47TH TURBOMACHINERY & 34TH PUMP SYMPOSIA
HOUSTON, TEXAS | SEPTEMBER 17-20, 2018
GEORGE R. BROWN CONVENTION CENTER

Crude Oil Non-Pusher Secondary Seal

Darin Rasmussen
New Product Development Engineer
John Crane
Morton Grove, IL, USA

Jim Wasser
Director, Design Engineering
John Crane
Morton Grove, IL, USA

John Morton
Manager, Product Line Management
John Crane
Slough, UK

Darin Rasmussen joined John Crane in the Chicago office in 1999 and has held a variety of positions within the engineering department and is currently in the Global Design Engineering Group.

John L. Morton is a Senior Product Marketing Manager for John Crane, based in Slough, Berkshire, United Kingdom. He is also the current Chairman of the European Sealing Association. Mr. Morton has held various posts within the John Crane global organization over the last 36 years, initially in engineering and product development, before taking on sales responsibilities and moving into a marketing role. This has included an extended period spent working in the Nordic regions. He has been involved in the design of, or has product managed many of the innovative John Crane products over the last 20+ years. Mr. Morton has represented the Sealing Industry at the World Trade Organization, and has participated in a European Union trade delegation. Mr. Morton obtained his MBA from the University of Reading in 1998

Jim Wasser joined John Crane in the Chicago office in 1986 and has held a variety of positions with the Engineering department. He is currently the Global Director of Design Engineering, responsible for wet seal development, material maintenance and the global test labs. He holds a BSME from Illinois State University.

ABSTRACT
Crude oil pipeline pumps traditionally suffer from seal leakage due to the fretting or wearing of the dynamic O-ring. A new concept secondary seal has been developed to eliminate this fretting occurrence. All major seal suppliers have products designed specifically for the operational challenges of crude oil pipeline services, some more than others recognize the application difficulties and have design characteristics that belie these. However none are designs based on non-pusher secondary seal concepts.

A "non-pusher" style represents a targeted response to customer dissatisfaction and root cause of failure where crude oil is sealed. A more robust non-dynamic secondary seal cannot hang up and a non-dynamic secondary seal does not wear the stub sleeve. Site

surveys and equipment Root Cause Analysis, have shown that solids & debris contained in the pumped product result in secondary seal damage and ultimately seal failure. Degradation of this secondary seal subsequently wears the stub sleeve making refurbishment more costly.

Providing a more reliable secondary seal, especially as pumping stations that serve transmission and gathering pipe lines can be very remote and often unmanned, increases both pump and pipeline reliability. Reduced sleeve fretting maintains seal integrity, delivering improved spill prevention, and is valuable to operators in terms of assets protected as well as critical asset expansion.

This paper will look at the design theory of a non-collapsible flexible sealing membrane, the subsequent successful development and testing of a non-pusher elastomer seal, and field deployment.

INTRODUCTION

Crude oil pipeline pump applications are unique in the hydrocarbon processing industry, the application, equipment and operating environment is very different from the typical hydrocarbon process pump in a refinery or petrochemical plant. The crude oil pipeline network is huge, and continues to grow as old lines are upgraded and new capacity is added [1]. Figure 1.0 illustrates the market percentages of pipelines and their regional location.

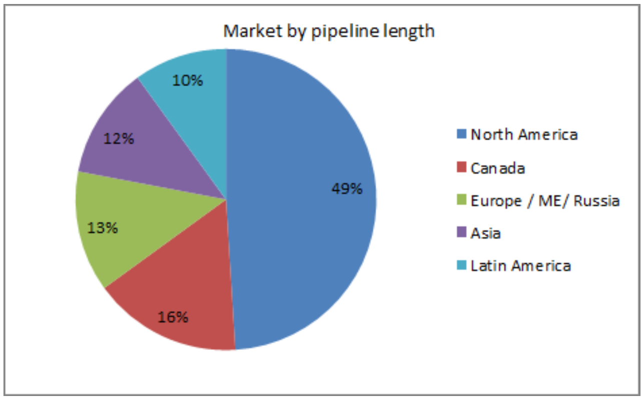

Figure 1 Regional Locations of Crude Oil Pipelines. [2]

The network of transmission lines across the world continues to grow involving an ever increasing population of pumping stations and fluid handling equipment. Pumping stations that serve transmission and gathering lines can be very remote and often unmanned.

There are operational and process differences in respect to how these assets are operated and managed, often as a result of the layout and location of these stations, as well as their position in the transmission network. Reliability problems in remote locations present a number of challenges, not the least of which is significant environmental impact. Remote locations add to the complexity and cost of repair and preventive maintenance. Therefore extending the MTBR of the sealing systems serving crude oil pipelines can represent savings to the operator that are orders of magnitude greater than in plant refinery systems.

A pipeline is made up of a series of pumping stations to maintain process flow. Many factors may dictate the spacing of these stations and the pump design / capacity. Generally pumping stations in a crude oil network will be about 50 miles apart and contain between 2 and 4 pumps in series. The pumps at each location form a pumping cycle, normally resulting in pumps within the same stations seeing suction pressures of around 100 psig (7 barg) and discharge of 500 psig (34.5 barg). Some pumps within the station seeing suction pressures as high as 900 psig (62 barg) and discharge pressures up to 1440 psig (100 barg).

Typically, though not exclusively, pipeline station design is symmetrical featuring equipment that is proven and preferred by the operating company. Redundancy and standardization is capacity driven and enables the company to continue transmission even in the event of a partial outage. This strategy is one that simplifies maintenance and while requirements may vary based on terrain, elevation and even seasonal differences, capacity impairment or limitations are preferred to cessation. Some stations will be provided with base load capacity to boost pipeline pressure and flow, and others will be designed to run in parallel to improve peek flows. Pump operational criteria will therefore be different station-to-station. With variables pipeline hydraulics, varying pump operating pressures, temperatures (also between summer and winter), variables speeds, and various 'upset' conditions. Pumps often have to run off from their BEP (Best Efficiency Point). Environmental regulations imposed on the operators in regard to leaks and spills are aggressive [3] so the provision of a reliable pumping solution is critical to an operators P&L. Lost production due to outages can quickly eclipse $150k / hour [4], and have non favorable market implications [5]. Seal repairs can have direct & indirect costs in excess of $1.3m per occurrence, excluding clean up or environmental costs that may be incurred [6].

Pipelines require seal solutions that deliver extended reliability in clean, dirty or mixed crudes, with a desire to run in excess of 9000 hours typical. Mechanical seals must have the ability to remain flexible and continue to move as designed in spite of liquid pumpage that presents significant solids content and changing viscosity. Both the primary sealing faces as well as the secondary sealing elements are required to perform without variation and ideally without failure. To meet the design requirements of the pipeline mechanical seals must be typically capable of operating at 1500 psig (100 barg) dynamic and 2200 psig (152 barg) static across a range of pump sizes diameters. The static duration can vary depending on shut down characteristics of the equipment and shut down procedures, whilst the characterization of leakage and the acceptable performance range varies by operating company. A typical large pipeline customer specifies "failure" criteria as approximately 10 drops per minute as reason to shut down equipment. These definitions and references are central to the seal design philosophy as well as the performance expectations assigned to a given design.

