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RULE BASED SEAL SELECTION FOR PIPELINE SERVICES

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ABSTRACT

Mechanical seal selection for pipeline applications can be a challenging task considering the application pressure ranges, speeds, and the properties of the fluids being moved. Variable fluid properties present their own challenges for operators and mechanical seal designs. Equipment operational challenges are also at the forefront and need to be addressed by the mechanical seal. Because of these challenges, there is an ever present and even increased emphasis on the criticality of selecting a robust mechanical seal design and associated support equipment to effectively manage the application demands.

Pipeline operators require reliable sealing systems that can handle the range of fluids and conditions while providing a leakage management and safety containment system that meets their local and federal regulations. Selecting the proper seal goes beyond picking one that can handle the pressure, temperature, and speed. Many other factors need to be considered to properly select the precise seal including materials, support system, and leakage management systems to accord with operator's specifications. The equipment itself may even impose limitations and restrictions to viable seal configurations that warrant consideration during the selection process. The impact of these complexities can be magnified through operational procedures (start-up, shutdown, and post upset or failure); all of which can greatly influence the selection decision.

The completion of an appropriate mechanical seal selection can be a daunting task for the user reviewing all potential influencing application factors. The process complexity is compounded when weighing the various advantages and disadvantages of numerous potential mechanical seal configurations and support system arrangements potentially suitable for the operating parameters.

This tutorial will identify the intricacies of selecting mechanical seals for varied hydrocarbon pipeline applications and will suggest a rule-based methodology for simplifying the seal and support system selection. The selection method will consider the pumped fluid variables, application, and operational regimes, as well as the legislative and non-legislative requirements for containment strategies and leakage management. The types of seal and support system solutions available by fluid type will be discussed, considering various local user or national preferences. The paper will identify the core attributes that affect seal selection and expand on the benefits and advantages on each solution.

INTRODUCTION

To be effective and acceptable pipeline sealing solutions must have:
- A solution that delivers extended reliability in all products
- Ability to resist hang up and wear of the secondary seal (typically an o-ring)
- Ability to work with marginal interface lubrication
- Ability to withstand contaminates and abrasives
- Pressure capability from around 2 bar (30 psi) to 100 bar (1450 psig) dynamic and 150 bar (2200 psig) static are the baseline across the size range for majority of pipeline applications. Operating pressures around 70 bar (1000 psig) dynamic is typical but pumping stations must be flexible in their ability to handle pressure.
- The secondary containment device should be suitable to effectively divert and contain most leakage to drain or vapor recovery system at full dynamic pressure and ensure there is no leakage to atmosphere.
- While the leakage definition / acceptability will vary by customer, province, region, or state it this the responsibility of the operator and the supplier to implement an acceptable leakage containment strategy. Therefore, design criteria need to reflect this as a typical shut down requirements and be flexible enough to deal with individual requirements or circumstances.

This paper will provide a methodology for providing best in class selection for the equipment user and define effective and efficient sealing strategies in all major midstream pipeline applications. It will consider the debates between single and dual seals, seal arrangements, secondary seal types and effectiveness of containment devices and seal support systems. This paper will discuss integrating relevant industry wide best practices and lessons learned from field installations combined with previous experience of the authors in addressing these applications. The authors would also like to bring attention to the use of API682 as defining performance requirements for pipeline duties when the size / pressures and performance needs are currently out of scope of 4th edition. It is anticipated that the next edition of API 682 (5^{th} Edition) will include coverage for mechanical seals in pipeline services.

PRESSURE RATINGS APPLIED TO PIPELINE SEALS

Pipelines are designed within certain operating pressure limits as necessary to transport products over long distances without an excessive number of booster pump stations. The limiting factor for many pipelines is the pressure rating of the pipe and fittings. As a practical matter, the required pressure ratings for pumps, seals and systems to be used in pipeline service are often based on pipe flange ratings:

- Pipelines using Class 300 flanges have a 720 psig rating
- Pipelines using Class 600 flanges have a 1440 psig rating
- Pipelines using Class 900 flanges have a 2160 psig rating

Although API 682 does not apply to all mechanical seals, it is useful to note that it defines terms that relate to the mechanical seal:

> *Maximum static sealing pressure (MSSP)* is "The highest pressure, excluding pressures encountered during hydrostatic testing, to which the seal (or seals) can be subjected while the pump is shut down."
>
> *Maximum dynamic sealing pressure (MDSP)* is "The highest pressure expected at the seal (or seals) during any specified operating condition and during start-up and shutdown. In determining this pressure, consideration should be given to the maximum suction pressure, the flush pressure, and the effect of clearance changes with the pump."
>
> *Maximum allowable working pressure (MAWP)* is "The greatest discharge pressure at the specified pumping temperature for which the pump casing is designed."
>
> *Static Sealing Pressure Rating (SSPR)* is *"*The highest pressure that the seal can continuously withstand at the pumping temperature while the shaft is not rotating. Thereafter, the seal must maintain its dynamic sealing pressure rating.
>
> *Dynamic Sealing Pressure Rating (DSPR)* is *"*The highest pressure that the seal can continuously withstand at the pumping temperature while the shaft is rotating. Thereafter, the seal must maintain its static sealing pressure rating.

The static and dynamic sealing pressure ratings of the mechanical seal should be greater than or equal to the Maximum Static Sealing Pressure (MSSP) and Maximum Dynamic Sealing Pressure (MDSP) as stated on the mechanical seal datasheet. These are process conditions that must be defined by the user.

When working with seal manufacturers on engineering a solution for a certain service, the manufacture will need to know the maximum pressure the seal will experience. It is easy to quote maximum line pressures, but that can result in added costs with limited benefits. Support systems will be designed to handle those pressures, when in fact they may never see full line pressure. Some seals are not designed to run at low pressure and speeds when they were designed for full line pressure.

MATERIAL CONSIDERATIONS

Within the context of seals used in midstream pipeline applications, this section will cover some common materials used and why specific materials are selected based on the unique application conditions of these services.

Face Materials

- Carbon-Graphite – To enhance the properties of carbon-graphite grades, they are typically impregnated with various substances to achieve the required chemical and physical properties. These adders are typically resins, ceramics, and metals. Metal impregnated carbons offer the highest strength and antimony is most used. Antimony impregnated carbon is the default carbon identified in API 682 when considering use in a light hydrocarbon service. Although antimony filled carbon-graphite primary rings can produce the best performance and lowest leakage, the relatively soft nature of the material in abrasive or viscous services can present limitations. In practice, two hard faces are usually necessary to prevent mechanical and/or abrasive face damage in crude oil service. Additionally, in pressures beyond 1200 PSIG (82 BARG). Evaluation of the carbon material is required to assure sufficient margin of safety against mechanical failure of the material and deficiencies in the seal performance from excessive distortion due to pressure.
- Sintered / Reaction-Bonded Silicon Carbide - Silicon carbide is an advanced ceramic material. Silicon carbide is extremely hard, being highly wear resistant and with good mechanical properties. It has high temperature strength and thermal shock resistance, maintaining its high mechanical strength at temperatures as high as 2550°F (1400°C). Above 2570°F (1410°C) the free silicon melts and strength decays. These advanced ceramics are routinely used in midstream applications as typically the mating face pair with a carbon ring in light hydrocarbon or finished products. The ceramic materials are typically run against one another (dissimilar grades) in high viscosity applications such as crude oil. Robust primary ring drive designs are recommended when using Silicon Carbide faces to avoid potential hang-up when contacting the comparatively softer metal components of the seal retainer.
- Tungsten Carbide - Cemented tungsten carbides are derived from a high percentage of tungsten carbide particles bonded together by a ductile metal. The common binders used for seal rings are nickel and cobalt. Tungsten carbide is an extremely tough material with good wear resistance, with Nickel bound being the most common material used in midstream pipeline applications. Seal rings in this material offer improved protection against mechanical or thermal shock but are more susceptible to heat checking damage when compared against advanced ceramics.
- Silicon Carbide / Graphite Composites - These are sintered silicon carbide composite containing free graphite. The free graphite reduces friction, improving dry run survivability and better thermal shock resistance than conventional sintered materials. Seal rings manufactured from graphite / silicon carbide composite materials are typically used in crude oil or finished products and light hydrocarbon service especially when operating pressures may exceed the limits of conventional metal-filled carbon grades.
- Diamond Treatment – A relatively new face treatment in the sealing industry. There are various versions of Diamond face treatments available depending on the seal manufacturer. Diamond face treatment may help with intermittent dry running and times of reduced face lubrication as well as dealing with abrasive particulate which is sometimes found in the pipeline applications. It may help extend the seal life in certain applications due to the hardness of the face treatment and the low coefficient of friction which is less than carbon. It may be applied to one or both seal faces depending on the manufacture's requirements. Diamond treated faces cannot be repaired and must be replaced after use.

