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EXPERIMENTAL TESTING OF A 1MW SCO2 TURBOCOMPRESSOR

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ABSTRACT

The Nuclear Energy Systems Laboratory (NESL) Brayton Laboratory at Sandia National Laboratories has been at the forefront of supercritical carbon dioxide (sCO2) power cycle development since 2007 when internal R&D funds were used to investigate the stability of sCO2 as a working fluid for power cycles. Since then, Sandia has been a leader in research and development of sCO2 power cycles through government funded research and by partnering with industry to design and test components necessary for commercialization of sCO2 Brayton cycles. Peregrine Turbine Technologies (PTT) is a small business working to commercialize sCO2 power cycles with their proprietary thermodynamic cycles, heat exchangers, and turbomachinery designs. Under a Small Business Innovation Research (SBIR) program with the United States Air Force Research Laboratory, PTT has designed a novel motorless turbocompressor for sCO2 power cycles. In 2017, Sandia purchased the first sCO2 turbocompressor from PTT and installed it into the 1-MW thermal turbomachinery development platform at Sandia. PTT and Sandia have worked together to experimentally test the turbocompressor to the limits of the development platform (932 F @ 2500 psi). This report will detail the design of the turbomachinery development platform, the novel process used to start the turbomachinery, and the experimental results to date. The report will also look at lessons learned throughout the process of constructing and operating an experimental sCO2 loop.

INTRODUCTION

In 2012, the NESL Brayton Lab at Sandia National Laboratories (SNL) completed the commissioning of a recompression closed Brayton cycle (RCBC) development platform (DP). The DP was used in both simple and recompression configurations to investigate key technical issues related to the power cycle components as well as to validate analytical models of system performance. The experience and intellectual property accumulated over years of testing turbomachinery, heat exchangers, heat input, heat rejection systems, and support equipment for sCO2 Brayton cycles has positioned the Brayton Lab at SNL as the premier facility for testing and development of Brayton power cycles.

In order to accelerate development and facilitate the commercialization of sCO2 Brayton technology, the Brayton Lab sought industry collaboration. Through an evaluation of responses to solicitations to industry leaders, PTT was identified as the most promising company to achieve the goals of both the Brayton Lab and the DOE. PTT is a small business from Maine, USA which has the object of commercializing closed Brayton cycle (CBC) power generation systems. The first activity of the collaboration was the purchase of the first of a kind sCO2 turbocompressor from PTT and the reconfiguration of the DP located at SNL for testing of the turbocompressor. The focus of this paper is the presentation of the preliminary testing and to highlight the important experimental results to date.

TEST OBJECTIVES

The turbocompressor, also referred to as the core, is designed to operate at 118,000 rpm, 1382 F (750C) turbine inlet temperature (TIT), and 6222 psi (42.9 MPa) compressor discharge pressure and is intended for use in a 1 MWe closed Brayton cycle engine also being developed by Peregrine. The Sandia DP where the core is installed limits the turbocompressor operation to approximately 2500 psi (17.2 MPa) with a TIT of 932 F (500 C). A P&ID of the system is shown in Figure 1 and of the secondary flows in Figure 2.

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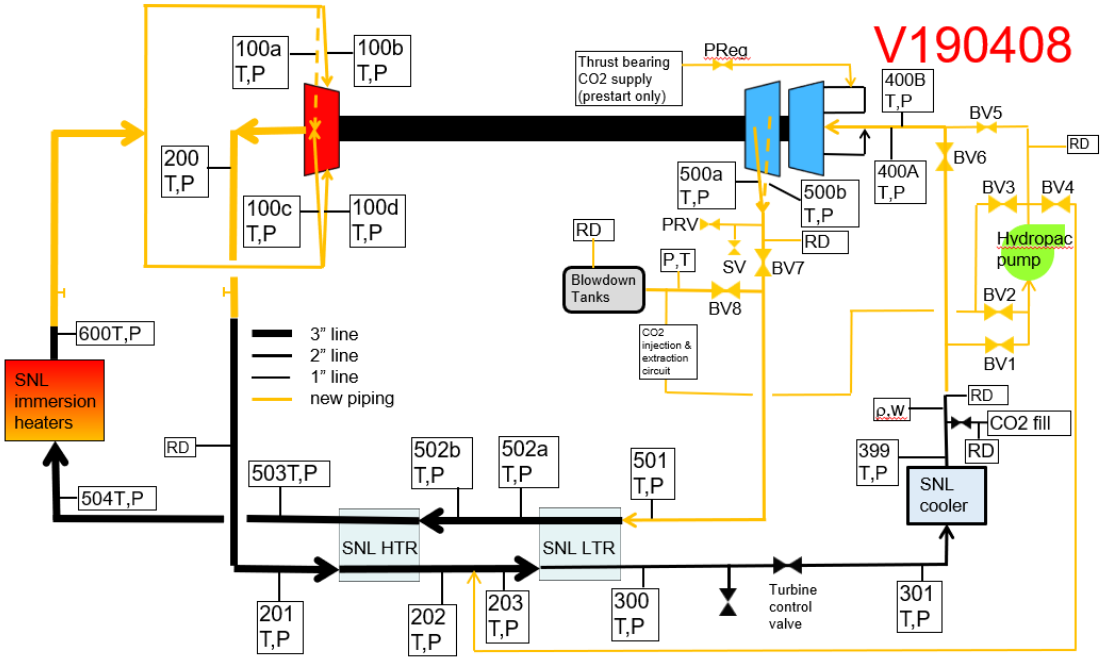

