45TH TURBOMACHINERY & 32ND PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 12 – 15, 2016
GEORGE R. BROWN CONVENTION CENTER

HIGH PRESSURE, HIGH TEMPERATURE SHAFT SEAL FOR A MULTIPHASE SUBSEA PUMP

Bernhard Gilch
Application Engineer
EagleBurgmann
Wolfratshausen, Germany

Bernhard Gilch is responsible for Technical Sales Support for Mechanical Seal applications at EagleBurgmann Germany in Wolfratshausen. He joined EagleBurgmann in 2008. His responsibilities include sales support, application engineering, and project fulfillment on sealing technology in upstream, midstream and downstream in the oil and gas, in the Petrochemical and Chemical market.
Bernhard Gilch has a Bachelor of Engineering from the Baden-Wuerttemberg Cooperative State University.

Simon Gassmann
Subsea Development Engineer
Sulzer Management Ltd, Pumps Equipment
Winterthur, Switzerland

Simon Gassmann started his career with Sulzer in 2008 as a Development Engineer where he is part of the "Subsea Pumps" team of the Product Development Unit in Winterthur, Switzerland.
Simon graduated as a Master of Science in Mechanical Engineering at the Swiss Federal Institute of Technology in Zurich in 2008 and as a Master of Science in Subsea Engineering at the University of Aberdeen in 2013.

Thomas Felix
Manager Subsea Pumps
Sulzer Management Ltd, Pumps Equipment
Winterthur, Switzerland

Thomas Felix studied Mechanical Engineering at the University of Magdeburg, Germany where he received his MSc in Turbo Machinery in 1994. He joined Sulzer in Brazil in 1995 as Product Engineer Pumps. Since 2000 he is working for Sulzer in Winterthur, Switzerland in several positions in the pump development as mechanical analyst and hydraulic engineer in the area of multiphase pumps. In 2007 he became Manager Subsea Pumps, heading up Sulzer's subsea pump development team.

Peter Droescher
Head of Engineering Pump Seals
EagleBurgmann
Wolfratshausen, Germany

Peter Droescher has more than 25 years of experience in sealing technology. After several years in engineering, R&D and compressor seals he took the engineering responsibility for the EagleBurgmann Pump Seal products.

Peter Droescher holds a Diploma in Mechanical Engineering of the University of Applied Sciences Munich

Marcelo Inforsati
Subsea Development Engineer
Sulzer Management Ltd, Pumps Equipment
Winterthur, Switzerland

Marcelo Inforsati studied Mechanical Engineering at the University of Sao Paulo State - UNESP, Brazil where he received his degree in Mechanical Engineering in 1993. He joined Sulzer Brazil in 1995 as Product Engineer Pumps. From 1998 he joined Siemens Ltd. in SP- Brazil as Design Engineer for high-voltage circuit breakers. In 2005 he joined Haver & Boecker Latinoamericana as Manager of Mechanical & Electrical Engineering Department and automation. Since 2008 he is working for Sulzer in Winterthur, Switzerland as Development Engineer, first in the design team for "Special Pumps" and in 2013 he joined "Subsea Pumps" team.

ABSTRACT

The paper describes a totally re-engineered mechanical seal for multiphase subsea pumps with a focus on extended metal and seal face material selection as well as more stable seal behavior, achieved with a new face concept, which provides enhanced reliability and robustness. It reports the design process, starting from the project description with definition of targets followed by a theoretical evaluation of the seal performance and a description of the final design features.
Increasing demand for high pressure and high temperature (HP/HT) pumps in Subsea multiphase applications requires the development of mechanical seals designed for pressure levels up to 15 kpsi and product temperatures up to 350°F with the capability to handle reverse pressure conditions. With the high pressure requirements the application of spring-energized polymer gaskets as static and dynamic secondary seals using a specific design for enhanced reverse pressurization capability were selected. To achieve enhanced robustness of the seal faces in transient dry running condition which may occur during upset conditions (such as reverse pressure) the design was optimized to include microcrystalline diamond coated seal faces. A detailed analysis of face deformation and seal performance under load with a combined structure and fluid analysis software together with an extensive test campaign and specific cooling jacket features lead to a robust mechanical seal design with optimized pressure distribution and mechanical contact zones.

1. INTRODUCTION

The sealing of the shaft in multiphase pumps for subsea oil exploration is a very demanding application for mechanical seals. The service requires extraordinary reliability, longevity and safety of the applied product. Subsea multiphase applications, planned for the near future will exceed the capability of state-of-the-art mechanical seals. The seal applications are characterized by the demand for highly reliable sealing of multiphase process fluids in corrosive environment with high pressure, high temperature (HP/HT) and significant rotational speed. During upset conditions dynamic reverse pressure can occur. This means the differential pressure will change from a positive level to a negative level while having a certain rotation speed of the shaft. This leads to a variation of hydraulic forces in the seal gap which can cause dry running of the mechanical seal faces at significant sliding velocity.
Available engineered mechanical seals for subsea applications are typically equipped with shrink fitted silicon carbide seal faces which are pressurized from the inside and with elastomers as secondary sealing elements. The targeted applications with higher differential pressure at significantly increased pressure level and elevated operating temperatures does not allow using elastomeric secondary seals due to the increasing risk of extrusion, deformation under static load or explosive decompression. State-of-the-art polymer gaskets which would be the preferred secondary seal solution for the application do not provide the requested reverse pressure capability.
For the sealing of high pressures with fluids containing abrasive particles, hard face materials are required to avoid unacceptable deformations or to minimize abrasive wear. Typically silicon carbide face materials are used which do not allow any dry running and have only limited capabilities with regard to poor lubrication. Microcrystalline diamond coated seal faces have been applied in multiphase applications with great success. For today’s state of the art shaft seals used for multiphase subsea boosting applications inside pressurized seal design has been applied. As the diamond face coating does not allow surface finishing after shrink fit process it is not applicable to inside pressurized seals.

The mentioned limitations were the main drivers to start a development of a new elastomer free mechanical seal without shrink fitted seal faces. A study was done on the possibility to re-engineer the mechanical seal design for this high duty application.

2. TARGETS OF THE DEVELOPMENT

The target of the development was to re-engineer the current mechanical seal design to enlarge the operating conditions for a pressure level of 1035 bar and for a maximum product temperature of 177°C with the capability to handle dynamic reverse pressure conditions. In subsea applications state of the art mechanical seals are strongly dependent on the properties of the main components of the mechanical seal which are the seal faces and the elastomers as secondary sealing elements. To enlarge the field of operation mainly the seal design concept had to be changed to have the possibility to use different seal face material and secondary sealing elements.

