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TESTING OF A 10 MWe SUPERCRITICAL CO_2 TURBINE

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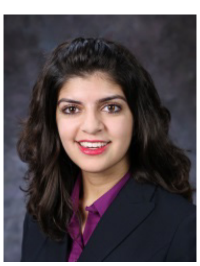
Mr. Stefan Cich is a research engineer and has been at SwRI for over 5 years. During his time, he has gained extensive experience in sCO_2 technologies. He has worked on the design and development of a 10 MWe turbine and power loop, a 5 MW compressor, heaters and recuperators, and the conceptual design of a 450 MW turbine. In addition, he has gained knowledge in superalloys and other materials that perform well in sCO_2 power cycles. This has also allowed him to gain much experience in the manufacturing of complex castings, forgings, and other technologies that can be used to develop new generation machinery.



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Dr. Douglas Hofer is currently Senior Principal Engineer in the thermal sciences organization at GE Global Research in Niskayuna, New York. His research interests are in the areas of turbomachinery aero-thermal fluid dynamics, advanced expander and compressor technologies, highly unsteady flows, two-phase flows, transonic and supersonic flows. He has deep experience in the steam turbine industry in both turbomachinery design and cycle analysis and innovation.

Mr. Jason Mortzheim is a senior mechanical engineer at the GE Global Research Center in Niskayuna, NY. He has a B.S. in aeronautical engineering from Rensselaer Polytechnic Institute and a M.S. in aerospace engineering from Georgia Institute of Technology. His area of research has been centered on improving fossil power combined cycle plant efficiency including both steam and gas turbines. Mr. Mortzheim initial area of research focused around advanced seal designs where he has held various roles spanning his 19-year professional career. He has utilized computational methods, both commercial CFD and in-house flow solvers, advanced experimental campaigns including industry leading test rig designs, complete to post commercial operation validation to develop state-of-the-art seal technology ranging from brush seals to film riding seals. Mr. Mortzheim is currently the Principal Investigator for DOE contract, EE0007109 related to advanced sCO_2 power cycle compression technology and the GE Principal Investigator supporting DOE contract, FE0028979, 10 MWe sCO_2 Pilot Plant Test Facility.

ABSTRACT

A new high temperature turbine was developed for use of an 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, but its application includes traditional heat sources such as natural gas, coal, and nuclear power. Traditional Rankine steam cycles exhibit thermal efficiencies in the 35-40% range (as high as 45% for advanced ultra-supercritical steam cycles). The sCO_2 cycle can approach 50% thermal efficiency using externally fired heat sources. Furthermore, this cycle is also well suited for bottoming cycle waste heat recovery applications. The lower thermal mass and increased power density of the sCO_2 cycle, as compared to steam-based systems, enables the development of compact, high-efficiency power blocks that can respond quickly to transient environmental changes and frequent start-up/shut-down operations, a particular advantage for solar, waste heat, and ship-board applications. 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 on the Space Shuttle Main Engines. This paper describes the design, commissioning, and initial testing of the 10 MWe turbine in a 1 MWe test facility including description of pressure containment, rotordynamics, thermal management, rotor aero and mechanical design, shaft-end and casing seals, bearings, and couplings.

INTRODUCTION

The author's company along with its partners developed a novel, high-efficiency supercritical carbon dioxide (sCO_2) hot-gas turbine 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). This 10 MW-scale sCO_2 turbine design advances the state-of-the-art of sCO_2 turbines 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 turbine and novel sCO_2 heat exchanger was 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. Since the turbine is being tested in a reduced flow facility, a flow-cut of the turbine flow path was made (to approximately 10%) in order to achieve similar velocities and pressure distribution as the full 10 MWe design. Therefore, the rotordynamics, thrust balance, thermal management and mechanical operation (of bearings, seals, etc.) will be similar to the full power design. This approach permitted a turbine design to be developed with industrial scale bearings, seals, and couplings and shortened the design cycle of the full 10 MWe design.

The scalable sCO_2 turbine 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 (2013 dollars).

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 (Kalra, 2014). 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.

Combs (1977) was the first to propose use of sCO_2 as a working fluid in a closed-loop recompression Brayton cycle for shipboard applications. Combs concluded that a substantial reduction in fuel consumption was possible. More recently, sCO_2 cycle testing has been performed at Knolls Atomic Power Laboratory (DOE, 2012; Kimball, 2012) and Sandia National Laboratories (Wright, 2010; Kolb, 2011; Conboy, 2012). Both of these turbines use a radial inflow turbine with diameter around 50 mm along with gas bearings. The data from these tests indicate that the basic design for the recompression cycle is feasible. However, the turbo-machinery layout used is not scalable multi-MW class applications like high efficiency CSP and utility scale power generation applications (Wright, 2010; Kolb, 2011).

This paper focuses on the turbine development and testing. The sCO_2 turbine 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. The size sCO_2 Brayton cycle is being designed to match current modular solar fields (10 MW) and has been identified as being commercially competitive (Ma, 2011). The oil and gas industry has developed technologies for pumping and compressing CO_2 at supercritical pressures for other applications, and hence, compression technology required for the sCO_2 Brayton cycle is considered a moderate risk (Moore, 2007). In contrast, industrial scale turbines for operation on sCO_2 do not have a precedent in the industry beyond some small demonstration radial turbomachinery units currently being run in labs (Kalra, 2014).

The current work is the first MWe-scale sCO_2 power cycle demonstration at 700ºC temperatures and has been tested to the highest temperature using sCO_2 to date. The turbine developed for this demonstration was optimized for the unique characteristics of CSP requiring rapid start-up/shut-down and transient thermal inputs. The unique turbine operating requirements, high pressure, high temperature, and supercritical working fluid are well beyond the current state-of-the-art in turbo-machinery design.

The project 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 turbine and heat exchangers were designed, and all engineering analysis and modeling was conducted. By the end of Phase I, a laboratory-scale prototype of the heat exchanger was tested, and the designs of the heat exchanger, turbine, and test loop were finalized. Phase II focused on fabrication and commission of the test loop and integration with the heat exchanger and turbine. Phase III was dedicated to testing of the facility. The performance and endurance of both main components will be documented to ensure they meet the operational requirements set during Phase I. Based on the test results, the final design will be optimized to meet all related goals, such as cost and efficiency.

TURBINE DESIGN

A custom design turbine was developed in Phase I of the project. The overall thermal design requirements were established by the thermodynamic cycle for a 10 MWe concentrated solar power plant, where the turbine output is approximately 14 MW (4 MW used to drive the compressor). In addition to the cycle requirements, mechanical design analysis was performed on the aero designs providing feedback on the levels of static stress due to rotation and gas bending. This resulted in the need to minimize the blade heights and tip diameters. Additionally, the blade dynamics were evaluated to avoid resonance, especially important given the high gas density of sCO_2 thereby minimizing the risk of blade failure. Throughout the conceptual design phase of the turbine several flow-path layouts were considered. Once the design configuration was down-selected to a multi-stage axial design, a study of several designs with 3 and 4 stage counts was conducted for mechanical shaft feasibility. To reduce mechanical stresses and improve efficiency a 4 stage design at 27,000 RPM was chosen, and the 3D airfoil designs are shown in Figure 1. Note the colored by axial position strictly for visualization purposes.

