UPSTREAM PUMPING TECHNOLOGY IN CENTRIFUGAL PUMP MECHANICAL SEALING APPLICATIONS—FIELD EXPERIENCE WITH HIGH DUTY SEA WATER INJECTION PUMPS

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ABSTRACT

Among the technologies developed for oil and gas production centrifugal pump sealing applications, active lift technology is one of the most promising. The reliability of any centrifugal pump in a critical application is strongly influenced by the performance of the mechanical seals. It is widely accepted based on extensive operating experience that mechanical seals are the most vulnerable components in the pump structure and their life is not easy to predict. Depending on the product being sealed and operating conditions, seals can last from a few weeks to 20 years.

Active lift, also known as "upstream pumping," seals represent a new approach to the liquid sealing technology with the potential to offer a step change in seal reliability and resulting production gains. Developed from dry gas seal technology its major advantage over conventional contacting mechanical seals is that the face separation is stabilized and there is no rubbing between the seal faces. As rubbing is completely eliminated face deterioration is eliminated and face life is significantly extended together with a considerable reduction in the heat generated by the seal faces. Another important advantage of this new concept compared to traditional pressurized double seals is the elimination of a sophisticated pressurized barrier fluid system in favor of a much more simple system.

It is obvious these days that "upstream pumping" sealing technology for higher pressure pumping applications is moving from small laboratory scale trials to wide industrial application. This paper takes the form of a general review of the product development and field experience accumulated during 18 months of operation with upstream pumping seals in a critical sea water injection pump application.

INTRODUCTION

Since their first introduction in the late 1930s mechanical seals have been used extensively in a wide range of pumps, mixers, compressors and other similar machines handling various fluids. In centrifugal pump sealing applications sealing is usually obtained via intimate contact between the opposing faces of the two annular rings. Usually one ring is fixed to the pump casing and is held stationary, while the other one rotates with the shaft.

The rotating face is held close to the stationary face via spring and hydrostatic pressure. The spring load acts as a closing force and is more significant in applications where the sealed pressure is low or zero. Seals are designed such that, during stable operation, a balance is struck between leakage and wear (Mayer, 1997).

There are a relatively large variety of mechanical seals of different types and classification is usually made first by design characteristics and then by positional arrangement. Wedge or O-ring, balanced or unbalanced; these are design characteristics. Another important aspect in classifying mechanical seals is by orientation such as rotating or stationary mounted, single, double or tandem. These parameters address positional arrangements regardless of their design (Netzel and Volden, 1995). In this paper the authors will focus on demanding double seal arrangements for critical petrochemical applications where even small amounts of leakage are not tolerated. Progress in sealing systems has resulted in the development of four primary sealing concepts based on the type of lubrication used:

- *Contacting liquid-lubricated*—seal faces are cooled and lubricated by the process fluid or barrier liquid being sealed. This system is a condition of mixed lubrication where the load at the faces is partly carried by the fluid film and partly carried by mechanical contact. This arrangement is still dominant in the industry for all types of rotating equipment.

- *Contacting gas-lubricated*—seal faces are designed to run dry with light contact. Cooling and lubrication are achieved from the process or buffer fluid being sealed. This is a condition of boundary lubrication where the seal faces are in contact, though separated from hard contact by material transfer films.

- *Noncontacting liquid-lubricated*—In this case, a geometry change is made to the seal face. Hydrodynamic lift is generated by spiral grooves or similar features incorporated into one of the seal faces. The hydrodynamic film separates the seal faces. This noncontacting seal concept is usually applied to specialized pumping applications to eliminate hazardous and toxic leakage or abrasive wear of the seal faces.

- *Noncontacting gas-lubricated*—This type of design is again based on the concept of hydrodynamic lubrication and incorporates geometry changes to the seal faces such as spiral grooves. It is very similar to the noncontacting liquid-lubricated concept with the difference that lubrication is provided by gas. The only heat that is developed is that of shearing gas at the seal faces. Therefore, it is the most energy efficient sealing system available to industry. This type of sealing system was initially developed for compressors, but is now being applied to difficult pumping applications to control emissions or maintain process fluid purity.