Secondary Sealing Elements

Specific elastomer usage varies in midstream pipeline applications depending on the process fluid and potential contaminants. Outside of material compatibility with the process fluid, for pressures above 1440 psig, a 90-durometer compound is recommended to prevent extrusion damage from pressure. In extreme cases, specialized back-up rings with an elastomer or energized polymer seals are utilized to accommodate pressures up to and beyond 2160 PSIG (149 BARG)

One of the more difficult challenges facing end users in the crude oil transportation market is the fretting wear of the dynamic O-ring secondary seal due to shuttling of pumps during operations. Shaft shuttling may occur when single stage double suction between bearing pumps are utilized. The balanced rotor is sensitive to hydraulic upset and more likely to shuttle when an upset occurs and during frequent starts and stops. An unstable axially load condition creates a cyclic side impact load on the thrust bearings and can cause a diminishing life expectancy. Shuttling of the rotor will damage bearings and reduce the mechanical seal performance. Eventually leading to higher leakage and failure, along with the bearings. It has been reported by many users of O-ring pusher seals that the main cause of seal failures has been the failure of the secondary seal due to fretting. Fretting occurs when the O-ring rubs against the metal sleeve which may incorporate a coating. The O-ring and sleeve begin to wear and may cause hang-up, which can cause the seal to overload the faces (high closing forces) or cause the seal to stick open during operation.

Most sleeves under the O-ring are coated with a hard material such as Tungsten Carbide (polished) to allow the O-ring to freely slide back-and-forth and reduce abrasive particulate wear. In crude oil applications, the crude oil and particulate 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. The hard coating on the sleeve has not shown to eliminate the fretting: it only prolongs the time to failure.

To eliminate failures due to shuttling or axial movement during operation, a non-pusher seal with a special elastomer seal can be used in these services. Figure 1 shows a single non-pusher single seal using a unique secondary sealing technology to prevent hang up and extend reliability in typical high-pressure crude oil applications where short seal life and 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.

The flexible connection portion flexes in response to axial movement of the axially shuttling 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 shuttling primary seal ring by the flange portion. 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.

This secondary elastomeric sealing element provides a more robust and reliable seal in the presence of the fluids containing high percentages and/or high hardness particles where O-rings have failed. With equipment frequent starts/stops, variable frequency drive, as well as pressure fluctuations that may occur on pipelines it is also common to experience pressure fluctuations within the system. An elastomer secondary sealing element shown in Figure 1 can aid in eliminating fretting wear, abrasive damage and hang up, but adds the ability to "dampen" axial cyclic loading and shuttling that can damage the sealing faces.

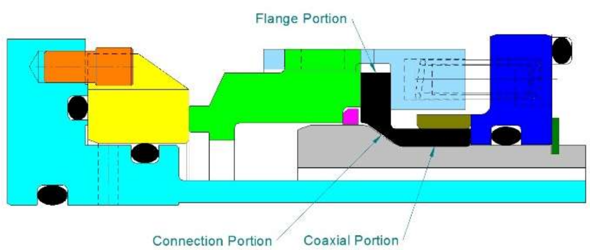

Figure 1: Non-pusher secondary seal concept

Drive Collar

The drive collar is an often-overlooked important element in successful seal design in high pressure duties. The type, material, quantity, and size of the set screws in terms of holding power can be the limiting factor in pressure ratings outside of any variable associated with the seal faces themselves. There are three common designs used. A double row collar may be used if the set screw holding power is verified as adequate to overcome the hydraulic thrust load of process pressure acting upon internal diameters of the mechanical seal. For maximum holding power, with reasonable corrosion resistance, knurled point set screws are recommended. Alternative drive/thrust collar arrangement have also started to gain popularity, such as a split ring or ring shrink disc collar design. It is worthwhile to note that on between-bearings pumps a shaft relief for set screw engagement is recommended to avoid passing the mechanical seal cartridge over raised metal in the shaft during maintenance activities.

VARIABLE SPEED OPERATION

The impact of variable speed drives on mechanical seal operation can be a separate topic, and for more information the reader is steered towards the references section where more information can be found.

The added complexity of variable speed or Variable Frequency Drives (VFD), in midstream pipeline applications warrants careful consideration with regards to the cooling flow requirements of the mechanical seal. Proper lubrication of mechanical seal faces is essential to prolonged reliability of the mechanical seal. The rate of the flush injection is specific to the operating conditions and can be moderately influenced by the pump type and impeller design. The pressure acting on the mechanical seal in the seal chamber can range from suction pressure to nearly discharge pressure. To determine the rate of injection, an estimate on the amount of heat generated by the mechanical seal and associated temperature rise in the fluid around the seal components is the most used criteria. Flush temperature rise is calculated based upon heat load, fluid properties, and process fluid flows into or out of the seal chamber. The heat load is a combination of seal face generated heat, heat soak, and churning or turbulent energy of the seal components rotating in the seal chamber immersed in the sealed fluid. In low viscosity fluids such as light hydrocarbons, this component can usually be ignored, but in the case of more viscous fluid such as crude oil the influence of churning is usually factored in especially as the seal size increases.

Recommended allowable temperature rise of the process fluid in the seal chamber varies based on the fluid being sealed – typical values would be 8°C (15°F) for water and low volatility hydrocarbons and 16°C (30°F) for lube oils. For volatile light hydrocarbons, the allowable temperature rise would be closer to 3°C (5°F). It is important to recognize that when determining flush flow rates that the target temperature rise is for the fluid surrounding the seal components and is not an indication of the actual seal face temperature. This is critical as designing for a minimum temperature rise in the flush may provide a limited margin of safety. For this reason, a more typical guideline would be to use the larger value of whatever is calculated or 0.15 lpm per mm (1 GPM per inch) of the seal size. In the case of a volatile light hydrocarbon service, the value should be increased to 0.30 lpm per mm (2 GPM per inch) of the seal size.

Varying speed in pipeline applications is especially common as the operation of the equipment is driven by downstream demand. There are often instances where the operation necessitates throughput, in which case it is advantageous for the unit operator to be flexible in how to operate the pump; increasing or decreasing speed to address requirements. This operational flexibility can be detrimental to the mechanical seal if not considered in the design and application of the seal and associated support system. Heat generation within the mechanical seal is proportional to speed; at low speed, mechanical seals require less flush flow while at high speed the flush flow requirements will be increased. In a typical pipeline application, the mechanical seals are supported by an API Plan 11 (see Figure 4), which uses the pumped product to cool and lubricate the seal faces. It takes the process from a high-pressure region of the pump through a flow control orifice and directs the flow into the seal chamber. The high-pressure region can be from the pump discharge, the pump discharge piping, or on multi-stage pumps from an intermediate stage with a suitable pressure differential above the seal chamber pressure. As the API Plan 11 flow is proportional to the differential pressure between the source location on the discharge side of the pump and seal chamber pressure, the influence of varying speed of the pump results in more flow to the seals at higher shaft speeds and less flow at slow speeds.

Outside of the concerns associated with the influence of the speed changes to the mechanical seal flush flow rate, the seal faces are continually compensating for thermal, and pressure induced changes as the speed varies. In many cases, the faces are subsequently 'wearing' against one another and as such take longer to establish a steady-state pressure profile. During these fluctuations, leakage rates can vary greatly along with frictional generated heat. Assuming one of the face materials is a soft component like carbon for example, wear-in effect is expected and can be reasonably compensated for in design evaluation by utilizing Finite Element Analysis (FEA) modeling techniques. However, in the case of an application where there are two hard face materials, such as in a crude oil or viscous service, the influence of variable speed in terms of face profile can be substantial especially at higher pressures. In these instances, a careful design review is critical to identify potential mitigating steps that can be taken to ensure stable operation across the speed and pressure ranges of the application. There have been many successful applications in variable speed operation where seal face geometry adjustments were made to mitigate significant pressure induced distortion effects at a reduced operating speed. Figure 2 outlines an example of a large diameter seal in a viscous oil service. Both simulations depict operation at 65.5 BARG (950 PSIG) @ 900 RPM; the simulation on the right is post modification to the face geometry to mitigate the pressure induced distortion effects, subsequently improving upon face generated heat and leakage.   Face temperatures reduced by 10% by using computer-aided software to simulate actual field conditions.

