DEVELOPMENTS IN HIGH PERFORMANCE SEAL DESIGNS FOR CRITICAL HIGH PRESSURE OFFSHORE AND PIPELINE APPLICATIONS

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ABSTRACT

Ever demanding business and operating conditions have highlighted the need for significant improvements in the reliability of high duty mechanical seals. With platform operators under pressure to minimize deferred oil production due to equipment failure, there has been, in recent years, a much closer link between user and vendor to jointly develop solutions to satisfy the business requirements. Using a root cause basis, and using advanced materials and technologies, the key problem areas affecting reliability have been addressed and solutions provided for some pioneering applications. This paper looks at the key problem areas of reliability and details solutions that have been applied to move mechanical seal operation to a new level.

INTRODUCTION

With approximately 2000 platforms or equivalent structures currently in operation, this equates to approximately 22,000 pumps operating on process critical applications, where downtime will directly influence the profitability of the asset. Of these facilities 10 percent plus (and this number is growing significantly) are now operating in areas where pressures and duty conditions are becoming more demanding (Figure 1). This equates to 2000 to 2500 pumps that are now operating in extreme duty services. As recovery depths get deeper, pressures get higher, not only for the main oil line (MOL) but also the produced water injection (PWI) or sea water injection (SWI) equipment. With pressures now commonly increasing toward 200 bar (3000 psi), the reliability of the equipment is becoming more of an issue, and therefore more critical.

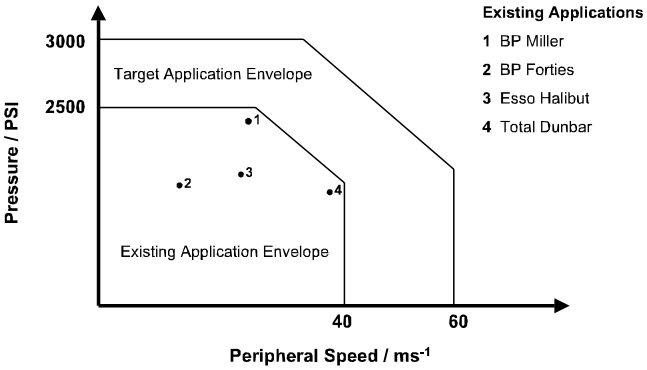

Figure 1. Model of High Performance Seal Performance Envelopes.

With the costs associated with managing these demanding applications escalating, the seal is often now being seen as the "problem." Though in consequence it is actually the weakest link, and highlights the problem rather than being the root cause of the problem. Weakest link or not, it cannot disguise the fact that it is possible to lose 100,000 bbl/day of produced water and 65,000 bbl/day of deferred oil production when critical equipment goes down. With these levels of potential loses, it is time to review the weakest link and improve its susceptibility to these changing operating environments.

If we choose to catalogue the symptoms of seal failure in this environment, they can be broadly categorized as failures associated with:

- Reverse pressure effects.
- Hang-up of sliding seal components.
- Material distortion or deformation.
- Interface running effects.

This paper will categorize and review the consequences of each of these.

HOW SEALS CAN BE DESIGNED TO OVERCOME POOR HIGH REVERSE PRESSURE OPERATION

What Is Reverse Pressure and How Has it Traditionally Been Addressed?

Dual pressurized mechanical seals rely on the pressurized barrier fluid to provide an effective fluid film between the seal faces (Figure 2). This fluid film is only formed when the barrier pressure is greater than the stuffing box pressure.

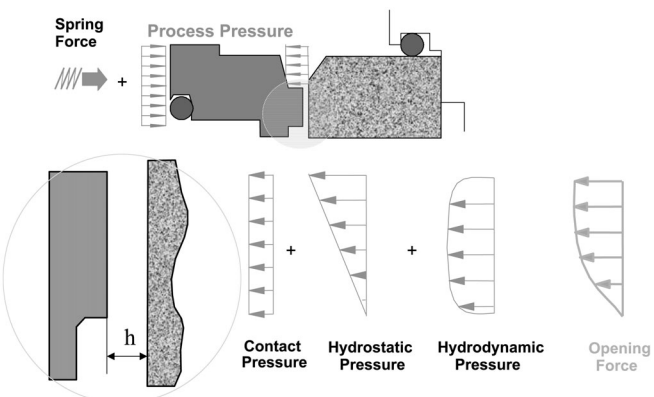

Figure 2. Pressure Distribution Model.

A seal is said to see a reverse pressure when the stuffing box pressure is greater than the barrier pressure. However it is highly undesirable for double seals to allow pumped product into the barrier system, and thus most seals are designed to seal tightly under these conditions. This high level of sealing, both dynamically and statically, is achieved by using a high seal face balance ratio approaching or, in some cases, exceeding 1.0.

With such a high balance ratio, while providing a strong resistance to product ingress, the resultant high closing force will generally lead to rapid face deterioration during the dynamic environment, as highlighted in Figure 3.

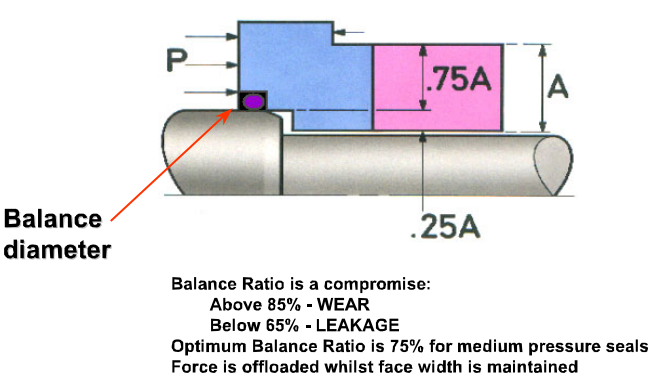

Figure 3. Model of Balance Ratio Understanding.

Why Do You Get Reverse Pressure Operating Conditions?

There are many operational reasons for seals being subjected to reverse pressure, but in general, with careful consideration of the design and good operational housekeeping, most could be prevented.

Looking first at the operation of seal systems, the simplest cause is basic operator error or mechanical failure allowing the pressure within the barrier fluid to drop below preset levels. Normally, on well designed systems, this occurrence has been acknowledged and alarms and/or switches minimize the chances of this occurring, but nonetheless it does occur with alarming regularity.

In the same dual arrangement, catastrophic failure of either the inboard or outboard seals can induce such a high support system leakage as to prevent corrective actions allowing pressure to be maintained within the system, subjecting the inboard seal often to extreme reverse pressure differentials.