Figure 1: Development Platform piping and instrumentation diagram

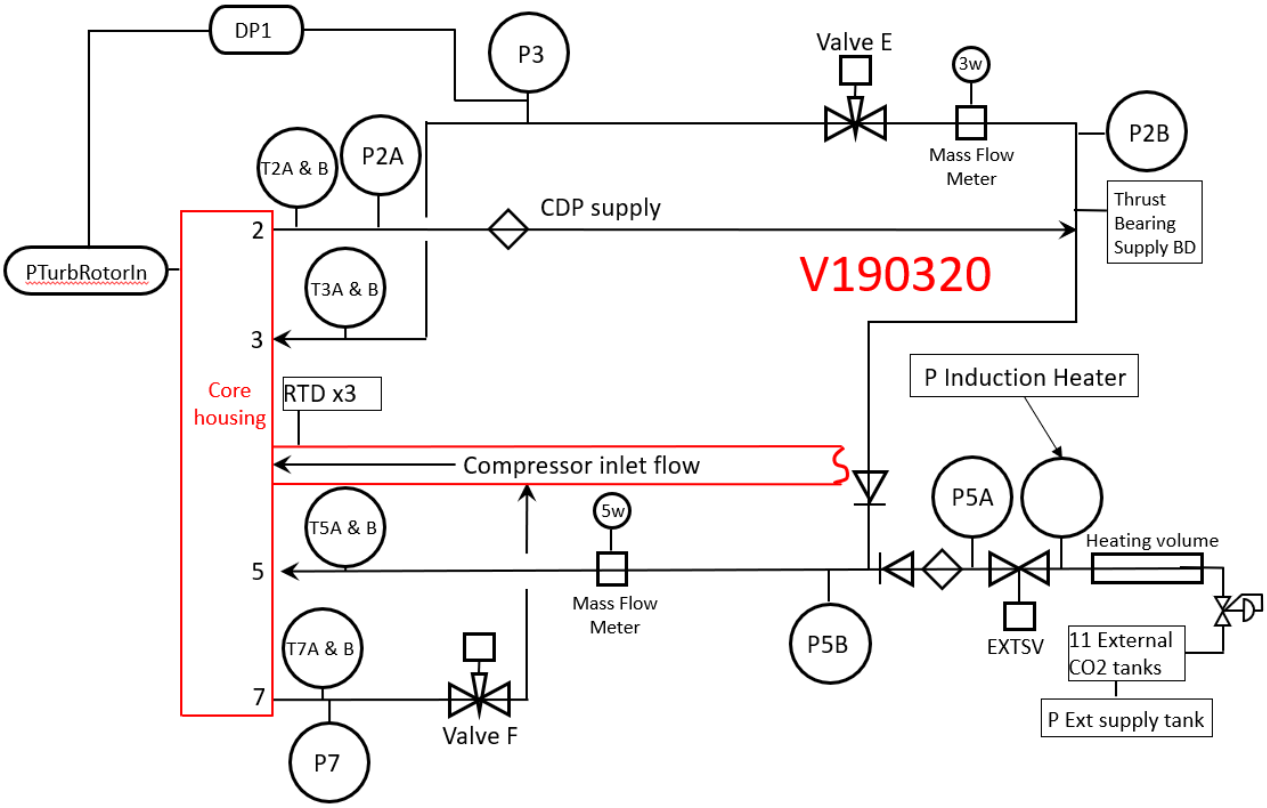

Figure 2: Secondary flows piping and instrumentation diagram

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Further details regarding the various components of the DP can be found in [1].

While the conditions achievable in the DP do not meet the specified design-point operating conditions of the turbocompressor, they are sufficient to test and validate the performance of the machine during start up, steady state operation, and shut down. The original objectives identified at the beginning of the test campaign were listed as:

1. Demonstrate the successful start to steady state of the PTT core.
2. Demonstrate repeatable start and performance operations.
3. Demonstrate operations of core internal functions
    a. Pressure activated leaf seals
    b. Radial bearings
    c. Thrust bearings
    d. Secondary flows
4. Demonstrate compressor controllability via back pressuring and TIT manipulation
5. Map turbine and compressor performance over a range of conditions
    a. Compressor inlet temperature
    b. Compressor inlet pressure
    c. Compressor pressure ratio
    d. Turbine inlet temperature
    e. Pressure loss effects
6. Support acoustical measurements for rotor speed

TESTING SUMMARY

To date, seven tests have been performed and data supporting objectives 1-4 has been successfully gathered. Objective 5 has just begun to be explored with the 8 hour long Test #7. The 6th objective is no longer applicable because an eddy current proximity probe was installed before Test #5 which is now used to measure both axial movement and rotational speed of the rotor. A summary of the tests performed is shown in Table 1.

Test Number	Test Date	Test Duration	TIT (^{o}F)	Compressor Discharge Pressure (psi)	Max PR
1	8/9/2018	00:00:32	225	1240	1.2
2	10/10/2018	00:18:40	420	1400	1.25
3	11/14/2018	00:20:46	645	1750	1.59
4	3/1/2019	00:03:32	610	1460	1.26
5	3/6/2019	00:08:23	530	1510	1.3
6	4/4/2019	00:03:18	530	1510	1.28
7	5/7/2019	8:05:44	570	1475	1.27

Table 1: **Summary of turbocompressor tests performed to date**

Initially, the tests were limited in duration due to difficulties in balancing rotor thrust which resulted in thrust bearing touchdown, axial rubs and subsequent failure of the turbine gas foil bearing. With several sets of test data in hand for data matching, secondary flow simulations gave additional insight into proper secondary flow and TCV valve settings in order to achieve complete thrust balance with no thrust bearing touchdown. Using this new information, the thrust bearing issue was resolved as of Test #4 and thrust balance was achieved as confirmed by proximity probe data. Tests #4-6 have experienced radial bearing failures after a period of stable running, which will be discussed in detail later. Test #7 demonstrated the successful operation of the radial bearings with an 8 hour test and controlled shut down.