The first target of the re-design was to change the design to a mechanical seal which is pressurized from the outside. The significant advantage of the externally pressurized seal is that loosely inserted seal faces without bandage can be used which allows the installation of microcrystalline diamond coatings. Crystalline diamond coatings offer an outstanding abrasive resistance due to their great hardness and, at the same time, a considerably improved performance under poor lubrication conditions due to the low friction coefficient. This makes the mechanical seal design more robust and capable to handle transient dry running.

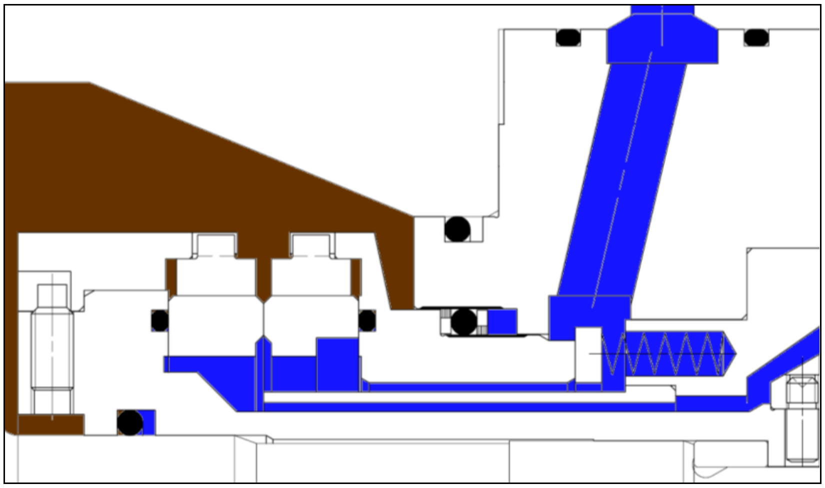

Figure 1: Principle of existing inside pressurized mechanical seal in normal operating condition (barrier pressure (blue) > product pressure (brown))

A second target in the specification was to develop an elastomer free seal. The reason is that elastomeric O-rings extrude at high pressure in combination with high temperature. Therefore spring-energized polymer gaskets as static and dynamic secondary seals with higher design limits should be used. The challenge has been the dynamic reverse pressure capability. Normal polymer gaskets are spring energized but are just capable to handle pressure from one side. In case of reverse pressure the spring force is not high enough to achieve a sealing effect of the lip seal. By changing the design of the polymer gasket to a semi-exposed spring will help to achieve a dynamic reverse pressure.

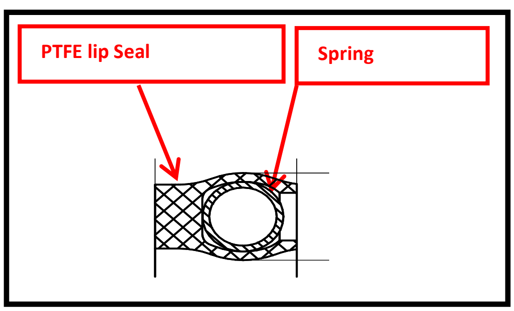

Figure 2: Spring energized polymer gaskets

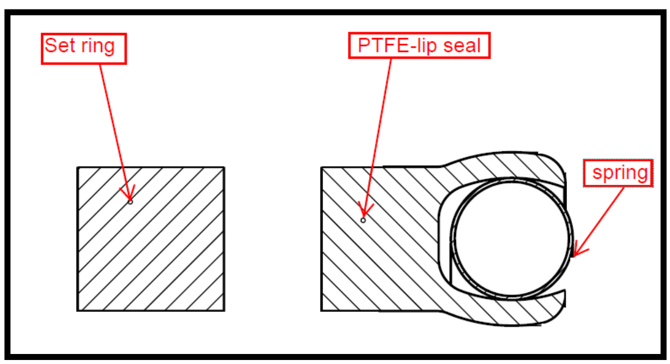

Figure 3: Polymer gaskets with semi-exposed spring and set ring

Furthermore, a new cooling jacket had to be designed and implemented into the pump in order to protect the barrier fluid and the mechanical seal components from temperatures above 120°C for product fluid temperatures up to 177°C. Above 120°C the water-glycol mixture used as barrier fluid becomes unstable and starts to chemically degrade which has to be avoided.