The paper will discuss the challenged associated with sealing crude oil applications, and will discuss the different seal technology used to seal crude oil pipeline applications. It will introduce a new technology being used today to solve a common problem experienced by users.

CURRENT TECHNOLOGIES FOR PIPELINE APPLICATIONS

Pipeline operating conditions vary greatly, but typical operating conditions for crude oil pipeline applications located in North America are:

- Fluid: Crude Oil
- Temperature vs. viscosities (from the Canadian border to the Gulf Coast)
  - -2.5 C @ 3169Cst
  - 15C @ 650Cst
  - 30C @ 215Cst
- Temperature: -4 to 400 F (-20 to 204 C)
- Size range: 2.625" to 6.130" (67-156mm)
- Speed: Up to 5000 fpm (25.4 m/s)
- Pump: Double-ended between bearings configuration
- Seal chamber pressure: Up to 1440 psig (100 barg) dynamic / 2200 psig (150 barg) static

Typical equipment used on crude oil pipelines.

Pumps used in crude oil pipeline service are almost always described and defined by the API 610 pump standard [7]. API 610 groups pumps types using a designation code. The following pump types are typically used in crude oil pipeline service:

- OH2 is one horizontal overhung impeller, centerline mounted and very common in refineries and offshore; sometimes called the "API 610 pump" or "process pump"
- OH3 is one vertical impeller, in-line mounted, very common in refineries and offshore
- BB1 is one impeller between bearings, axially split case, used in refineries and pipelines
- BB3 is multi-stage, impeller between bearings, axially split case, used in pipeline service
- BB5 is multi-stage, impeller between bearings, used offshore, called a "barrel pump"
- VS1 is a vertical, multi-stage pump, sometimes called a vertical turbine pump

Types BB1 and BB3 are the most common pump used in crude oil pipelines in North America.

Types of seal arrangements used in API pumps:

- Single seal – pusher liquid seal
- Dual unpressurized seals – pusher liquid seals (wet/dry/non-contacting containment seals)
- Dual pressurized seals – pusher liquid seals

Seal configurations vary globally, some markets prefer a single operating rotating pusher seal with some form of bushing or containment. All configurations are defined by API 682 seal configurations. [9].

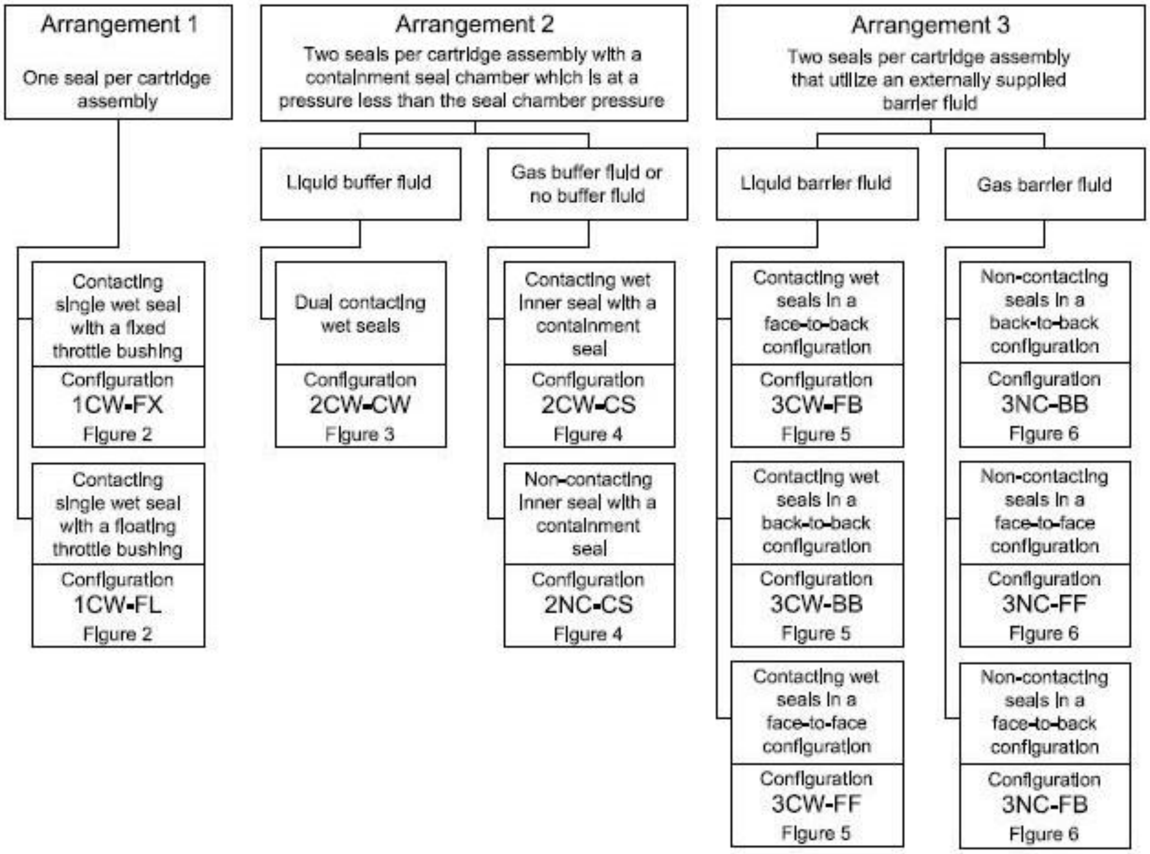

Chart 1: API 682 seal configurations

More elaborate single seals are often used, stationary mounted, linked to an API plan 66 or 66A dual bushing configuration. Both these single seals are easy to operate and require very little support providing they continue to run well. The seals used in this arrangement have been used successfully for many years. In most crude oil pipeline application, H2S is not a concern, but when it is present the generally accepted recommendation [8] is to use single seals when the H2S is below 500 PPM, single seals with a bushing between 500-1000 PPM, dual unpressurized seals between 1000-10,000 PPM, and dual pressurized seals when the H2S exceeds 10,000 PPM.

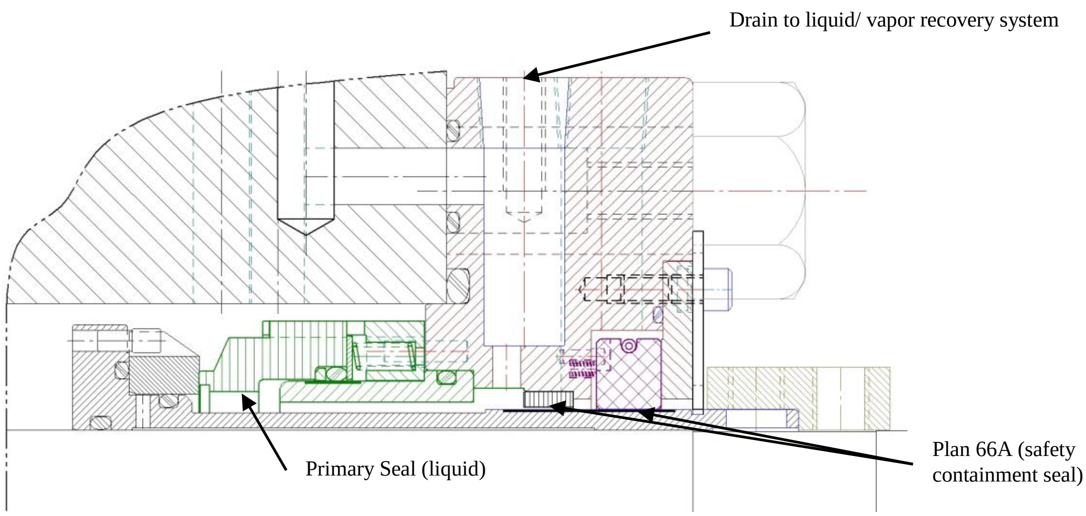

Figure 2.0 illustrates a typical single pusher seal with a safety containment bushing in a simple API plan 66A arrangement.