In some cases, poor system design can easily introduce the possibility of reverse pressure. Open loop API Plan 54 systems rely on an electrically powered pump to generate, via a back pressure control valve, the barrier pressure. Sudden or planned loss of the electric power will lead to an immediate loss of barrier pressure, unless valve and accumulators are included in the system to provide a static emergency backup.

These are just a few of the barrier pressure side influences on reverse pressure. From the product side, again there are numerous scenarios.

One common occurrence is where two pumps operate in parallel. Although check valves may be included on the discharge side of both pumps, failures of these can lead to the discharge pressure from the running pump applying full pressure to the standby pump, resulting in the reverse pressure on this piece of equipment.

Multiphase services are renowned for introducing reverse pressures, often by way of pressure spikes and abnormally high locked-in well pressures, which may only be drawn down after a period of pump operation.

Historical Reverse Pressure Seal Solutions

Reverse pressure is not a new phenomenon to mechanical seal design, and has been catered to in many ways in the past. However in most instances design applications were moderate (30 to 50 bar, 450 to 750 psi), and the ability of the seal to withstand these conditions with only minor design changes was never in serious question. However once the demand for higher pressures over these limits is called for, then the design requirements become completely different. Design effects used previously include:

- Material changes.
- Component shapes (either inert or flexible).
- Banding and composites.
- Reverse balancing.

Material Changes

These are normally associated with finding and utilizing a stronger material, one where there is greater tolerance to the mechanical forces being applied to the seal or the component. Traditionally these have focused on the use of carbides and later silicons to achieve the levels of performance sort (Table 1).

Table 1. Table of Properties Highlighting the Core Factors.

Material	Bend strength *MPa*	Modulus *GPa*	Therm.Cond *W/mK*
Carbon	83	33	12
Cranite	150	160	125
Silicon Carbide	240	360	150
Tungsten Carbide	480	600	75

It is perhaps no surprise that face materials based around tungsten carbides have been popular. It is also an added benefit that tungsten carbide has excellent running capabilities when paired with traditional seal face materials.

Component Shapes

After material changes, component shapes are probably the most commonly applied change in the search for enhanced reverse pressure ability. This is especially so with the development of finite element analysis, where more accurate modeling and therefore forecasting of deformation is possible. Shapes tend to be either inert, basic sections that react with minimum distortion and can be controlled further by the application of strategic pivot points, or they are flexible and designed to bend or rotate in a certain way within the material to give the required face presentation during operation (Figure 4). Both these design characteristics will only affect the component up to the maximum physical constraints applied to the material (allowing for an acceptable factor of safety).

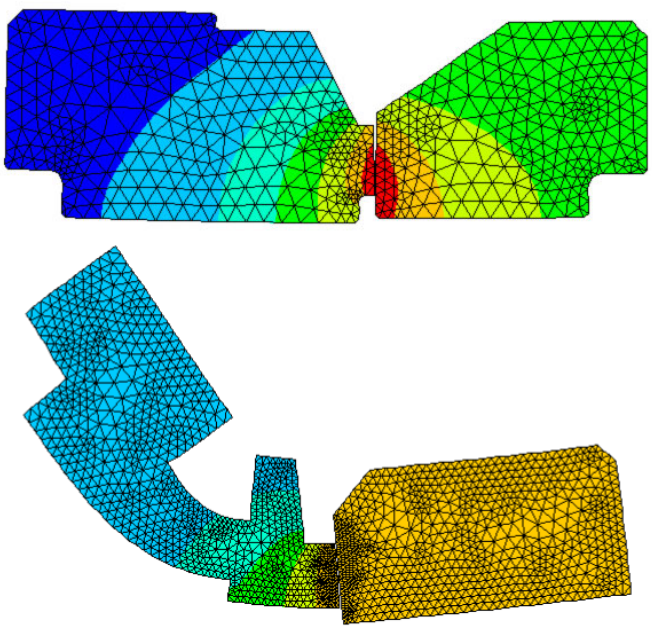

Figure 4. FE Example of Faces of Both Design.

Restraints or Shrouds

External restraints or shrouds have been used in many instances to help stabilize weaker materials when high internal pressures are applied. This shroud usually consists of a metal band of a slight interference fit applied to the outside of the component under pressure. While in practice this is a simple and robust solution, there are several downsides that will negatively affect the way the seal operates. Shrouding a face material with a steel band will, or can, develop the following problems:

- Face material shrouded from critical cooling flow causes overheating of the seal components and rapid performance degradation.
- Thermal effects on the components are different; as temperatures increase, the restraining effect of the band is reduced, shown in Figure 5. This relaxation introduces distortions that were lapped out when the components were assembled and finished, usually cold, also reducing the pressure retaining effect.

Reverse Balance

Reverse balancing takes into effect one of a mechanical seal's primary design features to allow the seal to operate in both standard and reverse pressure modes. The actual performance level

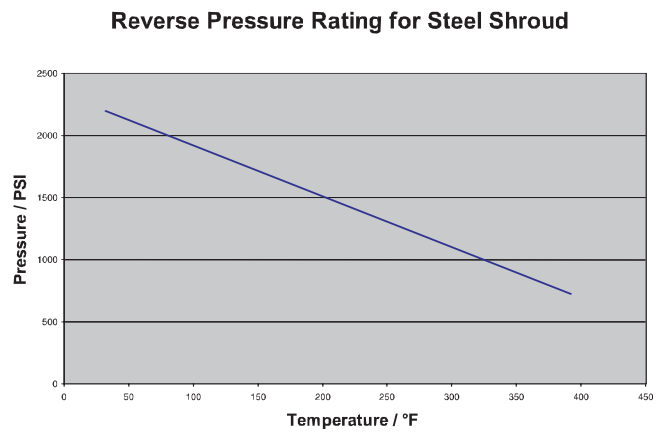

Figure 5. Graph of Steel Shroud Pressure and Temperature Limits.

is still governed by the physical attributes of the material used and how it distorts, but reverse balancing (Figure 6) allows the seal to remain hydraulically closed during a variety of operational regimes.

The effect relies on the movement of a secondary seal O-ring along the secondary sealing diameter. This diameter is different depending on the way the pressure acts. The skill in design is to balance the seal along both sliding diameters while maintaining optimum seal face width. This is best demonstrated in Figure 6.

