The 6th International Supercritical CO_2 Power Cycles Symposium
March 27 - 29, 2018, Pittsburgh, Pennsylvania

Commissioning of a 1 MWe Supercritical CO_2 Test Loop

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ABSTRACT

A new high-temperature expander turbine was developed for use of a sCO_2 closed-loop recompression Brayton cycle. This turbine was developed for Concentrating Solar Power (CSP) applications (700+ºC) with funding from the US DOE SunShot initiative and industry partners. The lower thermal mass and increased power density of the sCO_2 cycle, as compared to steam-based systems, enabled the development of compact, high-efficiency power blocks that can respond quickly to transient environmental changes and frequent start-up/shut-down operations. The power density of the turbine is significantly greater than traditional steam turbines and is rivaled only by liquid rocket engine turbo pumps, such as those used in the Space Shuttle Main Engines. This paper describes the commissioning and initial testing of the turbine verification of pressure containment, rotordynamics, thermal management, rotor aero and mechanical design, shaft-end and casing seals, bearings, and couplings.

INTRODUCTION

Southwest Research Institute® (SwRI®) along with its partnersdeveloped a novel, high-efficiency supercritical carbon dioxide (sCO_2) hot-gas turbo-expander optimized for the highly transient solar power plant duty cycle profile. Previous cycle demonstration loops are in the 100 kW scale with maximum temperatures of approximately 250°C (Kimball et al., 2012) [1]. This MW-scale sCO_2 turbo-expander design advances the state-of-the-art of sCO_2 turbo-expanders from a current technology readiness level (TRL) 3 (initial small-scale laboratory-size testing) to a full TRL6 (MW-scale prototype demonstration). A secondary objective of this project is to optimize novel printed-circuit heat exchangers for sCO_2 applications to reduce drastically their manufacturing costs. The sCO_2 turbo-expander and novel sCO_2 heat exchanger will be tested in a 1-MWe sCO_2 test loop, fabricated to demonstrate component performance and the performance of the optimized sCO_2 Brayton cycle over a wide range of part-load conditions and during transient operations representative of a typical CSP duty cycle.

The scalable sCO_2 expander design and improved heat exchanger address and close two critical technology gaps required for an optimized CSP sCO_2 power plant and provide a major stepping stone on the pathway to achieving CSP at \$0.06/kW-hr levelized cost of electricity (LCOE), increasing energy conversion efficiency to greater than 50%, and reducing total power block cost to below \$1,200/kW installed.

In 2011, the National Renewable Energy Laboratory (NREL) report sponsored by the Department of Energy (DOE) completed the evaluation of a sCO_2 cycle for CSP applications. The study concluded that the use of sCO_2 in a closed-loop recompression Brayton cycle offers equivalent or higher cycle efficiency when compared with supercritical- or superheated-steam cycles at temperatures relevant for CSP applications. The sCO_2 pressure is higher than superheated steam but lower than supercritical steam at temperatures of interest, making the use of sCO_2 in trough fields difficult. However, sCO_2 as a working fluid is well suited for the use in power towers. A single-phase process using sCO_2 as both heat-transfer and thermal-cycle fluid would simplify the power block machinery and is compatible with sensible-heat thermal energy storage. The study highlighted areas of uncertainty as to the high pressure required and the lack of experience with the closed-loop Brayton cycles, especially with the turbo-machines required.

The proposed CSP cycle uses sCO_2 as both the heat transfer fluid in the solar receiver and the working fluid in the power cycle. The lower thermal mass and increased power density of the sCO_2 power cycle, as compared to steam-based systems, enables the development of compact, high-efficiency power blocks that are compatible with sensible-heat thermal energy storage, and can respond quickly to transient environmental changes and frequent start-up/shutdown operations. These smaller, integrated power blocks are ideal for modular tower-mounted CSP solutions in the 5-10 MW range.

The team consists of SwRI; General Electric (GE); Thar Energy LLC (Thar); Bechtel Marine Propulsion Corporation, operator of Knolls Atomic Power Lab (KAPL); Aramco Services Co., and Electric Power Research Institute (EPRI). This MW-scale sCO_2 turbo-expander design advances state-of-the-art sCO_2 turbo-expanders from a current TRL3, initial small-scale laboratory-size testing, to a full TRL6 (MW-scale prototype demonstration). A secondary objective of this project is to optimize novel compact heat exchangers for sCO_2 applications to reduce drastically their manufacturing costs. The sCO_2 turbo-expander and novel sCO_2 heat exchanger were tested in a 1-MWe sCO_2 test loop, fabricated to demonstrate component performance over a wide range of part-load conditions and during transient operations representative of a typical CSP duty cycle.

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This is the first MWe-scale sCO_2 power cycle demonstration at 700°C temperatures. The turbo-expander developed for this demonstration will be optimized for the unique characteristics of CSP; quick start-up/shutdown and transient thermal inputs. The unique turbo-expander operating requirements, high pressure, high temperature, and supercritical working fluid are well beyond the current state-of-the-art in turbo-machinery design.

