DEVELOPMENT OF A 1 MWE SUPERCRITICAL CO_2 BRAYTON CYCLE TEST LOOP

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ABSTRACT

Recent studies have demonstrated that 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. With funding under the SunShot initiative, the authors are developing a high-efficiency sCO_2 turbo-expander for the solar power plant duty cycle profile and novel compact heat exchangers for the sCO_2 Brayton cycle. However, no test loop exists to test the turbine and heat exchangers under development. Therefore, a customized test loop is being developed at Southwest Research Institute that will accommodate the full test pressures (80 to 280 bar) and temperatures (45 to 700ºC) of the proposed Brayton cycle. The paper describes the design methodology to predict the pipe flow behavior and thermal growths as well as material selection. A customized natural gas fired heater has been designed, since no heater like it is available currently. Finally the test plan for testing the turbine and heater will be presented.

Background

One of the earliest uses of sCO_2 as a working fluid in a closed-loop recompression. Brayton cycle was proposed by Combs in 1977 (Combs, 1977) for shipboard applications, in which Combs concluded that a substantial reduction in fuel consumption was possible. More recently, sCO_2 cycle testing has been performed at Sandia National Laboratories (Conboy, 2012) and at Knolls Atomic Power Laboratory (DOE, 2012). As of 2012, the Sandia facility has achieved a turbine inlet temperature of 650°F (343°C) and generated 20kWe.

The Sunshot program is funded by the Department of Energy (DOE) Office of Energy Efficiency and Renewable Energy (EERE) SunShot office under the CSP power block Funding Opportunity Announcement (FOA). Co-funding is provided by our partners General Electric, Thar Energy, and Bechtel Marine. The thermal-to-electric efficiency of current CSP plants is 35 to 45% (DOE, 2012). The goal of this program is to meet these aggressive performance and cost goals:

- Net cycle efficiency > 50%
- Dry cooled
- Cost < $1,200/kWe

Southwest Research Institute (SwRI) in collaboration with General Electric and Thar Energy was awarded a Phase I award on the design and development of these tasks:

- Design Supercritical CO2 Brayton Cycle Power block to achieve FOA goals
- Proposed modular power block in 10 MWe range to meet CAPEX targets
- Compact power block for pre-fabricated tower mounted operation
- SwRI scope includes test loop design and operation, assist GE with expander engineering, manufacturing drawings, and expander fabrication.
- GE is responsible for the power block design, thermo-economic analysis, and test loop thermal design.
- GE to design the sCO2 turbo-generator to meet FOA targets.
- Thar Energy to design recuperator for the power block meeting the FOA efficiency and cost targets.

The team targeted a 30% reduction in recuperator cost from current state-of-the-art by implementation of advanced manufacturing processes. Table 1 outlines the schedule for the three project phases.

Table 1. Project Work Breakdown Schedule

22 months	12 months	6 months
Phase 9/12 – 7/14	Phase 2 8/14 – 8/15	Phase 3 8/15 – 2/16
• Test loop design & component/vendor identification (1 MWe)	• Test loop fabrication	• Expander assembly and shake-down testing Expander testing off-design at 1MWe scale.

22 months	12 months	6 months
• Expander engineering	• (1 MWe)	• Recuperator testing at 5MW-th scale.
• Test loop and expander manufacturing drawings	• Expander fabrication	
• Recuperator design and bench scale testing	• Recuperator fabrication.	
	• Test loop assembly	

Cycle Analysis

Various cycles and operating conditions were studied in order to achieve the FOA efficiency goals. The cycle selected is shown in Figure 1 named Recompression Supercritical CO_2 Cycle Model

The proposed CSP system uses sCO_2 as both, the heat transfer fluid and the working fluid.

