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Dry Gas Seals Design for Centrifugal Compressors in Supercritical CO2 Application

Detlev Steinmann	**Rafael Kassimi**	**Jonathan Kleiner**	**Paolo Susini**	**Alberto Milani**	**Matteo Dozzini**
Global tech. lead Compressor Seal	Testfield Manager	Engineering Manager	Senior Sales Engineer	Senior Systems Engineer	Design Engineer
Flowserve	Flowserve	Flowserve	Flowserve	Baker Hughes	Baker Hughes
Dortmund, Germany	Dortmund, Germany	Dortmund, Germany	Firenze, Italy	Firenze, Italy	Firenze, Italy

ABSTRACT

In the recent years, the interest in supercritical CO_2 cycle applications is growing consequently several funded programs all over the world have been started to analyze various cycle configurations, as well as involved equipment such as turbomachinery, primary heater, cooler, and recuperator, with the purpose to validate design methodology and equipment behavior.

Turbomachines normally use a sealing system to reduce as much as possible external leakage. Dry Gas Seals (DGS) are one of the most effective and widely used in turbomachinery application. Nevertheless, the experience with supercritical CO_2 conditions is limited.

Main compressor suction conditions are close to CO_2 critical point. In this conditions, possible co-existence of different phases may occur in same part of the machine. In particular, the DGS may operate in presence of liquid CO_2 or even in solid phase during transient conditions.
Moreover, in case of regenerative cycle where by-pass compressor is present, discharge temperature is in the range of 200°C. Due to the combination of pressure, temperature and speed, the operating conditions exceed the existing experience and may be critical for DGS integrity. For this reason, special attention during design phase must be taken to ensure safe and reliable DGS performance in HP CO_2 environment.

The present work will describe the challenges faced by DGS and compressor OEMs, working in close collaboration during the design and testing phase of turbomachinery for this application. In addition, to validate DGS behavior and performance in supercritical CO_2 environment, a dedicated test bench has been designed and manufactured to mirror turbomachinery conditions around the DGS. An extensive test program was developed to verify DGS performance in both design and off-design conditions. This test rig and the operating conditions that can be replicated set a benchmark for this kind of application in this industry sector.

The DGS design and test bench features will be presented as well as the results of the test carried out. The test highlighted the robustness and reliability of the DGS also in these challenging conditions.

Authors

Detlev Steinmann is a 1985 graduate from the University of Dortmund, Germany with a B.Sc. degree in Mechanical Engineering. He started his career 35 years ago in the engineering department of the Flowserve Corporation in Dortmund/ Germany. He was focused on compressor seal technology, including development of High-End Seals for the industry. After leading several engineering departments, in his current position he is the global technical lead for compressor sealing technology, covering R&D, design, standards (API 692 Task Force member), training and trouble-shooting support.

Rafael Kassimi is a graduate from the University of Gelsenkirchen, Germany with a B.Sc. degree in Mechanical Engineering. In 2011 he joined Flowserve, Dortmund in the compressor seal R&D team. During this time, he was working on product design projects and design verification testing for specialized DGS technology. In his current position he is managing the testfield and the compressor seal assembly team in Dortmund.

Jonathan Kleiner joined Flowserve Dortmund as an engineer in the R&D department in 2016. In parallel he did his Master of Science in mechanical engineering, which was accomplished in 2018. Between 2016 and 2020 his responsibility covered several special DGS technology projects, DGS performance and mechanical analysis and the development of an electrical driven seal gas booster. In his current position he is responsible as supervisor for Mixer, Specialty and Compressor Seal Order Engineering in Flowserve Dortmund.

Paolo Susini is currently Senior Sales Engineer for Compressor Seals & System at Flowserve. Paolo has over 30 years' experience in the turbomachinery industry as centrifugal compressor design engineer with BHNP, focused on DGS and System, field troubleshooter and trainer. He joined Flowserve in 2014 and he was part of the API692 task force committee.

Alberto Milani is currently Systems Engineer for compressor NPD projects at Baker Hughes. Alberto has over 17years experience in the turbomachinery industry (2y as Centrifugal compressor design engineer in GE O&G, 3y+ as Rotating Equipment engineer in Foster Wheeler Ltd, 4+y as Field Project engineer in GE O&G and 7y+ as NPD systems engineer in Baker Hughes)

Matteo Dozzini is a graduate from the University of Perugia, with a B.Sc. degree in Mechanical Engineering and PhD in Industrial Engineering. Is currently Senior Design Engineer for NPD projects in Baker Hughes. Matteo has over 15 years' experience in the turbomachinery (4y+ Auxiliary System Design Engineer, 2y Project Engineer, 8y+ Senior Design Engineer for Flange to Flange Prototype)

sCO2 APPLICATION IN CENTRIFUGAL COMPRESSOR AND CHALLENGES

In sCO2 application a Brayton recompression cycle with one expander, two parallel compressors and two recuperators is typically implemented. In this configuration the main compressor suction condition is in an area close to CO2 critical point, while the bypass one is far from the critical condition; the compressors discharge is typically around 250bar. Main compressor inlet temperature is kept constant using a cooler.
The main compressor takes advantage from the real gas behaviour of the working fluid near the critical point where isobars are thickened and following a polytropic compression for a given pressure ratio (FIGURE 1) the corresponding temperature rise is lower compared to the one obtained in the subcritical region. By the contrary, CO2 thermodynamic properties (e.g., compressibility, Mach number, etc.) near critical point show large gradients, and this may have impact on the design of turbomachinery and more specifically on compressor impellers.

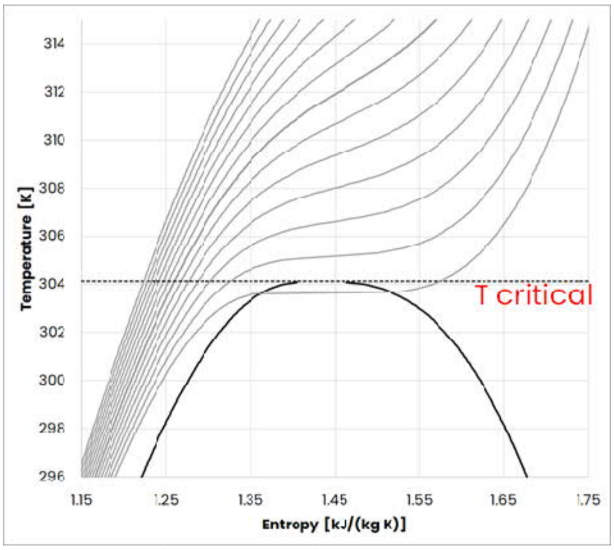


Figure 1: T-S Diagram for CO_2

The flow field on the first impeller could promote phase-change phenomena resulting in non-conventional flow configuration with coexistence of cavitation phenomena and compressibility effects. Close to the saturation line the thermodynamic quantities show high gradient, with speed of sound that drops significantly producing sudden increase of local Mach number. In this condition shock waves may occur, with direct implications on machine operating range and cycle efficiency [1]. For downstream impellers (in case of multi-stage machines) this phenomenon is less critical because proceeding with compression the flow conditions are far enough from saturation line and the risk of phase-change is highly reduced. Inlet guide vanes at compressor suction have been installed to allow for compressor suction condition fine tuning. For the above-mentioned reasons, compressor inlet temperature control is key for the good operation of the machine. This parameter must be taken in account carefully during the cooler design phase to guarantee that, in all operating conditions, steady state and transient, inlet temperature range (min/max oscillation) allow to avoid CO2 liquid formation at 1st impeller and to operate the compressors without any limitations.
The extreme power density (power to inertia ratio) for this application required special attention on mechanical verification of the main compressor components, on the compressor rotor dynamic behavior and on the evaluation of the thrust load acting on compressor thrust bearing.

On the other hand, the extreme power density, allow the selection of a smaller size compressor, providing benefits in terms of footprint and equipment cost.
In case of a prolonged pressurized stop, supercritical CO2 within the loop may be cooled, depending on the ambient conditions, below the critical temperature (31°C). In this case both liquid and gaseous phases will be present within the loop. A top process flange compressor arrangement would cause partial or total flooding of the compressor. This condition must be taken in account in the design of the seal gas system and may leads to flooding of the DGS as well. In addition, in case of DGS failure, venting supercritical CO2 to atmosphere can lead to cross the condensation line, entering in the multiphase area, where liquid and gas coexist together. Furthermore, below 5 bars, CO2 may become solid (ice formation) with possible clogging of the DGS vent lines. This is a typical condition that may occur in sCO2 application for which compressor and DGS OEMs have no or limited experience.

For all the above-mentioned reasons, a test on DGS in supercritical conditions was carried out to verify their behavior in design and off-design conditions.

Dry Gas Seal for sCO_2 Application

To ensure efficiency benefit, the sCO2 technology needs advanced rotor sealing capability for the turbomachines to handle the high pressure, the high operating temperature, the high speed and minimizing the leakages to avoid frequent circuit refilling and reduction emission to atmosphere. Dry gas seal (DGS) in the compressor world is already a wide utilized sealing technology for CO2 application since decades. But the sCO2 application window development is stretching the DGS technology to unexplored environments combining at the same time a supercritical gas, high pressure, an elevated temperature and highspeed range. Flowserve deliver since mid of the 90´ s dry gas seals for CO2 compressors, like integrally geared type, and partnered with compressor OEM´s, when there was a need for high end-seals for their projects in CO2 service, i.e. CO2 capturing /re-injection. In 2012 the first dry gas seals were delivered for sCO2 application, i.e. waste recovery cycle pilot projects. In recognition and to meet the rising technology trend and to support R&D projects in 2014 the exiting CO2 test loop, located in Temecula/ USA since 2009, was upgraded to generate sCO2 testing condition. Despite that over the last years the number of published papers related to dry gas seals in sCo2 service is increased, in majority those papers are explaining in detail sCO2 properties/ characteristics, analytical simulation (CFD) of gas flow through narrow sealing gaps. Few of these papers are giving an insight to practical experience during seal vendor testing or operational experience in several pilot plants. For this reason, a DGS functional test with sCO2 gas to mirror the turbomachine's operating conditions was considered necessary to gain the necessary industrial knowledge to support this technological step change. This article summarizes the DGS design, test rig arrangement and detailed test results.