To accomplish the stated objectives, the work was divided into three phases that roughly emulate the development process from TRL3 to TRL6, including proof-of-concept, basic and detailed designs, engineering analysis, and prototype testing. Phase 1 was scheduled to last 24 months, during which time the turbo-expander and heat exchangers were designed, and all engineering analysis and modeling was conducted. By the end of Phase 1, a laboratory-scale prototype of the heat exchanger was tested and the designs of the heat exchanger, turbo-expander, and test loop were finalized. Phase 2 focused on fabrication and commission of the test loop and integration with the heat exchanger and turbo-expander. This phase was scheduled to last 24 months. Phase 3 will be dedicated to testing and will last 6 months. The performance and endurance of both main components will be documented to ensure they meet the operational requirements set during Phase 1. Based on the test results, the final design will be optimized to meet all related goals, such as cost and efficiency.

RESULTS AND DISCUSSION

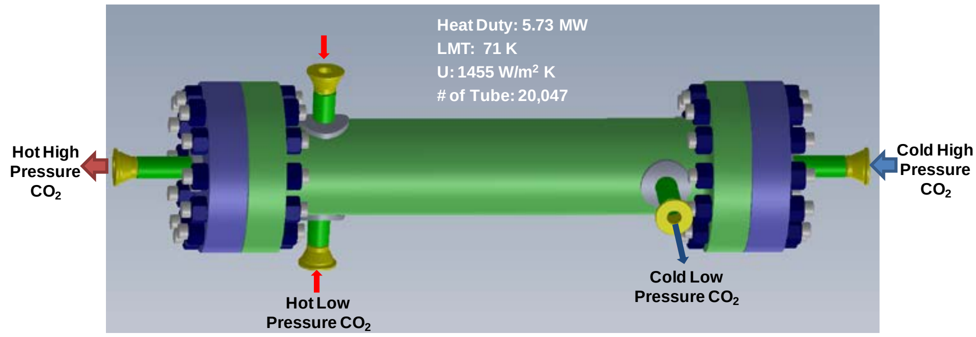
Heat Exchanger Fabrication

One of the tasks under the Sunshot program was development and fabrication of a 5-MWt recuperator led by Thar Energy. As a back-up, a second recuperator was procured from Vacuum Process Engineering, who designed and manufactured a printed circuit heat exchanger (PCHE). Below are the goals of the project and description of the development effort.

- Milestone: Delivery of 5-MWt compact recuperator and performance qualification testing to demonstrate that performance specifications are met.
- Completion Target Metrics:
    - Test unit size 50-kW scale up 100:1 for Phase 3.
    - Capacity (% of design) = Design goal is minimum of 80% of 35 MW/m^3 (i.e., 27 MW/m^3).
    - Pressure drop (% of design) < 1.5 times of bench-scale performance
    - Cost (% of design) < Goal is not more than 1.5 times $50/kW (i.e., no more than $75/kW)

The recuperator heat exchanger design was evaluated over the course of a number of design review meetings. The evaluation included mechanical, thermal, and flow distribution analyses. In addition, ASME and TEMA code calculations were reviewed for the recuperator shell, tube-sheet, seals, and clamps.

The recuperator shell and flanges were designed to meet ASME Section VIII, Division 1, where the pressure vessel is designed/rated to the highest pressure and highest temperature that will be seen anywhere within the heat exchanger shell and piping connections. Figure 1 shows a model of the shell design and the operating conditions. The shell (flanges and vessel body) is to be made of 316H stainless steel and is rated to the design conditions of 575°C @ 280 bar (1,053°F @ 4,116 psi). In addition, the pipe, pipe connections, all threads, and nuts are to be made of 316H stainless steel.



Low Pressure CO_2 (Hour-Glass)

- **9.910 kg/s**
- **T_{IN} = 567°C (1053°F)**
- **T_{OUT} = 80°C (175°F)**
- **P_{IN} = 86 Bar (1247 psi)**
- **ΔP = 1.5 Bar (22.4 psi)**

High Pressure CO_2 (Tubes)

- **8.347 kg/s**
- **T_{IN} = 29°C (85°F)**
- **T_{OUT} = 470°C (878°F)**
- **P_{IN} = 255 Bar (3698 psi)**
- **ΔP = 0.8 Bar (11.7 psi)**

Figure 1. Recuperator HX Configuration

The delta pressure across the HX was designed to meet the milestone of being less than 1.5 times the bench-scale performance. The capacity of the 20,000 micro-tube tube bundle is calculated at 89 MW/m^3, exceeding the design goal of 35 MW/m^3.

The pressure vessel fabrication was completed and passed the hydro test and received an ASME stamp, as shown in Figure 2. The tube bundle assembly was also completed, as shown in Figure 2. QA/QC of the tube bundle indicated that there was to be a delay in the delivery of the recuperator. This lead to a decision point to install a backup recuperator that had been ordered. Thar continues reviewing all stages in micro-tube recuperator fabrication in light of lessons learned from the fabrication of the first generation SunShot microtube bundle.