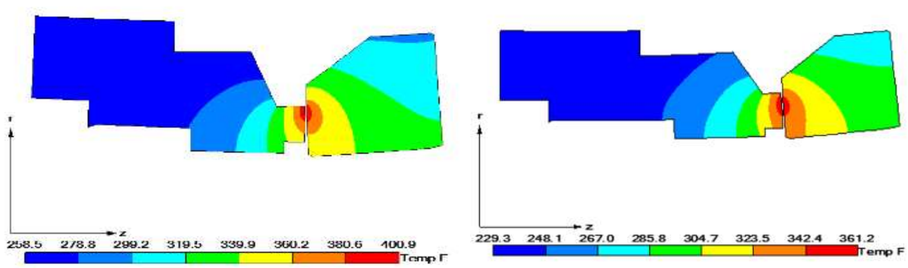

Figure 2: Geometry Modification for Pressure Induced Distortion – Pre & Post Modification Using FEA Software

SEALING STRATEGIES FOR MID-STREAM APPLICATIONS

Pumps used in midstream pipeline service are almost always described and defined by the API 610 pump standard. API 610 groups pump types using a designation code. The following pump types are typically used in midstream pipeline applications:

- OH2 is one horizontal overhung impeller, center line 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 and is most commonly used for mainline applications
- 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 vertical turbine pump and commonly used in booster applications

Types BB1 and BB3 are the most common pump designations in most pipeline applications handling crude oil and refined products. Traditionally high-pressure services associated with NGLs, and supercritical fluids were handled through BB3 or BB5 pumps. Recent installations have found BB1 pumps being specified for NGL services as well.

Types of seal arrangements used in API pumps in pipeline applications:
- Single seal or API 682 Arrangement 1
- Dual unpressurized seals or API 682 Arrangement 2 (wet/dry/non-contacting containment seals)
- Dual pressurized seals or API 682 Arrangement 3

CRUDE OIL PIPELINE SEALING STRATEGIES

When sealing crude oil, strong consideration should be given to the fluid properties as identified by the end-user, primarily the specific gravity and viscosity values at relative temperature. Crude oil generally has four levels of classification, to include light, medium, heavy, and extra heavy. Crude oil as a raw product can have varying chemicals compositions that contribute to establishing a functional API gravity classification, where the relationship between specific gravity and API gravity is defined as follows:

$$sg@60F = \frac{141.5}{API + 131.5} \qquad (1)$$

By this definition, crude oil classifications can be summarized as follows:
- Light crude has an API gravity higher than 31
- Medium crude oil has an API gravity ranged between 22 and 31
- Heavy crude oil has an API gravity ranged between 10 and 21
- Extra heavy crude oil has an API gravity below 10

Chemical components typically associated with crude oil in pipeline transportation can include low boiling point fluids (methane, ethane, propane, etc.) in low concentrations, sulfur, wax content, asphaltenes, sand and pipeline rouge. Crude oil is typically classified as a non-flashing hydrocarbon in that its respective vapor pressure is lower than atmospheric pressure. So, while raw crude oil as extracted from the well head contains entrained gases that are removed during the stabilization process, most remaining elements equate to a vapor pressure value that enable the crude oil to remain in liquid form even when pressure falls below this value.

The varying makeup of crude oil often requires a thorough application review by the mechanical seal supplier in terms of selecting seal arrangements, materials, and mechanical seal face profiles. Pumps operating in series and escalating seal chamber pressures within a single pumping station often require the manufacturer to evaluate performance at multiple operating points. Variances in speed and pressure, combined with the viscous nature of crude oil require the detailed consideration of seal design parameters to minimize seal leakage.

By this definition, typical operating parameters in crude oil pipeline applications can vary as follows:
- Suction Pressure: 3.5 to 100 BARG (50 to 1500 PSIG)
- Maximum Discharge Pressure: 152 BARG (2200 PSIG)
- Speed Range: 600 to 3600 RPM
- Temperatures: 10 to 38°C (50 to 110 °F)
- Viscosity @ Pumping Temperature: 3 to 1,000 cP
- Specific Gravity: 0.73 to 0.93
- Shaft diameter: 67 to 165mm (2.625" to 6.500") at the mechanical seal location
- Mode of operation: 2 to 6 pumps in series

Mechanical Seal Specifics and Method

Default seal face materials are dis-similar grades of silicon carbide run against one another. The common grades are reaction-bonded paired against an alpha-sintered material. Defaulting to hard face materials is a natural defense against both abrasive damage and face blistering failure modes that can plague comparatively softer face materials such as carbon. The choice of hard face materials eliminates the requirement of having to filter the primary seal flush stream that is both cooling and providing lubrication of the sealing interface.

The remaining components are consistent with standard materials in alternate pipeline segments, to include 316 or Duplex stainless-steel hardware and Fluoroelastomer secondary sealing elements. Applications containing H2S or amines have been successfully addressed utilizing Perflouroelastomer or TFE-Propylene secondary seals.

The nature of the process with inherent entrained solids and waxing can make survivability of conventional secondary sealing elements difficult. This natural abrasiveness of the process fluid coupled with inherent shaft movement of some common pump configurations in these services can accelerate secondary sealing element wear. Implementation of a non-pusher secondary sealing element to these services has proven to be highly effective when compared to potential alternatives such as utilizing higher durometer O-rings and hard coated adjacent surfaces.

Preferred Seal Arrangements

Preferred sealing platform for crude oil applications in midstream pipelines is a cartridge mounted, single seal, multiple spring design with engineered non-pusher secondary sealing element. Configuration of the seal faces should be oriented such that the seal head is stationary, and the mating ring is rotating. The orientation of the seal faces in this fashion is based on installation experience and considers the typical pump types (between bearings designs, axial, or radial split casings). Traditionally, stationary seal head configurations are better suited for compensation of potential perpendicular misalignment between the pump shaft and casing. Figure 3 shows a single seal with non-pusher secondary sealing element, with 66A (dual bushing containment).

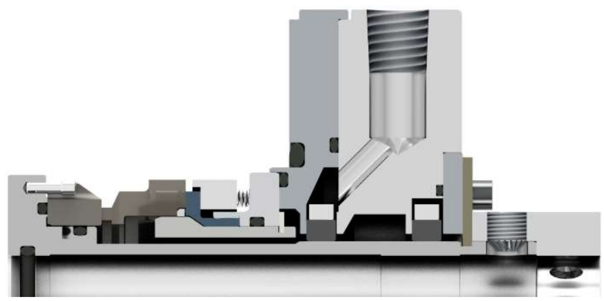

Figure 3: High-Duty Crude Pipeline Seal utilizing non-pusher secondary seal and dual bushing containment

The primary means of lubricating and cooling is achieved through API Plan 11, sourced from an appropriate location from discharge pressure regions of the pump. Care must be taken to ensure that cooling requirements are met across the entire operating window, considering both pressures, temperatures, and speeds. The method of flow control is most simply achieved with a drilled orifice plate or union; the sizing of the orifice should be appropriate to provide coverage of cooling flow within the defined operating parameters. It should be noted that implementing filters in viscous crude oil environment will lead to an extremely high fouling rate and therefore is not recommended. The separation efficiency of a cyclone separator as part of API Plan 31 also is severely impacted as liquid viscosity increases and therefore is not recommended. A simple sketch of API Plan 11 is shown in Figure 4

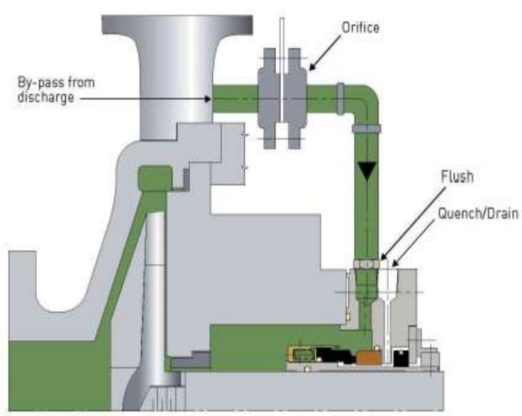

Figure 4: API Piping Plan 11

Containment strategies for single seals have been well-defined in API 682 4th edition, and piping Plan 66A and 66B are more commonly used in crude oil services today. Seal cartridges designed to include Plan 66A/B will be like the standard API 682 Arrangement 1 except that the gland plate for 66A includes an additional bushing and therefore is longer. Excessive leakage from the seal will be restricted by a bushing and increase the pressure within the monitoring cavity. This pressure increase will be detected by the pressure transmitter and is roughly related to the leakage rate. Leakage will then be directed into the drain. Plan 66A does have several control and design limitations to be considered based on each application. As an attempt to quantify seal leakage, the operator will have to consider variances in oil viscosities, operating pressures, bushing clearances and shaft speeds that impact overall leakage within both the primary seal assembly and the intermediate bushing. In determining suitable alarm and shutdown parameters, the operator will have to consider these variable factors. Because of the additional bushing and location of the drain, Plan 66A is much more sensitive to excessive leakage detection than Plan 66B. Figure 5 outlines the comparison between API Plan 66A and B.

