

## Behavior of a Solid Fuel Fired Primary Heat Exchanger in a 1.5 MW<sub>th</sub> Closed Loop Brayton Cycle Using sCO<sub>2</sub>

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research, biomass / coal co-firing for CO<sub>2</sub> mitigation, improved boiler metering and control for efficiency improvements at low load and through load ramping and advanced supercritical CO<sub>2</sub> power cycles.

### ABSTRACT

Most demonstration projects of closed-loop Brayton power cycles using supercritical CO<sub>2</sub> are fired on natural gas, requiring only a shell and tube configuration for the primary heat exchanger. Firing solid fuels is more complicated, where most of the heat transfer occurs

through radiant panels. The coupling of a solid fuel firing system with a radiant primary heat exchanger for supercritical CO<sub>2</sub> is more challenging than in traditional coal-fired boilers because there is no phase change occurring for the back-side fluid. Special consideration in the design and operation must be made to manage the coupling of the burner heat release profile and tube metal temperature and heat flux. The overall objective of our project was to perform the R&D necessary to mitigate the risk associated with the design of the primary heater for solid fuel-fired systems by utilizing small pilot-scale testing and advanced modeling to optimize the conceptual design process. The pilot-scale efforts were centered on a 1.5 MWth pulverized coal furnace (L1500) at the Utah San Rafael Energy Lab (USREL) in Orangeville, Utah. This reactor has been operated for over 400 hours while firing natural gas and pulverized biomass and coal. The finished sCO<sub>2</sub> conditions achieved in this facility include a temperature of 607 °C and a pressure of 20.3 MPa simultaneously at a flow rate of 5.5 kg/s CO<sub>2</sub>. During these trials, data were collected to evaluate the effect of load changes on tube metal temperatures in the near burner region. The tube metal temperature change as a function of an 11% load change was determined to be 107.6 °C/hr and the rate of change was determined to be 281.8 °C/hr. These values match those found in the literature for ultra-supercritical steam Rankine Cycle boilers.

## INTRODUCTION

The Rankine Cycle forms the backbone for electric power generation all across the world. It has poor cycle efficiency due the large heat rejection occurring in the condenser. This loss is thermodynamically necessary, but because of the shape of the water-steam vapor dome, the quantity of heat loss is comparatively large relative to the work generated. This limitation of the Rankine Cycle can't be avoided by creative engineering. Studies into advancing Rankine Cycle Efficiency focus on increasing the hot reservoir temperature through advances in materials. Other heat engines are possible in many different configurations, which leverage other working fluids. It is desirable to find a working fluid with potentially better properties for cycle efficiency and hardware size. Of particular interest are closed loop Brayton Cycles leveraging supercritical CO<sub>2</sub> (sCO<sub>2</sub>). sCO<sub>2</sub> is an ideal working fluid for closed loop Brayton Cycles because of its high heat capacity and density while still retaining compressive properties like a gas. When applied in a power cycle, a portion of the low-pressure operation (downstream of the turbine) can be devoted to heat recovery, or recuperation, which increases cycle efficiency.

Substantial research and development effort is being expended on coupling sCO<sub>2</sub> power cycles to various thermal sources for power generation. One of the first applications of interest was concentrated solar [1]. This interest has very quickly spread to other thermal sources including fossil heat [2], Nuclear [3] and geothermal [4]. Of more recent interest are applications for energy storage, which can focus on mechanical storage through compressing gas (supercritical fluid) [5] and thermal storage [6]. sCO<sub>2</sub> power cycles have not been rapidly deployed commercially due to technical challenges. Some of these challenges include careful control of operating conditions near the CO<sub>2</sub> critical point, controlling CO<sub>2</sub> leakage and developing mechanical equipment with high power requirements at small sizes. One specific example of this is the development of an sCO<sub>2</sub> turbine. To achieve high efficiencies at such high energy densities, the turbine must spin at tens of thousands of RPMs as opposed to 3600 RPMs for steam turbines in the United States. This poses special challenges for shaft seals and thermal and pressure gradients [7].

Despite a growing research focus on application of sCO<sub>2</sub> cycles to various thermal energy resources, little attention has been directed at our existing fossil-fuel power infrastructure, specifically coal. The U.S. Department of Energy developed a program titled 21<sup>st</sup> Century Power Plants, which had a key research thrust of coupling advanced power cycles with coal fired electric power generation. The results presented in this paper are derived from a project funded

under this program DE-FE0031928. The overall objective of this project was to perform the R&D necessary to mitigate the risk associated with the design of the primary heater by utilizing large lab-scale / small pilot-scale testing and advanced modeling to optimize the conceptual design process for a coal-fired heater intended for use in an indirectly fired sCO<sub>2</sub> power cycle. This project specifically developed a 1.5 MW<sub>th</sub> demonstration plant.

The crux of the demonstration project was developing and controlling a PHX which coupled solid fuel combustion with an sCO<sub>2</sub> power cycle through radiation heat transfer. In all traditional entrained flow, solid fired power systems, the heat exchange surfaces near the burners are water-wall radiation panels. In these systems, there is an enormous amount of heat required to overcome the latent heat of vaporization of the water. This is an excellent arrangement because the phase change occurring inside the radiative heat exchanger constrains the temperature of the tube metal, regardless of the proximity of the flame and the ensuing heat flux. Therefore, the shape of the flame, which is controlled by flame aerodynamics, and the ensuing heat release profiles are not of primary concern. Those variables are typically manipulated to control NO<sub>x</sub> formation. In an indirectly fired Brayton Cycle using supercritical CO<sub>2</sub>, the tube metal surface temperatures are not constrained by a phase change. The coupling of the flame heat release profile and radiative heat transfer surface becomes the primary consideration. Showing that this coupling is possible at industrial scales with relevant burner mixing scales and solid burner configurations represents a large step forward for the application of sCO<sub>2</sub> power cycles in the power industry.

