Mechanical Seal and Support System Considerations for Negative Temperature Hydrocarbon Services: NGL Processing and Ethylene Production Focus

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ABSTRACT

There are many challenging mechanical seal applications throughout industry and perhaps none more so than those with negative operating temperatures, more succinctly, a temperature value lower than zero on respective Fahrenheit or Celsius scales. The volatile nature of many process fluids yields the requirement of negative temperature operation so that the fluids may be processed and handled by different types of rotating equipment to achieve a desired result. At the forefront of most equipment selections is the associated method of shaft sealing, and with negative temperature fluids, there are especially unique challenges with applying mechanical seals. The process of selecting mechanical seal types and associated support systems for negative temperature applications requires a thorough evaluation of all aspects associated with functionality and ultimately long term reliability of these installations.

INTRODUCTION

The discussion of low temperature or negative temperature sealing can be a challenging conversation as these terms can mean many things to different individuals. In evaluating various sealing considerations, it is useful to begin by establishing criteria for the evaluation. API 682 4^{th} Edition states that operating temperatures below -30 °C (-20 °F) require special considerations with regards to materials of construction of the seal design, among other variables. The scope of API further specifies temperature ranges, operating conditions, recommended seal types, and special features with respect to certain fluids, such flashing and non-flashing hydrocarbons, wherein the lowest temperature range considered is -40 to -5 °C (-40 to 23 °F). There are applications that routinely fall outside of these temperature ranges in various NGL processing and Ethylene production facilities, particularly in the Demethanizer, Deethanizer, and C2 Splitter areas, where the 'intent' of API 682 would still be desired, even if the product temperature ranges would be between -20 °C (-4 °F) down to -140 °C (-220 °F) in some cases. Mechanical seals operating outside of the scope of the temperature ranges specified in API 682 4^{th} Edition would be classified as 'engineered seals', which only designates that the seals do not fall within the scope of the standard and are therefore not required to have completed qualification testing; engineered seal is not a specific seal type in API 682. In fact, when an engineered seal designation is applied, one must be cautious in how much of the actual standard can realistically be applied to the design. In relevance to the processing industry in question, light hydrocarbon fractions typically associated in these temperature ranges would be utilized in a wide array of applications ranging from fuel to raw material for further chemical synthesis such as polyethylene production. Figures 1 and 2 are typical process flow diagrams that might be associated with processing facilities of this nature.

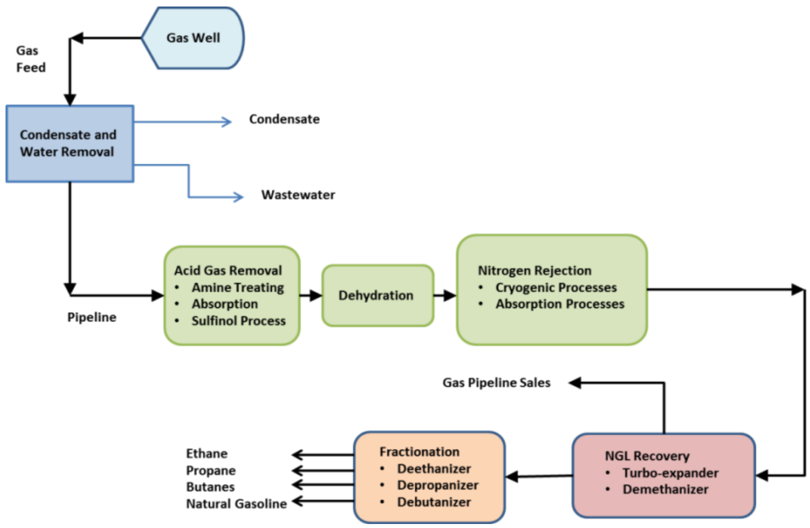

Figure 1: Simplified Gas Fractionation Process Flow

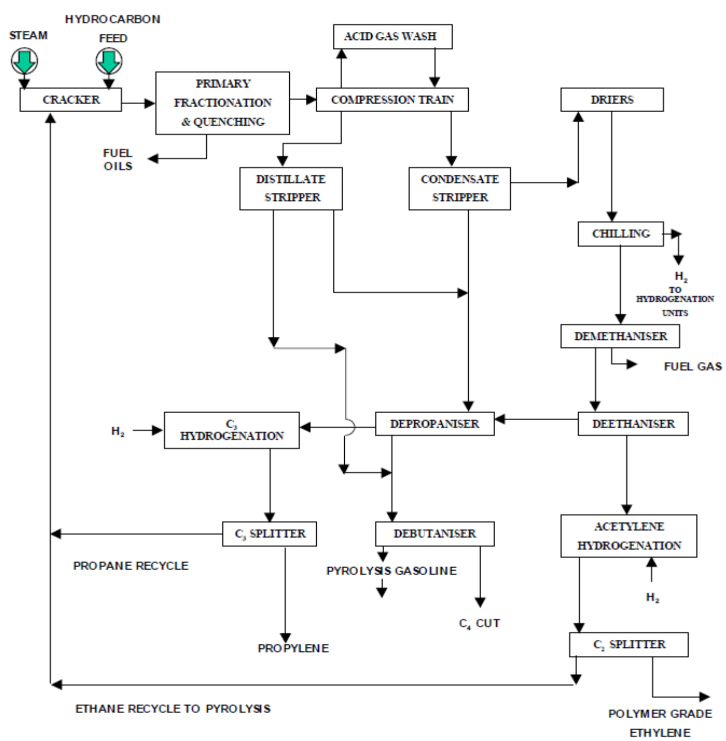

Figure 2: Typical Ethylene Production Process Flow Diagram

The challenge in processing these various constituent, low molecular weight hydrocarbons is their unique fluid properties, starting with the atmospheric boiling point. As an example, ethane, ethylene, and propane are all gases at ambient temperature and pressure since their atmospheric boiling point is less than normal ambient temperature. For this reason, these fluids and others similar to them would be processed at either very high pressures or very low temperatures. Figure 3 illustrates this point where the vapor pressure of ethylene at 0 °F is 390 PSIA, requiring that the ethylene must be pressurized greater than 390 PSIA in order to be a liquid at 0 °F. If however, the temperature is reduced to -50 F, the vapor pressure is reduced to 180 PSIA. This logic is indicative of many of these fluids in that they may be handled at lower pressures provided the temperature is reduced. In addition to the vapor pressure increase along with temperature, the specific gravity and viscosity would also decrease with a temperature increase. Additional details specific to the seal chamber conditions as it relates to these fluids will be addressed later in this tutorial.

	Ethylene	Ethane	Propane
Formula	C2H4	C2H6	C3H8
Molecular Weight	28.1	30.1	44.1
Boiling Point (°F)	-154	-127	-43
VP (PSIA)			
at 0°F	390	220	40
at 50°F	180	90	12
SG at 0°F	0.41	0.44	0.55
Viscosity (cP) at 0°F	0.05	0.07	0.15

Figure 3: Fluid Property Comparison

The services generally described to this point would be considered strictly API applications, and require the use of a current edition API 610 pump designs. API 610 defaults to mechanical seals that meet the standards and criteria of API 682. While there is some coverage for negative temperature applications as previously addressed, API 682 does not address many specific service directly although the majority of the applications described classify as flashing hydrocarbons. In these services, API 682 recommends a Type A seal, which is a pusher seal utilizing multiple springs and elastomer (O-ring) secondary sealing elements in a rotating or stationary seal head orientation.

The recommended use of a Type A seal is not without basis in field experience and various manufacturer testing. The requirements for 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; in these operating regions, the face materials are likely to experience higher wear rates due to increased temperature (from rubbing friction) and potential break down due to hydrostatic loading of the faces themselves, which is a function of the very high pressures typically associated with these applications. Further, balance ratio, which is 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. For Type A mechanical seals, the balance diameter is typically the diameter of the sliding contact surface of the dynamic O-ring. In a Type B and C seal, the balance diameter is the mean effective diameter of the bellows core. Due to the nature of the bellows geometry, the design is considered inherently balanced at low pressure. As pressure increases, the balance diameter decreases to a degree determined by the temperature, material characteristics, plate thickness, and geometry of the core, which leads to a net overall increase in balance ratio and face load.

Due to the nature of construction of the metal bellows core, which is comprised of individually welded plates, along with the likelihood of inadequate vapor suppression and potential for increased wear due to increased face loading, the conditions described will potentially lead to 'stick-slip' of the faces which transfers stresses to the undamped bellows welds closest to the face location. API 682 cautions against the use of metal bellows in flashing services due to these fatigue failure concerns as they relate to the metal bellows core. This concern would be most applicable to Arrangement 1 (single) and Arrangement 2 (dual unpressurized) seals, but not necessarily applicable to Arrangement 3 (dual pressurized) seals using a non-flashing barrier fluid. In these instances, Type C seals may be used, but the typical pressures of many fluids in these applications would be greater than the maximum pressure ratings of most metal bellows seal designs. For this reason, API 682 does suggest Type C seals are feasible in Arrangement 3 configurations, but they would fall under the designation of 'Engineered Seal'.