Dual unpressurised seals are often deployed, with either a liquid lubricated, dry running, or non-contacting containment seal outboard. Many pipeline operators prefer this offering as standard, [10]. Often where single seals are acceptable this configuration becomes popular when H2S is present.

Figure 3.0 illustrates a typical dual unpressurized dual pusher seal with a liquid safety containment seal outboard. Figure 4.0 illustrates a typical unpressurized dual pusher seal with a dry running safety containment seal outboard and Figure 5.0 illustrates the same seal with a non-contacting seal outboard

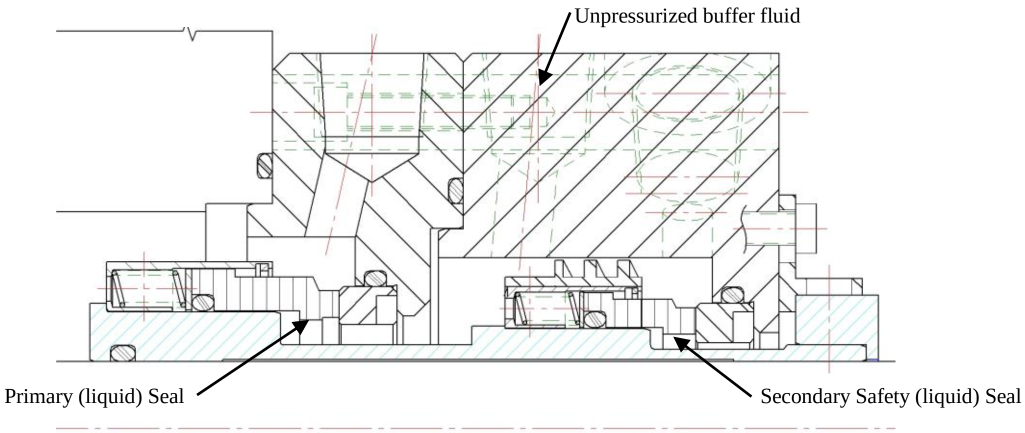

Figure 3 Typical dual unpressurized pusher seal arrangement using a liquid safety containment seal outboard

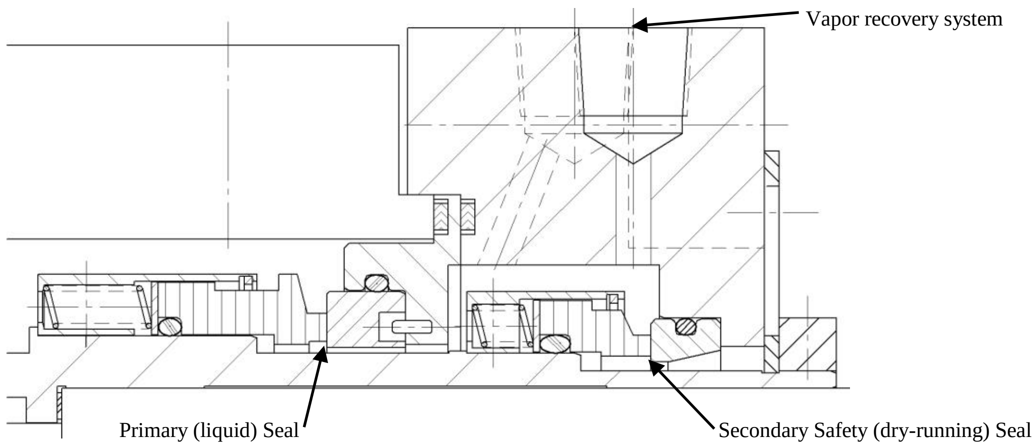

Figure 4 Typical dual unpressurized pusher seal arrangement using a dry-running safety containment seal outboard

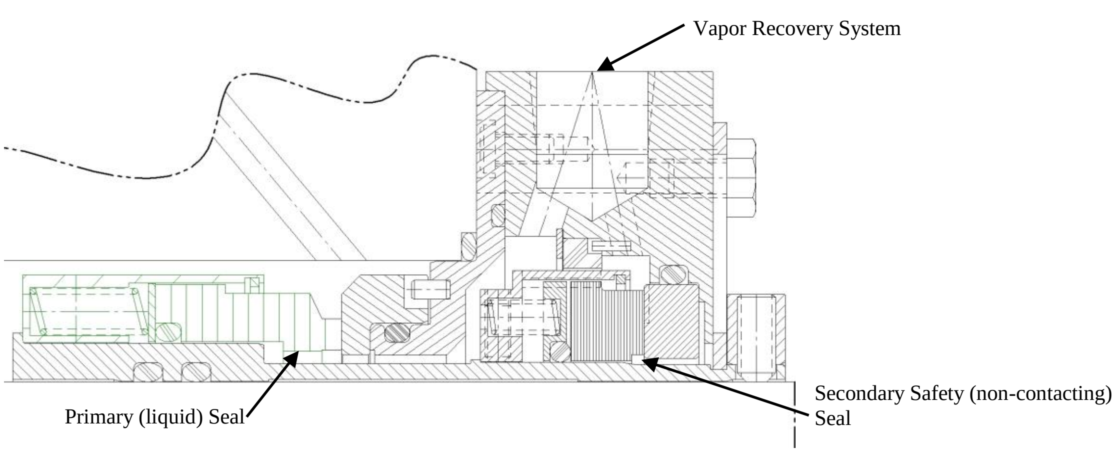

Figure 5 Typical dual unpressurized pusher seal arrangement using a non-contacting safety containment seal outboard

The most comprehensive solution would be the dual pressurized seal, two seals operating on a clean barrier fluid pressurized and supported by an elaborate API piping plan 53 A/B/C or Plan 54 configuration [9]. Often considered the ultimate in leakage containment and used with high concentrations of H2S or sever process contaminations that do not allow for good primary seal operation. Figure 6.0 illustrates a typical dual pressurized pusher seal.

