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Component Testing of a High Temperature Dry Gas Seal

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ABSTRACT

One of the most critical challenges in high-temperature supercritical carbon dioxide (sCO_2) turbine design is thermally isolating the dry gas seal (DGS) from excessively high turbine temperatures. Overcoming this challenge has led to the development of thermal seals, which attempt to use a combination of tuned hot and cold flows to isolate the thermally sensitive DGS and create a desirable thermal gradient that achieves acceptable shaft and casing stresses across the thermal transition. A DGS able to withstand high temperatures would eliminate the design challenges associated with thermal isolation. The temperature limiting component in a DGS is the balance seal, which minimizes leakage from bypassing the fluid film gap and going around the back of the stationary seal face. The balance seal must accommodate axial movement from both thermal growth (6mm over startup and cool down) and dynamic movement due to vibrations of the stationary ring (50 μm at 175 Hz).

This paper describes a novel test rig designed to test balance sealing elements up to 700°C and pressure to 26 MPa. The rig also allows for actuation of the balance seal to mimic both the slow and dynamic movement described above. The test rig was commissioned and used to test balance seal designs up to 500°C. Finally, the results of the testing show the balance seal was capable of maintaining pressure at temperatures up to 500°C while capturing the seal leakage.

INTRODUCTION

A DGS consists of two primary rings, one rotating and one stationary. A cross-section of a DGS is shown in Figure 1. In this image, the rotating pieces are yellow and light grey while the stationary pieces are blue and dark grey. The stationary seal ring is mounted on a preloaded spring, which acts to force the rotating and stationary faces together. The rotating ring contains grooves that draw in the fluid to create a pressure-dam between the rotating and stationary faces, providing hydrodynamic lift that opposes the force of the spring. The grooves and spring are designed in such a way that when these forces balance, the rotating and stationary rings are kept between approximately 3-5 microns from each other. This tight clearance limits leakage but prevents face contact that can cause rapid failure. The action of the spring and balance sealing element helps the seal follow the axial motion and expansion of the shaft and prevents lockup of the stationary ring.

The balance sealing element exists to prevent fluid from bypassing the fluid film gap and going around the back of the stationary seal face. During operation, the rotating ring, fixed to the shaft, has two modes of axial movement. The first of these is a high-frequency vibration due to dynamic shaft motion which has an amplitude of 10 to 70 microns peak-to-peak [1]. The second mode of axial movement is a slow axial translation and is mainly caused by thermal expansions of the shaft relative to the housing [2]. The amplitude of this second, slower, mode of axial movement is on the order of several millimeters and happens during transient conditions, such as turbine start up and shutdown. As the stationary face follows the axial movement of the rotating ring, these axial motions cause the stationary seal face to be in constant motion, necessitating a balance sealing element that can withstand consistent, high-frequency, small-scale sliding motion, as well as longer duration axial translation.