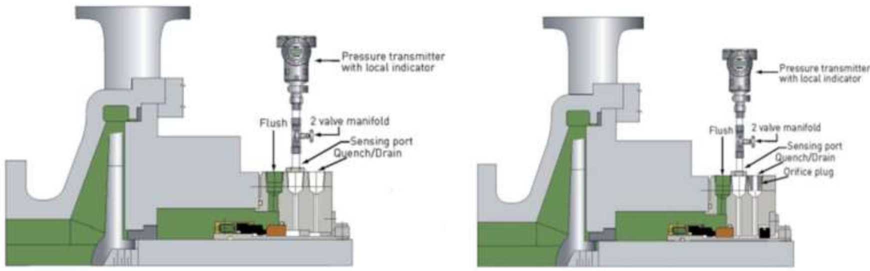

Figure 5: Comparisons between API Piping Plan 66A & 66B

Both Plan 66 arrangements only provide monitoring of the cavity, they are not leakage collection or management devices. To provide leakage containment, 66A or B would use Plan 65A or 65B (Figure 6) in conjunction with these monitoring plans. Note that 65A and B are both collection reservoirs with subtle differences. Plan 65A has the collection vessel with the drain open to sump/drain where the vessel drains through an orifice. The orifice restricts higher levels of leakage to build a level for identification of a failure allowing for a more controlled shutdown. The Plan 65B uses the same vessel but has a manual drain valve instead of the orifice. This requires manual intervention to drain leakage but can provide a more realistic measure of leakage over time as frequency of draining would indicate that the normal leakage levels have increased. Despite the monitoring capability, the remote nature of some pipeline installations makes the Plan 65B less desirable for this reason.

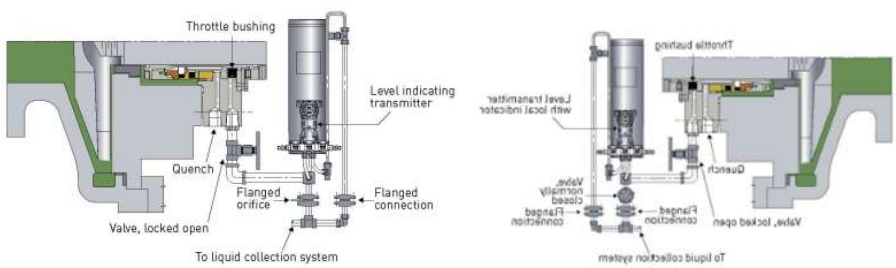

Figure 6: Comparisons between API Piping Plan 65A & 65B

Pipeline operators have deployed dual unpressurized sealing options where levels of H2S over 1000 PPM (Parts Per Million) are present or where a more robust containment sealing strategy is required. The preferred arrangement is to utilize a dual unpressurized seal with similar methodology outlined in Arrangement 1 configurations in addition to a dry-running containment sealing device and large volume collection reservoir. Various considerations with regards to materials, face balancing and spring load are implemented within a dry-running contacting seal concept, thus providing a positive end-face solution in lieu of a clearance bushing. Due to the nature of crude oil, a non-contacting containment seal option is not recommended as the micro-machined face structures would be susceptible to fouling. A dual unpressurized seal with dry containment design is shown in Figure 7 as an alternative solution; in this configuration both the process seal and containment seal utilize non-pusher secondary sealing elements to protect against fretting degradation damage from associated shaft movement.

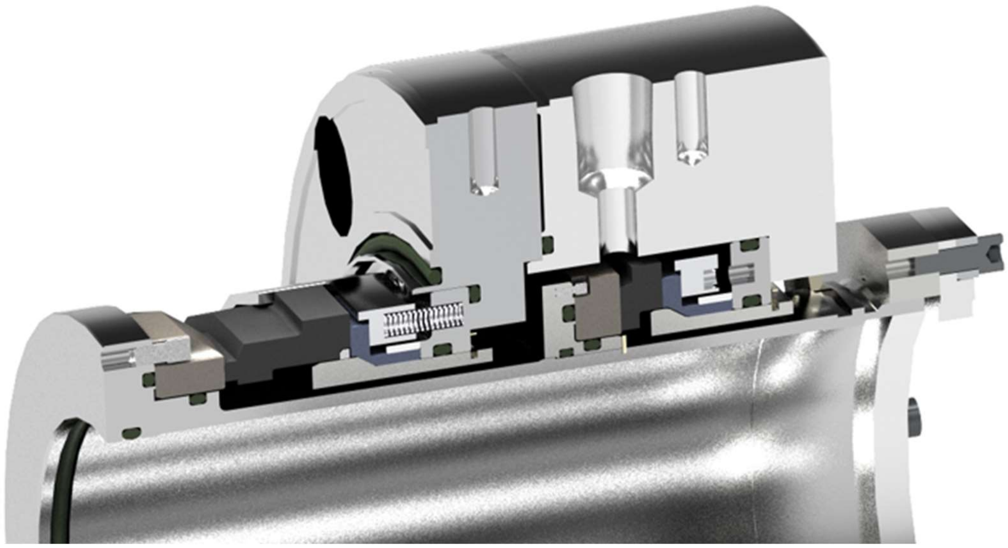

Figure 7: Dual un-pressurized seal with dry contacting containment & non-pusher secondary elements

API Plan 75 is intended for use in Arrangement 2 applications utilizing a dry-running containment seal concept to aid in managing inner seal leakage that would condense at atmospheric conditions. In this arrangement, the seal gland will contain a drain port located at the 6 o'clock position that is routed to a collection reservoir. The reservoir is typically supported with both a level and pressure transmitter. For applications where normal leakage remains in liquid form, the level transmitter will provide feedback in terms of a normal level rise on a time-based frequency. The operator should work with the seal supplier to establish acceptable leakage rate criteria and the Plan 75 system can then be monitored based on volume accumulation. Although API Plan 52 (liquid buffer) has been used in some installations, it is not recommended as a default piping plan for midstream pipeline applications and crude oil. While the reservoir with normal working level provides an indication as to the overall health of the containment seal, the buffer fluid environment must be maintained on a frequent interval. When evaluating API Plan 52 in crude oil or refined products pipeline applications, normal leakage from the primary sealing interface will eventually contaminate the buffer fluid, leading to frequent maintenance tasks of draining and re-filling. Figure 8 is an example sketch of API Plan 75.

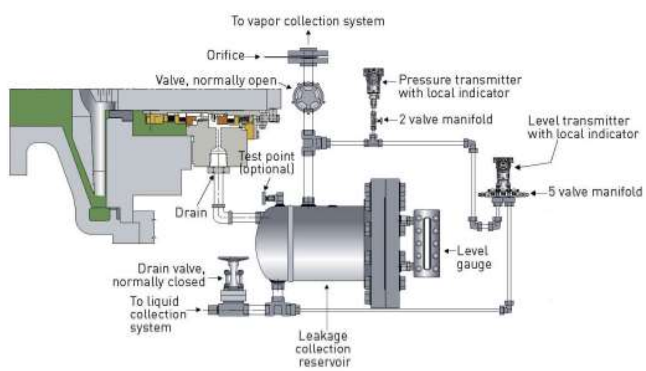

Figure 8: API Piping Plan 75

Dual pressurized configurations have been deployed in environments where the pump is subjected to variations in ambient site conditions, such as sub-zero temperatures or multiple API gravity liquids. Variation in ambient temperatures can impact the viscosity of crude oil when the pumping equipment is not operational, resulting in potential freezing of process within both the flush/drain lines and potential high torque scenarios at equipment start-up that result in either excessive face blistering or grain pull-out. The preferred sealing arrangement orients the seal in a way that allows for isolation of the primary sealing components from the varying process fluid conditions and thus provides a more stable interface. Design aspects associated with the Arrangement 1 and 2 seals should be carried over when considering dual pressurized configurations considering concerns associated with shaft shuttling and degradation of conventional O-rings. Successful design configurations are modeled after Figure 9 below.