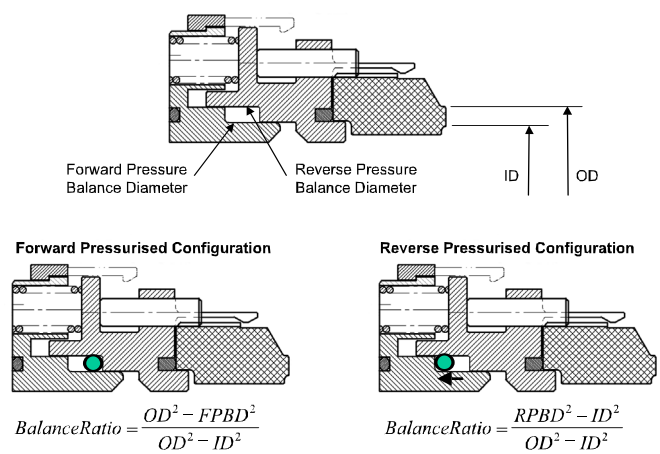

Figure 6. Reverse Pressure Designs.

How Can Seals Be Designed to React Neutrally to Reverse Pressure?

Seal Innovations—Armored Composite Rings

In recent years material development has moved on, and though the seal industry could be considered conservative in adopting these technologies, several notable improvements have recently been made.

Using the same principle that is used in prestressing concrete, this design of seal ring uses a strong outer band of carbon fiber/epoxy to induce a predetermined compressive stress within the inner core seal face material (shown in Figure 7). Due to its extremely high hoop stress capability, this carbon fiber material allows a very high degree of interference and precompression to be introduced into the inner ring. This can be achieved without resorting to large radial sections.

Using patented assembly techniques for this unique ring, reverse pressures up to 300 bar (4350 psi) have been achieved without overstressing or distorting the seal rings to prevent acceptable operation.

The operation and unique performance of this armored composite ring (ACR) are compounded by its thermal expansion

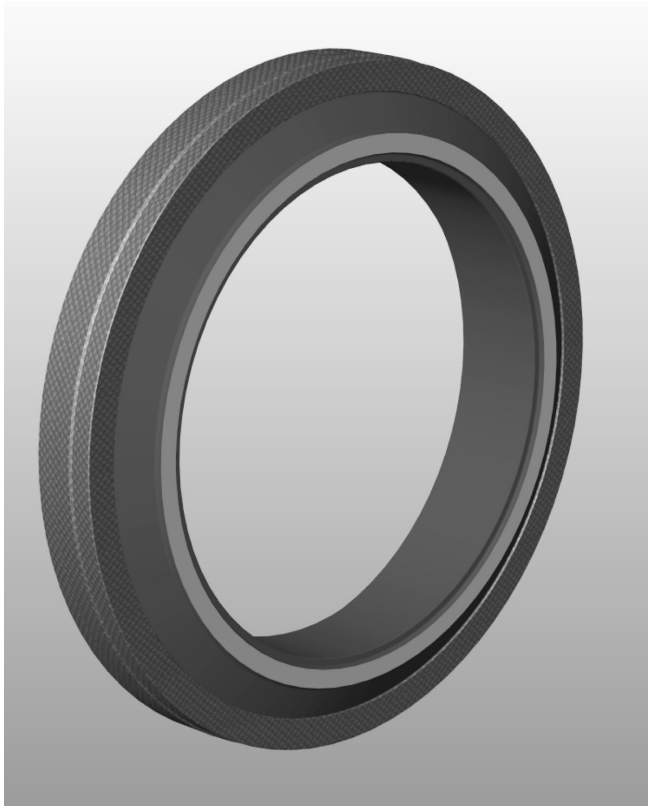

Figure 7. Armored Composite Ring.

coefficient, which being marginally lower then the inner seal face ring, actually results in an increase in reverse pressure capability with increasing temperature (Figure 8). This would not be achievable using a metal shroud, though these are recognized as having other design constraints.

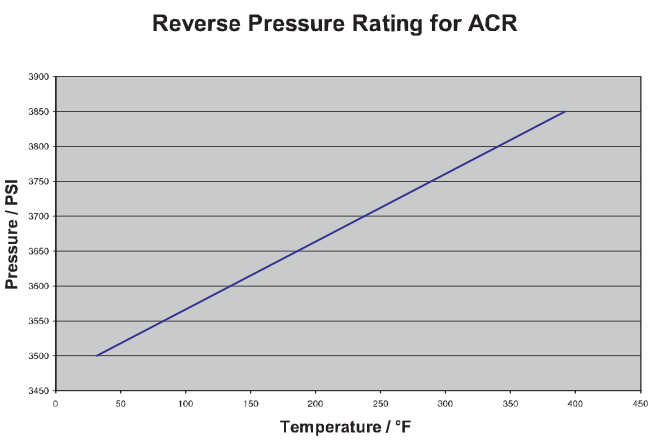

Figure 8. Graph of ACR Pressure and Temperature Limits.

Seal Innovations—A Different Design Philosophy

There are some operational instances where even the ACR style seal ring is not capable of withstanding the projected reverse pressures. To meet the ever increasing operational requirements, discharge pressures can now exceed 700 bar (10,150 psi). It is therefore theoretically possible for seals to now see reverse pressures of this level.

When pressures of this magnitude are encountered, the only option is to actually reverse the design of the seal, as shown in Figure 9, applying the barrier pressure from the inside. In this way, any reverse pressure results in the seal rings being pressurized from the outside rather than the inside (though this does create several new design problems when the equipment is seeing conventional pressure). This type of design puts the rings in compression, a condition that the materials are easily able to withstand. Such designs have now been supplied against high reverse pressure for deepwater injection projects in the Gulf of Mexico.

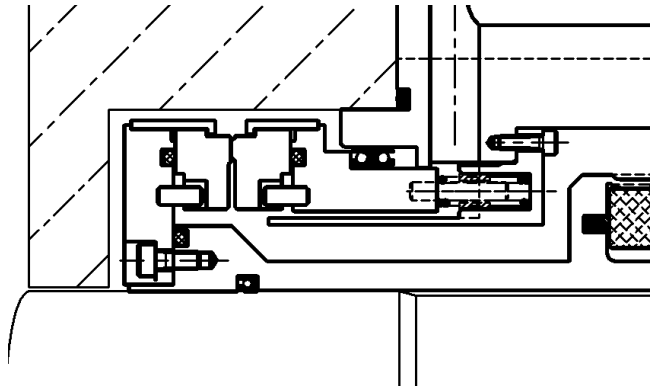

Figure 9. Design of Inside Out Head.

While this design of seal provides a simple solution to extreme reverse pressures, internally pressurized seals are far less tolerant to pressure and speed variations. Indeed, with the barrier pressure now being applied from the inside, care has to be taken not to over-pressurize the rings prior to pump pressure being introduced.

When in operation, internally pressurized seals need to be carefully optimized to ensure that the pressure and thermal rotations developed from the rotational velocity do not act together in such a way to preclude the fluid film from being generated and maintained between the faces (Figure 10).

