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Supercritical Carbon Dioxide Brayton Power Cycle Test Loop - Operations Review

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ABSTRACT

The design and operation of the supercritical carbon dioxide (sCO_2), Brayton power cycle heat exchanger test facility will be reviewed. The reconfigurable test loop, will be presented as a simple recuperated cycle, with a pump instead of a compressor and a pressure reducing valve used in place of the turbine. The test loop can accommodate test pressures up to 255 bar and temperatures up to 700°C. The sCO_2 mass flow rate can be adjusted up to 10 kg/min.

The system is sized to reduce cost barriers associated with evaluating innovative ideas in heat exchanger design. Its current focus is on collecting heat exchanger thermal performance measurements to evaluate compact Recuperator, and Heater heat exchangers.

System operations and safety consideration will be reviewed. This includes experiences with system startup and shutdown, transient analysis, heat exchanger thermal performance, and the performance of test loop component such as pumps, filters, valves, and sensors.

INTRODUCTION

The supercritical carbon dioxide (sCO_2) Brayton power cycle is adaptable to a variety of heat sources and has higher cycle efficiency compared to traditional power cycles. As such, it is being considered for multiple power generation applications such as waste heat recovery, fossil, concentrated solar power, geothermal, and nuclear.

A key determinant on power cycle efficiency and profitability is heat exchanger (HX) performance and cost. To evaluate heat exchanger performance, Thar Energy, LLC (Thar), with funding support from U.S. DOE NETL [1,2], has designed, built and operated a test loop simulating a simple recuperated Brayton cycle. The reconfigurable test loop has been sized to evaluate innovative ideas in HX design. Thar plans to evaluate two, 100 kWt prototype recuperators, as part of the development of the U.S. Department of Energy Modular High Temperature Recuperator program. [3]

RESULTS AND DISCUSSION

Test Loop Description

The sCO_2 HX test facility at Thar, is a simple recuperated cycle, Figure 1, with a pump instead of a compressor and a pressure reducing valve used in place of the turbine. The pressure reducing control valve available was limited to ~420°C operation, so this loop is set up to reduce the pressure before the heater, rather than after it, to prevent overheating the control valve.

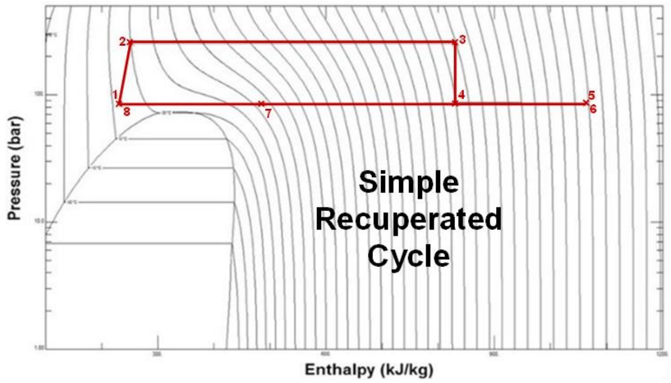


Figure 1. Test loop operates as a simple recuperated Brayton cycle.

The test loop flow diagram, Figure 2, shows that part of the facility is located outside for safety considerations. As shown, the test loop recuperator acts as a combined low temperature and high temperature recuperator, transferring on the order of 84 kWt from the lower pressure (86 bar), higher temperature stream to the higher pressure (255 bar), lower temperature stream.

