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Midstream Pipeline Applications – Design Aspects and Considerations for Mechanical Seals

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ABSTRACT

There are few applications that place a significant demand on mechanical seals such as those associated with the handling of various fluids through pipelines. Pipeline applications are unique in that they typically encompass variable fluid properties, along with fluctuations in pressure, temperature, and speed, sometimes through the same pumping equipment. There are additional challenges in the midstream pipeline sector associated with the remote nature of the installations and limited accessibility. Equipment monitoring and logistics of preventative maintenance support place an increased emphasis on the criticality of selecting a robust mechanical seal design and associated support equipment. The purpose of this tutorial is to serve as a guideline for the equipment user and define effective and efficient sealing strategies in midstream pipeline applications, integrating relevant industry best practices and lessons learned from field installations. In covering these topics, the tutorial will draw upon the combined previous experience of the authors in addressing these applications. The reader should review the content and consult as a reference, keeping in mind that not all of the content is applicable to every application and that each application should always undergo a thorough engineering review.

INTRODUCTION

Considering increased demand for product deliverables through midstream pipeline systems, there is a subsequent demand on the mechanical seals incorporated in the equipment within these networks. Many midstream pipelines handle multiple products with variable fluid properties and varying operating conditions. Pressure and temperature fluctuations along with variable fluid properties warrant special consideration into aspects of the mechanical seal in order to achieve satisfactory performance, achieving accpetable levels of compliance in terms of safety, reliability, and environmental benchmarks. Typical fluids encountered in midstream pipeline facilities range from crude oil, refined products (gasoline / diesel), flashing hydrocarbons (propane, butane, NGL), supercritical fluids (ethylene, ethane, and EP mix), and Carbon Dioxide.

Pumps used in midstream pipeline service are almost always described and defined by the API 610 pump standard, where API 610 groups pump types using a designation code. The following pump types are typically used in midstream pipeline service, with BB1, BB3, and BB5 being the most common for higher pressure and capacity considerations:

- 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, referred to as a "barrel pump"
- VS1 is a vertical, multi-stage pump, sometimes called a vertical turbine pump

One design aspect for consideration by the operator will be seal arrangement in terms of single seals vs. dual seals. Fundamentally, all mechanical seals have a calculated leakage rate (typically given over a range) as a function of adequate seal face lubrication to mitigate wear and dissipate heat. In a single seal arrangement (one set of seal faces), the leakage from the process will move to the atmospheric pressure region within the seal assembly. The operator must consider the nature of the leakage, whether it is hazardous or toxic, and how increased levels of leakage should be managed. Seal arrangements defined by API 682 4th edition are as follows:

Arrangement 1 – Seal configurations having one set of seal faces per cartridge assembly.

A rather large population of pipeline applications utilize a single mechanical seal strategy in conjunction with a containment device, such as a segmented bushing on the atmospheric side of the primary sealing interface. The bushing functions to restrict leakage during normal operation and as the health of the primary seal assembly begins to deteriorate. The primary advantages of implementing an Arrangement 1 design include a simplex control and support system and reduced total cost of equipment ownership. Arrangement 1 configurations will typically be designated as 1CW-FL as outlined in the standard.

Arrangement 2 – Seal configuration having two sets of seal faces per cartridge seal assembly, with the space between the seals at a pressure less than the seal chamber pressure.

Pipeline operators utilize dual unpressurized seal arrangements in efforts to enhance containment capability of primary seal leakage. Implementation of a secondary end-face mechanical seal rather than a simplex bushing reduces the potential exposure of leakage to the atmosphere. While the dual unpressurized arrangement requires a more complex and costly support system, it allows for improved health monitoring and allows for routing normal leakage to an acceptable collection area distanced from the pump skid. The redundant secondary containment seal does not require a pressurized source for lubrication. The containment seal can be of either a wetted or dry-running design concept. Detailed design considerations for containment seals will not be covered in this tutorial; however, a more detailed overview with regards to containment seal strategies (contacting vs. non-contacting) can be found in proceedings from the 31st International Pump Users Symposium (Kalfrin/Gonzalez, 2015). Arrangement II configurations will typically be designated as 2CW-CW, 2CW-CS, or 2NC-CS as outlined in the standard.

Arrangement 3 – Seal configuration having two sets of seal faces per cartridge seal assembly, utilizing an externally supplied barrier fluid at a pressure greater than the seal chamber pressure.

Pipeline operators generally utilize dual pressurized seal arrangements to isolate the sealing interface from multi-fluid applications that require varying sealing concepts. The pressurized arrangement allows for a controlled and consistent environment for the seal assembly over the complete range of operation of the main pumping system. In addition, the dual pressurized arrangement allows for full isolation of the process fluid from the atmosphere. Total cost of ownership for this arrangement is significantly higher compared to Arrangements 1 and 2 due to the complexity of the control logic and associated maintenance to ensure system functionality.

Regardless of arrangement selection, the seal design for BB1/BB3 axially split pumps should incorporate a stationary flexible element in accordance with best practice. The rotating seat or RS design concept accommodates both misalignment between case halves and provides added face stability where surface speed on larger shafts exceeds the maximum speed capability of a traditional rotating flexible element design. Arrangement III seals will typically be designated as 3CW-FF or 3CW-BB as outlined in the standard.

PRESSURE RATINGS OF 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.

The exact pressure rating of a flange varies with materials, temperature, etc. and has changed somewhat over time but the above ratings still apply to many pipelines, especially in North America. Whereas piping and pressure vessels are fixed equipment with ratings that can be well defined by structural limits, mechanical seals are dynamic with both structural limits and performance limits. In addition, the performance of the mechanical seal is dependent on the fluid that is being sealed and the operation of the equipment. Moreover, the overall performance of the seal may even be time dependent as the physical properties of various components, especially elastomers, change during operation. Finally, performance of a seal is often path dependent; that is, the steady state performance depends on the startup conditions and procedure.

The advantage of operating at higher pressures can be easily shown by realizing that the minimum number of pumping stations can be estimated as the total pressure drop through the entire pipeline divided by the safe working pressure of the pipeline. This means that, for example, the number of pumping stations necessary to operate at a 1440 psig rating is roughly half the number required if the pressure rating is 720 psig. For most pipeline seals, the above ratings will be static pressure ratings because the seal is usually near pump suction pressure. That is, for most pipeline pumps, if the seal were actually operating at maximum pipeline pressure then the pump discharge pressure would be higher than the maximum rated pipeline pressure. However, for some pump types, notably vertical pumps, the seal may be located near the discharge pressure of the pump. Therefore, it is very important to know the pump type and to confirm the seal chamber pressure before selecting the seal and support system.

As noted, pumps used in pipeline applications are almost always described and defined by the API 610 pump standard. Prior to the issue of API 682 specification for mechanical seals, mechanical seal specifications were included in API 610. Beginning with the 8th edition, API 610 refers to back API 682 for seal specifications. Because API 610 pumps can be used in very high pressure services, it is possible for API 610 pumps to be outside the scope of API 682. Even so, API 682 is often specified for pipeline seals even though exact compliance may not be possible because of the limited scope of API 682. In particular, API 682 is limited to services less than 40 bar (600 psig) and shaft sizes less than 110mm (4.3 inch). Because many pipeline seals are outside the scope of API 682, those seals may not have been qualification tested per API 682.

In addition, before service related ratings can be provided for the mechanical seal, it is first necessary to define expected performance. Without a specification, a default should be provided by the seal manufacturer. The only available standard for mechanical seals, API 682 requires that, "All seals should operate for 25,000 hrs. without need for replacement while either complying with local emissions regulations, or exhibiting a maximum screening value of 1,000 ml/m3 (1,000 ppm vol) as measured by the EPA Method 21, whichever is more stringent." The performance of a mechanical seal can be thought of as a compromise between leakage and wear. It is well known that the wear rate (and friction) of a seal that is leaking excessively can be very low. In general, when leakage rates are reduced, wear rates increase. Within this relationship, and using modern designs incorporating high performance materials, nearly any level of performance can be obtained through selection of the seal arrangement. For purposes of illustration, this section is primarily concerned with the performance of a single mechanical seal although the concepts explained herein are also applicable to multiple sealing arrangements.

PRESSURE LIMITS APPLIED TO PIPELINE SEALS

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

> *Maximum static sealing pressure* 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* 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."

API 682 also defines the pressure casing as including the seal chamber but excluding the stationary and rotating members of the mechanical seal. This means that there is no requirement that the seal have the same maximum allowable working pressure as the pump. Obviously, but not stated, the seal is expected to have a static pressure rating equal to or exceeding the maximum static sealing pressure for that service. In addition, the seal is expected to have a dynamic pressure rating equal to or exceeding the maximum dynamic sealing pressure for that service; again, this requirement is not stated in API 682. The *Seal Ratings* section in the data sheets of API 682 includes places to record the *maximum static sealing pressure* and *maximum dynamic sealing pressure* at the pump temperature. There is some confusion about whether the intention is to supply the maximum pressures that might occur in that service (as noted in the definitions) or the rated values for the seal. Because of the location in the Seal Ratings section, the requested information is assumed to be the rated values for the seal. Unfortunately, API 682 does not provide guidance about how these ratings might be consistently determined. That subject is a common point of discussion when reviewing pipeline mechanical seal applications.

Static Sealing Pressure Rating

The static sealing pressure rating *(SSPR)* should be defined as:

> *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.*

Along with the SSPR, there should be a hydrostatic test pressure rating for the seal that should be defined as:

> *The highest pressure that the seal can withstand for thirty minutes at ambient temperature on water while the shaft is not rotating. Thereafter, the seal must maintain its static and dynamic sealing pressure ratings.*

These definitions establish the difference between static and hydro test ratings. They also insure that performance is not affected by static conditions within the limits of the static sealing pressure rating. In many cases, the static rating will be equal to the hydro test rating. This is not the normal practice for pressure vessels in which a ratio of 1.5 is employed between maximum allowable working pressure and hydro test pressure. However, the static pressure rating can be significantly below pressures that result in structural damage or rupture of the seal.

Maximum Allowable Working Pressure (MAWP)

The Maximum Allowable Working Pressure (MAWP is defined in API 682 as:

> *Maximum continuous pressure for which the manufacturer has designed the equipment (or any part to which the term is referred) when handling the specified fluid at the specified maximum operating temperature.*

The MAWP only applies to the pressure casing, which is defined as:

> *Composite of all the stationary pressure-containing parts of the seal, including seal chamber barrier or buffer fluid chamber, containment seal chamber and seal gland plate, and excluding seal ring, mating ring, bellows, sleeves, miscellaneous internal seal parts and atmospheric side gland connections which cannot be isolated from atmospheric pressure.*

The application of applying the MAWP of the pump to the mechanical seal is not accurate and is often a mis-communication during bid stages in new projects or re-rates of existing equipment. It must be understood what components of the mechanical seal are required to be rated to what pressure and within context of the actual seal operation what the implications may be by applying higher pressure values than necessary to the mechanical seal. Typically, true high pressure seal designs will require hard face material combinations to resist distortion and optimized balance ratios to mitigate contact pressure and wear. While these parameters might be requirements for a high pressure design, they would contribute to elevated leakage at low pressures. Simply put, applying an MAWP rating of 1440 PSIG (100 BARG) to mechanical seal faces when the actual seal operating pressure is 50 PSIG (3.5 BARG) would result in less than desirable seal performance.