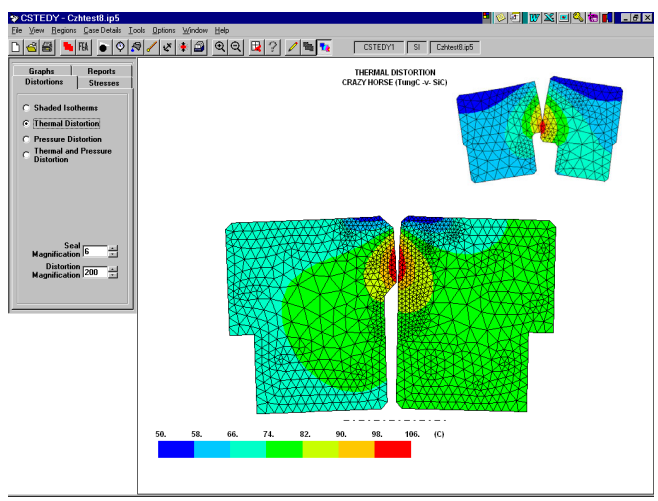

Figure 10. Finite Element Model of Design.

Using finite element techniques and novel design principles, these seals can be made to operate over fairly wide speed ranges up to 7000 rpm and with pressure differentials of up to 15 bar (220 psi).

HOW SEALS CAN BE DESIGNED TO OVERCOME HANG-UP

What Is Seal Hang-Up and How Is It Developed?

Within the design of a mechanical seal, the design must account for relative axial movement between the rotating and stationary

components. This movement can be high frequency—for example, vibration—or low frequency, possibly due to thermal expansion or changes in pressure, as well as during extended operation, the design wear of the mating components.

The primary seal face is designed to follow its mating face axially. A sliding secondary seal prevents leakage between the sliding and stationary components. Figure 11 shows one possible configuration. Although the shaft, sleeve, and carrier move, the primary face must follow the mating face.

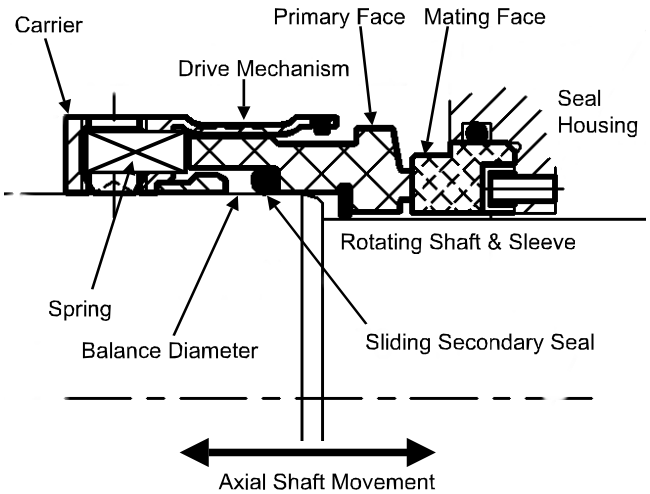

Figure 11. Example of Mechanical Seal Configuration.

Friction forces at these positions of relative axial movement exceed the closing forces—exerted by both spring load and hydraulic pressure (refer to Figure 2)—then the sliding components will not follow the mating seal face. This is seal hang-up.

There are many reasons for friction forces to reach a critical level.

Secondary Seal

The secondary seal is a relatively flexible element. Hang-up associated with the secondary seal usually occurs with large axial movements or during pressure changes. These cause distortion (or relaxation) of the secondary seal. If friction on the balance diameter is high, then the secondary seal may drag the faces open (Figure 12).

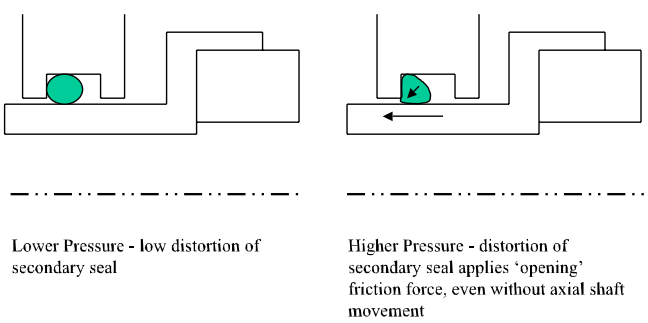

Figure 12. Secondary Seal Distortion Causing Hang-Up.

Because hang-up can happen during pressure changes, and closing forces are low when the seal is not pressurized, seal hang-up is more likely during a pump shutdown. The probability is increased by shaft movements due to either changing pressure or thermal expansion/contraction.

This goes unnoticed until an attempt is made to pressurize the seal, which then leaks heavily as the faces are held open. This can prevent repressurization of double seals.

Buildup of scale or other deposits on the sliding diameter, in front of the secondary seal, can increase friction dramatically, often totally preventing its movement.

Increased frictional effects can also occur when high operating pressures cause the elastomeric secondary seals to deform into microscopic surface roughness on the sliding diameter. The secondary seal can adhere to the sliding diameter even when pressure is subsequently reduced.

Balance Diameter

Depending on the design of the seal, either the sliding components or the axially static components will include the hydraulic balance diameter for the seal. This is critical to the operation of the seal.

The clearances between axially static and sliding components, including that at the balance diameter, are often designed to be very small. This is to reduce the risk of extrusion of the secondary seal.

Where very close radial clearances are present, hard contact—and therefore high friction—may be brought about by:

- Angular misalignment.
- Relative changes in diameter due to temperature and pressure.
- The introduction of solids—possibly suspended in the sealed fluid—into the close clearance.

The result is high face loading (often leading to seal failure), a large increase in seal leakage, or both.

What Are the Operational Symptoms of Hang-Up?

Before hang-up actually occurs, it is very difficult for the operator to detect any warning signs. As hang-up is essentially caused by friction, signs of hysterisis may indicate the problem in advance. This may be apparent if sealed pressures change regularly, and monitored parameters (leakage, heat generation, temperature, torque) do not always change consistently.

If a number of mechanical seals, which should be behaving identically, have monitored parameters that differ significantly, this may also indicate friction and the possibility of future hang-up.

During seal hang-up, excessive seal leakage is usually observed, and pressure within the seal cavity or barrier system drops due to this leakage.