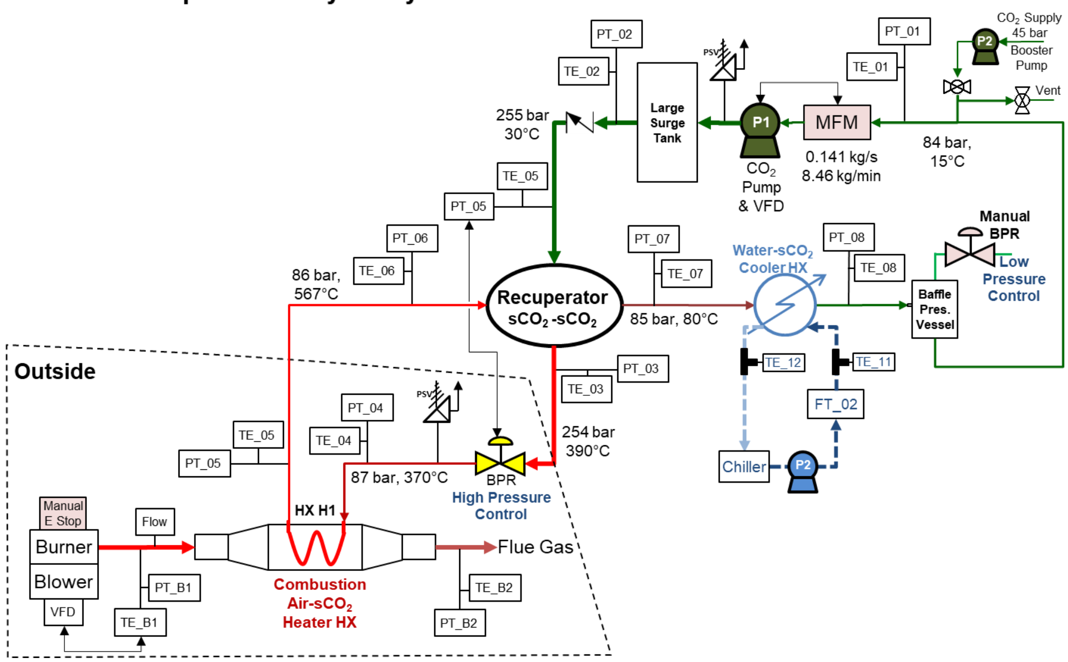
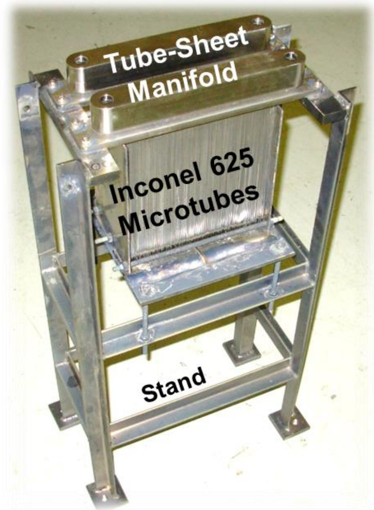


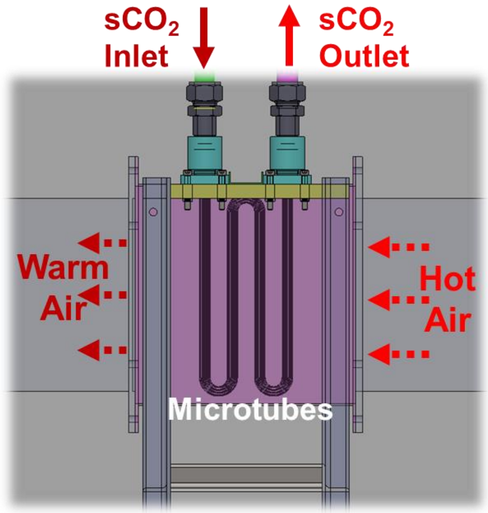
Figure 2. sCO_2 heat exchanger test loop flow diagram.