Supplying a support system to the full pump MAWP and temperature rating is often impractical, as it could lead to unnecessary overdesign, when compared to the actual operating conditions. Not only that but it is quite common that non-mechanical parts are not available at pump MAWT or they would require an unnecessarily complex arrangement. Instrumentation specifications are given in Section 9 of API 682 2nd, 3rd and 4th edition. Clause 9.1.7 of 4th edition refers instrument temperature ratings to the normal operating temperature of the buffer/barrier system.

Dynamic Sealing Pressure Rating

The Dynamic Sealing Pressure Rating (DSPR) should be defined as:

> *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.*

When discussing dynamic ratings, the relationship between leakage and wear becomes particularly apparent. In mechanical seal technology, the classical, but simplistic approach to analysis of wear is by using the Pressure-Velocity (PV) value. The PV value is widely used as a guideline for mechanical seal design and application. The standard calculation is described in by Schoenherr as

$$PV = [\Delta P(b - k) + Psp)V$$

The PV value can be important because it represents both wear and heat generation. Even so, it has only a very rough correlation with overall seal performance. In the above equation, the pressure gradient factor, k, is taken as ½ according to Schoenherr. In actual service, the value for k may vary from 0 to 1. Nevertheless, it is convenient and the usual practice is to use k = ½ when dealing with PV calculations. Industry practice has been to limit the PV value based on seal face material combinations and to attempt to couple this limit with fluid properties. For example, a typical limit is 500 KPSI-FPM for resin filled carbons versus tungsten carbide for seals in "non-lubricating" liquids in typical pump services. Higher values (usually by a factor of 1.6) are allowed for "lubricating" liquids. Figure 1 gives some examples of the recommended maximum PV values for various material combinations.

Material Pair	Maximum PV Value (PSI-FPM)
Carbon vs Silicon Carbide (or Tungsten Carbide)	500,000
Carbon vs Alumina	100,000
Silicon Carbide vs Tungsten Carbide	350,000

Figure 1: Maximum PV based on two year wear life in non-lubricating service for plain faced seals.

Figure 1 is not meant to be an all-inclusive reference; it merely illustrates the concept and general order of magnitude of PV limits for a material pair. In working with PV limits, it is important to recognize the basis and limits of this concept. Much of the data is based on tests of a commercially available 3.375” unbalanced seal at 1800 RPM in warm water. The tests are run for 100 hours and the wear is measured. The PV limit is defined by either excessive wear for the target design life or by damage due to some sort of “overload” -- usually thermal in nature. Some historical data at both 1800 and 3600 rpm indicates that the maximum recommended PV is not constant with respect to speed. In some cases it appears that the allowable PV value actually increases with speed. For example, to obtain a projected two year wear life, a particular combination of carbon and tungsten carbide was rated for a PV of 300 KPSI-FPM at a velocity of 1820 FPM, but 525 KPSI-FPM at 3600 FPM. The absolute maximums, based on face damage, were 467 and 587 KPSI-

FPM respectively. This type of relationship is consistent with hydrodynamic load support during the PV test.

When PV limits are used as the basis for wear, it is important to recognize that the maximum recommended values are often based on a *two year* life *with a 0.125" wear length.* This is 0.00071” (0.018 MM) wear during the 100 hour test -- a difficult measurement to obtain accurately. If a longer wear life is required, then the PV limit must be lowered. If the available seal wear length is less than 1/8" then either the PV limit must be lowered or a lesser wear life is projected. These are extreme extrapolations -- a 17,520 hour wear life is projected from a 100 hour test. In the 100 hour test, most of the wear undoubtedly occurs during the first few hours of the test. In actual long term service, mechanical seals rarely wear out.

It is important to realize that materials are available with higher (and lower) PV ratings than shown in Table 1. Seals with very good heat transfer designs may operate at higher PV values -- just as seals with poor heat transfer designs may be useful only at low PV values. Also, seals with special face design features such as recesses or grooves, may operate at higher PV values than indicated by the nominal PV calculation.

MATERIAL CONSIDERATIONS FOR PIPELINE SEALS

Comprehensive mechanical seal descriptions and properties are beyond the scope of this tutorial. 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 in order 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 commonly used; Antimony impregnated carbon is the default carbon identified in API 682 when considering use in a light hydrocarbon service. Antimony carbon is commonly used in various services in midstream pipelines. Specialized Nickel impregnated grades have also been used widely with good success, although lead times and costs of those materials make them less desirable options.

  Although antimony filled carbon-graphite primary ring 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 in order 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 also deficiencies in the seal performance from excessive distortion due to pressure. The majority of the carbon seal faces utilized in midstream pipeline applications employ engineered ring geometry and configurations to mitigate pressure induced distortion and optimize face contact pressure. The authors of this tutorial have extensively applied engineered carbon face geometry in these pressure ranges.

- Sintered / Reaction-Bonded Silicon Carbide - Silicon carbide is an advanced ceramic material. The earliest type of silicon carbide available for use in mechanical seals was reaction bonded and developments have made a number of variations available. 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 due to the exceptional PV characteristics of the material pairs. The ceramic materials are typically run against one another (dissimilar grades) in high viscosity applications such as crude oil. Robust drive designs for seal rings of this material are recommended 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. The resultant properties are dependent upon the tungsten matrix and percentage of binder (typically 6 to 12% by weight per volume). 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 will be limited in PV characteristics and 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. Grades are also available with a network of non-interconnecting pores, which entrap fluid to support hydrodynamic lubrication. These materials offer exceptional PV characteristics when paired with a corresponding advanced ceramic material. 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.

Adaptive Hardware

Predominantly stainless steels are used in the manufacture of mechanical seals. Other exotic alloys are also used on application where compatibility of seal material with the process fluid is to be considered. Stainless steel is advantageous for metallurgy construction as it has increased nickel content and the addition of molybdenum (Mo) to improve corrosion resistance, particularly in applications containing chlorides. Type 316SS (UNS S31600) contains 16 to 18% chromium, 10 to 14% nickel and 2 to 3% molybdenum. Type 316 and 316L (UNS S31603) are the most often used austenitic stainless steel in mechanical seals and are fairly common in midstream pipeline services.

Duplex stainless steels (UNS S31803) are magnetic and offer increased tensile and yield strength over austenitic stainless steels with improved resistance to localized corrosion, particularly pitting, crevice corrosion and stress corrosion cracking. These materials combine the beneficial properties of both ferritic and austenitic steels, resulting in a ferritic-austenitic structure with approximately 40-50% ferrite. Carbon levels are kept below 0.03% and chromium levels are usually high 21.0 to 26.0%, with nickel from 3.5 to 8.0%. They may contain molybdenum up to 5.0% and nitrogen 0.05 to 0.32%.

Adaptive hardware components, particularly sleeves and sub-sleeves benefit from being manufactured from Duplex materials especially in the context of midstream pipeline applications due to the increased material resiliency especially when considering exposure to pressures up to 2160 PSIG (149 BARG). Large diameter components, beyond 4.33” (110 MM), are recommended to be manufactured from Duplex materials when applied to midstream pipeline applications.

Secondary Sealing Elements

Specific elastomer usage varies in midstream pipeline applications depending on the process fluid and potential contaminants; the specific materials by process will be discussed later in this tutorial. Outside of material compatibility with the process fluid, for pressures above 1440 psig, a 90 durometer compound is recommended for the dynamic O-ring in Type A seals 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)

Drive collar

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. An alternative drive/thrust collar arrangement, such as a clamp-ring design has also been used extensively in midstream pipeline applications. Figure 2 high-lights some common drive collar arrangements used in midstream pipeline applications. 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. Note that the type, 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.

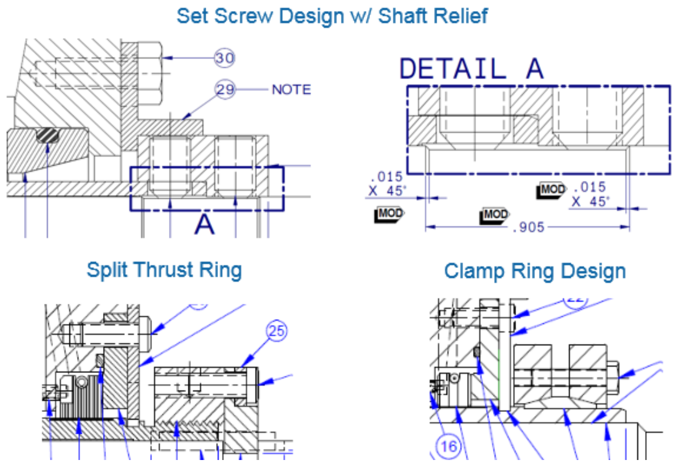


Figure 2: Seal Drive Collar Variations

VARIABLE SPEED OPERATION INFLUENCE ON MECHANICAL SEALS

Proper lubrication of mechanical seal faces is essential to prolonged reliability of the mechanical seal. In the case of many of the pipeline services discussed, the volatility of the process fluid in question makes determination of the flush rate and evaluation of the flush design even more critical. 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 commonly 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. The temperature rise can be calculated using the following equation:

$$\Delta T = (60000 \times P) \div (q \times sg \times cp) \quad \text{(metric units)}$$
$$\Delta T = P \div (q \times 500 \times sg \times cp) \quad \text{(imperial units)}$$

Where:

ΔT = temperature differential or temperature rise expressed in °K (°F)
P = heat load expressed in kilowatts (KW) or (Btu/hr)
q = flow rate expressed in liters per minute or US gallons per minute
sg = specific gravity of the fluid (dimensionless)
cp = specific heat capacity of the fluid expressed in joules per kilogram Kelvin (J/Kg·K) or (Btu/lb-°F)

For example, with a total heat load of 3 kilowatts (10,245 Btu/hr) the required flow rate for water with an allowable temperature rise of 8°C (15°F) would be 6.8 lpm (1.8 gpm), but for propane the required flow rate would be 68 lpm (18 gpm) based upon an allowable temperature rise of 3°C (5°F). Generally speaking, defaulting to a greater flow rate will improve seal life as the temperature rise in the seal chamber will be lower and therefore a greater margin of safety will be achieved, especially when the fluid being sealed has a vapor pressure greater than atmospheric pressure.

Keeping in mind these critical parameters, 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. 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 more or less 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, however, can be detrimental to the mechanical seal if not considered in the design and application of the seal and associated support system. Relating back to the heat load discussion, 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, 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.