Heat exchanger materials of construction are a primary focus for some researchers. Operation at high temperatures is necessary to achieve optimal efficiencies. In many cases this requires the use of advanced materials, including austenitic steels. Additionally, the steel interface with CO<sub>2</sub> may be subject to corrosion mechanisms that are not prevalent in water systems [8]. In any case, operation of austenitic steel heat exchangers requires careful control of metal temperatures, heat fluxes and thermal transients. The objective of this paper is to establish a relationship between changes in firing rate and tube metal temperatures for the radiant section of the PHX in a closed loop sCO<sub>2</sub> power cycle and compare them to the behavior of water wall tubes in a Rankine cycle boiler.

## **RESULTS AND DISCUSSION**

### **Methods**

The 1.5 MW<sub>th</sub> pilot-scale combustor (L1500) at the San Rafael Energy Research Center is a pulverized coal-fired furnace that was designed to simulate combustion in low emission, pulverized coal-fired boilers. This unit has been used for many investigations of technologies for NO<sub>x</sub> and particulate control, including staging, reburning, SNCR and burner development. The reaction zone of this furnace has a one-meter, square cross section and is approximately 14 meters in length. The length is divided into 10 sections, each with various sampling and injection ports. The furnace is refractory lined, with cooling panels in the first four sections to maintain realistic boiler temperature profiles. Multiple ports are in each of the reactor sections, allowing for numerous configurations of sampling, reagent injection and overfire air.

The burner installed on the L1500 is a low-NO<sub>x</sub> burner with multiple air registers for aerodynamic staging of the flame and is detailed in Figure 1. This burner configuration consists of multiple annular pipes and a quarl, for introducing the combustion reactants.

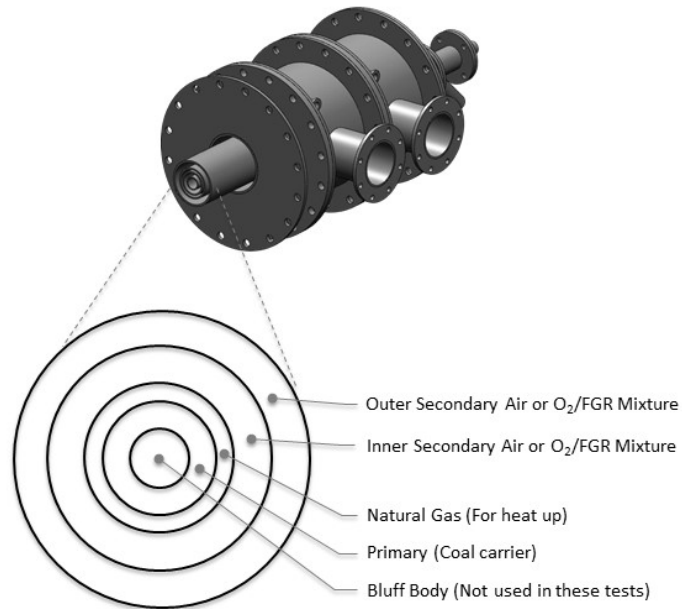


Figure 1. Annular register description of the low- $\text{NO}_x$  burner on the L1500

These annular registers include from inside to out: Bluff Body (Injection of any reactant or reagent), Primary (coal, air, FGR,  $\text{O}_2$ ), Gas (natural gas), Inner Secondary (air, FGR,  $\text{O}_2$ ), Outer Secondary (air, FGR,  $\text{O}_2$ ). The Inner and Outer Secondary registers have a swirl block which can induce a tangential velocity component to the gas flow. At full swirl (100%), the swirl number is approximately 1.2 and at 0% swirl the gas flow is purely axial.

The L1500 was retrofitted with a primary heat exchanger (PHX) to allow it to integrate with a skid containing turbomachinery, creating a closed loop  $\text{sCO}_2$  Brayton cycle. The PHX is comprised of a radiant and a convective module, both of which are in the radiant section of the L1500 as shown in Figure 1. A crossover tube, not shown in Figure 1, connects the headers of the convective and radiant modules of the PHX.

Material selection for the PHX was made to handle both the high temperature and pressure expected within a typical  $\text{sCO}_2$  system. Table 1 shows the designed conditions used when designing the PHX. The main considerations during the design process were the temperature and pressure, but flowrates, heat flux, and pressure drop were contributing factors. The design outlet temperature is limited to 873 K to eliminate the necessity for utilizing nickel alloys. The incremental expense of nickel alloys was determined to not justify the higher cycle efficiency achievable at higher temperatures [9]. Super 304H was selected since it has better stress characteristics at high temperatures than most comparable materials without going to nickel alloys.

