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Development of a Dry Gas Seal for high-temperature supercritical carbon dioxide (sCO2) turbines

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ABSTRACT

Sealing high-temperature supercritical carbon dioxide (sCO2) in turbines is a significant challenge. Current designs utilize thermal seals to safeguard the dry gas seal (DGS) against excessively elevated turbine temperatures. The thermal seals utilize hot and cold flows to create an optimal thermal gradient on the shaft that lowers the gas temperature while not creating excessive thermal stress levels in the shaft. Thermal seals are required because current DGS technologies are limited to approximately 200°C. But a DGS able to seal at higher temperatures would simplify the intricate task of thermal isolation in design. In addition, the high temperature DGS would reduce shaft axial length, allowing for extra turbine stages, or possibly higher speeds. This research paper introduces preliminary test results of a DGS that has been designed to operate at gas temperatures of 500°C. The paper outlines the iterative testing approach, static component test results and dynamic test results for the full DGS assembly. The DGS is tested at 21,000 rpm and 89 bar supply pressure. The working fluid is air-helium mixture, and the seal reaches a primary seal temperature of 250°C. The leakage of the seal is comparable to current DGS technologies.

INTRODUCTION

Dry gas seals started to replace oil seals in large turbomachines as far back as the 1950s [1].

Due to their contactless dynamic operation between rotating and stationary units their operating range rose from about 40 m/s up to 250 m/s of circumferential speed still consuming lower driver power. Moreover, with a sliding face design DGS technology shows a clear advantage over throttle seals regarding to low leakage rates at higher pressures (often above 200 bar).

Figure 1 shows the major components of a single DGS. The yellow components are attached to the machine shaft and thus are the only ones which undergo rotating motion. While in dynamic operation a narrow axial gap filled with sealing gas is created between the rotating ring and stationary ring due to a balance of opening (from gas compression on sealing gap) and closing (from hydrostatic pressure and spring deflection) forces. Sealing gap has a major influence on DGS leakage rates, but a gas lubricated interface must be ensured for seal faces integrity during rotation.