The differential pressure required to drive the seal flush flow will also be influenced by the variation in the shaft speed. Recall from pump Affinity Laws the following relationship:

$$H_2 = H_1 \left(\frac{N_2}{N_1} \right)^2$$

Where:

H = differential head (expressed in feet or meters)
N = shaft speed

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. Based on this relationship, what becomes the worse-case operating condition for the mechanical seal? The concept may be best illustrated by a real-life example from a midstream pipeline application in Y-grade NGL service. The process evaluation considered the following variables:

- 4.500” dual mechanical seal (wet contacting inner seal with dry running, non-contacting containment seal)
- 12x12x16, 3 stage API 610 style BB3 pump
- Suction Pressure conditions between 500 and 850 PSIG
- Fluid Specific Gravity = 0.46
- Fluid Specific Heat = 0.595 Btu/lb-°F

The end user in question was concerned regarding the performance of their mechanical seal, including premature wear of the inner seal carbon face. The mechanical seal was supported by a filtered API Plan 11 flush, with the flush sourced from the first stage discharge pressure region on the pump casing. Based on the pump configuration, the actual seal chamber pressure would be very close to the suction pressure conditions, assuming pump internals components including hydraulic thrust balancing mechanisms were not in a worn state. The end user varied the operating speed of the pump anywhere from 1450 RPM up to potentially 3600 RPM. There was a cause for concerning regarding the restriction orifice (RO) in the seal flush line as being sufficient to pass the required flush flow rate across the operating range. The existing orifice diameter supplied with the filtered flush support system was Ø 0.187” (4.7 MM) and the systems were supplied across multiple sites on the pipeline in question. The cross section of the mechanical seal is shown in Figure 3.

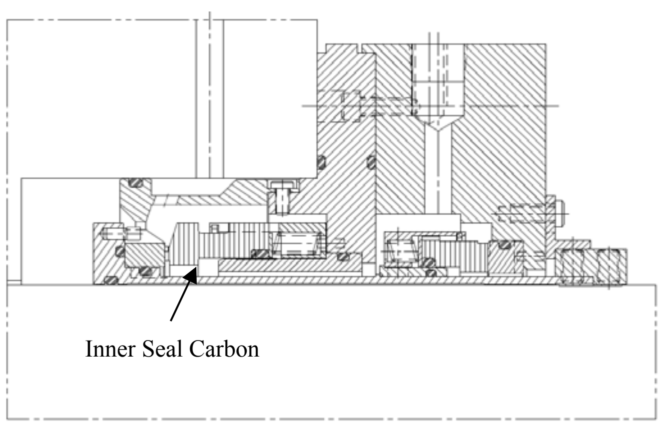


Figure 3: NGL Mechanical Seal Design

The evaluation of the variable speed influence focused on the associated heat load generated by the mechanical seal and the subsequent cooling flow requirements at varying speed intervals, keeping in mind the suction pressures remaining relatively constant depending on conditions. Figure 4 shows the varying heat load requirements assuming the initial 500 PSIG suction pressure condition.

Speed (RPM)	Total Differential Pressure (psig)	1st Stage Discharge Pressure (psig)	Total Heat Load (Btu/hr)	Seal Flush Required (5°F Temp Rise)	Flush Rate (.187" Orifice) 3/4" X 095WT Tubing
1400	120	40	4388	6.1	6.1
1450	129	43	4505	6.3	6.4
1800	198	66	5534	7.71	7.9
2100	270	90	6456	9	9.2
2400	353	118	7378	10.5	10.5
2700	446	149	8301	11.5	11.9
3000	551	184	9223	12.8	13.2
3300	667	222	10145	14	14.5
3600	793	264	11068	15.4	15.8

Figure 4: Seal Heat Load and Flush Flow at 500 PSIG in NGL Service

In this instance, it is apparent that the cooling requirements of the mechanical seal are being met with the existing support piping and orifice size in place. When comparing against the higher suction pressure condition, there are visible concerns. Figure 5 shows the varying heat load requirements assuming an 850 PSIG suction pressure condition.

Speed (RPM)	Total Differential Pressure (psig)	1st Stage Discharge Pressure (psig)	Total Heat Load (Btu/hr)	Seal Flush Required (5°F Temp Rise)	Flush Rate (.187" Orifice) w 3/4" .095WT Tubing
1400	120	40	6660	9.3	6.1
1450	129	43	6898	9.6	6.4
1800	198	66	8563	12	7.9
2100	270	90	9990	14	9.2
2400	353	118	11418	16	10.5
2700	446	149	12845	18	11.9
3000	551	184	14272	20	13.2
3300	667	222	15700	22	14.5
3600	793	264	17450	24	15.8

Figure 5: Seal Heat Load and Flush Flow at 850 PSIG in NGL Service

When evaluating the higher suction pressure condition, there are apparent concerns with regards to the temperature rise in the surrounding fluid in the seal chamber adjacent to the faces. Operation beyond this 5° recommended threshold increases the potential for vaporization of the flush fluid at the seal face outer diameter. Recall also that the temperature rise in the surrounding fluid speaks nothing to the actual seal face temperature. Increased temperature in the flush fluid subsequently translates to minimized fluid film support between the seal faces, increased frictional heat generation, accelerated face wear, and reduced seal life. When the results above are plotted graphically, the step-change in flush flow requirements are more visible. Figure 6 illustrates these results graphically.

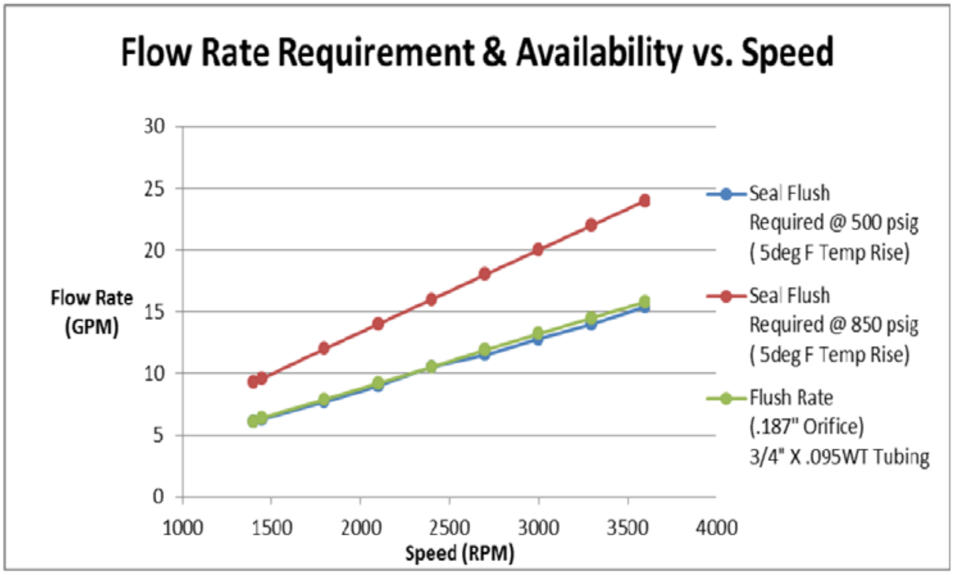


Figure 6: Flow Rate Requirment vs Speed

This example illustrates the worse-case scenario for the mechanical seal in this instance, which is actually operation at slow speed with the higher chamber pressure. In this case, the recommendation that was acceptable to the end user was to increase the size of the restriction orifice diameter in the flush support piping to satisfy the flow requirements of the mechanical seal at the slower speeds. This modification was selected as it was the least intensive to the existing pipework in terms of modification. The only potential drawback of this modification is that there will be a substantial increase in flow to the mechanical seal faces at higher speeds. Some midstream pipeline operators will specify seal flush sourcing from discharge regions based on the pump configuration to compensate for variable speed operation. One example would be to select sourcing of the flush from full discharge if the pump is less than 4 stages and from the high pressure cross-over portion on the casing if the pump is 4 stages or greater. This allows some degree of flexibility in orifice sizing as the available differential pressure will typically be sufficient across a variable speed operating range. Keeping this in mind, it is important to review the design of the mechanical seal internal flush porting. Efficient porting design ensures not only even flow distribution to the mechanical seals but also produces manageable flow velocities that will not induce erosion of components or create vortices that will hinder seal face lubrication.

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 7 outlines an example of a large diameter seal in a viscous oil service. Both simulations depict operation at 950 PSIG @ 900 RPM; the simulation on the left is post modification to the face geometry to mitigate the pressure induced distortion effects, subsequently improving upon face generated heat and leakage.

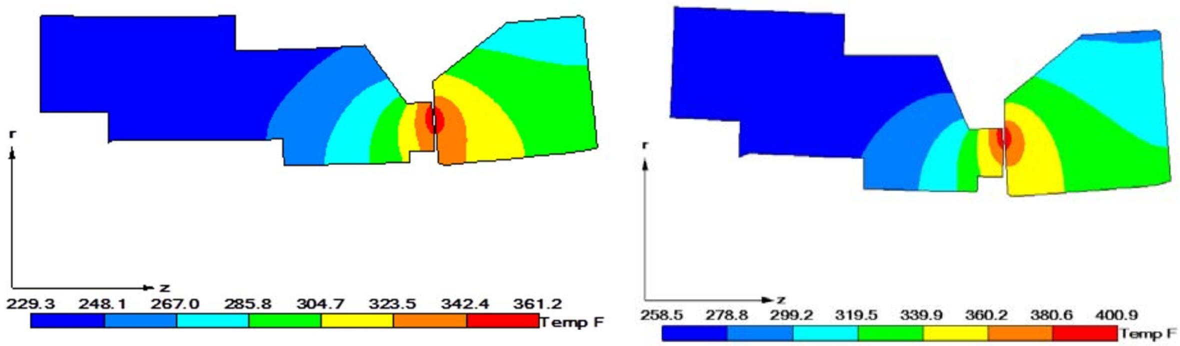


Figure 7: Geometry Modification for Pressure Induced Distortion Mitigation

One final comment associated with variable speed operation relates to the usage of API Plan 31. Plan 31 is a variation of API Plan 11 in which a cyclone separator is added to the flush line. In API Plan 31 the product, containing heavy, solid abrasives, is introduced to the cyclone separator from a discharge pressure region of the pump casing. The cyclone separator uses centrifugal forces to concentrate the solids into one stream and therefore produces a clean stream as well. The clean fluid is routed out the top of the separator and into the seal chamber. The stream having concentrated heavy solids is routed back to suction. For cyclone separators to be effective, the pressures on each respective outlet must be balanced to achieve the desired flow. At times, orifices may be added to respective clean and dirty outlets to bias flow to the mechanical seal. Cyclone separators can be very practical provided certain conditions are met – one of which is a constant speed. If the speed is varied with an API Plan 31 installation, the subsequent pressure balance across the cyclone separator will change as well. In these instances, there would likely be occurrences where the mechanical seals suffer from reduced flow rates for cooling and lubrication.