Table 1. Design conditions for the primary heat exchanger (PHX)

	Units	Value
sCO <sub>2</sub> Inlet Temperature	°C	415
sCO <sub>2</sub> Outlet Temperature	°C	600
sCO <sub>2</sub> Flow Rate	kg/s	5.5
sCO <sub>2</sub> Operating Pressure	MPa	20.4
Pressure Part Design Pressure	MPa	27.4

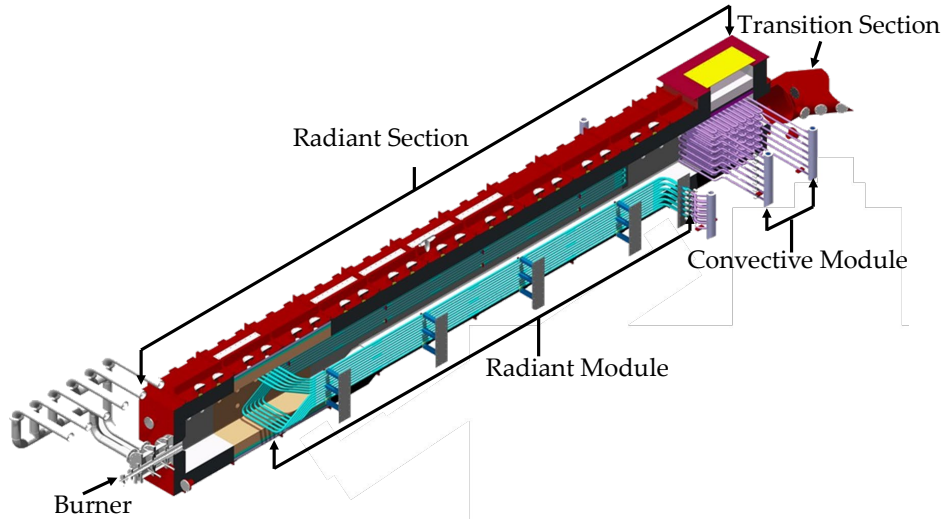


Figure 2. 3D Schematic of the primary heater (PHX) in the radiant section of the L1500. The radiant module of the PHX is shown in blue and the convective module of the PHX is shown in purple.

A thermal management system is coupled with the L1500 to form a pilot-scale sCO<sub>2</sub> Brayton cycle power plant. The turbomachinery provides the L1500 with sCO<sub>2</sub> at the desired flowrate, temperature, and pressure for testing. The turbomachinery includes a motor-driven positive displacement pump driven by a variable frequency drive, a printed circuit recuperative heat exchanger, and an active CO<sub>2</sub> inventory control system. A turbine is not used in this project as the main objective of the DOE project is to test the performance of the PHX. Instead, a throttling valve was used, providing more flexibility in testing. The last component is a printed circuit, water-cooled waste heat exchanger. The piping in the skid and interconnection piping with the PHX was designed to withstand high temperature and pressure, but unlike the PHX, it does not experience high radiative heat flux. SA312-TP316H, a high carbon variant of 316 stainless steel, was chosen for the piping in the skid. Additional details on the turbomachinery used can be found in the literature [10]

The system was configured as a simple recuperated cycle because the objective of this program was not to prove efficiency, but to show the coupling of a solid fuel-fired system with a sCO<sub>2</sub> power cycle. If proving efficiency were the objective, much more attention to heat integration would be necessary. A simple process flow diagram showing this cycle is included as Figure 2. The conditions of the CO<sub>2</sub> at each of the points indicated in Figure 2 are included in Table 2. The coupled process cycle conditions are illustrated on a P-H diagram in Figure 3. The

conditions chosen for the coupled process indicate a heat uptake of  $1.26 \text{ MW}_{\text{th}}$  by the PHX. From stream a to b, the sCO<sub>2</sub> is preheated in the recuperative heat exchanger and it continues to absorb heat as it passes through the PHX from points b to c. From point c to d, the sCO<sub>2</sub> flows through the throttling valve. From points d to e, the sCO<sub>2</sub> is cooled in the recuperative heat exchanger and additional cooling occurs from points e to f in the chiller. The sCO<sub>2</sub> is brought back up to pressure from points f to a from the compressor.

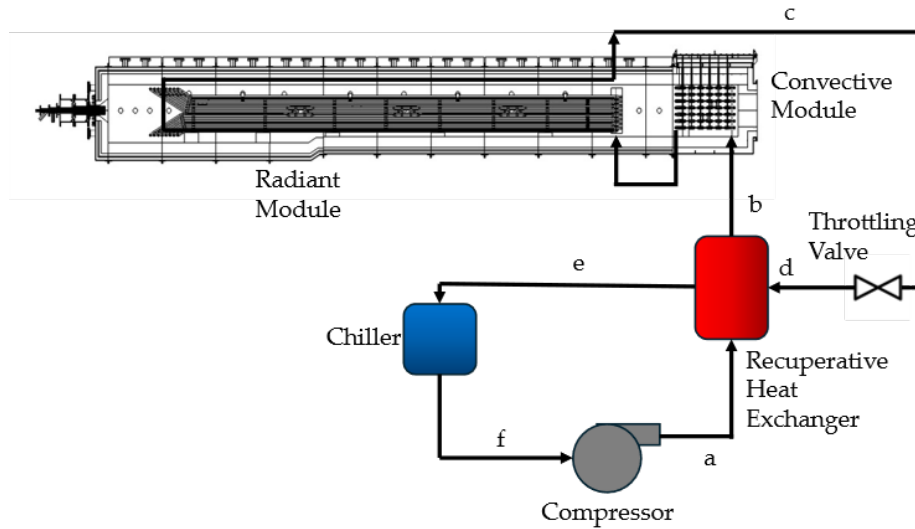


Figure 3. Schematic of the sCO<sub>2</sub> flow through the L1500 and sCO<sub>2</sub> Brayton Cycle Turbomachinery. The throttling valve mimics the pressure drop across a turbine in the Brayton cycle. The streams in the sCO<sub>2</sub> flow are numbered.