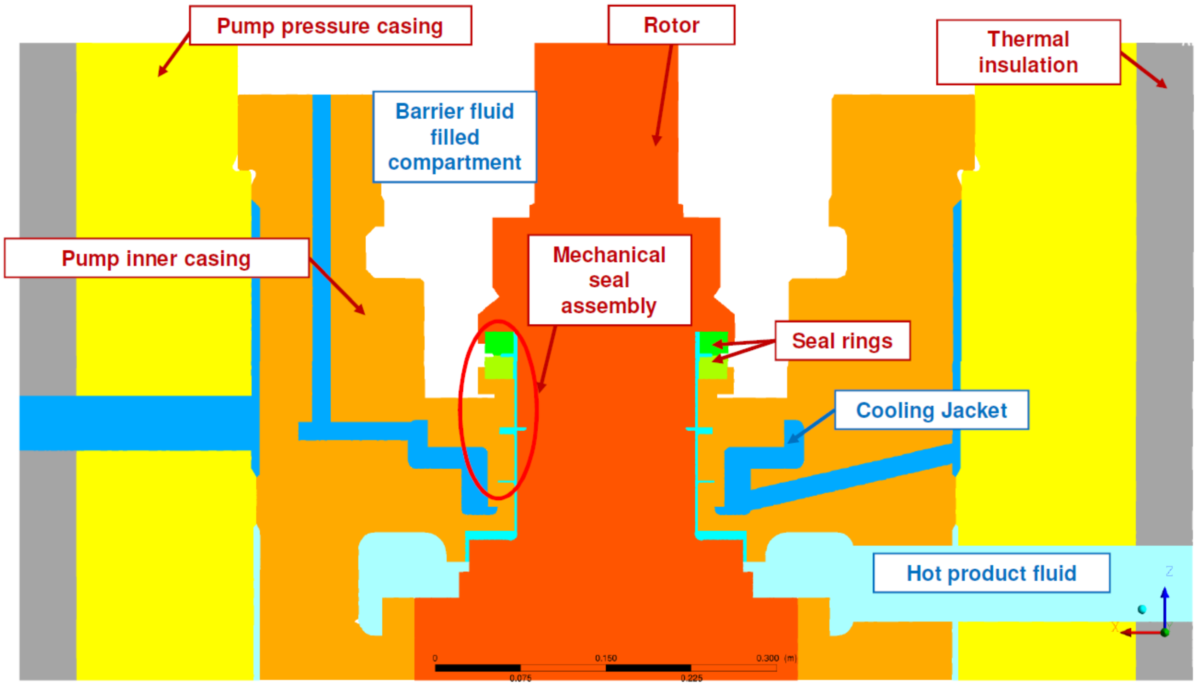

Figure 4: Integration of cooling jacket into subsea multiphase pump; Illustration shows extract of subsea pump cross section at the drive end (DE)

3. MECHANICAL SEAL DESIGN

The new developed mechanical seal is externally pressurized. This means for normal operation the seal faces are pressurized from the outside with the barrier fluid which leaks into the process medium contacting the seal faces at the inner diameter.
The key design feature of the mechanical seal for high duty applications is the independence from shrink fitted seal rings. The seal faces are loosely inserted into the face carrier, connected by a torque transmission system. Normally used shrink fitted seal faces inherently have temperature and stress limits which are not present with loosely inserted seal faces. A further design advantage is the fact that different seal face materials can be used. So far carbides have been used as seal face material. This material allows polishing after the shrink fit process. With microcrystalline diamond coated seal faces this production step is no longer possible, due to the reason that the grown diamonds on the surface are harder than the polishing process. In this case the flatness of the seal face has to be assured during the production for the diamond seal face ring. With the loosely inserted design these diamond coated seal faces can now also be installed.

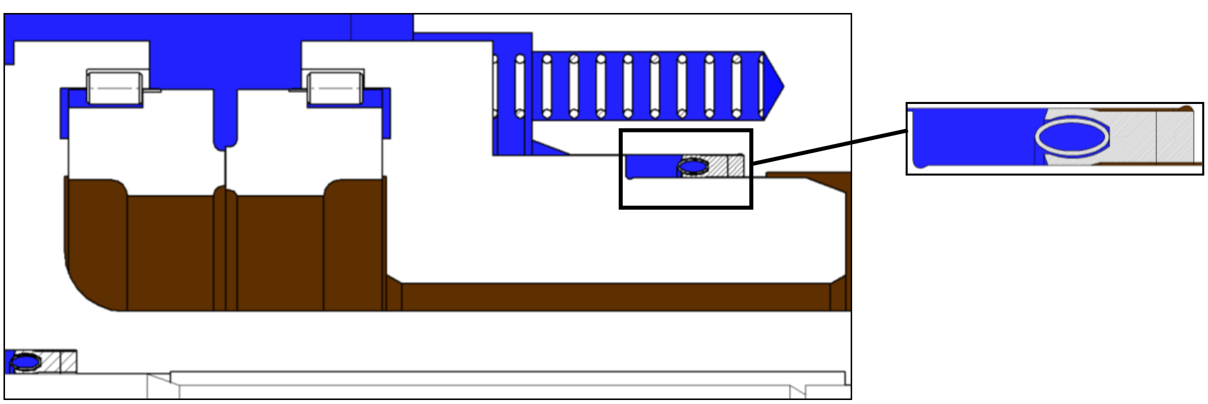

Figure 5: Principle of new developed external pressurized mechanical seal with loosely inserted seal faces in normal operating condition (barrier pressure (blue) > product pressure (brown))

To achieve maximum longevity the contact forces acting between the seal faces were reduced to a minimum with a controlled hydraulic balancing over a wide range of operating conditions. Also contact forces acting during reverse pressurization were reduced significantly.
For reverse pressure the product pressure is higher than the barrier pressure. The polymer gasket moves to the other side of the groove. Due to the pressure the semi-exposed spring will be axially compressed resulting into a radial force of the spring against the lip. Therefore the lips are pressed against the groove and a sealing effect is achieved.

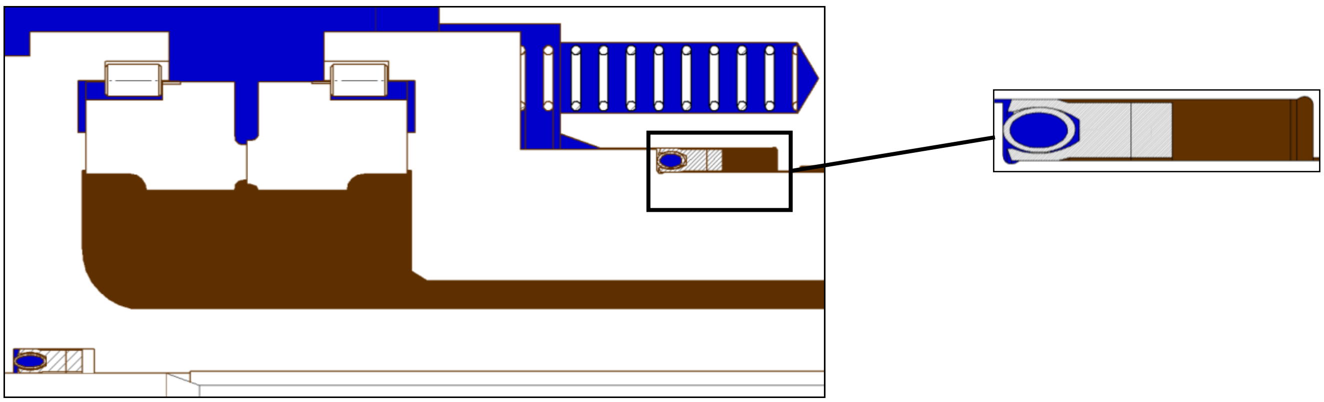

Figure 6: Function of the polymer gasket in reverse pressure condition (barrier pressure (blue) < product pressure (brown))

4. SEAL PERFORMANCE CALCULATION AND COOLING JACKET THERMAL ANALYSIS

After defining the design concept extensive seal performance evaluation has been carried out with a combined fluid and structure analysis.
Important was to evaluate the seal gap behavior at all operation conditions. The following plot illustrates the calculated face contact forces as a function of operating speed and pressure. In the below plot a pure gas has been considered on the product fluid side. The criteria for the judgement of the face contact are respecting the available data related to material performance.