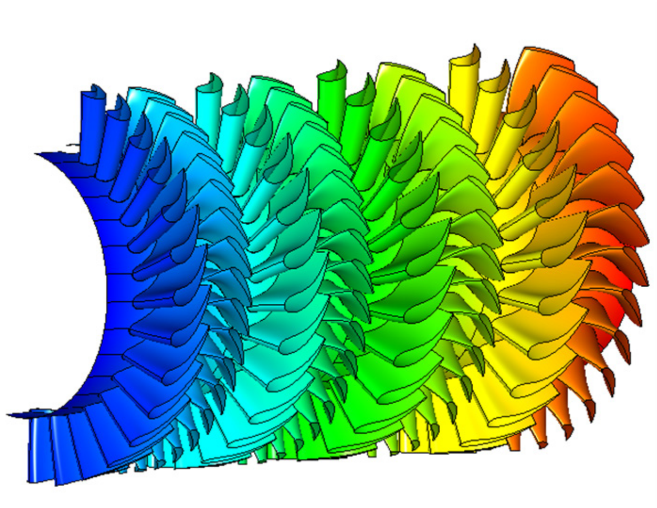

Figure 1: Airfoil designs for 4-stage SunShot turbine (Kalra, et. al, 2014)

Figure 2 shows the layout of the 4-stage axial flow turbine rotor. Key mechanical components are also identified including the balance piston (balances axial thrust), thermal management (controls temperature gradient), dry gas seal, radial and thrust bearings, and coupling attachment. A thorough rotordynamic analysis was performed to ensure good critical speed response and stability. One of the more challenging areas in the design of the turbine is in the thermal management region. Figure 3 shows the predicted temperature profile through the shaft. A steep temperature gradient exists in the thermal management region and is the results of careful management of cooling flow fed to the dry gas seals and through this region in order to maintain acceptable temperatures near the dry gas seal (350º or 177º).

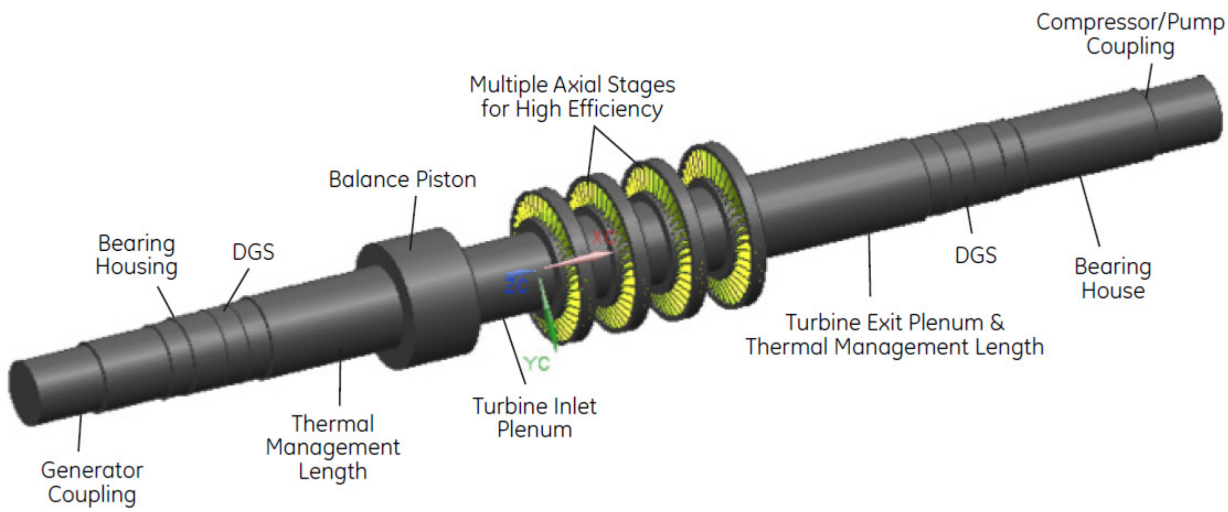

Figure 2: SunShot sCO_2 turbine rotor design (Kalra, et. al, 2014)

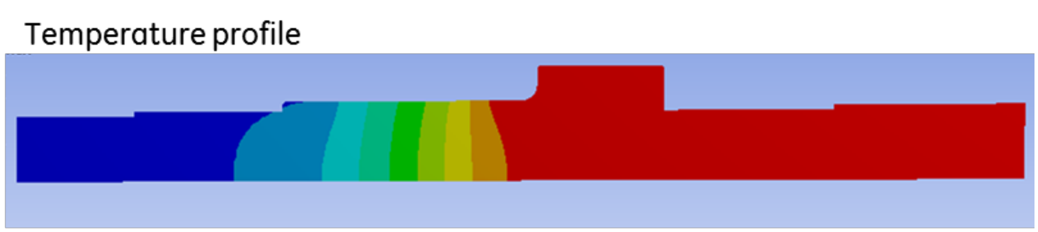

Figure 3: Temperature profile in the shaft and stator piece in the thermal management region (Blue = 50ºC, Red = 715ºC) (Kalra, et. al, 2014)

The following design metrics were satisfied as described by Kalra, et al., (2014).

- Pressure containment to ASME and API standards
- Lateral and torsional rotor-dynamics established per API and industry standards
- Rotor over-speed capability with 20% margin to all conditions anticipated in test loop
    - Thrust bearing and balance piston design expected to meet API 617 standard for anticipated operating conditions in test loop
- Creep and low cycle fatigue life sufficient for expected continuous and transient operation
    - Axial and radial clearances set to meet API 617 at all steady state and transient operating conditions
- Turbine isentropic efficiency greater than 85% based on mean line and CFD analysis (based on 10 MW design).

Figure 4 shows the resulting turbine cross-section. An axial stack of the casing was utilized to permit either casing or fabrication of the casing components. The internal components are labeled including the turbine nozzles with integral swirl-brakes for the turbine tip seals, hole pattern balance piston damper seal, and integral squeeze film damper (ISFD) bearings. These bearing dampers were tested by (Ertas, et. al, 2017) to verify the damping predictions. The turbine inlet and center spool casing components were fabricated from forgings using Inconel™ 625 material while the exit plenum was cast using stainless steel. They were designed in accordance with ASME Section VIII, Division 2 and used plated metallic C-seals at the flange joints. The dry gas seals employed pressure and spring energized polymer seals. To represent the 10 MW coupling, a dummy mass was added to the shaft. An air dynamometer (not shown) attaches to the 4 MW end of the turbine eliminating the need for couplings, gearboxes, and generators simplifying the packaging. Figure 5 shows the predicted unbalance response for mid-span unbalance (4 times API unbalance level) using minimum bearing clearance. Even this worst case shows a low amplification factor and response amplitude through the first critical speed and throughout the operating speed range. Figure 6 shows a rotordynamic experience plot showing how the SunShot turbine compares with other centrifugal compressor experience taken from Moore, et al., (2006). While near the boundary, it is within the experience of this manufacturer for centrifugal compressors. This chart was used since steam and gas turbines typically operate at significantly lower pressures and density. Stability enhancements including the hole pattern balance piston seal, swirl brakes at the turbine blade tip seals, and ISFD bearings were utilized.