The Challenge

End of 2018 BHNP contacted Flowserve regarding a DGS feasibility study for compressors, which will be installed in a sCO2 Re-compression Closed Brayton Cycle (RCBC) pilot project. The two compressors, Main and Bypass compressor, operate at same suction pressure ratio, different flows and suction temperatures.

Key design conditions

Shaft Size:	75mm
Seal type	Tandem L
Pressure static	285 bara
Pressure dynamic	200 bara
Temperature	-46to250°C
Speed MCS	28350 rpm
Speed Trip	32225 rpm

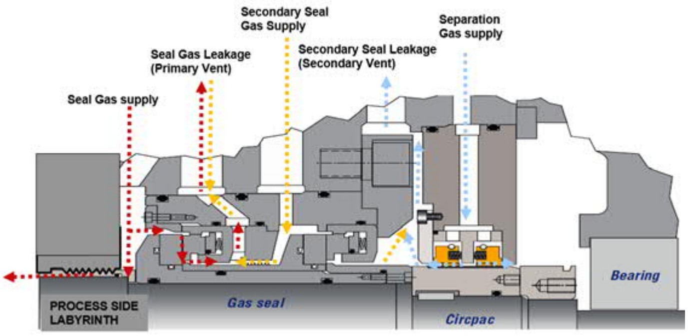


Figure 2: Dry Gas Tandem Design

The difference between the Main & Bypass compressor is the expected temperature of the compressor casing around the dry gas seal cartridge. An initial thermal calculation (BHNP) showed expected temperature distribution around the cartridge for the Main compressor ~100÷130°C, whereas temperatures of 200÷220°C are steady present for the Bypass compressor. [figure 3&4]

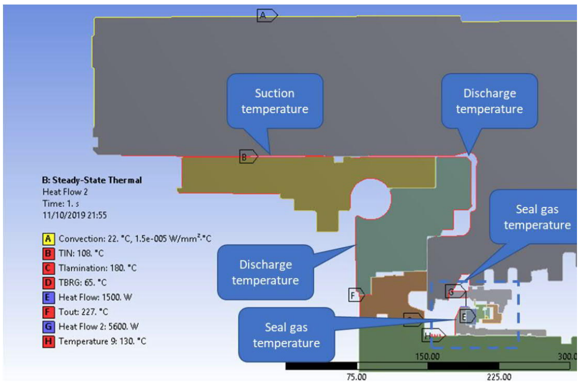


Figure 3: Temperature Map [1]

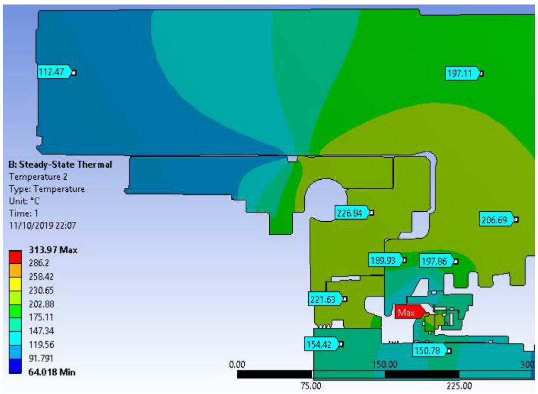


Figure 4: Cavity Seal Temperature profile [1]

The risk associated with high density gas like CO2 at high pressure, is related to the potential overheating of the DGS at the given speed/ pressure caused mainly by windage as heat source. The heat dissipation/absorption to the surrounding compressor cavity shaft is limited, when already here higher temperature levels are present. The expected thermal equilibrium for heat generation-absorption=> steady state temperature @ DGS showed as a result temperature at seal close to 300°C. 2014 BHNP/FLS already collaborated in an CO2 compressor project for re-injection. The available test results from the full acceptance, here Air/He mix was used as test gas, showed the high heat development @ seal during full load testing. The peripheral speed here was ~ 35% less (145m/s) vs the actual application@ 195m/s. Pressure in comparison are 240 bara vs 200 bara actual. But as known, the speed is the main driver for heat generation.
One requirement from BHNP, that DGS testing should simulated the "real Life" condition as close as possible, fits to Flowserve DGS development plan, to extend the testing capacity and R&D activities for sCO2 application. A new sCO2 testing loop at Flowserve facility Dortmund/ Germany was already in planning and construction started end 2019. Final commissioning was scheduled Q3/2020. Beside the design verification testing, one future benefit is the validation of the thermal model, which simulations start from a 2D approach with fluid and solid domains run as uncoupled, move forward through a conjugate run to conclude with a 3D approach zooming the full domain for the areas of higher relevance. The main purposes of this activity is to validate

the boundary conditions, from correlations, used, in general, during the DGS cooling system design phase and to develop a dedicated correlation for windage losses at high pressure in the analyzed fluid. Model description and validation is not the purpose of this paper and will be presented separately. Flowserve will use the gained experience for further development activities to push DGS technology to a new application window level and to meet the rising challenging industry demands for reliable shaft sealing solutions.

Dry Gas Seal Design

Based on Flowserve experience with high pressure, high speed and high temperature application the design process was possible to accelerate. The design for this new application follows the existing design standards for high pressure application up 650 bara, high temperature application up to 270°C and high-speed application up to 235m/s TIP. Here Flowserve already delivered proven and reliable sealing solutions to the market. [picture 5 & 6]

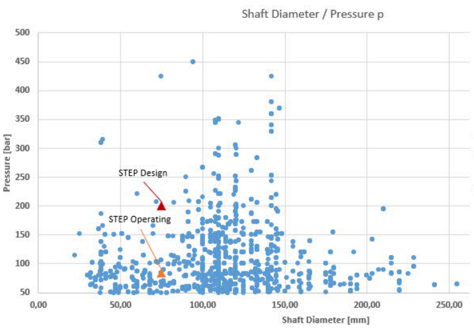


Figure 5: References High Speed

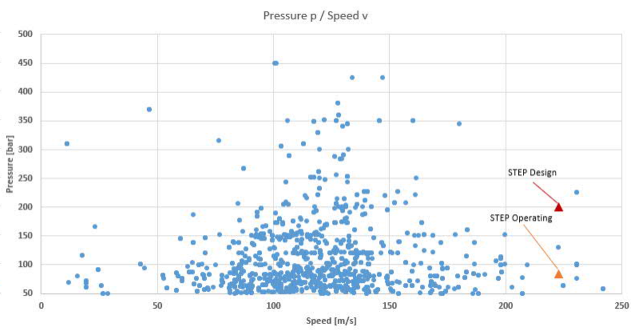


Figure 6: References High Speed

The design standard includes material selection, figure 7.

Main DGS Components

Stationary Face	:SiC/DLC
Rotating Face	:SIN
Sec. Sealing Elements	:PTFE

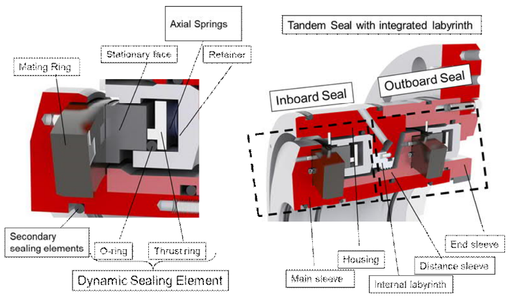


Figure 7: Overview DGS Tandem Arrangement

As design reference, figure 8, an existing standard High Pressure DGS (figure 8) for BHNP compressor, i.e. for natural gas application was available.
This design was successful tested up to 425 bara in 2015.

Shaft Size:	75mm
Seal Type*:	Tandem L
Pressure Static:	425 bara
Pressure Dynamic:	425 bara
Temperature:	-46 to 230°C
Speed MCS:	20500 rpm

*** Bi-Directional Design**

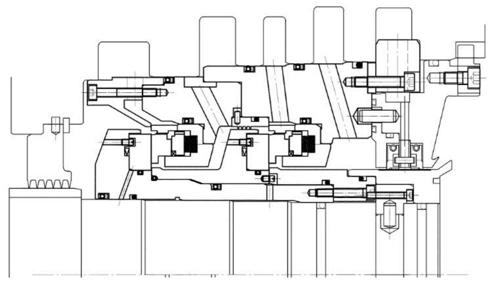


Figure 8: Design reference; 75 mm shaft size

Already combining high pressure, high speed and high cavity temperatures at seal are challenging and outside the application experience, but in addition, to ensure efficiency benefit, the sCO2 technology needs minimizing the leakages to avoid frequent close circuit refilling and so emission to atmosphere => this is to see as unique! [Figure 9]. Structural analysis (FEA) includes all single seal mechanical components and combine also functional units, which have an impact to the overall seal performance at the various operating conditions. The intensive review of existing analysis and performance results from previous projects leads at the end to the best design combination, which addressed all operating parameter and at the various condition:

Pressure static	0-> 285 bara
Pressure dynamic	0-> 200 bara
Temperature	-46 to 250°C
Speed	0-> 30500 rpm

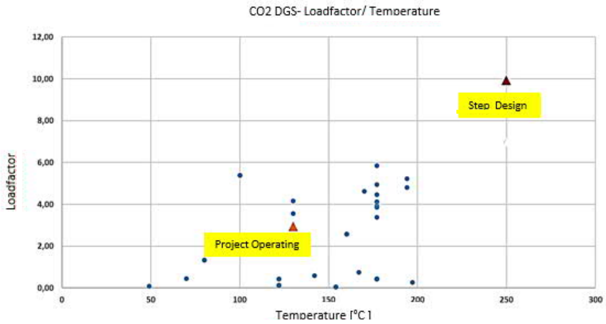


Figure 9: *Beyond the limits*

Seal Performance Analytic [2,3,4]

Parallel to the structural analysis available performance test results from those high-end references were integrated in the seal performance analytical process applying the sCO2 project operating condition.

Here the focus was i.e.:

- heat generation @ dynamic condition
- temperature profile within the seal cartridge
- leakage trend

As an example: during the test in Dortmund for the BHNP CO2 project in 2014 an extended test regime was focused on those impacts. The available test results were scaled to the new conditions (speed, pressure and size) and deliver solid valuable information, which were integrated in the sCO2 seal and test-rig design. In figure 10 temperature probe/ thermocouple typical arrangement for R&D testing is shown.

