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DEVELOPMENT OF A SINGLE MECHANICAL SEAL EQUIPPED WITH API PIPING PLAN 11/66A FOR LARGE MAINLINE CRUDE OIL PIPELINE PUMPS.

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ABSTRACT

Mechanical seals are an integral component of centrifugal pumps and are a critical component especially on a large oil pipeline pump with energy density in excess of 5.4 x 10E6 HP/Min. Therefore, TransCanada Pipelines undertook an initiative with the broad objectives of designing an application specific seal that has higher reliability, and superior capability to handle pipeline-operating conditions. With these objectives, a first of its kind API 682 4th edition plan 66A single mechanical seal with dual bushings was deployed for a large mainline pipeline pump with a shaft size of 6.130” in 2015. In developing these mechanical seals, an underlying philosophy that was closely adhered to was to test and validate the seal design in real pipeline like conditions.

This paper discusses the critical design aspects, laboratory verification testing and field validation experience of a single mechanical seal with an integrated containment system in the spirit of API piping plan 66A. Of particular importance is the shaft diameter - pressure relationship of the pump in this development project 155 mm x 100 bar (6.130” x 1450 psi). Besides the typical qualification testing of such engineered seals, application specific testing of the mechanical seal and API piping plan 66A as mandated by the pipeline operator was carried out. The tests resulted in a seal that could withstand pipeline transients, and frequent start stops better in addition to a custom-engineered bushing to contain oil pressures up to 100 bar.

INTRODUCTION

Crude oil pipeline applications can be very different from the typical 'process' pump in a hydrocarbon processing plant. There are a number of peculiarities regarding pipeline pumps related to the size of the equipment, process medium and plant operation that make these applications a challenge.

Pipeline operation

A pipeline will have two or more pumping units in a pump station in series over the length of the pipeline. Each station may be equal in terms of design and operation but with different normal operating conditions to suit geographical location, terrain between upstream and downstream stations, etc. Alternatively, the operating philosophy may vary from station to station; this is usually a function of some stations being base loaded with additional stations designed to run only when production is high and may result in some pumping in parallel, others in series, some with variable speed and others with fixed conditions. Consequently, duty conditions will change from station to station; operation at variable conditions; pumps running at maximum and/or minimum duty points are likely to be common operating modes. Which implies that a vast majority of the large pipeline pumps end running over a range of conditions and just not at the BEP. Last, depending on the hydraulic pipeline conditions, pipeline pumps are subject to various transient loads because of upset conditions and changing process variables.

Process fluid composition and properties

Generally, a pipeline handles export quality crude oil with very little water, H2S and salt content: so no problem for a single seal and it is not corrosive to '316' series stainless steels. Pipelines are welded together during construction; occasionally weld spatter may be present during commissioning and may appear during initial operations. Other debris such as the products of corrosion, any settlement could also be present. A typical crude will look like specific gravity of 0.85 to 0.94, viscosity of 5cst to 350cst and vapor pressure of 0,69 bara.

Pump location & Utilities

Pumping stations are often remote and lightly manned, possibly unmanned. This requires that the equipment be very reliable and that it be possible to condition monitor the equipment remotely. Pumps may be located outdoors or indoors. With outdoor locations, the effects of ambient temperature on the process fluid at standstill conditions should be considered. Pumping stations are often in remote locations with little other associated plant hence the availability of utilities will be restricted. Typically cooling water, steam and nitrogen are not available as plant utilities.

OPERATING CONDITIONS & SPECIAL REQUIREMENTS

Pumped fluid:	Crude oil + Diluted Bitumen
Seal chamber pressure range:	2.5 to 100 bar
Discharge pressure:	137 bar
Shaft diameter:	155 mm
RPM range:	720 to 1980
Temperature range:	10 to 70°C
Density:	0,87 to 0,93 kg/dm³
Vapor Pressure:	0,93bara
Viscosity:	71Cst