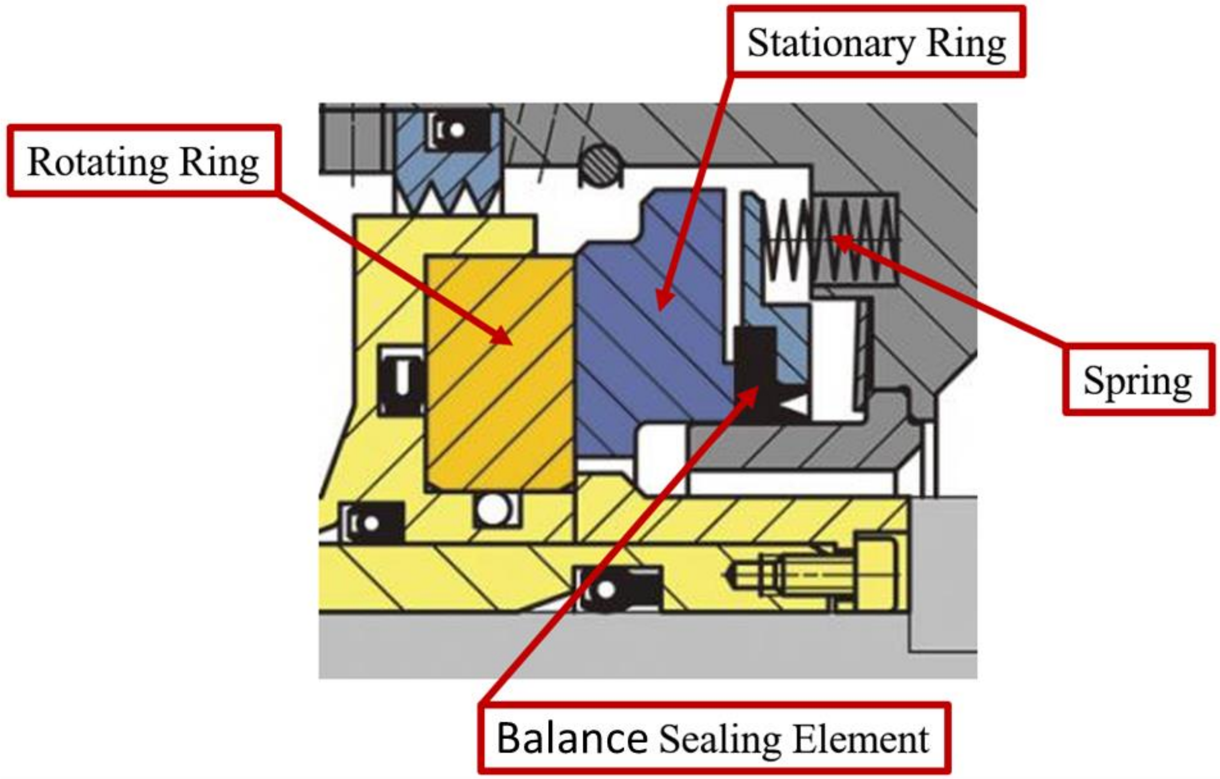

Figure 1. Cross Section of a Dry Gas Seal

The team used previous modeling studies and turbine designs to set the requirements of the DGS. Typical inlet and outlet conditions of an sCO_2 turbine based on cycle optimization are 25.6 MPa at 700°C and 7.4-8.5 MPa and 500°C[3–6]. These temperatures are well above the current limits of dry gas seals, which are usually 200°C. The temperature limiting component is the balance sealing element (typically an elastomer or polymer). Therefore, the team first designed and tested different balance sealing elements at high pressure and temperature in a CO_2 gas. Although the focus of the current project was designing and testing seals at the turbine outlet conditions, the test rig was designed for 25.6 MPa at 700°C for future design. The design requirements are shown in Table 1. The target friction leakage and friction force are 120 Nl/min and 1,000 N respectively. The target friction force was set based on an analysis of the balance of forces of the DGS faces. During operation, the DGS creates a pressure dam that applies a separating force. The springs provide a counter force to close the faces. The design of the DGS allows for the balance of forces to maintain a desired gap length between the faces. If the friction force is too high, the balance of forces will not allow the faces to maintain the desired gap length. Therefore, the target friction force was set based on the balance of forces, with a engineering safety factor. The leakage target is slightly more arbitrary. There is a tradeoff for having more leakage but removing the rotordynamic length, and every application will be different. That being said, the target leakage was set based on typical cold flows required to maintain a thermal management zone. The leakage target was also set such that the leakage of the balance seal did not more than double the total leakage of the DGS faces.

Table 1. Design Requirements for the Balance Sealing Element

Requirement	Unit	Value
Temperature	°C	500
Pressure	MPa	7.4
Leakage	Nl/min	120
Friction	N	1,000
Range of Movement (thermal expansion)	mm	6.35
Range of Movement (dynamic oscillations)	µm	50

Figure 2 shows the test rig setup used for testing the seals. The main pressure housing was rated to contain 25.6 MPa at 700°C. Inside the housing are inserts that contain the balance seal and balance sleeve. A ring that mimics the static dry gas seal is actuated via a plunger and actuator. The balance seal must seal against the balance sleeve at the OD, and the stationary ring on the axial face. The 16 kW radiant heaters add heat to the test rig. Thermocouples are placed at multiple locations on the exterior housing as well as the void that contains the pressurized fluid. The test rig is heated until the interior void reached 500°C.

Figure 3 shows the process and instrumentation diagram of the test setup. The gas is injected at two locations, 180° from each other. Table 2 shows the accuracy and type of all sensors used in the test setup. Gas leakage from the top seal is collected in a volume and tubed to a thermal based mass flow meter, M-250SLP, with an accuracy of 0.5% (FM-02). The flow into the test rig is a Coriolis flow meter, CMF050, with a 0.25% accuracy (FM-01). This flow meter picks up the total flow into the test rig, and the bottom seal leakage is assumed to be the difference between the two flow meters. It is noted that there is additional flow captured in this measurement, namely, the flow that is vented out the electronically controlled pressure regulator. N-type Thermocouples and RTDs were used to measure the temperature at multiple locations on the housing of the test rig. Two N-type thermocouples were also placed inside the housing and measured the pressurized gas. The pressure is measured at the inlet of the test rig, with a Rosemount 3051 transmitter that has a 0.04% of span accuracy. For the high temperature tests, the test rig was heated until the interior thermocouples read 500°C (the housing thermocouples read a higher temperature).

Table 2. Sensor Description and Accuracy

Sensor PN	Sensor Type	Accuracy
M-250SLP	Thermal Mass flow	0.5%
CMF050	Coriolis Mass Flow	0.25%
Rosemount 3051	Pressure	0.04%