Additionally, in volatile fluid applications, the cyclone separators may become prone to vapor lock which can further complicate the operation of the piping plan. Cyclone separators function efficiently when they are thoroughly vented – air / vapor at the top of the cyclone will hinder performance. In high vapor pressure applications, if more bubbles form at any point due to lighter constituents flashing off in the mixture, those bubbles would be more inclined to be channeled into the seal flush connections and eventually lead to dry running of the seal faces. This can be further complicated if unpredictable vortices form within the cyclone due to the pressure imbalances that can occur due to variances in differential pressures. In variable speed applications where there is a concern over abrasive contamination of the mechanical seal, a recommended alternative to a cyclone separator would be to utilize a filtered API Plan 11 seal flush or API Plan 99 (12). The filtration system should be rated for the application pressure, total seal flush flow requirements, and be equipped with double block and bleed capability for isolation and online element replacement. Figure 8 is an example of a duplex filtration system that has been successfully used in an NGL application where traditionally cyclone separators would have been supplied. This particular configuration includes an integral magnetic separation system to aid in isolating pipe scale from sensitive areas of the mechanical seal.

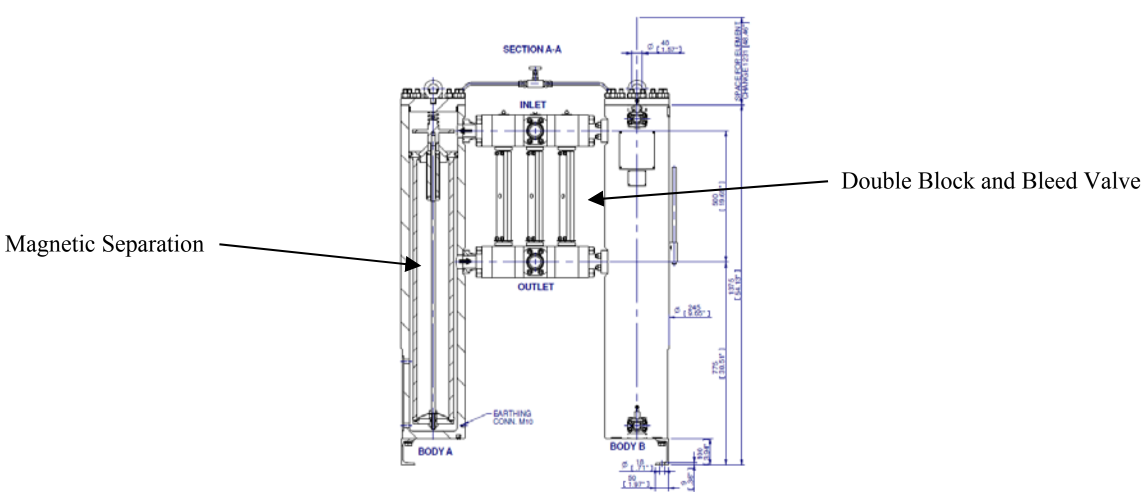


Figure 8: Duplex Filtration System with Double Block and Bleed, Magnetic Separation

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*, h*eavy* 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:

$$sg@60F = \frac{141.5}{API + 131.5}$$

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, the majority of 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 the following design parameters to minimize seal leakage:

- Balance ratio
- Face width
- Nominal face pressure
- Face material combinations relative to Pressure-Velocity limits (P-V)

Figure 9 below represents actual field applications that follow the recommended sealing arrangements within this tutorial.

Fluid	RPM	Seal Size	Temp (F)	Suction Pressure (psig)	Discharge Pressure (psig)	Specific Gravity	Visc. (cP)	Vapor Pressure (psia)	Mode of Operation (Pumps)
Crude Oil API Gravity < 21	790 – 1920	6.375”	90.0	50.0 – 1200.00	1480.0	0.83 - .93	350.0 – 1000.0	< 15.0	Series (4)
Crude Oil API Gravity 45	700 – 1800	4.760”	90.0	100.0 – 500.00	1200.0	0.825	7.0 – 11.0	< 15.0	Series (2)
Crude Oil API Gravity 21/30/45	900 – 1800	6.375”	60.0	60.0 – 950.0	1379.0	0.81	5.0	< 15.0	Series (3)
Crude Oil API Gravity 21/30/45	720 – 1800	7.125”	65.0	70.0 – 1278.0	1564.0	0.92	75.0 – 350.0	< 15.0	Series (4)
Crude Oil API Gravity 21/30/45	900 - 1800	6.375”	80.0	60.0 – 1100.0	1440.0	0.73 - .93	5.0 – 180.0	<15.0	Series (3)

Crude Oil API Gravity 21/30	3580	4.250”	50.0	450.0 – 900.0	1300.0	.83 - .93	6.8 – 500.0	8.1	Series (2)
Crude Oil API Gravity 45	2000 – 3560	4.125”	68.0	50.0 – 150.0	358.0	0.817	2.57	12.0	Parallel (2)

Figure 9 – Typical Crude Mainline Pump Applications

Single Seal Installations

The vast majority of mechanical seals in crude oil pipeline services are single cartridge arrangements using a Type A seal assembly as outlined in API 682 4th Edition. Where pumping equipment is subjected to high percentage of solids and shaft movement, an alternative Engineered Seal assembly can be implemented providing a non-pusher secondary sealing element in the critical sealing location. Type B seals are typically not an option due to their limited pressure capabilities. When utilizing single seal arrangements, various containment arrangements have been installed in accordance with specifications defined by the pipeline operator.

Seal face materials of choice typically default to a hard face combination, with options to include reaction-bonded silicon carbide, alpha-sintered silicon carbide, tungsten carbide or graphite-loaded silicon carbide. Where antimony filled carbon is utilized, establishing the correct face geometry will be critical to minimize face distortion in higher pressures, while limiting the exposure to higher viscosity oils that result in face blistering. 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. The use of Perfluoroelastomer secondary seals is an option when dealing with applications containing agents such as H2S or amines. The choice of hard face materials eliminates the requirement of having to filter or clean the primary flush stream in support of the primary interface. 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.

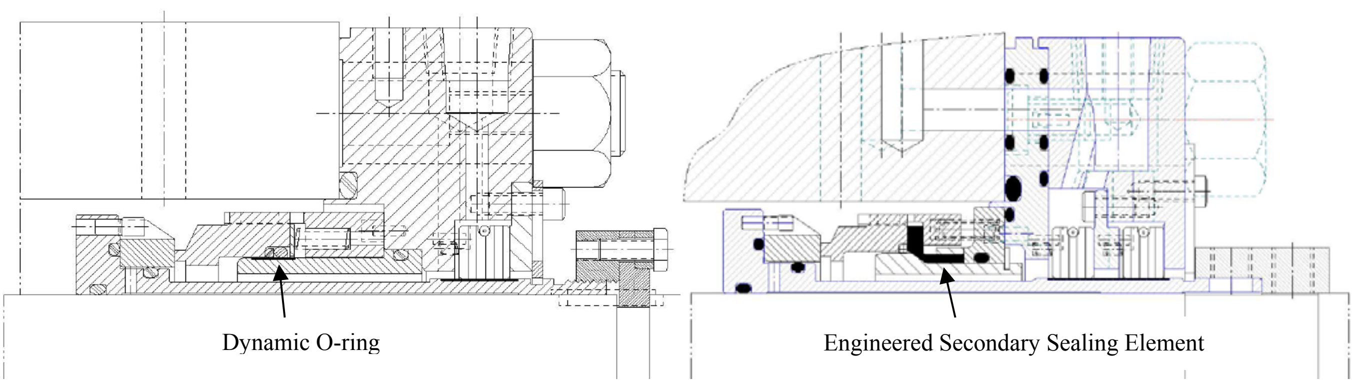


Figure 10. Type A with API Piping Plan 11/65 (left); Engineered Seal with API Piping Plan 11/66A (right)

Dual Seal Installations

Pipeline operators have deployed dual unpressurized sealing options where high levels of H2S are present or where a more robust containment sealing strategy is required. The preferred arrangement when utilizing a dual unpressurized seal is 2CW-CS, utilizing methods 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.

Dual unpressurized arrangements with 2CW-CW have also been utilized in crude oil pipeline services, but are often not recommended. This is primarily due to the lack of utilities and prevenative maintenance associated with the API Plan 52 buffer fluid. Due to the inherent dirty nature of crude oil, buffer fluid contamination and eventual degradation often leads to poor containment seal reliability. When considering normal primary leakage and potential for loss of buffer fluid level through the secondary containment sealing interface, maintenance tasks to include frequent draining and replenishment of the buffer fluid become a nusiance to the pipeline operator.

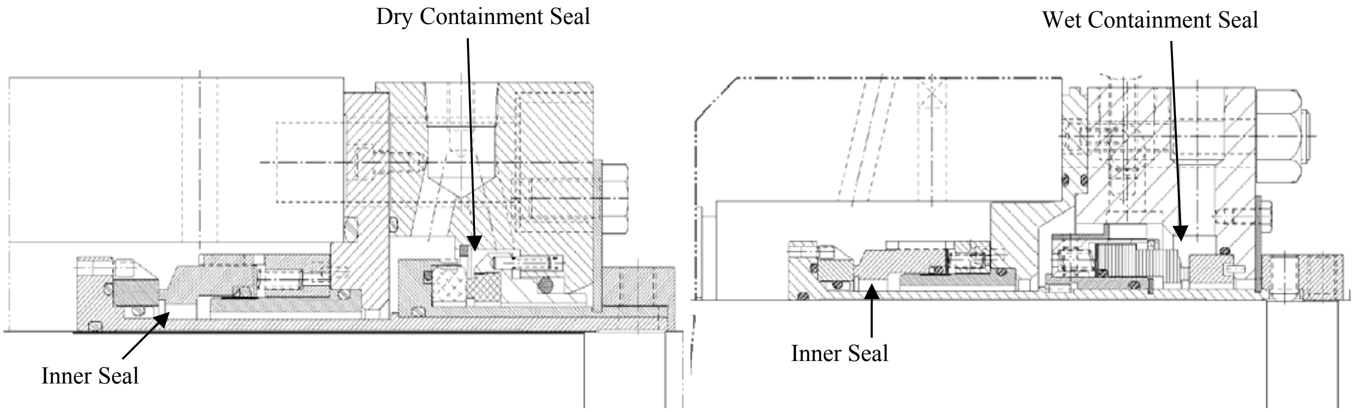


Figure 11: 2CW-CS Type A with API Piping Plan 11/75 (left); 2CW-CW Type A with API Piping Plan 11/52 (right)

Dual pressurized configurations defined as 3CW-FF are typically 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 3CW-FF sealing arrangement allows for isolation of the primary sealing components from the varying process fluid conditions and thus provides a more stable interface.

Where 3CW-FF units are installed, consideration should be given to maintaining a rotating-seat (RS) design concept, implementation of flow guides for the pressurized source, and an ID pumping scroll to displace solids from migrating to the flexible element. See Figure 12 for a recommended configuration.