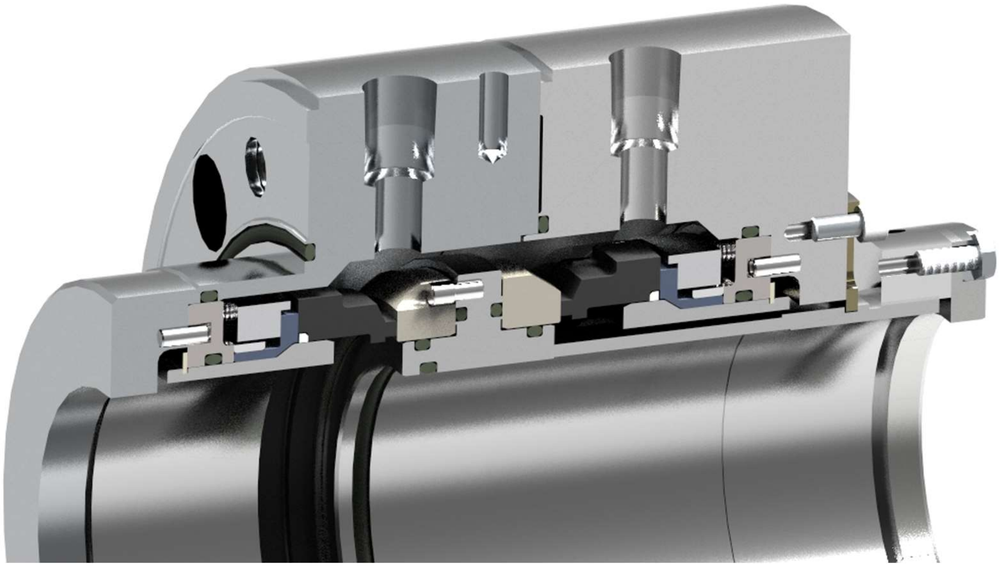

Figure 9: Dual Pressurized Seal Configuration with non-pusher secondary sealing elements

API Plan 53B is primarily intended for use in pressurized seal configurations. Plan 53B incorporates a closed loop circulating system and the bladder accumulator that provides a source of positive pressure. The circulation system entails all associated piping, a cooling mechanism typically in the form of a water or air-cooled exchanger and large sized reservoirs for added volume capacity. The barrier fluid requires circulation via an internal circulation device to deliver the required flow rate for heat dissipation through the seal assembly. The system pre-charge to the bladder accumulator is critical to ensure that as the working volume decreases due to seal leakage of both seal assemblies, the system pressure remains above the minimum allowable system pressure preventing loss of product containment. The bladder does act as a barrier between then pressurization gas and the barrier fluid, thus preventing gas absorption into the barrier fluid common with Plan 53A systems at higher operating pressures. Due to the combination of higher leakage rates associated with dual pressurized arrangements and the remote location of most pumping stations, it's common to incorporate a large volume auto-fill skid. Figure 10 depicts a typical API Plan 53B arrangement.

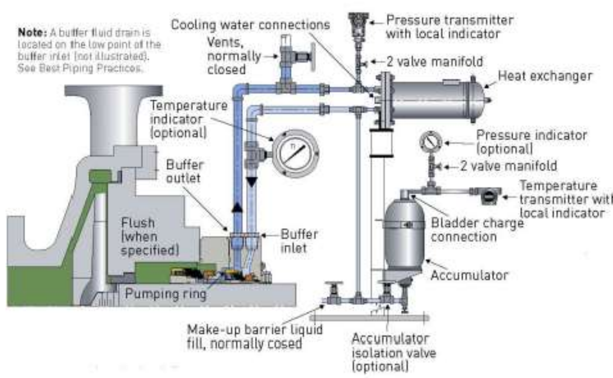

Figure 10: API Piping Plan 53B

An alternative option is to use API Plan 54, (Figure 11) which system provides a clean pressurized barrier fluid to isolate the sealing interface from the main process liquid and in addition, adds an additional level of redundancy in terms of process containment. In terms of pipeline installations, the scope of supply typically includes a large volume reservoir, redundant circulation pumps, duplex inline filtration, the use of an air-cooled exchanger and various transmitter components. The transmitters interface with the Distributed Control System (DCS) to provide equipment performance monitoring and assist with main process pump shutdown in the event of an upset condition. Increased seal leakage will manifest itself through decreasing levels in the operating reservoir, which should be sized adequately and implement a sequenced alarm strategy to allow for troubleshooting in the field prior to equipment shutdown. Plan 54 systems offer full process containment and can extend equipment reliability by providing a stable environment; however, there is a significant amount of upfront cost and complexity in terms of maintenance associated with this system. When implementing Plan 54, the pipeline operator should work through their equipment supplier to ensure suitable instruction, operations, and maintenance (IOM) documentation is provided, along with detailed training. There are numerous configurations available for stand alone seal lubrication systems that would be categorized as an API Piping Plan 54 console. Some suggested controls and monitoring logic for these types of systems were highlighted by Kalfrin and O'Brien in a tutorial from the 36th Pump Users Symposium (2020).

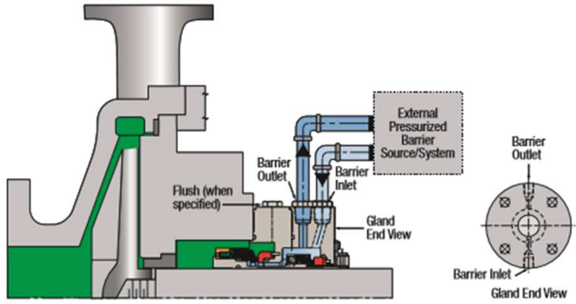

Figure 11: API Piping Plan 54 Diagram

REFINED PRODUCTS PIPELINE SEALING STRATEGIES

When reviewing pipeline applications defined as refined products within this paper, the focus will be geared towards petroleum derivatives processed at various refinery locations within the Oil & Gas industry. Crude oil located in various underground oil reservoirs is first extracted and then undergoes the initial processing at gathering centers prior to transport. By itself, crude oil is not a marketable product and requires further refining through the staged processes of separation, conversion, purification and upgrading. Refined crude oil will yield hydrocarbon products such as gasoline, diesel fuel and jet fuel, which typically have relatively low vapor pressure values. Most of these hydrocarbon by-products also have higher viscosities than NGL pipeline fluids, thus providing a more stable fluid film at the sealing interface.

Typical operating parameters in refined products applications can vary as follows:

- Suction Pressure: 3.5 to 100 BARG (50 to 1500 PSIG)
- Maximum Discharge Pressure: 1800 PSIG
- Speed Range: 1800 to 3600 RPM
- Temperatures: 16 to 38°C (50 to 110 °F)
- Viscosity @ Pumping Temperature: 0.5 to 3 cP
- Specific Gravity: 0.70 to 0.86
- Shaft diameter: 67 to 165mm (2.625” to 5.750”) at the mechanical seal location
- Mode of operation: 2 to 6 pumps in series

Mechanical Seal Specifics and Method

Default seal face materials typically antimony impregnated carbon paired against a hard face mating ring, typically silicon carbide. The common grades are reaction-bonded paired against an alpha-sintered material. The remaining components are consistent with standard materials in alternate pipeline segments, to include 316 or Duplex stainless-steel hardware and Fluoroelastomer secondary sealing elements. Applications containing H2S or amines have been successfully addressed utilizing Perflouroelastomer or TFE-Propylene secondary seals. As with crude oil applications, the implementation of a non-pusher secondary sealing element in these applications ensures secondary sealing element integrity if the seal is subjected to inherent shaft shuttling present in the pump design. Pipeline scale or ‘rouge’ can accelerate conventional secondary sealing element wear, making non-pusher secondary sealing elements advantageous to enhancing the inherent robustness of the seal design.