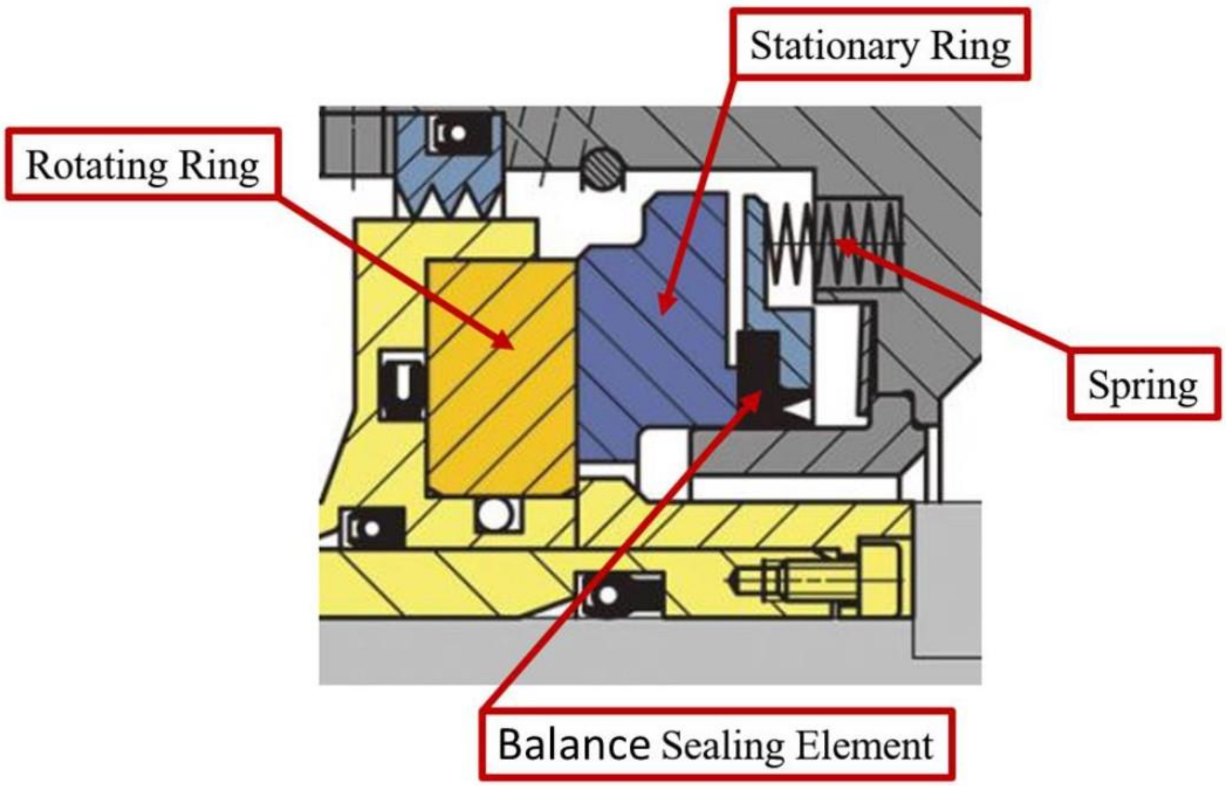


Figure 1: Cross-section of an ordinary DGS

Despite the rotating and stationary rings being of high-temperature resistant materials like ceramics, secondary sealing elements are usually made of elastomers or polymers, and limit operating temperature of ordinary DGS to about 200 °C. Temperature performance in DGS have been heavily studied in the literature. Golubiev [2] presents one of the first models for a mechanical seal. Energy is primarily added to the DGS from the windage between the rings. Both Golubiev [2] and Li [3] use a finite element method and assume a steady heat generation from the fluid of the DGS. From there, models began to look at conduction of the heat through the housing [4-5] and convection through the wetted surfaces in the seal cavity [6]. Hong et al. [7] use CFD to model the film heat generation, in an attempt to understand the effects of higher temperatures on the ring seals. Some work has even been done to evaluate the thermal performance of DGSs in sCO2 [8-10]. While there have been several studies predicting DGS performance at elevated temperatures, there are no experimental results in the literature.

Development of a new concept of dynamic sealing element

The most critical component for increasing the design temperature of DGSs (for being able to work on sCO2 turbines without thermal management) is the dynamic sealing element (also referred as balance seal, Figure 1). This element minimizes leakage around the back of stationary ring through the gap formed between the stationary ring and the balance diameter. Assembled as a rod seal around the balance diameter it is subjected to dynamic reciprocating motion of two types: quasi-static axial movement due to machine shaft thermal growth (up to 6 mm depending on shaft size and temperature span during startup and cool down) and vibration of the stationary ring transmitted from machine shaft through rotating ring (up to 70 $\mu m_{p\text{-}p}$ mainly at machine speed frequency) [11, 12]. Counteracting these axial movements of the stationary ring, the friction force between dynamic sealing element and balance diameter has a large influence on the balance of closing and opening forces that form the sealing gap, and consequently on its stabilization and on DGS leakage rate.

Designing a dynamic sealing element with proper friction force and the required leakage tightness for a high-temperature sCO2 turbine becomes therefore a challenging task. This work uses a concept of dynamic sealing element presented and tested in [13]. Since this is the first time a high temperature balance seal will be used in a DGS, the tests firstly evaluate its performance as a single component and afterwards as a part in a DGS for high-temperature sCO2 turbines.

Test conditions for dynamic sealing element

Figure 2 shows the test setup for the dynamic sealing elements. It is designed for a static test of a single dynamic sealing element. Test setup consists of a housing, ordinary DGS stationary ring and balancing sleeve, as well as a cover replacing a DGS rotating ring. These four components define two pressure chambers: gas supply and gas vent. Sealing among these components is performed by a polymeric seal, O-Rings and the dynamic sealing element to be tested. Air is supplied at room temperature on port A. Upstream a pressure control valve and a pressure transmitter respectively regulate and indicate the gas supply pressure. Downstream a flow transmitter indicates the standard volumetric leakage through port B.

