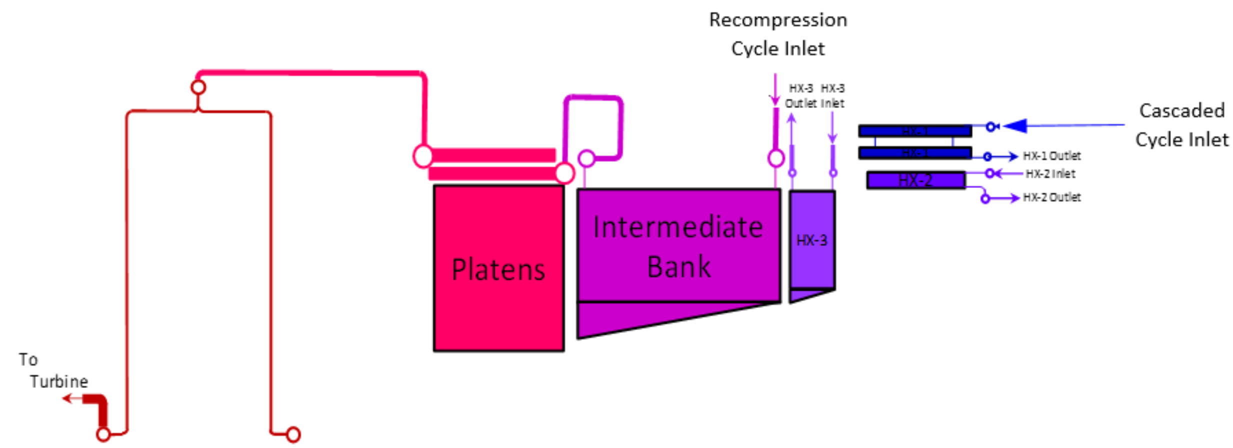


Figure 4
Coal-Fired sCO_2 Heater Working Fluid Flow Schematic

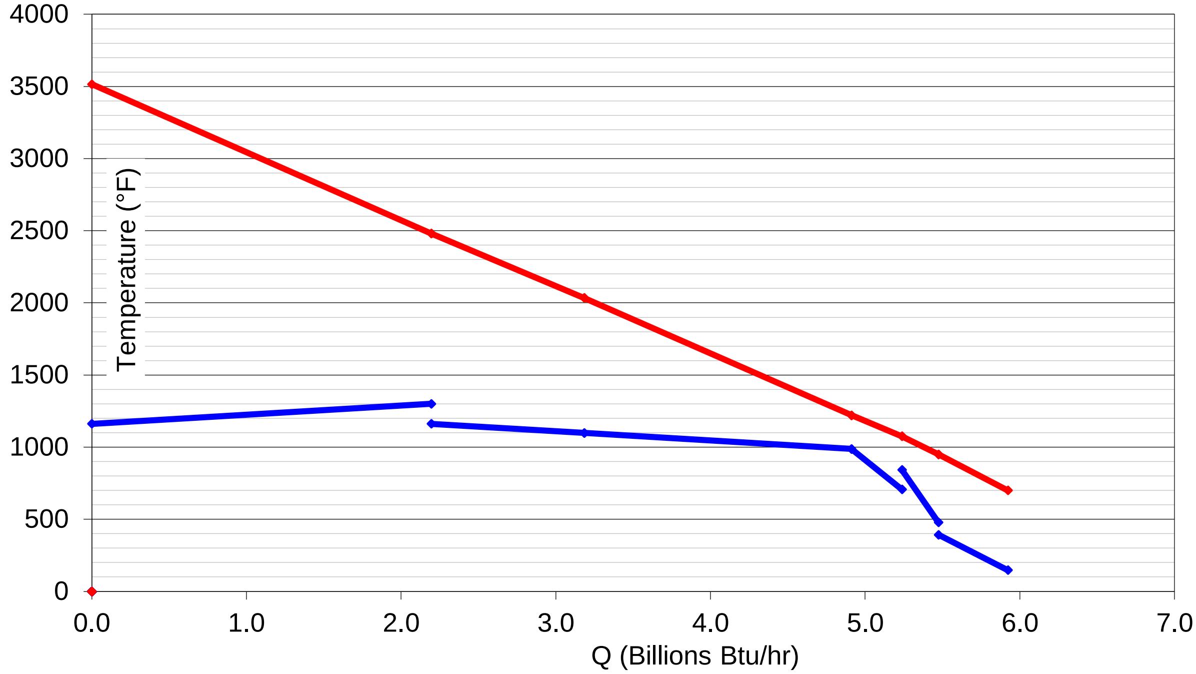


Figure 5
Flue Gas and CO_2 Temperature vs. Absorption of Heating Surface