Preferred Seal Arrangements

Preferred sealing platform for refined products applications in midstream pipelines is a cartridge mounted, single seal, multiple spring design with engineered non-pusher secondary sealing element. Configuration of the seal faces should be oriented such that the seal head is stationary, and the mating ring is rotating. The orientation of the seal faces in this fashion is based on installation experience and considers the typical pump types (between bearings designs, axial or radial split casings). Traditionally, stationary seal head configurations are better suited for compensation of potential perpendicular misalignment between the pump shaft and casing.

In support of the primary sealing interface, a simple piping Plan 11 flush is typically utilized based on the overall flow requirements. Refined products in pipeline service tend to operate at lower pressures and when combined with their enhanced lubricating properties, the overall heat load is reduced. The ideal temperature rise for the seal flush in refined products is generally targeted as 15°F (8 °C), which translates into a more manageable flush rate as compared to flashing hydrocarbon services. Some pipeline operators might also employ the use of API Plan 31 (Figure 12) with a cyclone separator to assist with removing solids from the flush stream in support of the primary sealing interface of either a single seal or dual unpressurized installation. The process fluid is introduced to the cyclone separator from a defined intermediate stage discharge and feeds into the top of the cylindrical cone of the separator. The inlet flow creates a vortex action that discharges the particulates to the wall of the separator and eventually pass downward out of the unit and routed back towards the suction side of the pump. The clean fluid moves inwards and is displaced from the top of the unit and to the seal chamber region as a means of cooling and lubrication. When considering the use of a cyclone separator, the service should be reviewed to ensure that the entrained solids have a specific gravity of at least twice that of the process fluid.

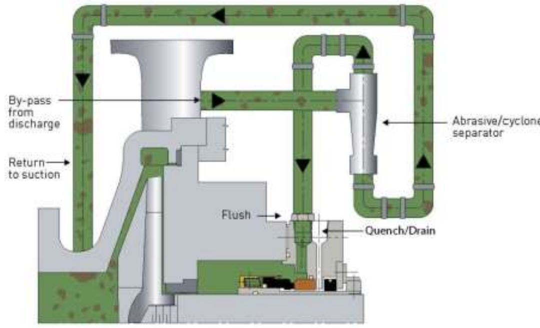

Figure 12: API Piping Plan 31

API Piping Plan 31 as applied within the pipeline industry does have several limitations that should be carefully managed during selection/installation. Considering the increased usage of variable frequency drive motors, implementation of fixed orifice plates or manual valves for flow control and limited pressure differential at low-speed operation can often result in loss of adequate flow rates. Combined with the remote nature of most pipeline applications, adequate balancing of the distributed clean/dirty flow rates as seal flush requirement varies is difficult to achieve as these stations are unmanned. Cyclone separators by design have limits in the maximum differential pressure required to perform properly, thus relocating the flush origin to final stage discharge to provide more additional flow is not recommended. The day-to-day variance in required pump throughput almost requires the use of flow control valves that interlock with the DCS system to determine flow rates required based on changing speeds and pressure. This adds both complexity in terms of control logic and cost that can be more reliably addressed through alternate means.

Most pipeline operators utilizing single seal arrangements in the refined products segment use a hybrid control and support system. While the front-end seal arrangement and containment bushing is rather consistent, the defining criteria for control will be dictated by the application requirements. In a heavy diesel application, normal leakage will remain in pure liquid form and utilization of a collection reservoir with a level alarm is most likely suitable. In a more volatile and unstable service such as gasoline, the combination of both an upstream pressure transmitter and level switch in a collection reservoir might be better suited.

As previously identified within this paper, pipeline operators have deployed dual unpressurized sealing options where a more robust containment strategy is desirable. The preferred arrangement when utilizing a dual unpressurized seal in refined products is to incorporate methods outlined with Arrangement 1 options in addition to a dry-running containment seal and large volume collection reservoir. In refined products applications, reliable seal configurations will mirror those depicted in Figure 3 (Arrangement 1) and Figure 7 (Arrangement 2) with appropriate considerations to face materials and secondary element design and material based on the pump configuration. The use of wetted contacting containment seals with conventional API Plan 52 support systems in these applications is typically not recommended due to the previous considerations related to maintenance requirements, buffer fluid contamination, and lack of support utilities on-site.

It should be noted that dual pressurized systems are very rarely offered in this segment of pipeline services. If selected, the applicable sealing arrangements and support system considerations discussed in this paper regarding Arrangement 3 seals would be options. Typically, the process contamination as a by-product of the pressurized barrier fluid makes this option less desirable for pipeline operators.

VOLATILE & FLASHING HYDROCARBON PIPELINE SEALING STRATEGIES

The focus of this paper will be on raw Natural Gas Liquids or "Y-Grade" variations that are byproducts from the upstream natural gas gathering of various wellheads and the subsequent gas purification processes. Once separated from the natural gas resource, the mixture of flashing hydrocarbon components is transported to fractionation plants for further separation into their base components via a boiling process. Once the various hydrocarbon components have been separated, they are then moved to storage prior to shipment as feedstock demand for the petrochemical industry.

When evaluating mechanical seal designs related to light hydrocarbons in the pipeline market segment, it's critical to understand the overall fluid make-up of the application. The definition of a flashing hydrocarbon according to API 682 4th edition is as follows:

3.1.36 Flashing Hydrocarbon – liquid hydrocarbon or other fluid with absolute vapor pressure greater than 0.1 MPa (1 bar) (14.7 psia) at the pumping temperature, or a fluid that will readily boil at ambient conditions.

Furthermore, API 682 4th edition defines flashing as follows:

3.1.35 Flashing – Rapid change in fluid state from liquid to gas.

Note – In a dynamic seal, this can occur when frictional energy is added to the fluid as it passes between the primary seal faces, or when fluid pressure is reduced below the fluid's vapor pressure, or a pressure drop across the seal faces.

Typical operating parameters in volatile hydrocarbon applications can vary as follows:

- Suction Pressure: 6.9 to 83 BARG (100 to 1200 PSIG)
- Maximum Discharge Pressure: 152 BARG (2200 PSIG)
- Speed Range: 1800 to 3600 RPM
- Temperatures: 15.7 to 44°C (60 to 110°F)
- Viscosity @ Pumping Temperature: 0.06 to .15 cP
- Specific Gravity: 0.42 to 0.58
- Y-Grade liquids with varying C2 - C8 compositions
- Shaft diameter: 67 to 165mm (2.625" to 5.510") at the mechanical seal location)
- Mode of operation: 2 to 4 pumps in series

Mechanical Seal Specifics and Method

Like the methodology for refined products, common seal face materials typically include antimony impregnated carbon paired against a silicon carbide mating ring due to the low coefficient of friction value of this material pairing. Mechanical seals operating in the NGL application range will typically do so with very little hydrodynamic load support due to the low viscosities in place. It is likely that the seal face in such applications will operate in a solid to mixed friction regime. The face materials are likely to experience higher wear rates in these regions due to increased temperature (from rubbing friction) and potential break down due to significant hydrostatic loading of the faces. Considering this, a suitable face treatment is often utilized to augment the lubricating fluid film between the seal faces. These treatments range from grooves, recesses, and various methods to increase fluid film to minimize contact pressure and wear. The remaining components are consistent with standard materials in alternate pipeline segments, to include 316 or Duplex stainless-steel hardware and Fluoroelastomer secondary sealing elements. Perflouroelastomer or TFE-Propylene secondary seals can be offered like both the crude and refined products offerings when the application is subjected to amines or H2S. As pump OEM's have most recently started to furnish BB1 style pump equipment, the implementation of a non-pusher secondary sealing element on NGL applications is becoming more prevalent to resist inherent shaft shuttling and the impact of pipeline scale or 'rouge', which accelerates degradation of the conventional secondary sealing component as a dynamic element

Preferred Seal Arrangements

Within NGL pipeline services, single cartridge assemblies serve as a viable platform for mechanical sealing in flashing hydrocarbon pipeline services. The primary sealing interface will typically incorporate a variety of face treatments to enhance the lubricating regime at the interface, implementation of a flow diverter component with multi-port injection for enhanced cooling and integral restrictive bushings for increased vapor suppression. The single seal offering should be supported with a tight tolerance bushing on the atmospheric side of the primary interface and venting primary seal leakage through an orifice to an acceptable vapor recovery system. The gland housing should be fitted with an upstream pressure transmitter to identify increases in seal leakage. A relatively narrow pressure band on the transmitter is recommended, 0 – 15 PSI (0 – 1 BAR) to capture appropriate changes in pressure through the intermediate cavity, triggering an alarm. Figure 13 outlines a sketch of a single, pusher, multi-spring arrangement, with Piping Plan 11 and 66B.