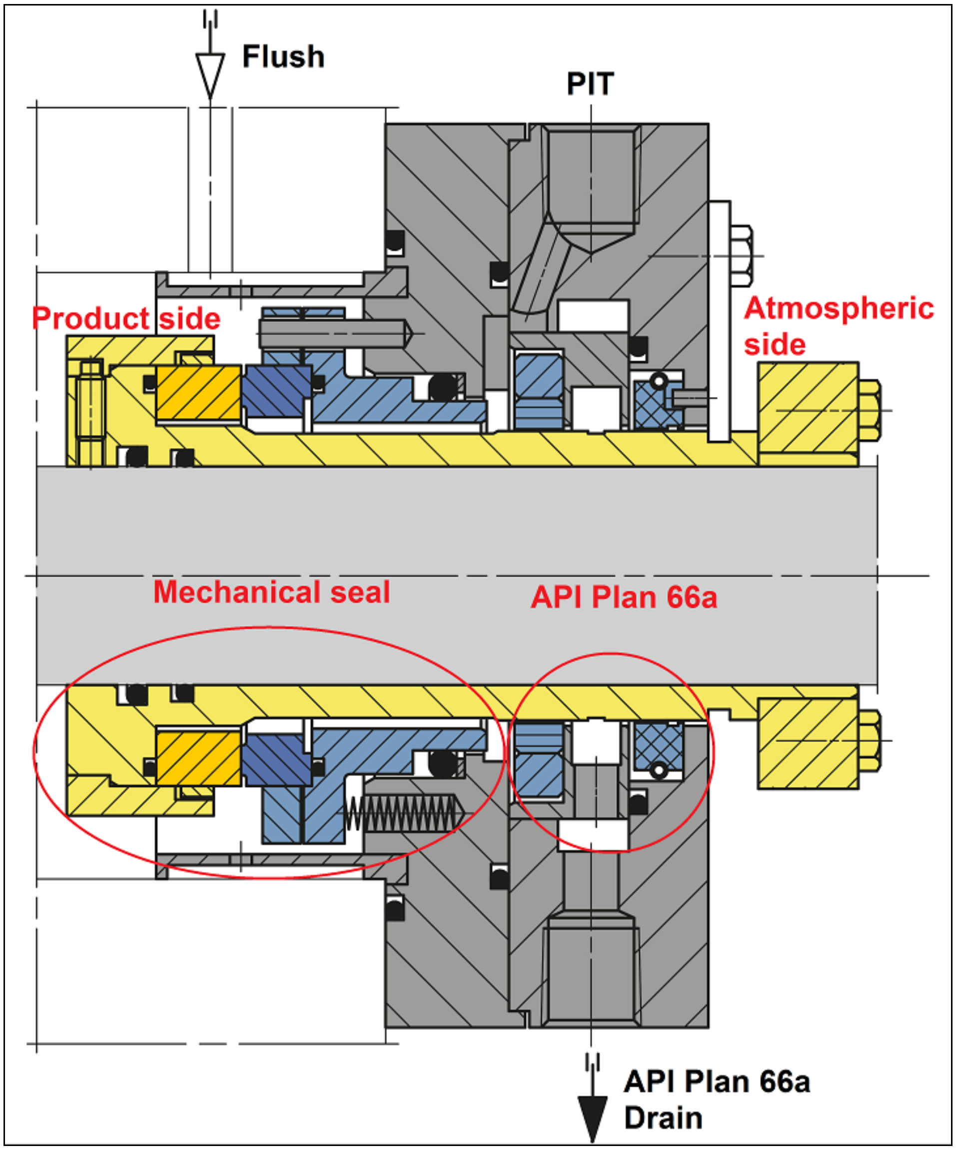
The wide operating range combined with a large shaft diameter implies the need for a relatively wide leakage rate allowance for a single mechanical seal to achieve long life. The containment system that collects and disposes the leakage must be designed to deal with these higher than typical leakage rates. However, in the improvement project the special requirement was that under no circumstances can a spray of oil into the bearing bracket or to the environment be tolerated. This requirement counts even in case of events such as seal installation errors, pump bearing malfunction or other events that could cause a complete loss of the mechanical seal sliding faces. This stringent requirement would normally direct the selection of the seal arrangement towards a dual seal, either pressurized or unpressurized. Yet, a single seal was the only possible option since utilities, which are typically required for dual seals are not available. In addition, modifications to the pump seal chamber was not an option, hence a dual seal option was not considered.

MECHANICAL SEAL AND PIPING PLAN DESIGN APPROACH

The initial evaluation of the application lead to two critical design features in a single seal arrangement that would be imperative to achieve the special requirements of the pipeline operator:

1. Make sure that the mechanical seal faces cannot break under any circumstance. Since seal, faces for such services typically fabricated in brittle materials such as silicon carbide, severe damages can occur when two hard materials are in hard contact. The worst-case scenario is complete loss of the sealing interface due to the faces breaking apart. Besides breaking one or both seal faces, complete loss of the sealing function can occur when faces separate because of hang up of the spring loaded seal face.
2. Ensure that the containment system behind the seal faces can handle a release of process fluid at 100 bar without any potential for collapse or failure of the components in the seal cartridge that will control the high pressure in the cavity between the seal faces and the environment. Such a containment capability shall last for a period of up to 15 min until the pumping unit can be safely depressurized.

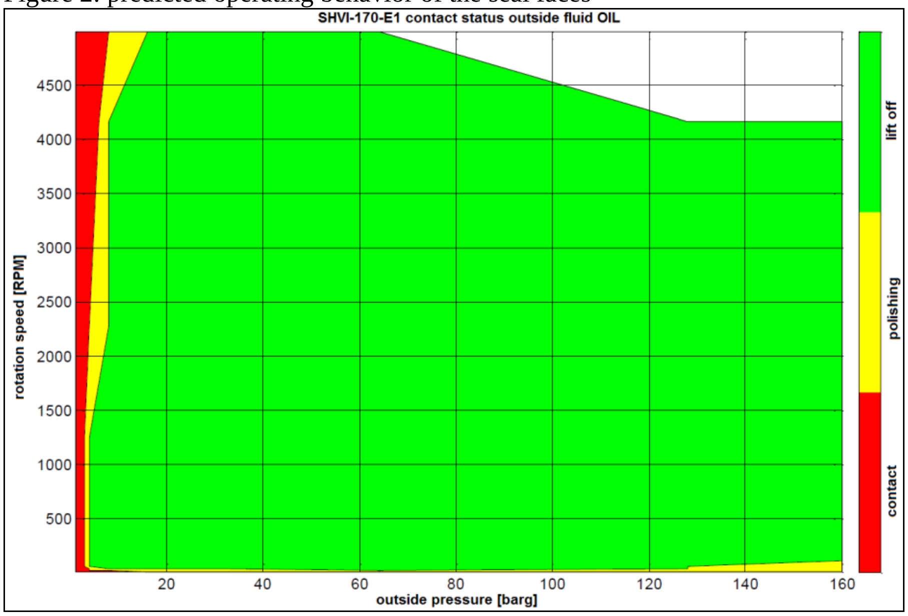
Fig. 1: Seal design that was field validated and met the test objectives of this project.



The seal face materials that were selected for this project are reaction-bonded silicon carbide for the rotating face (dark yellow) and a compound material of silicon carbide and carbon for the stationary face (dark blue). The latter material provides enhanced emergency characteristics if lubrication issues arise and is often used for high pressures and high speeds. Although the compound materials' strength properties reduced as compared to pure silicon carbide, the shrunk on steel ring (light blue) compensates for this compromise. Thermal conductivity and coefficient of thermal expansion are similar to silicon carbide.