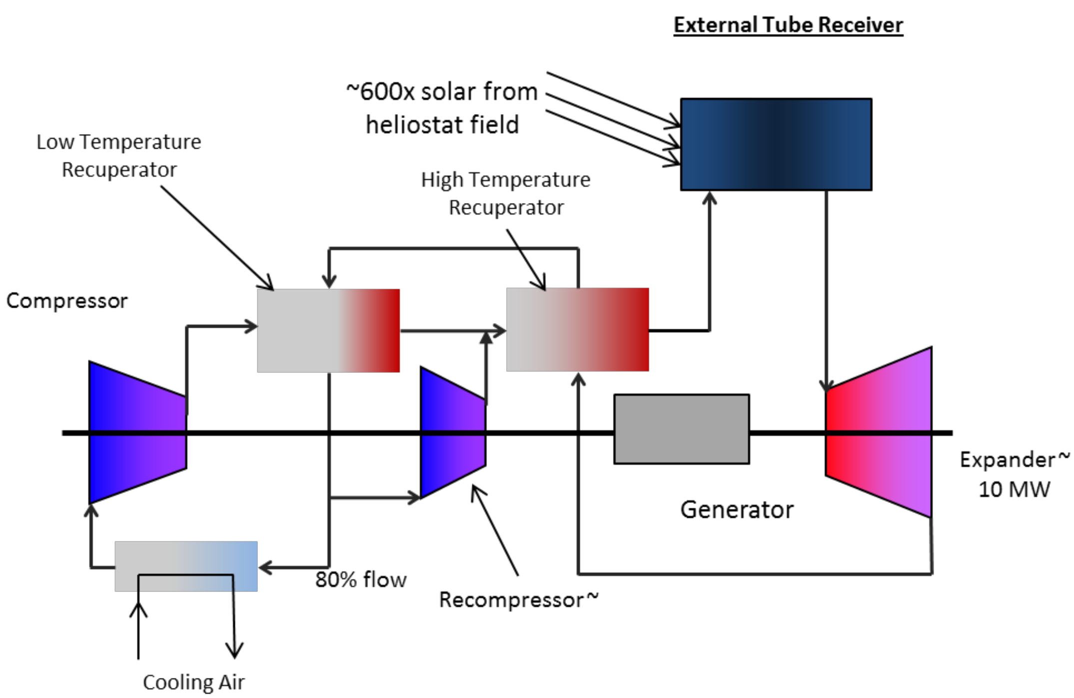

Figure 1. Recompression sCO_2 Cycle

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 than the dual recuperator shown above. The loop utilizes part of the existing CO_2 loop at SwRI including an existing shell-and-tube (wet) heat exchanger.

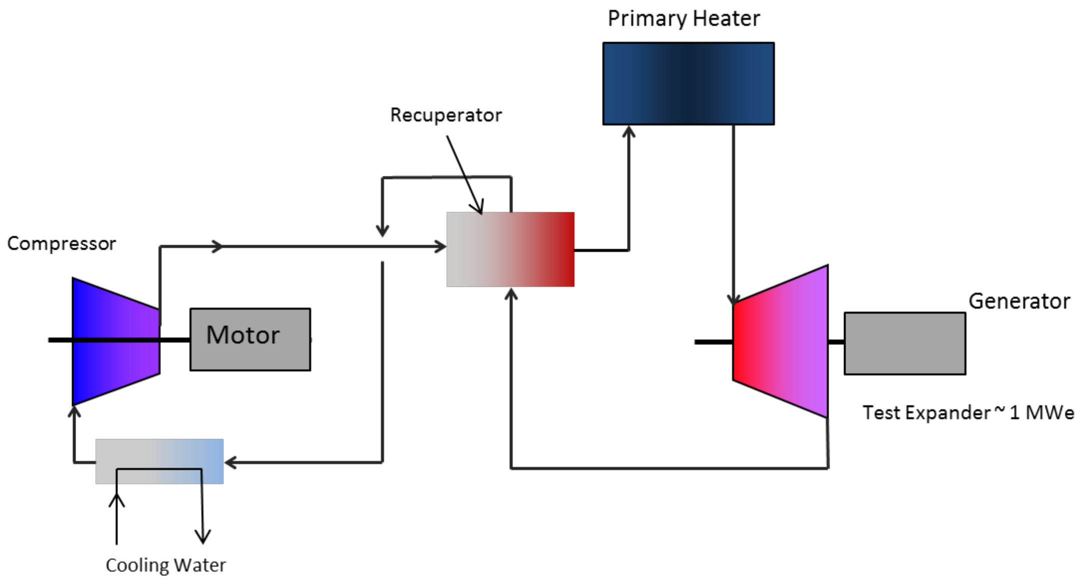

Figure 2. Simple sCO_2 Cycle for Test Loop

Test Loop Design

The operating conditions have been defined for all major components of the cycle and are summarized in Table 2. The pressure and temperature at the inlet and outlet of each component dictate the required materials of construction and define the density of CO_2 at that location. Based on the fluid density and the mass flow, the interconnecting piping can be sized to control the maximum flow velocity. The system which has been designed for construction and testing at Southwest Research Institute has been sized based on a maximum flow velocity of 100 ft/s. This limit is based on past experience and is expected to maintain a compromise between reasonable pressure losses and minimized pipe sizes, particularly for expensive nickel alloy and stainless steel sections.