Major components for the test loop, include:

- 500,000 BTUH Gas Fired Burner/Blower
- 40 kWt Combustion Gas/Air to sCO_2 Primary Heater HX, Inconel 625 construction, Figure 3
- 100 kWt sCO_2 to sCO_2 counter-current, Recuperator HX, Inconel 625 construction, Figure 4
- 40 kWt water to sCO_2 gas cooler, 304 stainless steel construction, Thar shell and tube design
- sCO_2 pump with VFD and Mass Flow Monitor
- Automated Back Pressure Regulator (e.g. Pressure Control Valve)
- PLC Controls and Data Acquisition System
- Pressure and Temperature Sensors
- Test Stand Hardware
- CO_2 supply hardware
- Water Chiller

The primary Heater HX, Figure 3a, used in this test loop was based on Thar's compact HX design. The Hot Air to sCO_2 Heater HX was designed to function in a cross-flow, counter-current arrangement, using serpentine formed microtubes, Figure 3b. The HX used ~1 mm O.D., Inconel 625 microtubes, with a HX face area of 17.8 cm x 17.8 cm, and an area density of ~1,800 m^2/m^3.

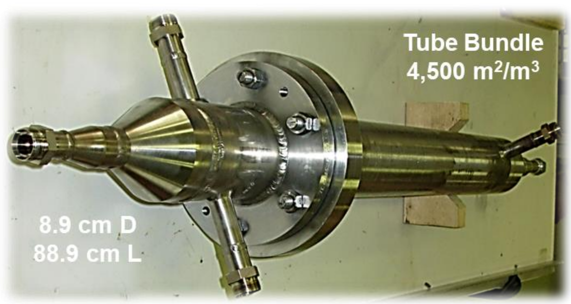


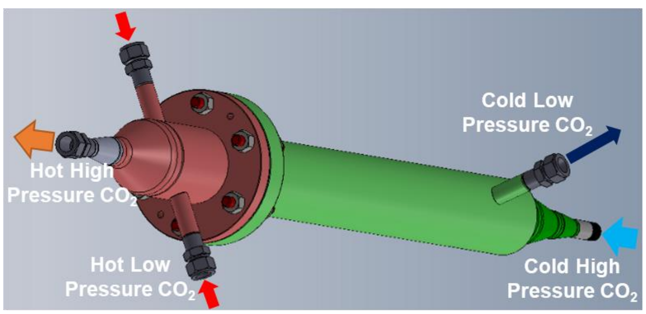


a) 40 kWt microtube heater HX **b) Cross-flow, counter current arrangement**

Figure 3. Hot air to sCO_2 compact microtube heater heat exchanger.

The microtube recuperator, Figure 4a, is a shell and tube HX that uses horizontal separators, versus vertical baffles, so that it operates as counter-current HX, Figure 4b. The high pressure sCO_2 travels in the microtubes and the lower pressure sCO_2 travels in the annular hour-glass space formed by the separator sheets and microtube outer diameter. It has a floating tube-sheet design to accommodate thermal stress and a replaceable tube bundle for ease of maintenance. The single flange design, optimizes materials' use for hot and cold sides, and uses fewer metal seals & bolts, for improved reliability and ease of assembly. Additional design and performance advantages of microtube heat exchangers have been presented in references 4-8.





a) 100 kWt microtube recuperator **b) Counter-current arrangement**

Figure 4. sCO_2 to sCO_2 compact, single flange, microtube recuperator.

Figure 5 shows the inside and outside portions of the test loop, after pressure testing and commissioning, but prior to the final installation of the insulation.

Prior to commissioning, all test loop components were installed and pressure tested. This included the tubing runs, manual valves, computer controlled valves, check valves, pressure safety relief valves, and the installation of temperature, pressure and flow sensors. The test loop was flushed to reduce particle contamination and filters cleaned. Temperature and pressure sensors were then calibrated and PID loops tuned.

a) Inside test loop **b) 100 kWt microtube recuperator** **c) Outside test loop**

Figure 5. sCO_2 heat exchanger test loop before final insulation installed.

Controls

The test loop data acquisition and controls system uses an Allen Bradley CompactLogix PLC controller. The PLC controller uses RSLogix for receiving data, processing, and sending logic signals to the various control devices in the loop. It has mixed modules using analog and digital Murr Elektronik I/O extension modules and Micro Motion Ethernet/IP Modules for communication. The operator interface is designed with Factory Talk, and includes instrumentation monitoring, PID control loops for the VFD controls for the pump and blower, valve and back pressure regulator control, and alarm and interlock notifications.