While it is often apparent at the time of failure that hang-up is probably the cause of seal leakage, it is often only evident during disassembly which element of hang-up has actually occurred. The following signs indicate hang-up:

- Springs do not move sliding components on disassembly.
- Fretting is apparent on the drive mechanism and close clearances.
- Sliding components are axially misaligned and/or jammed.
- Secondary seal does not slide smoothly over balance diameter.
- Evidence of scale, deposits of solids around balance diameter and other close clearances

Applications/Services Prone to Hang-Up

Suspended solids—Hard particles are trapped within close clearances causing sliding components to lock up. Abrasive particles damage sliding diameters, increasing roughness, causing increased friction. Particles become embedded in secondary seals causing fretting.

Stop start—Hang-up is most likely to occur during transient conditions when the equilibrium of constant running is disturbed. Intermittent services are more at risk.

Large axial movements—Since most hang-up occurs during axial movement, the larger and more frequent the movement, the higher the probability. Also, fretting further increases the likelihood of hang-up.

Scaling services—Buildup of scale on sliding diameters restricts movement of the secondary seal. Excessive scale can completely lock close clearances. Produced water injection is one notorious example.

Multiphase—These services are noted for large fluctuations in pressure, suspended solids and abrasives, and, in some pump designs, significant axial movement. All increase the risk of seal hang-up.

TRADITIONAL METHODS OF OVERCOMING HANG-UP

Increase Closing Force

Several methods of increasing the closing force between the seal faces have been traditionally used:

- Increasing the spring load is effective at low-pressure conditions. At higher pressures, however, the spring force often becomes negligible when compared with hydraulic forces and friction acting on pressurized secondary seals.
- Increasing the hydraulic balance ratio raises the closing force across the pressure range.
- Widening the seal-face increases both the opening and closing forces, making friction less significant.

However, each of the methods above compromise the tribology at the seal-face, increasing heat generation and moving away from optimum design conditions.

Increase Clearances

This makes the seal less prone to fretting provided that the components are centralized by another design mechanism. With increased clearances, diameter changes caused by temperature and pressure are much less likely to cause problems. However secondary seal extrusion is more likely, and antiextrusion devices have been known to contribute to seal hang-up and fretting, to the detriment of seal performance.

Reducing Friction in the Drive Mechanism

The use of low friction, nonfretting materials in the drive/antirotation mechanism most definitely has a positive effect on hang-up resistance and seal reliability. Provided components are resistant to wear, there are no negative aspects to this approach.

Resisting Fretting on Sliding Diameters

The use of hard coatings—for example, tungsten carbide—on sliding diameters and other close clearances, provides resistance to fretting. This eliminates the fretting/friction cycle that ultimately leads to seal hang-up. Friction is further reduced as these surfaces can be highly polished. This reduces the effect of elastomer seals "gripping" the surface or becoming embedded in the surface structure.

HOW ELSE CAN SEALS BE DESIGNED TO OVERCOME HANG-UP?

Hydraulic Balance Diameter Configuration

Figure 13 shows one common balance diameter configuration. At high pressure, the secondary seal is forced in the "opening" direction, toward the extrusion gap, becoming quite rigid. Friction on the balance diameter acts directly on the primary seal components. This impacts largely on net closing force, even for small displacements. Where this friction force is increased—for example, due to scale or fretting—hang-up can occur.

Figure 14 shows an alternative configuration. With this design, the hydraulic balance diameter is not part of the sliding primary seal components. This means that friction forces on this sliding diameter are not transmitted directly to the seal face, but act through the flexible secondary seal.

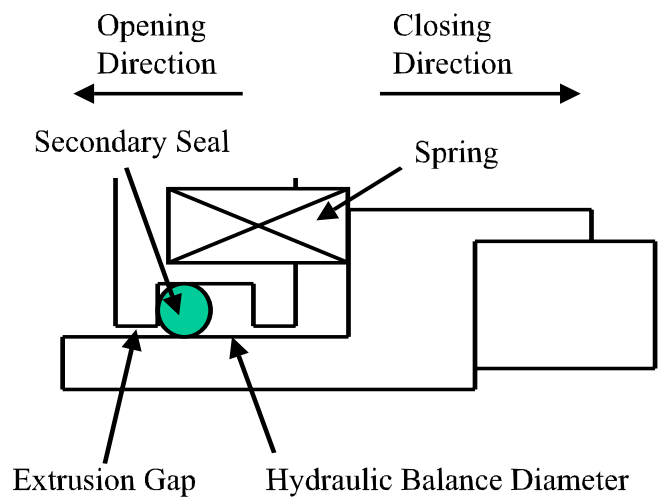

Figure 13. Balance Diameter on Sliding Components.

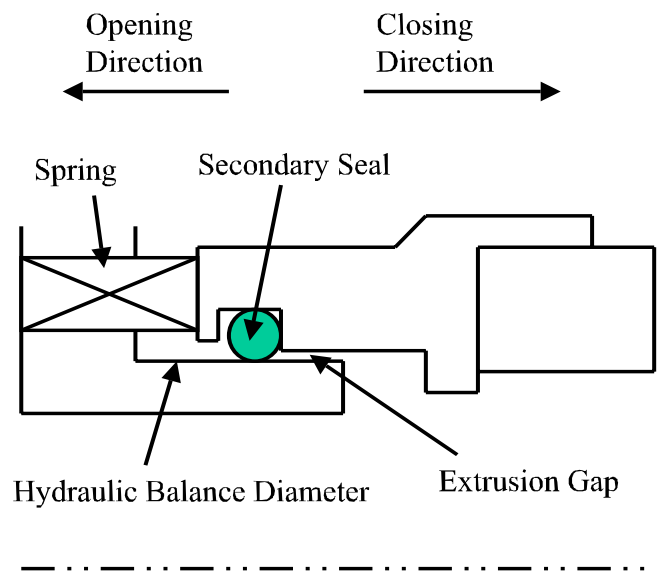

Figure 14. Balance Diameter on Axially Static Components.

In this configuration, the pressure is acting to push the secondary seal in the "closing" direction. This means the pressure assists the secondary seal in sliding or distorting. It does not behave rigidly. Larger displacements can occur before net closing force is compromised by friction.

Also, by decoupling the primary seal components from friction at the sliding diameter, the "secondary seal distortion" hang-up mechanism, shown in Figure 12, is effectively eliminated.

Polymer Seals

Polymer seals, for some time now, have been used very successfully in gas seal technology. They provide ultra-low friction at extreme pressures. Spring energized polymer seals have now been adopted for use in high duty wet seals, as shown in Figure 15. As well as offering lower friction and, hence, reduced risk of hang-up, polymer seals are extremely resistant to extrusion—requiring no additional antiextrusion feature. They also offer increased pressure, temperature, and chemical resistance capabilities.