TURBOCOMPRESSOR START

The Peregrine design is the first of its kind motorless sCO2 high-performance turbocompressor. Because there is no motor to start the turbocompressor, an alternative starting method is employed. This method is referred to as the blowdown start because inventory tanks (labeled “Blowdown Tanks” in Figure 1) are charged to a higher pressure than the loop and then are used to blowdown through the primary heat exchanger and then into the turbine to start the turbocompressor. Before the system can be started, the loop must be preconditioned such that the compressor inlet is slightly above supercritical conditions and the turbine inlet is heated to near the TIT that is desired for the test. An external positive displacement compressor is used to circulate CO2 for the preconditioning. Prior to the addition of the proximity probe the CO2 was circulated at a flow rate to minimize the time required to precondition the loop. However, when the proximity probe was added it was discovered that this flow rate was sufficient to slowly spin the shaft. The spinning of the shaft during preconditioning may have caused damage to the radial foil bearings. Once this was discovered, later tests utilized a reduced flow rate for preconditioning to protect the radial bearings.

Once the system was preconditioned, the valve positions were changed to execute a start. As can be seen in Figure 1, BV7 is first closed and then BV8 is opened to start the blowdown. This causes mass to flow through the LTR, the HTR, and the heaters before flowing into the turbine inlet. Once the compressor discharge pressure is greater than the blow down pressure by a specified margin, BV7 opens and BV8 closes and the system begins operating on its own power. The required blowdown pressure to successfully start the turbocompressor was unknown and not easily calculated. Thus, for the first tests the blowdown pressure was set to a high pressure to ensure there was sufficient energy to start and subsequently lowered for later tests, all of which had successful starts. For Test #7 the blowdown pressure had to be increased which will be explained in greater detail in the Radial Bearing Failure section. The values of blowdown pressure can be seen in Table 2.

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Test Number	Test Date	Blowdown Pressure (psi)
1	8/9/2018	1800
2	10/10/2018	1850
3	11/14/2018	1670
4	3/1/2019	1670
5	3/6/2019	1400
6	4/4/2019	1300
7	5/7/2019	1450

Table 2: Blowdown Pressures for successful starts

The blowdown process for the 11/14/18 test is shown in Figure 3. BV8 closes and BV7 opens when P500A-P501 > dP_min, where dP_min = 20 psi. This was done to ensure the pressure on the compressor discharge was high enough to prevent flow from the blowdown tanks from flowing back into the compressor discharge.

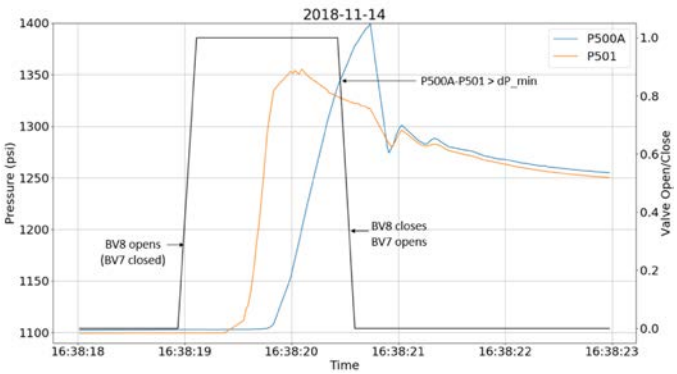

Figure 3: Blowdown start process - pressures and valve positions. Blowdown pressure = 1670 psi

This same process is shown for the 3/6/19 test in Figure 3, where the blowdown pressure was lowered to 1400psi from 1670psi.

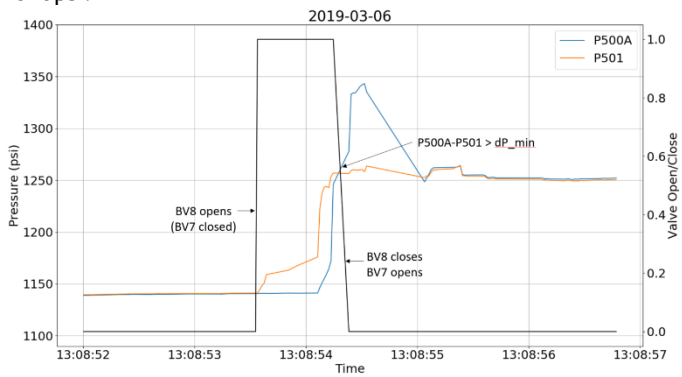

Figure 4: Blowdown start process - pressures and valve positions. Blowdown pressure = 1400 psi

As can be seen by comparing Figure 3 and Figure 4, the lower blowdown pressure reduced the pressure ramp rate of P501 from ~250psi to ~100psi. This decreases the severity of the blowdown event and reduces the possibilities that the radial foil or thrust bearings could be damaged during the blowdown process.

THRUST BEARING ISSUES

In Tests #1-3, both the thrust bearing and the radial bearings experienced rubs/failures. Figure 5 shows the compressor inlet and discharge pressures plotted versus time with spikes in pressure circled in red. These spikes correspond to thrust bearing touchdown rubs, where the rub caused the rotor to slow down momentarily.