The evolving tribological demands of mechanical face sealing applications such as those associated with high-duty fluid seals and gas sealing technology are driving the need for the development of new seal design and novel seal face material solutions. Many seals used in the past had carbon faces made of various carbon graphite compositions (Azibert and Chesterton, 1996). Under more severe conditions where the combination of high pressure and high surface speeds create high levels of pressure and velocity (PV) the seal designer will usually specify two hard faces, typically a combination of reaction bonded silicon carbide, alpha sintered silicon carbide or tungsten carbide (Jones, 2004; Enqvist, et al., 2000). Tungsten carbide is believed to be the best choice in applications that undergo considerable amounts of shock and vibration. Silicon carbide works well with light hydrocarbons that have a tendency to flash or turn into gas at the seal interface depending on pressure and temperature.

The performance of any mechanical seal is largely determined by the design of the sealing interface and the interface materials used. One of the major issues affecting seal performance and its service life is to find a balance between seal leakage rate and wear on the seal faces. The ideal situation would of course be one where the seal benefits from full fluid lubrication and exhibits low leakage rates. Extensive research in this direction resulted in creating a "laser face technology" (Wallace and Meck, 2003). The laser face seal uses two sets of recesses in one of the sealing faces. One set is used to introduce the fluid between stationary and rotating seal faces and the other to pump back fluid from the seal interface into the sealed space. It has already been proven that laser face seals can provide full face lubrication and low friction with low leakage rates.

Recently some research activities were focused on the study of laser texturing influence on the performance of sintered SiC stationary seal rings. Using laser texturing enables better control over producing spherical pores of selected diameter, depth and ratio. Compared with conventional SiC seals the laser textured seals demonstrated lower friction coefficient and friction torque. During fluid film lubrication each pore is acting like a microhydrodynamic bearing and due to hydrodynamic pressure build up over the pore and adjacent area, the mating rings separate under the action of this hydrodynamic pressure (Chen and Hsu, 2003).

The conventional approach to difficult service conditions involving toxic, nonlubricating or abrasive liquids is to use a double seal arrangement. Usually two seals mounted back to back operate in barrier liquid pressurized externally. In order to ensure that primary liquid between the seal faces is clean, barrier liquid is circulated externally and pressurized to maintain it higher than the process pressure. During steady process conditions the inner seal of the double seal prevents the barrier fluid leaking into the process side. The outer seal prevents the barrier fluid leaking to atmosphere. Since the barrier fluid for a double seal is maintained at a pressure above the process fluid pressure, it is normally assumed that leakage occurs only from the barrier into the process. Unfortunately in practice leakage still can occur in the opposite direction. Pressure reversal may take place due to process liquid pressure surging, poor maintenance of the system, accidental loss of pressure source or loss of barrier fluid. One of other main disadvantages of double seals is that the barrier fluid pressure must be maintained higher than process pressure, which requires that the outer seal be rated for a higher pressure differential. Additional auxiliary components for providing reliable functioning of the support system make these arrangements expensive. Also because of cross migration the barrier fluid can get contaminated with process fluid and controlled monitoring of the barrier fluid should be established via a periodic maintenance program.

The reliability of any mechanical seal strongly depends on the presence and stability of the load carrying hydrodynamic lubricant film. Therefore progress in mechanical seal performance was concentrated primarily in two directions. First, improving the stability of the lubricant film on the sliding interface in order to eliminate or minimize dry or boundary friction. Second, developing seal face materials capable of coping with local tribo-stresses and offering improved tribological properties. This approach resulted in the development of the spiral grooved seal concept for pump applications. The first patented upstream pumping seal is now applied to difficult sealing applications (Sedy, 1981).