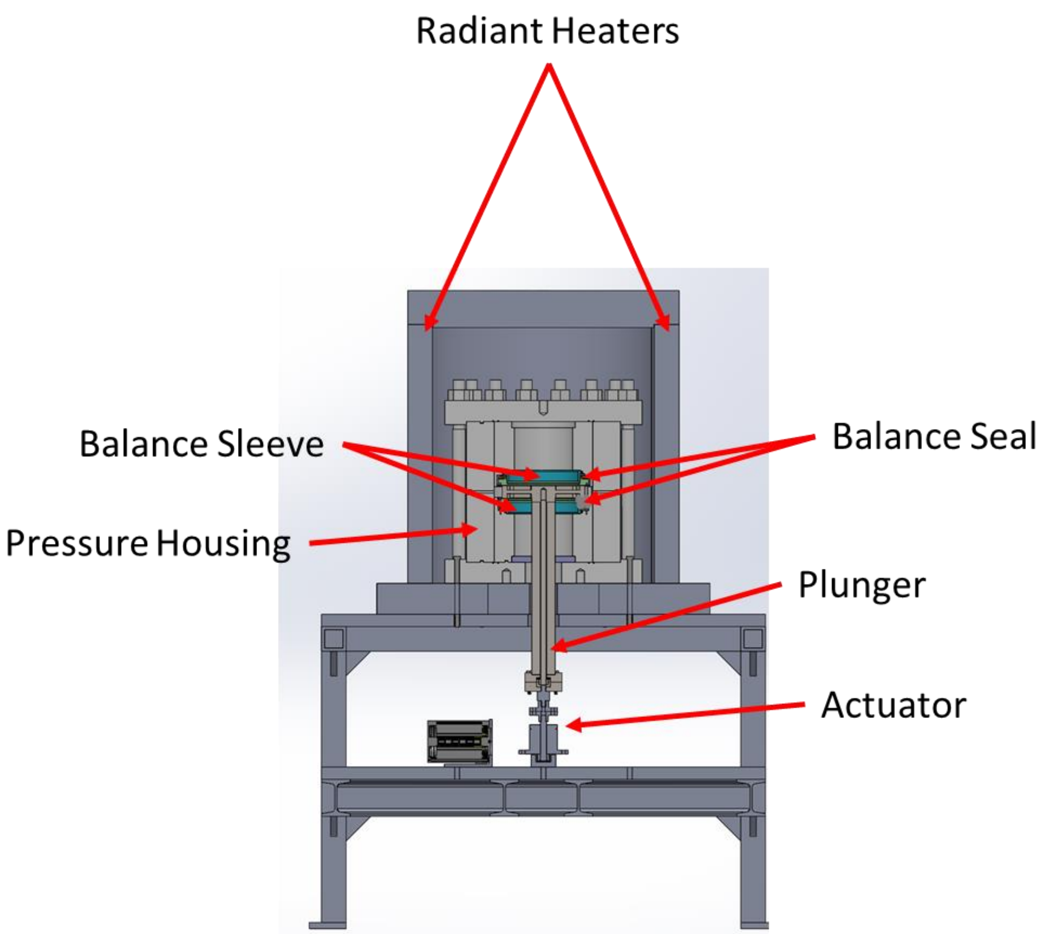

Figure 2. Test Rig Setup

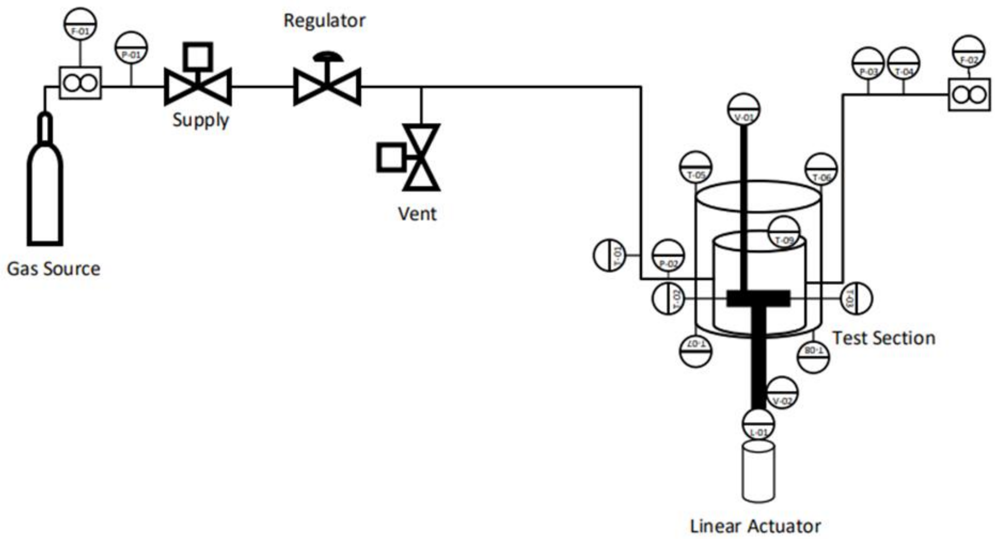

Figure 3. Process and Instrumentation Diagram for the Test Rig

RESULTS AND DISCUSSION

The team tested four types of balance seals with different designs, two balance sleeves, and two backing seals (static seal that seals the balance sleeve to the housing) as described in Table 3, which gives a summary of all the testing performed. Balance Seal Type 1 and Backing Seal 1 are both typical elastomer/polymer seals rated to 200°C. The leakage in the table is rated 1-5 where 1 is the baseline measurement, 2 means it meets the target +/- 15%, 3 is between the target and 2X the target, 4 is 2-5X the target, and 5 is greater than 5X. The force rating in the table is rated 1-5 where 1 is the baseline measurement, 2 means it meets the target +/- 15%, 3 is between the target and 2X the target, 4 is 2-5X the target, and 5 is greater than 5X.