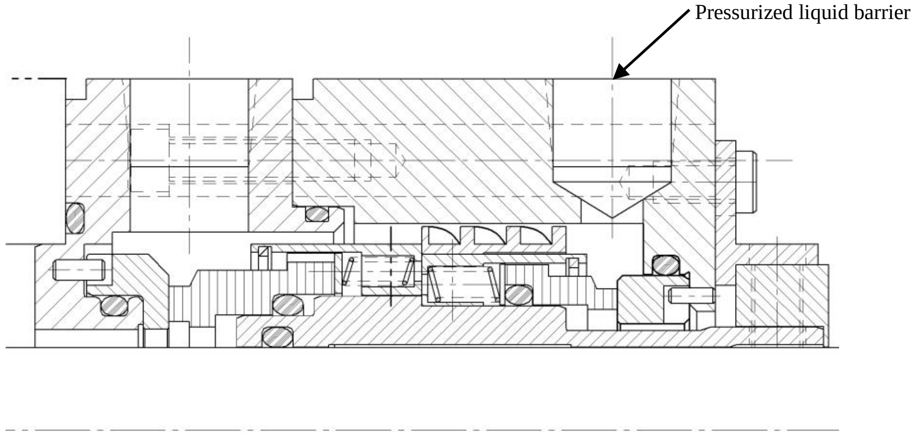

Figure 6 Typical dual pressurized pusher seal arrangement

Typical material of construction for pipeline seal running faces are Self-Sintered Silicon Carbide versus Reaction Bonded Silicon Carbide to provide optimum hard wearing interface characteristics, resist abrasion from the solids in the process stream and to handle the extreme pipeline pressures. Elastomers are typically 90 durometer Viton and 90 durometer Aflas [11] for the secondary seals, to provide rigidity under high operating pressures but still retain elastomeric characteristics. Unless corrosive contaminants then the metal hardware primarily consists of 316SS.

Most of the seals discussed above use API piping plans 11/66A to supply a flush to the seals while providing a safety containment device outboard. Where pipelines are using dual seals, they will use API piping plan 53 A/B/C or plan 54.

Challenges of Pipeline Applications

Pumping stations are often in remote locations, limiting their access and limiting the ability to incorporate support systems that require regular maintenance, such as the recharging barrier or buffer systems. These logistical and maintenance concerns have dictated that most of the seal arrangements used in North America are single seals with a bushing or dual unpressurized seals, incorporating a dry running secondary containment seal. These arrangements eliminate or greatly reduce the complexity of the seal support systems required. An API flush plan 11 (circulation of pumped fluid from a higher pressure (usually the discharge line)), is used to flush the primary ring / seal interface with product while the bushing or dry running secondary seal is connected to a drain recovery system. The drain collects any leakage past the primary ring and directs it to a holding tank or a recirculation system (sump pump). The holding tank or the discharge from the sump pump is piped to the suction of the pump to be reintroduced into the process stream. In some cases, a storage tank is used to collect the discharge before returning it to the suction. The drain must be sized properly to allow a large volume of fluid to pass through without creating too high of a back pressure, thus putting the primary seal under reverse pressure. A drain connection would typically be in the range 0.375” to 0.625” Dia, (9.5 to 16mm), [12] but will be sized to meet the expected leakage from the arrangement. A connection that is too small for the viscosity of the pumped fluid will clog quickly and fail to perform its function.

One of the more difficult challenges facing end users in the crude oil transportation market is the fretting of the O-ring secondary seal due to shuttling of pumps during operations. Shaft shuttling can occur when single stage double suction between bearings pumps (BB1) are utilized, the balanced rotor is sensitive to hydraulic upset and more likely to shuttle when an upset occurs. An unstable axially load condition creates a cyclic side impact load on the thrust bearings and may diminish life expectancy.

Whilst this shuttling of the rotor can lead to premature bearing damage if the severity of this movement is not taken into consideration at the design stage, this movement will reduce traditional mechanical seal performance and reduce life expectancy. Eventually leading to higher leakage and premature failure.

It has been reported by many pipeline operators of O-ring pusher seals that the main cause of seal failures they experience has been the failure of the secondary seal due to fretting [13]. Fretting occurs when the O-ring rubs against the metal sleeve, the O-ring begins to wear, no longer able to slide / roll as designed and hang-up occurs. This hang up manifests in 2 ways, i/ it causes the seal to over load the faces (high closing forces) or ii/ cause the seal to open during operation increasing leakage. In the event of over loading the seal faces (primary ring and mating ring interface), the seal will quickly lose lubrication. This results in high contact loads, high wear, increased temperatures, and dramatic reduction in seal life. If the faces remain open, a high and unstable fluid film between the seal faces will develop, resulting in higher than normal leakage to the atmosphere and high volumes of liquid filling the recovery system. Figure 7.0 highlights the O-ring (secondary seal) in a typical single pusher seal.

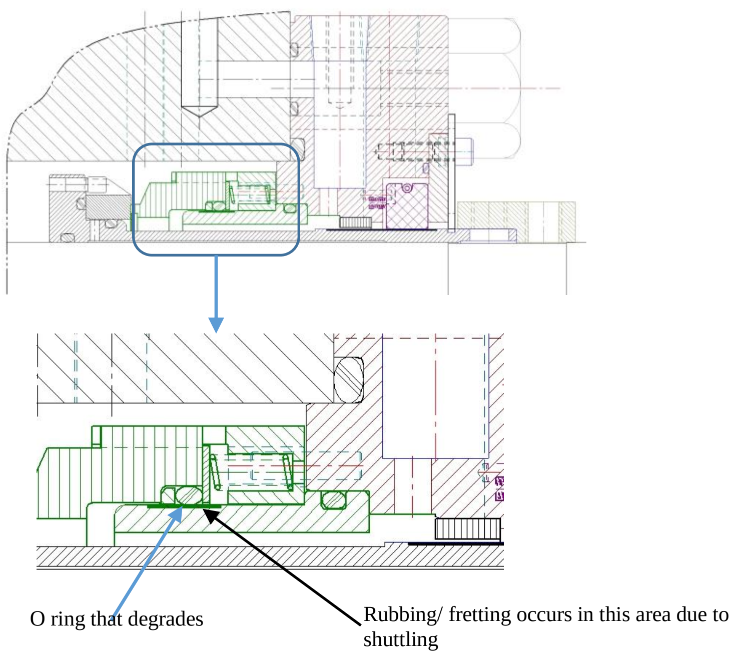

Figure 7 The O-ring (secondary seal) in a typical single pusher seal

The abrasion or fretting caused to the secondary seal as a result of pump shuttling is the root cause of accelerated seal failure in a high percentage of pusher seals deployed in BB style pumps on crude pipeline applications. [13].

By definition, a pusher seal is a seal that moves forward (with the O-ring) as the seal faces wear. In normal operation, the O-ring will push forward being forced by the spring load and hydraulic forces. In situations where the pump is shuttling, the shaft will move axially inboard and outboard, causing the seal to follow. This shuttling causes the O-ring to move across the metal sleeve. Most sleeves under the O-ring contact area are coated with a hard material such as Tungsten Carbide (polished) to allow the O-ring to freely slide back-and-forth. In crude oil applications, the product collects around the O-ring and will cause the O-ring to stick or it will leave dry crude oil remains on the surface in which the O-ring will slide. Over time this wearing action will damage the O-ring and cause a seal failure. Figure 8.0 illustrates an O-ring from a seal that failed prematurely due to O-ring fretting caused by pump shuttling.