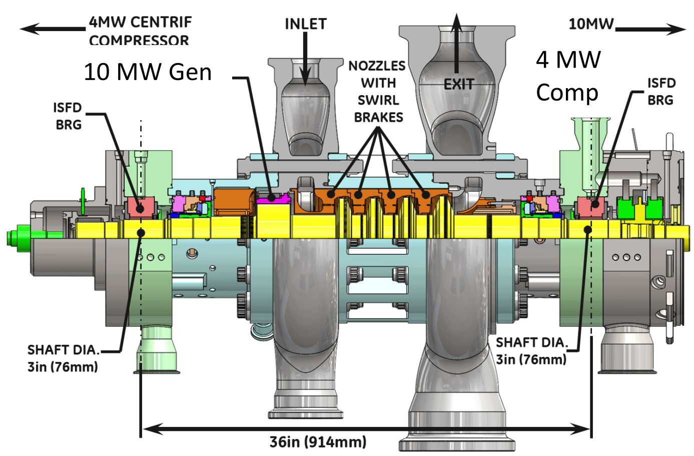

Figure 4: SunShot sCO_2 10 MWe 27krpm Turbine (Ertas, et. al, 2017)

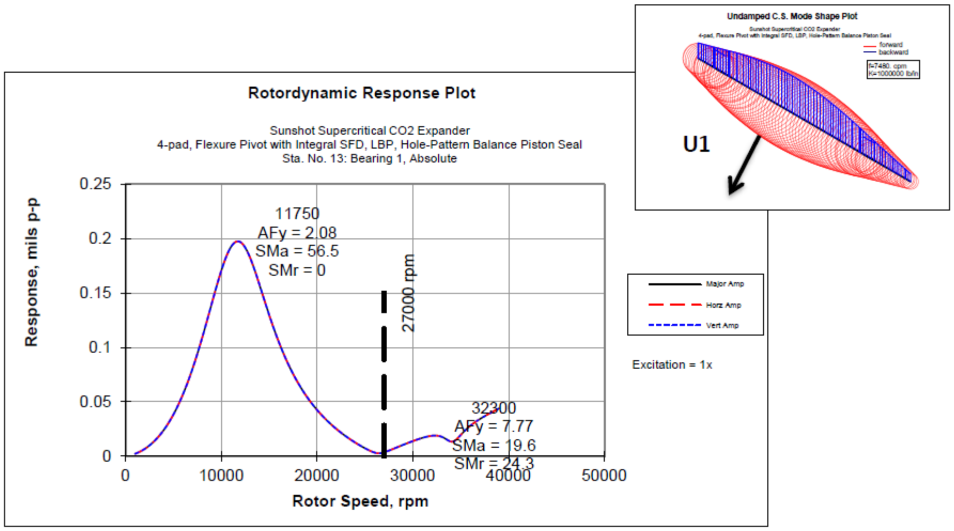

Figure 5: Rotordynamic Prediction for First Critical Speed (Min. Bearing Clearance with Mid-Span Unbalance)

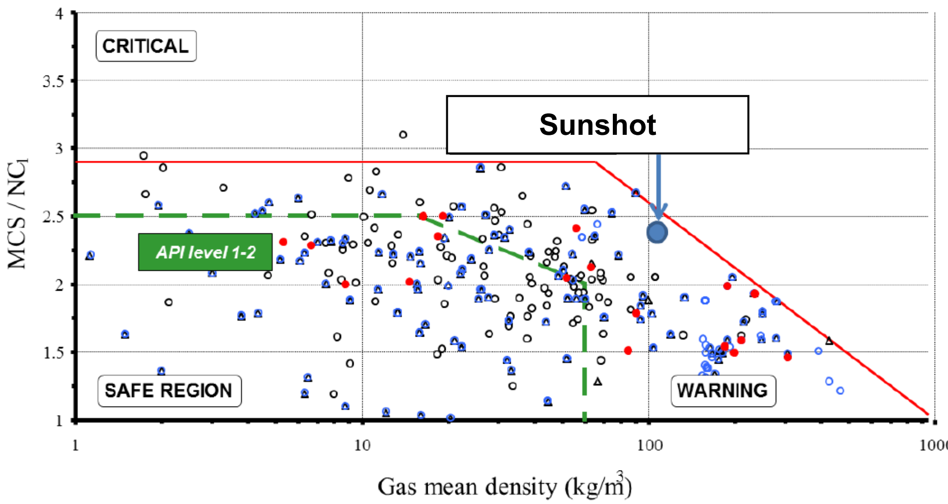

Figure 6: Rotordynamic Experience Chart from Moore (2006) with SunShot Turbine Rotor Added

HEAT EXCHANGER FABRICATION

A 5-MWt recuperator was procured from a heat exchanger vendor, who designed and manufactured a printed circuit heat exchanger (PCHE). This heat exchanger was a diffusion-bonded, micro-channel recuperator, which was completed and delivered after passing leak and hydro testing. Figure 7 shows the VPE recuperator on the stand in the author's laboratory. Grayloc^{TM} flanges were employed through the hot-section due to their high temperature and pressure capability. Insulation was added to the recuperator to minimize heat loss.