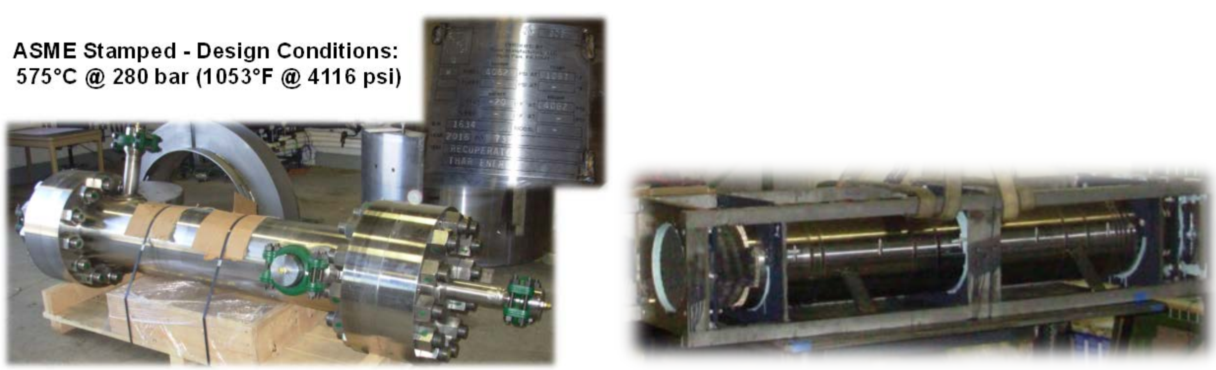


Figure 2. SunShot Recuperator Pressure Vessel with ASME Stamp

Recuperator Procurement

The alternative recuperator, manufactured by Vacuum Process Engineering (VPE), was installed in the test loop. This replacement, diffusion-bonded, micro-channel recuperator was completed and delivered to SwRI. The recuperator passed leak and hydro testing. Figure 3 shows the VPE recuperator on the stand in the SwRI Turbomachinery Research Facility Lab. Insulation was added to the recuperator.

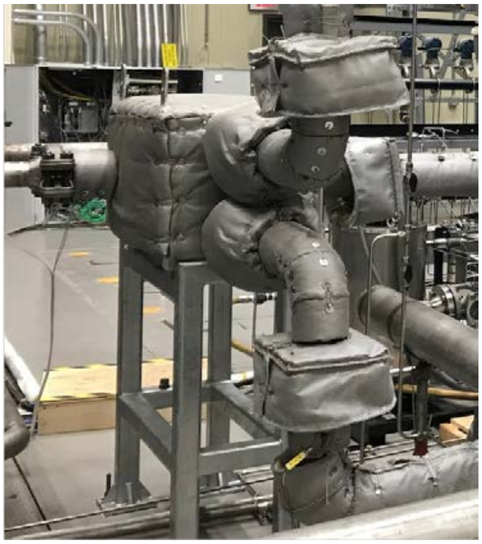


Figure 3. VPE Recuperator on the Stand in the SwRI Turbomachinery Research Facility Laboratory

Turbine Manufacturing and Assembly

SwRI has fabricated and assemble the complete turbo-machinery package, including skid, ancillary systems, and control system using in-house machine shops and contractors as necessary. The rest of the turbine parts were received, which included all of the casing components (inlet plenum, exit plenum, nozzle casing, and dry gas seal housing). With all of the parts in-house, an initial assembly was completed with both a high-pressure and low-pressure hydro test.

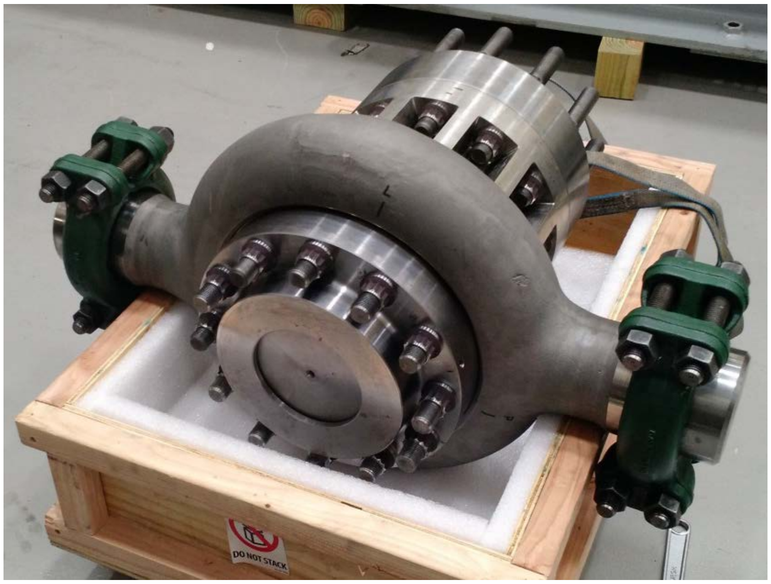


Figure 4. Assembly for High Pressure Hydro Test (9,000 psi)

Once the hydro testing was completed for all high-pressure piping and casings, he rebuilt turbine casing passed low-pressure and high-pressure hydro test. All field welds were completed at SwRI, and the SwRI weld shop passed a weld procedure to successfully weld INCO 740. All field welds passed 100% RT inspection and all required hydro testing. Once all the piping was fitted up, the turbine was disassembled, cleaned out, and prepared for final assembly with all internal components. The turbine was then assembled with the rotor, bearings, stator nozzles, dry gas seals, and all other internal turbine components. The turbine went together successfully and there was proper alignment for the critical spacing on the thrust bearings and the concentricity of the journal bearings. Below are some photos during the assembly:

With the turbine completed, the focus has shifted to final connecting of all the piping and instrumentation. It involves routing of all the dry gas seal lines, buffer air, and oil for the bearings. Testing of the buffer air and dry gas seals was performed after installation. After that, oil was slowly introduced into the system to ensure proper drainage and that oil is not creeping into the dry gas seals. After air and oil systems are tested successfully, instrumentation will be confirmed and the turbine will be slowly rolled around 1,000 rpm to test the rest of the system. For this test, the dyno wheel was not attached.

Figure 5. Turbine Assembly Showing Thrust Bearing Installation

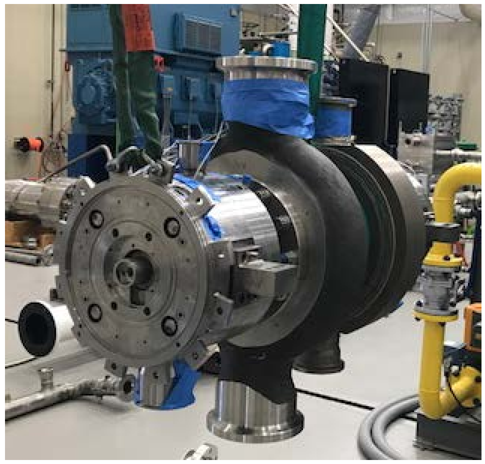


Figure 6. Fully Assembled Turbine

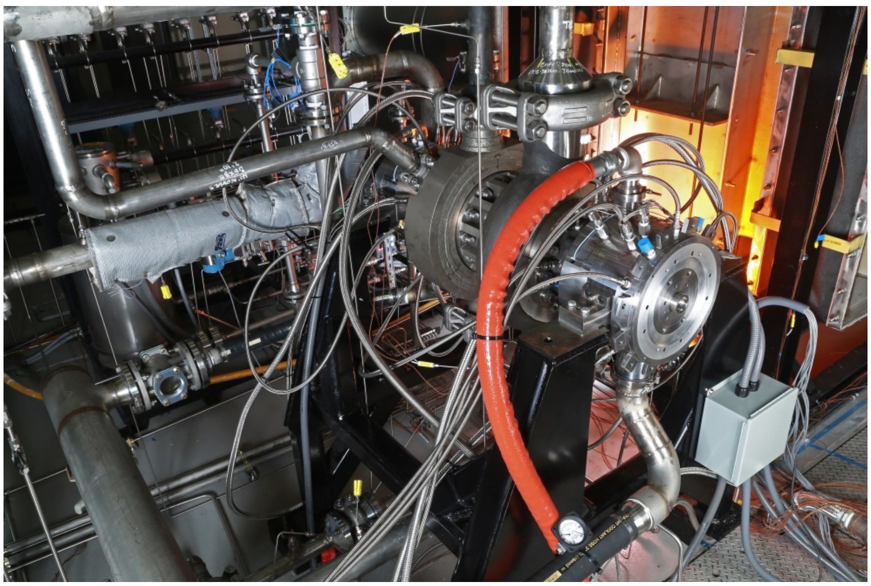


Figure 7. Assembled Turbine Casing on Operating Stand

With the turbine assembled, field welds were completed for the rest of the high-temperature piping. This included balance lines, piping to the recuperator, and piping from the heater and all instrumentation. All of the other small plumbing like dry gas seal supply and vent, buffer air supply, and lube oil supply and drain were connected to the turbine.

Test Loop Hardware Acquisition and Installation

The primary purpose of the SunShot test loop is to characterize the mechanical and aerodynamic performance of the recuperator and expander under development. Therefore, a simple recuperated cycle was chosen with a primary recuperator, an external heater to provide high temperature, and a separate pump to provide high-pressure CO_2 as shown in Figure 2. The simple cycle loop is less expensive and has less risk to implement. The turbine inlet conditions are identical to the recompression cycle. However, the single recuperator inlet conditions are different from the dual recuperator arrangement. The loop utilizes part of the existing CO_2 loop at SwRI including an existing shell-and-tube (wet) heat exchanger. More detail on the loop design can be found in Moore et al. (2014). [2]