The minimum required thickness for each section of piping is calculated based on ASME B31.3 for Process Piping. A safety factor of two was applied to the allowable stress and a corrosion allowance of 1/8" was added to all pipe sections, except the recuperator hot outlet to the heater, where a 1/16" corrosion allowance was added. Using a 1/8" corrosion allowance on this section would have required increasing the pipe thickness to a non-standard size, or changing to a more expensive material. A corrosion allowance of 1/16" is typically used in process piping, but given the novel nature of the test program and uncertainties in material durability when exposed to high temperature CO_2, a conservative corrosion allowance was chosen where possible. Piping will be regularly inspected for corrosion, and the testing may yield novel and valuable information about material durability. Pipe diameter, thickness, material of construction, and the predicted flow velocity for each section are shown in Table 3.

Table 2. Loop Operating Conditions

Component	T out (°C[°F])	P out (bar [psi])	Flow (kg/s [lb/s])
Pump	29.22 [84.60]	255.0 [3698]	9.910 [21.85]
Piping (1)	-	254.3 [3688]	
RCP-H	470.0 [878.0]	252.3 [3659]	8.410 [18.54]
Piping (2)	-	251.9 [3654]	
HT-HTR	715.0 [1319]	250.9 [3639]	
Piping (3)	-	250.6 [3634]	
EXP	685.7 [1266]	86 [1247]	
Piping (4)	567.3 [1053]	-	9.910 [21.85]
RCP-C	79.58 [175.2]	84 [1218]	
Piping (5)	-	-	
CLR	10.00 [50.00]	83 [1204]	

Table 3. Pipe Specifications

Section	NPS	Schedule	Thickness	Material	Predicted V (ft/s)	Flanges
Pump out	3	XXS	0.6	A106B	13.07	ANSI 2500#
Mixing line	1.5	XXS	0.4	316s	28.56	ANSI 2500#
Recuperator hot out	3	XXS	0.6	316s	59.19	Grayloc
Heater out	3	160	0.438	Inco 625	62.26	Grayloc
Heater out dual	2.5	160	0.375	Inco 625	47.47	Grayloc
Expander out double	3	160	0.438	Inco 625	84.73	Grayloc
Recuperator cool out	3	160	0.438	A106B	54.13	ANSI 1500#
6" to cooler	6	160	0.718	A106B	13.84	ANSI 1500#
Cooler out	8	120	0.718	A106B	1.37	ANSI 900#
Pump inlet	4	120	0.437	A106B	5.39	ANSI 900#

The process and instrumentation diagram in Figure 3 depicts the SwRI test configuration, process conditions, and required equipment (Table 3). Additionally, process measurement (pressure and temperature) locations are shown upstream and downstream of the major components, which will characterize their performance. Loop flow will be measured via an orifice flow meter downstream of the CO_2 pump (label A). An additional flow meter is located on the mixing line (label B), which provides cool flow from the pump outlet to the recuperator inlet, mixed with the expander outlet flow. This will allow the recuperator to operate at a safe temperature since the expander will be operating off-design during the mechanical test. A third flow meter is located on the pump outlet line leading into the recuperator (label C), after the split to the mixing line. The compact nature of the test loop requires this flow meter to be of the V-coneTM type, which minimizes the required upstream and downstream straight pipe lengths. These three meters will allow an accurate determination of total loop flow, and the proportion of flow going to the recuperator, heater, and expander.

The two expander balance lines (label D) have been routed from the turbine casing to the expander outlet piping where the cold mixing line ties into expander outlet piping (label E), forming a cross. These lines will be fitted with orifice plates for back pressure control and flow measurement. A high pressure feed pump (label F) (approximately 2300 psi) drawing from the CO_2 delivery system is used for filling the loop and initial supply to the expander dry gas seals. Once the loop is operating near the design point, the dry gas seals will be supplied by the GE CO_2 pump discharge, and the CO_2 delivery system will be used to maintain the loop pressure. The flow leaving the recuperator cold end connects to the existing shell-and-tube heat exchanger and returns to the pump.