During the design process, a failure mode effects analysis, was conducted to identify potential failure modes and the effects of those failures as well as to rate the severity of the failure. Some of the many possible failures identified that can occur were CO_2 gas leakage, loss of power, pump seal failure, cooling water failure, tube/pipe blockage, and valve or regulator malfunction. Sensors for pressure, temperature, mass flow, and liquid level are installed throughout the loop. The output of these sensors are evaluated by the control system to detect conditions out of the established operating range.

From this evaluation, alarm and trip points were established. Any sensor operating out-of-range will set off both an audible alarm and a signal in the operator interface. Out-of-range operating conditions detected by the sensors that required a trip condition were grouped into the following categories: Over pressure, High differential pressure, Low pressure, Over-temperature, Low mass flow, and Low liquid level.

For a trip condition, it was agreed that an immediate “pull the plug” by cutting the power to everything was not desirable as lack of cooling flow could cause failure to other components of the loop. It was determined that the trip conditions can be broken down into two shutdown scenarios:

1. Both normal shutdown and high temperature trip shutdown – the burner is shut off immediately and the pump continues to run until temperature sensors downstream of the heater continuously read less than 100°C.
2. Emergency shutdown button trip, over pressure trip, low pressure trip, high differential pressure trip, low mass flow trip and low liquid level trip – the burner and pump are both shut down immediately and the control valve slowly reduces pressure in the large surge tank allowing cool gas to continue flowing through the loop until temperature sensors downstream of the heater continuously read less than 100°C. In this scenario, the high pressure gas flows into the low pressure section of the loop and when the gas pressure exceeds the low pressure set point, the excess is vented through the manual back pressure regulator. It should be noted for this shutdown scenario that the emergency stop button configuration is in accordance with Category 1 of NFPA 79 9.2.2 Stop Functions Controlled Shutdown.

Commissioning of the above individual components includes leak and operational testing. System commissioning was also completed for the sCO_2 pump system, water chiller system, and the natural gas burner/blower system (which came equipped with its own logic permissives that controls alarms and trips).

System Operations

The following steps were taken to operate the test loop and to thermally characterize the HXs:

- Fill the system with CO_2
- Start the pump and adjust mass flow rate to the desired set point
- Adjust system pressures for both high and low sides
- Turn on the burner/blower and adjust burner air flow and temperature to achieve the desire sCO_2 temperature.

During system start-up and steady-state operation, 35 sensors were used to monitor system pressure, temperature, and flow. The temperature and pressure of the entering and exiting fluid streams are measured to assure a good energy balance is obtained.

A few lessons learned with regards to system start up include:

- Turning the burner on - The density of the sCO_2 flowing through the heater can quickly drop fivefold. This rapid expansion, if not accounted for, can increase the pressure such that the safety relief valves are set off. It was necessary to keep the CO_2 level, at a minimum, on the low pressure side, to allow for a safe start without setting off pressure relief valves.
- The CO_2 triplex piston pump worked best if the sCO_2 density was above 600 kg/m^3. During start-up, it was sometimes necessary to bleed off low density sCO_2 or run a bypass loop, to get higher density sCO_2 to the pump.
- It was observed that large, sudden changes, in system pressure, temperature, or flow could cause pump cavitation.
- Compression Fittings. The cone and thread compression fittings were best for providing leak free operation. Swagelok fittings held up well, but off brand compression fittings did not perform well at elevated temperature/pressure conditions. If thermal expansion was not properly taken into account, then even the Swagelok fittings would develop leaks.
- Measurement accuracy was reduced because of the combined bias of each instrument and/or sensor with the data acquisition system, as shown in Table 1. Senor and instrument calibrations were made to reduce the test loop bias.

Table 1. Test loop system bias.