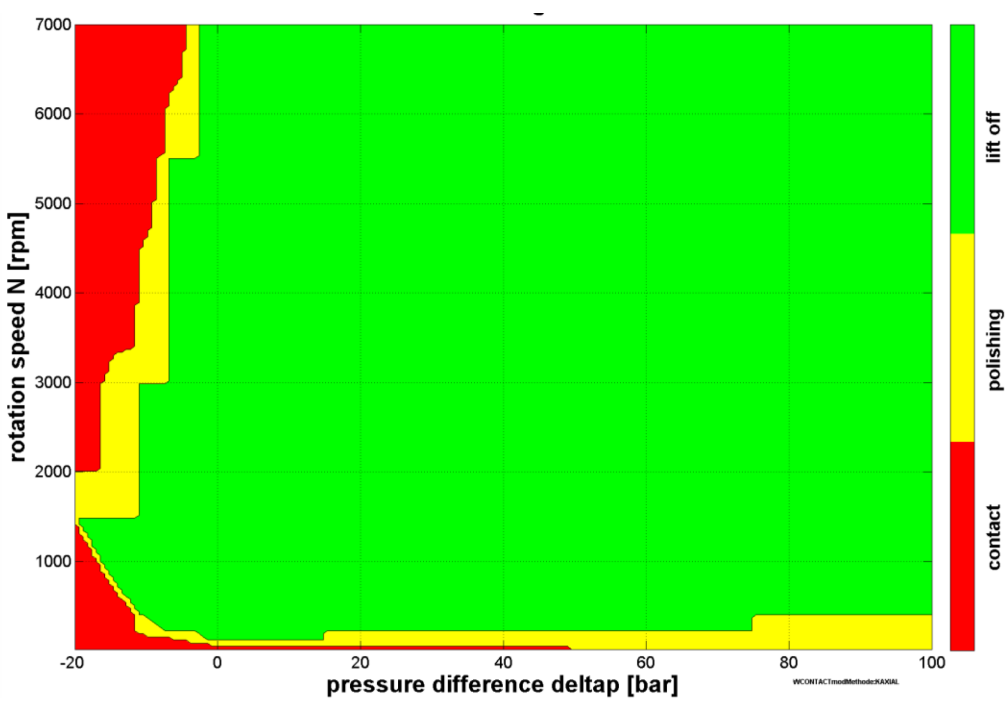

Figure 7: Seal gap behavior

The classification criteria for the different zones are based on theoretical and experimental studies. The seal is completely lifted off in the green zone which means a full hydrodynamic fluid film is established. At this condition the seal rings are not in contact and no wear is expected. In the yellow and red zone a certain contact force occurs between seal face und seat. The expected wear depends on the contact force, the sliding speed, the lubricating properties of barrier fluid, the amount of load, and on the surface hardness. With microcrystalline diamond coated seal faces the wear is significantly minimized. In the yellow zone the wear is limited. A full hydrodynamic fluid film is established after the polishing of surface roughness. In the red zone the level of the forces and the shape of the seal gap cannot be compensated by a hydraulic lift off.
The calculation plot illustrates a wide operating range of the mechanical seal. Even at reverse pressure and low speed a certain lift off the seal rings is achieved due to the hydrodynamic effect of the laser grooves at the seat.
The next plot shows the expected leakage rates, again under the assumption of pure gas on the product fluid side.