Table 3. Summary of Testing Performed

Test Nr	Balance Seal	Balance Sleeve	Backing Seal	Temp	Leakage rating	Force rating
1	Type 1	Sleeve 1	Seal 1	Ambient	1	1
1.1	Type 1	Sleeve 1	Seal 2	Ambient	2	--
2	Type 3 D1 M1	Sleeve 2	Seal 1	Ambient	4	2
3	Type 4 D1	Sleeve 2	Seal 1	Ambient	5	4
4	Type 3 D1 M2	Sleeve 2	Seal 1	Ambient	4	3
5	Type 2 D1	Sleeve 1	Seal 1	Ambient	3	3
6	Type 2 D2	Sleeve 1	Seal 1	Ambient	3	3
7	Type 3 D2	Sleeve 1	Seal 1	Ambient	2	2
8	Type 3 D2	Sleeve 2	Seal 1	Ambient	2	2
9	Type 3 D3	Sleeve 2	Seal 1	Ambient	3	3
10	Type 3 D2	Sleeve 1	Seal 2	500C	2	2

Figure 4 shows the baseline test (Test 1) using the Balance Seal Type 1, Balance Sleeve 1, and Backing Seal 1. This gives an example of how the testing is performed. The team tested multiple pressures for each test. At each pressure, the seals were actuated and the force was measured (meaning the friction force). During the pressure condition, and while actuating, the leakage is measured. Once multiple tests were performed, the data was post-processed to compare different seal designs.

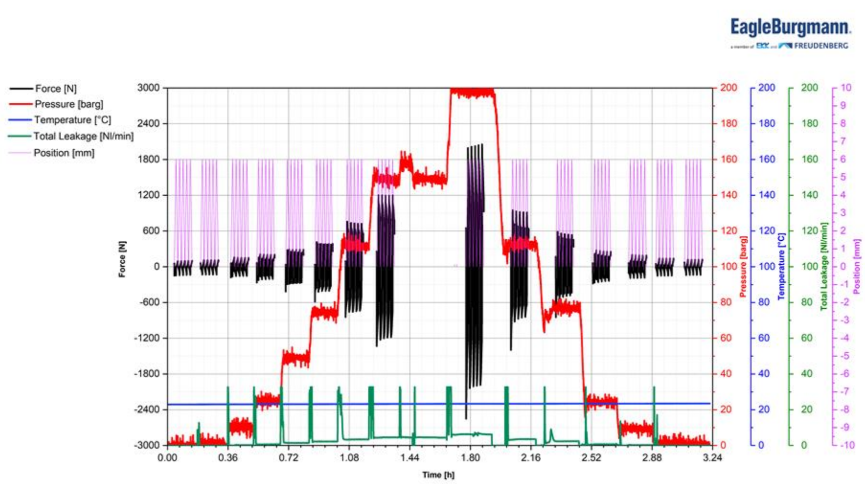

Figure 4. O-ring Baseline Test

Figure 5 and Figure 6 show the testing results for the ambient temperature tests compared to the target leakage and friction, respectively. These tests were performed with air as the fluid. Balance

Seal Type 1, which is a typical elastomer/polymer seal used for current balance sealing elements, shows very small leakage rates. The ability of these seals to deform allows good contact on both the OD of the balance sleeve and the face of the static ring. But this also limits their ability to achieve high temperatures. Test 1.1 compared the leakage from the two different types of backing seals (static seal that seals the balance sleeve to the housing). Seal 2, which is rated to 500°C, had additional leakage when compared to Seal 1. Seal 2 added 12.5 Nl/min at 80 bar (or 0.15 Nl/min-bar). The excess leakage most likely occurred due to the design of the seals groove, and has the potential to be reduced in a final design of a dry gas seal. Nonetheless, the final test results of the balance seal at temperature do include this excess leakage.

The leakage results show that Seal Type 4 D1 performed worse, with leakages ranging from 45-65 Nl/min-bar and a friction force in excess of the target. Seal Type 3 D1 showed better performance than Type 4 with leakages ranging from 5-7 NL/min-bar, and showed friction forces near the target. Seal Type 2 D1 showed better leakage performance with leakages ranging from 1.6-2.5 Nl/min-bar. Additionally, Seal Type 2 had large friction during actuation. The team identified the flaws in Seal Type 2 and 3 during initial testing and performed a redesign. Seal Type 3 D2 and Seal Type 3 D3 showed improved leakage rates ranging from 1.2-2 Nl/min-bar. All of Seal Type 3 showed friction measurements near or below the friction force target. The redesigned Seal Type 2 D2 did not show improved leakage or friction results. Ultimately, Seal Type 3 D2 was selected to be tested at full pressure and high temperature.