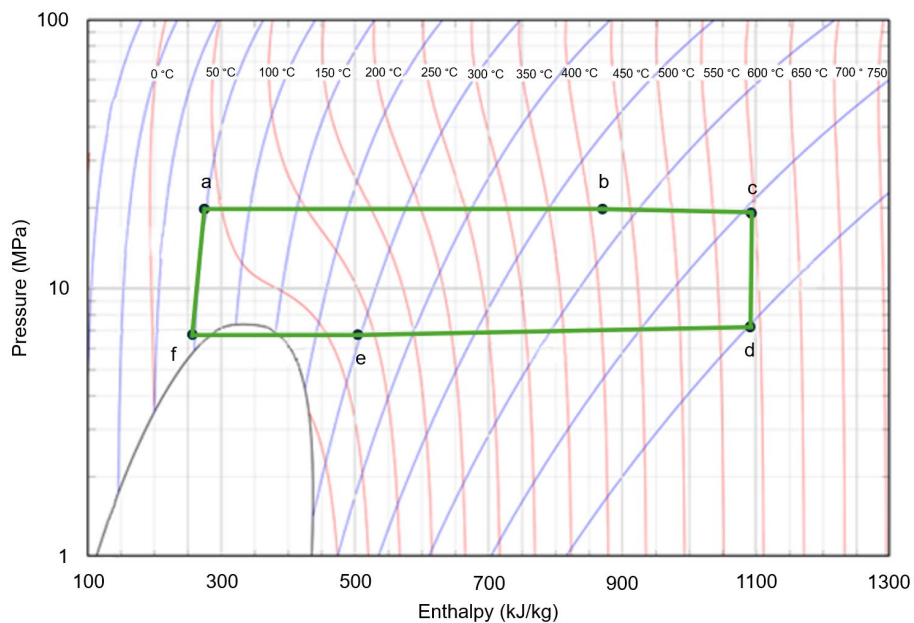


Figure 4. Pressure-enthalpy diagram of the sCO<sub>2</sub> Brayton cycle at the design conditions. Note that a throttling valve is used instead of a turbine.

Table 2. State points for the design conditions. State point numbers are labeled on Figure 2 and Figure 3.

State	P (MPa)	T (°C)	h (kJ/kg)	Flow (kg/s)	Duty (kW)
a	20.68	39	273.32	5.5	115
b	20.28	415	868.52	5.5	3274
c	19.88	600	1097.46	5.5	1259
d	7.18	593	1097.46	5.5	0
e	6.88	80	502.27	5.5	-3274
f	6.78	21	252.48	5.5	-1374

The PHX has six Clyde Bergmann heat flux sensors embedded in the tube walls. Each heat flux sensor is comprised of four thermocouples, two of which are shallowly embedded in the tube wall and two are more deeply embedded. These thermocouples allow for the calculation of the heat flux through the tube walls and for the extrapolation of the surface temperature of the tubes. These sensors allow the operator to determine if there is a risk of overheating the tubes at critical locations within the furnace. Figure 4 shows a schematic of the location of the six heat flux sensors on the radiant module. Three of the heat flux sensors are located near the burner and the other three are located further down the furnace. Sensors 1 and 2 are located on the wall in the background and the remaining sensors are along the wall in the foreground. Sensors 3 and 4 are at the same axial distance from the burner but are on different tubes with different elevations.

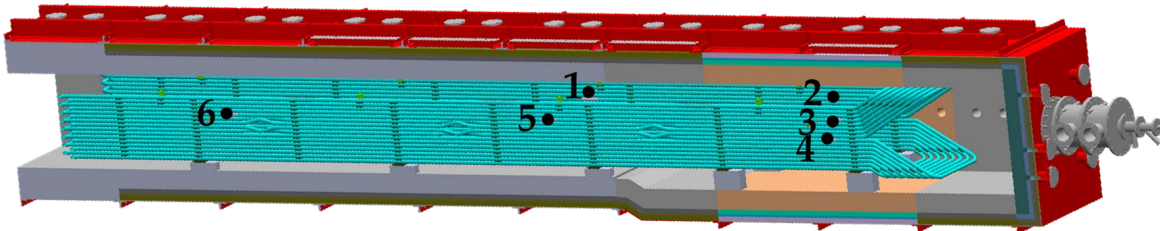


Figure 5. Schematic of the location of the six heat flux sensors on the radiant module in the radiant section of the L1500. Note that sensors 3 and 4 are at different elevations at the same axial location relative to the burner.

The completed and coupled system is detailed in Figure 5 and Figure 6.



Figure 6. View of the from the front end, with burner and secondary air flow mechanisms detailed. The outlet header from the radiant section of the PHX is detailed on the far left of the furnace.



Figure 7. The thermal management system, the CO<sub>2</sub> supply tank and transfer pump, pictured from outside of the building.

## Results

Once the pilot-scale system was constructed, the coupled system was operated for a total of 407 hours, with the longest continuous run of 248 hours. For 62% of the operational time, the unit was unmanned and in automatic control. The fuels used for the testing included natural gas, two Utah Bituminous coals, woody biomass, and bagasse. During the testing we were able to verify the 1.2 MW<sub>th</sub> heat uptake and we operated at a finished CO<sub>2</sub> temperature of 607 °C and a pressure of 20.3 MPa simultaneously. Specifics of this testing can be found in the literature [11].