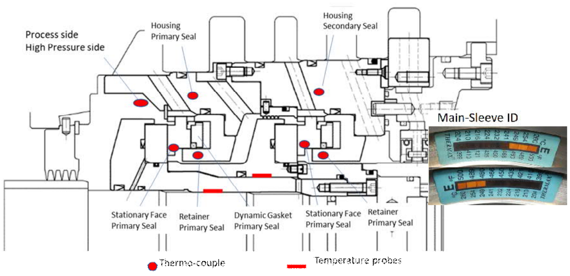


Figure 10: Temperature measurement positioning

The knowledge of the seal face temperature development @ the various operating conditions, including standstill, is essential to know. For a reliable DGS operation a stable gas film, separate the sealing faces, has to be achieved and maintained to avoid any instability like pumping effect = fluctuating or extremely rapidly increasing/ decreasing leakage trend. Here the CO2 special characteristic with changing properties depending on pressure & temperature is relatively unknown for narrow gap (3÷5 µm) analysis.

Flowserve is using for DGS performance MSTI [2] as standard tool for analysis beside further seal design/ performance simulation software. Performance test results were used to develop the software continuously, to improve the predictive accuracy. But it´s still a prediction if the application is beyond the experience. Taking in account manufacturing tolerances, slight differences in the structural design will influence the performance characteristic of a functional component design.
Therefore, that testing with air as reference was foreseen, in figure 11 calculation result are shown for seal-face temperature development CO2 vs Air.

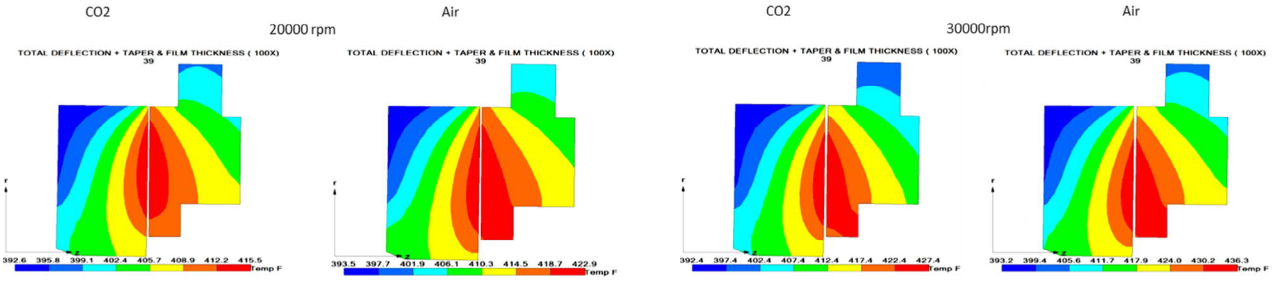


Figure 11: Seal-Face temperature development comparison sCO2 vs Air [2]
Note: @ 115bara // Assumption: 200°C constant @ seal for both calculations.

The final DGS design, figure 12 was released for manufacturing after ~ 4 months engineering time. Necessary design modification in comparison to the reference design addressed mainly higher operating temperature and increased speed.

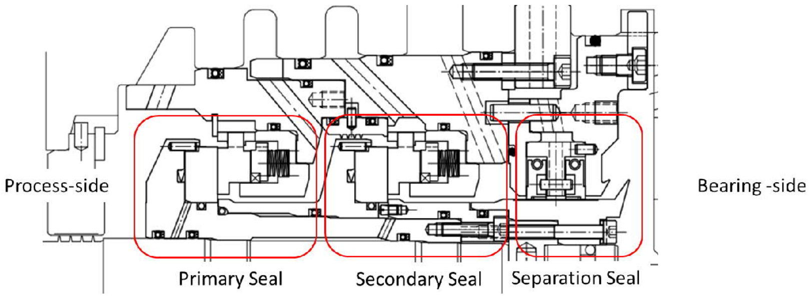


Figure 12: Final DGS design

DGS Pre-Test

The initial DGS design verification test started mid Q2/2020.
Purpose is to generate the first performance data, which were directly analyzed from the R&D department in relation to the theoretical calculation. The standardized test regime includes i.e.:

Test Medium: Air and or Air /He mix
Primary & secondary seal

- Static test: 0, 10, 20 , 30 ……. 100% of static design pressure
- Dynamic test
  - pressure variations 5; 10, 20, 30…..100% dynamic design pressure @ various constant speed
  - speed variations 60, 80, 100% of of mcs + trip speed @ various constant pressure
  - speed variations +/- 15% @ constant pressure and speed step
  - Start/stop cycles with pressure variation up @ dynamic design pressure

In the test regime integrated was also testing of *seal face layout variations*, i.e. balancing ratio, groove depth variations, to collect information regarding changes in the leakage characteristic and overall performance.

The performance results during the test were within the expectation.
and showed a stable "reaction" to speed or pressure variation, figure 13.

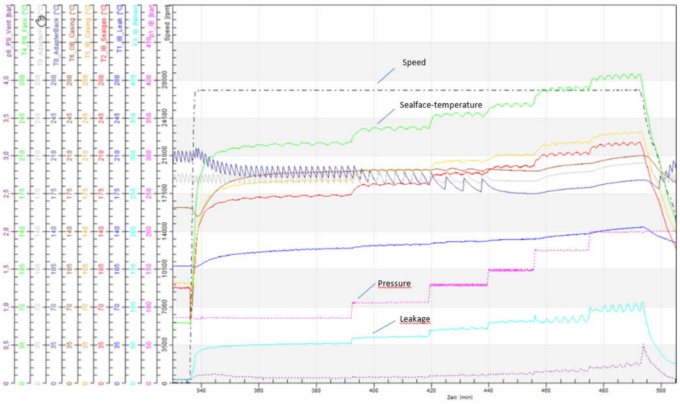


Figure 13: Test Diagram = Pressure variation test @ constant speed

Tester design considerations

Based on the information from BHNP regarding the temperature profile cavity, see figure 3 & 4, and the requirement to mirror those condition during the DGS testing, a detailed thermal analysis of the tester design was conducted during the design phase. As a result, to achieve stable conditions during steady state test steps and/ or to adjust to the various speed/ pressure condition an external- heat source was necessary to be integrated directly into tester housing, surrounding the DGS cartridge, to manage the thermal condition at seal. Design solution: heating rods (18 x 1 KW) were integrated directly in the tester adaptive part, figure 14.

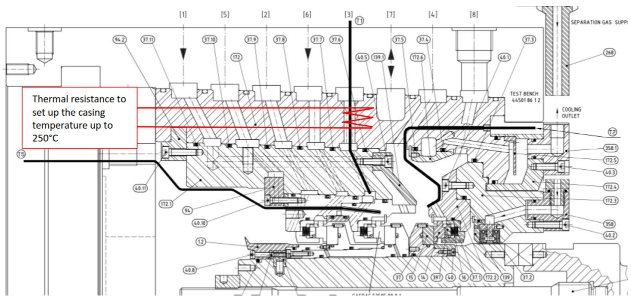


Figure 14: Tester Set Up

The function and efficiency were proven during DGS pre-testing. The on/ off was automatically controlled by a thermocouple, positioned at a reference temperature location. The adapter part temperature follows ~ simultaneous the heating cycles. Temperature variation, if needed, close to 5°C were so achievable, figure 15&16.

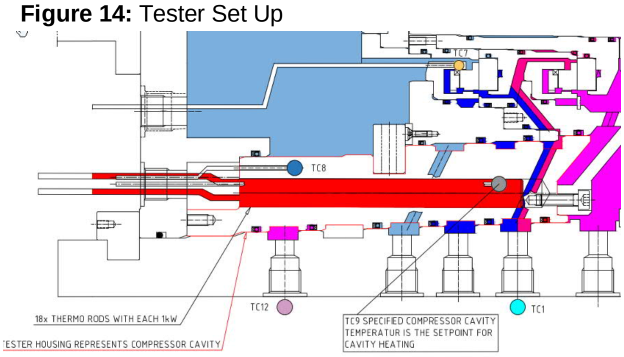


Figure 15: Tester set up with heating-rods

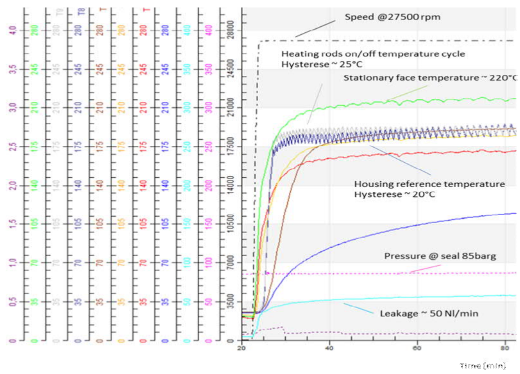


Figure 16: Diagram Temperature = f (heating rods)

DGS TEST BENCH DESCRIPTION

Ultra-Highspeed-Tester

For the CO2 test campaign, the ultra-highspeed-tester, located at Flowserve, Dortmund was used. The seals are assembled to the pinion and flanged in an overhung test cavity which is shown later in this chapter.