Sensor/System	Value
Coriolis flow meter	+/- .25%
K type thermal couple accuracy	Greater 2.2°C or 0.75%
Pressure transducer accuracy 0-2000 psi	+/- 1.0%
Pressure transducer accuracy 0-5000 psi	+/- 1.0%
Data acquisition system	+/-.5% FS

HX Thermal Performance Data Analysis

The test loop responded quickly to changes in flow, pressure, or temperature. At sCO_2 flow rates on the order of 7 kg/min, the system would stabilize within 15-20 minutes for a step change of ~50°C, Figure 6.

Figure 7 is a temperature vs time plot for the recuperator where the higher pressure stream is ~245 bar and the lower pressure stream is ~85 bar. The steady state portion of this temperature and pressure data is averaged, and along with mass flow data RefProp is used to calculate the fluid stream enthalpy. For each HX data set an energy balance is performed, to confirm system performance and data quality.

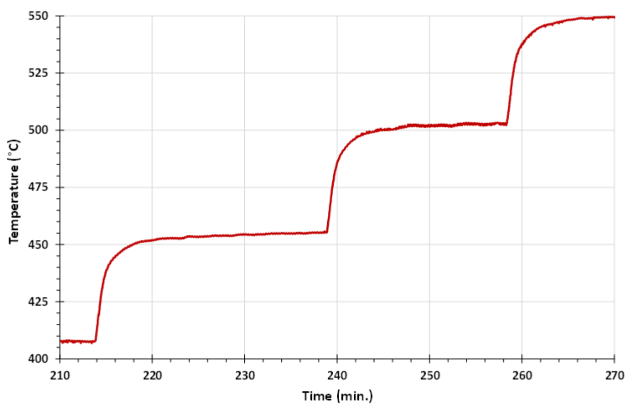


Figure 6. Recuperator transient time vs. temperature plot.

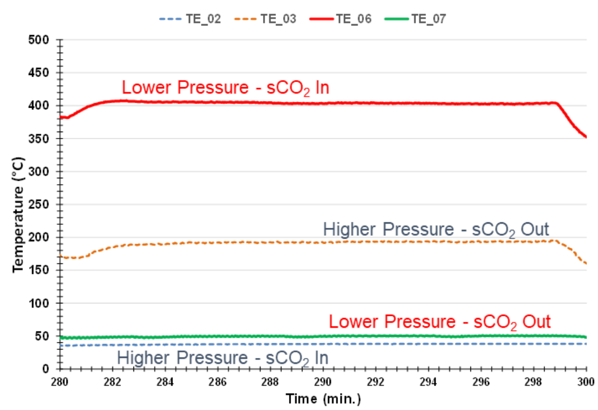


Figure 7. Recuperator steady state time vs. temperature plot.

Figure 8 presents a summary of the energy or heat balance for both the Recuperator and Heater HX data sets. For this series of tests, the sCO_2 flow was ~7 kg/min. The higher pressure sCO_2 stream was ~150 bar and the lower pressure sCO_2 stream was ~75 bar. The air temperature was raised from 200°C to 650°C in 50°C steps.

For the Recuperator, at lower energy loads, the heat absorbed from the colder, high pressure sCO_2 stream is almost the same as the heat given up by the hotter, lower pressure sCO_2 stream. At higher loads, the heat given up by the hot stream is slightly higher than the heat absorbed by the cold stream. This pattern is typical for HXs as some of the heat is lost to the environment through the surface of the HX and the surface of the piping between the instrumentation and the HX. The thermal images shown in Figure 9 provide an indication of heat lost by the Recuperator and Heater HX.

For the Recuperator data sets examined, the energy balance error was less than 5%. For the Heater HX, the energy balance error, between the air side and sCO_2, was between 5-10%. As with the recuperator, Heater HX heat loss increased with operating temperature.