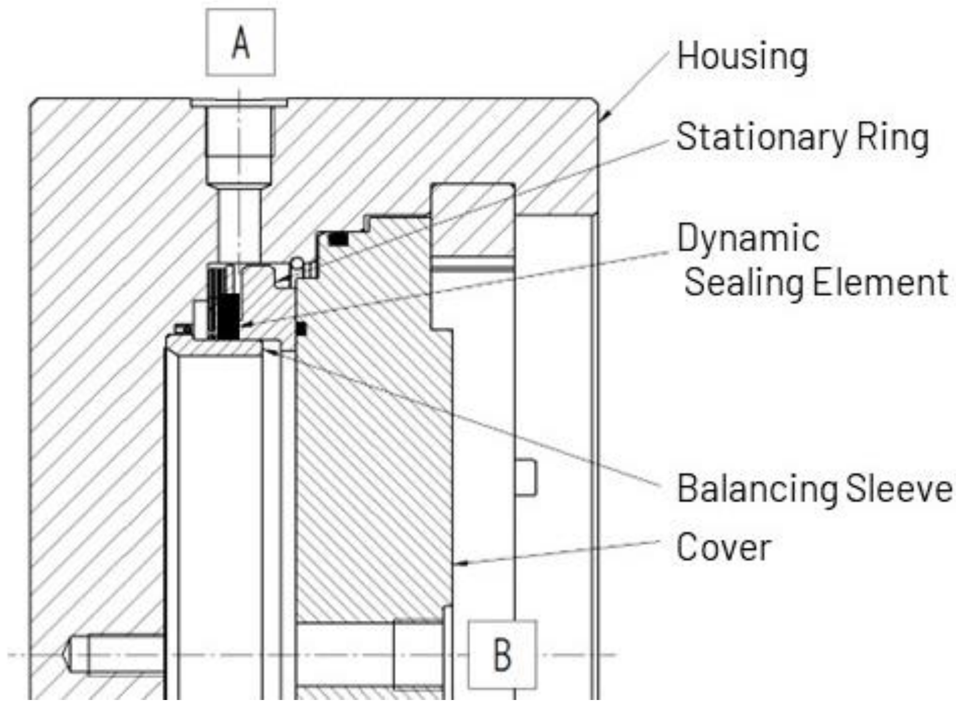


Figure 2: Test setup for dynamic sealing element

Despite not being the same parts used on the high temperature DGS, ordinary balancing sleeve and the stationary ring present the same profile at the interface with the dynamic sealing element. Leakage through both sealing elements (housing to balancing sleeve and cover to stationary ring) are also present in the flow transmitter readings but considered neglectable compared to leakage through dynamic sealing element.

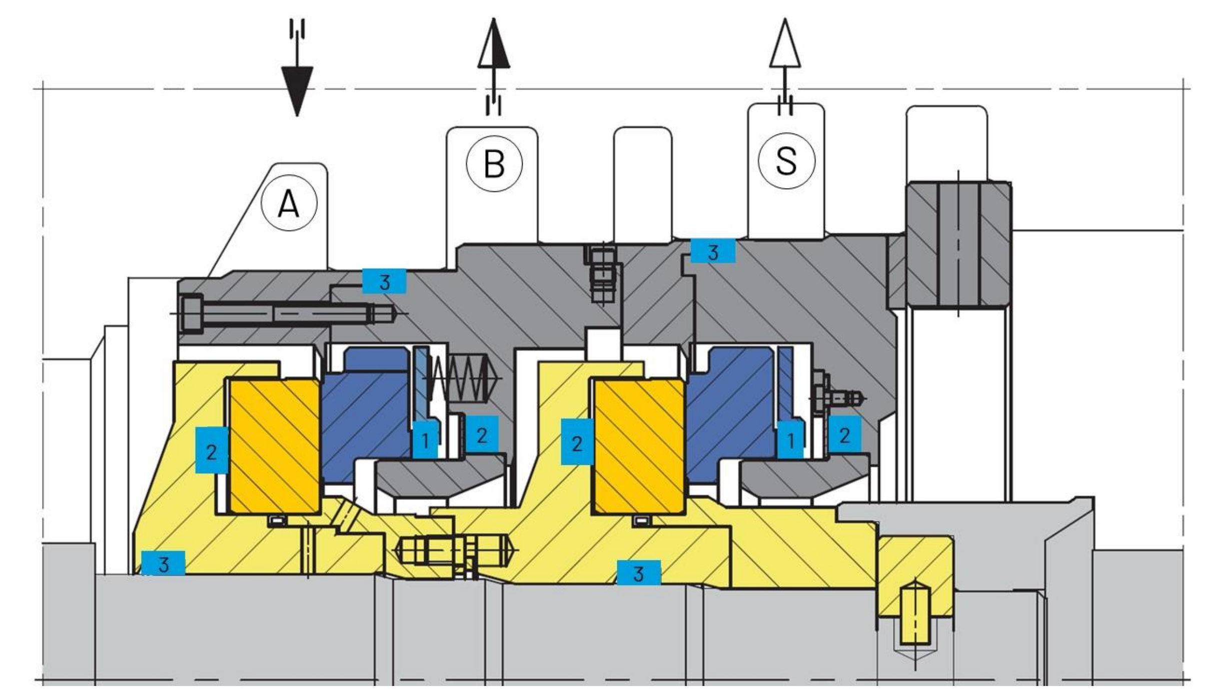
The test procedure consists of pressurizing the test setup in defined pressure steps to the static design pressure of dynamic sealing element (1650 psig – 113,8 barg) and depressurizing it using the same steps. Pressure is held at each step for at least one minute or until leakage stabilizes (no visual time variation). Leakage is noted immediately before switching to next step. The following pressure steps were used: 1, 2, 3, 4, 5, 10, 20, 40, 60, 80, 100, 113.8 barg.

Test conditions for DGS

Due to the technical difficulty of designing a high-temperature DGS the test procedure has been divided into two phases:

- Phase 1: tests at EagleBurgmann with helium-air mixture supplied at room temperature,
- Phase 2: tests at SwRI with sCO_2 supplied at up to 500°C.

Several design changes have been required to allow an upgrade from ordinary DGS design temperatures (about 200 °C) to the one required by this project (600°C). Since the impact of such novel design features on seal performance could not be exactly predicted, test phase 1 was split into several test steps using an iterative approach. This approach uses only one new design feature to the prototype as a replacement for the conventional DGS feature at a time and repeating the test procedure, keeping the successfully tested new features on the seal. This procedure is repeated until the final design (entire seal is high-temperature) is successfully tested, finishing phase 1, Figure 3.