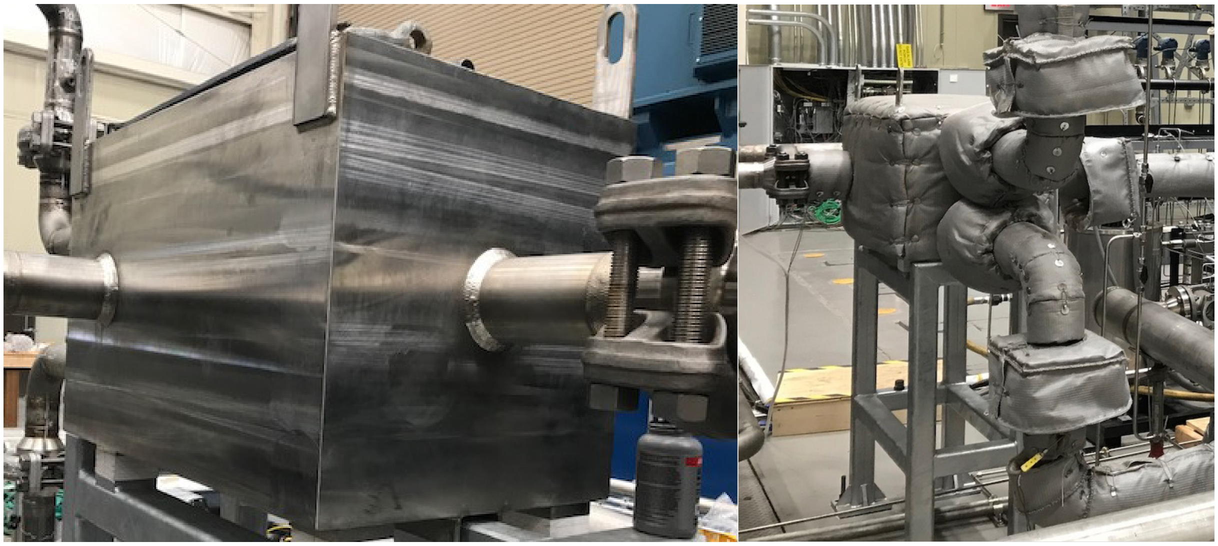

Figure 7: Recuperator in the Test Loop

TURBINE MANUFACTURING AND ASSEMBLY

The author's company fabricated and assembled 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 8 shows the assembly of the turbine inlet casing during hydro testing.

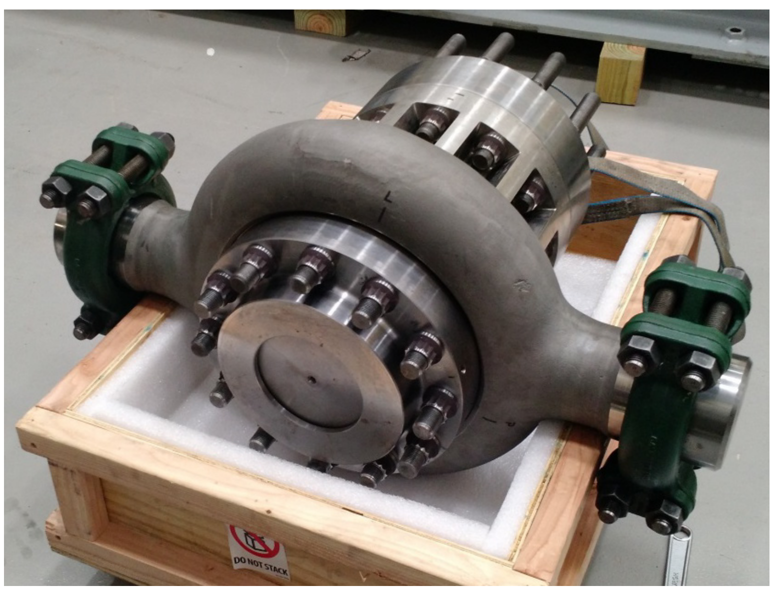

Figure 8: Assembly for High Pressure Hydro Test (9,000 psi)

The turbine casing was assembled (no internals) and placed on the test stand to complete all field welds, including the main inlet and exit pipes as well as the balance piston return lines. The INCO 740H inlet piping requires a special weld procedure and post-weld heat treat. All field welds passed 100% radiographic testing (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. Figure 9 shows the thrust bearing being installed after shimming for proper rotor endplay. Figure 10 shows the fully assembled turbine being moved to the test stand.

Figure 9: Turbine Assembly Showing Thrust Bearing Installation

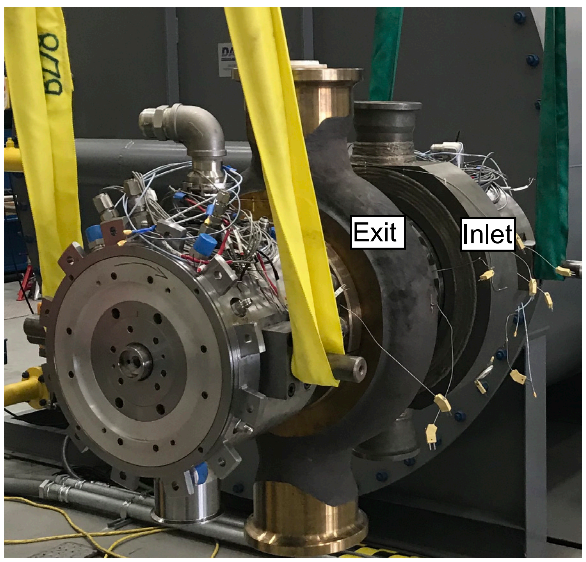

Figure 10: Fully Assembled Turbine

With the turbine assembly completed, the focus shifted to final connection of all the piping and instrumentation including 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. Lubricating oil was then slowly introduced into the system to ensure proper drainage and that oil was not migrating into the dry gas seals. After air and oil systems were tested successfully, instrumentation was confirmed and the turbine slow-rolled to test the rest of the system. For this test, the dyno wheel was not attached. Figure 11 shows the fully assembled turbine on the test stand prior to installing the pipe and heater insulation.

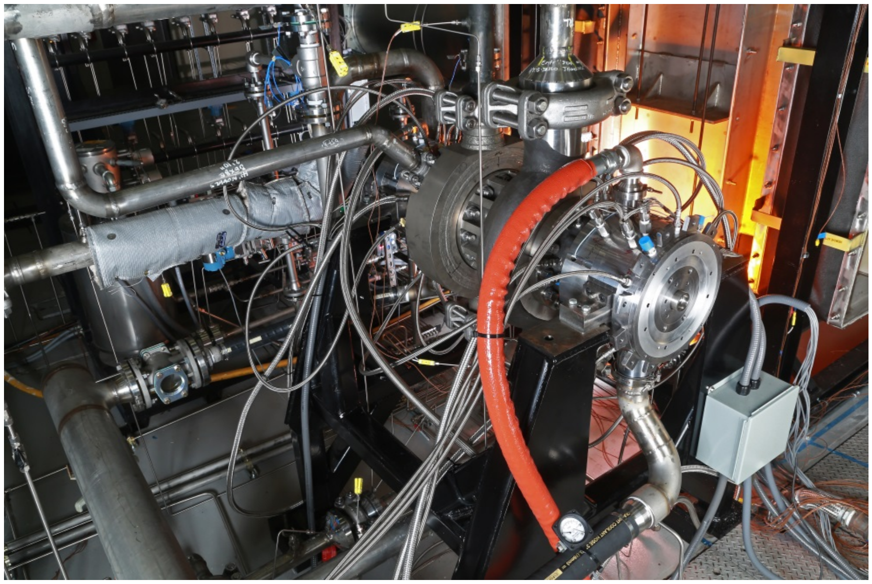

Figure 11: Assembled Turbine Casing on Operating Stand

TEST LOOP HARDWARE ACQUISITION AND INSTALLATION

The primary purpose of the test loop is to characterize the mechanical and aerodynamic performance of the recuperator and turbine 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 12. 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 an existing CO_2 loop including an existing shell-and-tube (wet) heat exchanger. More detail on the loop design can be found in Moore et al., (2014). The design test loop operating conditions are shown in Table 1.