Max. Speed
70.000 rpm

Max. Pressure
500 Barg (Air) / 270 Barg (sCO2)

Max. Shaft Diameter
~ 100 mm

Motor power (VFC driven)
200 kW

Torque measurement
dynamic

Data Acquisition
Multi-Channel-System

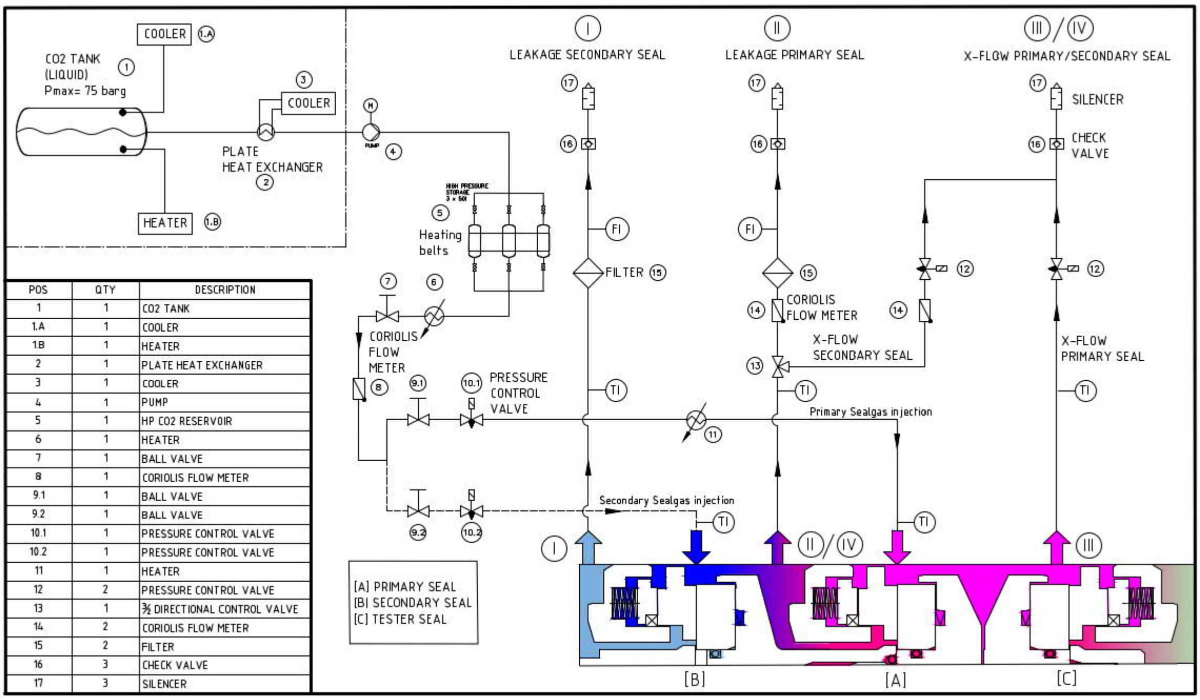
Picture 1: Tester setup / o*verall specification of the ultra-highspeed-tester*

CO2-Supply System

For testing with CO2, an additional system was implemented as shown in figure 17. As Flowserve has already a CO2-System in Temecula available, certain knowledge could be shared. During the implementation of the CO2 Supply, a few challenges needed to be overcome. As an example, some challenges were to maintain CO2 in liquid phase during compression by the pump or the selection of the pressure regulation valve. Additional experiences were gained by controlling the CO2 phases in the low pressure venting lines, to avoid ice formation.

System P&I (Figure 17)

Starting from the medium pressure CO2 tank with a capacity of 3 tons and a maximum pressure of 75 barg, the liquid CO2 is cooled down with a heat exchanger prior entering the piston pump. The pump compresses the liquid CO2 up to 270 barg into a heated bottle rack to keep the CO2 in supercritical state. Afterwards, the CO2 is controlled by a PCV to supply the seal with the correct pressure.



POS	QTY	DESCRIPTION
1	1	CO2 TANK
1.A	1	COOLER
1.B	1	HEATER
2	1	PLATE HEAT EXCHANGER
3	1	COOLER
4	1	PUMP
5	1	HP CO2 RESERVOIR
6	1	HEATER
7	1	BALL VALVE
8	1	CORIOLIS FLOW METER
9.1	1	BALL VALVE
9.2	1	BALL VALVE
10.1	1	PRESSURE CONTROL VALVE
10.2	1	PRESSURE CONTROL VALVE
11	1	HEATER
12	2	PRESSURE CONTROL VALVE
13	1	3/2 DIRECTIONAL CONTROL VALVE
14	2	CORIOLIS FLOW METER
15	2	FILTER
16	3	CHECK VALVE
17	3	SILENCER

Figure 17: CO2-Supply System

Between the compressor and the seal, two heaters have been integrated to keep the CO2 in the specified temperature range. The first heater is implemented directly at the compressor module and the second heater is implemented directly before the tester-cavity. This ensures that the CO2 is in the specified state during its flow through the piping-system. After the CO2 has passed the Coriolis inlet flow meter and has entered the tester cavity, it is either exiting through the leakage line I/II or as a sealgas cross flow on line III/IV. To prevent ice formation in the leakage or crossflow lines, the CO2 is kept with check valves at a pressure of ~6 barg. To measure the leakage, a Coriolis flow meter has been implemented in the line.

DGS Test set-up

The DGS test set-up simulates the compressor cavity, but contains further features, like tester seals or cooling chambers, which are necessary for the testing. Besides the actual DGS ([A]&[B], shown in Figure 17), a separate tester seal ([C], shown in Figure 17) is mandatory to create a pressurized volume. This tester seal, designed as a single plus an additional Circpac separation seal, has similar design features as the actual DGS. The Circpac separation seal allows to measure the leakage of the tester seal and helps to minimize the CO2 leakage into the tester room in case of a seal failure.

Flow description

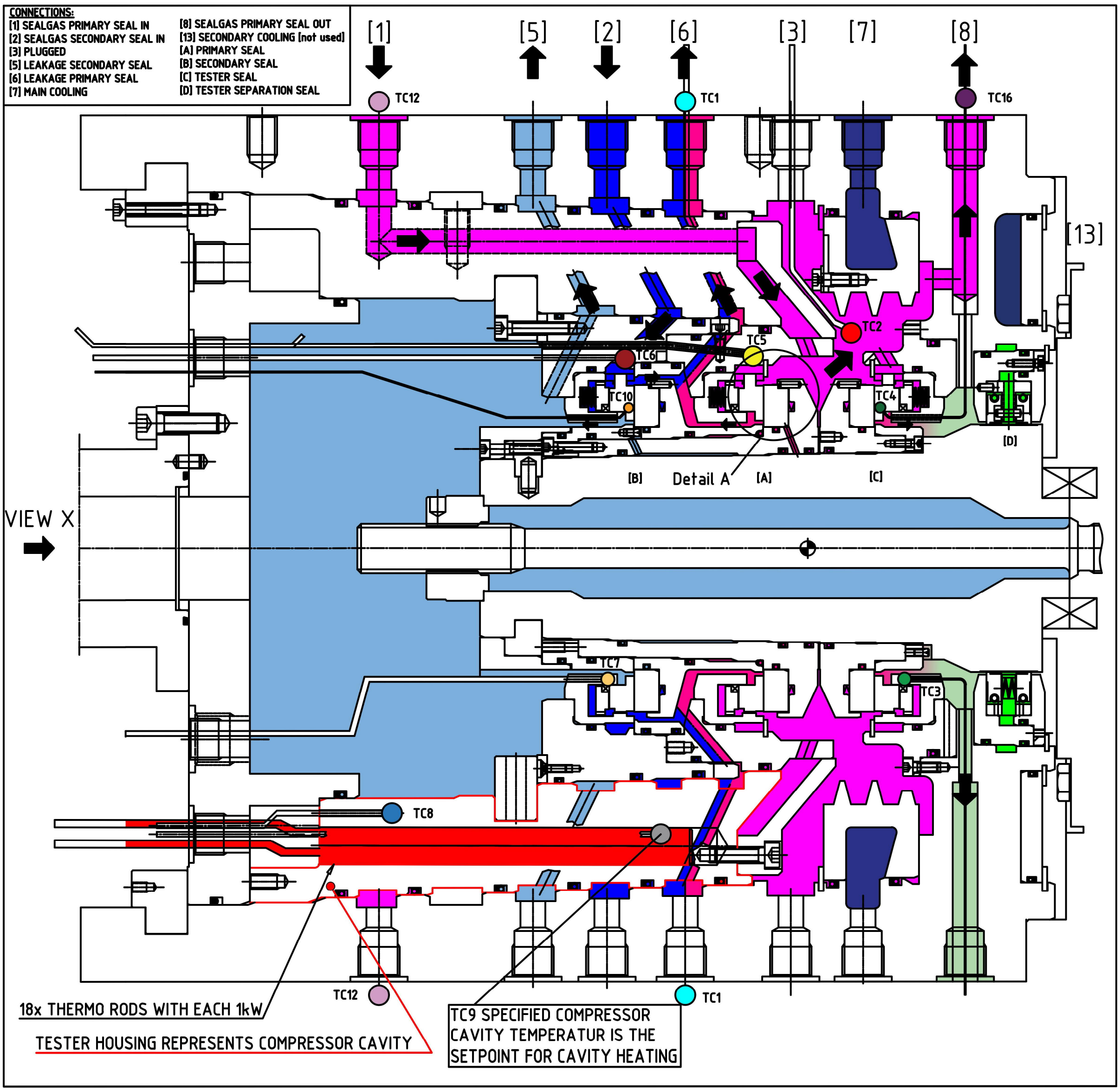


Figure 18-a: Test Cavity

The sealgas is entering into the tester cavity at port [1], is routed through the adapter housing with the heating rods (shown in red) and is then entering the actual sealing chamber of the inboard seal. Through the port [8] the sealgas can be flow-regulated to simulate a certain sealgas-flow. The leakage of the inboard seal is exiting the tester cavity on port [6] whereas the leakage of the outboard seal is exiting on port [5].