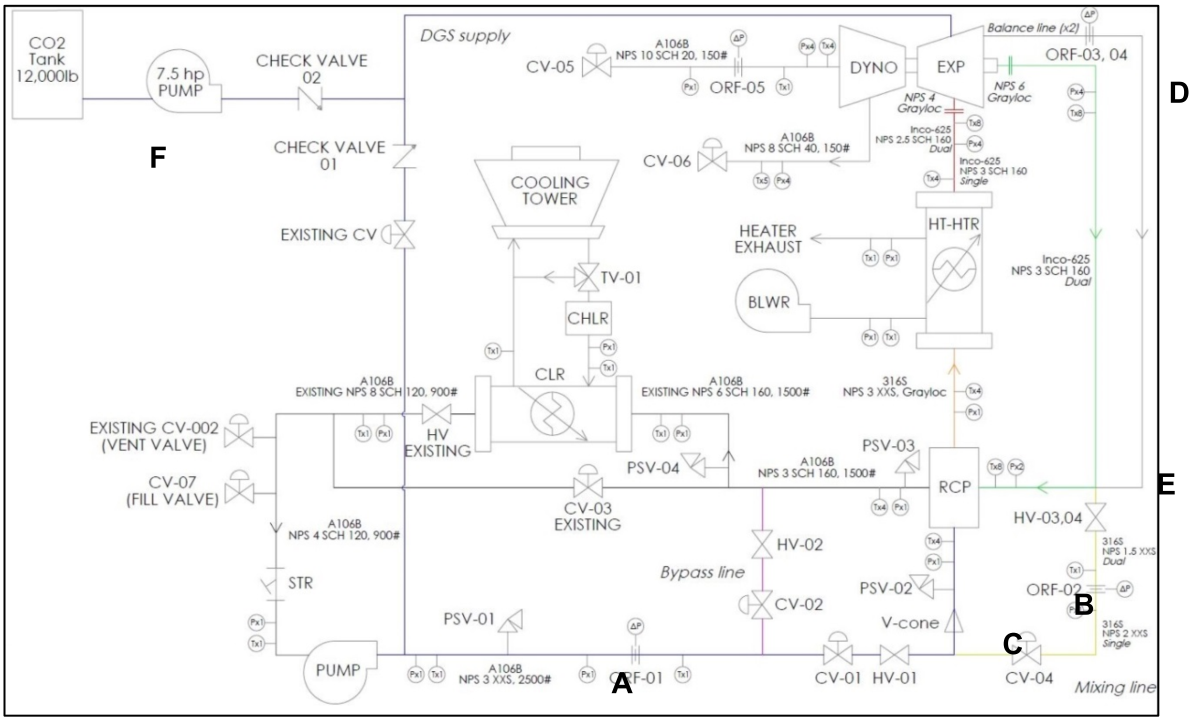

Figure 3. Test Loop Process and Instrumentation Diagram

Table 2. P&ID Equipment List

Component	Symbol	Description
Pump	PUMP	sC0$_2$ pump: GE Nuovo Pignone
Recuperator	RCP	Heat exchanger: Thar Energy
Heater	HTR	Gas-fired heater: Thar Energy
Blower	BLWR	Heater air supply blower
Expander	EXP	sC0$_2$ turbine: GE

Component	Symbol	Description
Dynamometer	DYNO	Load absorbing centrifugal compressor: SwRI
Chiller	CHLR	800 gpm cold water
Loop throttle	CV-01	Main loop throttle control valve
Compressor recycle	CV-02	$sC0_2$ pump recycle control valve
Cooler bypass	CV-03	Existing 3” Dyna-Flo, 900# process bypass
Dilution valve	CV-04	Mixing valve to control RCP-H inlet temp.
Dyno suction valve	CV-05	Dyno compressor suction throttle valve
Dyno discharge	CV-06	Dyno compressor discharge throttle valve
Cooling water bypass	TV-01	Existing 3-way cooling water bypass hand valve
Flow meter	ORF	Orifice plate flow meter
V-cone	V-CONE	Flow meter
Strainer	STR	4” Y-strainer
Relief valve	PSV-01	Set pressure = 4000 psig
Relief valve	PSV-02	Set pressure = 4000 psig
Relief valve	PSV-03	Set pressure = 1975 psig
Relief valve	PSV-04	Existing 2x3”, set pressure = 1975 psig

Piping Flow Simulation

A steady-state flow model of the system was created using Stoner Pipeline Simulator (SPS). SPS is a transient and steady-state hydraulic simulator that calculates dynamic flow of single phase fluids and fluid handling machinery such as pumps and compressors based on an equation of state. The model was tested on CO_2 using the Benedict-Webb-Rubin-Staring (BWRS) equation of state and compared to values from NIST REFPROP to verify its performance at supercritical conditions.