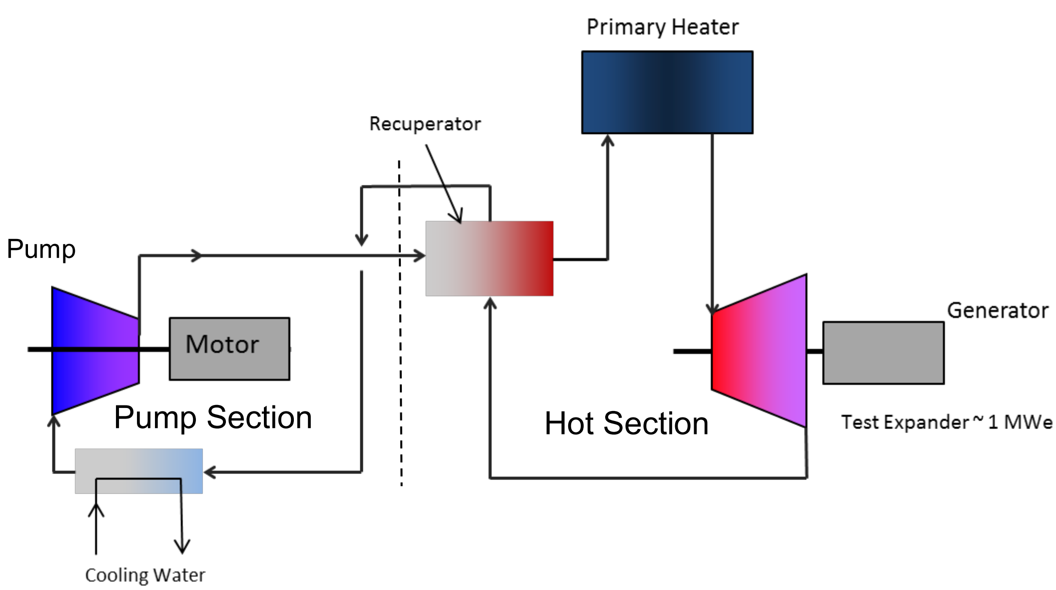

Figure 12: Simple sCO_2 Cycle for Test Loop

Table 1: Loop Operating Conditions (Moore, 2014)

Component	T out (ºC[ºF])	P out (bar [psi])	Flow (kg/s [lb/s])
Pump	**29.22 [84.60]**	**255.0 [3698]**	**9.910 [21.85]**
Recuperator-Hot Side	**470.0 [878.0]**	**252.3 [3659]**	**8.410 [18.54]**
Primary Heater	**715.0 [1319]**	**250.9 [3639]**	
Turbine	**685.7 [1266]**	**86 [1247]**	
			9.910 [21.85]
Recuperator-Cold Side	**79.58 [175.2]**	**84 [1218]**	
Cooler	**10.00 [50.00]**	**83 [1204]**	

PRIMARY HEATER OVERVIEW

For the heat source, a custom CO_2-to-air heat exchanger was developed, which was mated to a commercial natural gas fired furnace and exhaust ducting. 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 and fabricated by the project team. The overall intent of the heater for the 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 2.

Table 2: Test Loop Design Operating Conditions

	Recuperator Outlet/ Heater Inlet	Heater Outlet/ Turbine Inlet	Turbine Exit
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 13. This design allowed the tubes to be bent with weld joints only to the manifolds.

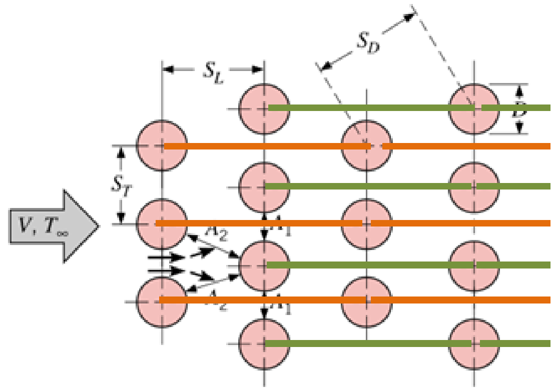

Figure 13: Heat Exchanger Tube Layout. Each Horizontal Row of Colored, Connectors represents a Single Tube

Figure 14 shows the completed heater heat exchanger using Inconel 740H tubing material. The tubing penetrations through the outer shell can be seen as well.

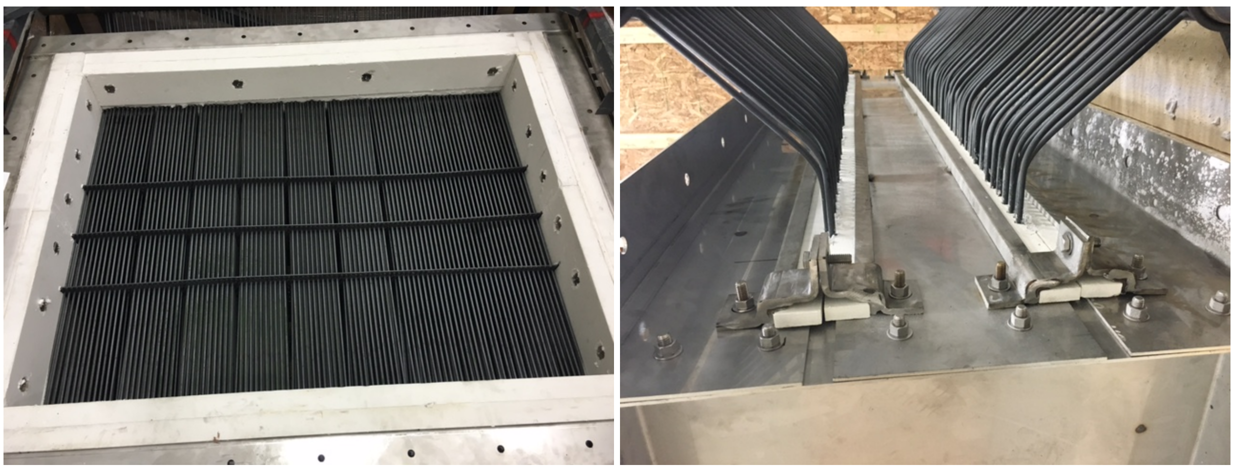

Figure 14: Heat Exchanger for Primary Heater

All of the piping was installed and the field joints were completed, including the Inconel 740H piping that connects the heater to the turbine. Figure 15 shows the 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.