UPSTREAM PUMPING SEAL—HOW IT WORKS

The idea of minimizing contact friction and therefore reducing the influence of thermomechanical stresses at the sealing interface is not new and was first applied in the early 80s on dry gas seals for centrifugal compressor applications. For noncontacting gas seals there are several different designs to achieve noncontacting operation. The most widely used are spiral grooves, U-grooves, T-slots, stepped grooves and wavy faces. The authors will mostly focus on the spiral groove design because at present this concept is used for liquid upstream pumping seals. The generalized concept of an upstream pumping seal versus a conventional pressurized double seal is shown in Figure 1. It consists of an inboard and outboard mechanical seal. The inboard mechanical seal consists of a pair of seal faces. The stationary seal is usually made of carbon or silicon carbide containing an increased amount of solid lubricant phase like graphite. Usually the rotating face is harder than the stationary face. Spiral groves are recessed onto the rotating face because of the specifics of the manufacturing process. During operation the rotating ring pumps barrier liquid from the seal gland cavity into the pump. Usually seal chamber pressure is higher than the barrier pressure; therefore barrier fluid is pumped against the pressure gradient—hence upstream pumping. Primary function of the outboard seal is to seal the barrier fluid in the seal gland cavity and serve as a tight seal in the event of an inboard seal failure. Cooling fluid will enter the grooves and, with rotation of the ring, will be moved outward toward the closed ends of the groves by viscous shear. The rotating face of the mechanical seal is illustrated in Figure 2. Depending on the seal duty the depth of these grooves may range from 2 to 6 microns. The area from the outer diameter of the spiral groove to the outside diameter of the face of the opposing sealing ring forms the sealing dam. Spiral groove seals operate by using the principles of fluid mechanics. With seal rotation, liquid flows into the spiral groove by a viscous shearing action and is moved outward to the end of the grooves. For normal seal operation the opening forces and closing forces should stay in equilibrium. However a significant proportion of the opening force is from the pressure generated by the grooves and this leads to a thicker fluid film than would be the case with an ungrooved seal. The combined film pressure results in an opening force greater than the closing force that separates the faces approximately 98.42 μin (2.5 μm). At shutdown, hydrostatic forces along with the spring load act to close the faces. Seal balance and the design of the grooves prevent damage to the faces at startup and shutdown prior to separation. The sealing dam plays an important role in the performance of the upstream pumping seal. It is believed that spiral grooves act to restrict the exit of liquid from the groove tips, as pump operating against closed discharge, and therefore considerable pressure can be generated.

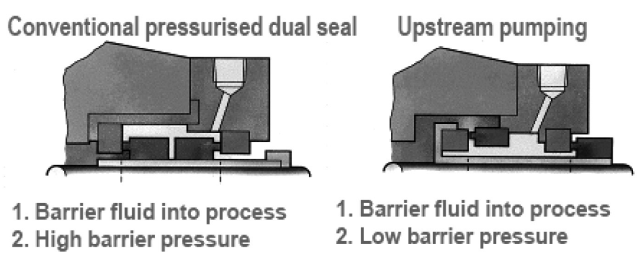

Figure 1. Simplified Scheme of Upstream Pumping Versus Conventional Double Pressurized Seal Working Principle.

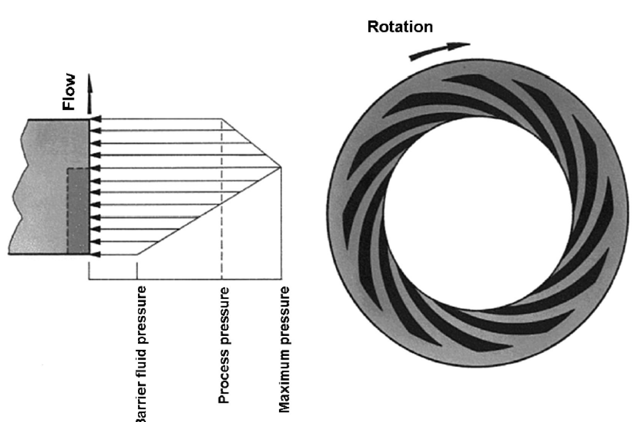

Figure 2. Spiral Grooved Mating Ring.