Temperature measurement

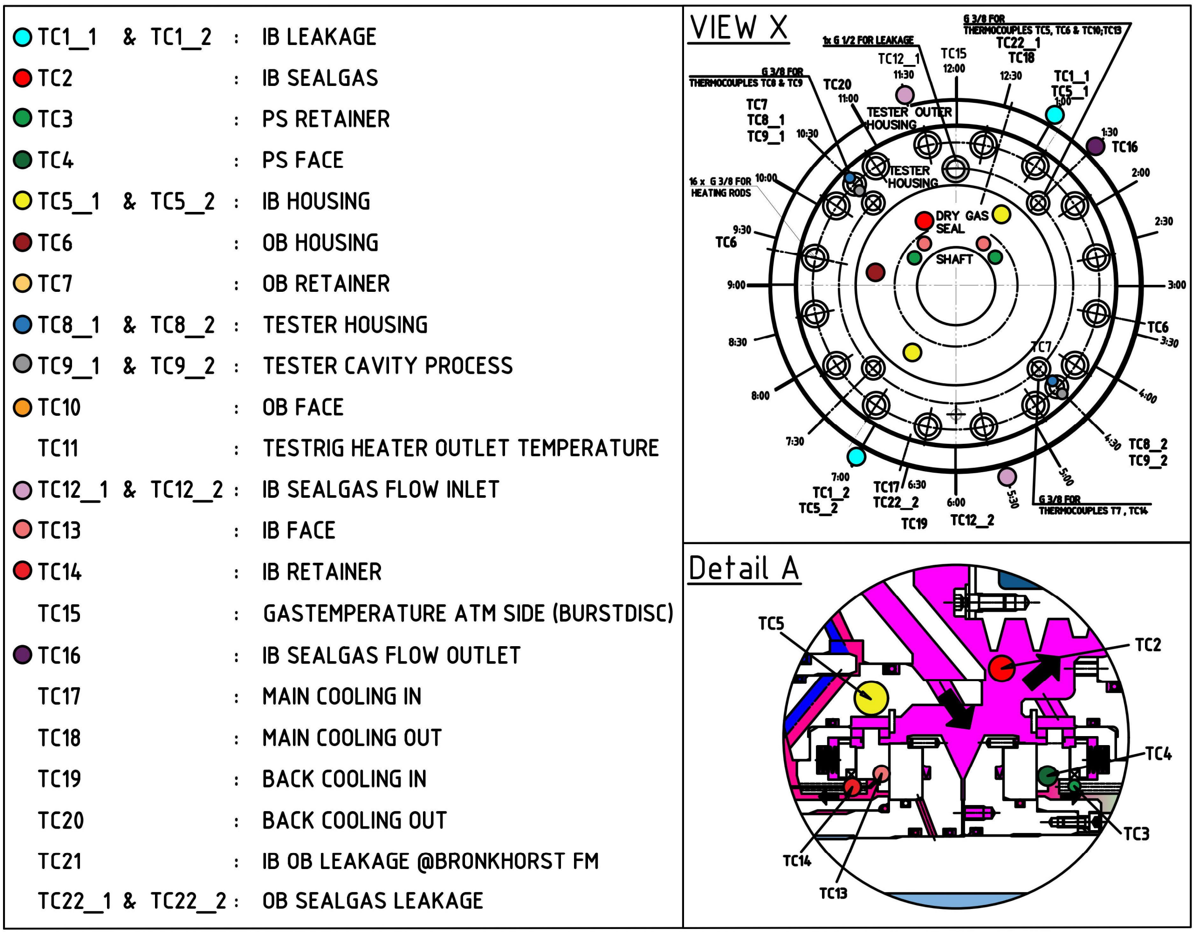


Figure 18-b: Test Cavity -View X, Detail A and Thermocouple description

Multiple temperature probes were installed into the seal and the tester parts, to create an overall thermoprofile during the testing. Whereas in the plots of the next chapter the quantity of the displayed temperature probes was limited, to gain a clear overview, as following:

- TC1_IB Leakage
- TC2_IB Sealgas
- TC4_PS Face
- TC5_IB Housing
- TC6_OB Housing
- TC9_Tester Cavity Process
- TC12_IB Seal
- TC13_IB Face
- TC16_IB Seal Flow Outlet

Picture 2: Tester set-up for hot sCO2

The tester main housing and sealfluid injection lines are insulated. The tester adapter-housing (reflecting the housing of the compressor) is equipped with 18 heating rods.

The seal and the tester adapter-housings are equipped, with 28 thermocouples.

In picture 2 the cabling of the thermocouples and the heating rods, show the complexity of the test set-up to monitor and control the test conditions.

Picture 3: Monitoring of Testing

During the complete test campaign, the tester itself, the CO2 System and the seal were monitored, to perform accurate system control and data acquisition.

DGS TEST PROCEDURE

Beside an extended design validation testing, done with air and CO2, an intense collaboration between BHNP and FLOWSERVE was carried out by following a special CO2 test plan (CO2 Matrix Test). This test plan reflects different test conditions which simulate various operational scenarios, as well as design parameter, to validate the seal's capability coping with the fluid in various phases (See Fig. 19). In general, the Matrix Test distributes into three main parts, with different temperatures of the tester housing, reflecting the compressor cavity (Shown in Fig. 18-a). The test campaign gained information about the temperature development of the seal in the tester, due to the inner energy, produced by the seal itself and by implementing external energy into the tester, when heating up the cavity to different temperature setpoints. Also, the sealgas was constantly heated up to the specified setpoints.

After setting the constrains for the seal, by controlling the cavity and fluid -temperature, also the influence of different sealgas massflows crossing the cavity, under constant pressures, could be recorded at various temperature probe locations.

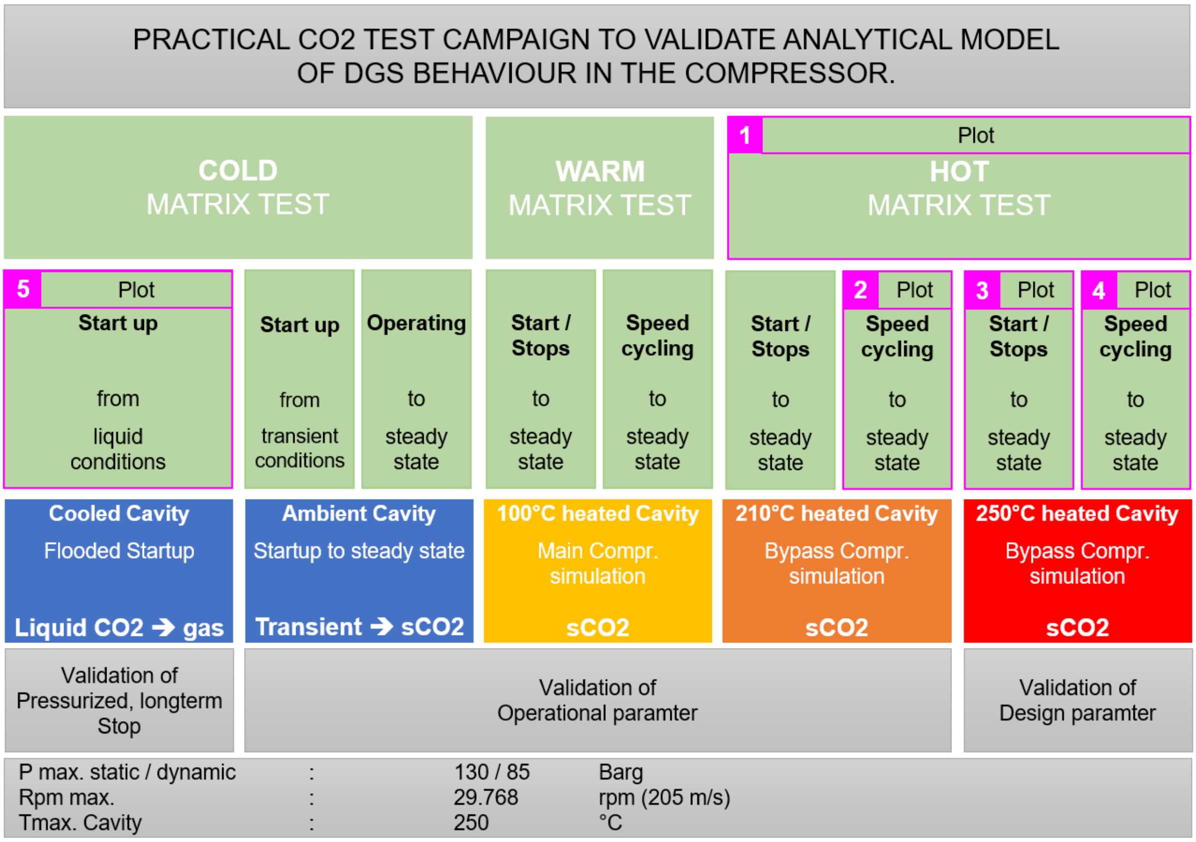


Figure 19: Overview of CO2 testing to validate the boundary conditions of the thermal model

COLD MATRIX TEST

The Cold Matrix Test consists of two scenarios with an unheated cavity, whereas the startup performance of the seal was measured. To simulate a prolonged, pressurized stop of the compressor, the inboard seal cavity was fully flooded with liquid CO2 (See Fig. 21). After static testing a startup out of those conditions was performed up to certain speeds, until the liquid turned into transient and gaseous CO2 phases, here the pressure was kept constant (Details will be shown in the following chapter). The second Cold Test was performed with an ambient tempered cavity. The sealgas was heated up to 130 °C, measured at cavity injection. The seal was started up from static conditions to normal operative speed at normal pressure, to see the temperature development of the seal in the system up to steady state conditions, due to rotor windage, only.

WARM MATRIX TEST & HOT MATRIX TEST

To reflect the Main and Bypass Compressor types, the two *Warm Matrix Test* scenarios differ by different setpoints of the cavity temperature. Prior performing various dynamic test steps, the temperature was regulated to:

- 100 °C housing temperature*, reflecting the Main Compressor cavity and to
- 210 °C housing temperature*, reflecting the Bypass Compressor cavity.

For the *Hot Matrix Test* scenario, the cavity temperature was set up to:

- 250°C housing temperature*, to gain test data at DGS design conditions.
  **** See Fig. 18-a and 18-b**, temperature measured at TC9_1 and TC9_2*

After static testing at settle out pressure, different dynamic tests have been performed until the reach of steady state temperatures, in the system, for each test step.