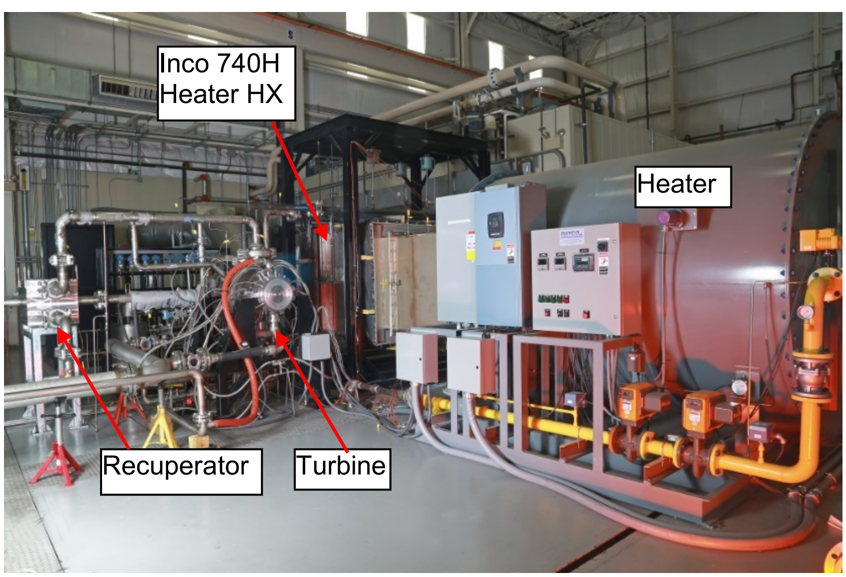

Figure 15: SunShot Test Loop

To provide the pressure rise in the loop, a dense-phase pump was procured. This is a 12 stage back-to-back, 3,600 rpm pump that requires that the CO_2 be below the critical temperature (< 20ºC). Figure 16 shows the piping that connects the pump to the loop and the recuperator. This piping followed the same process using field welds, RT inspection, and hydro testing. To minimize heat loss and minimize risk to personnel, insulation has been added to the heater and pipe loop as shown in Figure 17.

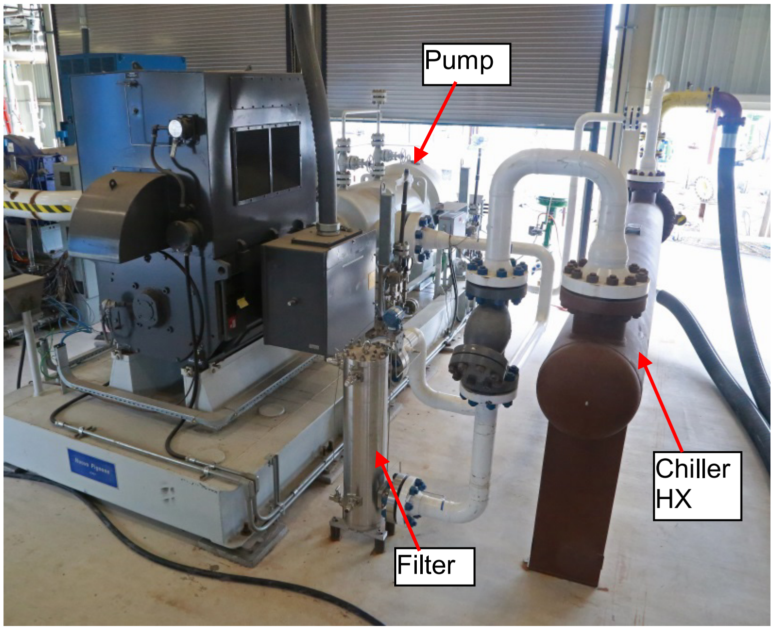

Figure 16: sCO_2 Pump and Chiller Heat Exchanger

Figure 17: SunShot Turbine Assembly with Insulation

TEST LOOP COMMISSIONING

As part of the commissioning and initial testing, the dry gas seal supply, buffer air, and lubricating oil systems were leak tested. Care was taken to adjust the buffer air to prevent oil migration into the dry gas seal cavity, made challenging given the relatively high oil flow rate (42 gpm on thrust end of turbine) in the small machine. The dry gas seal vent manifold contains low-point drains and remained free of oil. The dry gas seal panel contains a large electric heater, which vaporizes and super-heats the CO_2 to prevent dry ice formation in the seal vents. Independent control to each dry gas seal is accomplished using two control valves with a flow control strategy.

Pressurizing the test loop starts with the main loop pump section with the hot section near the turbine isolated. Fill is accomplished using a small reciprocating pump fed from a 1,500-gallon liquid CO_2 storage tank. A loop fill valve is used, as well as, feeding warm gas to the pump seals. This strategy of filling the loop with both liquid and vapor helps achieve the desired mass in the loop and prevents excessive venting after starting the pump and turbine. Also, by feeding the dry gas seals with warm seal gas, dry ice formation in the vents can be avoided. The seal gas is heated using a 55 kW electric heater rated for the high pressure supply. A chiller is started flowing chilled water through the heat exchanger upstream of the pump. The pump is then started at low speed until the desired pump inlet temperature of below 20ºC is achieved. It is then accelerated to its design condition of 3,600 rpm and 3,600 psia (250 bara) by throttling the recycle valve around the pump. The hot section is pressurized next by switching the warm seal gas to feed the heater (refer to Figure 12).

Since the turbine utilizes abradable seals, spinning tests began with a break-in procedure that consisted of incremental speed targets followed by shutdowns to prevent overheating during seal rubs. All safety checks were performed first including verification of emergency shutdown buttons, over-speed protection, and loop venting. More detail on the control system and emergency shutdowns scenarios by the control system can be found at Allison (2018). Multiple vent valves are used to prevent rotation of the turbine during venting. Break-in procedures are followed until the first critical speed (10-12,000 rpm) is traversed. 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 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 (55℃)	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 (177℃)	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	1700 °F (926℃)	Reduce fuel flow rate.

*DGS=Dry Gas Seal

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 (715℃)	Reduce heater fuel flow
Turbine exhaust temperature limit	1270 °F (688℃)	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 (260℃)	Reduce heater fuel flow

TEST RESULTS

Testing reached the first design point with the conditions shown in Table 5. This first design point is representative of a waste heat recovery application in terms of temperature with a speed of 21,000 rpm at a turbine inlet temperature of 550ºC. This condition represents the highest temperature condition achieved to date for a sCO_2 turbine. Figure 18 is a test data screen capture for key turbine parameters taken during the test while reaching this condition. It shows an inlet pressure to the turbine of 2,200 psi at a speed of 16,600 rpm. The rotor exhibited low vibrations, good thrust balance, and bearing temperatures well within limits. Thermocouples were installed at critical locations on the casing including the case near the dry gas seals and on the flange bolts. The bolt temperature is compared to the adjacent flange temperature to ensure that proper bolt tension is maintained (or overstretch prevented) during thermal transients.