A conventional mechanical seal usually has lapped faces to within two light bands (flatness). Light band is a commonly used measure for flatness in the lapping process and it numerically equals to 11.6 μin (0.295 μm). During operation the hydrodynamic action of the seal allows a gap of 19.68 to 118.11 μin (0.5 to 3 μm) to form between the seal faces. This interface film lubricates, cools and prevents mechanical contact between the faces. Cooling fluid should therefore be stable and clean to allow a good interface film to form. The pressure drop profile across the faces will generally vary with liquid properties throughout the seal life wear of the flat seal faces. Seal designers always try to ensure that under the worst operating conditions the pressure penetration of the faces will not overcome the closing forces to the extent that gross leakage will take place. Sometimes the seal designer may face a conflicting situation between force balance and efficient control of the pressure profile across the faces.

In the upstream pumping concept the clear area of face running around the grooves provides the dam that creates a higher pressure in the grooves. The physical presence of the sealing dam ensures there is no leakage in static conditions, not "higher pressure," which in any case, is only generated by and in the grooves during rotation.

Usually seal power consumption is defined by the sum of rotational turbulence, interface viscous shear and interface mechanical friction. It has been established that the viscous shear of the liquid between the faces and mechanical rubbing of the rotating face against the stationary face are the major sources of the heat generated on the sealing interface. In general heat generated by a mechanical seal can be described by the following equation:

$$H = P_f V A_f f + U A_f V^2 / h \tag{1}$$

where:

P_f = Average mechanical load on seal face/face area
V = Mean peripheral velocity of seal face
A_f = Seal face area
U = Viscosity of barrier fluid
f = Coefficient of friction
h = Seal face separation

The first term in Equation (1) represents heat generated by mechanical rubbing. The second term represents heat generated due to viscous shear. Although it is generally assumed that the heat generated by viscous shear in an upstream pumping seal might be 10 to 20 percent higher compared to a conventional seal because of the higher face separation and its variation over the grooved area, there is no mechanical rubbing and all heat is generated by viscous shear. Therefore, an upstream pumping seal will generate less heat compared to a conventional seal due to elimination of face rubbing. Therefore reduced film temperatures can be expected (Morton, et al., 2005).

It is generally proposed that upstream pumping seals should demonstrate a number of advantages over conventional tandem seals. These main advantages are:

- Reduced power consumption.
- Extended seal life.
- Effective seal performance at difficult sealing applications.
- Elimination of barrier fluid contamination.
- Reduced heat generation. The groves contribute to a noncontacting operating mode (like a dry gas seal). Seal face temperatures are reduced because the seal gland is at lower pressure than the barrier fluid pressure used in a double seal. This also reduces the risk of scaling on the seal process side when water is used.
- Reduced maintenance. Once the pressure and flow rate are set before start-up, no adjustment is required (unless there is a significant change in the supply pressure). No need to top-up the tank with oil, or set accumulator precharge or barrier pressures.

A double seal has proven to be a viable sealing concept and can deliver many years of reliable service when it is maintained and operated correctly. However, a double seal is completely reliant on the Plan 53 barrier system, which can suffer a number of problems. These are:

- Reverse pressure.
- Accumulator bladder deterioration.
- Air pump (or air supply) failure.
- Barrier oil contamination.
- Venting problems.

One of the characteristic features that double seals can demonstrate in service is their vulnerability to scale and solids build up on the wetted parts of the seal, causing hang-up and poor film formation between the inboard faces.

EXPERIENCE WITH SEA WATER INJECTION PUMPS

Two high pressure multistage centrifugal pumps installed on the offshore oil and gas platform in the Caspian Sea were fitted with upstream pumping seals in February and July 2007. These injection pumps raise the pressure of the deaerated sea water for injection into the reservoir. Each pump is rated for a flow of 192,866 gal/hr (730 m^3/hr) at 4,164.7 psia (28,620 kPa) differential pressure. These are eight-stage barrel pumps on continuous duty driven by a 10,000 hp (7.6 MW) gas turbine via speed reducing gearbox. The pumps take deaerated sea water from booster pumps, and increase the pressure to approximately 4,654.7 psia (32,101 kPa) for injection into the reservoir.