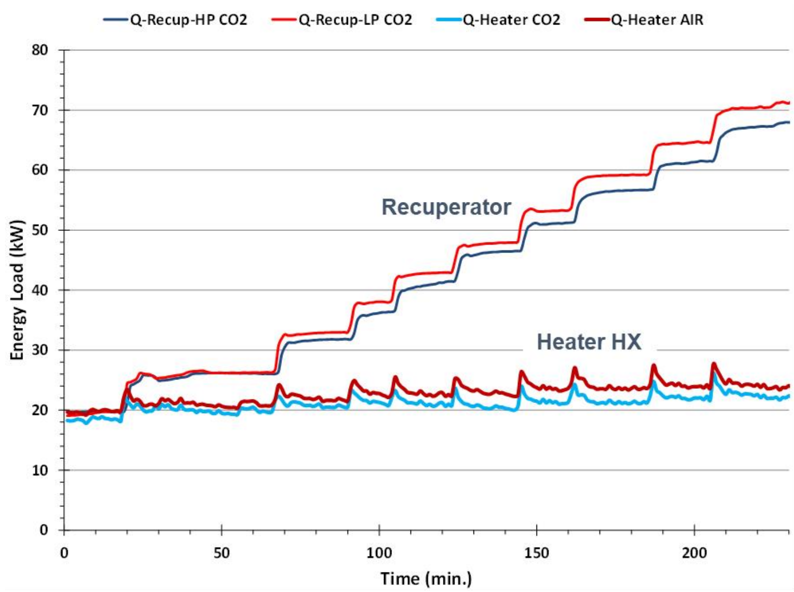


Figure 8. Recuperator and Heater HX energy balance plot.

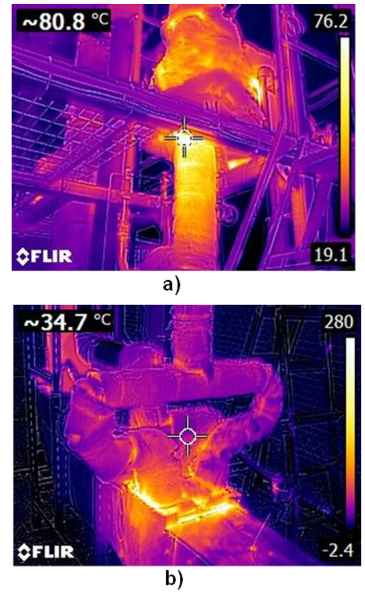


Figure 9. Thermal IR image. a) Recuperator b) Heater HX

To characterize the Recuperator and Heater HX, thermal models based on code written in Matlab, employed discretized calculations. This was done in part to compensate for large sCO_2 property changes that can occur with small variations in temperature and pressure.

A number of heat transfer models were evaluated. The models selected are well established, but based on experiments with tube sizes larger than 6 mm in diameter. As such, errors can be induced using these models with the smaller diameter microtube used in the compact HXs evaluated. Further investigations on how to modify these models for smaller diameter tubing is warranted.

On the CO_2 side, or internal microtube side, the Petukhov [9] equation was selected to calculate the Nusselt number and the Bhatti and Shah's model [10] was selected to calculate pressure drop. For the air side, the Martin model [11] was selected to calculate the Nusselt number and the Zukauskas model [12] was selected to calculate pressure drop.

Figures 10-11 provide a summary of the compact, counter-current, sCO_2-sCO_2, microtube recuperator performance. This Recuperator was designed to operate as a combined low temperature and high temperature recuperator. The design efficiency, in a fouled condition, is in the low 70% range.

Figure 10 shows how the inlet and outlet temperatures of the two recuperator sCO_2 streams, change with time, as the combustion gas/air temperature rises from 200°C to 650°C in 50°C steps. Also plotted in Figure 11, is the heat or energy transferred, represented by the green shaded area (right side y-axis). As the air temperature increases, so does the temperatures of low pressure sCO_2 entering the recuperator and the high pressure sCO_2 stream exiting the recuperator, and the energy transferred in the recuperator. Also, one can observe the close approach temperature of the low pressure sCO_2 exiting the recuperator and the high pressure sCO_2 stream entering the recuperator rises from less than 10°C to 15°C.