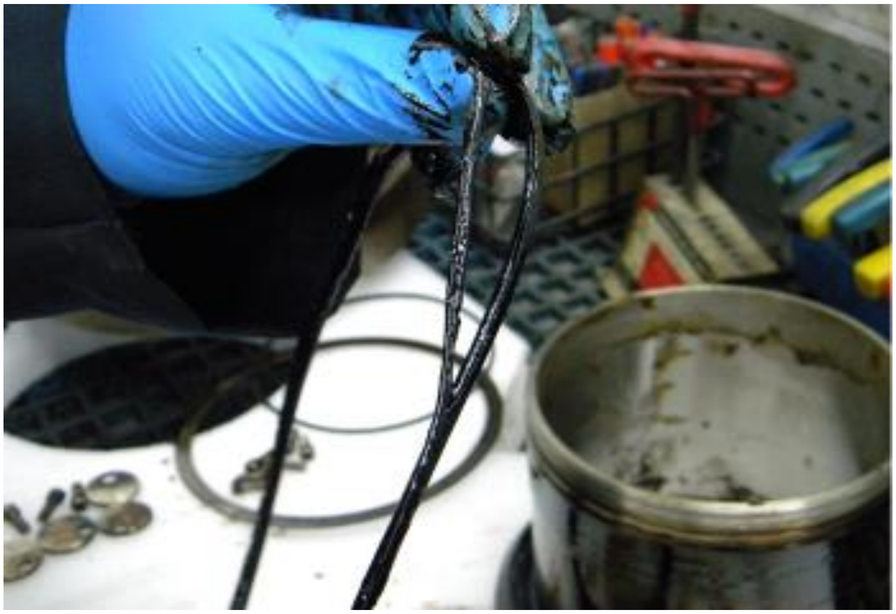

Figure 8 A typical O-ring failure due to fretting caused by pump shuttling

Removal of the O-ring and replacing it with another pusher type secondary seal (PTFE wedge, spring energized seal) would result in similar failures. To eliminate failures due to shuttling or axial movement during operation, a non-pusher seal configuration was development incorporating a special elastomer seal. Conventional bellows have included a convolution that can expand to accommodate axial motion of the axially shiftable seal ring relative to the sleeve.

Figure 9.0 illustrates a conventional full convolution (traditional) style elastomer bellows seal. The technology used in full convolution bellows has been used in mechanical seals for over 50 years. Recent studies have shown under the high pressures of crude oil pipeline conditions, the bellows will collapse. A collapsed bellows can become axially rigid and unable to accommodate axial motion of the axially shiftable seal ring. This axial rigidity can cause the closing force to increase, eventually leading to seal burn out and failure. Figure 10.0 illustrates the effects of high pressure on an elastomer bellows.

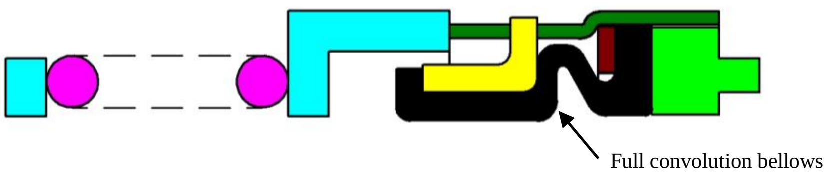

Figure 9 Conventional full convolution elastomer bellows (bellow highlighted)

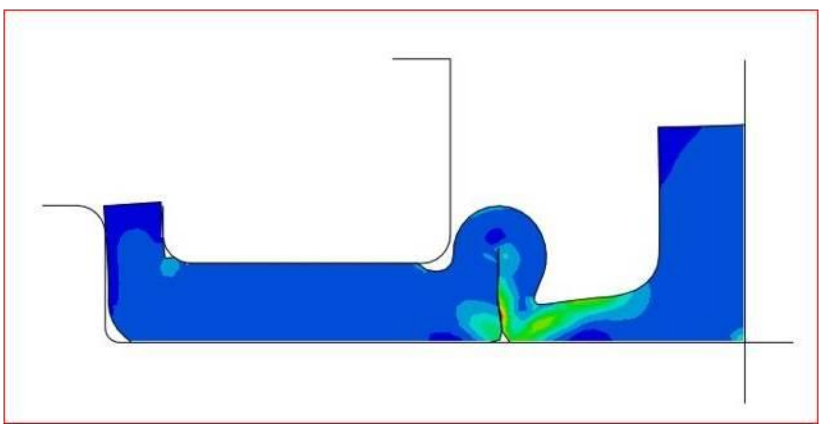

Figure 10 A collapsed elastomer bellows under high pressures

Figure 11.0 shows a single non-pusher single seal using a unique secondary sealing technology to prevent hang up. This design will extend reliability in typical high pressure crude oil application where short seal life and excessive leakage are prevalent. The unique feature is a non-pusher secondary seal elastomeric element that stretches and compresses to take up shaft/axial movement or wear. The sealing faces of a non-pusher seal tracks each other while in operation, but the secondary seal remains in place. This eliminates the wear or fretting caused by the constant shuttling or movement of the pump. Replacing a bellows that has a convolution with a flexible sealing membrane (non-pusher secondary seal), that lacks a full convolution, eliminates bellows collapse and axial rigidity. As shown in Figure 11.0 the flexible sealing membrane includes a flange portion, a coaxial portion, and a connecting portion extending between the flange portion and the coaxial portion. Each of the flange portion, connecting portion, and coaxial portion lie against a portion of the axially shiftable primary seal ring, an anti-extrusion ring, or a stub sleeve and in this respect are supported against collapse due to high process chamber pressures.

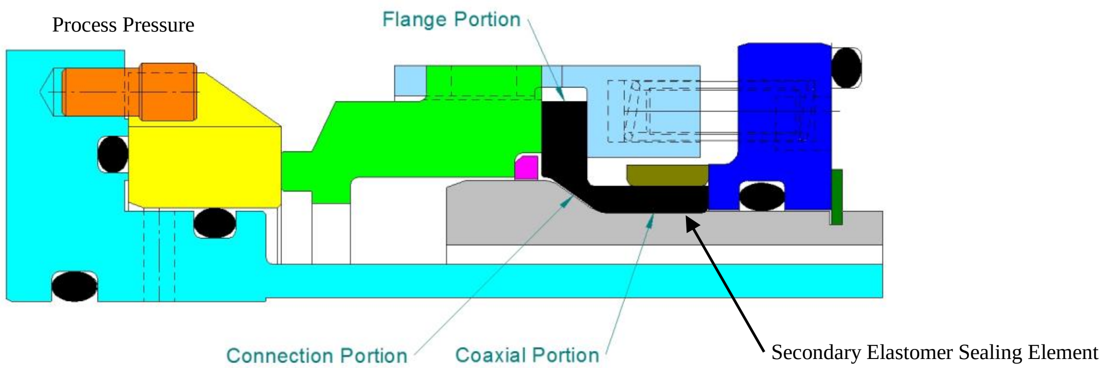

Figure 11 The new flexible elastomer sealing element incorporated in a single non-pusher seal

The flexible connection portion flexes in response to axial movement of the axially shiftable primary seal ring and the flange portion. This occurs without compressing the axial portion of the annular flexible sealing membrane and thus without altering the closing force applied to the axially shiftable primary seal ring by the flange portion.