Originally both pumps were fitted with double seals, which proved to be unreliable for this application. Most of the problems were recorded on outboard seals and were manifested by fretting at the outboard antirotation keys. Although all seal faces were usually found to be in good condition, the outboard rotary seal ring was often found displaced. Considering the business impact from frequent seal failures it was decided to modify existing double seals to single seals. This decision was supported by simplicity of single seals compared to double and by the fact that the same seals installed on a similar duty pump at one of the platforms in the North Sea were demonstrating satisfactory performance. Modified single seals were demonstrating improved service life of about 18 months between failures compared to originally installed seals. In January 2007 the water injection system underwent a major upgrade. Although main injection pumps were designed for a nominal flow of 192,866 gal /hr (730 m^3/hr) operations has been complaining about a shortfall in the performance, which was as low as 136,327 gal/hr (516 m^3/hr). It has been concluded that this underperformance was a result of a power shortage by the gas turbine driver exacerbated during the summer season due to elevated ambient temperatures. Objective of the upgrade was replacement of the existing booster pumps by units generating higher head at the same flow rates. Due to a fourfold increase in booster pump differential head main injection pumps had to be operated at new suction conditions. This made existing main injection pump shaft seals unsuitable for new service conditions. Therefore the decision was made to modify the main injection pump shaft seals in order to meet new operating conditions.

The seal of interest was the double mechanical "upstream pumping" seal incorporating a spiral grooved inboard mating ring made of reaction bonded SiC and a primary ring made of SiC-based composite containing up to 30 percent graphite. The outboard primary ring was fabricated from super duplex stainless steel with a mating ring made of SiC. A cross-sectional drawing of the seal is given in Figure 3.

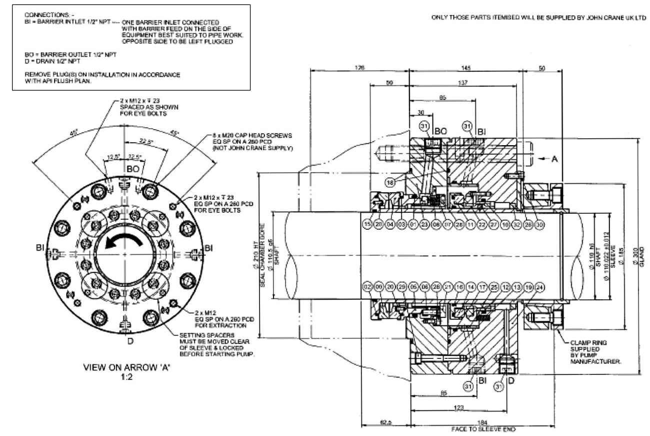

Figure 3. Upstream Pumping Seal.

Sealing conditions are given below:

- Product conditions

Sealing product:	Deaerated sea water
Process temperature:	41°F (5°C) to 64°F (18°C)
Specific gravity:	1.008
Viscosity:	1.2 cP

- Pump data and operating conditions

Shaft diameter:	3.937 in (100 mm)/4.33 in (110 mm) at seal
Speed:	6063 rpm
Suction pressure:	732.25 psia (50.5 bara)
Discharge pressure:	4857.5 psia (335 bara)
Seal barrier pressure:	29.2 psia (2 bara)
Seal chamber pressure:	732.25 psia (50.5 bara)

In this particular arrangement the upstream pumping seals use a continuous flow of filtered sea water. This is introduced through the inlet port on the seal gland plate, which is located between the inboard and outboard seals. Sea water serves to remove the heat generated on the sealing interface and also lubricates the seal faces. After picking up heat from the seal the buffer fluid leaves through the outlet port on the gland plate. Usually around 0.132 gal/hour (0.5 liters/hour) of the buffer fluid is consumed for lubrication of the inboard faces. This fluid is pumped directly into the process.