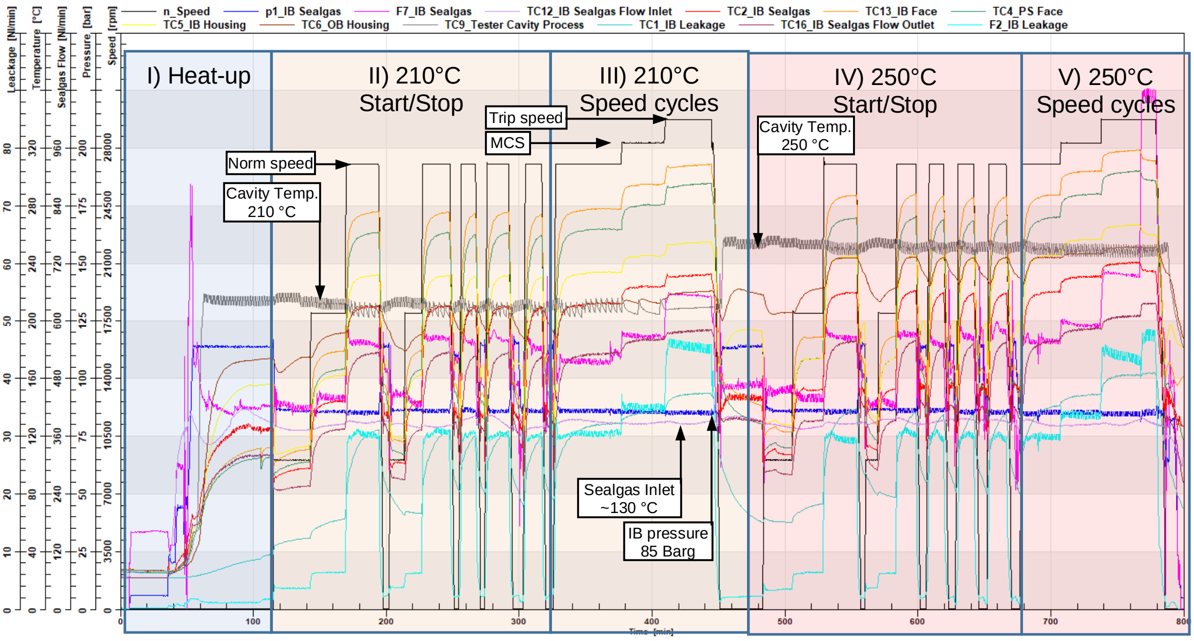
For all tests, the sealgas was heated up to 130 °C, measured at both cavity injection ports (see Fig. 18-a and 18-b, TC12_1 and TC12_2). To compare the data of the *Warm* and the *Hot -Matrix Test* scenarios, the tests were performed similar, following the shown principle:

Static test conditions SOP for the Main / Bypass -Compressor		130 / 114	Barg
Setpoint of Sealgas injection temperature		130	°C
Dynamic test conditions The pressure was reduced from static SOP to dyn. operating pressure		85	Barg
Setpoint of **Sealgas injection temperature**		130	°C
Setpoint of **operating cavity temperature** for the Main / Bypass -Compressor		100 / 210	°C
Setpoint of **cavity design temperature**		250	°C
Dynamic test sections *Each speed was hold until reaching steady state temperatures* 1. Hot start-up simulation up to normal speed		0 ➔ 27.000	rpm
2. Multiple Start/Stops, with steps	Step 1	0 ➔ 9.000	rpm
	Step 2	9.000 ➔ 18.000	rpm
	Step 3	18.000 ➔ 27.000	rpm
3. Multiple Start/Stops, without steps		0 ➔ 27.000	rpm
4. Dynamic test at maximum continuous speed		28.350	rpm
5. Dynamic test at trip speed		29.768	rpm

Figure 20: Warm and Hot -Matrix sCO2 Test data

DGS TEST RESULTS

In this chapter some extracts out of the *Hot Matrix Test* will be represented by different plots (See Fig. 19 for orientation). The focus is on the tests with the highest temperature readings, during testing with sCO2. The readings shown in the plots will be commented to focus on the outcome. A comparison of the readings from the tests with different cavity temperatures is possible, as the tests were performed with similar speeds, pressure, and flowrates.
A further focus is on the cold / ambient testing, without any external heating of the cavity, where the fluid was injected into the seal cavity in liquid phase, flooding the seal.

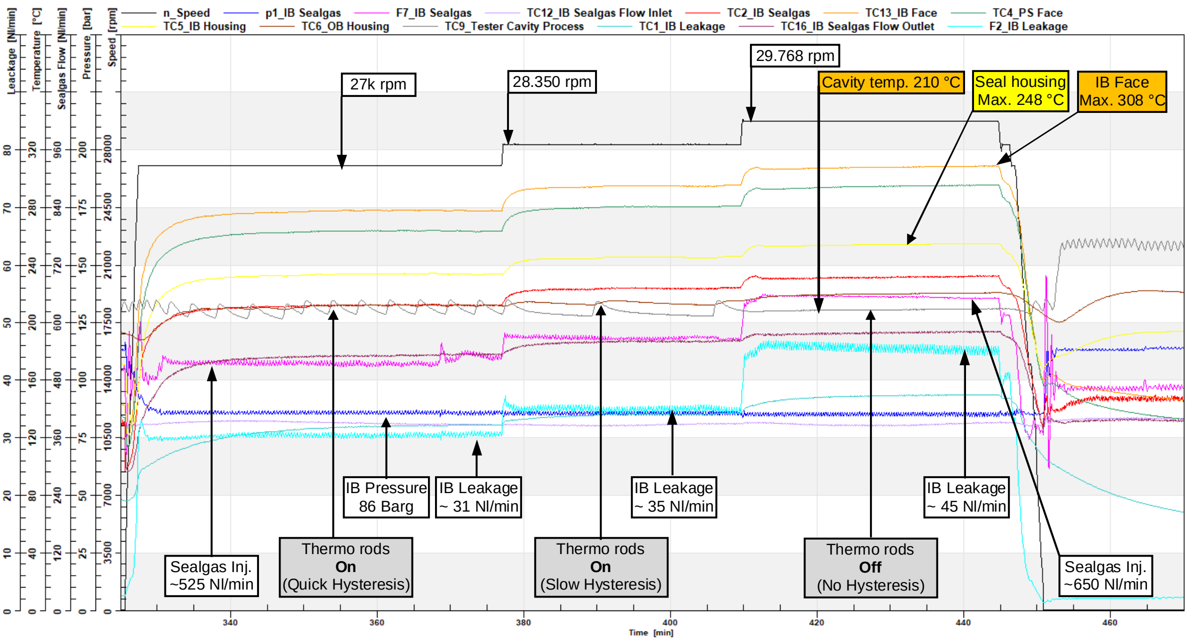


Plot 1: Overview – Extract of the Hot Matrix Test

In general, the first letter of the probes -listed in the legend, on the top of the plot- indicates which parameter is shown with the color of the graph.:

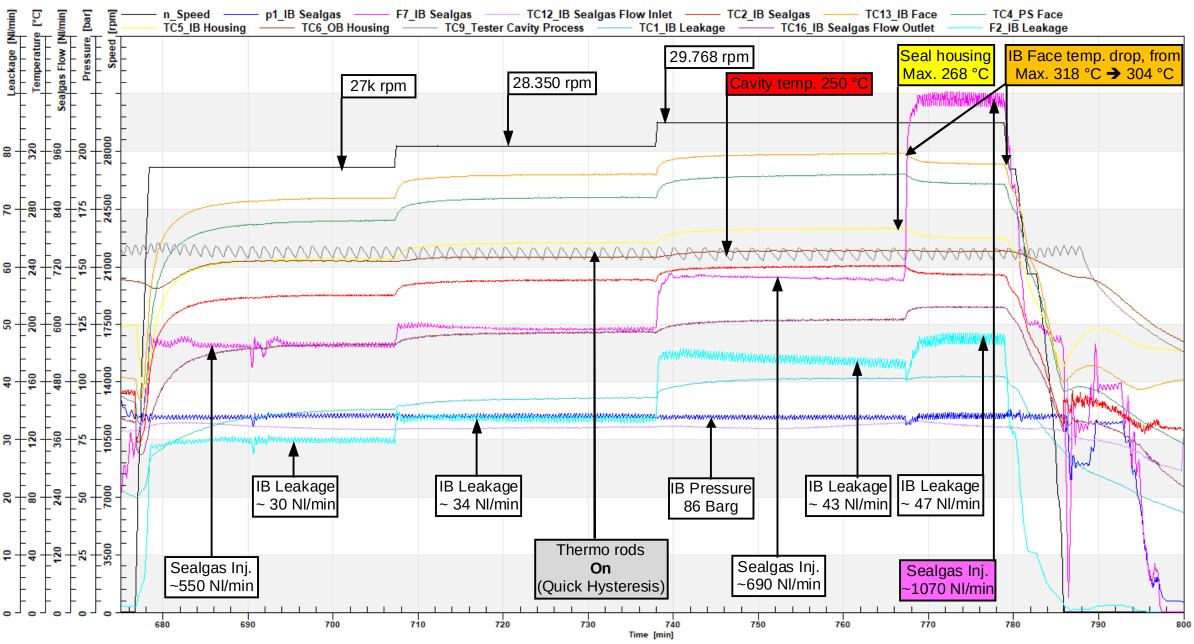
n	=	speed	[rpm]
p	=	pressure	[Barg]
F	=	flow	[Nl/min]
TC	=	temperature	[°C]

Plot 1 shows an extract of the readings from the *Hot Matrix Test*, as an overview. The different sections are highlighted. In the heat-up phase the temperature setpoints were reached by heating the sealgas-inlet-flow up to 130 °C. Also, the cavity was heated up to a temperature of 210 °C.
During section III) and V) The normal speed, MCS and trip speed was set. For the sections IV) and V), the cavity was set up to a temperature of 250 °C. The sealgas-flow was regulated to control the pressure and the overall temperatures in the system.
The readings of the temperature probes (TC1, TC2, TC4, TC5, TC6, TC13, TC16) are reflecting the temperatures as result from the above-described setup values.



PLOT 2: Speed cycling test at 210 °C heated cavity.

Plot 2 and Plot 4 represents, that all the steady state temperatures were reached at the speeds of 27.000 rpm, 28.350 rpm and a 29768 rpm. Also, the seal leakage was stable at each speed and shows the typical increase in correlation with speed.
In plot 2 we can see the performance of the heating rods. The heating rods need to implement additional energy into the cavity up to a test speed of 28.350 rpm, as the inner energy, produced by the seal due to windage, is not sufficient to keep the cavity at 210 °C. That changes, as the seal is operated at trip speed of 29.768 rpm, where the heating rods turn off, as the inner energy of the seal is high enough to keep the cavity at 210 °C by the seal's windage.