One focus of the test plan was to evaluate operational variables that were expected to have a strong correlation with heat flux distribution on the radiant section of the PHX and the ensuing tube metal temperatures. Through CFD modeling that was employed during the design phase of the PHX we learned that the most important operating parameters were firing rate, excess air and CO<sub>2</sub> flow rate. Our testing focused heavily on controlled, and sometimes uncontrolled, changes in the firing rate and CO<sub>2</sub> flow rate. In practice, changes in firing rate were simultaneously coupled with changes in CO<sub>2</sub> flow rate in an effort to mitigate the impact of firing rate on tube metal temperatures while also controlling the finish CO<sub>2</sub> temperature. Across the 407 hours of operation, there were many instances of change in firing rate. For purposes of this investigation, two instances of firing rate change were selected which were controlled and included only changes to fuel feed and CO<sub>2</sub> flow rates.

Data presented in Figure 7 show the coal and natural gas flow rates over the full period of one day's operation. Figure 8 shows the CO<sub>2</sub> pump pressure and CO<sub>2</sub> finish (return) temperature for that same period. On this particular day of operation, the CO<sub>2</sub> pressure was operating at well below the design pressure. These data show the following manipulations to operation. Overnight unattended operation at 1.25 MW<sub>th</sub> while firing natural gas until about 8:15 AM. At about 8:15 AM the firing rate on natural gas was increased from 1.25 to 1.4 MW<sub>th</sub>. The firing rate on natural gas was again increased from 1.4 to 1.6 at about 9:00 AM. The following data show conversion from natural gas firing to coal firing. At 9:25 AM there was an attempt to change from gas to coal in small step changes, which resulted in a solid-fuel feeder trip. Therefore, the natural gas firing rate of 1.6 MW<sub>th</sub> was restored. At about 10:30 AM a second conversion to coal firing was initiated, which was ultimately successful. Coal-fired operation ensued until approximately 4:00 PM when the fuel feed was converted back to natural gas at a firing rate of 1.25 MW<sub>th</sub> for overnight unattended operation. Throughout the operational changes previously described, the CO<sub>2</sub> flow was changed in order to manage tube metal temperature and ultimately finish CO<sub>2</sub> temperature.

Conversion from natural gas to solid fuel firing requires several subtle changes to the burner operating conditions which can also have a dramatic effect on burner aerodynamics and in turn flame shape and heat release profile. First, air must be redistributed in the burner registers. The secondary air flow is reduced and the difference is directed through the primary air register. Additionally, the secondary air preheaters are turned on and set to operate at 260 °C. This is necessary to preheat and ignite the solid fuel particles. Additional adjustments are made to the secondary air distribution between the inner and outer burner registers and to the swirl blocks in order to stabilize the flame. All of these changes are impactful to the flame heat release profile and on the PHX tube conditions. The surface temperatures at the locations of the heat flux sensors detailed in Figure 4 for the same period of operation are presented in Figure 9.

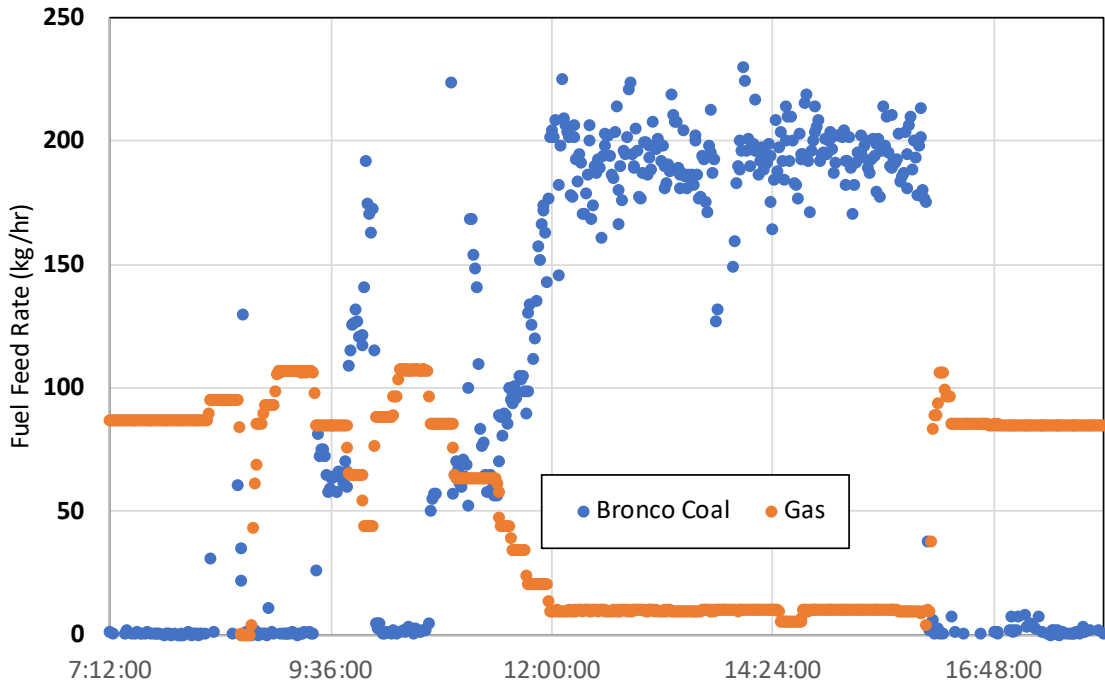


Figure 8. Natural gas and coal feed rate for a 10 hour operating period exhibiting changes in firing rate and fuel blend.