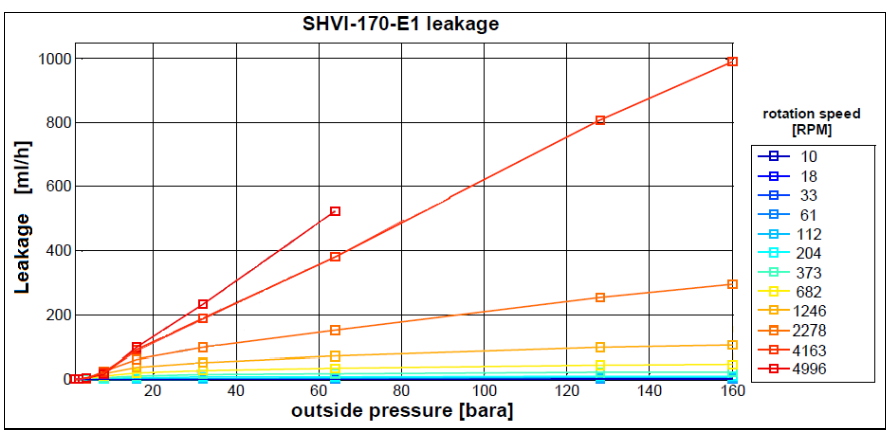
The seal design is of the stationary spring style (blue) of which the seal faces are shrunk in stainless steel rings or bandages. This design approach provides several benefits as compared to homogenous face designs. Torque is transmitted from the interface via the shrink fit to the seal face carriers (light blue) and anti-rotation pins in the seal gland (grey). This means that the brittle seal faces have no holes or slots where fractures typically initiate when lubrication of the interface is interrupted due to unforeseen events in the pump or pipeline. Secondly, the bandages provide additional stiffness and robustness to the seal faces which reduces face distortions. Maintaining a stable sealing gap geometry in all possible operating modes is the key to consistent leakage and friction behavior of the seal faces. The plot in Fig. 2 illustrates the predicted lubrication behavior of the seal faces as a function of operating speed and pressure. The classification criteria for the different zones is based on theoretical and experimental studies. The seal faces lift off in the green zone that means no contact hence no wear. In the yellow zone, a small contact force could occur and some polishing happens whereas in the red zone contact will definitely occur which will result in wear of the seal faces with an increased likelihood for face damages. By knowing all the possible operating conditions, i.e. normal as well as unusual, it is possible to avoid operation of the seal in conditions that would make the seal faces prone to damages and wear. Because of the wide operating span in this particular application, special face lubrication grooves were incorporated at the compromise of higher leakage rates.

Figure 2: predicted operating behavior of the seal faces



The diagram in Fig. 3 shows the calculated expected leakage rates of the mechanical seal relative to seal chamber pressure for various shaft speeds. In general, the seal is optimized with a certain leakage rate for safe operation in all possible operating modes. In other words: at 20 drops per milliliter, one would always expect to see oil drops going into the collection system via the drain connection of the plan 66A.

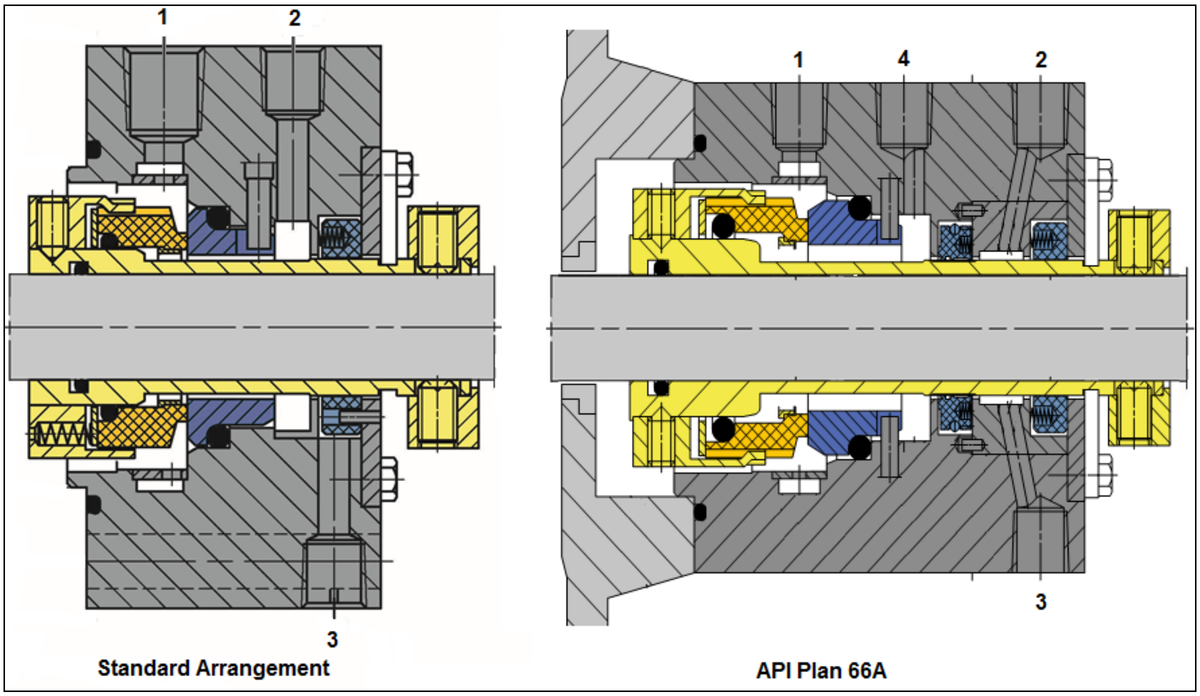
Figure 3: Mechanical seal predicted leakage rates



The multiple springs in the seal and the area of the dynamic O-ring are continuously flushed with a multi-port injection via the plan 11 to minimize the possibility of hang up due to accumulation of solids.