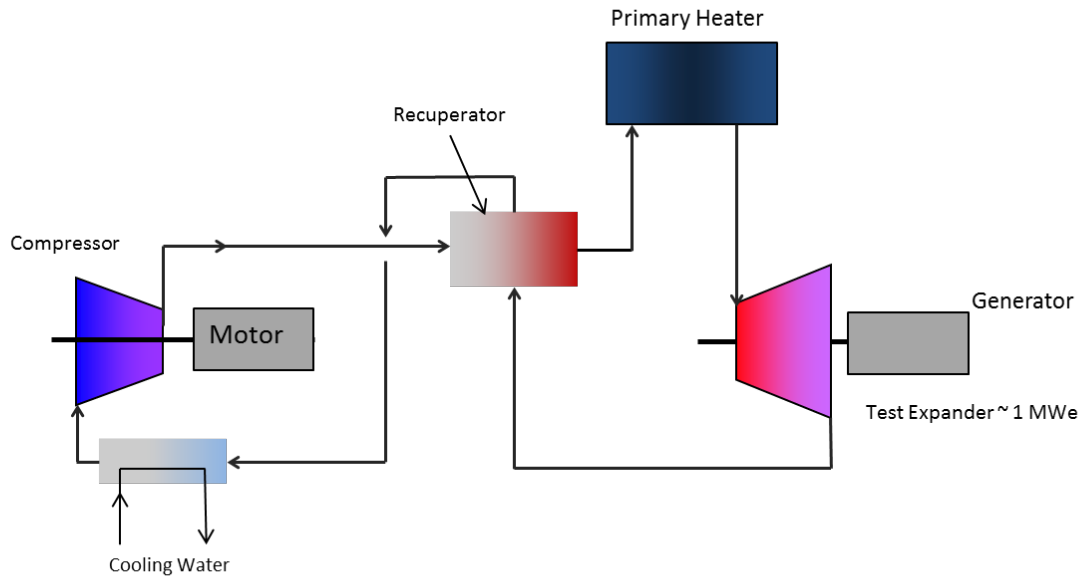


Figure 8. Simple sCO_2 Cycle for Test Loop

Primary Heater Overview

For the heat source, a custom heat CO_2-to-air heat exchanger was developed, which was mated to a commercial natural gas fired furnace. The heat exchanger was built using Inconel 740H tubing and piping and represents the first heat exchanger fabricated from this material. The heat exchanger was designed by SwRI and Thar Energy, and it was fabricated by Thar Energy. The overall intent of the heater for the SunShot test loop is to increase the CO_2 temperature exiting the recuperator discharge to the required temperature for the turbine inlet while not exceeding 1 bar of CO_2 pressure drop. These operating conditions are outlined in Table 1.

Table 1. Heater Operating Conditions

	Recuperator Outlet/ Heater Inlet	Heater Outlet/ Turbine Inlet
Temperature	470°C	715°C
Pressure	251.9 bar	250.9 bar
Mass flow rate of CO_2	8.410 kg/s	8.410 kg/s

These requirements were satisfied by using a natural-gas-fired heat exchanger consisting of a staggered array of tubes carrying the CO_2. The resultant tubes are arranged in an alternating pattern with half of them staggered as shown in Figure 9. This design allowed the tubes to be bent with weld joints only to the manifolds.

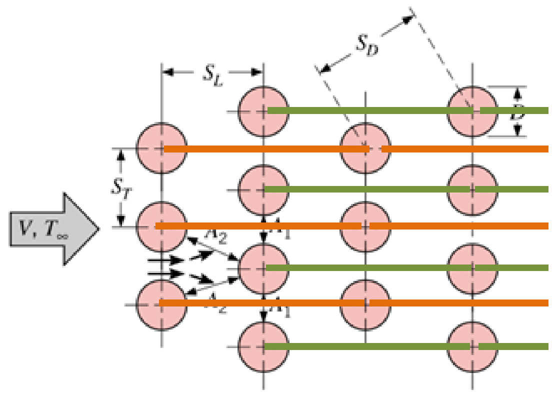


Figure 9. Heat Exchanger Tube Layout. Each Horizontal Row of Colored, Connectors represents a Single Tube. Different Colored Tubes will be fed from Different Headers

Figure 10 shows the completed heater heat exchanger using Inconel 740H tubing material. The tubing penetrations through the outer shell can be seen as well.

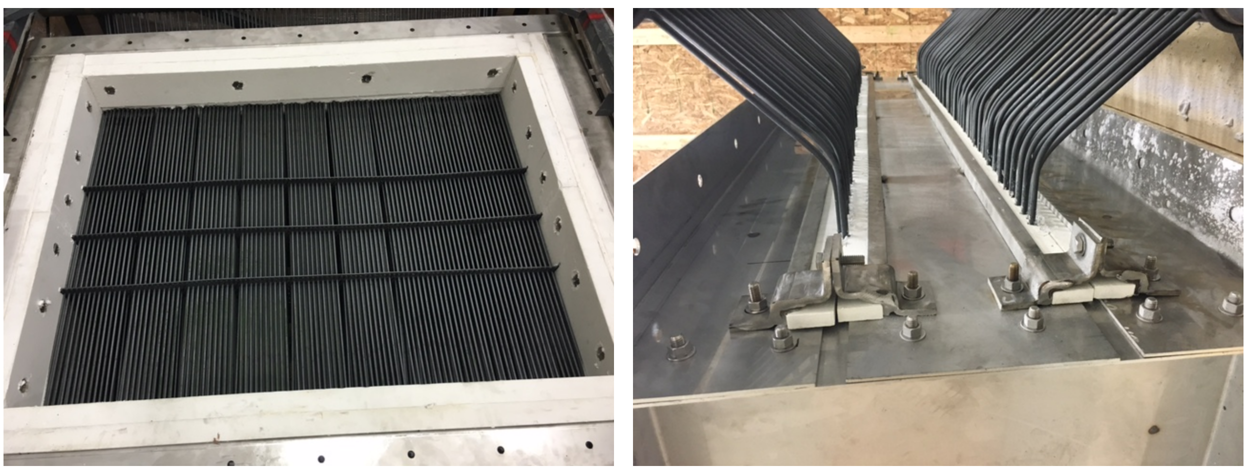


Figure 10. Heater Heat Exchanger

All of the piping was installed and the field joints were completed, including the Inconel 740 piping that connects the heater to the turbine-expander. Figure 3 shows the Sunshot test facility. The piping includes the connections from the pump (located outside the high bay) to the recuperator, from the recuperator to the heater, and from the recuperator back out of the high bay to the loop in the yard outside.