Moreover, an anti-extrusion ring is positioned within a groove of the axially shiftable primary seal ring and adjacent to the stub sleeve to prevent extrusion of the annular flexible sealing member. Not allowing the face loading (closing force) to be altered during changes in pressure, allows for stable seal performance even in the extreme conditions of crude oil pipeline applications. Figure 12.0 shows the elastomeric flexible secondary seal in a single non-pusher single seal with a segmental bushing as the secondary containment seal.

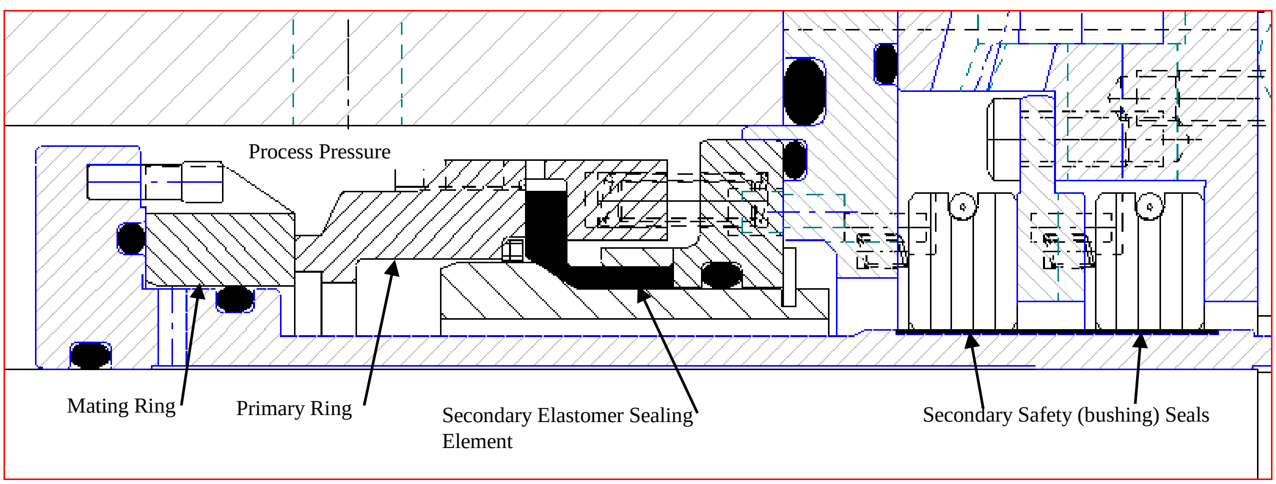

Figure 12 A typical non-pusher single seal incorporating the flexible elastomeric secondary seal as it is installed in service

This secondary elastomeric sealing element provides a reliable seal in the presence of the fluids containing high percentages and/or high hardness particles where O-rings have failed. With a high number of starts/stops and with pressure fluctuations that occur on pipelines, it is not uncommon to experience pressure spikes within the system.

Figure 13.0 shows a chart of typical pressure spikes a pump in a crude oil service could experience, [13]. This coupled with the fluids solids content present in the crude oil, and the high possibility of axial shuttling, previously detailed, can cause the contact area of a dynamic o-ring on a metal surface to abrade and degrade very quickly.

The new elastomer secondary sealing element shown in Figure 11.0 can aid in totally eliminating the abrasive damage and hang up, while providing a "dampen" against axial cyclic loading of the sealing faces.

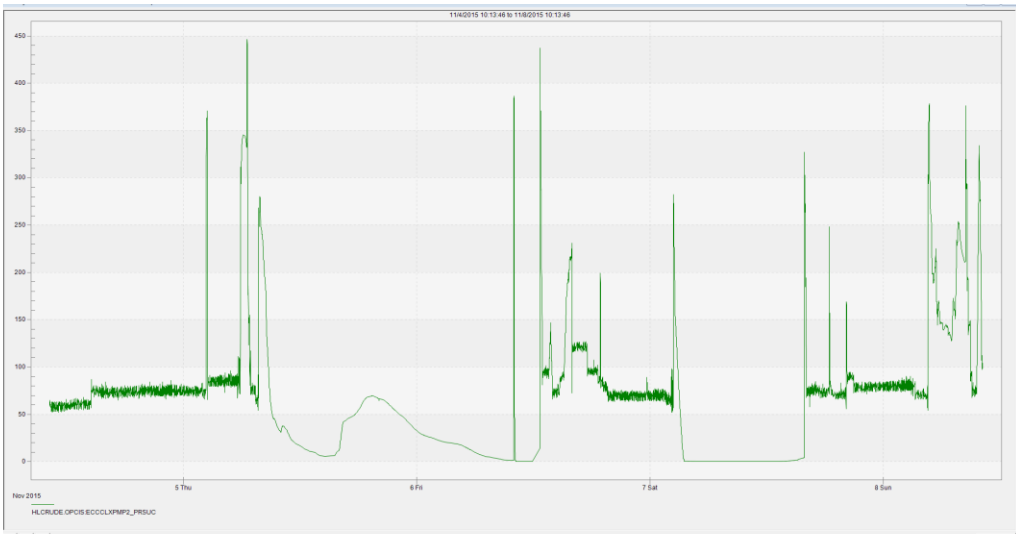

Figure 13 Pressure spikes normally experienced on crude oil pipeline rotating equipment

Internal Development Testing

Over 6000 hours of testing has been complete on this new seal design; three shaft sizes. 2.625", 5.510" and 6.130" (67mm, 140mm, and 156mm) were chosen. Test full test conditions were established in conjunction with a significant pipeline operator based on operating conditions at their pumping stations.

Dynamic testing included seal pressures from 50 to 2200 psig (3.5 barg to 152 barg) during dynamic testing and a static test up to 3300 psig (228 barg). Rotational velocities up to 5000 fpm. Over 500 start/stop tests were completed at full pressure without any negative effects to the seal's performance, and no change in the leakage from the first start to the last start. The seal was also dynamically tested with the seal head offset and at maximum pressure (1440 psig/100 bag). During the offset testing, the seal was axially mis-aligned by 0.031" (0.8 mm) in positive and negative axial direction. Again, no change in seal performance (leakage) was observed.

The fluid used as the process was Royal Purple 910, a high viscosity oil recommended by end users that possesses many of the thermal characteristics of a typical crude oil. Testing with such a viscous fluid had its challenges, such as keeping the system cool enough so not to change the properties of the fluid (reduced viscosity). A chiller and heat exchanger were used to keep the inlet temperature of the test fluid at ambient conditions. The non-pusher secondary seal underwent a cyclic test of more than 4.25 million cycles of +/- 0.031" (0.8mm) movement from nominal to check for fatigue on the elastomer and no fatigue or degradation was found.

Figure 14.0 shows the viscosity chart for RP 910. Figures 15.0 and 16.0 shows the test seal, with the leakage collection setup. Leakage never exceeded 35 ml/hour. Figure 17.0 shows the cyclic test rig.