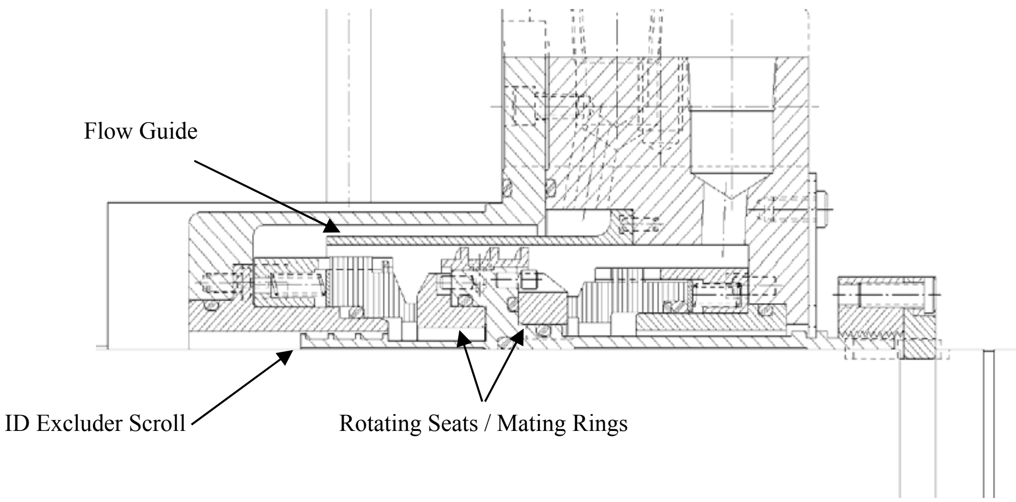


Figure 12: 3CW-FF Type A with API Piping Plan 54

REFINED PRODUCTS PIPELINE SEALING STRATEGIES

When reviewing pipeline applications defined as refined products within this tutorial, 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. Figure 13 represents actual field operating conditions and fluid compositions in today's pipeline industry that have been successfully sealed with extended reliability utilizing the methods recommended within this tutorial.

Fluid	RPM	Seal Size	Temp (F)	Suction Pressure (psig)	Discharge Pressure (psig)	Specific Gravity	Visc. (cP)	Vapor Pressure (psia)	Mode of Operation (Pumps)
Diesel	1800	5.750"	90.0 - 100.0	50.0 / 250.0	950.0	.70 - .85	.5 - 3.0	< 15.0	Series (2)
Gasoline Jet Fuel	2200 - 3600	3.500"	70.0	145.0	1135.0	0.73	0.3	7.0	Series (2)
Gasoline	3870	3.260"	70.0	30.0 / 200.0	1453.0	0.73	0.57	3.0	Series (2)
Gasoline Diesel	2500 - 3760	2.635"	60.0	50.0 / 250.0	1775.0	0.86	3.0	N/A	Series (2)
Gasoline	3560	4.250"	60.0 - 70.0	50.0 / 444.0	874.0	0.74	1.0	< 15.0	Series (2)

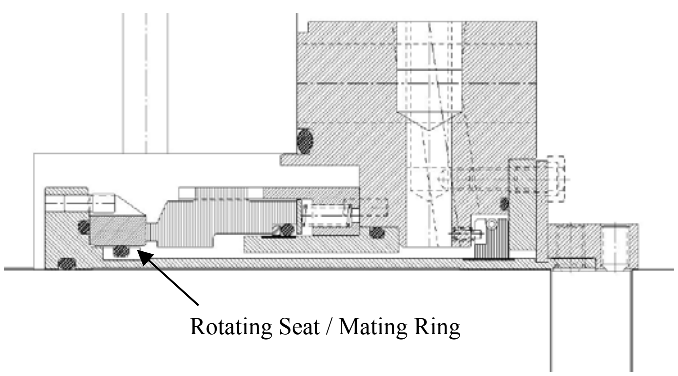
Figure 13: Typical Refined Products Mainline Pump Applications

Single Seal Installations

The vast majority of mechanical seals in refined products pipeline services are single cartridge arrangements, identified as 1CW-FL configuration and utilizing a Type A seal. Seal materials of choice typically default to antimony impregnated carbon paired against a hard face mating material, 316SS hardware and Fluoroelastomer secondary sealing elements.

In support of the primary sealing interface, a simplex 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 with a cyclone separator to assist with removing solids from the flush stream.

The majority of 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 particular 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.



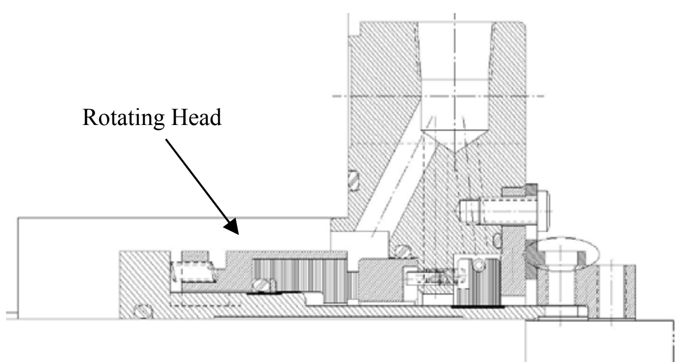


Figure 14: 1CW-FL Type A (RS) with Piping Plan 11/65 (left); 1CW-FL Type A (Non-RS) with Piping Plan 11/65 (right)

Dual Seal Installations

As previously identified within this tutorial, 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 2CW-CS, utilizing methods outlined with Arrangement 1 options in addition to a dry-running containment seal and large volume collection reservoir. The use of 2CW-CW in this particular application is typically not recommended due to the previous considerations related to maintenance requirements, buffer fluid contamination, and lack of support utilities on-site.

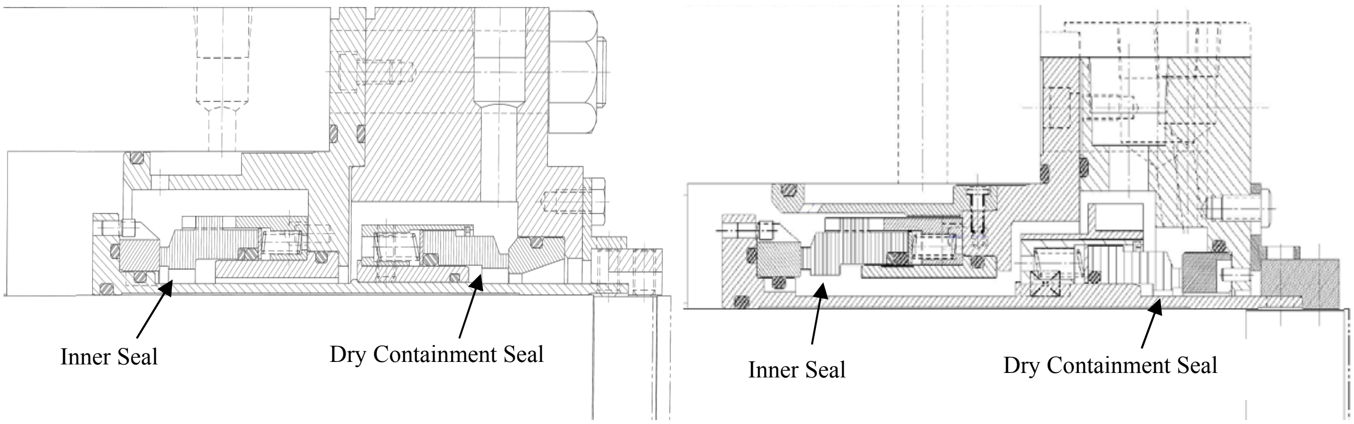


Figure 15: 2CW-CS Type A with API Piping Plan 11 / 75 (left); 2CW-CW Type A with API Piping Plan 11 / 52 (right)

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

VOLATILE / FLASHING HYDROCARBON PIPELINE SEALING STRATEGIES

For the purpose of this tutorial, we will focus on the NGL liquids that are critical components of the downstream refinery, petrochemical and specialty chemical processing industries. The raw NGL's or "Y-Grade" variations, are byproducts from the upstream natural gas gathering of various wellheads and the subsequent gas purification and separation processes. Once separated from the natural gas resource, the mixture of flashing hydrocarbon components are then transported to fractionation plants to further separate the hydrocarbons into their base components via a boiling process. Once the various hydrocarbon components have been separated, they are then moved to storage. Hydrocarbon components are transported based on demand via pipeline pumping stations from the natural gas processing facilities to fractionation plants and refining destinations.

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 .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.

Noted in a tutorial from the 32nd Pump Symposium, the requirements for effective sealing of very light hydrocarbons in terms of the mechanical seal is a maximization of both seal face stability and lubrication. Mechanical seals operating in the services described will 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. Further, balance ratio, which is a dimensionless value associated with closing and opening areas of the seal face geometry, must be optimized to minimize the face generated heat and loading in order for the seal to have a reasonable chance to survive (Kalfrin 2016).

Figures 16 and 17 below represents actual field operating conditions and fluid compositions in today's pipeline industry that have been successfully sealed with extended reliability, utilizing the methods recommended within this tutorial.

Fluid	RPM	Seal Size	Temp (F)	Suction Pressure (psig)	Discharge Pressure (psig)	Specific Gravity	Visc. (cP)	Vapor Pressure (psia)	Mode of Operation (Pumps)
Butane Isobutane	1800 - 3600	4.250"	70.0 / 90.0	425.0 / 925.0	1250.0 / 1350.0	.55- .58	0.14	40.0	Series (2)
Polymer Grade Propylene	2200 - 3600	5.250"	95.0	700.0	1620.0	0.502	0.06	212.5	Parallel (2)
Propane	3570	4.250"	95.0	236.0/ 375.0	1370.0	0.48	0.07	186.0	Parallel (2)
NGL / Y-Grade	1450 - 3650	4.500"	60.0 / 85.0	500.0 / 900.0	1460.0	0.48	0.09	300.0	Series (3)
NGL / Y-Grade	1800 - 3600	4.500"	70.0 / 80.0	375.0 / 900.0	1350.0	0.50	0.1	240.0	Series (2)
NGL / Y-Grade	1450 - 3650	5.375"	80.0	1330.0	1750.0	0.48	0.09	300.0	Series (3)
NGL / Y-Grade	1800 - 3600	5.125"	60.0 / 85.0	450	1050	0.45	0.1	396.0	Series (2)

Figure 16: Typical Flashing Hydrocarbon Mainline Pump Applications

Component	Y-Grade #1 % Composition	Y-Grade #2 % Composition	Y-Grade #3 % Composition	Y-Grade #4 % Composition
CO2	0.0	0.0	0.0	0.0
Methane	0.2	0.0	0.2	0.0
Ethane	31.2	6.4	23.3	51.8
Propane	36.5	54.4	43.2	25.8
Isobutane	6.3	11.1	6.4	3.8
Butane	14.5	16.4	16.6	8.2
Isopentane	3.8	4.6	3.5	0.1
Pentane	3.7	3.4	3.9	10.3
Hexane/C7+	3.8	3.7	2.9	0.0
Vapor Pressure (psia)	330.0	175.0	214.9	349.0

Figure 17: Y-Grade Compositions

The vast majority of mechanical seals in flashing hydrocarbon pipeline services are identified as an Arrangement 2, 2CW-NC or 2CW-CS configurations and utilizing a Type A seal configuration, though Arrangement 1 identified as 1CW-FL are also utilized within the industry. The seal design concepts for flashing hydrocarbon service, whether utilizing a single or dual arrangement, will consist of similar design criteria for the inner seal. These enhancements can include 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. Seal materials of choice typically default to antimony impregnated carbon paired against a hard face mating material, 316 or Duplex stainless steel hardware and Fluoroelastomer secondary sealing elements.