Table 5: Turbine Design Operating Points

	Speed (rpm)	Turbine Inlet Temp. °C (°F)	Turbine Inlet Pressure bar (psi)	Turbine Exit Pressure bar (psi)
1st Design Point	21,000	550ºC (1,022°F)	~200 bar (3,000 psi)	80 bar (1,160 psi)
2nd Design Point	27,000	715ºC (1,319°F)	~250 bar (3,625 psi)	80 bar (1,160 psi)

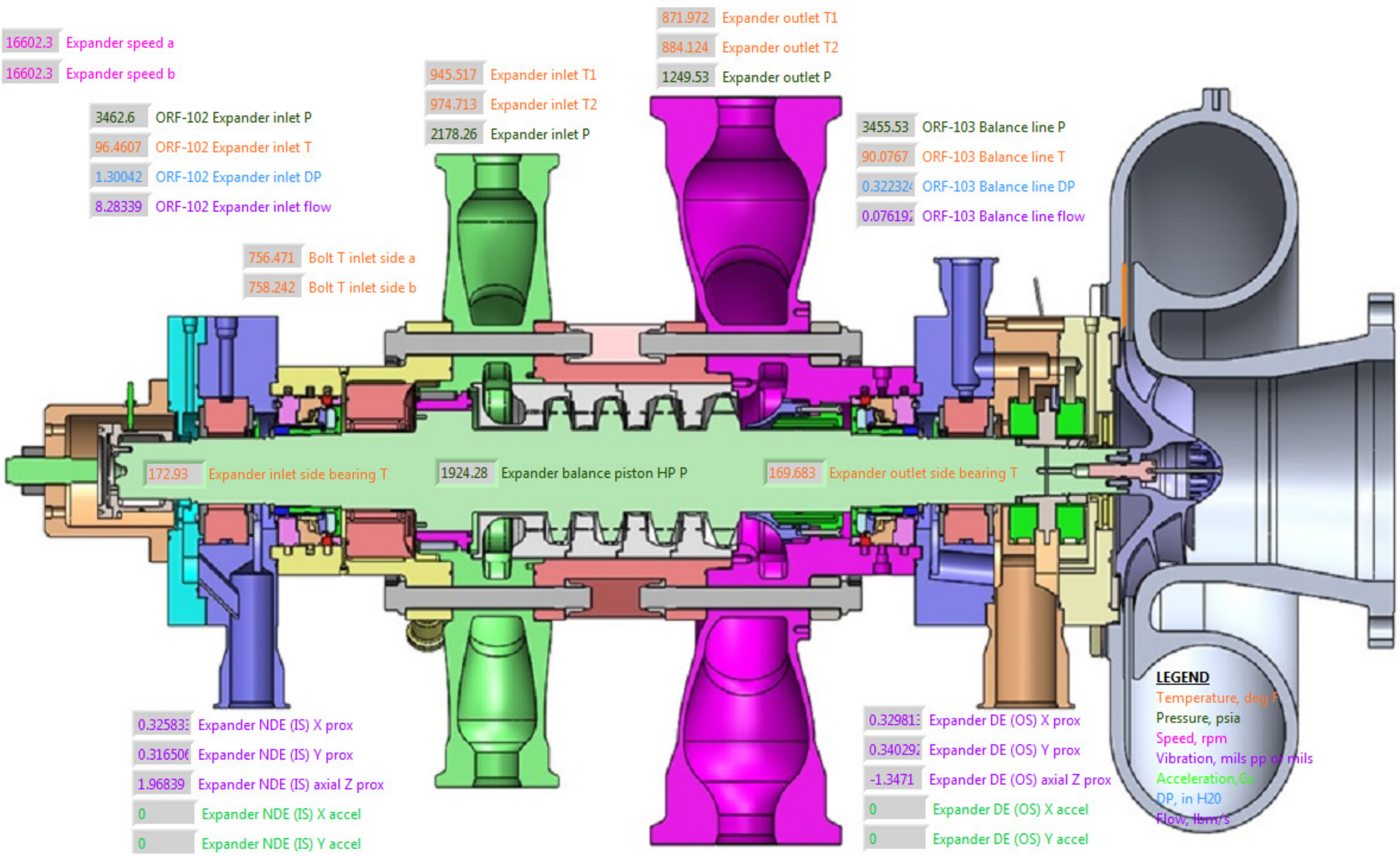

Figure 18: Screen Capture of Data Acquisition System during Initial Turbine Testing

Figure 19 shows a time trend plot of the turbine inlet and exit pressures along with turbine speed. Both the loop fill and start-up can be seen. Some speed transients occur as the pressure and temperature at the turbine inlet is stabilized. Figure 20 shows the turbine inlet and exit temperature for this same period. Note that the main heater is fired (to about 260ºC, 500ºF) even during the filling period to begin preheating the turbine case. As CO_2 is flowed through the turbine leading up to break-away, the temperature and pressure undergoes changes as the gas in the loop is mixed requiring either fill or venting to obtain the correct mass in the loop. Once this is stabilized, the flow is further increased leading to break-away of the turbine to an initial idle speed near 5,000 rpm.

The temperature of the turbine inlet is slowly increased staying within the limit 670ºC/hour (1,200Fº/hour) shown in the table above. This rate minimizes thermal stress in the rotor and casing during these early tests. As the temperature increases, the speed also increases. After the temperature is stabilized at 550ºC (1,022ºF), the speed is further increased by opening the turbine throttle valve to a speed of 21,000 rpm reaching a turbine inlet pressure of around 180 bar (2,600 psi) with an exit pressure of 80 bar (1,160 psi).

Figure 21 shows the trend plot for radial vibration and axial thrust position showing less than 0.5 mils p-p (12 um p-p) throughout the operating speed range including transition through the first critical speed (10-12 krpm). Thrust position show no unusual behavior as the turbine is loaded. Figure 22 plots the vibration spectrum at 21,000 rpm indicating only a small synchronous vibration with no signs of subsynchronous vibration, indicating a stable rotor. Future testing will gradually increase speed and temperature to a maximum value of 27,000 rpm and 715ºC, respectively.