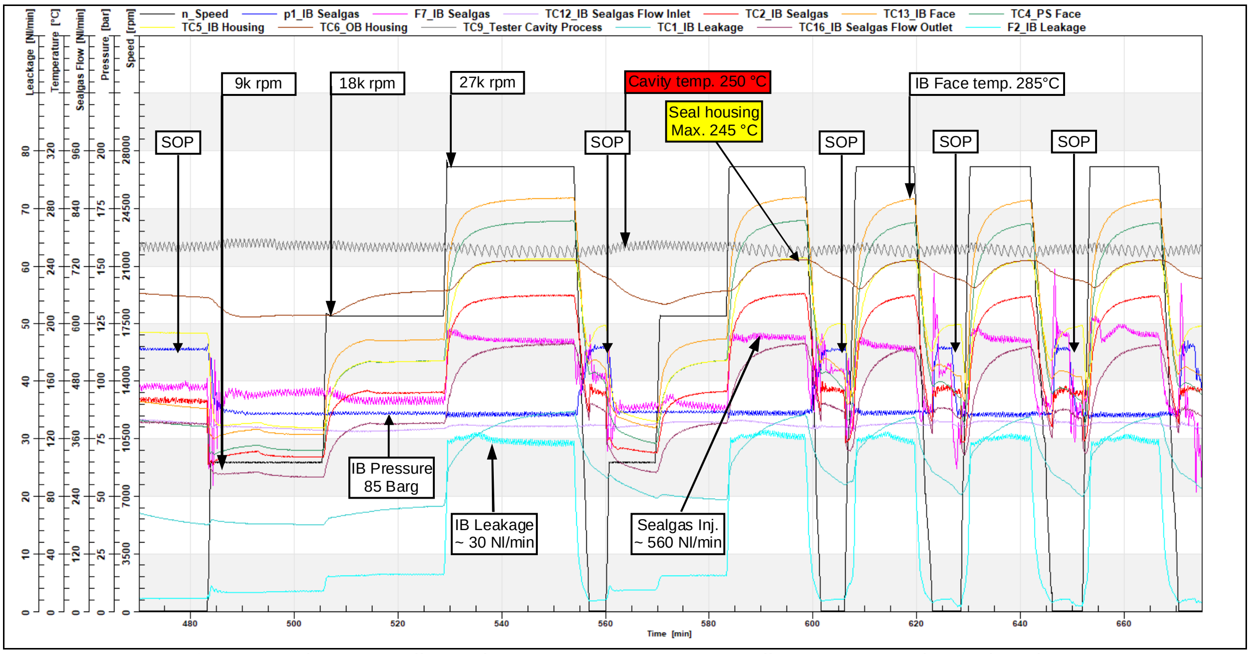
PLOT 4: Speed cycling test at 250 °C heated cavity.

Plot 2 and Plot 4 comparison

From comparison of trip speed conditions (29.768 rpm), while the housing temperature was increased by 40 °C (from 210 to 250 °C) the overall primary seal temperature (TC 4, 7, 11 and 13) increased by 10 to 20°C only.

For both tests, the cooling flowrate conditions were kept stable, whereas the heating rods had to be switched-on only to reach the 250°C housing temperature. From this comparison, it is evident that the seal temperature increase is not proportional to the housing temperature increase. In addition, the heat generated by the seal due to windage comes to a balance as the external heat source provides constant energy, but the seal temperatures do not increase and reach to a steady state.

During the 250°C test, the sealgas flowrate was increased from ~690 to 1070 Nl/min to see the cooling effect on the seal. As a result, the seal temperature decreased by ~10 °C, whereas the heating rods heat supply was reduced from 2,34 to 1,35 kWh, still adding additional external energy to the system.



PLOT 3: Extract from Start / Stops until steady state of temperature readings at 250 °C heated cavity 470-675 min

Plot 3: The cavity was heated up to the setpoint of 250 °C, with a SOP of 130 Barg. Afterwards the speed was increased with 500 rpm/sec up to max. 27.000 rpm and the pressure was reduced progressively to 85 Barg, to simulate the compressor start-up. The dynamic conditions were kept until the temperature readings reached steady state.

During the test sequence the seal showed a reliable and repeatable behavior, from temperature and leakage flowrates point of view. The maximum temperature of the Inboard seal face reached ~285 °C, steady state. The primary seal leakage was at ~30 Nl/min, when operating at 27.000 rpm.

Liquid CO2 flooded test

To simulate a prolonged, pressurized stop of the compressor at ambient to cold housing temperatures, the inboard seal cavity was fully flooded with liquid CO2 during static conditions. After the static test sequence, the seal was started up until the liquid phase changed into transient / supercritical -CO2. Beside the seal performance, also the startup torque was measured to indicate the impact of liquids and potential ice formation at the primary vent.

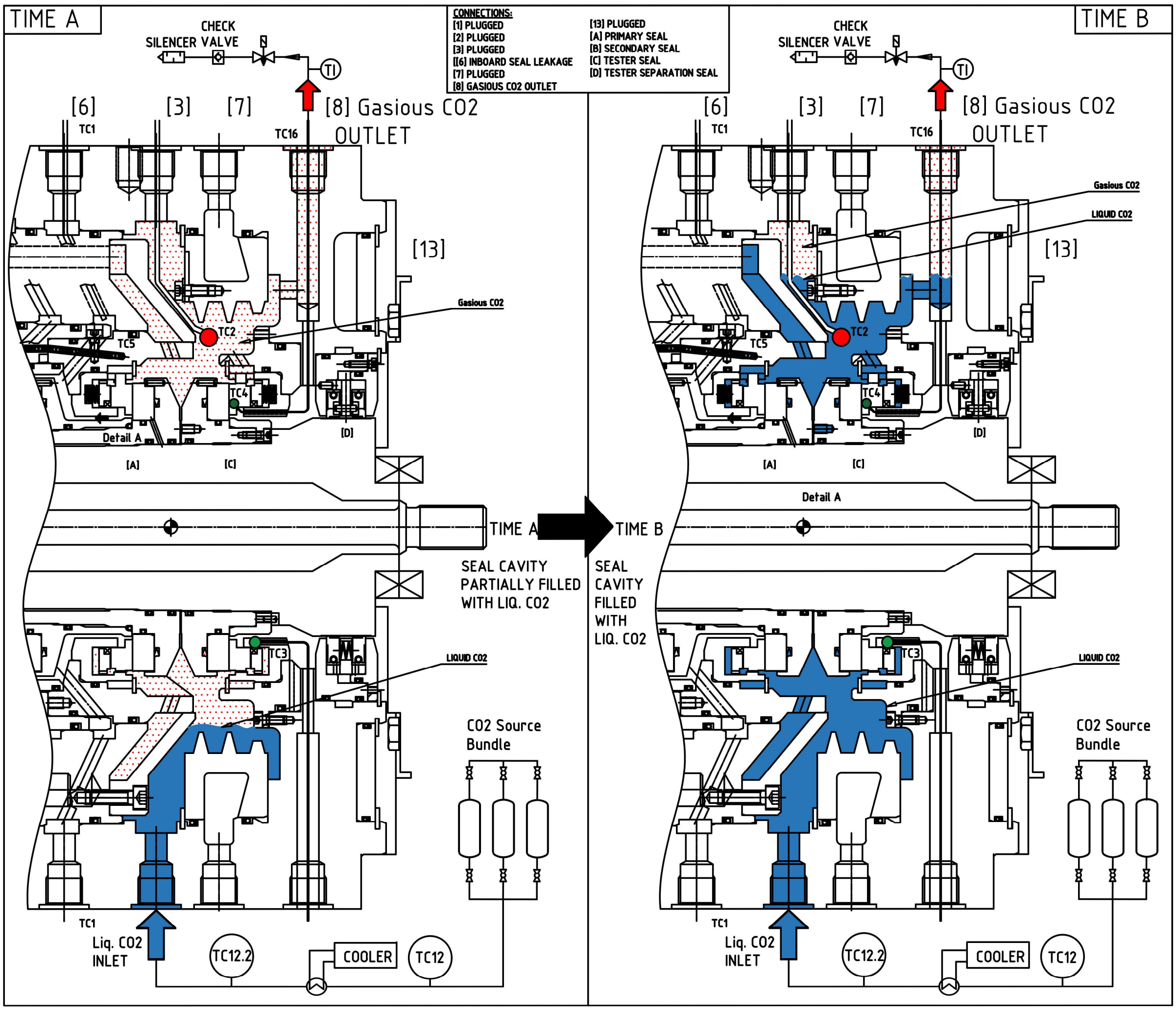


Figure 21: Tester setup with liquid CO2, flooding the seal cavity.

As the supplied CO2 is stored at the boiling line, a special test set-up was chosen, to change the fluids phase (See figure 22) and maintain it liquid at the seal for several hours. Here for the fluid was cooled down to ~15°C @ ~57 Barg, prior entering the tester cavity (See Figure 22). The cavity was filled from the bottom (6 o'clock) and vented at the top (12 o'clock) to achieve full flooding of the seal area (See Fig. 21, blue = liquid CO2). To monitor the phase condition and flooding process, additional thermocouples were installed (See figure 21, TC2, TC12 and TC12_2).