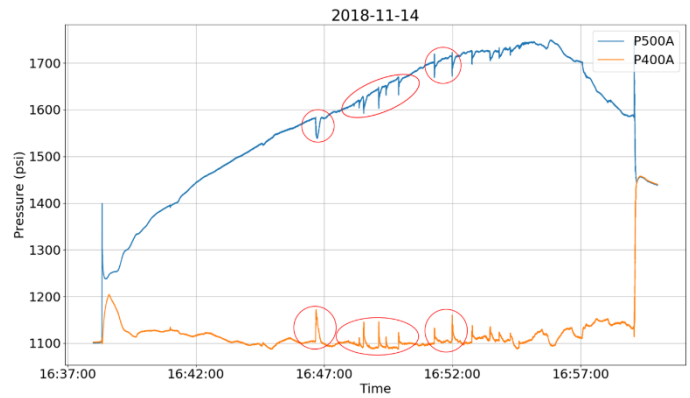

Figure 5: Compressor inlet and discharge pressure showing thrust bearing rubs

By adjusting the back pressure on the turbine using the turbine control valve (TCV) as well as adjusting the setting for Valve F in the secondary flows, the thrust balance on the rotor can be equilibrated. Valve F is part of the secondary flow control system and regulates the venting pressure on the aft side of the thrust disk of the rotor. This serves as a balance piston in the design and permits real-time adjustment of rotor thrust within a certain range. The testing on 2018-11-14 was performed with Valve F completely closed, so for the next test, both TCV and Valve F were opened using prescribed settings from the updated secondary flow analysis. Additionally, a proximity probe was added to monitor the axial movement of the shaft during the test. This allowed the proximity to be monitored and valve adjustments to be made in real time should the axial position start to change during a test. The plot of compressor inlet and discharge pressure for Test #5 is shown in Figure 6 and the proximity data is shown in Figure 7.

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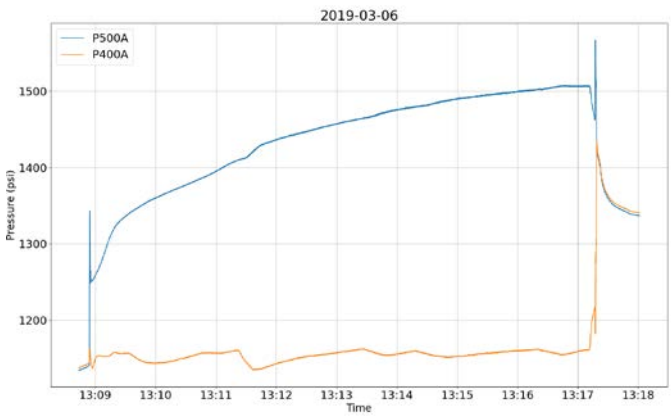

Figure 6: Compressor inlet and discharge pressure – thrust bearing issue resolved

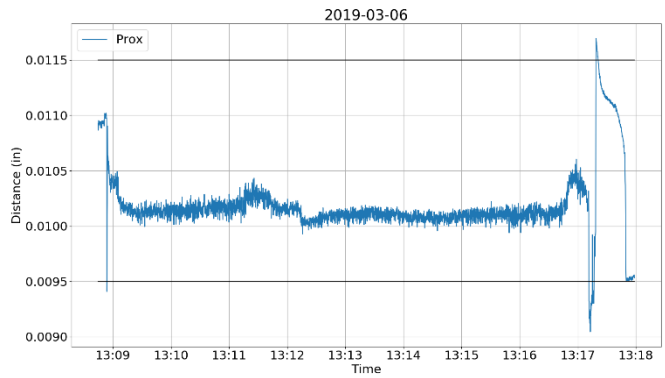

Figure 7: Proximity probe data for Test #5 showing axial position of the shaft with lines showing high and low proximity limits

As can be seen in Figure 6, there are no longer spikes in pressure due to a thrust bearing rub. Additionally, when the turbomachinery was disassembled after the test there was no indication of thrust bearing contact. In Figure 7, the horizontal lines indicate the minimum and maximum proximity before a rub occurs; that is, at 0.0095in and 0.0115in, a thrust bearing rub would be expected. The proximity sensor data shows there was very little axial movement during the test. The initial drop in distance was due to the blowdown start and the position quickly recovers to approximately 0.0101in. The other dramatic change in proximity is during shutdown. The method used to shut down in Test #5 was closing BV7 and BV6 at the same time. This causes the shaft to stop spinning abruptly and thus some axial movement is expected during this event. The proximity probe data, as well as the lack of physical evidence of thrust bearing wear, gives confidence that the thrust balance problem has been resolved. However, the turbine end radial bearing continued to fail at some point during each test until the bearing design was improved for Test #7.

RADIAL BEARING FAILURE

The core features two radial gas bearings, one at the cold end near the low-pressure compressor and the other at the hot end near the turbine. With the exception of the last test, Test #7, the aft (turbine side) radial bearing has failed and limited the duration of the test. While the root cause of those failures is still under investigation, presented here are some of the potential causes:

One hypothesis as to why the radial bearing is failing is the severe nature of the blowdown start. An attempt was made to reduce the severity of the blowdown by decreasing the blowdown pressure as described in the "Turbocompressor Start" section. Additionally, the valve which controls the blowdown start, BV8, was switched from a fast-acting valve to a slower actuated valve in attempt to further reduce the abrupt pressure change to the system.

In Tests #5 and #6, it was attempted to keep the test conditions such as starting turbine inlet temp and valve settings consistent and only change the blowdown pressure and valve actuation speed. In this way, if the bearing did not fail, then it might be reasonably concluded that the blowdown process was the culprit. However, the bearing did fail at a similar rotational speed in both tests. Plots of speed for Test #5 and #6 are shown in Figure 8 and Figure 9.