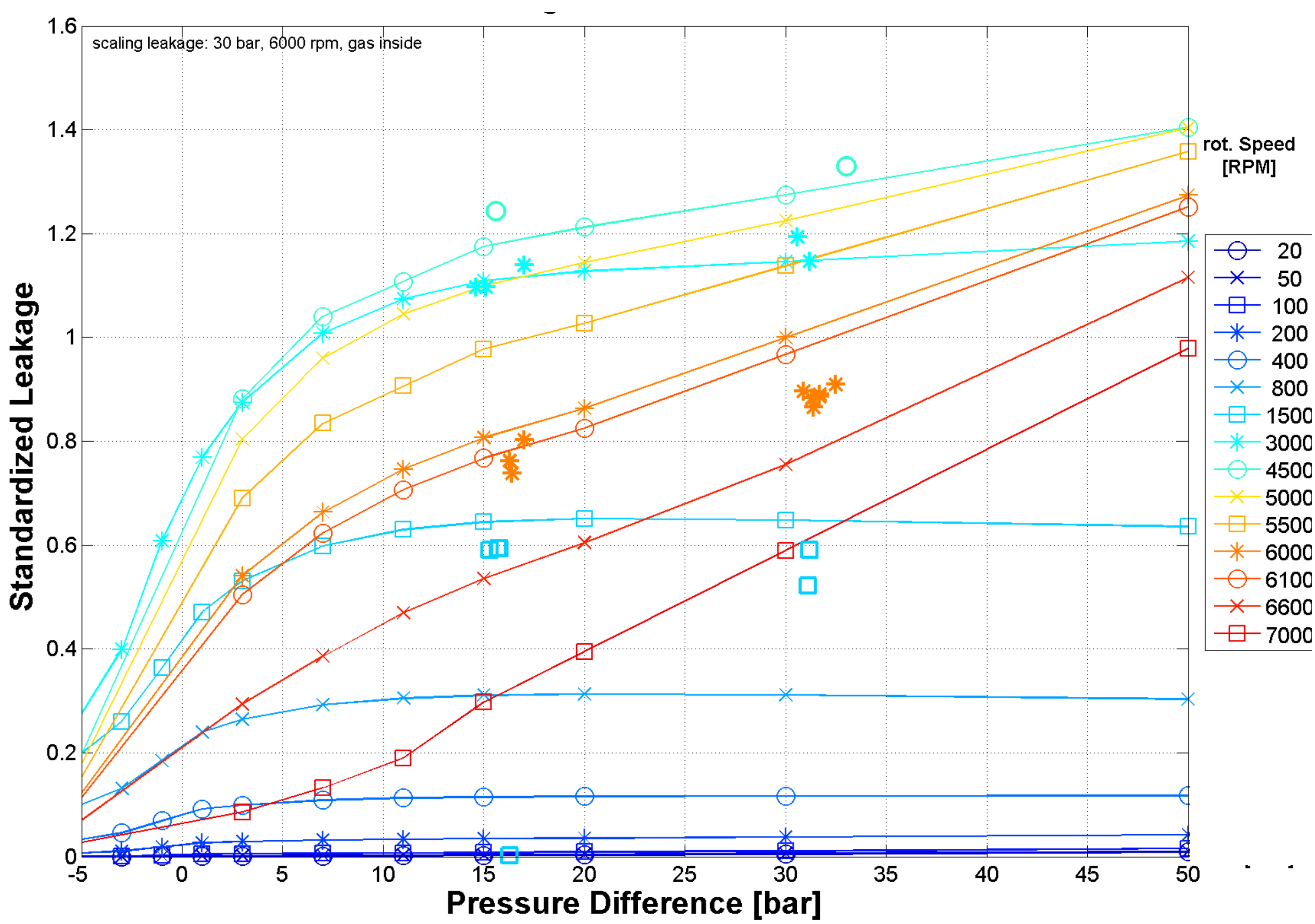

Figure 8: Mechanical normalized seal leakage rate (points next to chart lines represent measurements)

The variation of rotation speed is illustrated by different colored lines. In general the seal has been optimized with certain leakage level for a safe operation. At low reverse pressure the seal is still lubricated and cooled by the leakage.

The power generation of the mechanical seal leads to a temperature rise of the seal. The temperature difference between the barrier fluid and the hottest point on seal ring has been evaluated. For all listed operation condition the seal gap is open to the outside diameter (V-gap) and the seal is in a stable behavior. In case of an external pressurized seal, a V-gap leads to a higher leakage, better cooling and hence reduced temperatures which finally results into a stable behavior, see schematic explanation below.

Figure 9: Tendency of seal gap geometry for an external (left) and internal (right) pressurized seal. The red arrow indicates the leakage path (leakage from high pressure barrier fluid towards lower pressure process fluid).

For an internal pressurized seal, thermal deflection of the seal rings might open the gap towards the process fluid and narrow towards the (inner) barrier fluid side, this can lead to a cut-off of the leakage and hence to even higher temperatures; therefore it is called "self-amplifying".

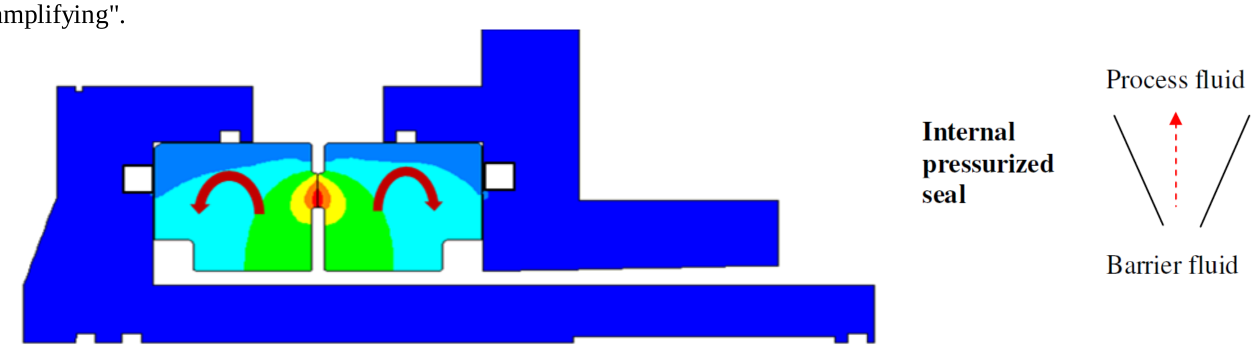

Figure 10: Thermal deflection of seal rings

The next plot shows the temperature distribution of the seal rings for one of the operating conditions (6000 rpm, 30 bar differential pressure, mixture of water-glycol on barrier fluid side and liquid on product side, barrier fluid temperature 50°C, product fluid temperature 120°C).

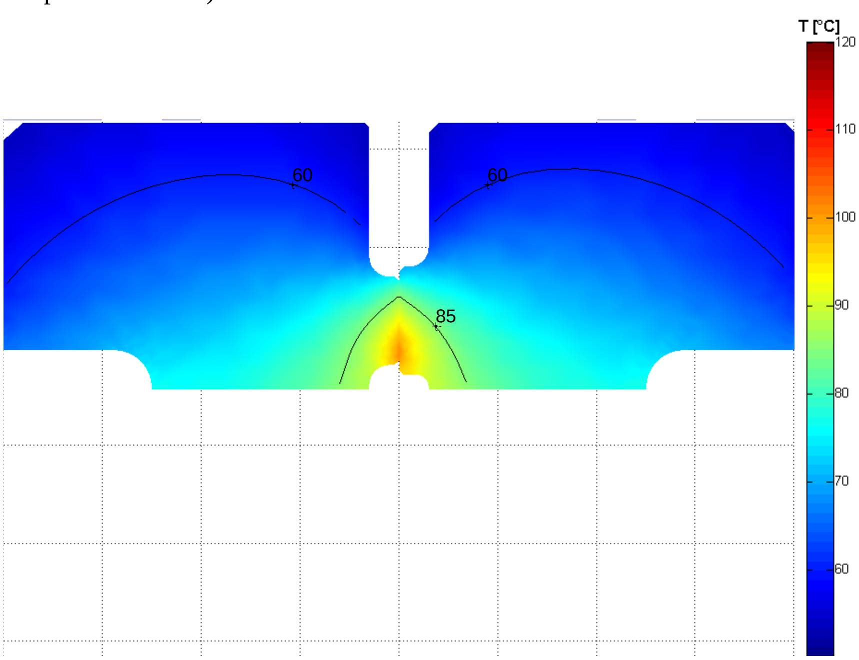

Figure 11: Temperature distribution at the mechanical seal rings

The temperature distribution is important to avoid vaporization and exceeding the ageing temperature of the barrier fluid.
The FE calculations showed that the influences of the temperature on the seal face have only minor impacts. Due to the reason that the seal rings are no longer combined with a shrink fit connection, less radial deflection goes into the seal face. This makes the loosely design more stable in operation.

In addition to the mechanical seal performance calculation extensive CFD calculations have been done to define and validate the cooling jacket temperature distribution in the vicinity of the mechanical seal when the pump is being operated and after shut down (transients). The below CFD results show the temperature distribution around the pump DE mechanical seal and the cooling jacket for one set of operating conditions (6000 rpm, pure gas on product side at temperature 177°C). The below temperature plot shows that the maximum temperature in the vicinity of the mechanical seal and the cooling jacket is well below 120°C.