The nominal amount of the clean sea water required to keep the seal faces from overheating should be higher than 1.56 gal/min (6 liters/min).

The seal support system is fairly simple and was designed for reduced maintenance and operating costs (Figure 4). It incorporates two main assemblies: a filter package and buffer flow/pressure control panel. Sea water is supplied from the ring main supply. It is then fed into the filter package via a normally open ball valve and filtered through a 394 μin (10 μm) filter. The filter is a duplex type with a manual crossover valve, so the active filter cartridge can be changed over during operation without interruption to the buffer fluid supply. A differential pressure indicator across the inlet and outlet of the filter indicates when it is becoming blocked. A pressure indicator shows the pressure leaving the filter. From the filter, the clean fluid is fed to the inlet port of the upstream side of the control panel. It passes through a ball valve, a needle valve on a tee, and a ball valve, which acts as a block and bleed assembly for isolation, bleed and test purposes.

The ball valves will be normally open, and the bleed valve normally closed. Next are a nonreturn valve (NRV), and then a needle valve, which will be normally open but partially throttled to set the flow rate. After the needle valve, there is another NRV to give 100 percent redundancy. The sea water is then taken to the "buffer inlet" (BI) connection on the seal plate.

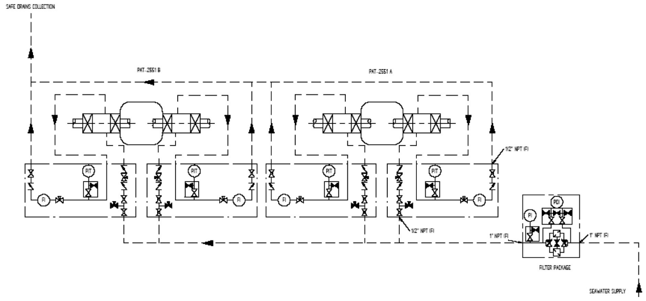

Figure 4. Filtered Sea Water Flow Diagram.

Leaving the seal through the "buffer outlet" (BO) connection, the buffer fluid is taken back to the inlet of the downstream side of the control panel. The buffer fluid passes through a ball valve (normally open), and a tee to the pressure indicating transmitter. Next is a needle valve that is used to set the back pressure of the buffer fluid. The flow is measured locally via a flap type flow gauge. Finally there is a nonreturn valve and a ball valve (normally open). The panel outlet should be piped to safe drains for buffer fluid disposal.

The system was initially commissioned with filters of 394 µin (10 µm) absolute rating. However, due to their short service life it was decided to replace them with 985 µin (25µm) filters. These filters provided improved service life ensuring reasonably clean buffer fluid. These filters were replaced every 10 to14 days. This was the only operating cost associated with the system. After a year of reliable service the filter service life impaired significantly. For a couple of months filters were being replaced on a daily basis, which had a significant cost impact on the maintenance budget. Careful analysis of the sea water samples collected from different points of the system showed that the particulates plugging the filters had an organic source and basically originated from sea water plankton. It became obvious that the system was extremely sensitive to seasonal blooming effects affecting sea water cleanliness. Various options were evaluated for reducing the operating costs without compromising reliability. As the system was designed for a continuous flow a large amount of filtered sea water was wasted. Converting the existing continuous flow system into a closed loop system could significantly increase the risk of salt precipitation. Therefore this option required that the working fluid should be a distilled water/monoethylene glycol mixture. The major disadvantage of closed loop seal support system is the requirement for a reasonably large reservoir and additional main and standby pumps to ensure fluid circulation. Considering the space and weight constraints and requirement for additional instrumentation this option did not seem to look as the first choice at that time. Another option was to use a self cleaning filter with a fully automated feature with existing continuous flow system. One of the fundamental design flaws with the original system was absence of unit trip function from the reduced buffer fluid pressure that could potentially address the seal cooling fluid starvation due to filter or other blockage in the fluid supply line. Initial seal protection logic was relatively operator dependant and was annunciating alarm when the pressure sensed in the "buffer outlet" line was falling to 20.5 psia (141.325 kPa). Each shift on the platform was responsible for visual observation and recording of the cooling sea water filter differential pressure.