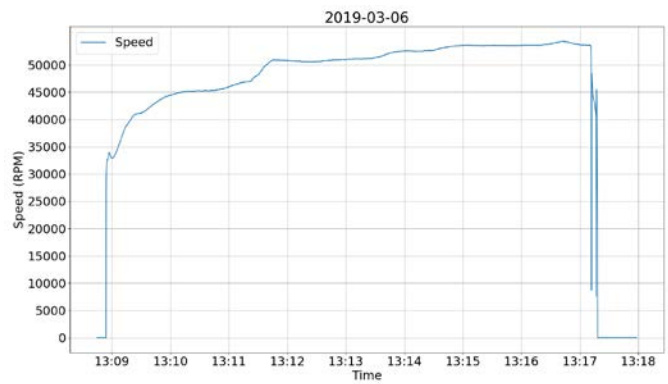

Figure 8: Rotor speed for Test #5

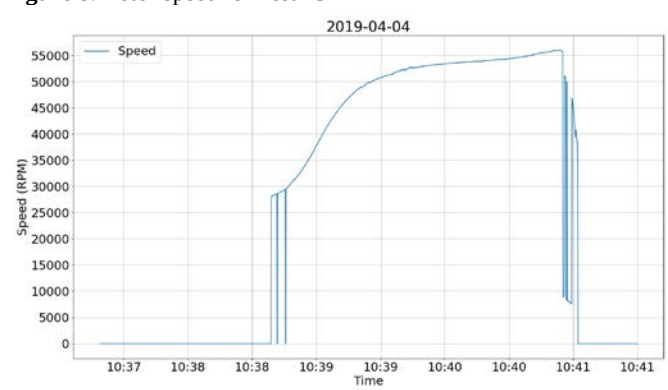

Figure 9: Rotor speed for Test #6

During Test #6, the proximity probe malfunctioned at the start of the test. This can be seen in the lack of data until approximately 10:38 in Figure 9. However, once the sensor began working the speed can be seen increasing until reaching approximately 56,000 rpm. Comparing Figure 8 and Figure 9, the plots show the bearing failed at approximately 55,000 rpm in both tests. Normally, this might indicate there is a critical speed

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near 55,000 rpm that is causing instability of the shaft and failure of the bearing. However, the physical evidence of the bearing failure only shows yielded foils at a specific clocking of the bearing; that is, the foils are not yielded uniformly around the entire circumference of the bearing. If the failure was due to a critical speed instability of the shaft, the expected result would be yielded foils around the entire circumference of the bearing instead of only at a particular clocking. It should be noted that rotordynamic analysis indicates that there are no mode crossings in the operating range above 35,000 rpm. This points to the failure being caused by a radial side load, which may be caused by non-uniform turbine inlet temperatures and/or flowrates.

The design of the PTT turbocompressor features 4 turbine inlets which are shown in Figure 10.

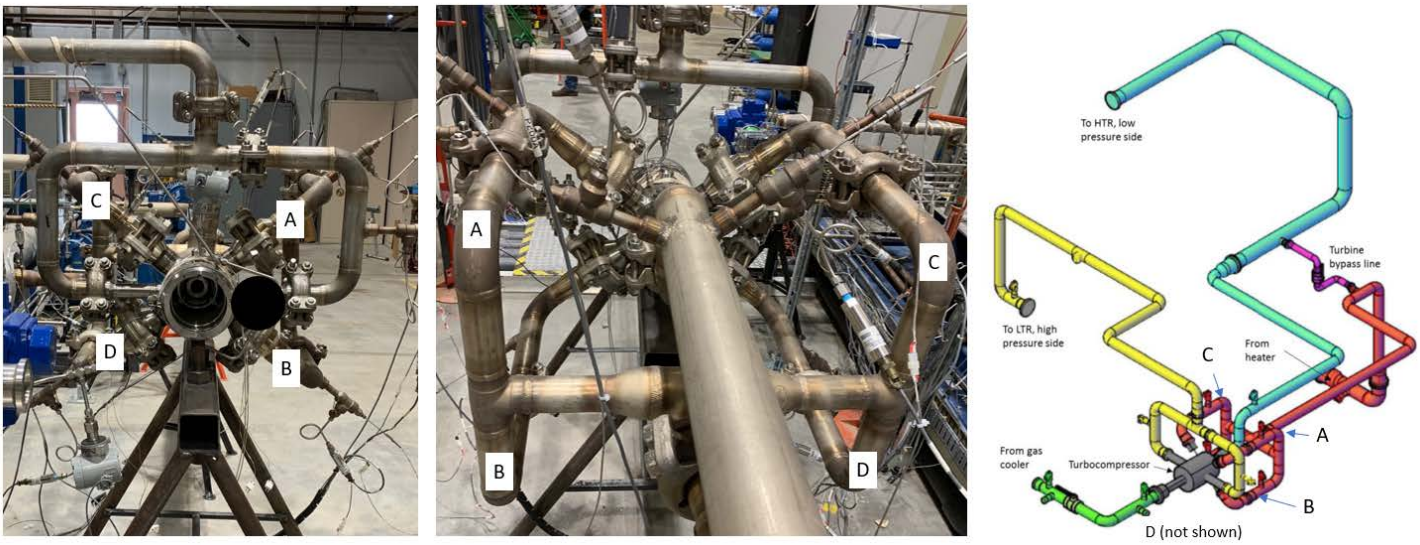

Figure 10: Left- Core case with compressor inlet and turbomachine cartridge removed. Center -Turbine inlets. Right-Isometric of piping around core

These inlets are designed to be symmetric and should have equal mass flow and temperature. However, during testing the legs appeared to be inconsistent in temperature, with variations as high as 50F. Figure 11 through Figure 15 show the TIT plots for Test #2-#6.