Single Seal Installations

Due to the volatile nature of the fluid with normal leakage present in the form of a vapor, a rigorous monitoring and support system should be implemented. The typical Arrangement I offering should utilize a tight tolerance bushing with a vent connection 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 cavity, triggering an alarm.

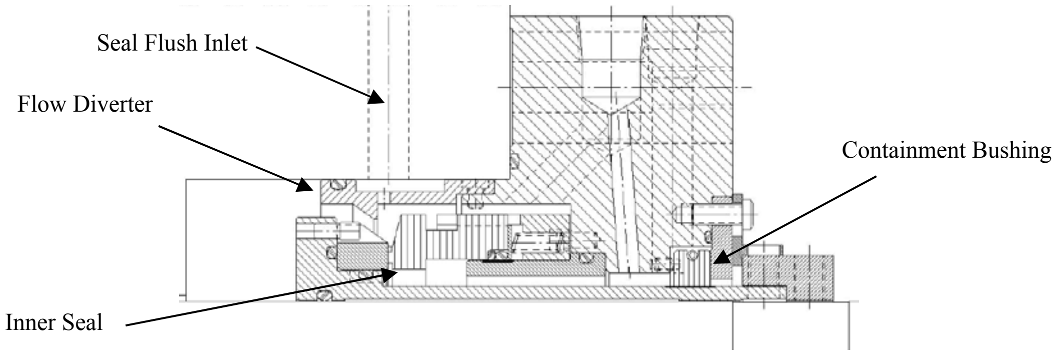


Figure 18: 1CW-FL Type A with API Piping Plan 11 / 66B

Dual Seal Installations

The preferred arrangement when utilizing a dual unpressurized seal in flashing hydrocarbon services is 2CW-NC or 2CW-CS, utilizing design criteria outlined with Arrangement 1 options, but including a containment seal. Though both contacting 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. In support of Arrangement 2 in flashing hydrocarbon services, API Plan 76 is the most common support system, routing primary leakage to an acceptable vapor recovery system through a control panel. API Plan 75 offers flexibility in managing both vapor and liquid leakage in heavier hydrocarbon applications where the inner seal leakage may condense.

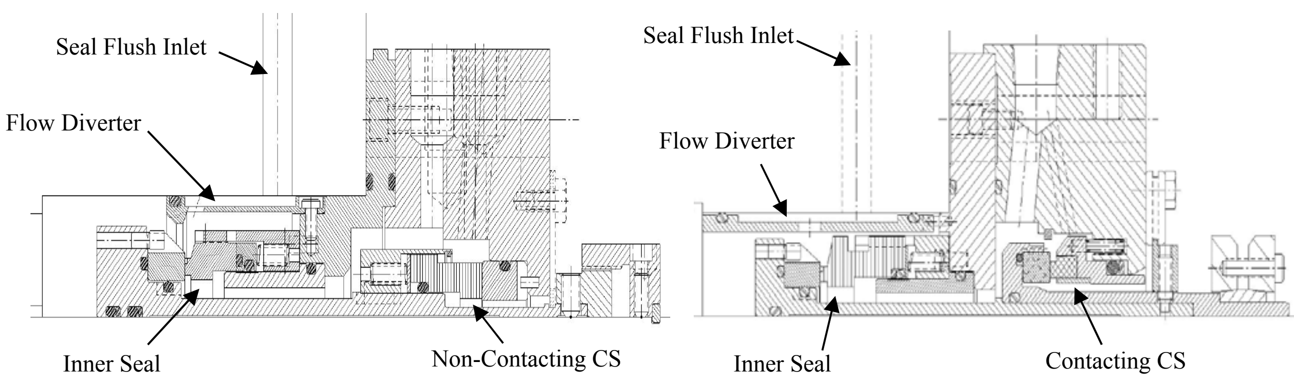


Figure 19: 2CW-CS Type A with API Piping Plan 11 / 76 (left); 2CW-CS Type A with API Piping Plan 11 / 75 (right)

SUPERCRITICAL FLUID PIPELINE SEALING STRATEGIES

Similar to the flashing hydrocarbon services previously outlined in this tutorial, pipeline applications with supercritical fluids like ethylene, ethane and CO2 are exposed to the same challenges related to seal design. 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. 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 and any temperature above 49.82 ºF, 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 (2NC-NC) and dual pressurized (3CW-FF) arrangements.

Dual Unpressurized Seals

Dry running non-contacting gas seals 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 a antimony carbon paired against premium grade tungsten carbide, while hardware and secondary sealing elements are consistent with materials described for flashing hydrocarbons. 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. 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)

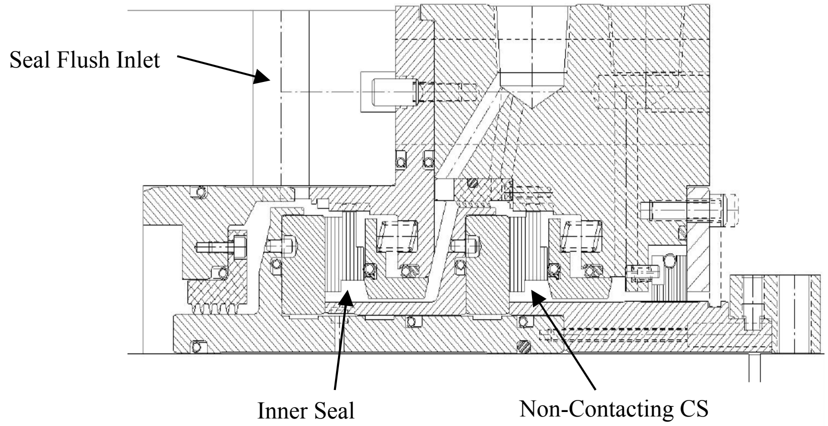


Figure 20: 2NC-NC (Engineered Seal) with API Piping Plan 99 (12) / 76

Dual Pressurized Seals

The alternative to the dry running non-contacting option is to deploy a 3CW-FF arrangement with an external pressurized source to isolate the seal interface components from the supercritical fluid. The Arrangement III design considerations and support system has been outlined previously within this tutorial; however, supercritical fluid applications can operate at significantly higher pressures and require an even more intricate support system. 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 2NC-NC arrangement, the presence of a supercritical fluid within the operating parameters typically dictates that a dual pressurized system be utilized.

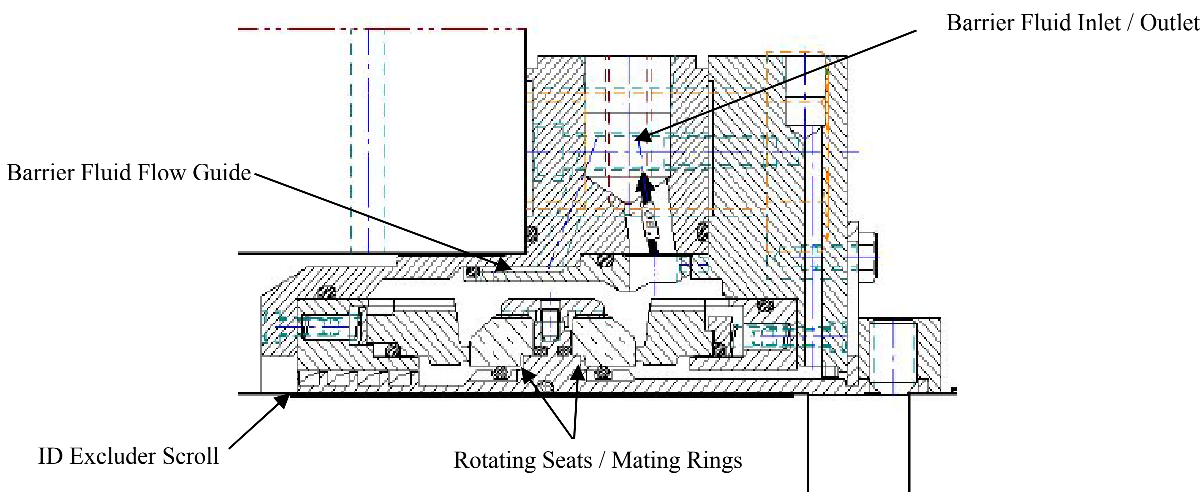


Figure 21: 3CW-FF Type A with API Piping Plan 54 or 53B

PIPING PLAN SUMMARY

Commonly used API Piping Plans in the midstream pipeline sector will be discussed briefly in the following section. The overviews provided are in no means comprehensive. Several of the piping plans discussed have been covered in greater detail in previous tutorials presented at the Texas A&M International Pump Users Symposium and those documents can be found by reviewing previous symposium proceedings.

API Plan 11

Plan 11 is the most common piping plan in the pipeline industry, utilized on a variety of process fluid applications. The primary objective of Plan 11 is to provide adequate cooling and lubrication to the primary sealing inteface of either an Arrangment 1 or Arrangment 2 installation. Typically defined as a discharge bypass, Plan 11 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. The pipeline operator should work closely with their seal manufacturer to determine the suitable flow rate to maintain a predetermined allowable flush temperature rise. This value will be based on total heat load associated with face frictional heat, viscous shearing and heat soak components, typically measured in BTU/hr.

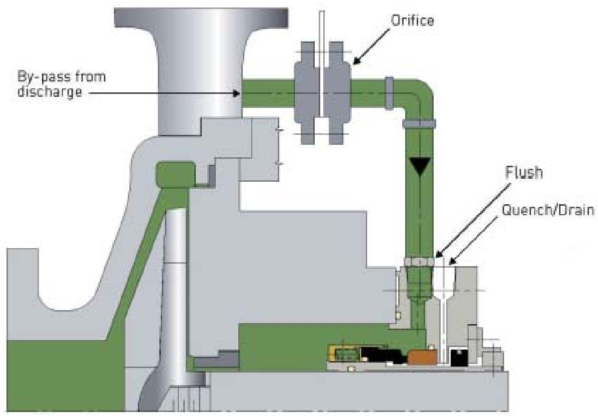


Figure 22: API Plan 11 Diagram

Plan 11 is easily maintained with flow control typically through an orifice or choke tube and properly sized piping (or tubing). In some cases, Plan 11 will be used in conjunction with an internal throat bushing to assist with building seal chamber pressure as an effort to enhance vapor pressure margin. In addition, Plan 11 utilizes the process fluid for mechanical seal lubrication, thus there is no issue of product contamination.