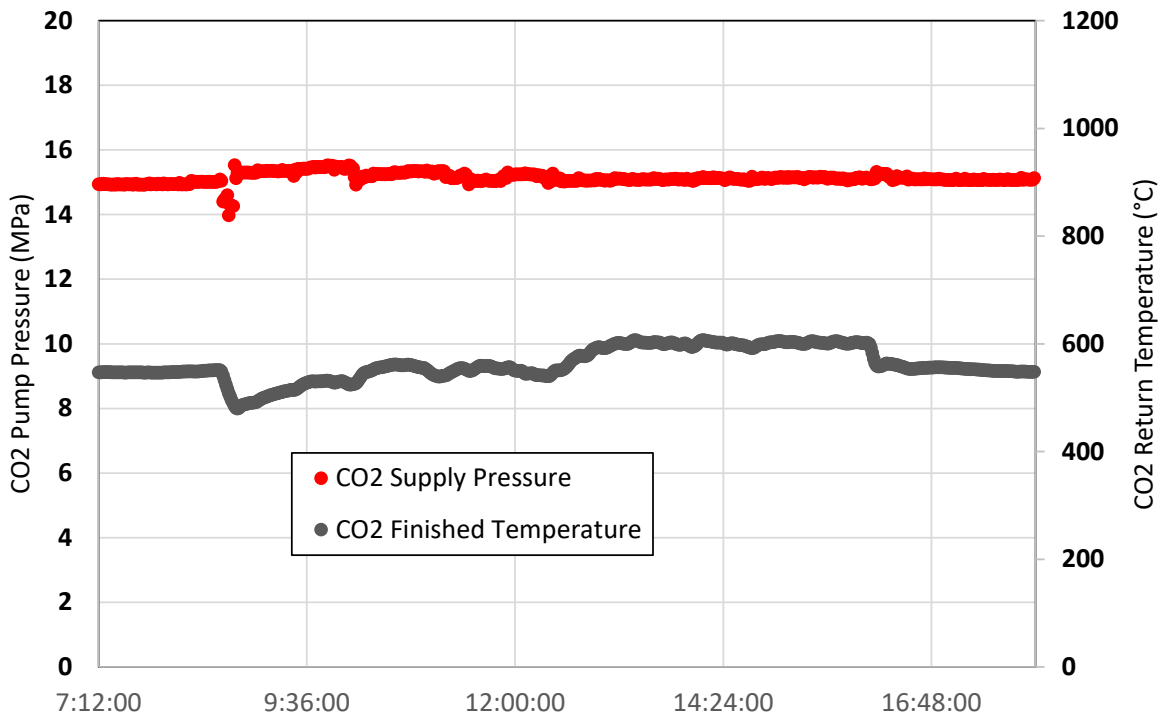


Figure 9. CO2 Pump pressure and return temperature across a 10 hour operating period exhibiting changes in firing rate and fuel blend.

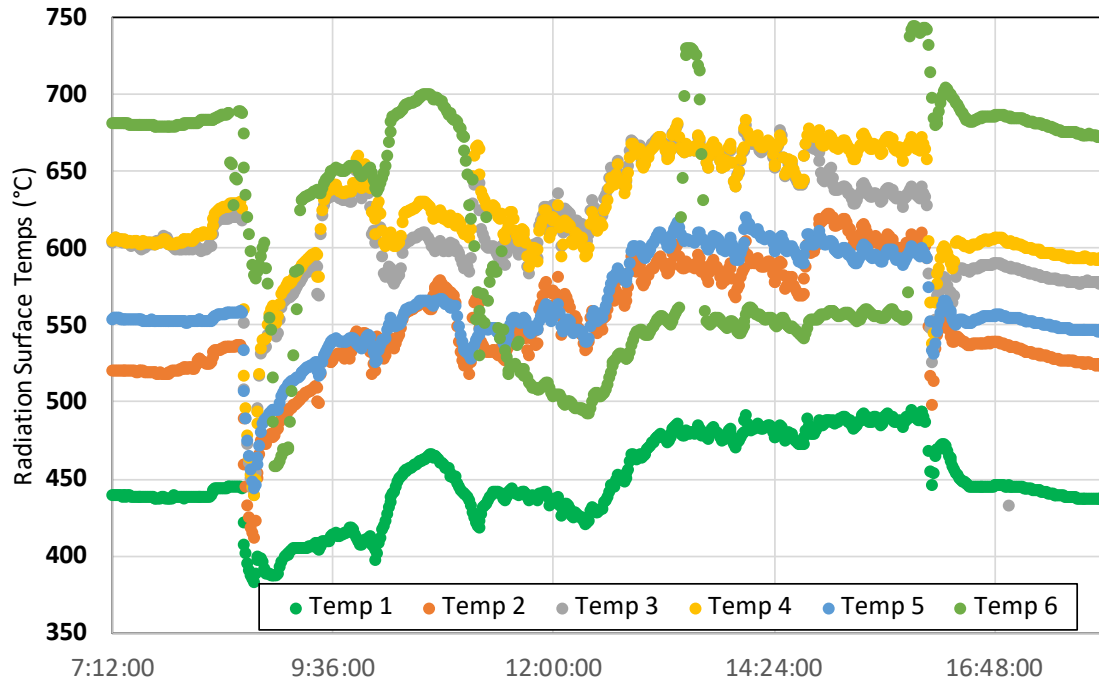


Figure 10. Tube metal temperatures in the radiant section across a 10 hour operating period exhibiting changes in firing rate and fuel blend.