The model was built using pipe and valve elements with the expander inlet and outlet conditions being achieved via a heat-exchanger and valve combination. The pump suction and discharge pressure, temperature, and flow were imposed via fixed boundary conditions. A schematic of the model is shown in Figure 4 and the resulting predicted density at each section is shown in Table 5. The conditions predicted by SPS agree well with the reference. The greatest error is observed at the cooler outlet/pump inlet, where the CO_2 is at its coldest, lowest pressure state (nearest the critical point) but is considered acceptable. This steady-state hydraulic model was be used to verify flow velocities, pressure losses, and thermal exchange with the surrounding environment throughout the system. Based on results of this simulation, the pump discharge pipe was increased from 2” to 3” to reduce pressure losses from 70 psid to 10 psid.

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                                                                 /-----|
                                                                /      |   1174.00 DF
                                                 3681.71 PSI   /       |   1228.22 PSI
                                                         |---|EXPANDER|----|
                                                         |     \       |    |
                                                         |      \      |    |
                                           1320.00 DF    |       \-----|    |
                                            2.2051 PSID  |                  |
                                                   |HT HEATER|              |
                                                         |                  |
                                                         |                  |
                                                         |                  |
                                              874.60 DF|                    |
            50.00 DF                                     |                  |
           0.2837 PSID                108.00 DF          |                  |
   |-----<-----|COOLER|----------------<-----|RECUPERATOR|-----<-----|
   |                                  |                  |
   |                         0.0000  |                   |
   | 1204.00 PSI                    |CV 02|              |
   |                                  |                  |
|------|                              |                  |
| PUMP |---->--------------------------|CV 01|-->-|
|------|                                  1.0000
3768.20 PSI
  84.60 DF
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Figure 4. Stoner Pipeline Simulator Model Schematic

Table 3. Comparison of CO_2 Density - Stoner Pipeline Simulator and REFPROP

Device	P (psia)	T (°F)	Density [lb/ft3]		Error
			Stoner	REFPROP	
Pump out	3697.91	143.17	49.387	48.883	1.03%
LT-HTR out	3690.28	356.90	22.9	22.554	1.53%
RCP 1 out	3681.14	988.00	10.306	9.9871	3.19%
HT-HTR out	3678.02	1320.00	8.259	8.0043	3.18%
EXP out	1213.73	1190.00	3.014	2.9711	1.44%
RCP 2 out	1205.08	366.10	6.621	6.5232	1.50%
CLR out	1204.00	86.00	47.988	44.789	7.14%

A thermal stress analysis was performed using Caesar II piping modeler to predict nozzle loads and determine support placement. Operating temperatures are input to the model and thermal expansion is computed based on the pipe geometry and material. The predicted loads are shown in Table 6, where the expander loads are within the allowable described in the NEMA 23 standard for steam turbines.

Table 4. Predicted Nozzle Loads

	Fx (lb)	Fy	Fz	Mx (ft-lb)	My	Mz
RCP-C in	66	-50	-120	-144.3	-130.8	-89
RCP-H out	74	-305	-129	456.2	48.7	-185.4
HTR in	17	-92	-85	143.4	290.4	-253.4
HTR out	305	605	-252	-325.4	1166.5	490.7
EXP in, top	670	-143	-143	-105.6	-457	-777.2
EXP in, bottom	-365	-149	-109	53.1	-393.1	-477.7
EXP out, top	333	898	-170	-1613.1	175.1	164.8
EXP out, bottom	102	-781	-82	1414.5	121.9	-618.3
RCP-H in, top	232	-2537	101	2405	-386.2	287.4
RCP-H in, bottom	118	2271	-94	-2334.2	-292.2	-522.7
RCP-C out	55	169	165	195.3	-686.6	457.4

Test Loop Layout and Integration into Existing Infrastructure

A 3-D solid model has been created incorporating the existing Turbomachinery Research Facility and existing piping at the SwRI campus. The expander test rig will be located adjacent to a centrifugal compressor skid which utilizes piping designed for testing CO_2 compression technologies. Most of the major components including the heater, recuperator, and expander will be placed in close proximity to one another inside the lab. Locating the expander near the heater is important since it is necessary to minimize the lengths of the hottest sections of piping. This will reduce material costs of high temperature components and help manage thermal stresses.