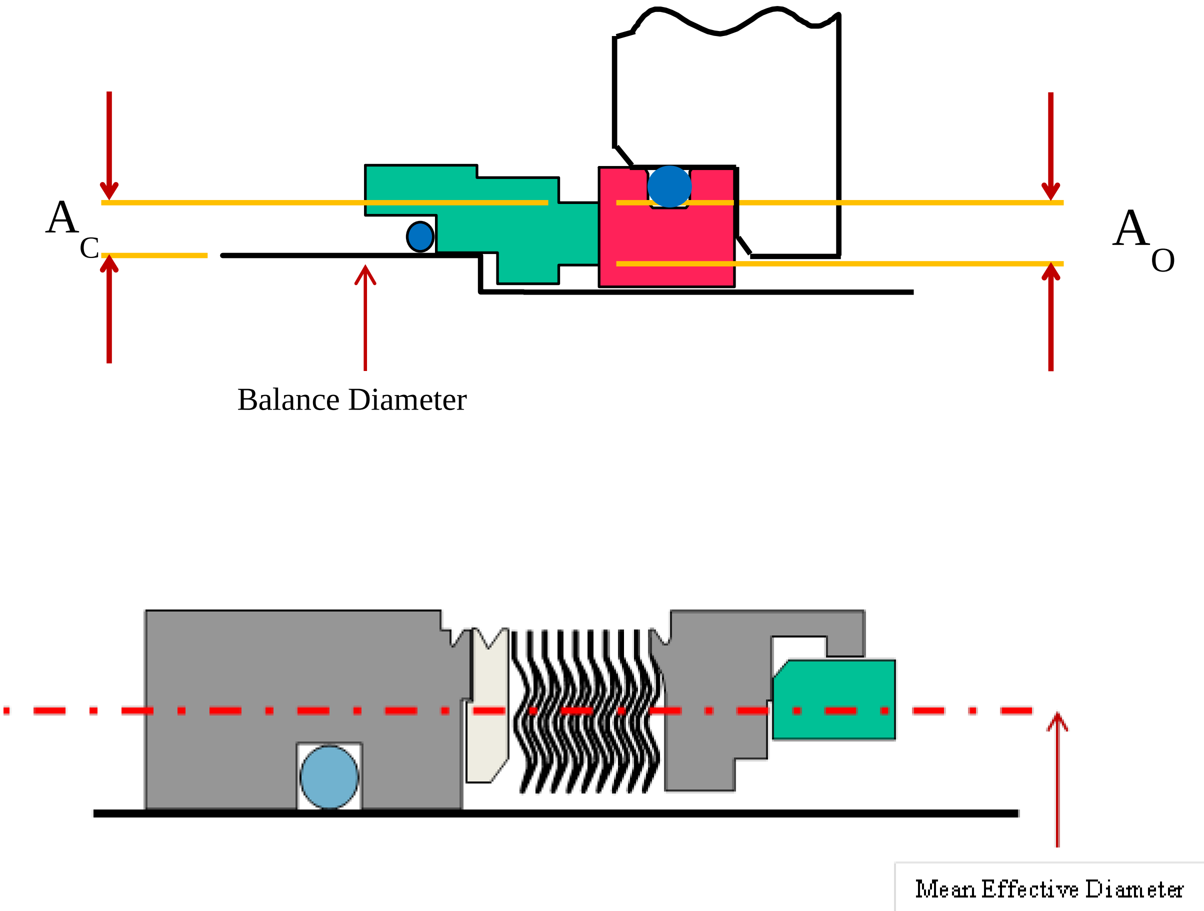

Figure 4: Type A (pusher) and Type B & C (metal bellows) balance diameter comparison

MECHANICAL SEAL MATERIAL CONSIDERATIONS

Essential to the design of the mechanical seal is the materials of construction acceptable for use in the application. When reviewing specialized applications such as those discussed in the introduction, considerations for how the process conditions can influence the seal materials are very critical. To evaluate the potential impacts to the mechanical seal design under these conditions, it is beneficial to look at several key components independently, specifically secondary sealing elements and metallurgy.

Secondary Sealing Elements

Secondary sealing elements are divided into sub-groups in API 682: elastomers, energized seals, flexible graphite rings, and flexible graphite filled spiral wound gaskets. Selecting the appropriate secondary seal material is important to the functionality and reliability of a mechanical seal in any application, but perhaps even more so in these negative temperature services. A suitable secondary sealing element must be chosen based on the application temperature and compatibility with the process fluid. Below are some general temperature limits associated with some common secondary seal materials:

Material	ISO / ASTM Designation	Minimum Temperature °C (°F)
Fluoroelastomer	FKM	-7 (20)
Perfluoroelastomer (high temp)	FFKM	0 (32)
Perfluoroelastomer (chemical resistant)	FFKM	-7 (20)
Nitrile	NBR	-40 (-40)
Ethylene Propylene Diene	EPDM	-50 (-58)
Tetrafluoroethylene Propylene	FEPM/TFE	-7 (20)
Polytetrafluoroethylene	PTFE	-270 (-454)
Flexible Graphite	-	-240 (-400)
Fluorosilicone	FVMQ	-60 (-76)

Figure 5: Secondary Seal Temperature Limits

The list above is intended as a general guide. There are, in fact, many manufacturers of several of the materials listed above and some manufacturers producing specialized grades that can function outside of the ranges above. Common examples are specialized NBR compounds suitable for use in temperatures down to -54 °C (-65 °F) in static applications and some Perfluoroelastomer grades are being manufactured with capability of sealing application temperatures down to -42 °C (-42F). Traditionally, FKM was considered limited in low temperature applications, but advancements in curing processes have allowed specialty materials with low temperature capability down to -30 °C (-22 °F) to be developed and utilized.

One key element to note with negative temperature applications in relation to the secondary sealing element is the potential impact associated with the commissioning process. Because of the low temperatures involved, it is important to remove all moisture from the pump and mechanical seals prior to commissioning and start-up. The process of removing moisture is referred to as dry out, and in the past involved flooding the pump, seals, and piping with methanol. Methanol has a great affinity for water and is very effective when used as a drying agent; however, methanol is on the Volatile Hazardous Air Pollutant (VHAP) list and alternative methods for dry out involving nitrogen are more commonly used today. This being said, it is important to verify the dry out mechanism for any new negative temperature application as the chemical compatibility with secondary sealing element materials may be compromised if methanol is used.

Special design considerations are required for elastomers when used at temperatures below 0 °C (32 °F). One variable that must be considered is the squeeze. Thermal expansion and contraction of O-rings can significantly affect squeeze. For applications in the ranges discussed already, the minimum and maximum squeeze of each O-ring should be determined at the minimum operating temperature. In some cases, it may be necessary to increase the initial O-ring squeeze in order to have an acceptable squeeze at the minimum operating temperature. In any case, the O-ring squeeze should not exceed 30% at the maximum operating temperature in order to prevent damage to the O-ring. Additionally, if a special FFKM grade was used below 0 °C (32 °F), determination of squeeze would be especially important. The linear coefficient of thermal expansion for FFKM material is approximately two times the coefficient of thermal expansion for FKM by comparison. Careful design of O-ring grooves for these types of materials is required and slightly higher squeeze values may be necessary to ensure seal force retention at lower operating temperatures. As noted, the initial higher squeeze value may cause compression set problems if the O-ring is cycled between very low and very high temperatures, which is not likely in a typical application, but should be considered as squeeze values above 30 % at maximum operating temperature will damage the O-ring. At lower temperatures, compression set of the elastomer material is typically compounded by a reduction in squeeze due to thermal contraction; therefore, elastomer squeeze should be verified at the minimum operating temperature of each application.

While the available list of elastomeric secondary sealing materials is fairly comprehensive for these specialized applications, there are services that will occasionally fall outside of these material limits. The scope of API 682 covers application temperatures down to -40 °C (-40 °F), so application of the standard in temperature ranges outside of this limit denote 'engineered seal' descriptions applied to the mechanical seal. Utilizing specialized secondary sealing elements in these temperature ranges may require additional considerations which reinforce the engineered seal descriptor with regard to groove design and surface finish, among other factors.