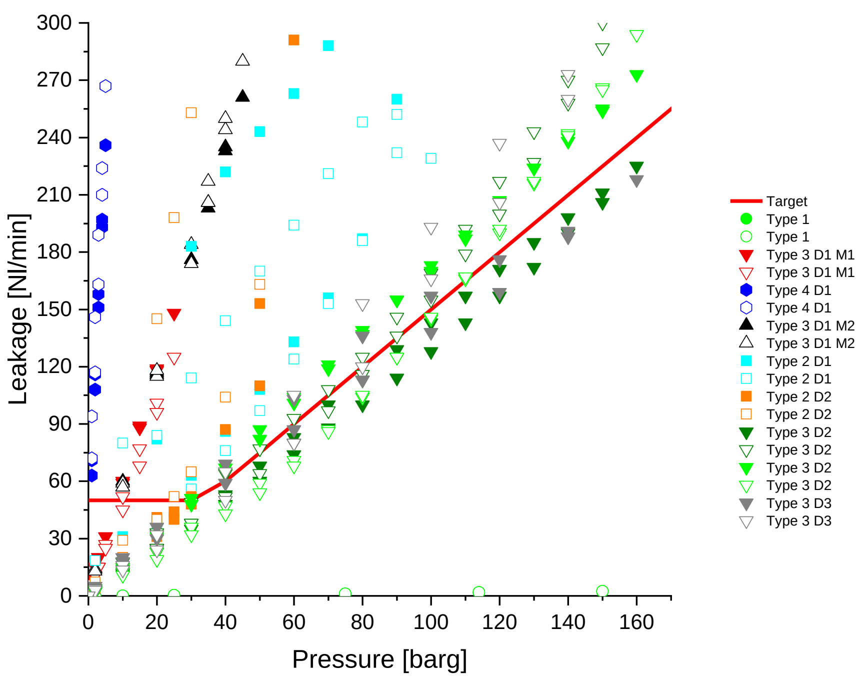

Figure 5. Ambient Temperature Leakage Results

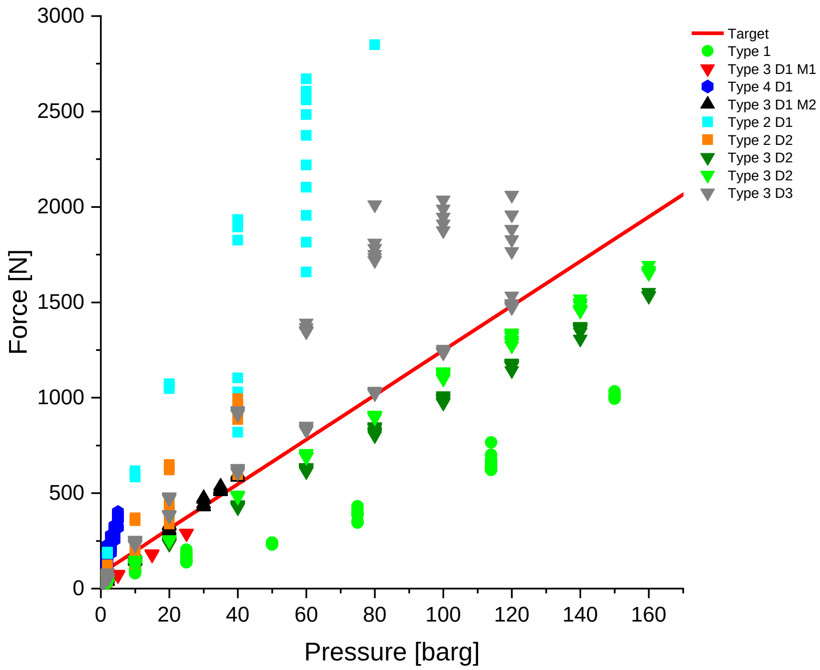

Figure 6. Ambient Temperature Friction Results

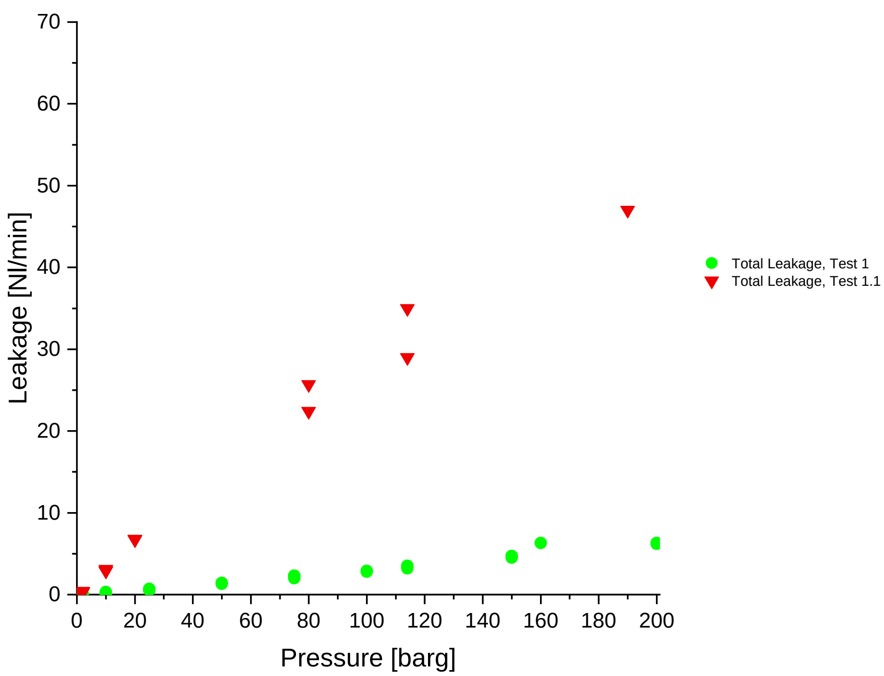

Figure 7. Test 1.1, Leakage Comparison Between Backing Seal 1 and Seal 2.