HOW SEALS CAN BE DESIGNED TO OFFER 200 BAR (3000 PSI) PERFORMANCE

As indicated in the INTRODUCTION, with applications getting deeper and demands greater, the need for seals to more regularly

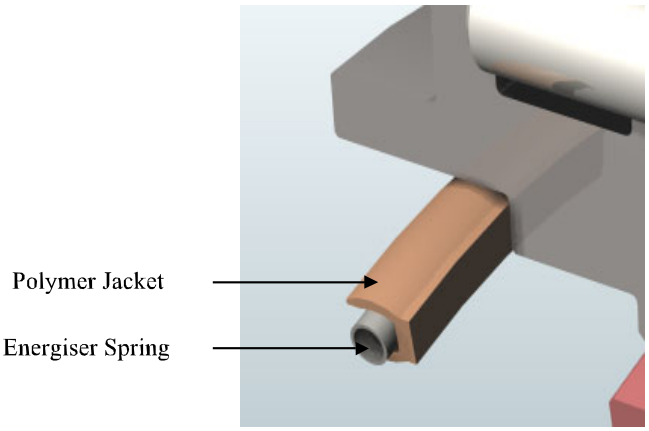

Figure 15. Model of Polymer Seal.

withstand extreme pressures up to and often exceeding 200 bar (3000 psi) is becoming more common. Therefore it is becoming essential to establish best practice in this field.

Distortions and Rotations

In mechanical seal operation the lubricating film between the faces is typically on the order of 1 µm. This means that distortions of the seal face as low as 0.1 µm can have marked effects on seal performance. When a seal design operates over a wide range of pressures and temperatures, distortion control becomes paramount.

Seal face distortion can be divided into two main categories:

- *Radial distortion*—Often called rotation
- *Tangential distortion*—Sometimes called face waviness

Radial Distortion

Distortion is usually caused by temperature gradients within the seal ring and by pressures and forces (loads) that exert moments about the cross sectional centroid.

Temperature distortion has the effect of increasing the diameter of the seal ring adjacent to the sealing face. This is where heat is generated during operation, due to friction and liquid shear between the faces. A convex rotation in the seal ring is developed known as coning, as shown in Figure 16.

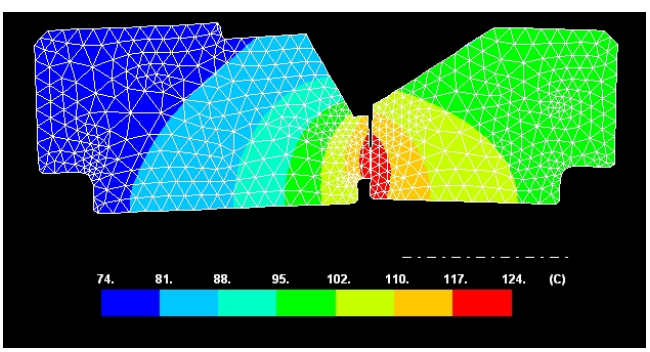

Figure 16. Temperature Rotation.

For externally pressurized seal rings, this opens the faces to the seal fluid (or barrier liquid) providing increased hydrostatic support. This reduces heat generation a little, and a stable operating point is achieved.

For an internally pressurized seal, increased heat generation again causes convex rotation. This now has the effect of *closing* the faces to the sealed fluid (which is on the inside diameter) and reducing hydrostatic support. Heat generation is now further increased. Although in most instances a stable operating condition is reached, this is much hotter than that achieved with a similar externally pressurized seal.

It is also not unique, since reducing the heat generation of a mechanical seal causes less convex distortion within the face components. This, in turn, provides greater hydrostatic support, further reducing the generated heat. There is therefore also a cold operating point, which is equally as stable.

It is common for internally pressurized seals to switch between operating points. This however makes accurate control of the sealing interface tribology more difficult.

In mechanical seal design it is always beneficial to reduce temperature effect rotation. This allows for predictable operation across a large range of speeds and pressures. If the majority of cooling occurs very close to the seal faces relative to the overall length of the face component, stabilizing thermal stresses help minimize thermal rotation.

Pressure distortion can be accurately controlled at the design stage with careful consideration of face loading. The use of finite element analysis/computational fluid dynamics (FEA/CFD) means that seal face geometry can be optimized. This means pressure rotation can provide an effective balance to thermal rotation, across a range of operating pressures and speeds (Figure 17).

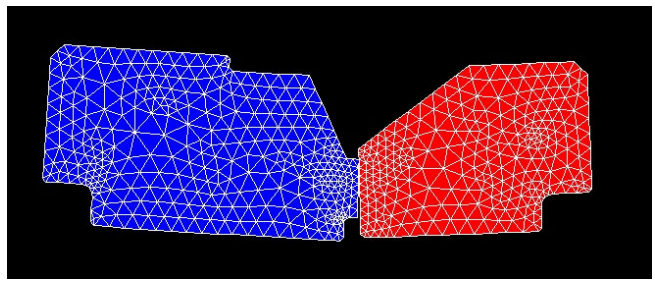

Figure 17. Pressure Rotation.

The key in design is to make sure the computer model boundary conditions, particularly forces caused by friction on the static or secondary seal component, and supports that come into contact with the seal ring are accurately reflected in the actual seal design.

For maximum predictability, hysterisis effects should also be minimized. Reducing coefficients of friction and support loading effectively will achieve this state.

Tangential Distortion (Waviness)

Uneven loading or support around the seal ring will result in unwanted face waviness, which can lead to unpredictable and high leakage. This effect can be reduced and often eliminated by the use of elastomeric soft supports (Figure 18) or by lapping both the seal faces and the metal support abutment.

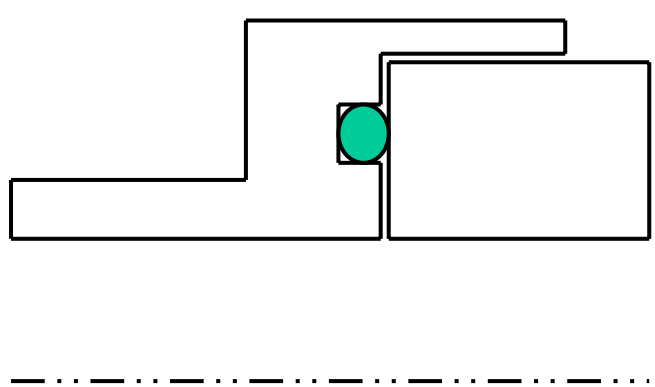

Figure 18. Example of Soft Support.

Soft support is achieved by careful control of tolerances allowing radial extrusion gaps to be minimized while eliminating the chance of hard contact between the seal ring and carrier.