The disadvantange of utilizing Plan 11 is that the potential presence of pipeline rouge, sand and various foreign particulate can be abrasive to critical sealing surfaces and lead to potential clogging of the flexible element. Furthermore, pipeline equipment continues to grow in terms of operating pressures and pump size to meet the requirements of product delivery. The impact on mechanical seals translates to higher flush rates being re-circulated through the Plan 11 system, thus decreasing the overall pump efficiency.

API Plan 99 (12)

Plan 99 (12) is a varation of Plan 11, implementing the use 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 mechancial seal assembly within Arrangement 1 and Arrangement 2 installations. Typically defined as a discharge bypass, Plan 99 (12) 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. Either a simplex or duplex filtration system is installed within the associated plumbing upstream of the seal chamber, with flow rate being controlled by a restriction orifice (RO). A minimum requirement for pipeline applications should be the implementation of a differential pressure transmitter across the filter housing. This serves as the primary indicator that an element is fouled signficantly and that seal flsuh flow has been compromised.

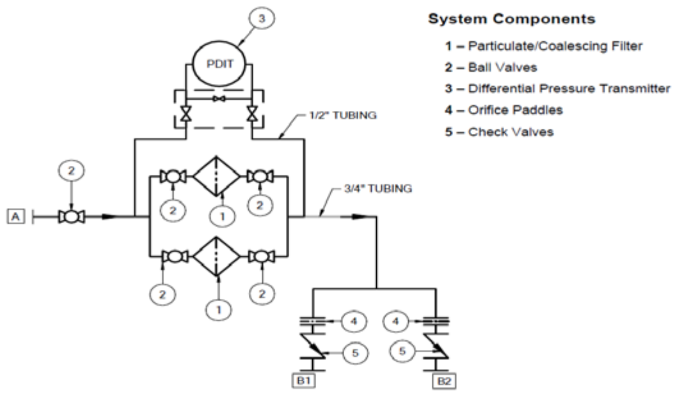


Figure 23: API Plan 99 (12) Diagram

The design of the filtration system should take into account 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 maximim 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
- Filtration arrangement (i.e., simplex vs. duplex)
- Implementation of transfer valve for non-interrupted service
- Implementaion of magentic separator
- Control logic regarding equiment shutdown or maintenance required due to plugged elements

The aformentioned are critical in terms of establishing a suitable mainteance schedule to address fouled elements. Pipelines are subjected to varying degrees of pipeline rouge, pigging frequency and overal flow demand; however, maximixing 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.

API Plan 31

Plan 31 is a variation of Plan 11, with the addition of an in-line cyclone separator. Utilization of Plan 31 attempts to mitigate the amount of solids and abrasives injected at the primary sealing interface of either an Arrangement 1 or Arrangement 2 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. Plan 31 is advantageous in that solids are removed from the primary flush source, while offering a solution that requires minimum maintenance in the form of cleansing or frequent element replacement.

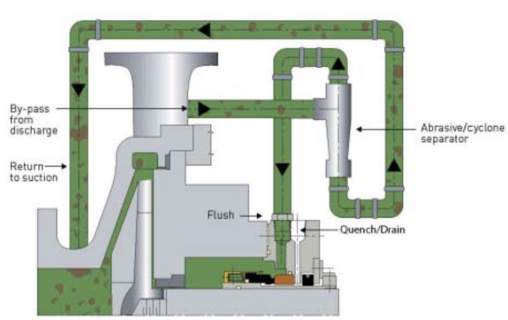


Figure 24: API Plan 31 Diagram

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 adequte 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 means of a dedicated filtration system (99/12). Finally, the operator should understand that the use of a separator does not assist with removing the contaminates from the main process stream. The contaminates are merely directed away from the primary sealing interface and remain in suspension, routed back to suction pressure regions on the pump.

API Plan 52

Plan 52 is intended for use with Arrangement 2 installations, using an external reservoir to provide buffer fluid for the outer seal of the dual unpressurized assembly. During normal operation, the reservoir maintains a pressure lower than the seal chamber pressure and is continuously vented to the appropriate vapor receovery system. Circulation of the buffer fluid is maintained by the use of an integral pumping device and provides adequate lubrication for the containment seal. The typical Plan 52 reservoir will be fitted with internal cooling coils, an orifice to create back pressure as primary leakage increases, local instrumentation and various transmitters required to trouble-shoot potential failure modes. The alarm/shutdown logic will have to consider the characteristics of the main process as it migrates past the primary sealing interface. For fluids that remain in liquid form, the appropriate alarm strategy will be built on high level transmission, while applications dealing with vaporizing hydrocarbons will require the combination of high pressure alarms working in conjunction with the installed orifice plate.

While API Plan 52 has been used successfuly in various oil & gas applications, the utilization of Plan 52 is not preferred in pipeline applications for various reasons. 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 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. In addition, the amount of working volume in the reservoir is typically limited when compared to predicted leakage rates of high pressure or viscosity applications. When evaluating Plan 52 in flashing hydrocarbon services, bulk temeperatures within the Plan 52 reservoir due to lack of adequate cooling can lead to expedited buffer fluid decomposition and in some cases impact the lubrication regime at the inner diameter of the inner seal faces. When considering the high maintenance requirement and lack of additional utility support, Plan 52 is not included as a preferred piping plan relative to the pipeline industry.

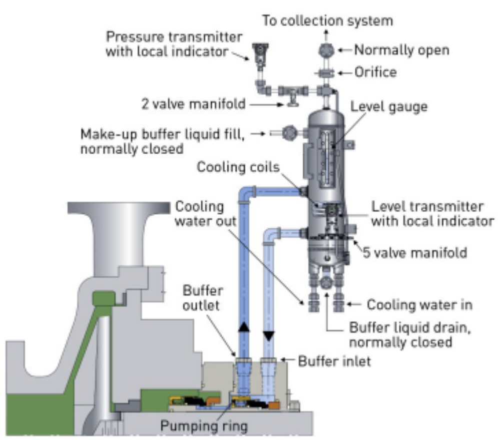


Figure 25: API Plan 52 Diagram

API Plan 53B

Plan 53B is primarily intended for use in Arrangement 3 applications, where the seal consists of two separate sealing interfaces that are operating in a pressurized environment. 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 asssemblies, 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 absoprtion 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 incoportate a large volume auto-fill skid.

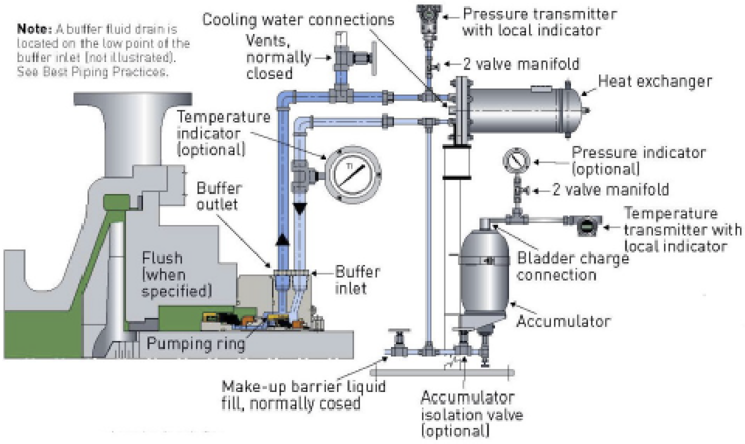


Figure 26: API Plan 53B Diagram and Auto-Fill Skid

When evaluating implementation of the Plan 53B skid, the pipeline operator should work with their equipment supplier to evaluate pumping ring performance vs. the system curve, which takes into account tubing size, tubing length, flow resistance from the addition of ancillary equipment, retention time based on reservoirs, etc. At higher heat load requirements, API Plan 54 might be another design consideration that ensures positive seal flush flow without the complexity of the Plan 53B maintenance and system curve requirements.

API Plan 54

Plan 54 is primarily intended for use in Arrangement III applications, where the seal consists of two separate sealing interfaces that are operating in a pressurized environment. The Plan 54 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 containtment. 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 DCS to provide equipment performance monitoring and assist with main process pump shutdown in the event of an upset condition. Increased seal leakage will mainfest itself through decreasing levels in the operating reservoir, which should be sized adequtely and implement a seqenced alram strategy to allow for troubleshooting in the field prior to equipment shutdown.

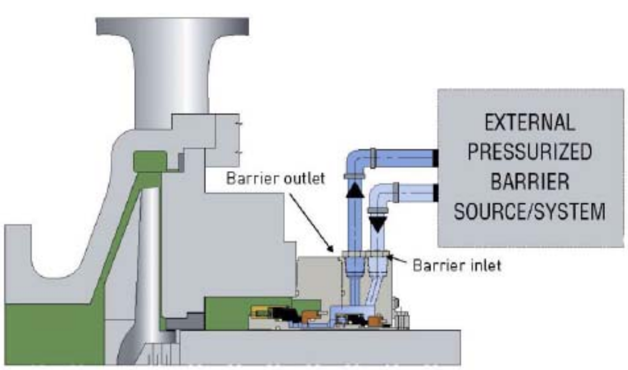


Figure 27: API Plan 54 Diagram and Example System

Plan 54 systems provide a continuous pressurized barrier fluid to the seal assembly and are viable solutions where the pipeline operates with varying process fluids. As the sealing interfaces are isolated from the main process, applications with dirty processes or unstable lubricity regimes can now be managed by providing a stable operating environment with specific seal design and material criteria. The operational scheme requires consideration of varying seal chamber pressures associated with different process applications and the pumps operating in series configuration. Determining the maximum setpoint will be critical to ensure positive pressure bias relative to both seal assemblies. An alternative, though more complex, is to provide a system with pressure tracking capability. Once again, in determining the overall flow requirements, the operator and supplier should consider maximum operating pressure to properly size and design the system.

Plan 54 systems offer full processs containment and can extend equipment reliability by providing a stable enivornment; however, there is a signficant amount of upfront cost and complexity in terms of maintenance asssociated with this sysem. When implementing Plan 54, the pipeline operator should work through their equipment supplier to ensure suitable instruction, operations, and mainteance (IOM) documentation is provided, along with detailed training.

API Plan 65A

Plan 65A is primarily intended for use in Arrangement 1 applications and provides leakage detection and management for fluids normally in a liquid state. The system uitlizes a close-clearance containment bushing and requires the leakage be directed from the gland plate drain connection to a liquid reservoir and through an orifice to an acceptable sump or drain recovery system. The liquid reservoir design should include a level transmitter, local level indicator and a high-point bypass line. During normal operation, leakage will flow through the orifice to the recovery system. As leakage increases, the orifice downstream of the reservoir restricts flow and allows the liquid level in the reservoir to increase. The rise in level activates the high level alarm and controlled shutdown of the equipment for maintenance can be executed.