API plan 66A, introduced in the 4th edition of API 682 was selected to avoid oil release into the pump bearing bracket and environment. API plan 66A, see Fig. 4 principally provides for two throttle bushings on the atmospheric side in the seal housing. A drain port connection 3 is located between the two bushings to carry away the leakage to a collection system. Excessive leakage from the seal faces is indicated when the pressure rises in the space between the seal faces and the first throttle bushing. The pressure measurement with a PIT (Pressure Indicator Transmitter) in connection 4 relates to the leakage value of the mechanical seal and informs the operator of potential troubles. Normal leakage of the seal will pass through the inner bushing with very little or no pressure build up.

Figure 4: Concept of API plan 66A



A catastrophic failure of a mechanical seal occurs when there is a total loss of the mechanical seal faces, i.e. fractures of one or both seal faces. The standard arrangement bushing is designed to contain normal leakage under atmospheric pressure but not the full product pressure. In this project, pressures up to 100 bar must be contained. Therefore, the inner bushing of plan 66A had to be optimized to withstand the high pressure, limit the oil flow towards the drain connection that is connected to a central collection sump in the pumping station. On the atmospheric side, a special segmented carbon ring avoids any release of oil into the bearing bracket and atmosphere.

MECHANICAL SEAL QUALIFICATION TEST

The mechanical seal and API plan 66A were qualified by doing separate performance tests on each of the two seal components. The mechanical seal was verified on a horizontal high-speed-pressure test rig (fig. 5 & 6). The API plan 66A qualification test was completed on a different test rig suitable to handle high thrust loads (fig. 8 & 9).

The purpose of the mechanical seal qualification test is to verify the theoretical performance calculation with respect to leakage and friction under conditions, set as close as possible to the field operating conditions as shown in tables 1 & 2.

Test parameter	Value
Test fluid	Oil ISO VG 68
RPM	800 - 2000
Pressure	10; 50; 100bar; and short time 150bar
Temperature	Approx. 60 °C

Table 1: Test parameters for the mechanical seal qualification test

Figure 5: Test rig for the mechanical seal qualification test

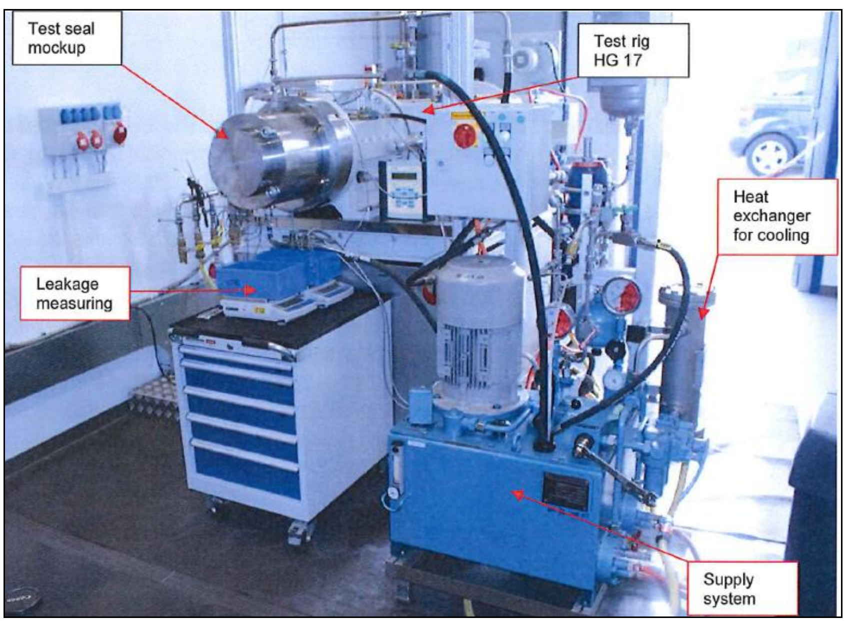


Figure 6: Cross-section of original seal design in the test rig setup.