Since installation in February 2007 there have been two failures. The first failure occurred on the nondrive end seal of the second train injection pump after 15 months of continuous service. The second failure occurred on the drive end seal of the first train injection pump after 18 months of continuous service. The nondrive end seal from the first train injection pump and drive end seal from the second train injection pumps are still in service.

Both seal failures had common visual evidences and were thoroughly investigated in order to identify a root cause. Both failed seals were analyzed in the pump contractor workshop. After disassembly of the major components it became obvious that the inboard mating ring was completely destroyed and sleeve antirotation pins were damaged. The inboard primary ring was severely damaged with a chipped appearance around the entire surface (Figure 5). Copper-like metal fragments were also found in the seal gland. The shaft sleeve was discolored under the inboard seal (Figure 6). The outboard seal rotary seal ring had scale formation and a damaged stationary seal ring. The inboard flow guide was severely corroded. It was obvious that the seal faces had suffered from lack of cooling. There were visual heat marks, thermal rotation of the IB and OB seal rings witnessing lack of cooling caused by starvation mode. More evidence for heat stress was the discoloration of the shaft sleeve under the inboard seal. The complete disintegration of the inboard SiC mating ring and cutting of all antirotational pins in the shaft sleeve indicates overtorque due to heavy friction under dry running conditions. The fractured SiC seal ring fragments have caused heavy damage to the shaft sleeve during further pump rotation. Based on the root cause failure analysis findings it was proposed to restore unit trip function from low barrier fluid pressure, which would ultimately offer a better protection during reduced buffer fluid flow due to filter or cooling fluid supply line blockage.

Figure 5. Damaged Inboard Primary Ring.

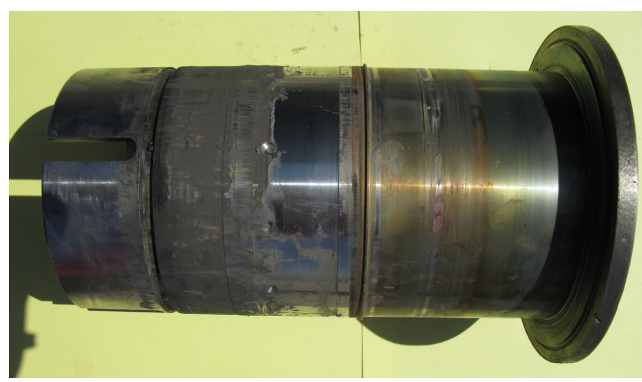

Figure 6. Damaged Shaft Sleeve.

In order to reduce the maintenance cost it was also proposed as a short term solution to use stainless steel fiber filters of the same absolute filtration range. Also higher in initial cost these filters were offering a multiple use feature due to their capability to recover performance after treatment in the ultrasonic bath. The long term solution was installation of the "backflush" filter arrangement working on the automated self-cleaning principle.

CONCLUSIONS

This paper explains the basic operating principles of upstream pumping mechanical seals and presents actual operating experience with a high duty centrifugal injection pump installation on an offshore oil production platform. It was shown that although the upstream pumping seal technology has been available in various applications for a comparatively long time its wider use in demanding industrial applications has only started to gain success recently. It was shown that upstream pumping mechanical seals offer the potential for improved reliability and relative simplicity. However in a real application there is a risk of high operating costs if the system is not carefully designed. Particular attention was given to the fact that continuous flow systems relaying on filtered sea water can become extremely sensitive to seasonal blooming effects affecting sea water quality. Nevertheless with a carefully designed seal support system a significant step change in the seal reliability and reduction of maintenance cost could be achieved.

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