Test Step	Dynamic sealing element (pos 1)	Core sealing areas (pos 2)	Seal housing sealing areas (pos 3)
1st	O-Ring	O-Rings	O-Rings
2nd	Final design	O-Rings	O-Rings
3rd	Final design	Final design	O-Rings
4th	Final design	Final design	Final design

Figure 3: Iterative approach depicting four test steps of phase 1 on a schematic representation of the prototype

The high temperature prototype DGS is a tandem layout without secondary seal gas injection. Its design is intended to address the main challenges for supercritical carbon dioxide turbines: resistance to high temperature, low leakage rates and corrosion resistance even at moderate pressures (design static 113,8 barg, design dynamic 89 barg) and high speeds (192 m/s on rotating ring outer diameter).

A schematic view of the test setup for the DGS in phase 1 is shown in Figure 4. The test bench is an overhung design, with the electric motor as driver, transmission, and bearings placed on the left side (not shown). An auxiliary seal (AS) identical to the primary seal (PS) protects the bearings from the high-pressure gas. Pressure (p1), temperature (T4) and flow (seal gas consumption – VL4) of test gas are measured by sensors placed upstream the test rig. The primary seal crossflow valve acts as a by-pass for both the PS and AS, and is kept closed for this project – this valve is intended to reduce the temperature resulted from friction power of both pair of seal faces by increasing the test gas consumption. Its counterpart, the AS crossflow valve, is also kept closed all the time. When testing the PS the main by-pass valve is also kept closed.

In this case the sum of leakage through the PS and AS results in the test gas consumption, VL4. Furthermore, PS leakage is the sum of leakage to primary vent (VL1) and to secondary vent (VL2 through AS). A pressure transmitter on primary vent (p2) indicates the back pressure set for the AS (at least 6 barg for this prototype). Transmitters installed about five millimeters from the sleeve shrouds indicate the temperatures on the PS (T1) and AS (T2) sealing chambers. When testing the AS, the main by-pass valve is kept partially open creating a differential pressure of about 5 bar over the PS seal for its protection, while p2 is set to the testing pressure. The valve to primary vent is closed so there is no reading on VL1. Finally, a tachometer measures the shaft rotation speed.