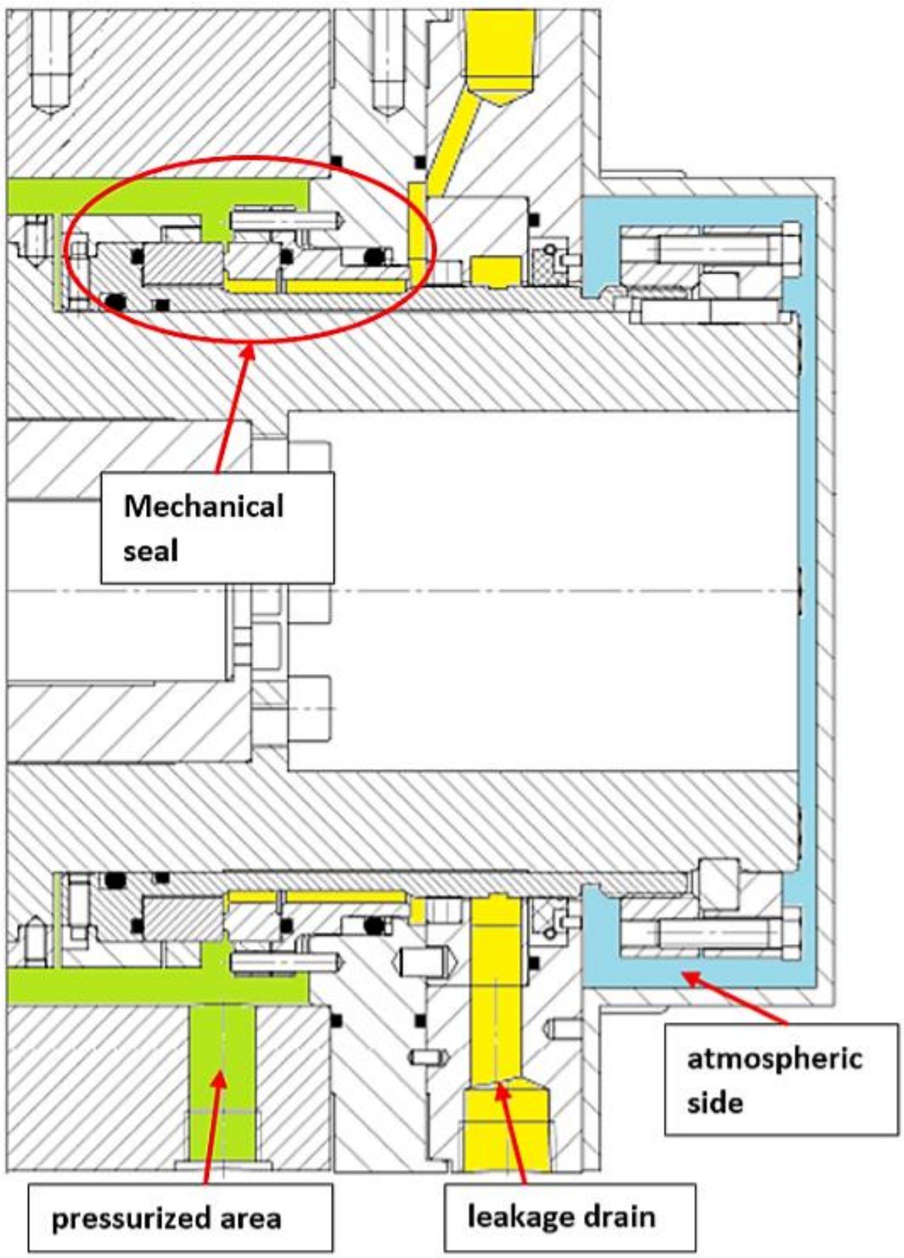


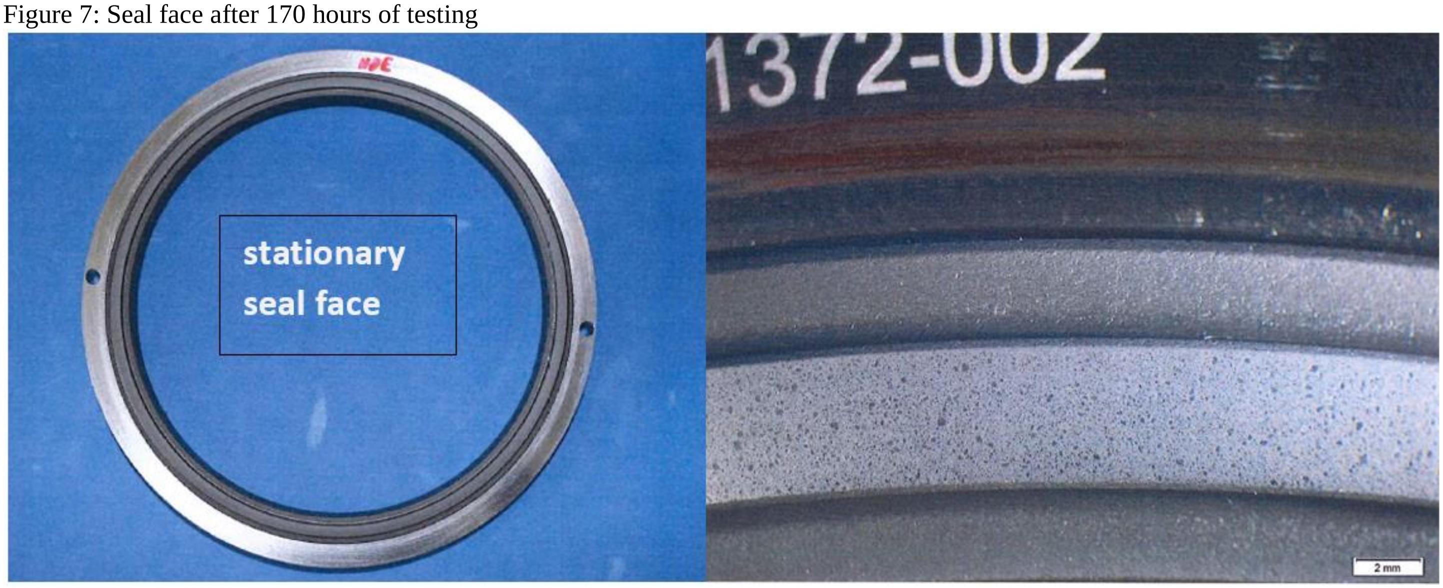
Table 2 shows the speed and transients conditions that the seal was exposed to during 170 hours. The basis for the program was the API 682 qualification test protocol but adjusted with the specific conditions of the pipeline application.

Test No.	Speed [rpm]	Pressure [bar]	Runtime [hour]
1	0 (Static)	100	0,5
2	800	100	0,5
3	800	100 -> 150	10 sec
4	800	100	0,5
5	800	100 -> 150	10 sec
6	800	100	0,25
7	2000	100	0,25
8	2000	100 -> 150	10 sec
9	2000	100	0,5
10	2000	100 -> 150	10 sec
11	800	10	6
12	2000	10	6
13	800	50	6
14	2000	50	18
15	800	100	24
16	2000	100	40

Table 2: Requested Test Cycles

The test results showed an operating behavior as predicted by the analytical calculations in all operating states. There was no contact or mixed lubrication detected during the mechanical seal test. Fig. 7 shows the stationary seal face after 170 hours runtime. Only small traces of contact coming from startups and shutdowns were observed during the inspection of the seal faces upon completion of the test program. The test results demonstrate the value of analytical simulation, because the calculation predicted a seal face lift off at any operating condition.

Figure 7: Seal face after 170 hours of testing



API PLAN 66A QUALIFICATION TEST WITH CATASTROPIC FAILURE SIMULATION

The second test was requested to verify the leakage containment capability of the API plan 66A and to verify the performance of the high-pressure bushing. This is the first bushing that is exposed to oil at 100 bar. It was done in two different test runs with low pressures and high pressure (Table 3). The first test run of the API plan 66A was to determine the leakage rate/pressure relationship for the pressure indicator transmitter of plan 66A. The PIT will monitor excessive leakage of the mechanical seal by measuring the pressure in the cavity between the seal faces and first bushing. The second test was set up to simulate a catastrophic failure of the mechanical seal and to assure that the containment system and the high-pressure bushing can withstand the full pump/pipeline pressure.