Figure 11. SunShot Piping in the High Bay Laboratory of Building 278

To provide the pressure rise in the loop, a dense-phase pump provided by Baker-Hughes General Electric was procured. This is a 12 stage back-to-back, 3600 rpm pump that requires that the CO_2 be below the critical temperature (< 20°C). Figure 12 shows the piping that connects the pump to the loop and the recuperator. This piping is complete and fully installed.

Figure 12. SunShot Piping Connecting the Pump to the Loop

To minimize heat loss and minimize risk to personnel, insulation has been added to the heater and pipe loop as shown in Figure 13.

Figure 13. SunShot Turbine Assembly with Insulation

Test Loop Commissioning

Table 2 lists the spinning tests that will be conducted as part of the commissioning and initial testing. The spinning tests will start by slow rolling the turbine and continue in progressively more advanced tests until all of the testing is completed. Items S1-S3 have been completed as part of the commissioning. Currently testing up to S7 conditions have been completed, including operation up to 550°C and 21,000 rpm. Table 3 lists the turbine operating limits that will be respected while the turbine is operating. Operating limits for the test-loop conditions are listed in Table 4 along with appropriate actions if these limits are reached.

Table 2. Summary of Spinning Tests

#	Test Name	Test Description	Dyno	Heater	Speed	TIT	TEP	Exit Criteria
S1	Slow roll, init. seal break-in, and rotordyn. instrum. check	Spin turbine up to 2,000 rpm and immediately shut down. Confirm that tachometer was triggering and proximity probes were functioning properly. Repeat spin up and shut down procedure with increasing target speeds in approximately 1,000-2,000 rpm increments up to 5,000 rpm. This will allow abradable seals to safely cut-in during transients in case there is rubbing.	Impeller removed.	Off	5,000 rpm	47°C (or max pump discharge)	80 bar	Observe all rotordynamics instrumentation functioning properly.
S2	ESD and overspeed trip check	Spin turbine up to 5,000 rpm and activate the ESD switch to confirm its function. Set overspeed trip value to 4,000 rpm. Attempt to spin turbine up to 5,000 rpm and confirm that the overspeed trip functions properly.	Impeller removed.	Off	5,000 rpm	47°C (or max pump discharge)	80 bar	Observe ESD and overspeed trip sequences are functioning properly.
S3	Seal break-in, trim balance	Spin turbine up to target speed and immediately shut down. This will allow abradable seals to safely cut-in during transients in case there is rubbing. Execute trim balancing as necessary to reduce vibration levels below limit. Increase target speed from 5,000 rpm in approximately 1,000-3,000 rpm increments up to 18,000 rpm. Capture coast down Bode plot from maximum speed.	Impeller removed.	Off	18,000 rpm	47°C (or max pump discharge)	80 bar	Observe normal vibration within amplitude limits and without signs of rubbing across speed range. Confirm bearing and SFD performance. Confirm rotordynamics model prediction of unbalance response *without dyno impeller*; tune model if necessary.
S4	Cold test, limited speed	Operate up to 21,000 rpm. Achieve steady state conditions. Capture coast down Bode plot from maximum speed.	Impeller removed.	Off	21,000 rpm	47°C (or max pump discharge)	80 bar	Successfully operate at 1 steady-state condition.
S5	Dyno impeller break-in	Spin turbine up to target speed and immediately shut down. Execute trim balancing as necessary to reduce vibration levels below limit. Increase target speed in approximately 1,000-3,000 rpm increments up to maximum speed (power-limited without heating turbine flow). Capture coast down Bode plot from maximum speed.	Impeller attached; adjust flow control valves for min power (within safe operating range).	Off	12,000 rpm (or maximum power-limited speed)	47°C (or max pump discharge)	80 bar	Observe normal vibration within amplitude limits and without signs of rubbing across speed range. Confirm rotordynamics model prediction of unbalance response *with dyno impeller*; tune model if necessary.
S6	Warm test, limited speed	Operate up to 21,000 rpm (or maximum power-limited speed) with heater off. Fire heater to increase TIT to 150°C (rate < 5°C/min), verify thermal seal performance, and make appropriate adjustments to DGS cooling flow. Slowly increase TIT to 550°C (rate < 5°C/min), modify cooling flow as necessary. Obtain 5 steady-state operating points from min to max dyno flow; adjust TIP to maintain speed, TIT, and TEP.	Impeller attached; adjust dyno flow from min to max (or within safe operating range) for 5 steady-state points.	Fired	21,000 rpm	550°C	80 bar	Successfully operate at 5 steady-state conditions. Document transient and steady state performance for all components in the loop. Document required DGS flow rates.
S7	Warm test, full speed	*This test may be a continuation of a successful <u>limited speed warm test</u> without shutdown (if not, refer to previous procedures).* Increase speed to 27,000 rpm. Achieve steady state conditions. Obtain 5 steady-state operating points from min to max dyno flow; adjust TIP to maintain speed, TIT, and TEP.	Impeller attached; adjust dyno flow from min to max (or within safe operating range) for 5 steady-state points.	Fired	21,000 rpm (or maximum power-limited speed)	550°C	80 bar	Successfully operate at 5 steady-state conditions. Document transient and steady state performance for all components in the loop.
S8	Hot test, full speed	*This test may be a continuation of a successful <u>full speed warm test</u> without shutdown (if not, refer to previous procedures).* Slowly increase TIT to 715°C (rate < 5°C/min). Achieve steady state conditions. Obtain 5 steady-state operating points from min to max dyno flow; adjust TIP to maintain speed, TIT, and TEP.	Impeller attached; adjust dyno flow from min to max (or within safe operating range) for 5 steady-state points.	Fired	27,000 rpm	715°C	80 bar	Successfully operate at 5 steady-state conditions. Document transient and steady state performance for all components in the loop.
S9	Normal shutdown from max TIT	*This test may be a continuation of a successful <u>full speed hot test</u> without shutdown (if not, refer to previous procedures).* Slowly decrease TIT to 200°C (rate < 5°C/min). Turn off burner, but maintain air blower flow. Reduce speed to 5,000 rpm. After heater air exit temperature reaches 150°C, stop turbine and pump.	Impeller attached.	Fired	27,000 rpm	715°C	80 bar	Successfully shut down. Document transient performance for all components in the loop.