The temperature data represented in Figure 9 are quite messy. There are several operating issues that can explain the behavior here. The significant drop in temperature at 8:37 AM happened when the burner was being prepared for the switch from gas to coal. As the air distribution was being changed, the flame became unstable and extinguished. During the 9:40 – 10:20 AM period, as the burner was being switched to solid fuel, there were some problems with the baghouse and ID fan which made it difficult to control the furnace pressure. Then, the successful conversion to solid fuel had a drastic impact on the heat release profile. Fortunately, these conditions were held at a relatively steady state for several hours. The dramatic nature of the temperature data in Figure 9 show the strong relationship between tube metal temperature and operating conditions, particularly perturbations in operation.

Despite some of the difficult conditions and somewhat messy data, there were some operational changes that were clean and useful for diagnosing the relationship between tube metal temperature and firing rate. One of these occurred at 8:15 AM while firing natural gas. At this time the firing rate was increased from 1.25 to 1.6 MW<sub>th</sub>. During this increase in firing rate, the only other variable that was manipulated was the CO<sub>2</sub> flow rate. The operating data for this operational change are presented in Figure 10 and Figure 11. Figure 10 shows the natural gas and CO<sub>2</sub> flow rate for a 30-minute period centered on the firing rate change. Figure 11 shows the tube metal temperatures for the same period.

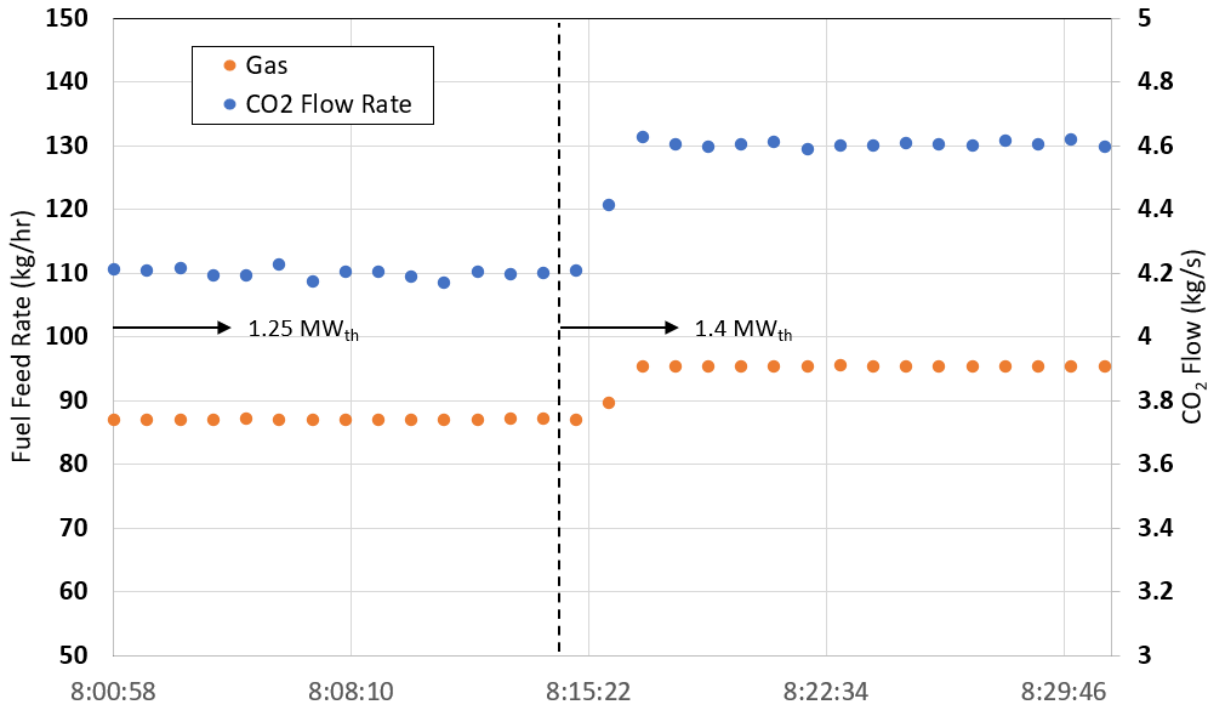


Figure 11. Firing rate and CO2 flow rate while other operating conditions were held constant.