Test parameter	Value
1st Test run	
Product medium	Oil ISO VG 68
RPM	1000
Pressure	0,1; 0,2; 0,3; 0,4; 0,5; 1,0 bar
Temperature	Approx. 60 °C
2nd Test run	
Product medium	Oil ISO VG 68
RPM	1000 for 10 min
Pressure	100 bar
Temperature	Approx. 60 °C

Table 3: Test parameters for API plan 66A

The test set up of the plan 66A consists of the fixtures as shown in Fig. 9 in which oil is pressurized and circulated by a hydraulic supply system as shown in Fig. 8. A camera was installed in the upper bearing housing to visualize the leakage if any, coming out of the upper, low-pressure bushing.

Figure 8: Test rig for API plan 66A

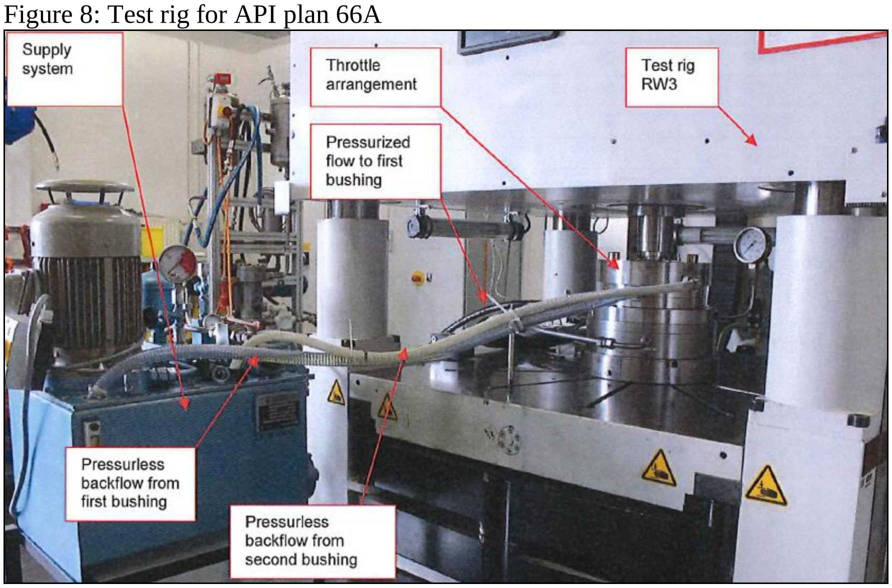
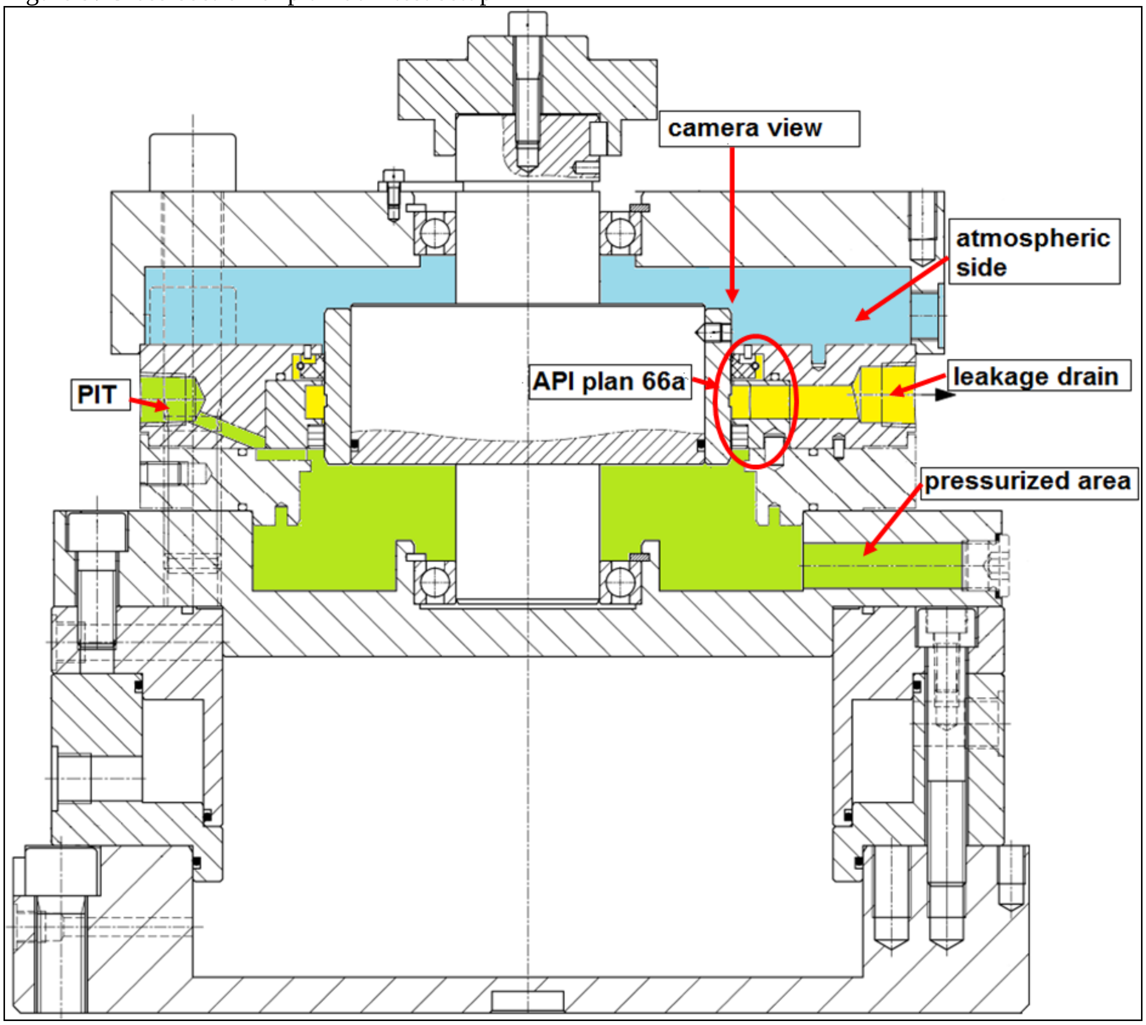