When considering these specialized elements, attention should be paid to these finer points in the design of the components. The most typical type of specialized secondary sealing element utilized outside of elastomer temperature limits is a PTFE encapsulated seal design, either elastomeric or metal spring core. These designs are acceptable alternatives for static applications, but dynamic seals must utilize an energized polymer seal to provide the method of sealing between the primary elements and hardware.

An elastomer core encapsulated O-ring is an O-ring seal having an elastomer core to provide memory and a PTFE jacket to protect the elastomer and provide sealing. While these designs have been successfully used in the past, they are susceptible to pre-mature failure of the elastomer core through cracks in the PTFE jacket. PTFE by its nature is not an elastomer and has no memory – therefore mishandling during installation can promote damage as the material is not very forgiving. Minute cracks in the jacket can allow the process fluid to come into contact with the elastomer core. Should one consider utilizing this type of secondary sealing element, evaluation of the core compatibility with the process fluid should completed – at a minimum, the elastomer core should be compatible, but in instances where the temperature of the process exceeds core material limits, this may not be practical.

Based on these limitations, a more acceptable alternative would be to utilize a metal spring core encapsulated O-ring which is an O-ring seal having a helical wound metal spring core with a PTFE jacket. The PTFE jacket continues to provide the sealing function; the difference in this design is that the metal spring core provides some memory to the overall design. Utilizing the capability of the metal spring to provide memory to the seal ensures a more positive seal to the groove and retains seal force retention.

Despite these advantages, the metal spring encapsulated O-ring can still only be used in static applications. To address dynamic applications, a polymer seal is typically used. The term polymer seal is the general name for a spring energized PTFE lip seal. Polymer seals are recommended for dynamic mechanical seal applications (balance diameter seals) because they have a considerably lower sliding friction than encapsulated O-rings. There are various types and designs of polymer seals; one more commonly used is a U-section radial lip seal having a PTFE outer lip seal with an inner metal spring to load the seal lips against the mating hardware. Polymer seals as described are generally available to fit gland or groove dimensions similar to those used for O-rings. Although the required gland dimensions are similar to O-rings, polymer seals typically require specific shaft and bore dimensions for proper fit and radial squeeze. Special shaft and bore surface finishes are also required for proper performance. One design feature that has been successfully utilized in the past is to hard coat the sliding surface beneath the seal to obtain a super polish surface finish for proper operation of the polymer seal. Figure 6 depicts a seal installation utilizing metal spring core PTFE encapsulated O-rings in static locations with a polymer seal in the dynamic position beneath the primary seal face.

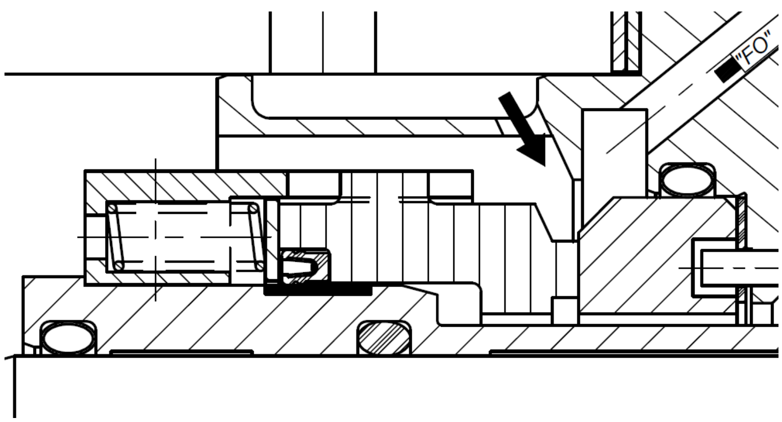

Figure 6: Secondary Seal Examples

In dynamic applications, the surface finish beneath the sealing element is critical as the roughness can have an impact on the wear rate of the PTFE material. Recall that the material properties are less forgiving than an elastomer, so wear from friction is especially critical for consideration. When evaluating surface finish, a common term used is Ra (Arithmetic Average Roughness Height); this factor alone may not be sufficient to properly specify the required finish as some polymer seal manufacturers require additional parameters for evaluation. Two other commonly used surface finish indicators are Rz and Rsk, where Rz is the average of the five greatest peak-to-valley separations and Rsk, or skewness, defines the symmetry of the finish about its mean line. Figure 7 is a table of commonly recommended surface finishes specified for one particular polymer seal design.

Media Sealed	Static Surface	Dynamic Surface
Cryogenic and Critical Sealing of Light Gases	Ra 6 - 12 µin	Ra 2-8 µin
	Rz < 50 µin	Rz < 30 µin
	Rsk -1.0 to -4.0	Rsk -1.0 to -4.0
Less Critical Gas Sealing	Ra 16-32 µin	Ra 6-12 µin
	Rz < 80 µin	Rz < 50 µin
	Rsk -1.0 to -4.0	Rsk -1.0 to -4.0
Fluid Sealing	Ra 20-63 µin	Ra 8-16 µin
	Rz < 80 µin	Rz < 50 µin
	Rsk -1.0 to -4.0	Rsk -1.0 to -4.0

Figure 7: Polymer Seal Surface Finish Examples

Metallurgy

Most low temperature applications use austenitic (300 series) stainless steels for pressure containing and/or structural seal components although other metallurgy may be considered under specific conditions. For metals typically used in sealing applications, Martensitic and Ferritic (400 series) stainless steels should be avoided. While they show excellent properties at room and high temperatures they lose most of their fracture resistance at low temperatures. API 682 standards are written for specific range of pumps, but can still be specified even though the equipment may fall outside of the categories noted. These standards require Charpy V-notch impact testing when the minimum design metal temperature is below -28 °C (-20ºF) unless they are exempt in accordance with ASME VIII, Division 1, UHA-51. It also specifies that the impact test results shall meet the requirements of ASME VIII, Division 1, UG-84. If one of the above standards is imposed it does require the purchaser to specify the minimum design temperature to be used to establish impact test requirements.

While Charpy impact testing is not a common requirement, if it is specified per API, ISO, or ASTM standards, the requirements and conflicts between standards can be confusing and time consuming to work out the details. If there is a requirement that the Charpy impact test be done at a specific temperature it will conflict with one of the ASTM material standards that normally specifies the testing to be done at 21–27°C (70-80ºF). In addition, ASTM standards require testing and certification of mechanical properties such as tensile, yield, elongation, and potentially hardness testing. Depending upon what specific tests are required by the purchaser, the use of UNS specifications will alleviate additional testing for mechanical properties that may not be necessary.

Charpy V-notch tests require that the test specimen is 2.165” long by 0.394” square with a 0.079” x 45º V-Notch in the center with the specimen taken from a representative section of the bar or tube. If a bar is used then the sample should be taken from the average diameter of the finished component being manufactured from the bar. The Charpy test requires 3 test specimens to be tested: if testing is done at a specific low temperature according to the ASTM standard the time between removing the specimen from the cooling medium to testing is to be 5 seconds; strict adherence to the testing criteria requires careful consideration the material supplier’s qualifications. If the machine is not set up correctly wide variations in results can occur and it has been noted in previous evaluations of the test that variations of 0.005” in the depth of the V-notch can alter the results by almost 10% of the impact resistance.

If the ASME Section VIII requirement are imposed then it is necessary to obtain the latest copies of UHA-51 and UG-84 to review all requirements. The data in Figures 7 and 8 are all longitudinal V-Notch results that are typically taken from 1 or 1-1/4” bar. Results obtained from large bar stock will be less and may require further discussion and evaluation.

Condition / Temperature	-40 C (-40 F)	-79 C (-110 F)	-196 C (-320 F)
15-5 PH (H1100)*	54	27	3.5
15-5PH (H1150-M)*	167	152	33
15-5PH (H1150-M)**	----	75	20
17-4PH (H1150)*	75	47	7.5
17-4PH (H1150)*	75	60	20

Figure 8: 17-4 and 15-5 PH (average longitudinal results in foot-pounds)
***Average value for 1" bar. **Average value for 6x6 section**

Condition / Temperature	-51 C (-60 F)	-101 C (-150 F)	-196 C (-320 F)
304 SS Annealed	82	84	80
316 SS Annealed	61	59	59
347 SS Annealed	71	71	65
410 SS Annealed	18	15	4
430 SS Annealed	8	5	2

Figure 9: Various Stainless Steels (average longitudinal results in foot-pounds)

For low temperature applications such as those discussed the materials to consider are 15-5PH (H-1150 & H-1150-M), 17-4 (H-1150 & H-1150-M), and austenitic stainless steel grades. Based upon the customer requirements the material can be specified by ASTM or other industry standards for chemical and physical requirements. Alternately, the material can be certified to UNS specifications with specific test requirements such as hardness.