The ambient seal tests were also compared using a statistical analysis. Figure 8 and Figure 9 show the box and whisker plots for each of the nine ambient tests, which show the mean, upper quartile, lower quartile, and outlier data. These data are normalized by the pressure and force respectively to get a larger number of samples for the statistics. When comparing type 3 and 4 (tests 2-4) seals to type 2 (tests 5 and 6) seals, we point out the statistical difference. These tests mostly show that the data is skewed toward the lower quartile. Type 3 and 4 seals held a smaller standard deviation, and therefore a smaller variation between the upper and lower quartile. Although the type 2 seals performed better than the initial Type 3 seals, they also had a significantly larger variation.

Tests 7-9 show the testing of the improved type 3 design. The variation of the updated designs remains small, with little variation between the upper and lower quartile, similar to tests 2-3. Those statistics show that the mean values did meet the target leakage, although the upper quartile did fall slightly outside the target leakage. The type 3 D2 (Test 7 and 8) showed better results that type 3 D3 (test 9), whose mean fell above the target leakage rate. This variation is fairly small as the upper quartile of test 7 and 8 is above the lower quartile of test 9.

Similarly, the friction was compared in Figure 9. The type 4 (test 3) and type 2 (test 5 and 6) showed the widest variations of the upper and lower quartile, and were well above the target friction force. Type 3 D1 (test 2) showed that the mean was below the target, but the upper quartile remained above. Type 3 D2 (test 7 and 8) and Type 3 D3 (test 9) had very narrow variations between the upper and lower quartiles. The complete data set for the Type 3 D2 seal remained below the target, while the complete data set for the Type 3 D3 seal remained above. The Type 3 D3 seal showed higher friction than when compared to the baseline seal (test 1), but still fell within the functioning limits for the dry gas seal operation. The ambient temperature tests showed that the Type 3 D2 balance seal performed best, and therefore, was selected for high temperature testing.

