MECHANICAL SEAL PERFORMANCE AND RELATED CALCULATIONS

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INTRODUCTION

A short collection of mechanical seal performance calculations has always been included in the earlier and current editions of the seal standard API 682 and the co-branded version of ISO 21049. The new draft of the Fourth Edition of API 682 and the planned update of ISO 21049 include a significantly expanded version of these calculations plus associated explanations in its Annex F. The new Annex topics pay particular attention to subsections covering piping plans for dual seal configurations. This tutorial is intended to present some new material on leakage management and dual seals, review some existing Annex F topics, and include sample calculations. Some of the topics covered include:

- Leakage management
- Seal face generated heat
- Heat soak
- Seal flush flow rate
- Plan 52 and 53A Piping system curves
- Flush flow rates for Arrangement 3CW seals at different temperatures
- Piping Plans 53A and 53B operation and alarm strategy

SEAL PERFORMANCE: PART 1—SEAL LEAKAGE AND LEAKAGE MANAGEMENT

Seal Leakage

There is always a mass flow rate across the face of a mechanical seal, so all seals "leak" to some extent. Some seals, particularly noncontacting seals, are designed to have a certain flow between the faces. Nevertheless, for the vast majority of pumps there are normally no visible seal leakage. Leakage can occur regardless of seal category, type or arrangement; however, with Arrangement 2 and 3 dual seals, the leaked fluid may be buffer or barrier fluid instead of process fluid. Buffer and barrier fluids are often lubricating oils, which are not volatile, and wetting of the gland plate may occur resulting in occasional visible droplets. However, visible leakage in the order of drops per minute is normally an indication of a seal problem. Sometimes visible leakage is apparent only over time, as the nonvolatile components of the process stream or buffer/barrier fluids accumulate.

Contacting seals may use features such as variable or low seal balance ratio, or face enhancing features such as scallops, matte lapping or preferential lapping to reduce wear and extend the design envelope; however, leakage can be slightly higher than similar seals using plain faces under less difficult conditions. Seals designed for high pressures but actually used at low pressures may have unacceptable leakage. A single contacting wet seal (1CW) sealing water at a vendor pump test ordinarily leaks a fluid that is volatile and is not visible. The aforementioned design features, necessary for specific process reliability, can in a water-sealing environment alter leakage levels such that a slight visible leakage can occur at the vendor pump test.

Factors other than design features can result in increased leakage as well; however, these may be the result of aberrant system conditions. In particular, after a contacting seal has worn in to match a certain set of operating conditions, changing those conditions can result in increased leakage until the faces have worn to match the new conditions. Such changes include fluid type, viscosity or density in either the process or buffer/barrier fluid. Operating conditions such as temperature or pressure outside its design envelope can damage the seal and result in greater leakage rates. Other system factors that affect seal leakage rates, besides condition of the seal parts, include pump operation at off-design conditions, pipe strain, bearing problems, fitting leaks at the seal gland (often mistaken as seal leakage), impeller or sleeve gasket damage, etc.

Leakage Management

End face mechanical seals and devices used on the atmospheric side of these seals are a subset of the larger topic of leakage management. Depending on local laws and fluid properties different levels of leakage of the process fluid to the atmosphere or drain may apply. Leakage management might include the selection of a sealless pump, or a pump with additional containment using a bushing, packing or another end face seal of either contacting or non contacting design.

For example, when containment of the process fluid is required (zero leakage to atmosphere is required) a sealless pump or pressurized dual seal may be the right choice. At the other extreme, the use of a bleed bushing in a vertical cooling water pump instead of an end face seal may be appropriate since water leaking past the bleed bushing could be directed back to the sump.

Leakage management auxiliary systems can also be attached in series with mechanical seals. With these systems, leakage can be diverted to a location determined by the plant operator. Some examples of auxiliary systems include a separate buffer liquid lubricated "outer seal" and the associated support auxiliary system or a containment chamber and a sealing device for the containment chamber with its auxiliary support system. While there are many types of containment devices, three types are most common: 1) simple fixed bushings, 2) floating bushings and 3) special purpose mechanical seals called "containment seals." Selection of the appropriate containment sealing device and system depends on the requirements for leakage control as well as expectations during normal operation and upsets.