SEAL FACE DESIGN CONSIDERATIONS

As discussed to this point, the task of applying mechanical seals to negative temperature hydrocarbons is not an easy one. There is a great level of detail required when reviewing just the materials of construction of the mechanical seal, to say nothing of the level of detail required when reviewing the seal faces and how they will perform in the application. The scope of this paper is not intended to provide a step by step detailed approach to designing seals to perform reliably in these challenging applications, but some guidelines and suggested criteria can be useful in anticipating mechanical seal performance based on the application. Prior to discussing these concepts in detail, some definitions are useful:

- **Flashing** – rapid change in fluid state from liquid to gas; most commonly associated with liquid vaporization between dynamic seal faces due to a reduction in pressure or increase in heat.

- **Flashing Index (Fi)** – Fi is the relative vapor portion of an isothermal seal face when there is no heat generation. This is a dimensionless number and while somewhat of an idealized situation; it can be a useful criterion in evaluating sealing applications. Services with very high flashing indexes require special attention to design details such as balance ratio, face design, and flushing mechanisms. As an example, a service with a Flashing Index of 1 would have an all vapor interface. More suitable Flashing Index values would be less than from 30 to 50% for a light hydrocarbon. Flashing Index is calculated based on the molecular weight of the fluid, process temperature, specific gravity, liquid viscosity, vapor viscosity, vapor pressure, and sealed pressure.

- **Product Temperature Margin (PTM)** – PTM is the difference between the saturation temperature of the fluid at the seal chamber pressure and the actual temperature in the seal chamber.

- **Vapor Pressure Margin (VPM)** – VPM is the difference between the seal chamber pressure and the vapor pressure of the fluid at the seal chamber temperature.

Per Annex F of API 682 4th Edition, previous editions of API 682 required a PTM of not less than 20 °C (36 °F) or a ratio of seal chamber pressure to maximum vapor pressure of 1.3 (130%), while the 1st Edition of API 682 required a vapor pressure margin of 3.5 bar (50 PSI). The current edition of API 682 reverts back to the 3.5 bar (50 PSI) vapor pressure margin from 1st Edition as it is often a simpler approach to performance evaluation for most hydrocarbons, but may be limited when considering high vapor pressure services. As an example, if the vapor pressure is 6.9 bara (100 PSIA) then the seal chamber pressure should be at least 10.3 bara (150 PSIA). A problem with this simple rule of thumb is it implies that seal chamber pressure must always be at least 3.5 bar (50 PSI) and this may not always be realistic or necessary. Because of this, the ability to evaluate vapor suppression based on the guideline that the ratio of seal chamber pressure to vapor pressure should be at least 1.3 to 1 can be more encompassing. For example, if the vapor pressure is 6.9 bara (100 PSIA) then the seal chamber pressure should be at least 8.9 bara (130 PSIA).

Looking further, assuming the application being reviewed was ethylene at -23 °C (-10 °F), the vapor pressure at this temperature is 21 bara (300 PSIA). Under these conditions, the Flashing Index would be 100 %, Product Temperature Margin zero, and Vapor Pressure Margin zero. Using the pressure margin requirement of API 682, 3.5 bar (50 PSI), the required seal chamber pressure would have to be 24 bara (350 PSIA) at the operating temperature. Comparisons of pressures and Flashing Index values are shown in Figure 10.

Pressure, Bara (PSIA)	Temperature, °C (°F)	Flashing Index
24 (350)	-23 (-10)	0.56
31 (450)	-23 (-10)	0.30
21 (300)	-40 (-40)	0.23

Figure 10: Flashing Index Comparison

In pursuing ideal conditions for the mechanical seal, the preferred method for improving vapor suppression in the seal chamber is typically to lower temperature. In practice, it would be very difficult to apply additional cooling in this example and therefore the default option would be make an effort in adjusting the pressure in the seal chamber. Seal chamber pressure adjustment can be achieved in different ways depending on the configuration of the pump design, and one more common method is to use a close clearance throat bushing in the chamber with an injection piping plan such as API Plan 11. While reasonably effective, the use of a close clearance bushing in conjunction with a high flow rate can have a diminishing rate of return in terms of bushing clearance increase over time from wear and erosion, depending on the bushing material. Because of this fact, it is practical to recommend considering seal face design effects that limit frictional heat influences should pressurization in the chamber diminish over time due to clearance increase in the bushing, pressure fluctuations in the process, and so on.

There have been many seal face design treatments used to help promote lubrication in volatile applications such as these negative temperature services described. For the most part, fluids described in these applications have very low viscosities and offer little in terms of hydrodynamic load support at the seal faces, so the potential for wear is likely. Many face treatments have been utilized successfully to mitigate these concerns as they are very prevalent in the applications discussed. Face treatments would be considered as engineered recesses, slots, grooves, or other patterns machined into one of the seal faces. The purpose of applying the face treatment is reduce seal face frictional heat by augmenting hydrodynamic lift, which in turn promotes improved face lubrication. Simple OD recesses, or hydropads, are one such treatment that has been incorporated into many seal designs with good reliability. The OD recesses promote interface lubrication by allowing fluid penetration in the face area. While this does minimize frictional heat by promoting enhanced lubrication the drawback is that the design may be susceptible to more increased detectable leakage levels. Leakage management is a very important aspect to the design of seals in these negative temperature fluids which will be discussed later; however, improvements to the face treatment can and has been implemented to maximize benefits in both areas.

As outlined by Wallace and Muller in proceedings from the Eleventh International Pump Users Symposium, defining consistency and stability of the seal interface gap is essential to minimizing wear and leakage. To preserve the advantages of full fluid film lubrication while at the same time avoiding elevated leakage rates requires an interface pumping mechanism which returns fluid from the seal gaps to an area of higher pressure. Different concepts were evaluated and tested, all based on shear-flow pumping of the sliding face surface that draws fluid through the gap towards the space to be sealed (Wallace and Muller). Through testing and design, a reverse flow face treatment was conceived to address concerns utilizing entry-flow (EF) structures connected to areas of higher pressure in unison with return-flow (RF) structures which collects liquid from the high pressure edge and guides it tangentially along the trailing edge of the recess which is very close to the high pressure side of the sealing gap, or sealed space. The fluid pressure in the RF recess can be much higher than the fluid in the sealed space and as a result much of the flow is returned to this location as it is the path of least resistance (Wallace and Muller).

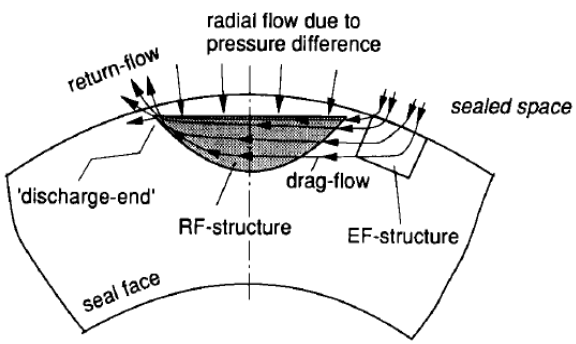

Figure 11: EF and RF Seal Face treatment as described by Wallace and Muller (1994 TPS)

The benefits of the described face treatment structure are numerous in that the seal reliability is improved by utilizing the hydrodynamic features to minimize frictional heat while at the same time maintaining low leakage levels. The nature of the design allows the feature to be utilized in either direction, which can also be advantageous in certain equipment applications. More advanced computer modelling techniques have allowed for flexibility in evaluating performance criteria in various applications, comparing number, distribution, and shape of the profile. Manufacturing utilizing laser etching has aided in profile depth consistency which has yielded repeatable results in terms of leakage rates (Wallace and Muller).

When considering applying seals in negative temperature hydrocarbons, the relationship between temperature and pressure is very critical to the successful operation of the seal. In many instances, the conditions surrounding the process or primary seal faces may only be a few degrees below the saturation temperature of the fluid, while the average seal face temperature may consistently be above the surrounding temperature by a significant margin, 4 to 5 times greater. When evaluating these applications, it is beneficial to consider what benefits may be gained by utilizing a specific face treatment such as the design described previously. As an example, an application with the following conditions was reviewed a few years ago as part of an upgrade project to improve the existing mechanical seal design, which was performing unreliably:

Fluid:	Methane & Butane
Temperature:	-149 °F
Suction Pressure:	487 PSIG
Discharge Pressure:	575 PSIG
Sealed Pressure:	525 PSIG
Vapor Pressure:	461 PSIA

Under the current conditions, the Flashing Index is reasonable at 0.26, but the comparison between seal chamber pressure and vapor pressure only yields a 17% margin. To consider increasing the margin by raising seal chamber pressure through the use of an injection flush may be possible, but impractical considering the minimal amount of differential pressure across the pump that would be available to support the seal flush flow rate. An alternative in this application was to consider a laser machined EF and RF structure as highlighted in Wallace and Muller's paper to reduce temperature rise at the sealing interface through minimized asperity contact. When reviewing FEA analyses of both a conventional face pair and optimized face pair using EF and RF structures, the advantages are noticeable.