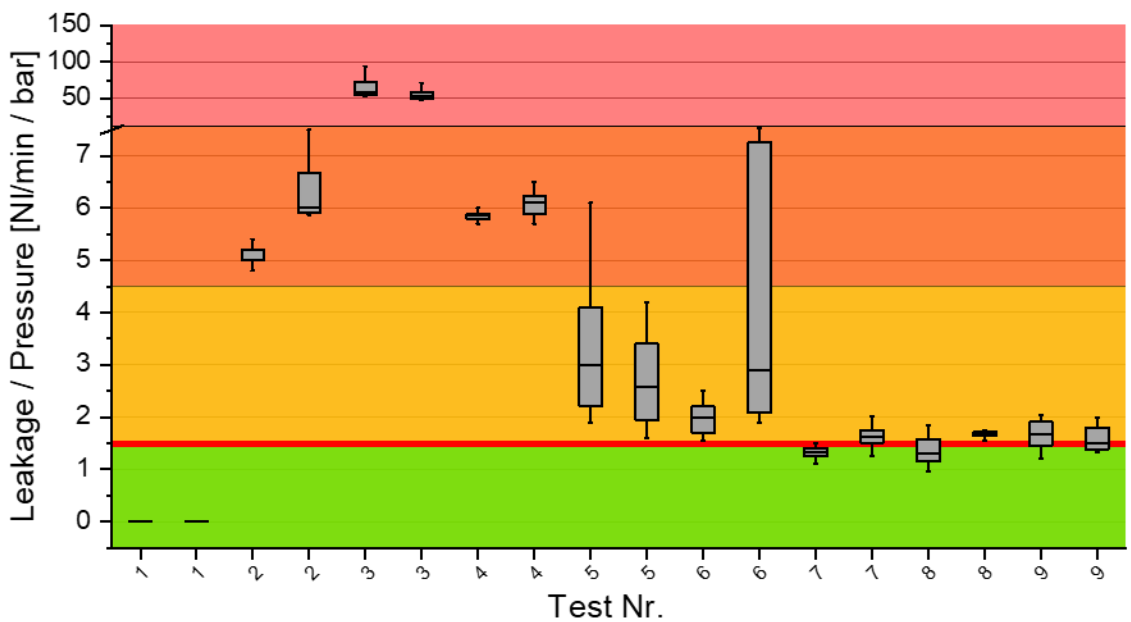

Figure 8. Statistical Comparison of the Leakage for the Ambient Tests

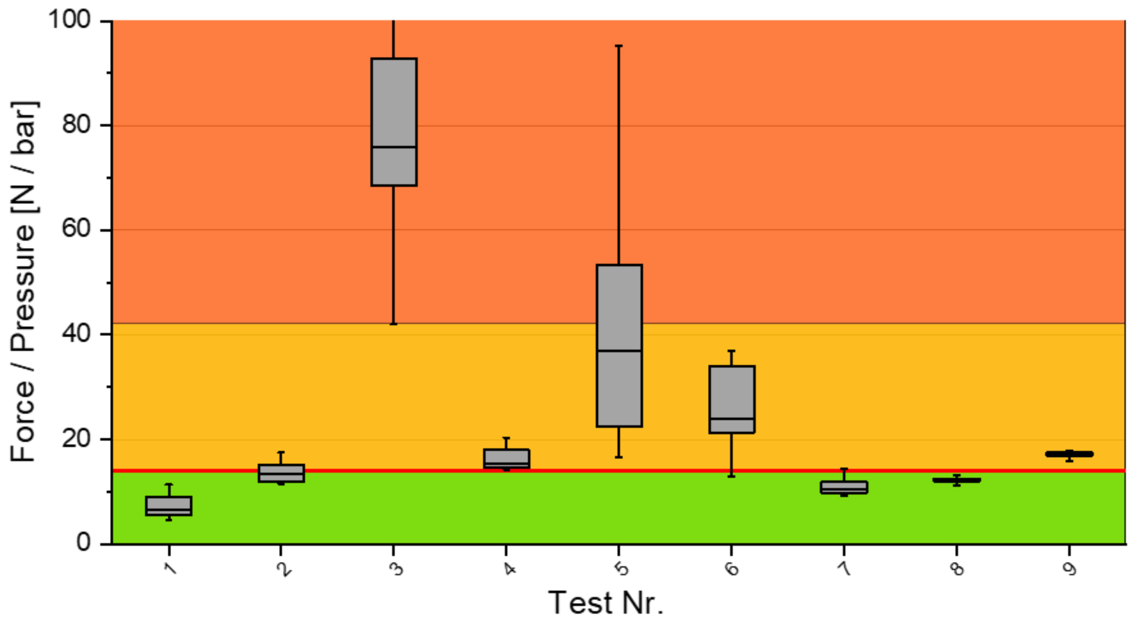

Figure 9. Statistical Comparison of the Friction for the Ambient Tests.

Figure 10 and Figure 11 show the results of the high temperature CO_2 test for Seal Type 3 D2 leakage and friction respectively (Balance Sleeve 1). The figures compare the seal results with the original ambient temperature air tests. An initial ambient test was performed using CO_2 to compare with ambient air. The test was performed with both Seal 1 and Seal 2 as the backing seal (seals the balance sleeve and balance seal housing). The ambient CO_2 results were above the target, and the results show the difference in leakage between Seal 1 and Seal 2 as the backing seal (discussed above in Figure 7). The ambient CO_2 tests also show an inflection point as CO_2 reaches the critical pressure.The test rig was heated to 500°C and the leakage and friction were measured at full temperature. The increase in temperature reduced the leakage of the seal. A contributing factor is likely related to the increase in viscosity of CO_2, which increases by about 2X from 25°C-500°C. The final leakage was 0.706 NL/min-bar with a 99% confidence interval of 0.695-0.722 Nl/min-bar. Additionally, temperate had no significant increase on the friction force of Seal Type 3 D2.

Additionally, tests were performed with the Balance Sleeve 2. The material properties of Balance Sleeve 2 and Seal Type 2 D3 were such that the difference in thermal growth caused additional leakage above the target. Additionally, Balance Sleeve 2 had a failure at high temperature.