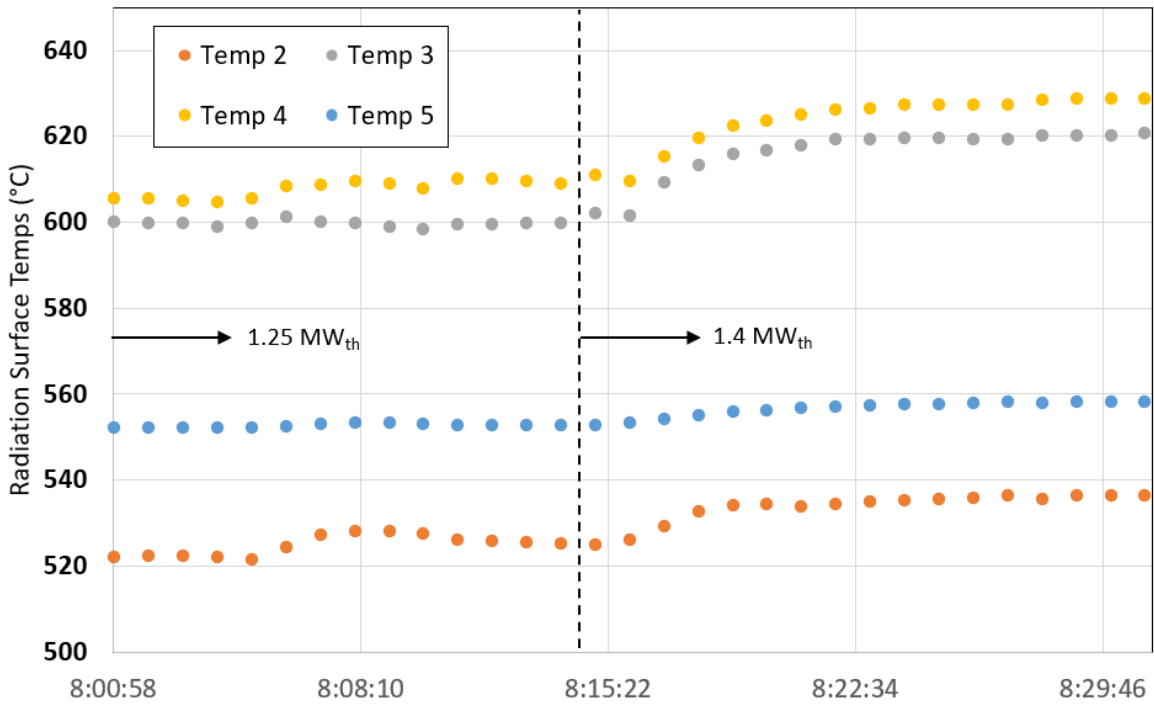


Figure 12. PHX surface temperature at each heat flux sensor location during an increase in firing rate and CO2 flow rate.

There are many instances throughout the 407 hours of operation where condition changes like this have been made. These data have been evaluated to determine the effect of load changes on tube metal temperature the time scale for these effects to reach steady state and the rate of

change. This evaluation was done for the heat flux sensors in the near burner region. It was also useful to compare these values to what is reported in the literature for both water wall steam systems running the Rankine Cycle and sCO<sub>2</sub> systems. Yan reports experimentally measured water wall tube metal temperatures in a 1000 MW Ultra-Supercritical Boiler with load changes from valves wide open (VWO) to 22% full load [12]. These data represent real operating conditions, where firing and steam rate were adjusted simultaneously similar to the operating conditions that we evaluated. Wang reports model predictions for wall temperature change rates and duration of transient for step changes in load of 10% from 100% to 40% full load operation of a sCO<sub>2</sub> closed-loop Brayton Cycle [13]. This information is compared to the operating data from this experimental study in Table 3. In this table, the temperature response to load change is calculated as the temperature change divided by the % load change. In this way, the temperature change is normalized and does not count absolute load in MW.

*Table 3. Comparison of the transient tube metal temperature behavior to load change in the 1.5 MW<sub>th</sub> L1500 sCO<sub>2</sub> system to 1000 MW<sub>e</sub> ultra-supercritical steam Rankine Cycled data (Yan) [12] and to the modeled behavior of a 370 kW<sub>e</sub> sCO<sub>2</sub> closed loop Brayton Cycle system [13].*

	<b>L1500 sCO<sub>2</sub></b>	<b>Yan 1000 MW</b>	<b>Wang Model</b>
<b>Temperature Response to Load Change (°C/Load Change)</b>	107.6	104.9	
<b>Temperature Ramp Rate (°C/hr)</b>	281.8		288
<b>Transient Duration (sec)</b>	450 (90% steady)		201

The data presented in Table 3 suggest show that the response of the tube metal temperature in the 1.5 MW<sub>th</sub> sCO<sub>2</sub> system is nearly identical to that measured in a 1000 MW<sub>e</sub> ultra-supercritical steam Rankine Cycle boiler. This was surprising considering the number of variables that would have been exercised during a load change in a full-scale boiler. However, in both cases the working fluid is supercritical, under similar operating conditions, experiencing similar heat flux variations and encased in metal tubes of similar properties.

The temperature ramp rate was also nearly identical to what was predicted in the Wang model for a similarly configured sCO<sub>2</sub> power cycle and for nearly identical step changes in firing rate as those measured in this data set. The transient response time was over two times longer in our system as compared to the predicted result. This is not surprising as the L1500 is refractory lined and contains significantly larger thermal mass.

## **Summary and Conclusions**

A demonstration unit for a solid fuel fired sCO<sub>2</sub> closed loop Brayton Cycle power system has been developed at the scale of 1.5 MW<sub>th</sub>. This system has been operated for over 400 hours while firing natural gas, coal and biomass. These are the only data sets available anywhere showing the behavior of an sCO<sub>2</sub> system coupled with solid fuel firing. Great care was taken in designing the primary heat exchanger (PHX) such that the effects of burner behavior and flame radiation heat release on tube metal temperatures could be managed. Heat flux sensors were placed at various locations along the radiant section of the PHX which also have the capability of reporting tube metal temperature. Operating conditions were evaluated including variation in fuel feed and CO<sub>2</sub> flow rate, and load change, for their effect on tube metal temperature in the

near burner region. Major conclusions from this evaluation include:

- Tube metal temperatures were shown to be extremely sensitive to operating conditions.
- The tube metal temperature change as a function of load change was determined to be 107.6 °C/load change, which matches the behavior of tube metal temperatures in full-scale ultra-supercritical coal-fired utility boilers.
- The rate of change was determined to be 281.8 °C/hr when making load variations of 11%, which matches predicted rate cooling/heating rates in the literature for sCO<sub>2</sub> systems
- The duration of the transient for the L1500 system is longer than predicted in the literature, likely due to the large refractory thermal mass in the L1500

This facility will be used in the future for long-term demonstration of sCO<sub>2</sub> turbomachinery. It will also be adapted for thermal demonstration of advanced nuclear reactors.

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