For many decades process leakage management has been achieved using an outer mechanical seal, lubricated by flow from a separate liquid buffer or barrier auxiliary system. The process leakage from the inner seal mixes with the buffer liquid and is separated and safely removed within the buffer liquid circuit or the barrier liquid (or gas) lubricates the seal faces and prevents process liquid leakage.

A containment sealing device does not necessarily have the performance or rating of a mechanical seal. There are many types of containment devices but fixed bushings typically have the highest release rates. Floating bushings leak significantly less than fixed bushings. Containment mechanical seals have the lowest leakage rate. Containment devices may also be used to manage quench fluids such as steam or water.

Mechanical seals used as dry running containment seals may be similar in appearance to conventional face type seals, but they include special features and materials. Although there are many variations, containment mechanical seals are designed to operate without the presence of a lubricating liquid. This ability to operate dry is possible because face material pairs have been specially developed and heat generation is very low. Containment seals may be further classified as having either contacting or noncontacting seal faces. Whereas contacting seals usually have a plain, flat face, a noncontacting seal face includes features to create aerodynamic lift that separates the faces. Noncontacting containment seals leak more than the contacting type; however, contacting containment seals have a finite wear life. Whether contacting or noncontacting, containment seals can have low leakage and long life.

Auxiliary systems used to contain process leakage from emission to the atmosphere are usually supplied with equipment that can enable the plant operator to monitor the process seal leakage rate and alarm when levels are considered excessive.

Arrangement 1 seals are usually fitted with either a fixed or floating bushing as the containment device. Optional leakage management systems for Arrangement 1 seals are Plans 62 and 65.

Arrangement 2 uses two mechanical seals; the outer seal can be either a conventional wet mechanical seal or a dry-running containment seal. Optional leakage management systems for Arrangement 2 are Plans 52, 71, 72, 75 and 76.

Predicted Leakage Rates

All mechanical seals require face lubrication to achieve reliability; this results in a minimal level of leakage. On a water pump test of a contacting wet seal (1CW), the leakage typically evaporates and is not visible. Face design features, however, can increase leakage levels and visible droplets may occur. Pressurized dual contacting wet seals (3CW), when used with a nonevaporative, lubricating oil barrier fluid, can also produce visible leakage in the form of droplets, but typically at a rate less than 5.6 grams/hour (2 drops per minute).

In the choice of seal type and arrangement, the purchaser may benefit by consulting the applicable seal vendor's qualification test results. The leakage value obtained will give a guide as to what may be expected after an adjustment is considered for differences in sealing pressure and fluid viscosity. The seal vendor should be consulted about predicted leakage rates.

Noncontacting inner seal designs utilize a liftoff face pattern, such as grooves or waves, which can provide reliable operation in liquid or gas service. Often it is difficult to provide an adequate vapor pressure margin when sealing clean high vapor pressure or mixed vapor pressure fluids with contacting wet face designs. A noncontacting inner seal can give the option of sealing a liquid/gas mixture by allowing the product to flash into a gas across the seal faces, effectively using the noncontacting design inner seal as a gas lubricated seal. The leakage rate from a noncontacting design is normally higher than a contacting wet design.

Noncontacting containment seals utilize a face pattern (grooves, waves, etc.) to provide an aerodynamic lift of the seal faces. Contacting containment seals use the face material properties and often specific molecules in the gas such as humidity to manage the wear rate and achieve the seal life expectancy of most users. Noncontacting face designs have the following benefits:

- Lower wear rate in operation
- More tolerant to higher pressures and pressure spikes created by the downstream leakage management system such as a flare or relief system
- Do not require maintenance check on their wear condition and function
- More tolerant to a Piping Plan 72, which utilizes low humidity gas

Contacting containment seals have different benefits, which are:

- The leakage rate to atmosphere, in normal and alarm conditions, is much lower (Figure 1 and 2). This is particularly significant when sealing a process with a high liquid content at atmospheric conditions in the inner seal leakage (Figure 2).
- The flat face design is more reliable when there is a significant liquid content in the inner seal leakage.