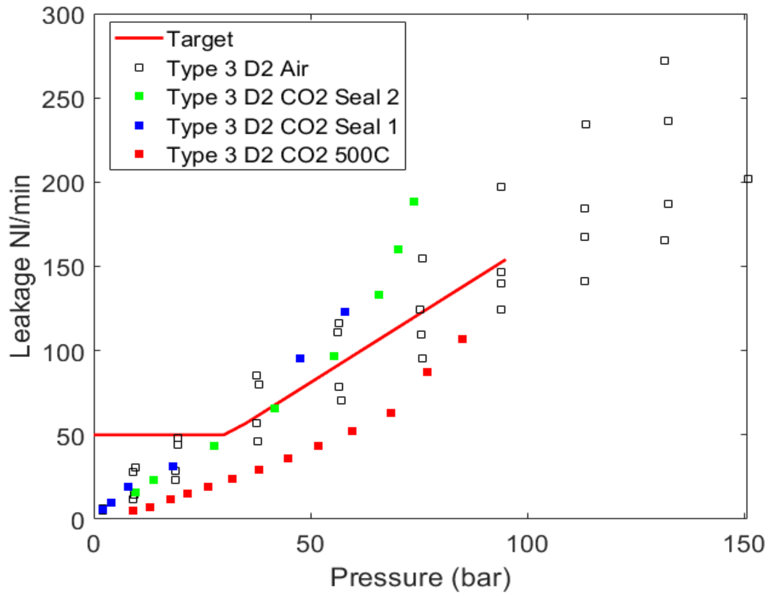

Figure 10. High Temperature Leakage Test

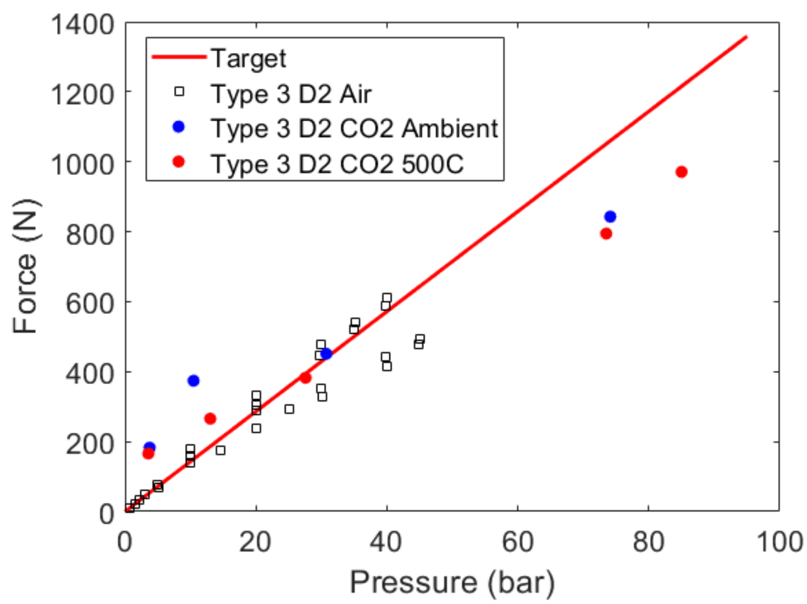

Figure 11. High Temperature Friction Test

The team also performed high temperature testing of the dynamic/high speed movement of the seal. The team performed a 36 hour test while actuating the seals between 40 and 70 um peak to peak at a frequency of 175 Hz. The movement was measured with vibration probe. An integration and Fast Fourier Transform were performed to quantify the peak to peak displacement. Figure 12 shows the measured leakage during heat up (while applying the dynamic movement). The leakage vs temperature results show how the leakage reduces with temperature, as the heating of the test rig occurred during the first 12 hours. Once at steady state temperature, the leakage was collected for 24 hours while dynamically moving the seals. The results are shown in Figure 13. The leakage did not significantly change during the 24 hours at high temperature, even with the dynamic movement. The seals were disassembled and showed no significant signs of damage.

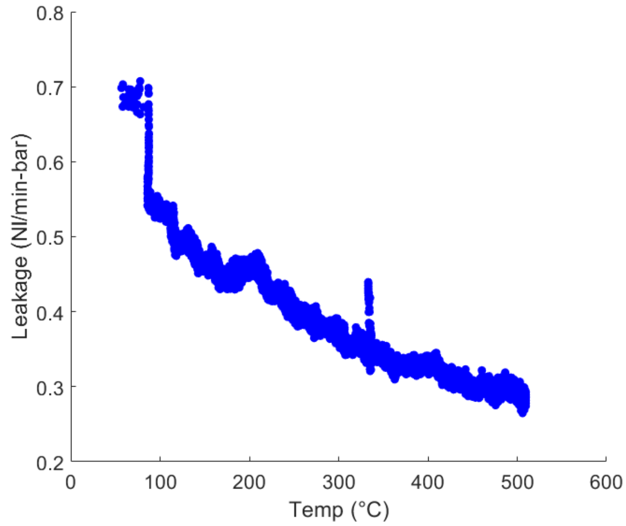

Figure 12. Leakage measurement during heat up for the long duration leakage test.

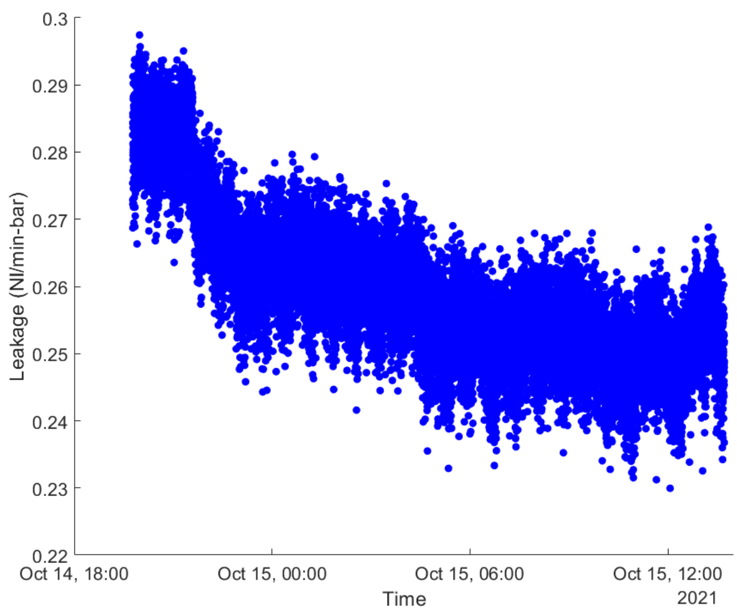

Figure 13. Leakage During the Long Duration Dynamic Movement Test

CONCLUSIONS

The Project Team was successful in designing and testing a balance seal capable of 500°C at 7.4 MPa that met both the leakage and friction target. The leakage was not affected by high frequency movement expected for a typical turbine design. It should be noted that the high temperature balance seals have an additional leakage penalty compared to the baseline low temperature balance seal, which utilizes elastomer seals. The advantage of the high temperature DGS, even with some additional leakage, comes with the ability to eliminate rotodynamic length from a thermal management zone used to reduce the gas temperate to an acceptable limit. Alternatively, the excess rotodynamic length could be used to add additional stages of expansion, which has the potential to improve isentropic efficiency 2-4%. This adds both robustness and flexibility to current sCO_2 turbine designs.

Moving forward, the project team plans to incorporate the balance seal into a full DGS assembly. The full DGS will be tested at full speed (rotational), full pressure, and full temperature.

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