The pump will be placed outside the lab in between the process cooler and the heater. The heater exhaust will be vented through the building wall via ducting to an exhaust stack which will direct the hot combustion products up and away from any occupied areas. The heater shown in the figures below represents a design produced by Thar Energy employing a blower, combustion chamber, and a heat exchanger similar to the technology used in the recuperator. The facility configuration is shown in Figures 5 and 6. The piping sections are identified by colors corresponding to the labels in Table 7. Instrumentation taps (pressure, temperature, and flow) have been located to meet ASME Performance Test Code (PTC) 10 for rotating machinery and other components of interest including the recuperator, heater, and pump.

Table 5. Pipe Section Key

Pipe Section	Color
Pump to recuperator	Dark blue
Mixing line (pump to expander outlet)	Yellow
Recuperator to heater	Orange
Heater to expander	Red

Expander to recuperator	Dark green
Recuperator to existing	Light green
Existing facility piping	Dark gray
Existing piping to pump	Light blue

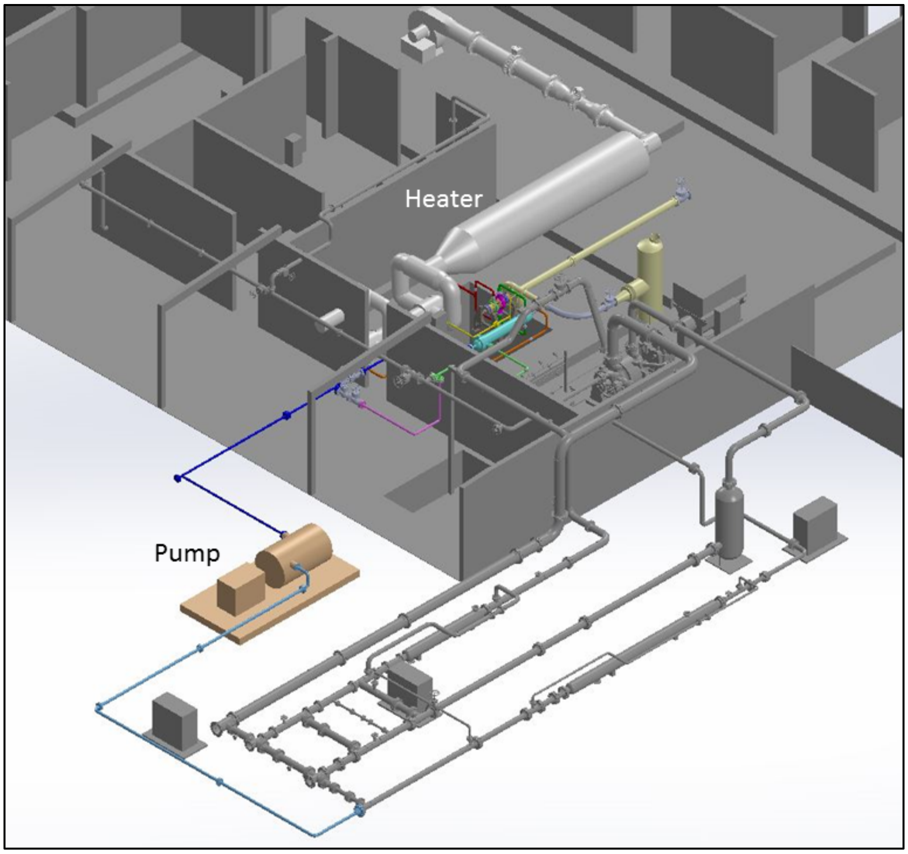

Figure 5. Existing Facility Piping and Expander Piping

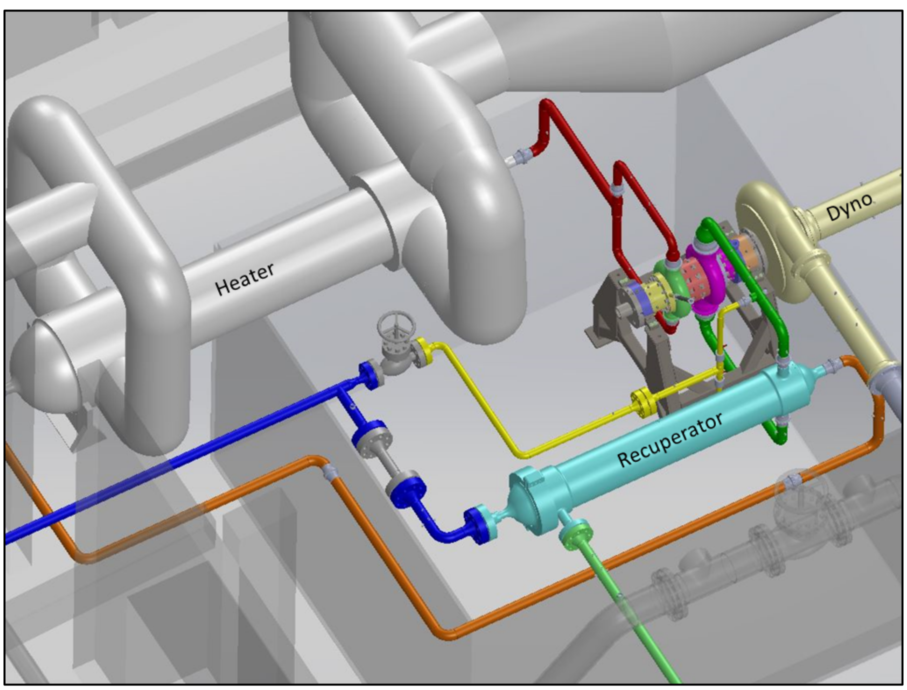

Figure 6. Expander Piping Detail

A custom engineered air dynamometer (dyno) will absorb the power produced by the expander during testing rather than an electric generator. The air dyno can be mounted directly to the expander shaft (replacing what would normally be the compressor drive coupling) and is designed to mimic the rotordynamic behavior of the coupling. Unlike an electric generator, an air-dyno cannot be suddenly unloaded eliminating the need for fast acting, high temperature turbine trip valves (TTV) to prevent over-speed of the system. The air dyno, designed by SwRI, is based on a single stage centrifugal compressor drawing in ambient air with both suction and discharge throttling to maximize turn-down. The discharge feeds into a silencer and is exhausted into the high-bay.