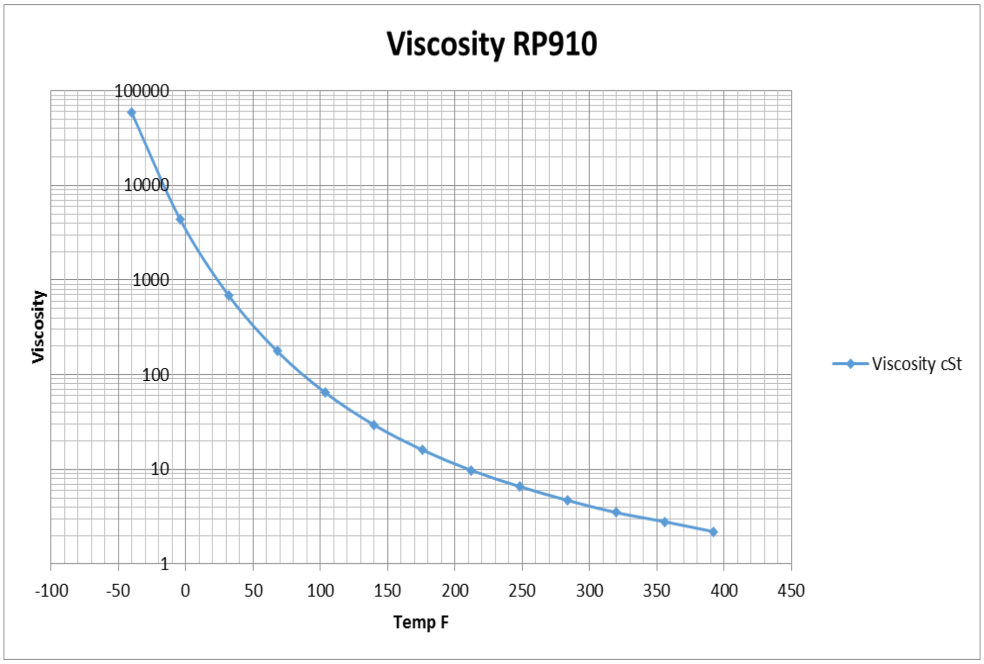

Figure 14 Viscosity chart for RP 910

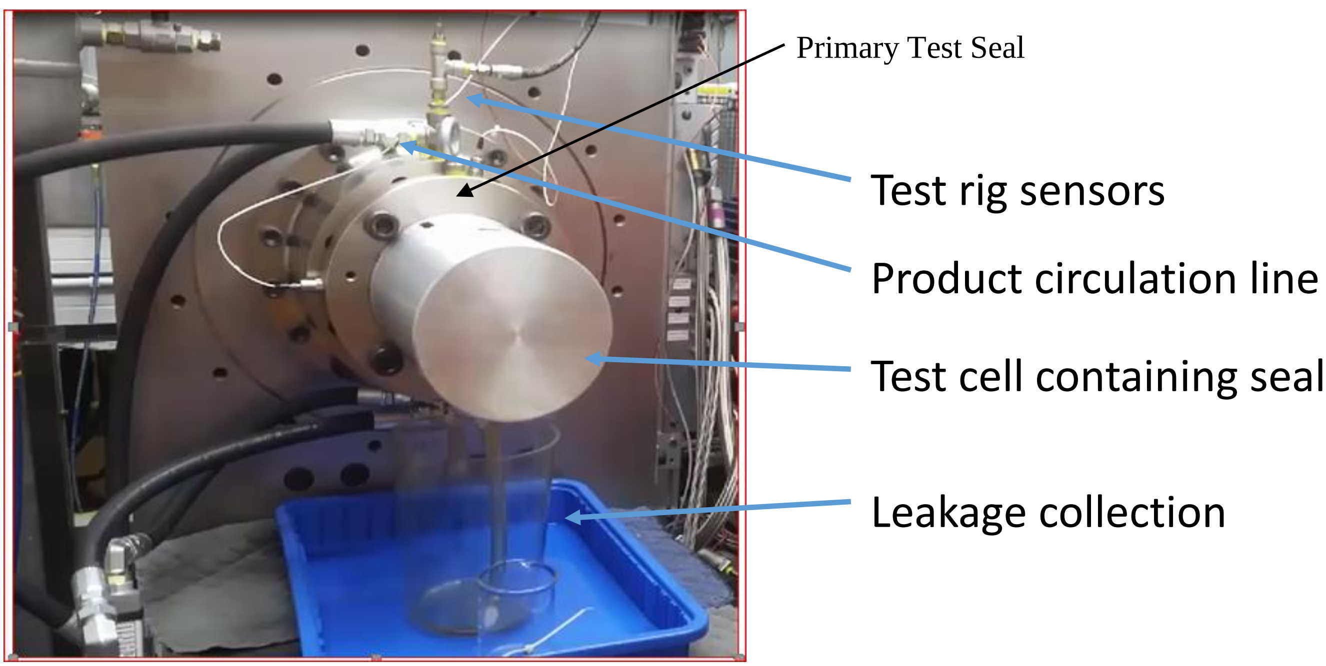

Figure 15 Test Rig

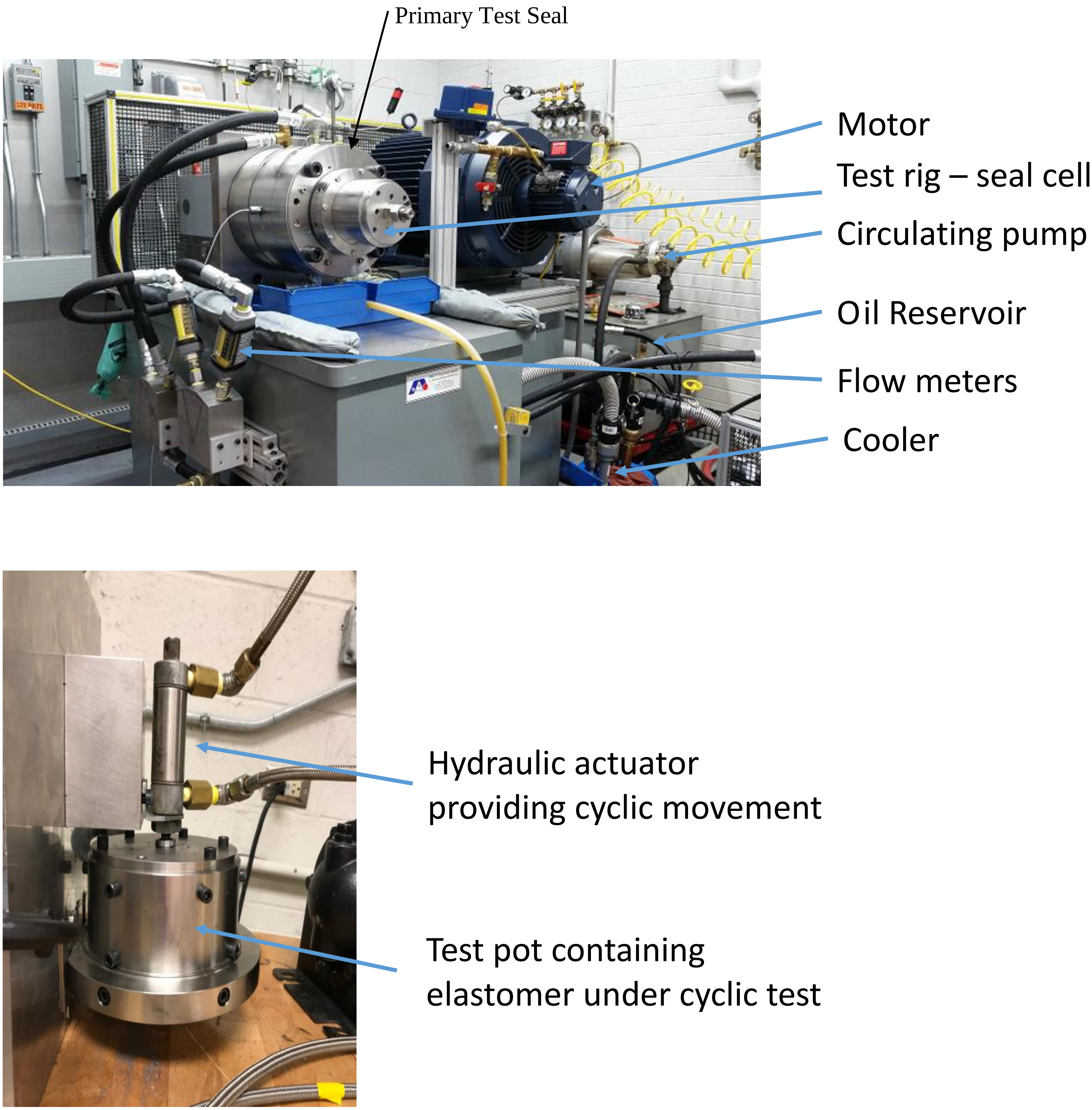

Figure 17 Cyclic Test Rig

Field Testing
At the time this paper was submitted, about 12 seals are installed at several locations within North America. The first installation being in November 2017. This is the largest seal installed, being a 5.510” (140mm) shaft. At one location, the seals were installed

onto pumps that were fitted with monitoring equipment to record any shaft shuttling. Those pumps were recently retrofitted with new bearings to eliminate shuttling, but recent failures still showed signs of O-ring fretting on the old pusher type seal causing the user to question the success of the bearing upgrades.

In all installations, no leakage was detected in the recovery system. Pressure range on the seals installed was 35 psig to 1198 psig (2.4 to 83 barg), and temperature ranges from 0 F to 110 F (-18 to 44 C). Speeds were from 720 rpm to 3580 rpm. Several new installations are scheduled throughout the US and Canada using the new elastomer secondary sealing element.

CONCLUSIONS

The use of non-pusher sealing technology to enhance the reliability of crude oil pipeline applications represents a novel approach to the pipeline industry. Seal designs that employ a non-pusher secondary sealing element have proven effective for decades in challenging material mining and high performance chemical applications where solids and high viscosity fluids are encountered.

The design philosophy replaces the single O-ring sealing device in favor of a semi-static non-pusher secondary seal. The solids rich environment and dynamic requirements faced by the single O-ring seal is managed far better by a large non-pusher secondary seal that only flexes where an O-ring was required to 'roll'. The non-pusher secondary seal continues to seal through hydraulic changes and dynamic demands that would quickly damage an O-ring. The secondary seal device used in this design presents a shape and cross section that is many times the mass and volume of a single O-ring, this enables the non-pusher secondary seal to maintain a high integrity seal through environments that would destroy a single O-ring. Ultimately, the reliability of the sealing system is enhanced many fold as a result of these design changes.

FIGURES

Figure 1**-** Regional Locations of Crude Oil Pipelines
Figure 2 - Typical single pusher seal with a safety containment bushing in a 66A arrangement
Figure 3 - Typical dual unpressurized pusher seal arrangement using a liquid safety containment seal outboard
Figure 4 – Typical dual unpressurized pusher seal arrangement using a liquid safety containment seal outboard
Figure 5 – Typical dual unpressurized pusher seal arrangement using a non-contacting safety containment seal outboard
Figure 6 – Typical dual pressurized pusher seal arrangement
Figure 7 – The O-ring (secondary seal) in a typical single pusher seal
Figure 8 – A typical O-ring failure due to fretting caused by pump shuttling