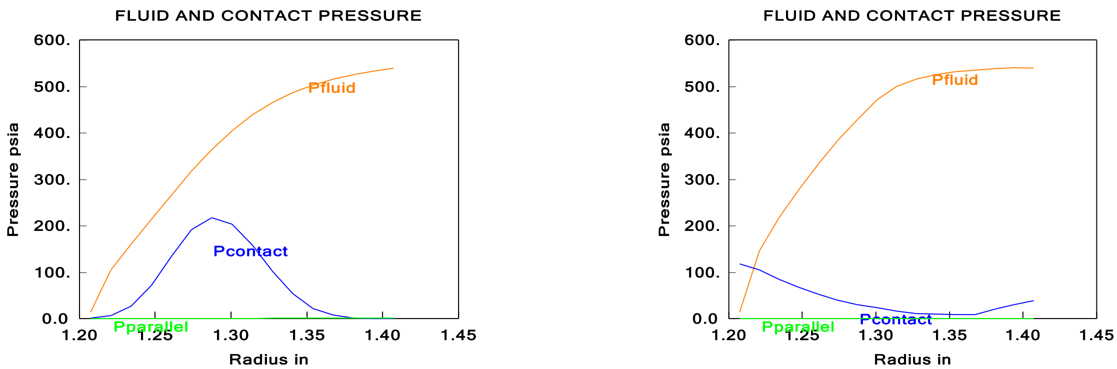

Figure 12: Conventional face vs optimized face pattern

The significant difference is in the relationship of the face contact pressure to the fluid pressure, representative of hydrodynamic load support in this application. The optimized face pair using the EF and RF structures greatly minimizes asperity contact pressure, reducing the likelihood of face material wear in this location which in turn keeps frictional heat generation to a minimum. The comparison of the average seal face temperatures was also significant in that the optimized face pair using the EF and RF structures resulted in an average face temperature of -144 °F, which is only a 5°F increase over the process temperature. By comparison, the conventional face pair yielded nearly a 60°F temperature rise over the process in terms of the average interface temperature. The application of this particular design approach proved to be advantageous based on the process conditions and limitations as noted with regards to the vapor suppression. The replacement mechanical seal has proven to be a good performer for past few years it has been in service.

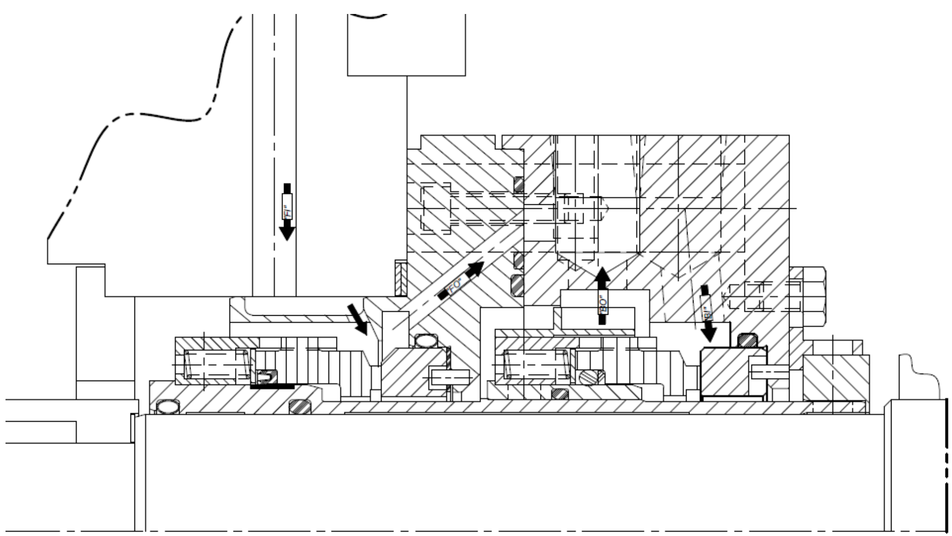

Figure 13: Mechanical Seal Design from Methane Application

Despite these advantages, a well-designed method of lubricating such a seal configuration is still required, meaning flush rates must be considered and calculated, although the rate of injection may be reduced when compared to more conventional designs. In applications where the seal chamber conditions are close to or below saturation temperature, it will be very difficult to utilize an Arrangement 2 seal even with the enhancements discussed. Some pump configurations will even add to the difficultly of utilizing this type of seal, which will be explored later. In instances where the seal chamber conditions are above or near the saturation point of the process fluid, either a pressurized seal (Arrangement 3) must be utilized or a non-contacting vaporizing liquid seal design may be considered as well. Dry running, non-contacting seals are based on design principles of compressor gas seals using modified grooving to allow for complete vaporization of the liquid prior to entering the seal face gap, or immediately after (Goodenberger). Non-contacting seals designed for sealing fluids near or above their saturation point rely on vapor formation to ensure a long, dependable seal life. A grooved pattern on the rotating face pumps fluid down from the outside diameter of the face to the root diameter of the groove. The pumping action generates an increase in pressure, which creates a small gap between the face, resulting in no face wear and essentially zero heat generation due to friction (Goodenberger). API 682 4th Edition allows the use of these seal types (2NC-CS); when considering these designs allowances in the support system design need to be accounted for as the leakage rate from these seals will be elevated when compared to a contacting design. Allotment for inner seal leakage, purge rates, and sizing of associated piping for realistic and accurate alarm points was discussed in detail by Kalfrin and Gonzalez (2015 TPS).

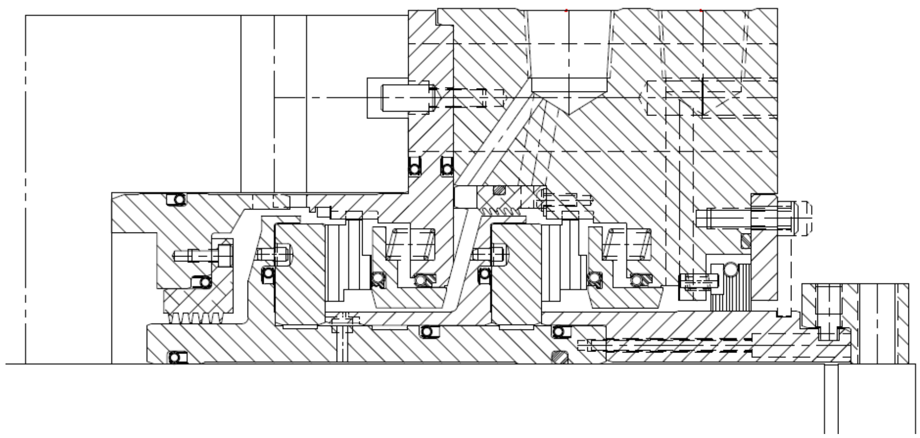

Figure 14: 2NC-CS in Ethylene service – note energized polymer secondary seals for temperature

SUPPORT SYSTEMS

In unison with concepts and aspects pertinent to the seal design, support of the mechanical seal in terms of seal face lubrication and leakage management are equally as important when sealing these negative temperature fluids. Some key concepts and some more common piping plans associated with these applications will be discussed, although this is certainly not an all-inclusive list. Based on the volatile nature of the majority of the services discussed and need for additional safety and leakage management, Arrangement 2 seals are typically used, which do require an inner seal lubrication mechanism in additional to a containment seal piping plan. Depending on the pump design and application conditions, an Arrangement 3 seal may be used if the seal environment is not suitable to support an Arrangement 2 configuration especially in terms of the inner seal face lubrication and vapor suppression.

Flush Rates and Types

Proper lubrication of the mechanical seal faces is essential to prolonged reliability of the mechanical seal. In the case of many of the applications 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 in that 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 those already discussed, this component can usually be ignored. Heat soak can be an interesting component in the case of negative temperature applications when considering heat loads, which will be discussed shortly.

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, like those common in some of the applications discussed, 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.

API Plan 11

API Plan 11 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. Plan 11 is the most common piping plan for single seals in use today. It can also be used as the piping plan for the inner seal of an Arrangement 2 or Arrangement 3 seals where a face-to-back seal arrangement is used.