Future aerodynamic tests may be performed on a 10 MW variation of the turbine by driving unit to full flow using the existing Datum D12 centrifugal compressor in a closed loop, low temperature test (~400°F). This test would permit the isentropic efficiency of the turbine to be directly measured. Figure 7 shows the Dresser-Rand 3 MW 6-Stage back-to-back centrifugal compressor. Figure 8 shows an image of the existing CO_2 compressor test loop. The high pressure portion of the loop is equipped with 1500# ANSI flanges and Schedule 160 pipe and will be used for low pressure portion of the Sunshot loop. The cooler is rated to 2000 psi, which provides sufficient margin for settle-out conditions of the Sunshot loop.

Figure 7. Exist 3 MW CO_2 Compressor

Figure 8. Existing CO_2 Pipe Loop

CONCLUSIONS

The design of a high-pressure, high-temperature $_SCO_2$ flow loop has been completed to measure the mechanical and flow performance of a custom $_SCO_2$ turbine expander and recuperator. The flow capacity of the loop is equivalent to a 1 MWe size. The loop also employs a custom high-temperature natural gas heater to achieve the desired turbine inlet temperature. The goal of the test loop is not to demonstrate a particular cycle performance, but rather is to provide a platform to perform mechanical and performance testing of the expander and recuperator. The test loop design has sized the pipe to maintain acceptable flow velocities and pressure drop. A thermal piping analysis was performed to demonstrate acceptable pipe loading on the expander and recuperator nozzles. The quantity of expensive Inconel piping was minimized by co-locating the heater, expander, and recuperator in a compact arrangement. All of the test loop design objectives were satisfied. Manufacturing will commence in Phase 2 of the program.

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