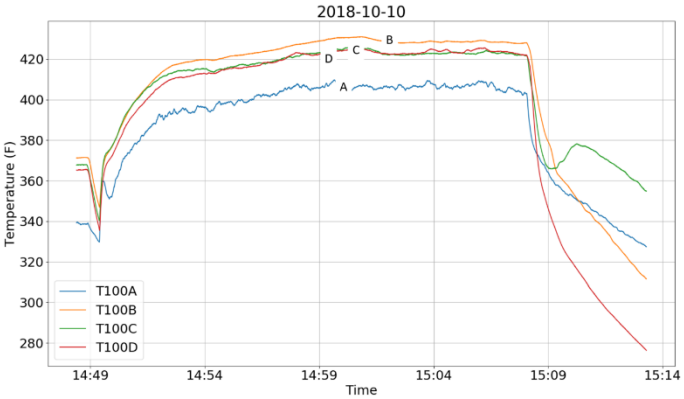

Figure 11: Turbine inlet temperature variations Test #2

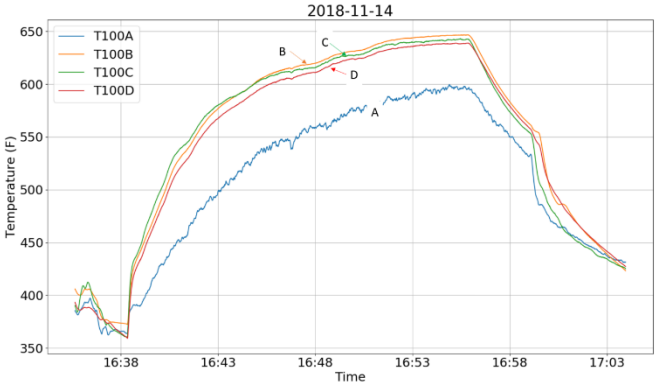

Figure 12: Turbine inlet temperature variations Test #3

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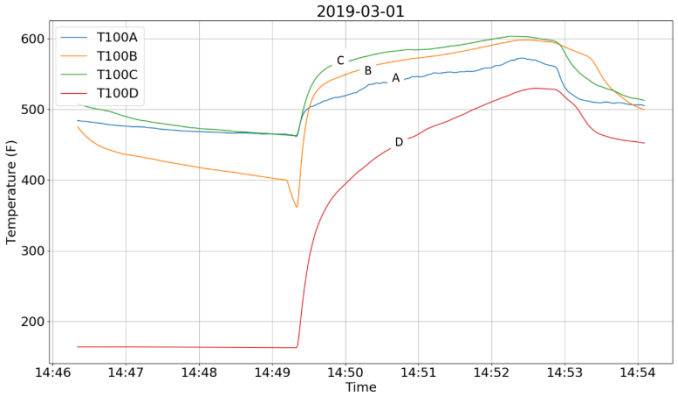

Figure 13: Turbine inlet temperature variations Test #4

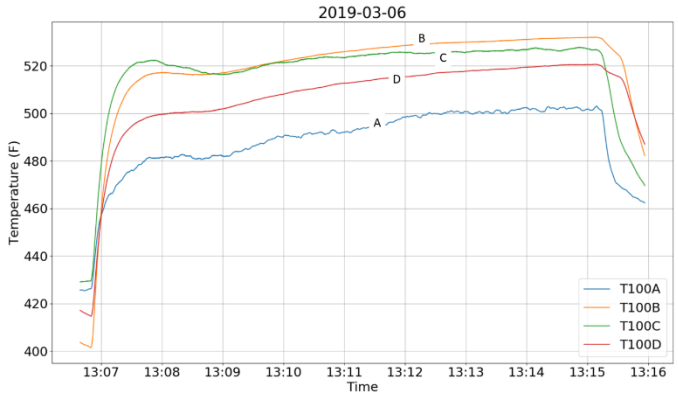

Figure 14: Turbine inlet temperature variations Test #5

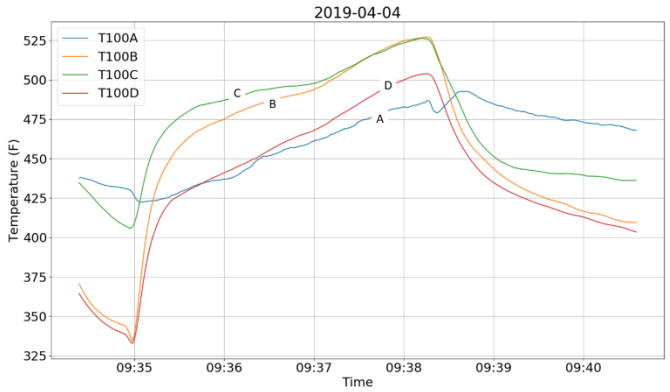

Figure 15: Turbine inlet temperature variations Test #6

As can be seen in the plots of TIT, leg "A" is often significantly cooler than the rest, and in later tests leg "D" is also cool. It is believed that Leg "D" in Test #4 had a cold, dense slug that did not have sufficient time to clear. The temperature can be seen to increase rapidly once the turbocompressor is started and the leg experiences a higher mass flow rate. The temperature profile of leg "A" shows a very jagged trace, which would indicate the instrument was not inserted into the flow adequately. When the lengths of the RTD's were measured, they were found to be within ¼" inch of each other. However, the piping was not removed to physically see insertion depth of the instrument until after Test #6. When that was done, it was found that T100A was not inserted correctly. The original placement of T100A is shown in Figure 16 on the left, and the new insertion depth is shown on the right.

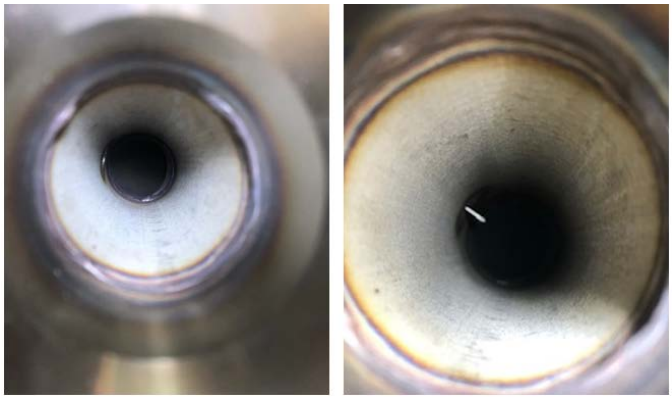

Figure 16: Insertion depth of T100A RTD; original insertion on left (not visible), new insertion shown on right.