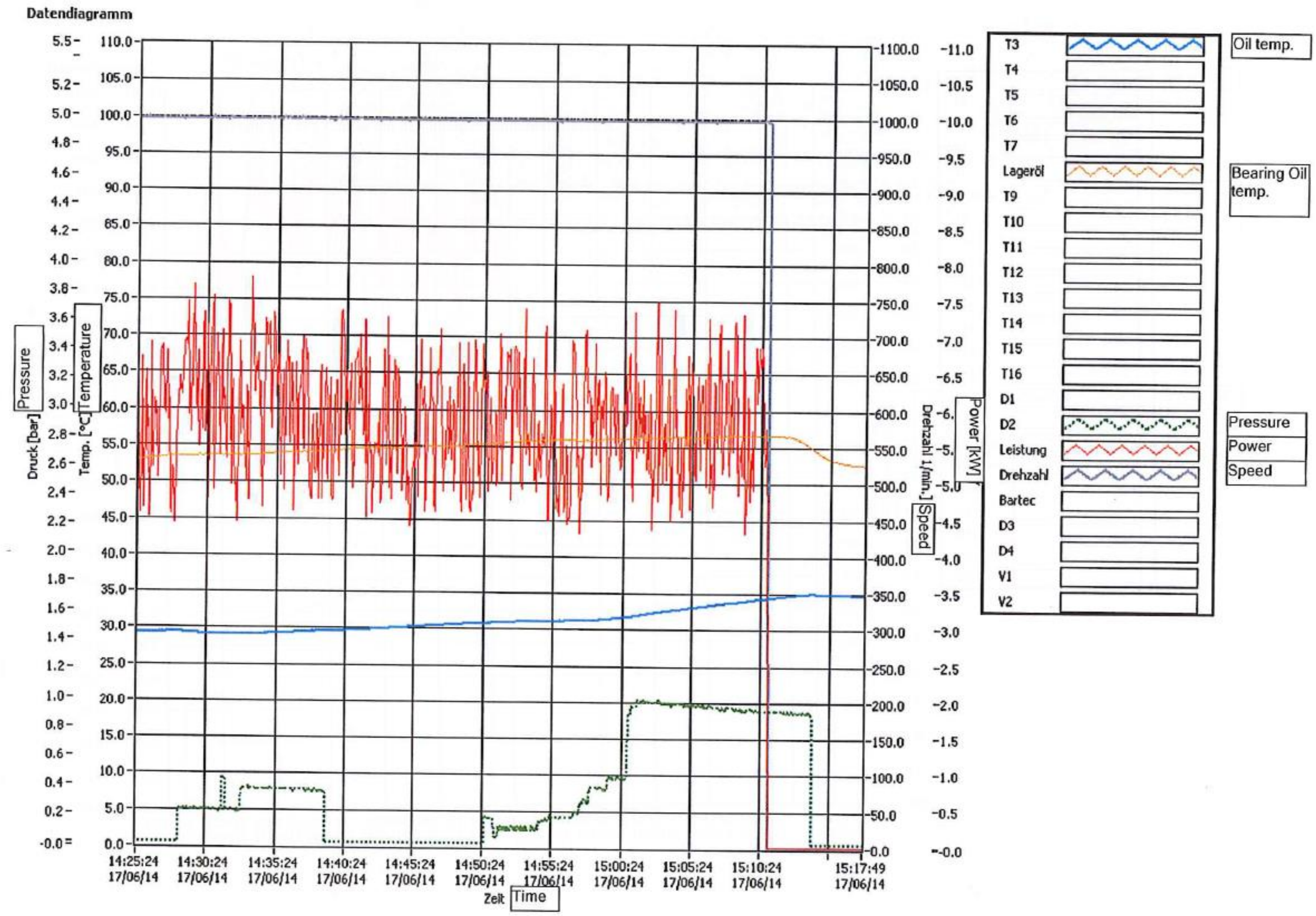
Figure 9: Cross-section of plan 66A test setup



The diagram in Fig. 10 shows the following operating parameters: power consumption of the complet test rig in red, RPM in grey, oil pressure in green, bearing oil temperature in orange, and the oil pressure in blue.