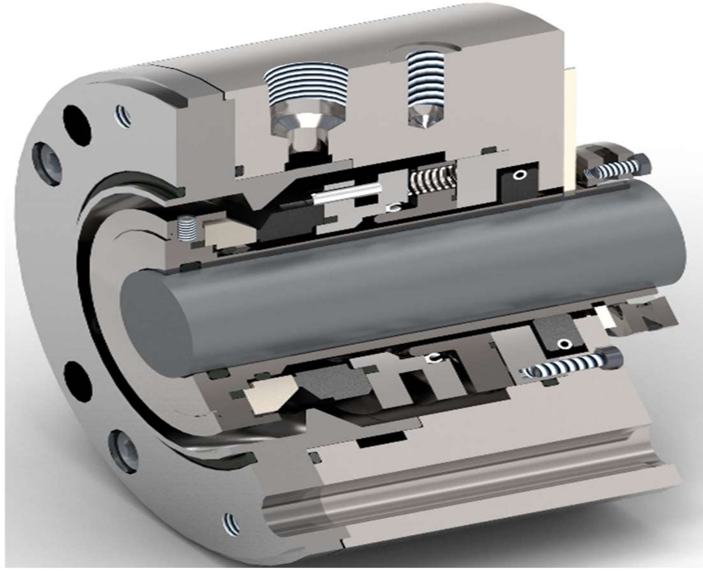

Figure 13: Single, multi-spring, pusher configuration with API Plan 11 / 66B

The primary seal interface should be supported with Piping Plan 99 (12)1, which is a variation of Plan 11, but inclusive of an inline filtration system. The use of filtration is becoming more common in pipeline applications due to the influx of pipeline rouge and various seal design concepts being susceptible to abrasive wear and hang-up of the flexible element. The primary function of the filtration system is to remove various particulates from the flush stream supplied to the mechanical seal assembly within single seal and dual unpressurized configurations. Typically defined as a discharge bypass, Plan 99 (12)[1] routes an appropriate amount of fluid from either the discharge of the pump or in the case of BB3/BB5 style machines, an intermediate stage discharge of the pump to the seal chamber. This flow rate is controlled by either a restriction orifice (RO) or variable flow control valves. As the ideal temperature rise for NGL applications is targeted as 5°F (-15°C) to maximize reliability, a duplex filtration system (Figure 14) is recommended to accommodate the higher flush rate requirement while maintaining continuous operation once the filter element begins to foul. A minimum requirement for pipeline applications should be the implementation of a differential pressure transmitter across the filter housing, which serves as the primary indicator that an element is fouled significantly and that seal flush flow has been compromised.

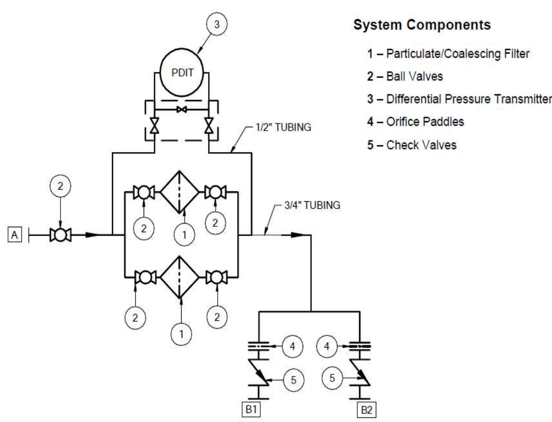

Figure 14: Duplex filtration system

[1] *Plan 99 designation. A Plan 99 is simply an engineered piping plan that is not defined by any of the existing plans in the standard. It is not sufficient to indicate "Plan 99" on a seal data sheet or even on a seal layout drawing as a lone descriptor. A drawing of the Plan 99 and notes about its operation should be supplied. Good practice when adopting this designation is to include a descriptor along with the "99" designation to provide clarity if the proposed piping is a variation of an existing piping plan, such as Plan 99 (12) for example*

The design of the filtration system should consider the total seal flush requirement for both seal assemblies on a between bearings pump. Once again, the pipeline operator should work with their seal supplier to determine maximum flow requirements across the spectrum of all operating parameters. Once the total flow rate is established, the pipeline operator should work with their equipment supplier to determine the following:

- Filter element micron rating
- Filtration surface area
- Implementation of transfer valve for non-interrupted service

- Implementation of magnetic separator
- Control logic regarding equipment shutdown or maintenance required due to plugged elements

The items above are critical in terms of establishing a suitable maintenance schedule to address fouled elements. Pipelines are subjected to varying degrees of pipeline rouge, pigging frequency, and overall flow demand; however, maximizing the run-time of the filtration system can be achieved with a robust design. While the upfront cost of filtration is higher than Plan 11 on its own or even Plan 31, minimizing equipment down-time and maintenance tasks can provide valuable return on investment in short order.

The majority of NGL pipeline operators in North America have defaulted to a dual unpressurized seal arrangement, utilizing dry-running containment seals in lieu of the tight tolerance bushing see Figure 16. Though both contacting (Figure 16), and non-contacting secondary containment design concepts have been implemented successfully, a thorough application review should be completed by the seal manufacturer to establish containment seal type in accordance with best practice and user preference. Advantages and disadvantages for both non-contacting and contacting containment seals are highlighted by Kalfrin and Gonzalez in a tutorial from the 31st Pump Users Symposium (2015).

Figure 15 illustrates the typical unpressurized dual pusher seal with a dry running non-contacting safety containment pusher seal. The outboard seal utilizes a spiral groove faces pattern to provide hydrodynamic lift-off to separate the faces while running at low pressure operation. The faces run on a gas film generated by the vicious shearing of the gas at the interface (between the sealing faces). This increase in pressure is enough to separate the faces, resulting in a non-contacting condition. This technology was first introduced to mechanical seals over fifty years ago for turbo gas compressors and then to the pump industry 25 years ago. Being non-contacting offers many advantages, such as, long life and less expensive and low maintenance support systems. Typically, they are pusher seals using an O-ring as the secondary sealing element. For enhanced reliability in specific pipeline pumps, the design in Figure 15 is depicted with a non-pusher secondary seal instead of an O-ring. Details of the development around this configuration specifically focused on usage within NGL pipelines are detailed in a lecture by Wasser Et al. in proceedings from the 36th Pump Users Symposium (2020). Designs can be uni-directional or bi-directional. Like the dry running contacting secondary seal, the non-contacting arrangement using an API 11/75 or 76.

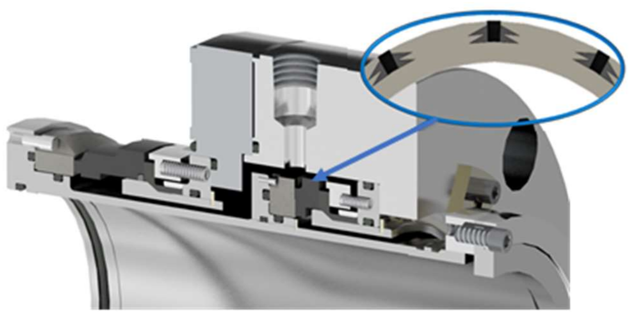

Figure 15: Dual Unpressurized seal with non-contacting containment seal for NGL Service

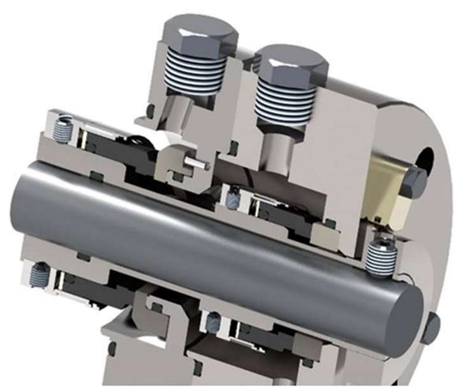

Figure 16: Dual Unpressurized configuration with contacting containment seal

API Piping Plan 76 (figure 17) is intended for use with dual unpressurized arrangements utilizing a dry-running containment seal that assists with leakage management where inner seal leakage remains in vapor form and will not condense at lower temperatures or pressures. Piping plan 76 requires the seal gland be equipped with a vent port at the 12 o'clock position that is routed to a flare or vapor recovery system. The vent system includes an installed orifice to induce back pressure at higher leakage rates with an upstream pressure transmitter to provide indication of compromised primary seal integrity. Typical orifice diameters are sized at 0.125" (3mm), but smaller sizes may be necessary to provide a realistic leakage alarm point. A word of caution, however, is to be mindful of the orifice diameter considering many midstream pipeline applications being susceptible to contaminants by nature. From experience, orifice diameters ranging from 0.093 to 0.125" (2.4 to 3mm) are typically sufficient for these services. The pipeline operator should work with the mechanical seal supplier to define the normal leakage range across the operating parameters and then evaluate the vent system and orifice size to setup alarm and shutdown parameters through the DCS.