As can be seen in Figure 16, the T100A was not inserted into the flow at all. Shown on the right in Figure 16 is the new RTD which was inserted to a depth to match the other three legs. A test was run without the turbocompressor in the loop in order to record the turbine inlet temperatures with the new T100A inserted to match the depth of the other legs. The results are shown in Figure 17.

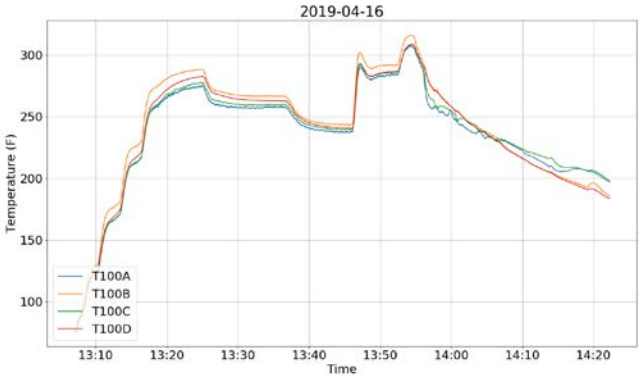

Figure 17: Turbine inlet temperatures with T100A replaced and inserted correctly into the flow

Figure 17 shows that the temperature discrepancy of T100A is no longer a problem. New RTDs were ordered and the existing T100A-D were replaced with closer detail paid to the insertion depth. Figure 18 shows the original and new insertion depths of T100A-D.

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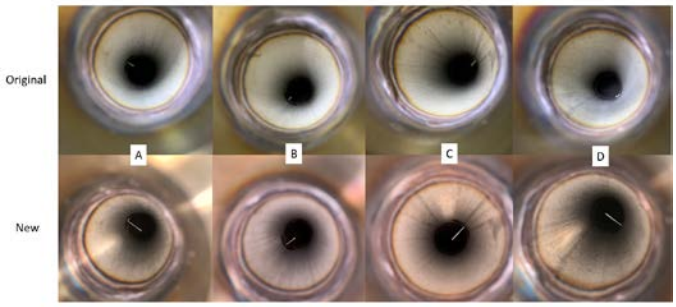

Figure 18: Original (top) and new (bottom) insertion depths of T100A-D

The next test of the turbocompressor demonstrated that the insertion depth of the RTDs was the issue and was resolved with the installation of new RTDs. Figure 19 shows the turbine inlet temperatures are in very good agreement with each other during Test #7.

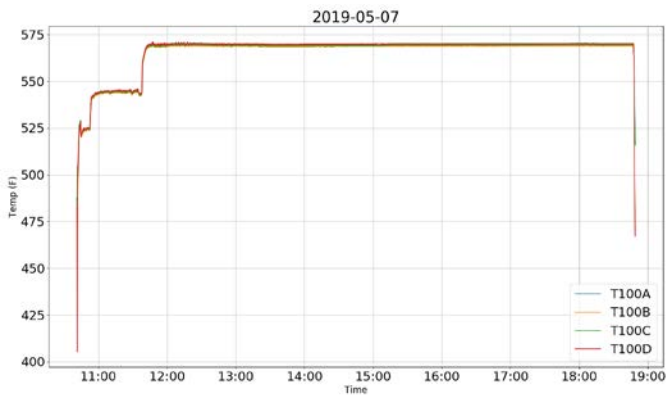

Figure 19:Turbine inlet temperatures with new insertion depth

With the insertion depth of the RTDs in mind, it is no longer believed that there is a significant temperature variation between the legs during the test. However, there does exist a temperature difference between the top and bottom legs at the start of the turbocompressor due to stratification of the fluid due to temperature/density changes. This occurs during the time between ending the preconditioning of the system and the blowdown. If the cooler, denser fluid sinks to the bottom legs during this time and then is injected into the turbine during startup, this could cause an unbalanced radial load during the blowdown event. Perhaps this is sufficient to cause damage to the radial bearing which then fails later in the test.

It was decided to increase the length over diameter ratio of the aft radial bearing to increase the load capacity as well as the damping of the bearing. This resulted in the successful operation of the turbocompressor for 8 hours with a controlled shutdown. The speed vs time for Test #7 is shown in Figure 20 and the proximity data is shown in Figure 21. As can be seen in Figure 20, there were three unsuccessful start attempts before the turbocompressor was started successfully. The conditions for the attempted starts are shown in Table 3.

Start Number	Turbine Inlet Temp (F)	Blowdown Pressure (psi)	Outcome
1	460	1330	Unsuccessful
2	500	1360	Unsuccessful
3	525	1425	Unsuccessful
4	550	1450	Successful

Table 3: Conditions for attempted starts during Test #7

It is interesting to note the peak speed for each unsuccessful start attempt in Figure 20. It is clear that increasing the TIT and blowdown pressure impart more energy to the turbocompressor, but the conditions were not sufficient to achieve a successful start until the fourth attempt. It is believed that the stiffness and damping added by the new bearing increased the energy needed to start the turbocompressor. Further tests will explore the minimum TIT and blowdown pressure that is required for a successful start with the new aft radial bearing.