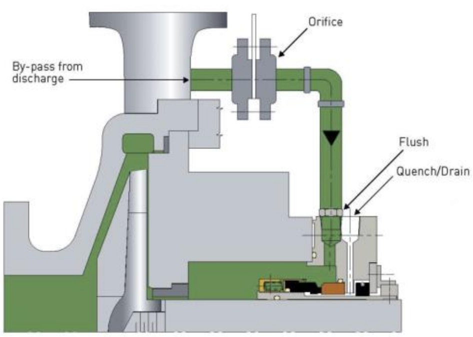

Figure 15: API Plan 11 layout

In the case of volatile services with low vapor pressure margins, the use of a Plan 11 injection with a distributed or multiple port baffle that directs flow at the faces is recommended. The use of these types of flush arrangements help mitigate erosion concerns due to high velocity impingement at the faces, ensure fluid distribution on larger diameter seals, and help force vapor bubbles that may form on adjacent components away from the seal face area. One of the challenges with Plan 11 especially in volatile services would be managing the differential pressure based on the pump configuration. API 682 recommends orifice diameters not be less than 3 MM (0.125 inch) to prevent clogging. High differential pressure applications may require multiple orifices in series, larger bore orifices, or choke tubes sized for the pressure drop. Similarly, on low developed head pumps, the orifice can potentially be omitted in some cases. The key element is that the flow rate should be sufficient to provide not only cooling to the seal but substantial enough to move vapor away from the faces.

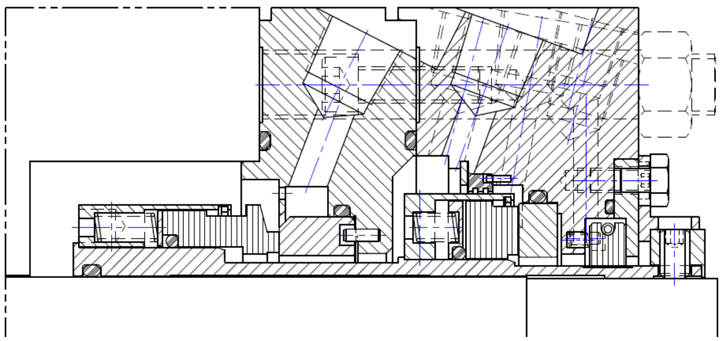

Figure 16: Arrangement 2 with distributed flush on inner seal

API Plan 13

Plan 13 is similar to Plan 11 with the exception that the flow is reversed. In a Plan 13 the flow exits the seal chamber, goes through the orifice, and is returned to the pump suction or suction piping of the pump. It is most commonly used on vertical pumps, where the seal is located at the top of the pump and is subject to discharge pressure below the seal chamber throat bushing. It should not be used on vertical pumps, utilizing a bleed bushing below the seal chamber, which would use a Plan 11. Depending upon the throat bushing clearance and the flow rate this piping plan can reduce the seal chamber pressure, which is of particular importance when considering applications where vapor pressure margin may be minimal.

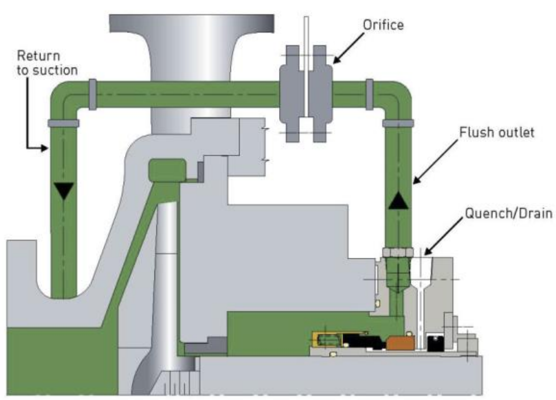

Figure 17: API Plan 13 layout

When properly piped Plan 13 is self-venting. Venting and potential dry running of the seal becomes an issue if the Flush outlet (F connection) is not at the upper most position in the gland plate above the seal faces. For this reason a Plan 13 should never be piped using a connection in the pump seal chamber/stuffing box. The Flush "F" connection on the gland plate should be a single port outlet. A distributed flush or multi-port arrangement should not be utilized with this plan as it can allow air to be trapped in the seal chamber. On volatile hydrocarbon applications an optimized Plan 11 design would have the flush impinging on the seal faces to assist in removing vapor bubbles that may occur due to seal generated heat. In a Plan 13 the flow path does not impinge on the seal faces to assist in removing vapor bubbles that may form and stay around the seal faces while the shaft is rotating. For this reason is it good practice to increase the flow rates to make up for the inefficiency of this piping plan. In narrow vapor pressure margin applications it is a good idea to make sure that the Plan 13 piping does not form a trap. To avoid this, the piping often cannot terminate on the pump, but will have to be at an elevation above the seal connection. This logic makes the flush self-venting and thermosyphon while on standby so there is no vapor on start-up.

API Plan 52

Plan 52 uses an external reservoir to provide buffer fluid for the outer seal of an unpressurized dual seal arrangement. During normal operation, circulation is maintained by an internal circulation device commonly referred to as a pumping ring. The reservoir in the system is usually continuously vented to a vapor recovery system and is maintained at a pressure less than the pressure in the seal chamber. Liquid buffer fluid systems utilizing a Plan 52 have been used for many years, and are advantageous in terms of both the ability to provide a reduction in overall leakage when compared to a single seal and redundancy in the event of an inner seal failure.

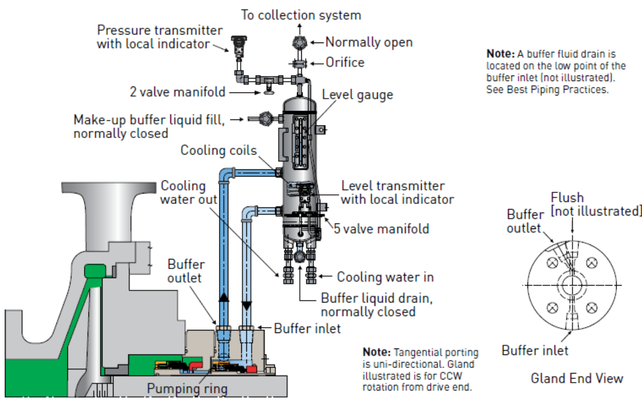

Figure 18: API Plan 52 layout

As mentioned, Arrangement 2 seal configurations are commonly used in negative temperature applications primarily due to concerns associated with leakage across the seal faces. Assuming an Arrangement 1 seal was used in one of the services described to this point, the combination of low temperature operation plus refrigeration effects due to the pressure drop across the seal faces would cause ice to form on the atmospheric side of the seal. To fully eliminate ice or prevent the formation of ice utilizing an Arrangement 2 seal with a liquid buffer can be very effective as the inclusion of the buffer fluid excludes moisture from the atmospheric side of the inner seal. In addition to the exclusion of moisture, the use of a liquid buffer fluid in these applications will also provide a built in additional source of heat to prevent ice formation by warming the inner seal. The heat source is from the frictional heat generation of the wetted outer seal operating within the confines of the containment seal cavity in the buffer fluid.

The heat generated by the outer seal in the buffer fluid will influence the inner seal area in the form of heat soak (from the buffer to the process seal area). The addition of this heat must be accounted for when considering the required flush rate for the inner seal in these services. In the case of an application where there is low vapor pressure margin, the additional heat from the buffer fluid can reduce margin further and potentially cause the inner seal to operate with partial to full vapor between the faces. To this point, consideration of the fluid properties and the associated seal chamber conditions with the added heat from the buffer fluid is crucial. Minimizing frictional heat generation at the inner seal faces is very critical in this instance and incorporating the various methods discussed in terms of the seal face design may be required. Another point of consideration with API Plan 52 when used with low temperature, high pressure volatile fluids is in regards to the reservoir design. Based on the refrigeration effects from normal leakage and also considering the scenario of a significant inner seal leak where the buffer fluid may be forced from the reservoir and to the flare system, the outer seal would be operating in a vapor pocket and degrading rapidly. The sudden in rush of the process fluid to the outer seal cavity will significantly lower the temperature of the reservoir. Considerations regarding the reservoir materials of construction need to be made to ensure compliance with section 6.1.6.11.2 in API 682 4th Edition, which address minimum design metal temperature.

Part of the selection process in opting for a Plan 52 support system is selection of the buffer fluid. The term buffer fluid would be associated with an unpressurized system, and Plan 52 systems with wetted seals utilize a liquid buffer. When considering buffer fluids for negative temperature applications such as those described, some key considerations would be:

- Safe to use, handle, and store
- Fluid is not a VOC (Volatile Organic Compound) or VHAP (Volatile Hazardous Air Pollutant)
- Non-flammable
- Good lubricity
- Good heat transfer properties
- Compatible with the seal materials and the process fluid
- Good flow qualities at very low temperatures
- Remains a fluid at very low temperatures
- Non-foaming when vaporized process leakage is added (from the inner seal)
- Low solubility of vaporized process
- Inexpensive

In more traditional applications, general purpose lube oil would be sufficient for a buffer fluid as it hits many of the criteria already listed. However, in negative temperature applications, these oils may not flow well when exposed to reduced temperatures and subsequent viscosity increases may cause blistering to occur on carbon seal faces exposed to the buffer fluid. In the past, Methanol has been used as buffer fluid for negative temperature applications, but as noted previously it is a VHAP and is no longer recommended. In addition to toxicity concerns, methanol has a low viscosity and low boiling point and would not be a good seal face lubricant.