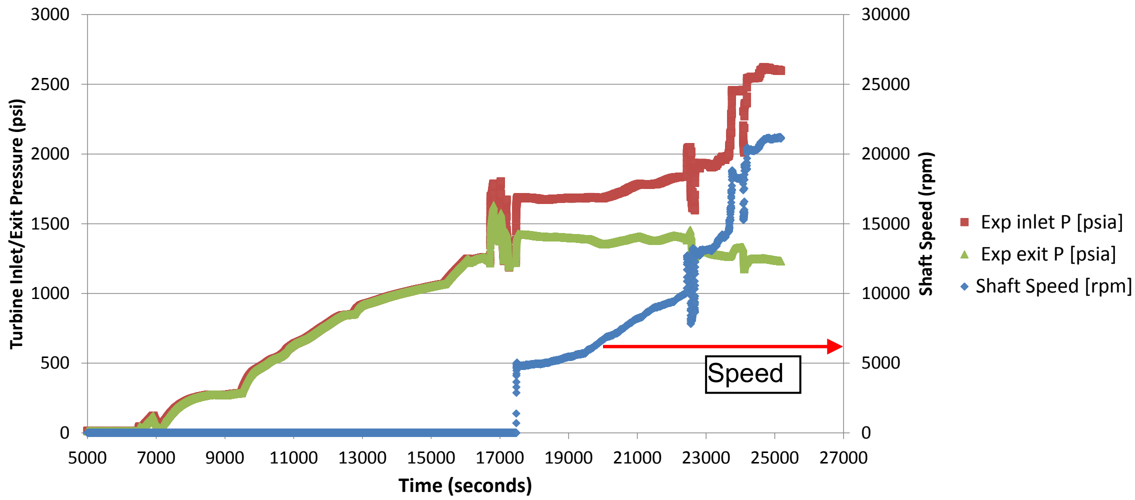

Figure 19: Trend Plot: Turbine Pressure and Speed

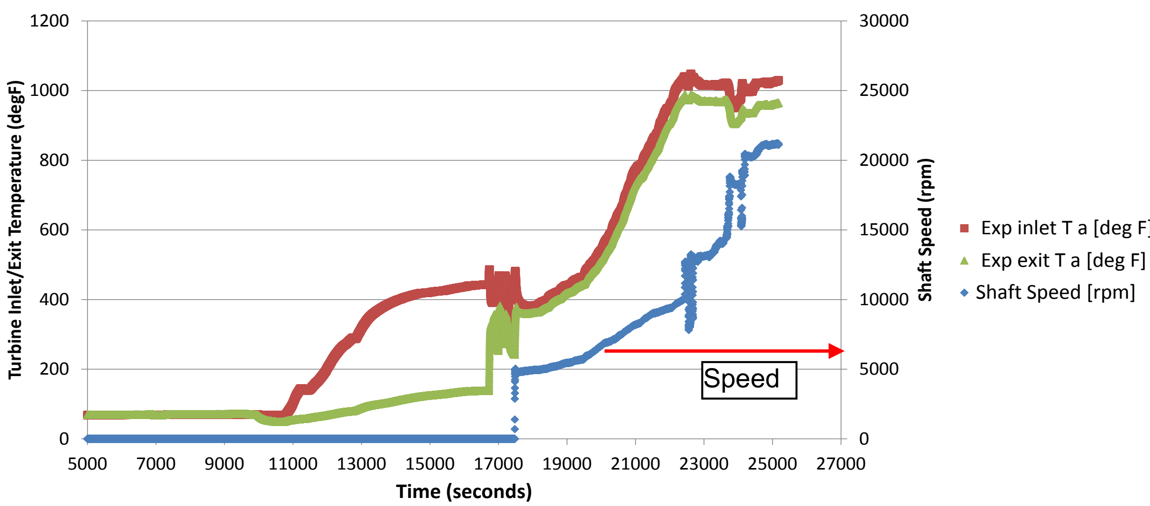

Figure 20: Trend Plot: Turbine Temperature and Speed

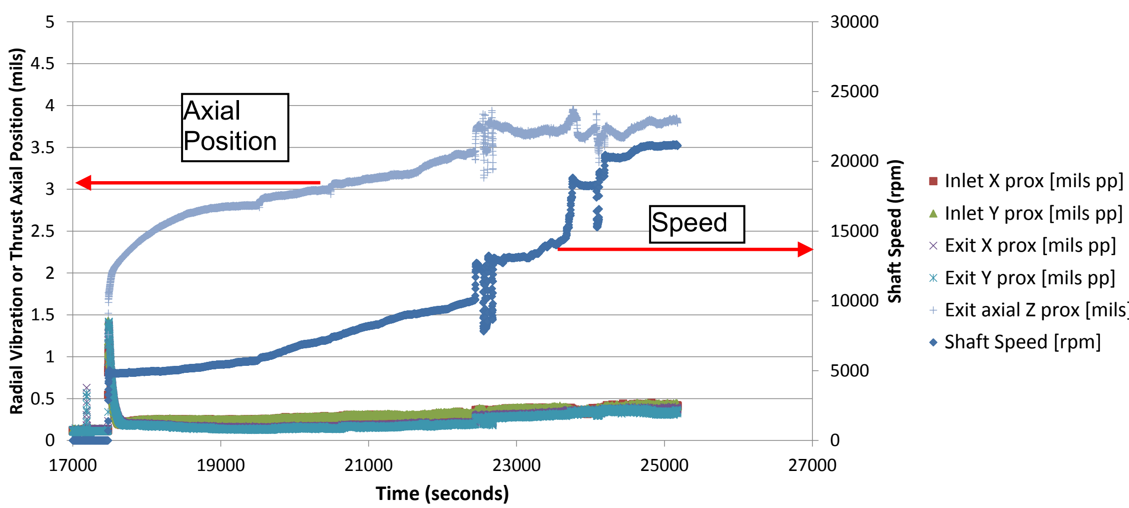

Figure 21: Trend Plot: Turbine Vibration, Thrust Position, and Speed

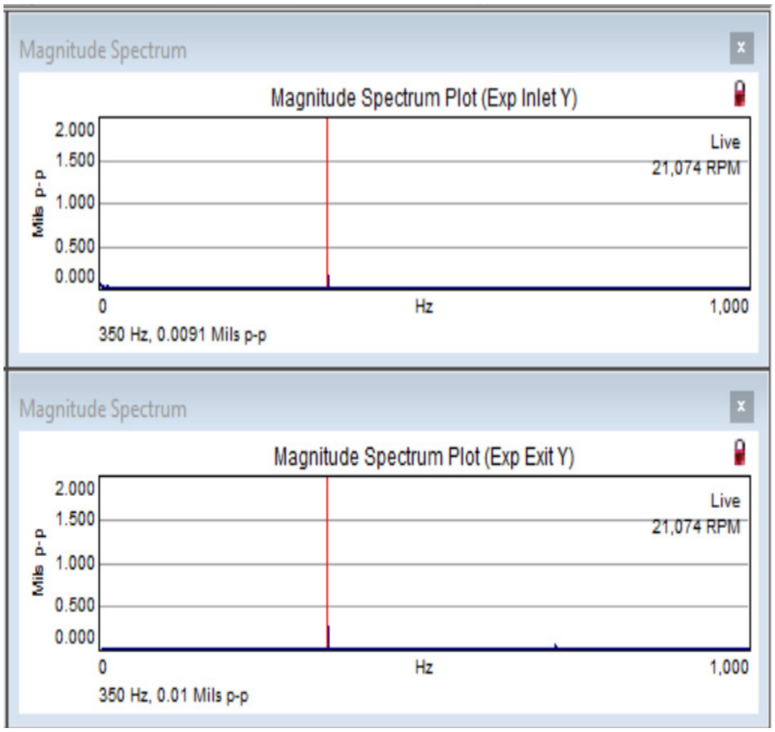

Figure 22: Turbine Radial Vibration Spectra

CONCLUSIONS

This paper outlines the assembly, commissioning, and initial testing of a 1 MWe sCO_2 test loop consisting of a novel 1 MWe turbine (10 MWe frame size), a novel air-to-CO_2 heat exchanger made of Inconel 740H, commercial dense-phase pump, and a commercially sourced PCHE recuperator. The test loop components met all applicable codes (hydro test, weld procedures, etc.). 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 followed by mechanical testing to monitor vibrations, critical speeds, and bearing temperatures. Initial testing of the turbine has achieved its first design point at the turbine inlet of 550°C and 180 bar 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.

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