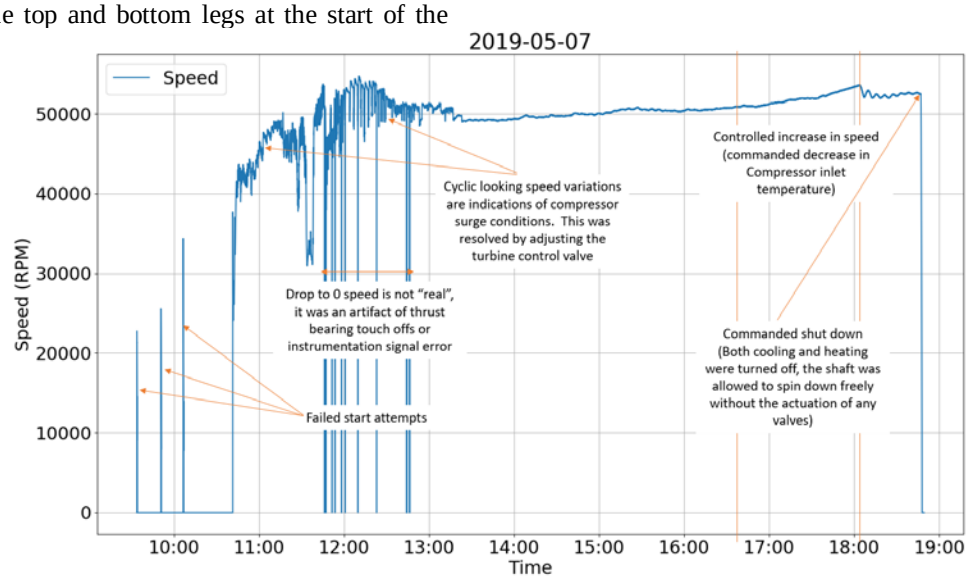

Figure 20 – Speed plot for Test #7

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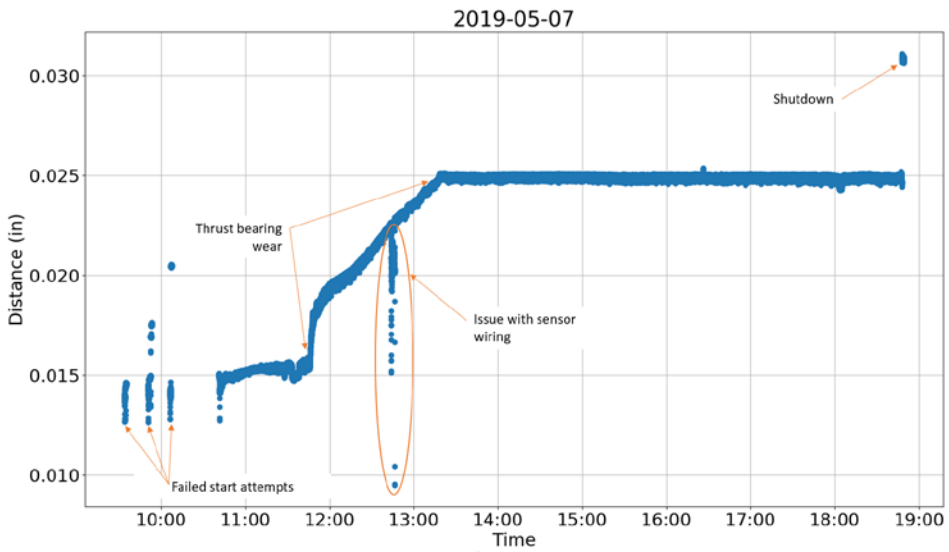

Figure 21 – Proximity probe plot for Test #7

After the successful start of the turbocompressor, it took some time to achieve steady state operation. As can be seen in Figure 21, there was very little axial movement for approximately the first hour of operation. However, during that time the operation was not steady state as the compressor was seeing surge conditions indicated by the fluctuations in speed in Figure 20. At approximately 11:45 the rotor began slowly moving in the aft direction and moved approximately 0.01 inches in the aft direction until it settled into an axial position of 0.025 inches. This movement was the result of a force unbalance and was corrected by adjusting the TCV and Valve F. The movement resulted in the wearing away of the thrust bearing which was evident in the post-test tear down of the turbocompressor. However, the ability of the thrust bearing to withstand wear and continue to perform acceptably is an impressive display of the robustness of the thrust bearing. In future testing, the valve positions will be adjusted more aggressively to attempt to arrest the movement of the shaft faster and prevent wear on the thrust bearing.

Now that the turbocompressor has been operated successfully for an extended duration of time and the ability to control the thrust balance is better understood, another test is planned to operate at higher inlet temperatures and corresponding compressor discharge pressure.

CONCLUSIONS

The SNL Brayton Lab has completed 7 tests of a new turbocompressor from Peregrine Turbine Technologies that is deigned to process sCO2 for a CBC operating at 1382 F, 6222 psi, and 12.1 lbm/s (750 °C, 42.9 MPa, and 5.5 kg/s) generating 1.0 MWe at nominally 45% thermal efficiency based on LHV of natural gas. The testing has demonstrated the successful startup using the blowdown method, resolved thrust bearing issues, and has resolved the issues with the aft radial bearing. Testing will continue and will focus on exploring the performance of the turbocompressor over a wide range of conditions and for longer durations of time. Looking ahead, Peregrine Turbine Technologies is currently constructing a complete 1MWe power system which will use the turbocompressor currently being tested at SNL. Through a Cooperative Research And Development Agreement (CRADA), SNL and PTT are working to develop a test plan for testing the 1MWe unit at PTT facilities in Maine.

NOMENCLATURE

BV	Binary Valve
CRADA	Cooperative Research and Development Agreement
DP	Development Platform
HTR	High Temperature Recuperator
LTR	Low Temperature Recuperator
NESL	Nuclear Energy Systems Laboratory
MPa	mega-Pascal
P	Pressure
psi	pounds per square inch
PTT	Peregrine Turbine Technologies
RD	Rupture Disk
sCO2	Supercritical Carbon Dioxide
SNL	Sandia National Laboratories
T	Temperature
TIT	Turbine Inlet Temperature
TCV	Turbine Control Valve
w	mass flow

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