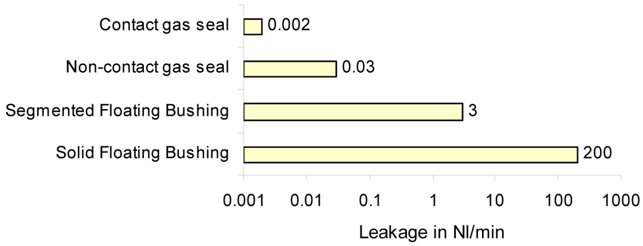

Figure 1. Estimated Gas Leakage for 50 mm Shaft at a Gauge Pressure of 0.07 MPa (0.7 bar) (10 psi) in NL/min.

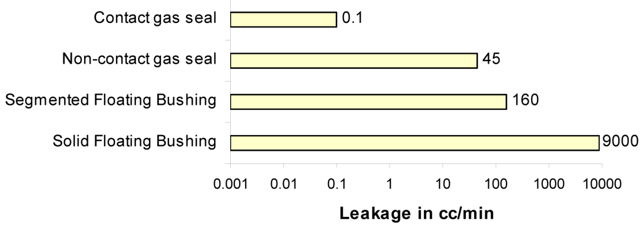

Figure 2. Estimated Liquid (Water) Leakage for 50 mm Shaft at Gauge Pressure of 0.275 MPa (2.75 bar) (40 psi) in cc/min.

SEAL PERFORMANCE: PART 2—LUBRICATION BETWEEN THE SEAL FACES ON VAPORIZING SERVICES

Lubrication Between Seal Faces

It is assumed that reliable seal performance requires liquid between the faces for lubrication. Since most seals have no visible leakage, we accept that the liquid between the faces vaporizes at some point as it travels across the face to the atmospheric side of the seal. The amount of gas between the seal faces of an idealized seal depends on the fluid properties, sealing pressure and sealing temperature. For example, high vapor pressure fluids like propane will have a large percentage of the seal face width operating with gas between the faces. The hydrocarbon processing industries use this ratio of liquid/gas as the basis for criteria used to predict seal face performance. It is reinterpreted as a vapor pressure margin (see below). Most seal vendors have modeling programs to estimate the fluid state transition point. However, when dealing with fluid mixtures or pump systems designed to handle more than one fluid, optimizing seal selection and piping plans can be more involved.

Vapor Pressure Margin and Product Temperature Margin

A pressure margin between seal chamber pressure and the maximum liquid vapor pressure is a basic requirement for pump and seal system design, has proved to be easy to administer, and it correlates well with other methods of evaluating seal suitability for given service conditions as measured by seal life at an acceptable seal leakage rate.

The pressure margin between seal chamber pressure and the maximum liquid vapor pressure applies to contacting wet single seals and the inner seal of a dual unpressurized configuration. This margin is considered a threshold below which seal vendors must more closely consider the seal piping plan, seal selection, and configuration of adaptive hardware to achieve an acceptable service life.

Pumps that develop low differential pressure and pumps that handle high vapor pressure fluids may not achieve the required margins. For contacting wet seal designs, maintaining an adequate vapor pressure margin helps protect the seal faces against excessive levels of localized boiling of the process fluid at the seal faces. Boiling of the process fluid at the seal faces can cause loss of seal face lubrication and subsequent seal failure. Low density fluids that typically are pumped with low vapor pressure margins are some of the most troublesome fluids to seal and account for a high percentage of seal repairs.

Methods for achieving the required pressure margin may utilize one or a combination of the following options. The selection and application of these solutions are usually the result of mutual agreement between the purchaser and the seal and pump vendors.

- Lowering the seal chamber fluid temperature by cooling the flush fluid
- Raising the seal chamber pressure by removing the back wear ring and plugging impeller balance holes
- Utilizing an external flush fluid
- Raising the seal chamber pressure through the use of a close clearance (floating) throat bushing

Lowering the flush fluid temperature (seal chamber fluid temperature) is always preferable to pressurizing the seal chamber by using a close clearance throat bushing. Bushing wear over a period of time inevitably results in a decreased seal chamber pressure and margin over vapor pressure.