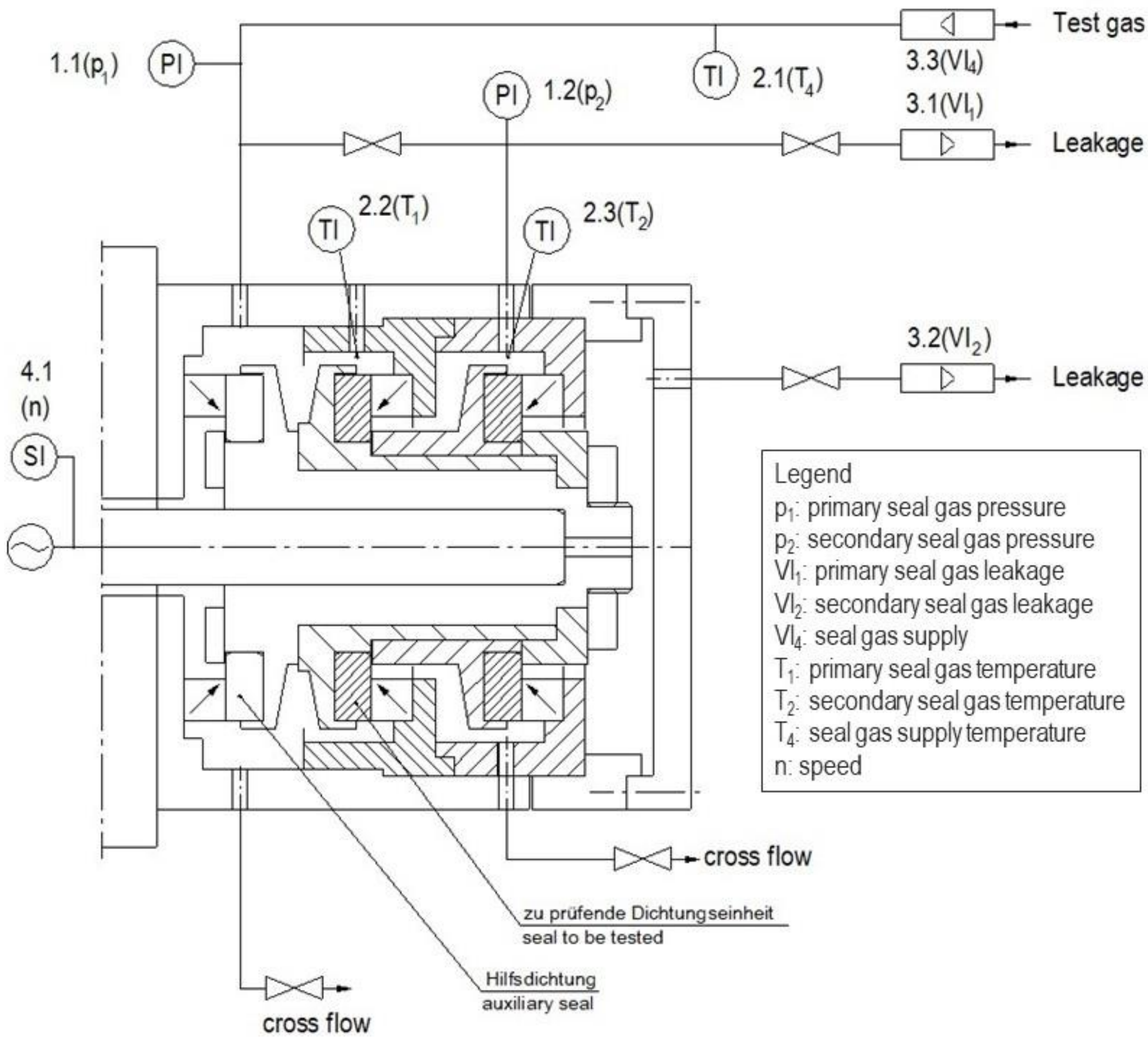


Figure 4: Schematic view of DGS test setup (phase 1)

For keeping the test rig within its operating temperature limit (250°C) an air-helium (75% and 25%, respectively) mixture was set for the test gas. The test procedure for each test step of phase 1 consists of:

- Static test at room temperature (static cold): pressurization to the static design pressure (113.8 barg) with the same pressure steps used for testing the dynamic sealing element.

Pressure is held at each step for at least one minute or until leakage stabilizes (no visual time variation).

- Dynamic test: pressure is set to the operating design (p1=74 barg and p2=7 barg) and speed is increased (10,000 rpm – 92 m/s, 15,000 rpm – 137 m/s, 18,000 rpm – 165 m/s and 21,000 rpm – 192 m/s), the test conditions are then held until leakage stabilizes; afterwards the pressure is changed to the dynamic design (p1=89 barg and p2=6 barg) and the same procedure with switching speeds is repeated.

- Static test (static hot): identical to static cold, but immediately after dynamic test.

RESULTS AND DISCUSSION

Test results for dynamic sealing elements

Figure 5 presents the test result of one dynamic sealing element. Its inner diameter is approximately 150 mm. Leakage rate is stable over time.

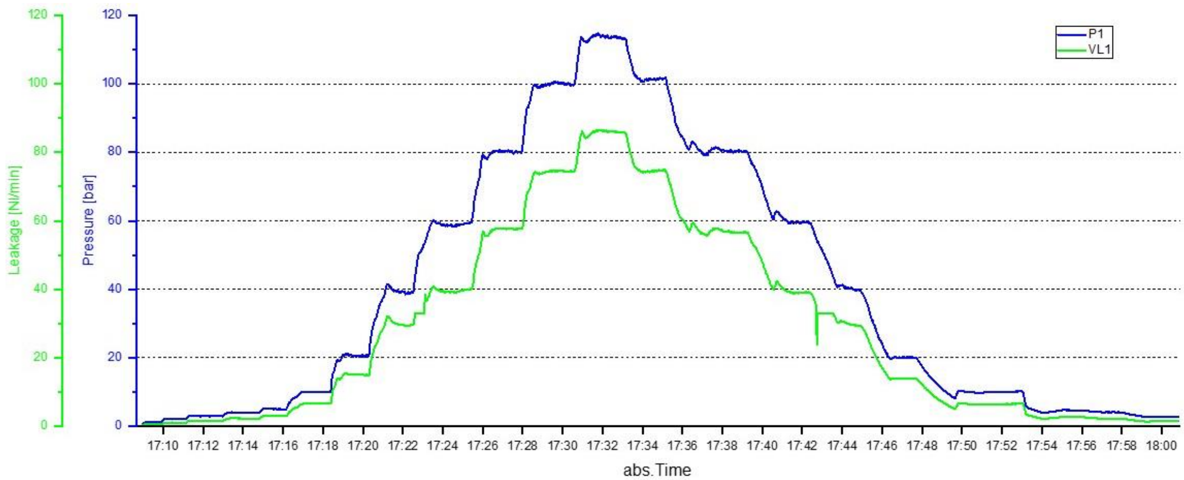


Figure 5: Static test of dynamic sealing element SN 105421800-4

Figure 6 presents an overview of the tested dynamic sealing elements. It was possible to remain under the target leakage for all pressure steps. As a rule of thumb, leakage rates are of about 1 Nl/min per 1 bar sealing pressure.

Figure 6: Overview of tested dynamic sealing elements

Test results for DGS

Since the AS is designed only as a back-up seal and thus undergoes different conditions compared to the PS, this paper will focus on test results of PS seal. Figure 7 shows the static cold test for the first test step (DGS equipped only with O-Rings). Leakage though seal faces is a majority of VL1. Its levels (about 0,01 Nl/min per 1 bar sealing pressure) are in accordance with seal design. Leakage remains stable over time. VL1 decreases back to zero when the valve to primary vent is closed for testing the AS.