Waviness is also caused by tangential variation in cross section, due to drive slots, for example. Decreasing the size of the features is the first step in reducing this effect, increasing the number is the next. This has the effect of increasing the modal frequency; effectively stiffening the ring against pressure induced waviness.

Extrusions

Both secondary seals and elastomeric support rings are susceptible to extrusion. The higher the differential pressure and/or temperature, the more resilient the design must be.

Methods of increasing resistance to extrusion include reducing effective clearances. The smaller the gap, the less likely extrusion will occur to a detrimental level. As well as reducing diametrical clearances, accurate centralization of components can also reduce effective radial clearances. (Note: Effective radial clearance is equal to the diametrical clearance if no other centralizing mechanism exists.)

In understanding extrusion mechanisms it is necessary to recognize the main causes:

- *Classical extrusion*—Where the elastomeric seal "flows" into the extrusion gap, shown in Figure 19
- *Scrolling*—Where the edge of the extrusion gap cuts the elastomeric seal causing it to unravel, shown in Figure 20

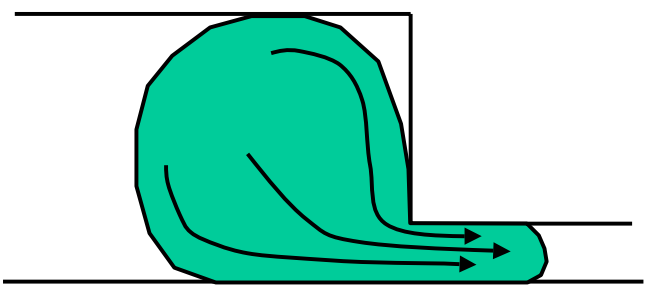

Figure 19. Classical Extrusion.

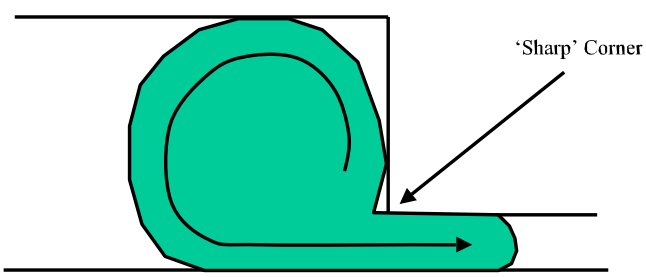

Figure 20. Scrolling Extrusion.

Increasing the hardness of the elastomeric seal can effectively eliminate classical extrusion, providing the extrusion gap is accurately controlled to be small enough. Scrolling is then the only mechanism by which extrusion can occur.

Increasing the material hardness also helps guard against scrolling, but sharp edges on the extrusion gap entrance must be eliminated to guarantee safety. Applying a 45 degree chamfer to the edges (the normal recommendation) is not sufficient as pressures approach 200 bar (3000 psi). The corners must be radiused, with low surface roughness to eliminate cutting of the elastomer. Care must be taken not to provide a lead for classical extrusion. Radius sizes must be chosen carefully.

Using noncircular sections also helps resist scrolling, as it becomes difficult for the seal to unwind. Figure 21 shows one such section, used as a soft support. This has the additional advantage that very little compression occurs when the seal is pressurized. Close control of both groove and support ring dimensions ensure the elastomeric seal behaves predictably over a large pressure range.

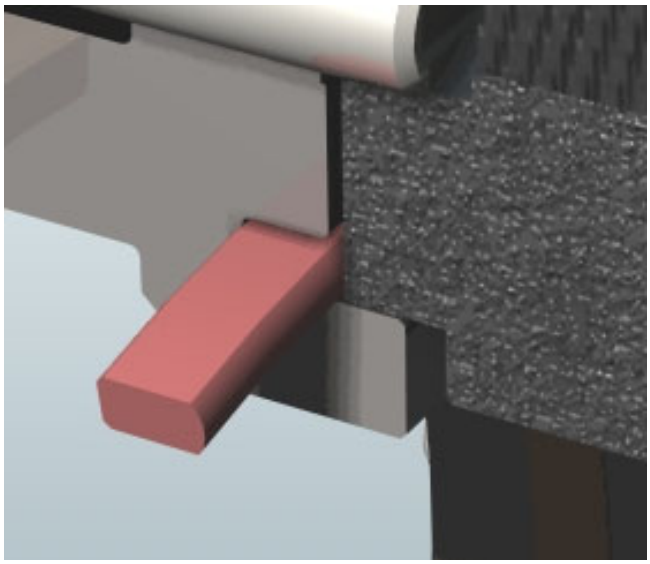

Figure 21. Elastomeric Support Ring Section.

Nonface Distortions

As already recognized, hang-up can be caused by deformation due to pressure and temperature, when close clearance diameters change relative to each other causing hard contact.

During seal design it is necessary to establish limits of pressure and temperature acting on components. Care should be taken that, for worst case tolerances, such components never contact while within these limits.

Using materials with similar coefficients of thermal expansion and matching component sections and modulus of elasticity can assist in achieving this objective. Careful design ensures nonface deformation is *not* a limiting factor in establishing the performance range for the seal.

Loads and PV factors

Using a low balance ratio can reduce loading of the seal face, though this dramatically increases the seal face leakage as it increases face separation.

Seal leakage is proportional to the cube of the face gap. If the face separation doubles, the leakage increases by a factor of eight.

High duty seals have high hydraulic balance ratios, in the range 75 to 80 percent, to achieve a balance between load and leakage.

At 200 bar (3000 psi), present value (PV) factors are way above that which can be tolerated by even the most innovative materials during a boundary lubrication regime. It is therefore essential that contact during normal seal operation is eliminated to preserve the integrity of the seal face materials. High duty seals must rely on hydrostatic and hydrodynamic support only.

With high balance ratios, high film stiffness is necessary to achieve stability and reliability. Hydrodynamic support generated using advanced lapping techniques and specialist hydrodynamic structures can increase film stiffness with little or no change to nominal film thickness and seal leakage. An example of a unique structure is shown in Figure 22.

Innovative composite materials provide additional tolerance to high loading during upset conditions when face contact can occur. Examples are during reverse pressure operation and during startup and shutdown, when hydrodynamic support is insufficient at very low speeds.

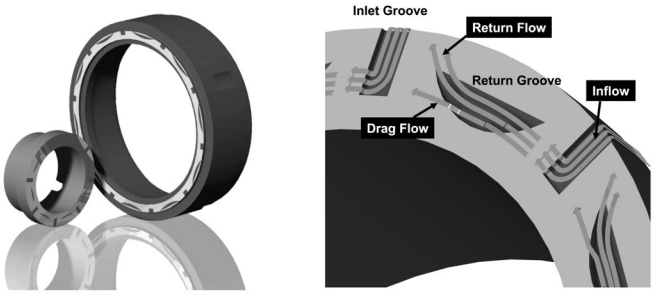

Figure 22. Examples of Specialist Hydrodynamic Structures.