TIT = Turbine inlet temperature
TIP = Turbine inlet pressure
TEP = Turbine exhaust pressure
DGS = Dry gas seal
SFD = Squeeze film damper
ESD = Emergency shut down

Table 3. Summary of Turbine Operating Limits

Measured Quantity	Limit	Action if Limit Exceeded
Synchronous (1x) vibration factory acceptance with proximity probes	0.7 mil p-p (ISO 7919 Zone A)	Reduce speed or change load condition to try to eliminate high vibration, or shut down. Evaluate possible causes for high vibrations and correct (e.g. trim balance)
Subsynchronous (*n*x, *n*<1) vibrations measured with proximity probes	0.18 mil p-p	Reduce speed or change load condition to try to eliminate subsynchronous vibration presence, or shut down. Evaluate possible causes for subsynchronous vibrations and correct, if possible, or justify setting higher limit.
Overall vibration Amplitude measured with proximity probes	1.5	Alarm – Reduce speed or change load condition to try to eliminate high vibration, or shut down. Evaluate possible causes for high vibrations and correct, if possible, or justify setting higher limit.
	1.8 mil p-p *Note: Based on experience chart.*	Shut down – determine the cause for the high vibration and correct it or justify setting a higher limit.
Housing acceleration measured with accelerometers	0.25 in/s RMS velocity (factory acceptance).	Reduce speed or change load condition to try to reduce acceleration amplitude, or shut down. Evaluate possible causes for high acceleration amplitude and correct, if possible, or justify setting higher limit.
Oil supply temperature	130 °F	Evaluate lube oil cooler and reduce supply temperature
Oil drain temperature	50 °F above oil inlet temperature.	Shut down and determine the cause for the high oil temperature rise and correct it or justify setting a higher limit.
DGS Casing temperature	350 °F	Decrease turbine TIT or increase seal gas flow rate.
DGS pressure	1500 psi	Decrease loop pressure.
Casing temperature?	1320 °F (715°C)	Reduce burner temperature
Casing temperature rate	670°C/hr, increasing	Turn down heater temperature or flow.
	670°C/hr, decreasing	Turn up heater temperature or flow.
Heater air temperature?	1700 °F	Reduce fuel flow rate.

Table 4. Summary of Loop Operating Limits

Measured Quantity	Limit	Action if Limit Exceeded
Loop low pressure limit	1600 psi (PSV set point)	Vent loop
Loop high pressure limit	3950 psi (turbine case rating)	Vent loop
Turbine inlet temperature limit	1320 °F	Reduce heater fuel flow
Turbine exhaust temperature limit	1270 °F	Reduce heater fuel flow
Carbon steel pipe temperature limit	600 °F (900# ANSI Flange Limit)	Reduce heater fuel flow
Precooler inlet temperature limit	500 °F	Reduce heater fuel flow

Initial Testing

Initial testing of the turbine has begun, including break-in of the abradable seals. The rotor has reached a speed of 21,000 rpm at a turbine inlet temperature of 550°C. The rotor exhibited low vibrations, good thrust balance, and bearing temperatures well within limits. Figure 14 is a test data screen capture taken during the recent testing while reaching this condition. It shows an inlet pressure to the turbine of 2,200 psi at a speed of 16,600 rpm. Vibrations on the turbine were low. This test completes the commissioning activities of the turbine and test loop. Future testing will include performance testing, with gradually increasing speed and temperature to a maximum value of 27,000 rpm and 715°C, respectively.