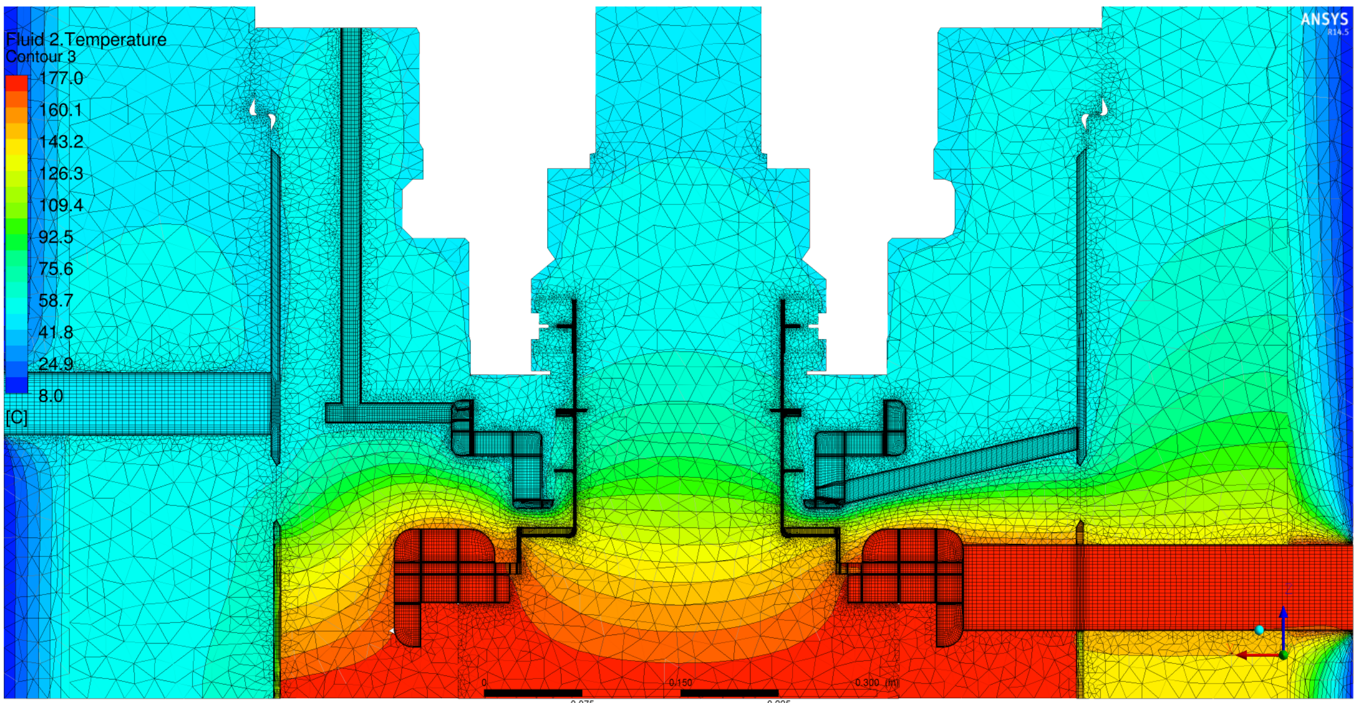

Figure 12: Temperature distribution of pump and mechanical seal components at pump DE domain for 6000 rpm and with gas at 177°C on product side

5. SEAL QUALIFICATION

The secondary sealing elements of the mechanical seal assembly have been validated to the full design pressure of 1035 bar (differential pressure maximum 250 bar) before doing the performance tests of the complete seal assembly. The secondary seal tests also included a reverse pressure test and the evaluation of the friction forces under dynamic axial movement for different absolute pressures (up to 1035 bar) and for various differential pressures.

To verify the design conditions and calculations of the mechanical seal several R&D performance tests for the complete mechanical seal assembly have been done.

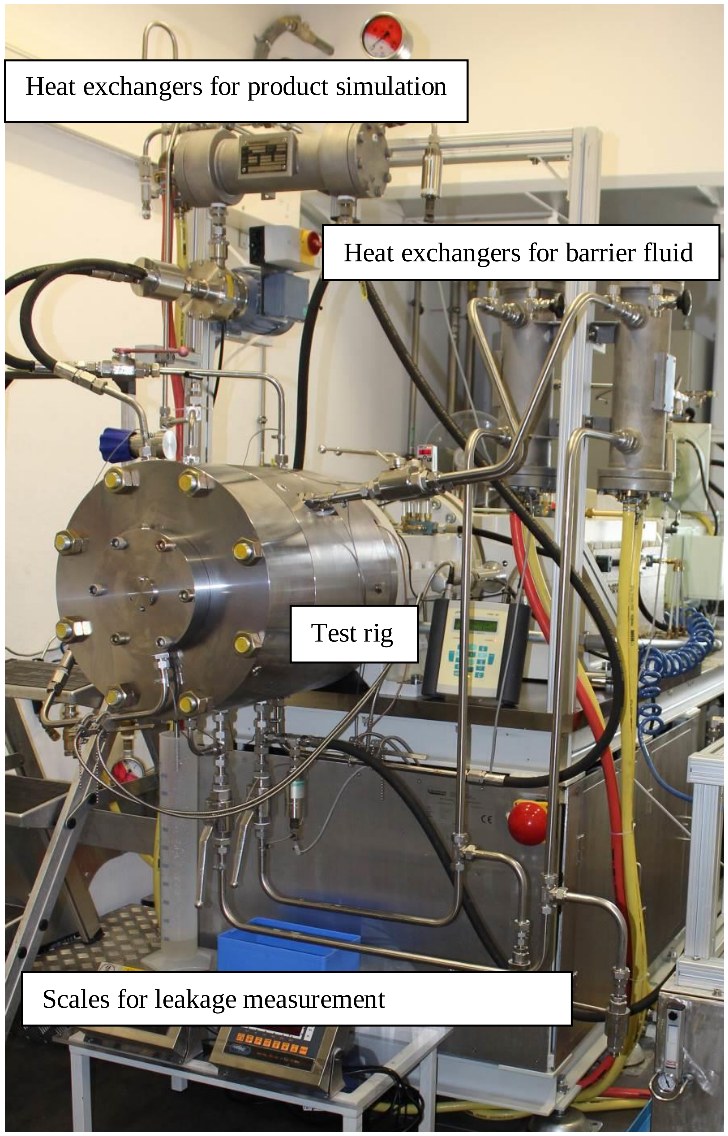

Figure 13: Test rig for the mechanical seal performance test

The first test was to verify the performance of the mechanical seal without product simulation:

Test parameter	Value
Product medium	Air at 1 atm
Barrier medium	Water/Glythermin NF 30/70
Speed	1500; 3000; 6000 1/min
Barrier pressure	15; 30; 80 barg
static pressure	30; 80; 250 barg
Temperature	< 80 °C
Start-Stop	10 times at 30 bar and target speed of 6000 1/min

Table 1: Test parameters for seal performance tests

The second test was to verify the performance of the mechanical seal with product simulation:

Test parameter	Value
Product medium	Water/Glythermin NF 30/70
Barrier medium	Water/Glythermin NF 30/70
Speed	1500; 3000; 6000 1/min
Barrier pressure	20; 35; 85 barg; 20 → 4 bar for simulation of reverse pressure
Product pressure	5 barg
static pressure	30; 80; 250 barg
Temperature	< 80 °C

Table 2: Test parameters for seal performance tests

During the test the temperature of the media as well as the temperature of the seal seats was measured. After the test the mechanical seals have been inspected. The inspection was an overall inspection of the mechanical seal as well as measurement of the seal face flatness and roughness. The seal face and seat inspection confirm a good running behavior under acceptable leakage rates over the whole test period. Figure 14 shows the inspection of the seal face and seat after the test run.

Figure 14: Seal face (left) and seat (right) inspection after the performance test

After successful completion of the mechanical seal performance tests, the mechanical seals have been additionally tested on a dedicated high temperature test loop. Two full size mechanical seals and cooling jacket has been used for the qualification tests.

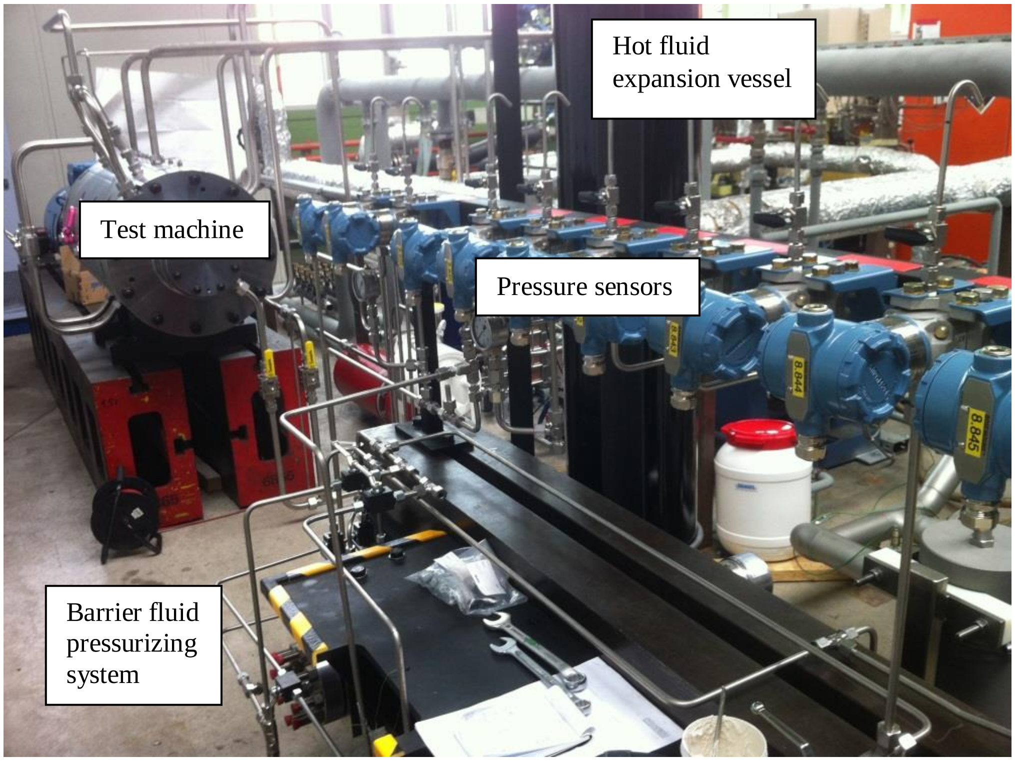

Figure 15: High temperature mechanical seal test rig

The test program that the mechanical seals have been undergoing is summarized in the below table:

Test parameter	**Value**
Product medium	Air and high temperature heat transfer fluid
Barrier medium	Water/Glythermin NF 30/70 vol.%
Speed	1500; 3000; 6000 rpm
Barrier fluid pressure	15; 30; 80 barg
Barrier fluid temperature	-2°C to +70°C
Product fluid temperature	20°C to 177°C
Starts/Stops	250 times from 0 to 1500 rpm

Table 3: Test parameter overview for mechanical seal and cooling jacket performance tests

The test loop is instrumented with different sensors at various locations as summarized in the below table:

Measured value	**Description**
Pressure (9 sensor locations in total)	Barrier fluid pressure (various locations), product fluid pressure (various locations)
Temperature (20 sensor locations in total)	Barrier fluid supply & return temperature, product fluid temperatures, cooling jacket hot spot temperatures, seal face temperature
Flow (4 sensors)	Cooling flow rates to mechanical seals and cooling jackets
Speed	From VFD
Test machine torque	From VFD
Accumulated leakage	Mechanical seal leakage measurement by a scale
Casing acceleration	Accelerometers at test machine casing NDE
Shaft displacement	Shaft position measurement at test machine DE

Table 4: Instrumentation used for hot mechanical seal tests

In order to validate the mechanical seal and the cooling jacket performance, temperatures are measured at multiple locations within the test machine and at various locations of the test loop. The below cross section of the test machine illustrates the test concept with 2 mechanical seals in a back-to-back arrangement (for axial thrust compensation) as well as the different locations of temperature measurements (highlighted with a green X).