THE COST BENEFIT IMPACTS OF SEAL UPTIME

The review in the previous sections has shown there to be many forms of equipment failure as a result of poor seal performance in increasingly demanding applications.

Regardless of the actual failure modes, the impact of these failures on the facility or the asset is always significant in financial terms compared with the component cost. Therefore no matter what the cost of achieving longer seal life, this is always inconsequential to the actual costs of not improving the reliability of the asset. This does not only impact the cost of support, which when considering an offshore asset can in itself be considerable, but also the costs associated with the production and loss of produced water and/or the costs associated with deferring oil production.

To highlight this impact on users, it is necessary to look in more detail at some typical industry cases. (Note: Cases are associated with UK sector North Sea, though the cost models would be applicable to most similar environments.)

In case 1, by reducing the cumulative pump outages from seven to two on the produced water injection pumps reduces the consequential loss in deferred oil from 5.1m bbl to 1.4m bbl (approximate operational improvement of $17m ($24m to $7m) per annum and the loss of injection water from 7.7m bbl to 2.2m bbl. The percentage improved against the cost associated with improvement is 0.1 percent, when compared with a total preventive cost of $17k per seal unit in upgrade and maintenance costs. An outage schedule and typical cost estimate model are shown in Figure 23 and Table 2.

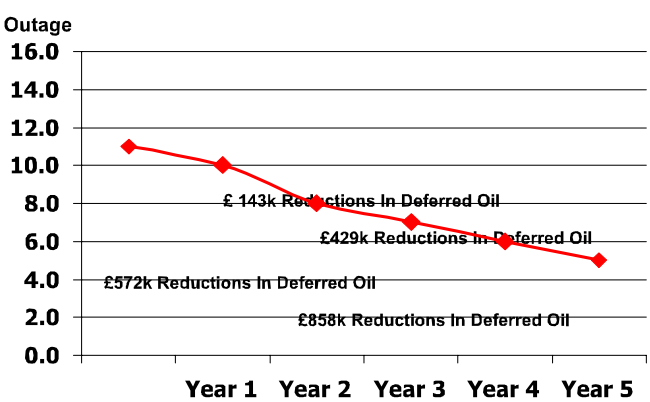

Figure 23. Outage Curve.

In case 2, the failure history of three offshore water injection pumps was again poor due to the factors already analyzed. The typical run rate was two to four failures per annum (based on six seal units) with an immediate history prior to action of eight failures in 12 months. The review of the situation indicated that the problem was getting worse, with root cause analysis (RCA) pointing at several more common failure modes associated with varying but consistent failure mechanisms.

Table 2. Typical Example of Financial Model.

	History	Year 1	Year 2	Year 3	Year 4	Year 5
Direct costs						
Outages pa	7	2	1.3	1.3	1.3	1.3
Pump days lost / outage	11					
Injection water lost (b/d)	100000					
Seal repair cost	$20,300					
Pump repair costs	$5,075	$1,450	$967	$967	$967	$967
Offshore support (pump)	$31,973	$9,135	$6,090	$6,090	$6,090	$6,090
Offshore support (seal)	$9,947					
Total Maintanence Costs	**$67,295**	**$10,585**	**$7,057**	**$7,057**	**$7,057**	**$7,057**
Saving		$56,710	$60,239	$60,239	$60,239	$60,239
Cumlative saving		$56,710	$116,948	$177,187	$237,426	$297,663
Upgrade costs (fee based)		**$69,600**	**$69,600**	**$69,600**	**$69,600**	**$69,600**
Total direct costs		**$80,185**	**$76,657**	**$76,657**	**$76,657**	**$76,657**
Saving		$12,891	$9,361	$9,361	$9,361	$9,361
Cumlative saving		$12,891	$22,252	$31,613	$40,974	$50,337
Production costs						
Injection water lost (b/d)	$145,000					
Total injection water lost	$11,165,000	$3,190,000	$2,126,667	$2,126,667	$2,126,667	$2,126,667
Deferred oil (b/d)	$7,443,333	$2,126,667	$1,417,778	$1,417,778	$1,417,778	$1,417,778
Value of deferrer oil	$23,843,546	$6,812,442	$4,541,628	$4,541,628	$4,541,628	$4,541,628
Saving		$17,031,106	$19,301,919	$19,301,919	$19,301,919	$19,301,919
Cumulative saving		**$17,031,106**	**$36,333,024**	**$55,634,943**	**$74,936,861**	**$94,238,780**

Review of the last seven outages, shown in Figure 24, accumulates the following cost pile:

Seal outages	7
Down time per outage	11 days total
Injection water lost	100k bbl/day
Deferred oil	65k bbl/day
Deferred oil value	$24m
Upgrade costs:	
Six replacement seals	$234,000
Support systems	$45,000
Three year MTBF over five years' operation (5.3 overhauls)	$83,000
Total cost	$361,000
	$60,250 per seal

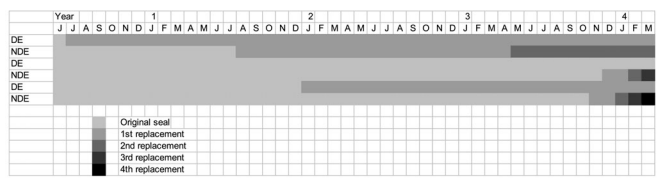

Figure 24. Outage Chart.

Based on maintaining an increased seal performance to an average four year mean time between failure (MTBF), the outcomes in real terms can be quantified as:

- Reduced maintenance budget by approximately 25 percent, $2m to $1.4m over four years
- Produced water reinjection availability increased from 41 percent to 90 percent
- Direct sea water injection pumps' maintenance costs reduced by 77 percent

CONCLUSIONS

- While a mechanical seal is often seen as the cause of the problem with high duty pumps, it is in most instances a symptom of unstable or unplanned running, and the failure acts as a warning.
- By focusing on the key areas that affect reliability, it is possible for seals to operate for extended periods in these "upset" conditions, which is fundamental to improving asset reliability.
- Reverse pressure and seal hang-up have been clearly identified as the main challenges, and by combining new materials and carefully developed sealing arrangements, it is possible to produce more tolerant sealing arrangements.

• The cost of making such changes, when compared with the operational losses or deferrals, is minimal.

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