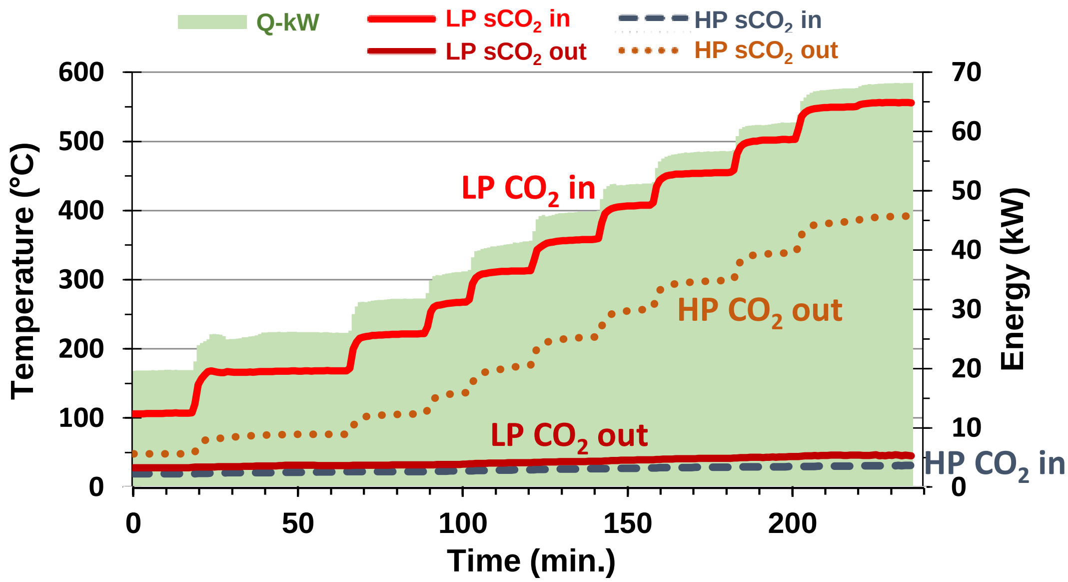
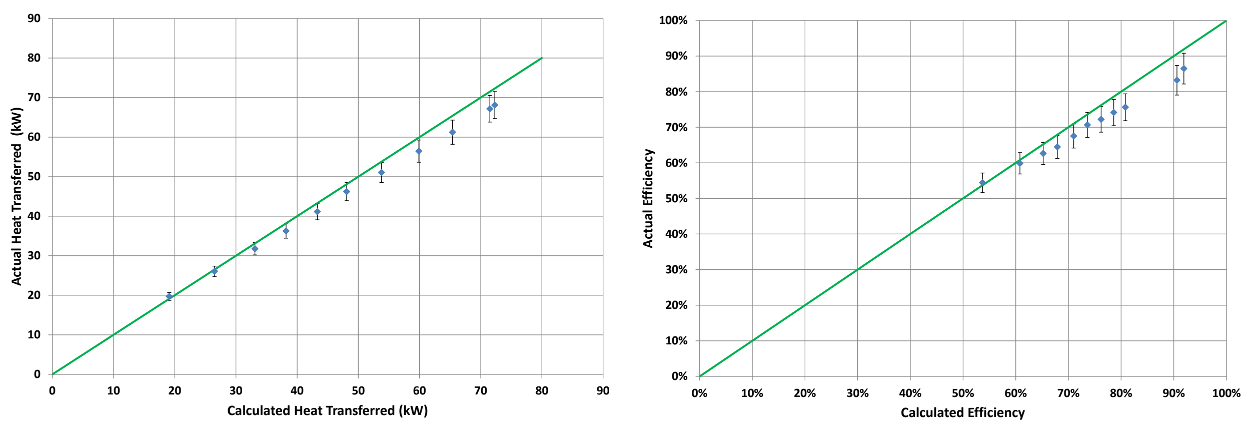


Figure 10. Recuperator temperature (left-axis) and energy (right-axis) vs. time plot.