Propanol, or propyl alcohol, is one of the most commonly used buffer fluid in negative temperature applications and has shown to provide good reliability in many applications of this type. Propanol has a pour point of -126 °C (-195 °F). Another common option is a 50 / 50 mixture of propylene glycol and water, which has replaced ethylene glycol and water as the ethylene glycol is classified as a VHAP as well. Propylene glycol and water has a pour point of -33 °C (-28 °F). In addition, there are many more synthetic lube oils and specialized heat transfer fluids with varying degrees of low temperature resistance. To list each fluid in detail would be beyond the scope of this paper. It is recommended that the application conditions and acceptable fluid options be reviewed during the seal selection and application process.

API Plan 72 (Gas Buffer Systems)

An alternative to a liquid buffer system would be to utilize a gas buffer instead, most typically referred to as an API Plan 72; Plan 72 can be implemented to support both contacting and non-contacting containment seals. Plan 72 uses an external low pressure buffer or purge gas which is regulated by a control panel and then injected into the outer seal cavity. Plan 72 can be a viable alternative to a wetted system as the injection of nitrogen into the containment seal cavity does warm the area sufficiently to prevent ice formation. While the injection of nitrogen into the containment seal cavity provides warming, it is not at the expense of added heat to the inner seal similar to the concerns with the Plan 52 system due to the low heat generation associated with dry running containment seals.

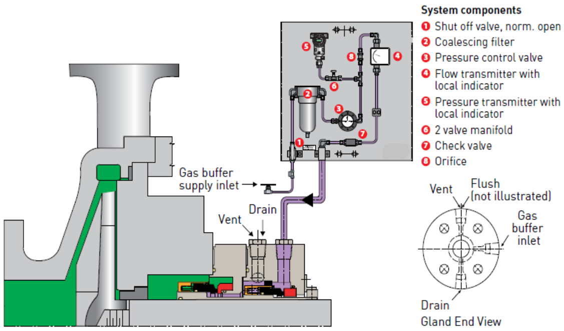

Figure 19: API Plan 72 layout

As the majority of the applications discussed are considered non-condensable leakage, Plan 72 will typically be used in conjunction with API Plan 76. Plan 76 is intended for use when the process sealed by the inner seal will not condense at lower temperatures or pressures. In this arrangement the vent is located at the top of the outer seal gland and is routed to a flare or vapor recovery system through an orifice, with upstream pressure monitoring and alarm. 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. The estimated leakage rate of the inner seal depending on the design (contacting or non-contacting), can directly influence the orifice diameter on the Plan 76 system.

Dry running seals with a Plan 72 and 76 can effectively be used in negative temperature applications down to temperatures of -40 °C (-40 °F); below these temperatures, and a Plan 52 would likely need to be considered. When evaluating a dry running containment seal and Plan 72, while the nitrogen purge into the containment seal cavity provides warming and prevents moisture ingress, there is potentially a concern for ice formation due to moisture in the atmosphere that would come into contact with the atmospheric side of the containment seal. To alleviate this concern, it is a good practice to utilize a nitrogen quench downstream of the containment seal to sweep the atmospheric side and displace moisture in this location. This is an application of API Plan 62 in addition to the Plan 72 and Plan 76 already described, and one of the requirements in terms of the mechanical seal design is sufficient axial space to accommodate the required gland plate porting in addition a throttle bushing to minimize quench flow to the atmosphere. The added benefit of this configuration would be an additional containment device in the form of the bushing that is sealing in a different plane than the containment seal faces.

As noted, the quench medium injected in the case of negative temperature applications must be nitrogen and not steam for obvious reasons. This is noteworthy only for clarification as many have the misconception that the term quench or Plan 62 denotes the use of steam; it only designates the injection location of the medium and not necessarily the medium being used. The use of multiple piping plans in one seal configuration can get confusing, so seal installation requires attention to ensure each gland port is connected to the correct location and that the gland ports are labelled correctly. Figure 21 is an example of a low temperature ethylene seal using API Plan 11 / 72 / 76 / 62 for support – note the use of the distributed flush baffle for Plan 11 flush injection at the faces along with a restrictive clearance throat bushing for increased vapor suppression. This design has yielded good reliability in this service.

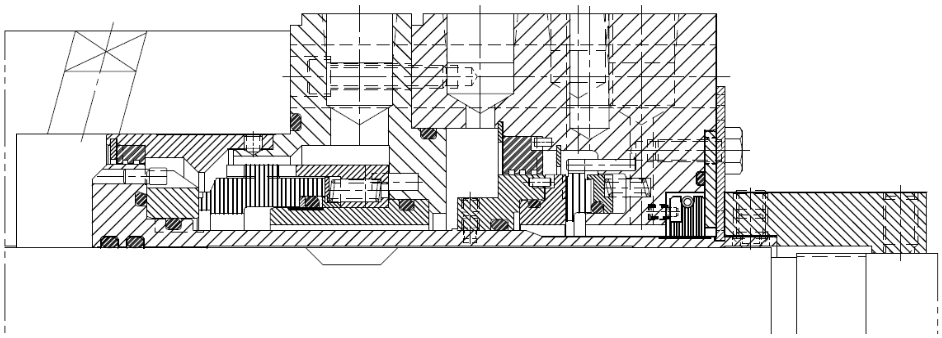

Figure 20: Ethylene Seal cutaway with API Plan 11 / 72 / 76 / 62

Figure 21: Nitrogen quench – seal area comparatively warmer than the pump

Pressurized Systems

Any discussion around negative temperature sealing applications in the ranges described should include mention of pressurized systems that would support the mechanical seal. Based on the severity of the service, the majority of pumps specified for duty in these applications would be API 610 designs, with the configuration varying based on the requirements of the service in terms of flow rate, developed head, or NPSHA (Net Positive Suction Head Available). Some more common pump configurations used in negative temperature hydrocarbon applications would be:

- API Style OH2 – horizontal, single-stage, overhung
- API Style BB2 - horizontal, between bearings, radially split
- API Style BB3 – horizontal, multi-stage, axially split
- API Style BB5 – horizontal, multi-stage, double casing
- API Style VS6 – vertical, double casing, diffuser style

As noted, when applying mechanical seals to horizontal pumps in the temperature ranges noted, there are inherent challenges related to the sealing of the process fluid based on the chamber conditions driven by the fact that the mechanical seal is essentially exposed to the full process temperature in a horizontal configuration. In a vertical pump configuration, either a VS6 style or variation, there have been unique features incorporated to many OEM designs that provide a more suitable environment for the mechanical seal. These specific features incorporate what is commonly referred to as a 'coffer dam' within the pump which is used to bleed off a small amount of the pumped product flow rate. As the product passes through a throttling device, or throat bushing, the pressure is dropped from discharge pressure to just above suction pressure. In essence, much like the use of a coffer dam within or across a body of water within which a dry environment is created, the same effect takes place within the pump design. In Figure 22, the throttled mixture of liquid and gas would be in Chamber 1 and as the mixture enters Chamber 2, the temperature has increased sufficiently in the mixture for it to be predominantly a vapor. The creation of this vapor space insulates the mechanical seal from the cold temperatures of the process, and makes the use of conventional materials feasible which will save on cost and manufacturing of the mechanical seal.