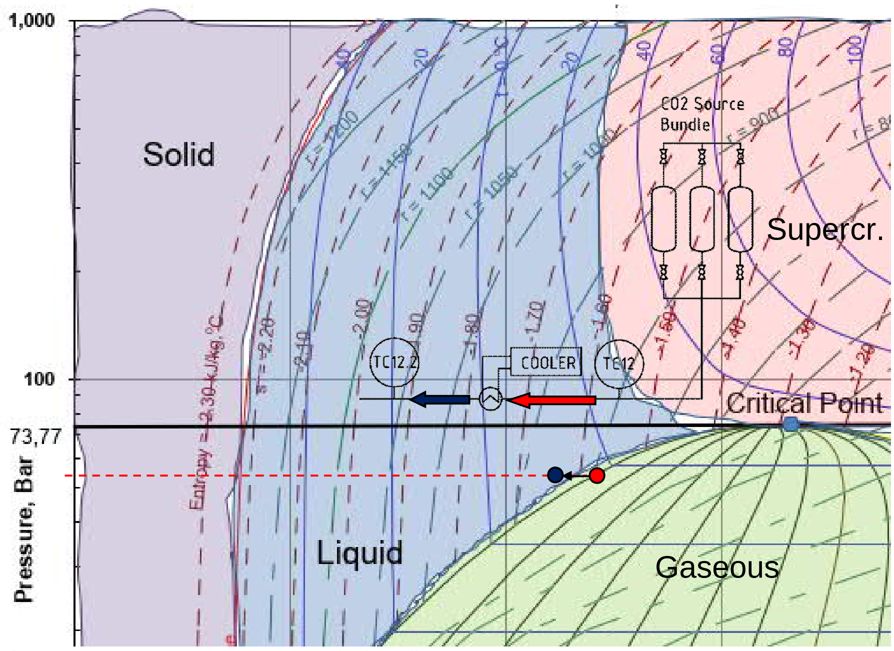
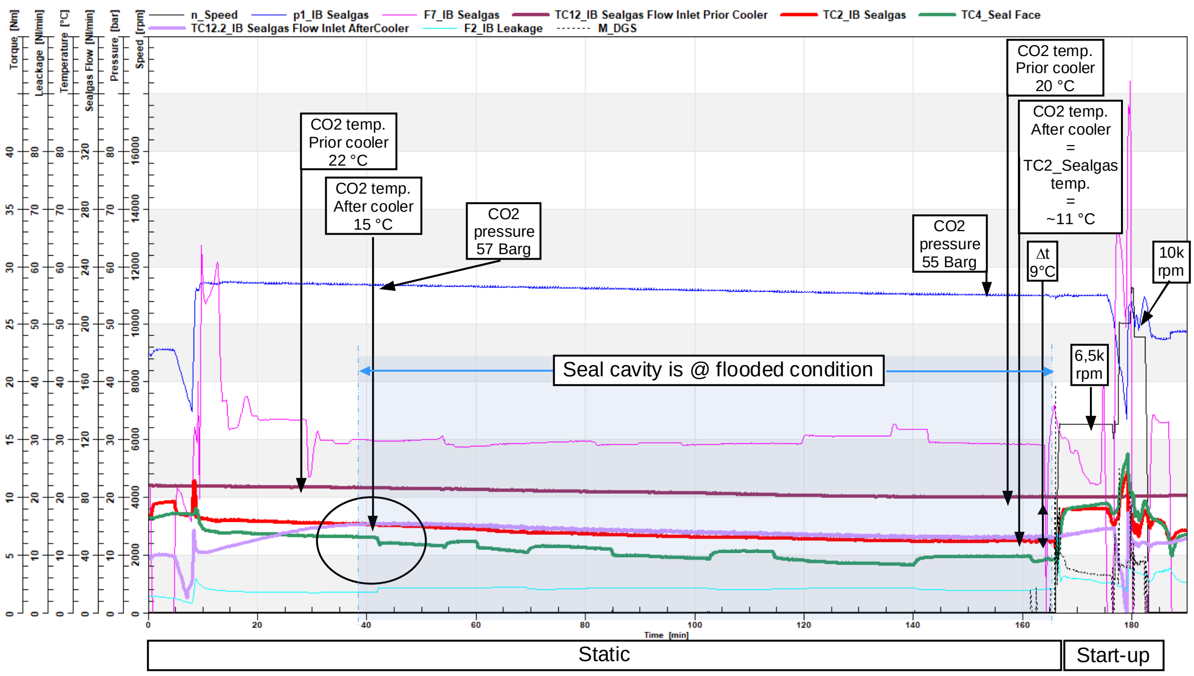
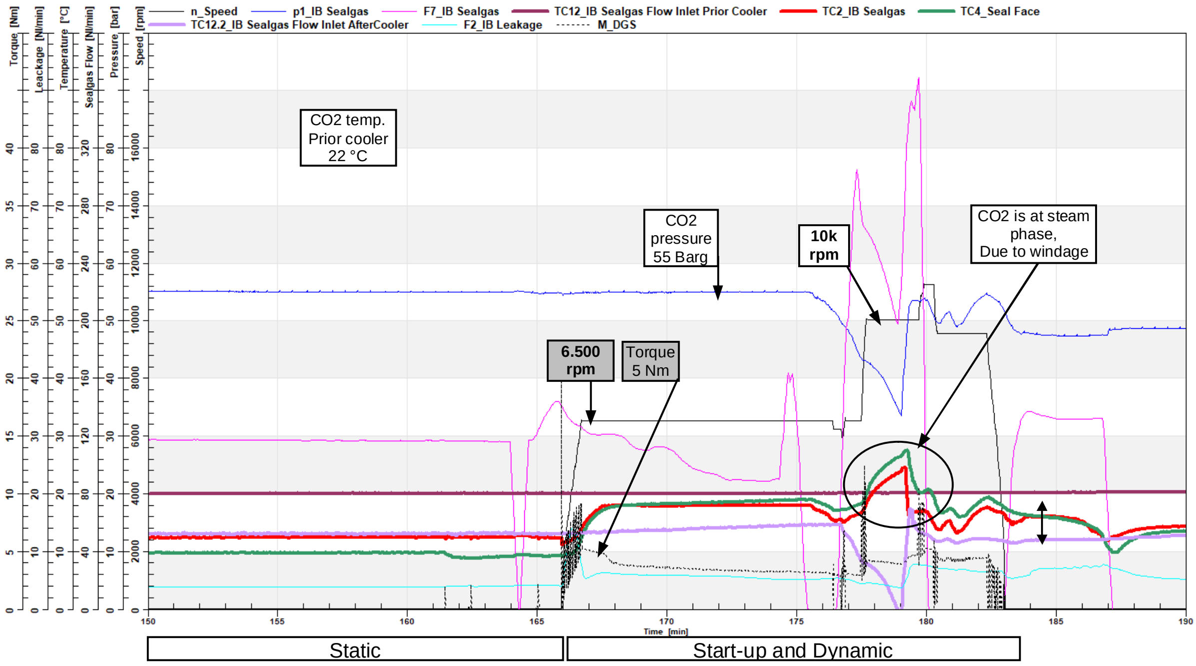


Figure 22: Extract from CO2 Enthalpy-diagram

The trends of **PLOT 5** show that the seal is fully flooded with liquid CO2. When the readings of the thermocouple (TC2_IB Sealgas), at the 12 o'clock position of the seal cavity, show the same temperature as the thermocouple (TC12_2), after the CO2 cooler.



PLOT 5: Filling process of seal cavity with liquid CO2



PLOT 6: DETAIL A: Start-up under liquid CO2 conditions

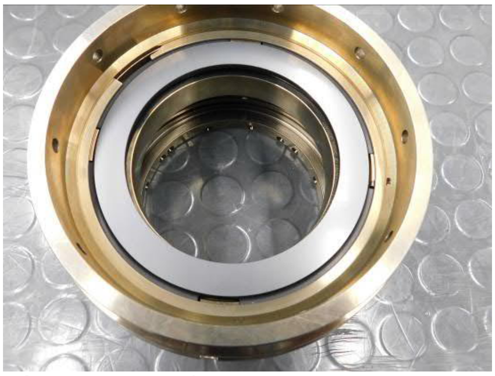
The start/up sequences were several times repeated. The torque development during start/up was stable, uncritical and comparable to the startup measurements at sCO2 condition.

Summary of the test campaign

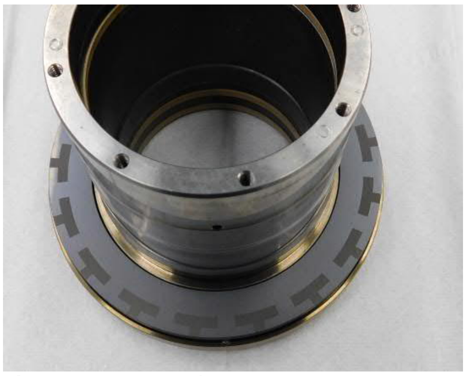
The test campaign was finished successfully, matching the agreed test matrix (See Fig. 19), including: COLD-, WARM- and HOT Testing.

The overall test results met the predicted performance, obtained through analytical studies, calculations, and the experience from previous, successfully implemented sealing solutions. The suitability of Flowserve sealing technology has been proven by stable and repeatable trend data, measured at different, severe test conditions (including the start-up @ flooded conditions) in combination with the visual appearance of the seal components, post testing.

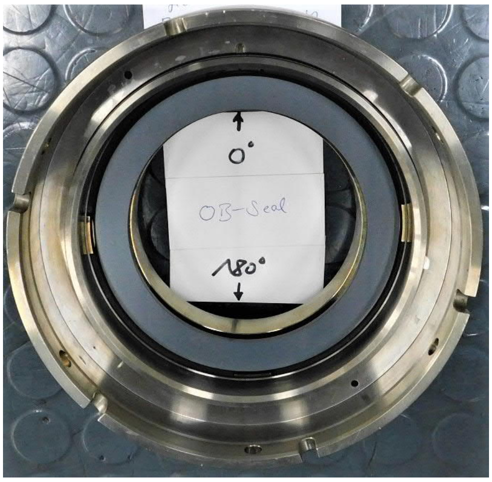
- Visual appearance of the seal after the test campaign



Picture 4: IB Stationary Face



Picture 5: IB Rotating Face



Picture 6: OB Stationary Face

Picture 7: OB Rotating Face

Summary & Conclusion

The high demand from the emerging renewable energy industry, utilizing sCO2, is closely related to the availability of high-end shaft sealing solutions for the turbomachinery equipment. DGS technology development was driven in the past mainly by Oil& Gas Industry – i.e. higher operating pressures - but it is a key partner also for the success of energy transition process.

The paper gave a brief insight regarding analytical & design process and final performance testing of a DGS for a sCO2 Re-compression Closed Brayton Cycle (RCBC) pilot plant.

The conducted test campaign mirrored as close as possible the operational conditions in the compressors and provided needed information and performance results to close the gap between analytical studies and real application testing, allowing to improve knowledge and experience for both Compressor and DGS OEMs.

The seal was designed to meet the actual sCO2 requirements according to Flowserve design standard and based on existing experience in CO2 pump & compressor industrial applications, where Flowserve cumulated experiences in sCO2 applications for pilot projects.

The seal shows overall stable and repeatable performance, with a very low leakage level which is fundamental for the Close Loop Process to minimize emission and reducing refilling needs. At the same time, the low leakage reduces risk of ice formation on the primary-vent-side, improving seal reliability, as well minimizes seal system complexity.

The extensive DGS test campaign included several operational and transient conditions at the most severe scenario for a dry gas seal, such as liquid phase CO2 @ DGS. The positive result of all tests confirmed that Flowserve seal technology can be safely operated at multiphase CO2 conditions for turbomachinery.

The seal test data available are being analyzed by FLS in collaboration with BH and are of extreme importance to validate numerical calculation programs and thermal models for the prediction of seal performance for high temperature turbomachinery, where the DGS is a key element to ensure efficiency, safety and reliability for emerging renewable energy industry.

Acknowledgement
The authors would like to acknowledge and thank both FLS and BHNP team members for the hard work and collaboration that lead to the success of this project.

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