Figure 9 – Conventional full convolution elastomer bellows
Figure 10 – A collapsed elastomer bellows under high pressures
Figure 11 – The new flexible elastomer sealing element incorporated in a single non-pusher seal
Figure 12 – A typical non-pusher single seal incorporating the flexible elastomeric secondary seal as it is installed in service

Figure 13 – Pressure spikes normally experienced on crude oil pipeline rotating equipment
Figure 14 – Viscosity chart for RP 910
Figure 15 – Test Rig
Figure 16 – Test Rig
Figure 17 – Cyclic Test Rig
Figure 18 – Leakage testing

REFERNCES

[1] World Pipelines November 2017 page 12
[2] Collection of data from www.theodra.com December 2017
[3] https://www.reuters.com/article/us-bp-spill-penalty/bp-to-pay-25-million-penalty-over-alaska-oil-spill-idUSTRE7424DR20110503
[4] Customer interviews John Crane
[5] https://www.ft.com/content/24f8024a-de92-11e7-8f9f-de1c2175f5ce
[6] Operator interviews with John Crane
[7] American Petroleum Institute standard 610, 11th edition 2010
[8] EagleBurgmann Mechanical seal technology and selection. DMS_TSE / E5 / PDF / 04.17 / 9.7 c EagleBurgmann Group Marketing, Germany
[9] ADVANCEMENTS IN MECHANICAL SEALING - API 682 FOURTH EDITION. 43rd Turbomachinery & 30th Pump Users Symposia (Pump & Turbo 2014) September 23-25, 2014| Houston, TX.
Michael B. Huebner, Gordon S. Buck, Henri V. Azibert

[10] John Crane Market research
[11] AFLAS® is the tradename for a unique fluoroelastomer based upon an alternating copolymer of tetrafluoroethylene and propylene (TFE/P).
[12] Review of several 100 John Crane installed applications in crude pipeline pumps
[13] JC information collected in site surveys 2013-2017

BIBLOGRAPHY

(1) Muijderman, E. A., Spiral Groove Bearings, Springer-Verlag, New York, 1966.
(2) Schoenherr, Karl 'Fundamentals of Mechanical Seals" reprinted from Iron and Steel Engineer, reprinted in Engineered Fluid Sealing, John Crane UK 1976
(4) Buck, G S Selection and Design of End Face Mechanical Seals for Common Refinery Services, Louisiana State University, 1978.
(5) Rodgers, P R, 'The development of profiled mechanical seal faces for positive liquid transfer', 1lth international Conference of the British Pump Manufacturers' Association Churchill College, Cambridge18-20 April, 1989
(6) Morton, J L, Evans, J G, Developments in high performance seal designs for critical high pressure offshore & pipeline applications. 20th International pump users symposium, Houston Tx, March 2003
(7) Evans, J G, Janssen, S, Developments in sealing technology within multiphase pumping. 19th International pump user's symposium, Houston Tx, February 2002
(8) 'Quantifying Reliability Improvements through effective pump, seal and coupling management' – Process pump reliability conference, Texas 1999
(9) American Petroleum Institute, Pumps – Shaft sealing systems for centrifugal and rotary pumps. API standard 682 4th edition, July 2014
(10) Pehl, A; Szwarz M; Satish H; Vanhie E. Developments of a single mechanical seal equipped with API piping plan 11/66A for large mainline crude oil pipeline pumps. 46th Turbomachinery & 33rd Pump Symposia Houston December 2107