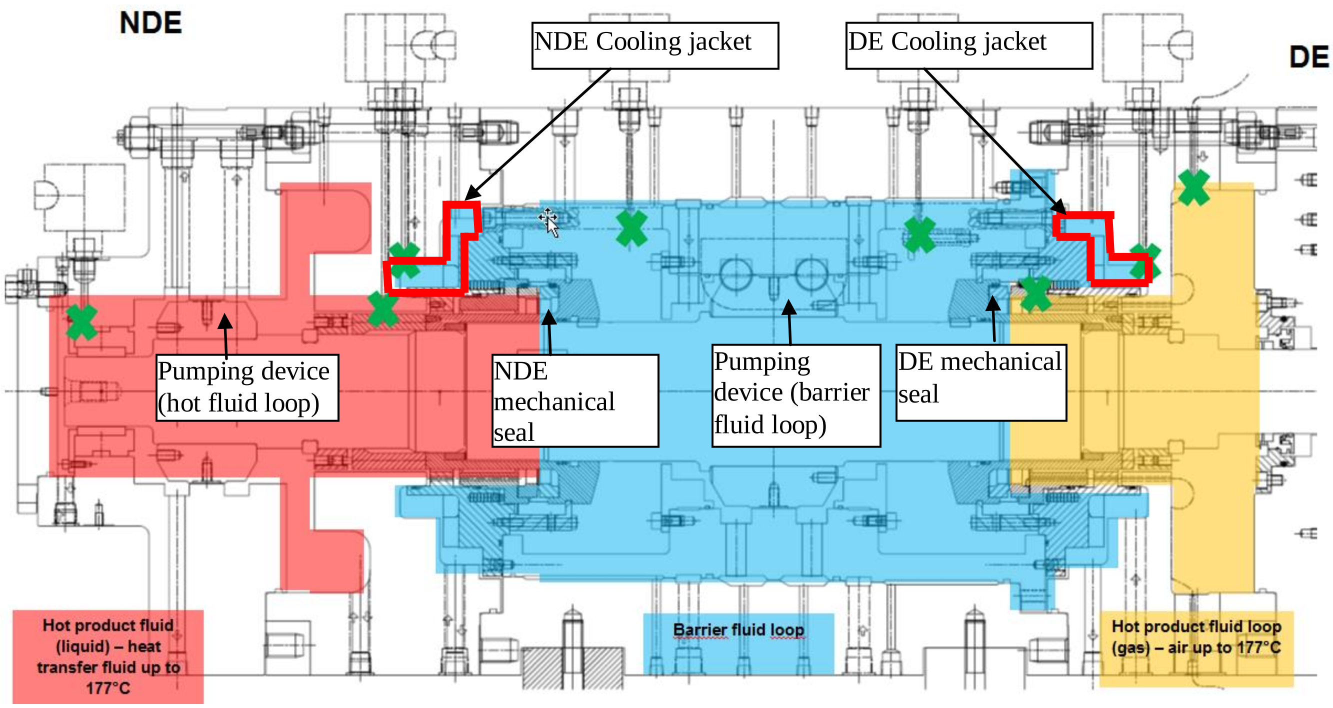

Figure 16: Cross section of test machine used for hot mechanical seal tests. Temperature sensors inside the test machine are highlighted with a green "X".

The obtained test results are used for comparison with the predicted (calculated) values and to validate the mechanical seal and the cooling jacket for high temperature operation.

The below table summarizes a few test points and shows the comparison to the predicted (calculated) values:

Parameter	**Unit**		**Value**		
Test point	-		1	2	4
Speed	[rpm]		1500	3000	6000
Differential pressure DE mechanical seal	[bar]		21	21	19
Differential pressure NDE mechanical seal	[bar]		18	17	15
Barrier fluid inlet temperature	[°C]		22	15	45
DE product fluid (gas) temperature	[°C]		102	177	178
NDE product fluid (liquid) temperature	[°C]		103	175	168
Standardized Leakage DE mechanical seal (normalized at Δp 30 bar, 6000 rpm, gas inside)	--	Measured	0.47	1.13	1.43
		Predicted	0.60	1.03	0.88
Power losses per mechanical seal	kW	Measured	2.2	9	12.6
		Predicted	2.5	6	17
DE seal face temperature	°C	Measured	35	37	73.6
		Predicted (FEA)	25 1)	40 2)	65

DE cooling jacket hot spot temperature	°C	Measured	42	22.5	50.2
		Predicted (CFD)	n.a.	n.a.	80
NDE cooling jacket hot spot temperature	°C	Measured	80	66.5	85.8
		Predicted (CFD)	n.a.	n.a.	~85 3)
Wetted shaft temperature NDE	°C	Measured	80	100	130.1
		Predicted (CFD)	n.a.	n.a.	~170 3)

1) FEA done for barrier fluid inlet temperature 15°C
2) FEA done for barrier fluid inlet temperature 30°C
3) CFD done for product fluid temperature 177°C

Table 5: Overview of test points

The overall test program consists of 34 testing points with an overall test duration of approximately 250 hours, including 50 hours endurance test at 6000 rpm and maximum product fluid temperature of 177°C, as well as 500 starts/stops.

6. CHALLENGES, ACHIEVEMENTS AND CONCLUSION

As a result of the seal tests the parameters applied in the seal performance calculation were adjusted to achieve an optimum correlation of test results and measured performance data. One of the main challenges was the adjustment of the heat transfer parameters in order to achieve an acceptable correlation for both the operation with 100% process gas and 100% liquid. Heat transfer from the seal faces to the fluid influences the deformation of the face and the operating taper and hence affects the hydrostatic and hydrodynamic lift off of the faces.

A second important result of the comparison of the test data with the calculated results was the consideration of centrifugal effects. Centrifugal effects occurring due to the specific weight of the accelerated sealing fluids have a significant effect on hydraulic forces.

The new re-engineered mechanical seal for high pressure and high temperature applications performed as expected and all targets for the development have been achieved. Also the inspection of the seal faces after test run showed that the seal faces have been in very good condition without any negative influences of the new design.
These results give the confidence to install the mechanical seal in field operations.

NOMENCLATURE

HP/HT = High Pressure/High Temperature
JIP = Joint Industry Project
R&D = Research & Development
DE = Drive End
NDE = Non Drive End
VFD = Variable Frequency Drive
CFD = Computational Fluid Dynamics
FEA = Finite Element Analysis

ACKNOWLEDGEMENTS

The authors thank Andrei Strikovski of FMC Technologies and Christian Eisfeld of EagleBurgmann for their continuous support during this seal development project.