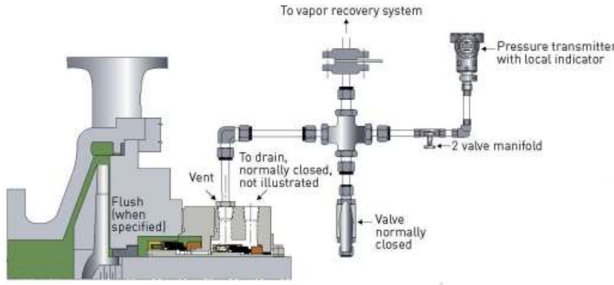

Figure 17: API Piping Plan 76

Plan 76 is a low-cost alternative to wetted dual unpressurized systems using Plan 52 and in addition has lower maintenance costs and requirements. Note that for ranged liquids that could potentially condense as normal leakage or when subjected to cold ambient environments, implementation of the previously mentioned Plan 75 adds flexibility to allow for level measurement as an indicator of a compromised primary seal along with a continuous vent as part of the same containment system. As Plan 75 and 76 both avoid use of a buffer fluid, routine field testing should be executed to evaluate integrity of the secondary containment seal assemblies. Suggested containment seal testing protocols were highlighted by Kalfrin and Gonzalez in a tutorial from the 31st Pump Users Symposium (2015).

Dual pressurized arrangements are typically not utilized in flashing hydrocarbon services, unless the operator has a zero emissions requirement for process release or if there is a major variance in the type of liquid being pumped through the same pumped equipment. This could entail a heavy oil, light condensate or process liquids classified as supercritical fluids. For those applications requiring dual pressurized arrangements, please refer to the methodology incorporating Plan 53B or Plan 54 previously described in this paper.

SUPERCRITICAL FLUID PIPELINE SEALING STRATEGIES

Pipeline applications with fluids designated as supercritical fluids like ethylene, ethane and CO2 present additional challenges related to seal design when compared to NGL fluids. The use of higher flush rates and adjusted face loading concepts are often utilized to maintain a sufficient lubrication regime and to promote face stability when evaluating NGL. Supercritical fluids pose a problem in that they are often operating near their critical point. As noted in a tutorial from the 20th Pump Symposium, for ethylene the critical point is at 742.1 PSIA (51.1 BARA) and 49.82oF (10°C), thus at any pressure above 742.1 PSIA (51.2 BAR) and any temperature above 49.82oF (10oC), ethylene is a compressible material and acts like a vapor, not like a liquid. The lubricating characteristics and the specific heat of the fluid are more like a vapor than a liquid, so attempting to seal the fluid as a liquid often results in poor reliability and a short life expectancy. Methods to mitigate the strenuous conditions described above include the use of two alternatives from traditional contacting-wet seal strategies, dual unpressurized with gas seal technology and dual pressurized arrangements.

Typical operating parameters in supercritical fluid applications can vary as follows:

- Suction Pressure: 41 to 97 BARG (600 to 1400 PSIG)
- Maximum Discharge Pressure: 152 BARG (2200 PSIG)
- Speed Range: 1800 to 3600 RPM
- Temperatures: 10 to 33°C (50 to 90°F)
- Viscosity @ Pumping Temperature: .02 - 0.04 cP
- Specific Gravity: 0.30 - .40
- Shaft diameter: 67 to 165mm (2.625" to 4.760") at the mechanical seal location)
- Mode of operation: 2 to 3 pumps in parallel

Mechanical Seal Specifics and Method

Dry running non-contacting Gas Seals (DGS) for vaporizing liquids are based on the design principles of compressor seals, with face treatments on the sealing surface modified to allow complete vaporization of the flush fluid. The seal assembly utilizes rotational energy from the pump shaft and the swirling effect, or windage, which takes place in the area adjacent to the rotating face to ensure a stable gas film at the interface. Induced face separation with non-contacting technology eliminates face wear and minimizes heat generation, extending the life of the seal assembly. The materials of construction include an antimony carbon paired against premium grade tungsten carbide, while hardware and secondary sealing elements are consistent with materials described for flashing hydrocarbons.

Seal Arrangements

Primary sealing options include dual unpressurized arrangements with both seals incorporating DGS features. Filtration on the primary flush source is required to ensure the micro-structures on the mating ring are not subjected to fouling, which hinders face lift-off. Furthermore, DGS technology typically incorporates a dynamic O-ring that is susceptible to hang-up without means of filtration as a support system. A more comprehensive overview with regards to design considerations in supercritical fluids can be found in proceedings from the 20th International Pump Users Symposium (Goodenberger/Barron/Marquardt, 2003). Figure 18 represents a typical dual unpressurized DGS arrangement:

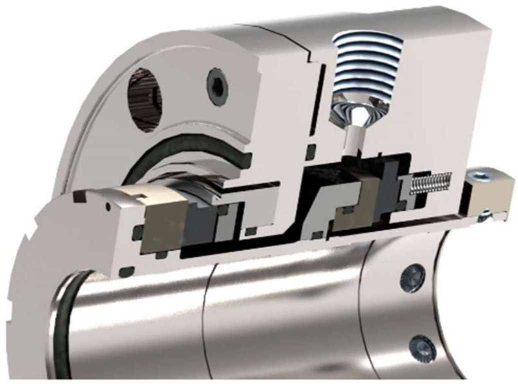

Figure 18: Typical Dual Unpressurized DGS configuration

The alternative to the dry running non-contacting option is to deploy a dual pressurized arrangement with an external pressurized source to isolate the seal interface components from the supercritical fluid. This arrangement and support system have been outlined previously within this paper; however, supercritical fluid applications can operate at significantly higher pressures and require an even more intricate support system with added system capacity. It should be noted that with pipeline operators looking to operate across a broad range of process fluids in the same pipeline, where heavier hydrocarbons are processed and not compatible with the dual unpressurized gas seal configurations, the presence of a supercritical fluid within the operating parameters typically dictates that a dual pressurized system be utilized. Please refer to the previous section related to Plan 53B and Plan 54 design considerations. Note that the respective pressurized systems require significant capital investment and a focused maintenance schedule to ensure continuous reliability. Figure 19 represents a dual pressurized wet concept with high pressure capability

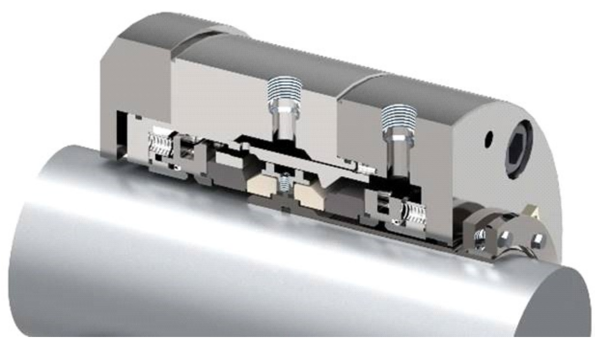

Figure 19: A dual unpressurized wet seal configuration with high-pressure capability

CONCLUSIONS

As energy demands increase, the demands for pumping equipment and especially mechanical seals in the midstream pipeline segment to accommodate higher design pressures, speeds, and extremes of variable fluid properties that are now becoming more common, will continue to evolve. The seal selection for the various and varying applications found in liquid hydrocarbon mid-stream duties can therefore prove challenging to operators and designers. Applying the most appropriate state of the art technologies correctly can produce world class equipment reliability while meeting all environmental requirements impacting the industry.

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LIST OF FIGURES

Figures courtesy of John Crane

LIST OF EQUATIONS

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