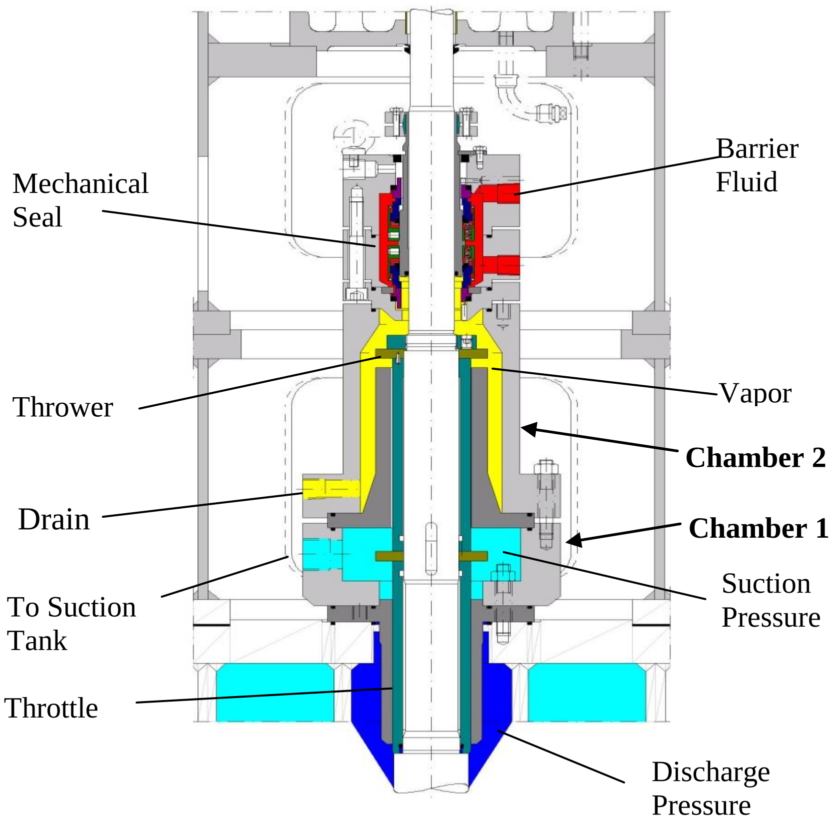

Figure 22: Coffer Dam in Vertical Pump

While the creation of this vapor space simplifies the mechanical seal design in terms of the materials of construction, that challenge is the vapor now occupying the chamber below the seal would not provide an acceptable lubricant for the mechanical seal faces. For this reason, an Arrangement 3 seal with a pressurized support system is the most common sealing configuration in these examples. An Arrangement 3 seal would use a pressurized barrier fluid supplied to the cavity between the two mechanical seals at a higher pressure than the process, or the pressure in the vapor space in this instance. As the fluid is maintained at higher pressure, it provides the lubrication for both sets of seal faces. Maintaining the barrier pressure above the pressure within the pump is essential to prevent cross-contamination and ensure that the seal receives sufficient lubrication. There are several API Piping Plans designated for use with Arrangement 3 seal designs, not all of which will be covered in this tutorial. What is most commonly used in these types of applications is either a traditional or very close approximation of an API Plan 53C.

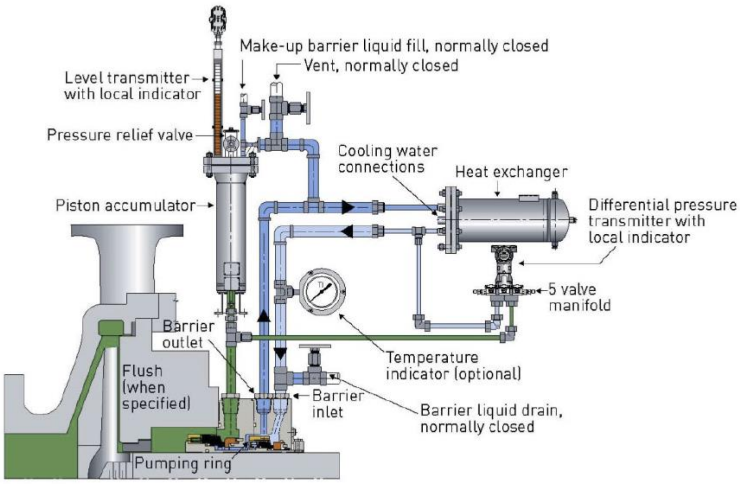

Figure 23: API Plan 53C layout

API Plan 53C features a reference line from the seal chamber to a piston accumulator teed into the closed loop system that provides a constant pressure to the loop above the seal chamber pressure. The basic setup is comprised of two parts: the closed loop circulating system and the piston accumulator. In this system, the barrier fluid pressure automatically “tracks” the reference pressure, which would be the pressure in the vapor space below the mechanical seal. This is especially useful for pumps having variable suction pressure conditions, which is not uncommon in some of these services. The barrier pressure for Plan 53C is a function of the piston accumulator design. Typically barrier pressure is about 10% above the reference pressure and some piston accumulators include a spring to provide a slightly higher barrier pressure and mitigate concerns with delays in tracking that may result in a loss of positive pressure across the seal faces. Delays can occur due to spikes or drops in the system and the seal chamber area.

The circulation rate in a Plan 53C system is usually not controlled directly; it depends on the performance of the pumping ring within the particular closed loop system. The pumping ring, reservoir and piping are selected to produce the desired operating conditions. Total heat loads of the seal must be considered in determining the desired circulation rate. All components should be located as close as possible to the seal and the interconnecting piping should be large in diameter and minimum length with long radius bends to minimize pressure drop and promote good flow through the closed loop. The comparatively warmer vapor space beneath the mechanical seal allows the use of more conventional barrier fluids and light viscosity oil is a common option. The use of pressurized systems is not isolated to vertical pumps with these special design features; any instance where the vapor suppression inside the seal chamber is a concern may be a good application for consideration. As noted, however, horizontal pumps will have seal chambers more likely in direct contact with the low process temperatures, meaning that considerations for the design of not only the seal components but associated support system components in a pressurized system require consideration for temperature resistance. In vertical pump designs, either with a vapor space feature or without, an alternative arrangement that has been used successfully with good reliability is a non-contacting inner seal with hydrodynamic face features incorporated to operate in a non-contacting nature on a vapor film; these would fall back into the Arrangement 2 configuration. Figure 24 is an example of such a design used in a vertical pump in LNG service at 15 bar (218 PSIG) and -30 °C (-22 °F).

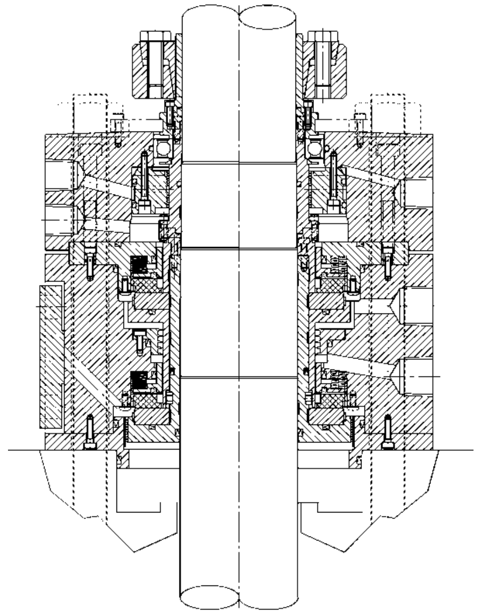

Figure 24: Non-contacting Arrangement 2 seal in vertical application

SUMMARY

A summary of some of the key topics discussed in this tutorial has been provided:

- Product Temperature Margin (PTM), Flashing Index (Fi), and Vapor Pressure Margin (VPM) are all factors that can be evaluated to determine suitability of a mechanical seal for a negative temperature hydrocarbon service and can be considered independent of actual mechanical seal details as they are all functions of temperature and pressure.

- Secondary sealing elements require careful consideration in terms of elastomer squeeze (when applicable) and surface finish requirements for spring energized polymer seals.

- In low differential head pump designs, consider a face treatment design similar to those described with entry flow (EF) and return flow (RF) structures to add gap stability and minimize frictional heat generation and flush flow requirements.

- When seal chamber conditions approach saturation conditions, a non-contacting vaporizing liquid seal design (Arrangement 2) or dual pressurized seal design (Arrangement 3) should be considered.

- Seal flush flow rates should be at least 2 GPM / inch of seal size if not higher to limit temperature rise in the surrounding fluid. Higher flow rates may be required to move bubbles away from the seal face area.

- Distributed flush arrangements are recommended for improved face cooling, flow distribution, and mitigation of vapor bubble accumulation around the seal face area.

- Consider heat soak from a wetted buffer fluid system (API Plan 52) when evaluating the heat load for an Arrangement 2 configuration in these services.

- Utilization of an API Plan 62 nitrogen quench is a good practice to prevent ice formation on the atmospheric side of both wet and dry containment seals.

- Vertical pumps – determine if the pump design incorporates a vapor space feature or if a conventional piping plan is to be used. If a vapor space exists below the seal, consider a dual pressurized (Arrangement 3) or non-contacting (Arrangement 2) design. If a Plan 13 flush is utilized, route the Plan 13 piping to an elevation above the seal chamber to promote more efficient cooling and venting of the seal chamber.

CONCLUSIONS

The application of mechanical seals in negative temperature hydrocarbon services is not without challenges. 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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ACKNOWLEDGEMENTS

The author would like to thank John Crane and Alan O’Brien in particular for support in the development of this tutorial. Additionally, a great thanks to Gordon Buck for his valuable insights over the years, which were beneficial in formulating this tutorial.