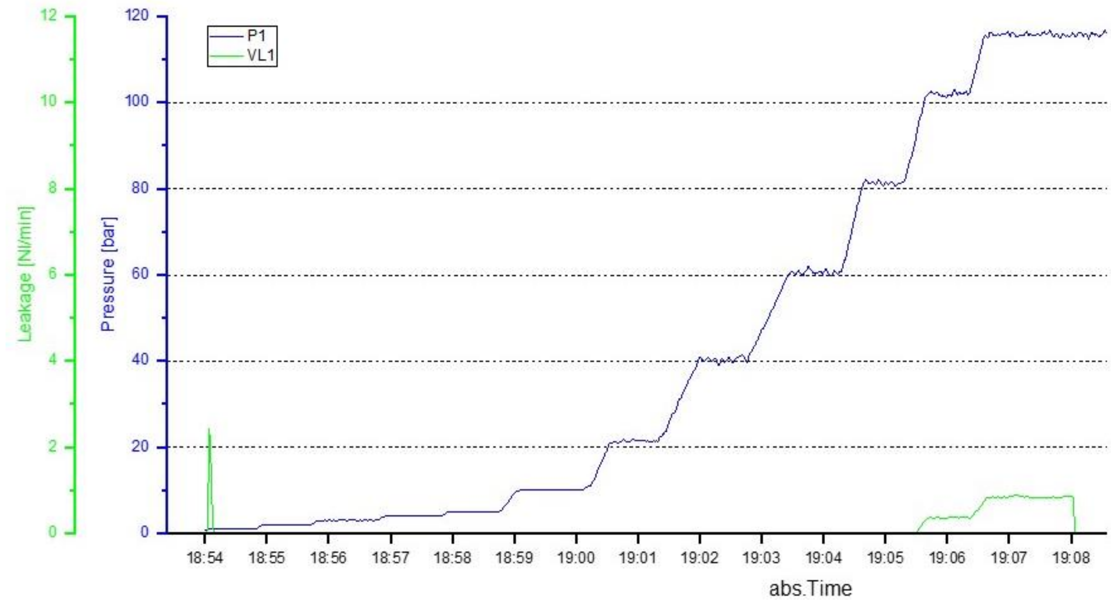


Figure 7: 1st test step – static cold

Dynamic seal tests are shown in Figure 8. Leakage is also far lower than 1 Nl/min per 1 bar sealing pressure and stable over time. This test step was limited to 18,000 rpm (165 m/s) because of limitations on cooling capacity of the available test rig.

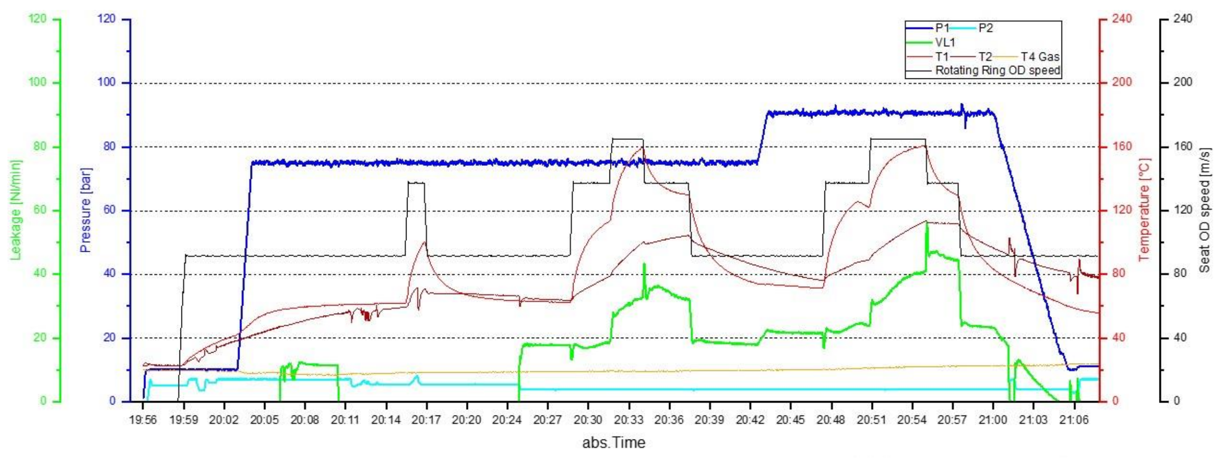


Figure 8: 1st test step – dynamic PS

Figure 9 shows the static test at room temperature for the second test step (DGS equipped with final dynamic sealing elements and O-Rings for the remaining sealing positions). The increase of leakage compared to the first test step (with O-Ring as dynamic sealing element) is remarkable. Still leakage remains slightly below 1 Nl/min per 1 bar sealing pressure.