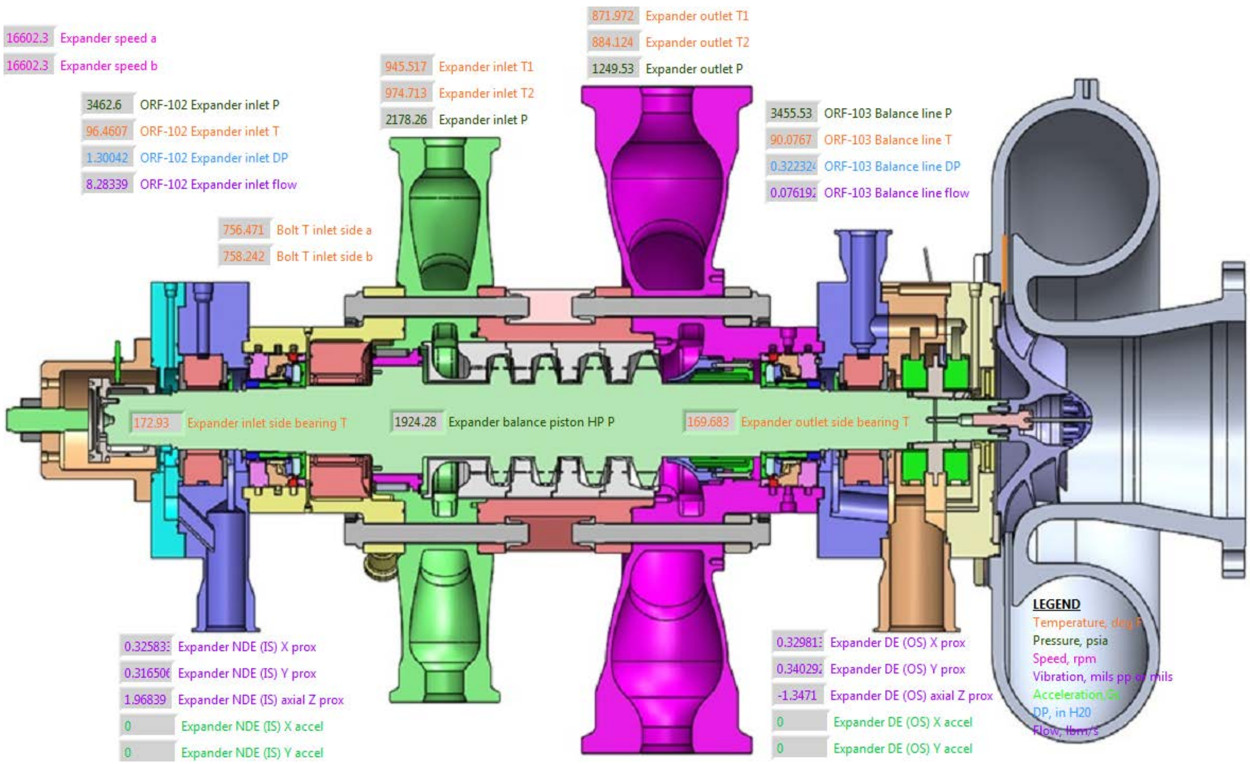


Figure 14. **Screen Capture of Data Acquisition System during Initial Turbine Testing**

CONCLUSIONS

This paper outlines the assembly and commission of a 1 MWe sCO_2 test loop consisting of a novel 1 MWe turbine expander (10 MWe frame size), a novel air-to-CO_2 heat exchanger made of Inconel 740H, and a commercially source PCHE recuperator. The test loop components met all applicable codes (hydro test, weld procedures, etc.) and all leaks were addressed. Extensive instrumentation was installed to monitor critical temperatures, pressure, flow, and vibration and a micro-processor controller monitors all critical channels. Initial commissioning steps included breaking in the turbine (uses abradable seals) followed by mechanical testing to monitor vibrations, critical speeds, and bearing temperatures. To date the turbine has achieved its initial test point of 550°C at 21,000 rpm. Continued testing will seek to reach the maximum design conditions of 715°C, 250 bar turbine inlet conditions at 27,000 rpm.

REFERENCES

[1] Kimball, K.J and E.M. Clementoni, "Supercritical Carbon Dioxide Brayton Power Cycle Development Overview," Proceedings of the ASME Turbo Expo, Copenhagen, Denmark, June 11-15, 2012.

[2] Moore, J.J., Evans, N., Brun, K., Kalra, C., 2015, Development of 1 MWe Supercritical CO_2 Test Loop, Proceedings of the ASME Turbo Expo, Paper #GT2015-43771, June 15-19, 2015, Montreal, Quebec, Canada.

ACKNOWLEDGEMENTS

The authors would like to thank the Office of Energy Efficiency and Renewable Energy (EERE) within the U.S. Department of Energy, General Electric, Thar Energy, Aramco Services Co., Bechtel Marine, and EPRI for providing guidance and funding for this research.