Figure 11a compares the actual test data to the calculated model prediction for the total heat transfer. Figure 11b compares actual to calculated recuperator efficiency. In both examples there is a good linear comparison between the test and calculated data.

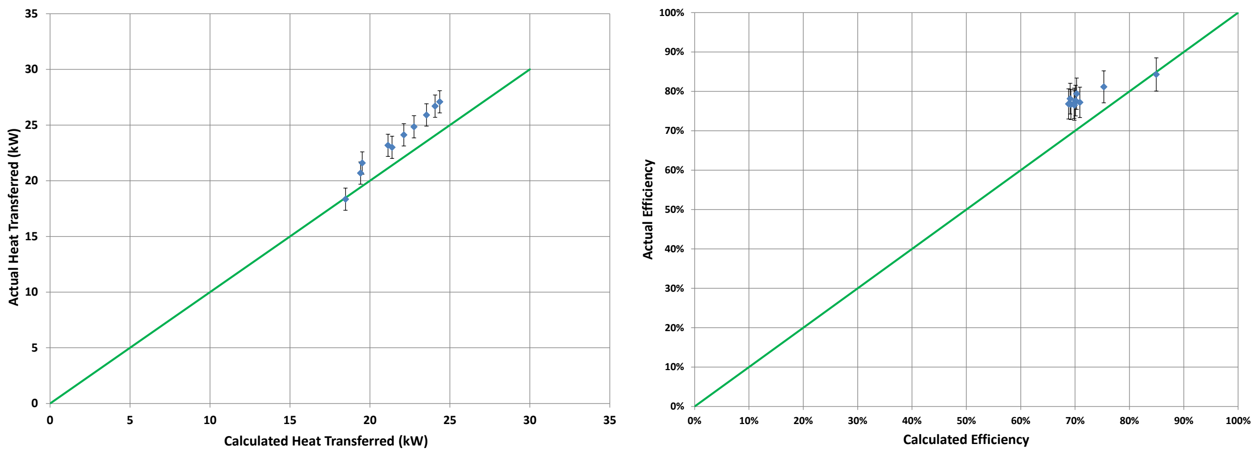


a) Recuperator total transferred heat **b) Recuperator efficiency**

Figure 11. Recuperator thermal performance – calculated vs. experimental data.

Figures 12 present a summary of the compact, cross-flow, counter-current, Air-sCO_2, microtube Heater HX performance. Figures 12a and 12b compare the actual and predicted energy transfer and HX efficiency. The plots indicate that the actual overall Heater HX performance is better than model predictions by ~10%. Heater HX efficiency or effectiveness, ranged between 75-85% which is promising for an Air-sCO_2 HX.

The graphs also indicate reasonable linearity with heat loss increasing with operating temperature. While the Heater HX is wrapped in two layers of insulation, thermal imaging, Figure 9, still shows signs of heat loss. From early test runs it was determined the insulation made a significant difference in obtaining a good energy balance. IR photos also showed pipe insulation butt joints had significant heat leaks and that joints could be improved with tapered overlaps on the piping.



a) Heater HX total transferred heat **b) Heater HX efficiency**

Figure 12. Heater HX thermal performance – calculated vs. experimental data.

CONCLUSIONS

The design and operation of the sCO_2, Brayton power cycle heat exchanger test facility has been reviewed along with initial operational lessons learned. Key lessons learned included: accommodating the rapid decrease in sCO_2 density upon heater start-up, making sure the sCO_2 density at the inlet to the pump was above 600 kg/m^3, and using appropriate fittings for the higher temperature connections. Also, additional layers of insulation would help to minimize heat loss.

The current system cannot detect a small leak until an energy balance has been performed. Having a mass flow detector, able to function at the higher temperature and pressure conditions would help to find leaks in real time.

Thermal performance data on two compact HXs (microtube Recuperator and Heater HX) have been reported. Overall, the test program has demonstrated that the predicted thermal performance of the compact microtube Recuperator and Primary Heater HX has been confirmed.

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