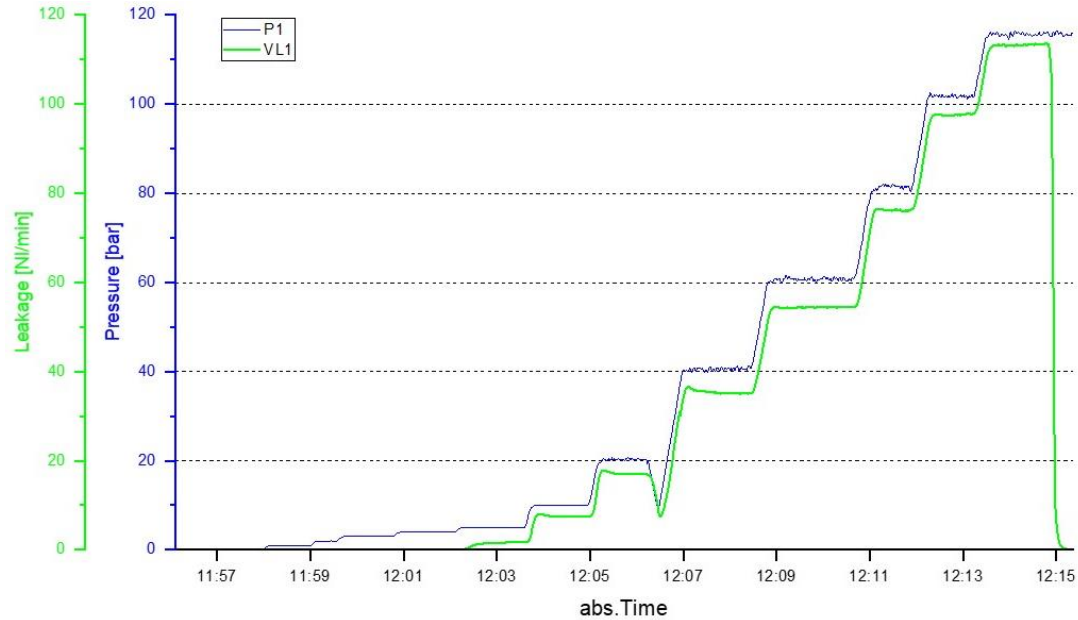


Figure 9: 2nd test step – static cold

Figure 10 presents dynamic PS test of second test step. Leakage is about tripled when replacing the O-Ring as dynamic sealing element by the high temperature resistant design. Nevertheless, it remains close to 1 Nl/min per 1 bar sealing pressure and stable over time. Again, this test must be limited to 18,000 rpm due to the available test rig.

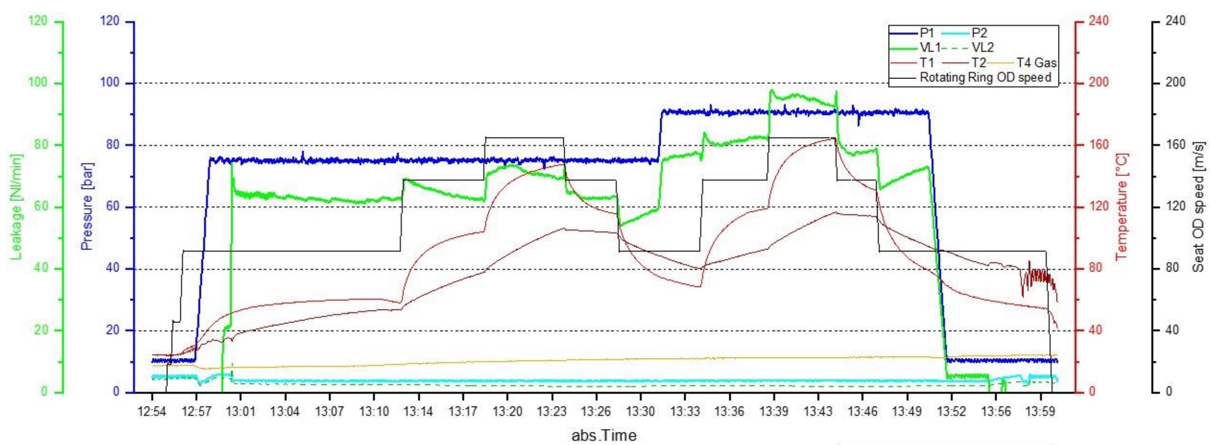


Figure 10: 2nd test step – dynamic

Figure 11 shows dynamic PS test for the third test step. For this test the dynamic sealing element and the core sealing positions (shaft sleeve to rotating ring and housing to balancing sleeve) utilize the final design seals. When comparing to Figure 10 it becomes evident that the adopted design for both core static sealing positions does not impair leakage level nor stability.