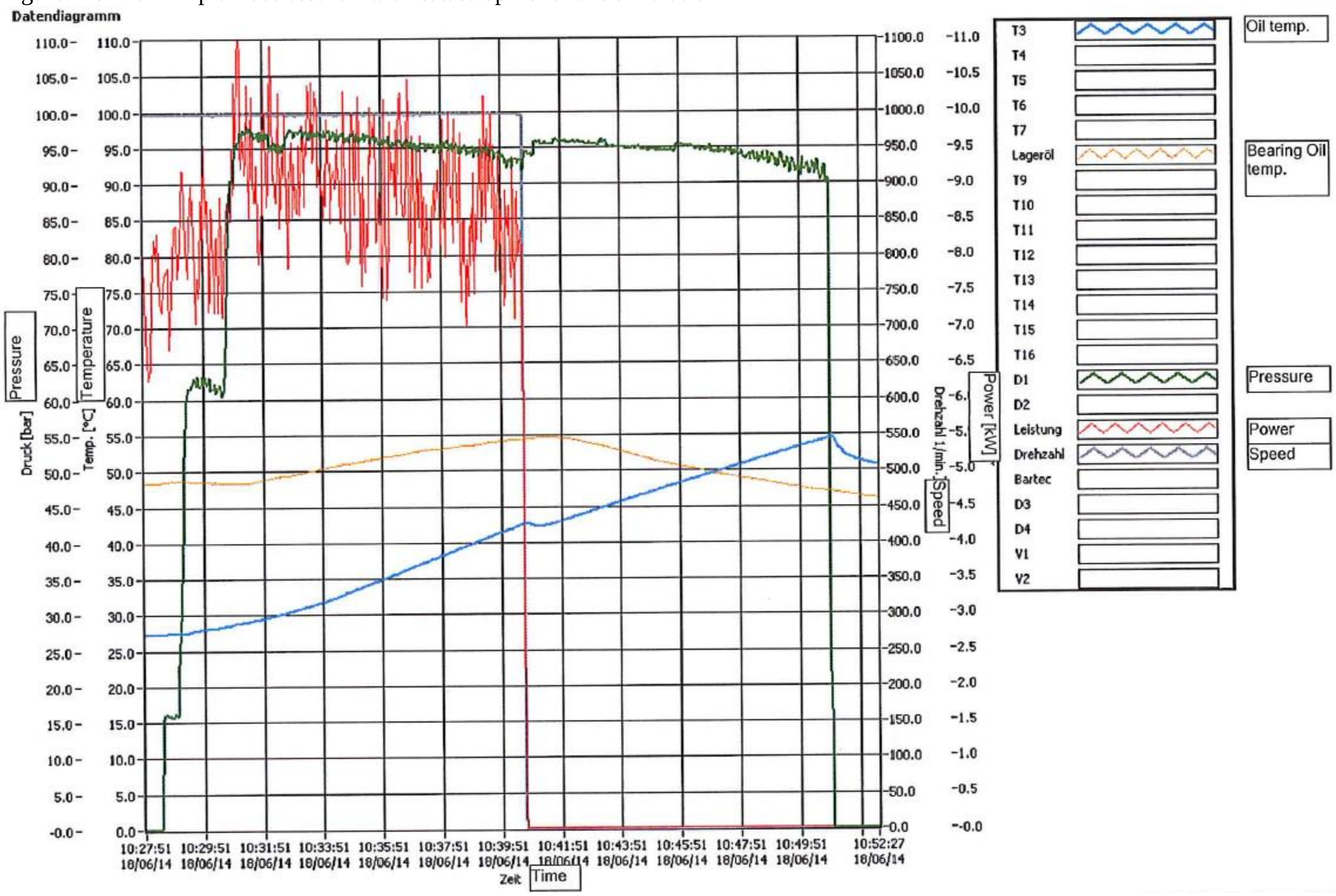
The alarm set points for the PIT can be determined on the basis of the flow/pressure relationship relative to the predicted maximum leakage rate of the seal faces in all operating modes of the pumps. The corresponding flow rate across the bushing into the drainage port is the actual leakage rate during normal field operation. By using the PIT measurement, the operator can remotely identify which of the two seals on a BB pump is in trouble and measure the magnitude of the leakage.

Figure 10: API plan 66A test run to determine the leakage rate/pressure relationship



The 2nd test run simulates a catastrophic failure of the mechanical seal which would result in a release of oil at 100 bar for 10 minutes while the shaft is spinning. After the shaft was stopped, full pressure was still applied for another 10 minutes. The complete test cycle is shown in the diagram of Fig. 11. Oil leakage in the atmospheric cavity of the test rig is monitored with a camera and at no time was there any evidence of oil coming across the low pressure bushing. This test result indicated that a short bushing can be used to reduce the full pipeline pressure safely and reliably for at least 10 to 20 minutes, static as well as dynamic. However, in 2015 field experience indicated that a floating bushing is preferred above a fixed design for reasons of alignment in the pump and seal.

Figure 11: 2nd API plan 66a test run with catastrophic failure simulation



FIELD EXPERIENCE

Following the successful laboratory testing field trials were undertaken to further assess the proposed API plan 66A seal design. The seals were installed into the drive end and non drive end side of the operator’s mainline pump and monitored throughout its operation. After approximately 3000 hours a thrust bearing event initiated an investigation into the machine. This subsequently indentified areas for improvement within the mechanical seal resulting in the implementation of the floating bushing . The floating bushing design ensured that radial movements of the pump shaft due to operational and dimensional tolerances are accounted for and maintain its containment function. The mechanical seal with the new high pressure floating bushing design was also subjected to the same kind of qualification testing as described in the earlier sections. After validating the results by TransCanada Pipelines and EagleBurgman, the enhanced seals were placed in service on a different pump station within TransCanada’s Oil Pipeline and has since been accumulating operating hours without any known issues.

CONCLUSIONS

The results of the test program and acquired field experience validates that it is possible to mitigate the potential for any oil sprays from large crude oil pumps under full pipeline pressure with a single mechanical seal equipped with API plan 66A. Even under severe upset conditions that can lead to loss of the seal faces altogether, safe containment of the pumped media can be guaranteed. Secondly, API plan 66A allows for early detection of seal troubles based upon the predicted leakage rates through analytical methods which provides for greater protection and reliability for the user. The testing of the mechancial seal and API plan 66A went over and beyond a standard qualification test. The goal was qualifying the mechancial seal by simulating real pipeline conditions. It is one of the largest mechanical seals produced with an API plan 66A for pressures up to 100 bar.

NOMENCLATURE

API	= American Petroleum Institute
BEP	= Best Efficiency Point
PIT	= Pressure Indicator Transmitter
BB Pump	= API Pump Designation for Between Bearing Pumps.

REFERENCES

G. Lederer, 2014, R&D Report 1753,
T. Holzapfel 2015, Performance test 5000003428

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