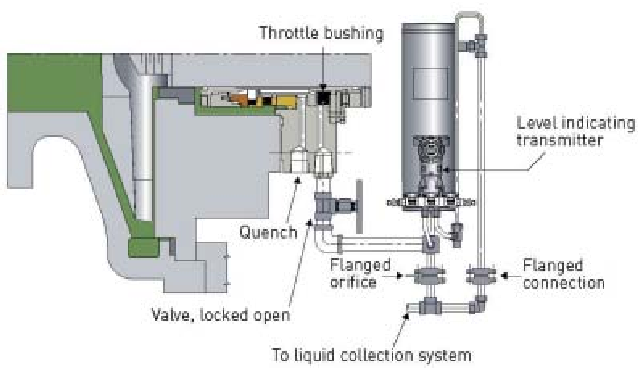


Figure 28: API Plan 65A Diagram

Design consideration should take into account varying leakge rates, changing liquid viscosities and presence of contaminates when determinig the gland drill-thru diameter, associated tubing or pipe size and perhaps most critical, the drainage orifice diameter. Typical orifice diameter is 0.250" (6.4 MM) and is susceptible to plugging in more viscous oils or low ambient temperatures. When compared to Plan 66 systems, the Plan 65A arrangement does entail higher upfront costs associated with the reservoir.

API Plan 65B

Plan 65B is also primarily intended for use in Arrangement 1 applications and provides leakage detection and management for fluids normally in a liquid state. The system uitlizes a close-clearance containment bushing and requires the leakage be directed from the gland plate drain connection through a liquid reservoir and to an acceptable sump or drain recovery system. The liquid reservoir design should include a level transmitter, local level indicator and a high-point bypass line. Plan 65B differs from Plan 65A in that the outlet from the liquid collection reservoir has a valve instead of an orifice. The rise in level activates the high level alarm and controlled shutdown of the equipment for maintenance can be executed, with the advantage being in this case leakage increases over time can be monitored more accurately.

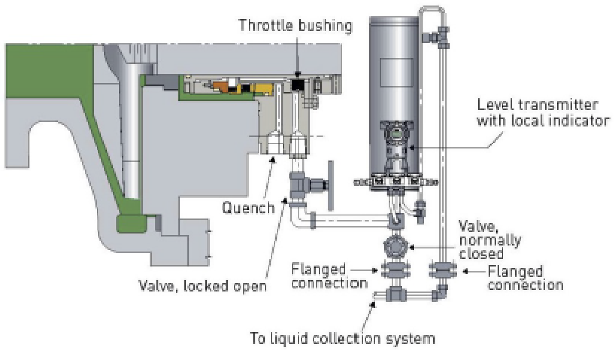


Figure 29: API Plan 65B Diagram

In the Plan 65B there are two valves. The valve between the gland drain connection and the reservoir should be open to allow process leakage into the reservoir. The valve below the reservoir should be closed allowing leakage to accumulate in the reservoir that will eventually set off an alarm. The first valve should only be closed to perform maintenance on the piping plan. The second valve may be opened periodically to allow the collected leakage to be drained according to the end users standard practice. Identical to Plan 65A, when compared to Plan 66 systems, the Plan 65B arrangement does entail higher upfront costs associated with the reservoir.

API Plan 66A

Plan 66A is intended for use in Arrangement 1 applications and provides two main feaures, including atmospheric detection of seal leakage by utilization of a pressure transmitter and minimizing seal leakage to atmosphere. The system is uitlized in conjuction with a single mechanical seal arrangement, supported by two throttle bushings operating in series. During normal operation, leakage will flow through the clearance between the primary bushing and the sleeve diameter. The region between the two bushings utilizes a large diameter drill-thru in the seal gland housing and installation of large-bore piping to direct leakage to an acceptable sump or drain location at the facility. The secondary bushing should be designed with a close-tolerance fit to the sleeve diameter to prevent migration of process leakage towards the backside of the gland plate. As the sealing system begins to experience an increase in process leakage, the pressure transmitter upstream of the primary bushing will detect an increase in pressure and trigger the respective alarm to allow for equipment shutdown and maintenance.

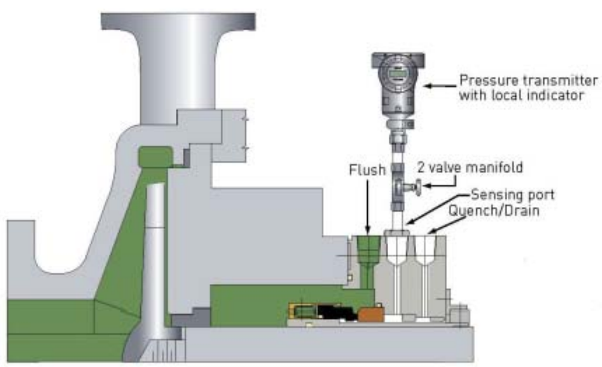


Figure 30: API Plan 66A Diagram

Plan 66A does have several control and design limitations to be considered based on each application. As an attempt to quantify seal leakge, 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 shutdwon parameters, the operator will have to take into account these variable factors. The added bushing and gland porting requires additional axial space that can be limited on older equipment. The distance to nearest obstruction and shaft undercut locations relative to the seal chamber face will require review to ensure proper fit of the longer mechanical seal assemblies related to Plan 66A design concepts.

API Plan 66B

Plan 66B is intended for use in Arrangement 1 applications, providing atmospheric detection of seal leakage by utilization of a pressure transmitter and containment of seal leakage to atmosphere via the use of a single floating bushing. Whereas Plan 66A utilizes two separate monitoring and drain cavities, in Plan 66B those two cavities are combined. The Plan 66B system does require implementation of an orifice plug to be installed within the gland plate drain connection. As the sealing system begins to experience an increase in process leakage, the pressure transmitter upstream of the orifice plug will detect an increase in pressure and trigger the respective alarm to allow for equipment shutdown and maintenance.

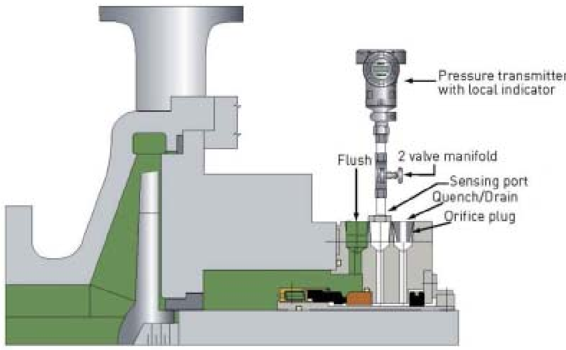


Figure 31: API Plan 66B Diagram

Due to the single bushing design concept, Plan 66B might be a more viable option when compared to Plan 66A in equipment installations with limited axial space. With regards to design considerations, sizing of the orifice diameter should take into account the viscosity of the process fluid, varying exposure to pipeline contaminates, and variances associated with changes in ambient temperatures. An orifice diameter between 0.062 - 0.125” (1.5 to 3 MM) could be prone to plugging, thus the implementation of a simplex flow meter should be considered to help troubleshoot potential nuisance alarms due to fouled orifice plugs. The aforementioned scenario allows for process accumulation in the intermediate cavity and possibly results in higher leakage rates through the containment bushing to the atmosphere. Incorprorating orifice designs with larger diameters will allow for higher leakage rate values to trigger the pressure alarm set-point and thus increase the potential for atmospheric exposure. When utilizing Plan 66B, the operator should be prepared to develop preventive mainteance tasks geared towards inspection of the orifice and drain system prior to equipment maintenance.

API Plan 75

Plan 75 is intended for use in Arrangement 2 applications utilzing 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 feebdback in terms of a normal level rise on a time based frequency. The operator should work with the seal supplier to establish accpetable leakage rate criteria and the Plan 75 system can then be monitored based on volume accumulation.

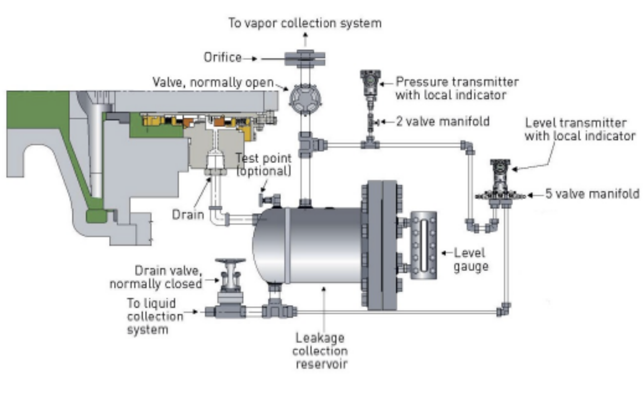


Figure 32: API Plan 75 Diagram

Plan 75 offers flexibility with fluids such as butane and iso-butane, that depending on ambient site conditions can sometimes yield condensable leakage. Additionally, when comparted to API plan 52 it provides the elimination of a supplemental buffer fluid with its maintenance and disposal concerns. Plan 75 does require a vent to a vapor recovery or flare, which when viewed separately would become API Plan 76. For this reason, selection of a Plan 75 is often advantageous as Plan 76 is provided as well.

API Plan 76

Plan 76 is intended for use in Arrangement 2 applications utilzing 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. In this arrangement, the seal gland will contain a vent port located 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. API 682 requires a minimum orifice diameter of 0.125" (3 MM), 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 3 MM) 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.

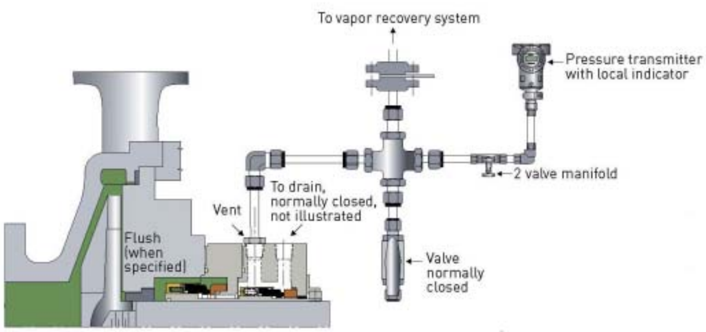


Figure 33: API Plan 76 Diagram

Plan 76 is a low cost alternative to wetted Arrangement 2 systems using Plan 52 and in addition has lower mainteance costs and requirements. As Plan 75 and 76 both do not utilize a buffer fluid, routine field testing should be executed to evaulate 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).

CONCLUSIONS

As energy demands increase, the need for pumping equipment and mechanical seals in the midstream pipeline segment to accommodate higher pressures, speeds, and extremes of variable fluid properties will continue to evolve to meet those requirements. Through careful consideration of several parameters discussed in this tutorial, a successful and reliable mechanical seal and support system can be implemented in these applications.

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LIST OF FIGURES

Figures courtesy of John Crane

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