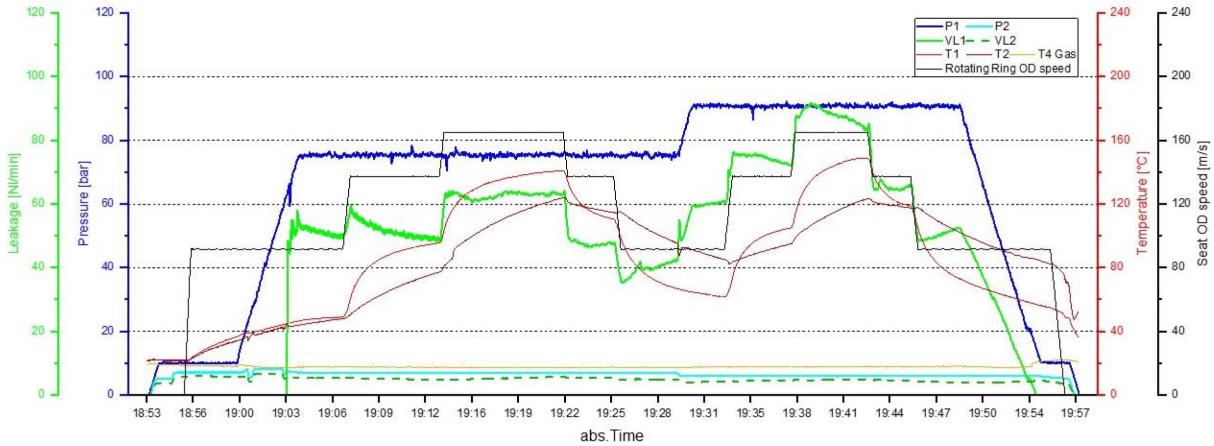


Figure 11: 3rd test step – dynamic

Figure 12 shows dynamic PS test of fourth test step. For this test the dynamic sealing element and core sealing positions are the final design. Furthermore, seal housing sealing positions (see Figure 3) for PS seal are also high temperature resistant. For AS, however, O-Rings were still used. Comparison to the previous test steps demonstrates no impairment of DGS performance by the high temperature features.

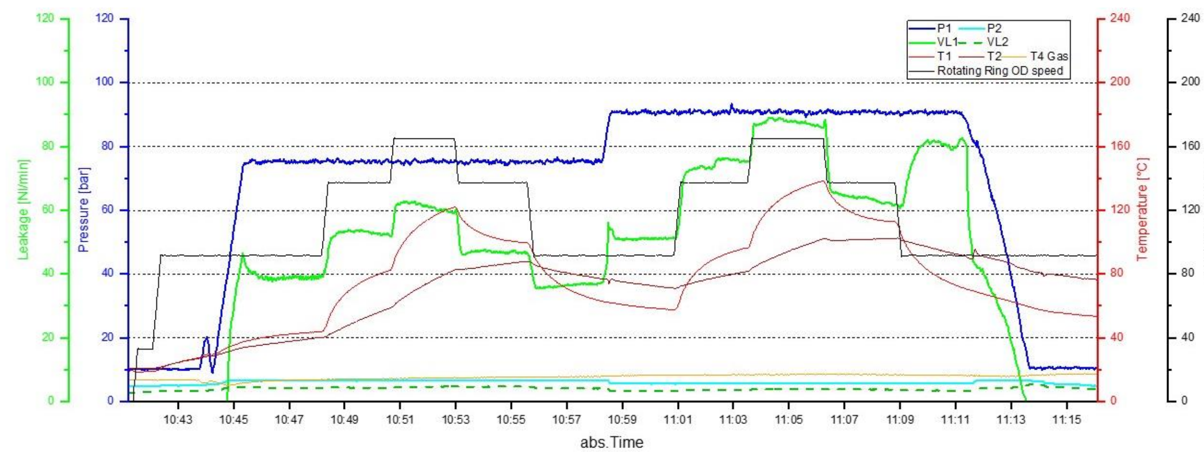


Figure 12: 4th test step – dynamic

By the time this paper has been submitted for publication tests with high temperature resistant sealing elements as replacement for the O-rings on AS housing sealing positions have not been successfully tested. Moreover, a full-load test at maximum speed (21000 rpm – 192 m/s) and a start-stop test must still be performed.

CONCLUSIONS

Preliminary tests have shown the technical feasibility of a novel concept of a DGS for high-temperature supercritical carbon dioxide (sCO2) turbines. The prototype is a mid-sized (ID of dynamic sealing element of about 150 mm) tandem seal without intermediate labyrinth and designed for moderate pressures (design static 113,8 barg, design dynamic 89 barg) and high speeds (192 m/s on rotating ring outer diameter). At this first test phase, the high-temperature resistant features as dynamic sealing element, core sealing areas and seal housing sealing areas were static and dynamic tested with and air-helium mixture up to design pressures at usual DGS temperatures (from room temperature up to 250 °C on primary seal). Due to limitations on cooling capacity of the available test rig the tests must be limited to 18000 rpm (165 m/s). Leakage rates remained stable and below the target of 1,5 Nl/min per each bar sealing pressure also after changing operating conditions as sealing pressure and speed. Final tests with sCO2 supplied at 500°C and up to design speed are still to be performed at SwRI.

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