The idea of a vapor pressure margin requirement dates to the Fifth Edition (1971) of API 610 pump specification (if not earlier) requiring seal chamber pressure to be 0.172 MPa (1.72 bar) (25 psi) above suction pressure (assumed to be roughly equal to seal chamber pressure). API 610, Sixth Edition, contained the same requirement. API 610, Seventh Edition, called for conditions leading to a stable film at the seal faces to be jointly established by pump and seal vendors. The Eighth Edition of API 610 referred to API 682, First Edition, which required a margin of at least 0.35 MPa (3.5 bar) (50 psi) above the maximum vapor pressure.

Figure 3 graphically represents the different methods of calculating the actual operating margins and the vapor pressure ratio for a specific process and operating point. The minimum operating margins stated above and the values discussed in the next section are performance recommendations for each method to achieve reliable seal face function. Figure 4 uses the value(s) discussed in the next section and it illustrates how the pressure and temperature margins between process liquid vapor pressure and minimum recommended seal chamber pressure vary between the three calculating methods for a propane service.

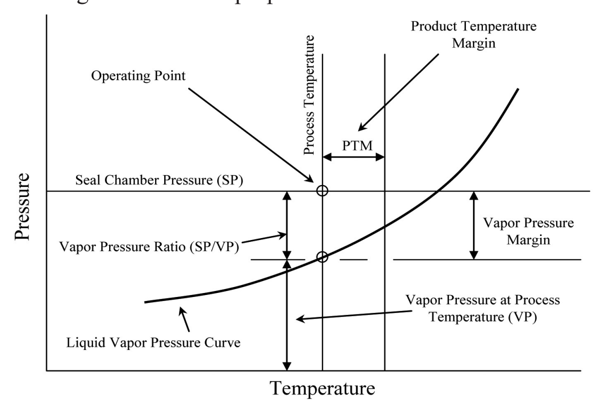

Figure 3. Operating Margin Calculation Methods.

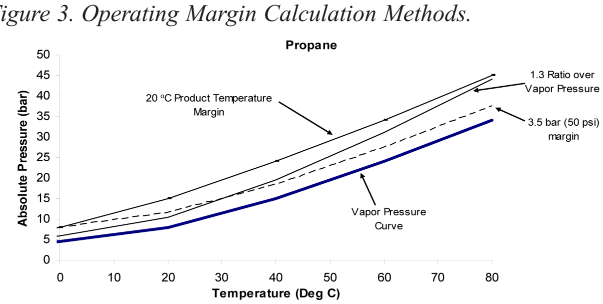

Figure 4. Propane (Operating Margin Calculation Methods).

The vapor pressure margin recommended in API Standards is primarily aimed at hydrocarbon services where the process liquid is often pumped close to its saturated vapor pressure. Sealing of water-based liquids becomes more sensitive to vapor pressure margin and they are typically rated to operate reliably with a temperature margin below their atmospheric boiling point.

Fixed Ratio or Product Temperature Margin in ISO 21049/API 682, Second and Third Edition

Although temperature and vaporization are probably better indicators of reliability, pressure has become the parameter of choice. The pressure margin in API 682, First Edition, of 0.035 MPa (3.5 bar) (50 psi) can be viewed as a "pressure interpretation of a temperature requirement." For example, the Second and Third Edition of API 682 and ISO 21049 required a product temperature margin (PTM) of not less than 20°C (36°F) or a ratio of seal chamber pressure to maximum vapor pressure of 1.3 (30 percent). PTM is the difference between the process temperature in the seal chamber and the saturation temperature of the process liquid at the seal chamber pressure. As an example, the API 682/ISO 21049 qualification tests on propane are at 32°C (90°F) and an absolute pressure of 1.8 MPa (18 bar) (261 psia). The saturation temperature of propane at 1.8 MPa (18 bar) (261 psia) is 52°C (126°F). Therefore, the API 682/ISO 21049 tests are based on a PTM of 52–32 = 20°C (126–90 = 36°F). Although PTM is a single component concept, for mixtures it can be based on the bubble point, but this can be a complex calculation.

Seals with good heat transfer designs (wetted area, thermal conductivity, convection heat transfer) and reduced heat generation (low speed, low pressure, low balance ratio, hydropads, narrow faces, low spring loads, good tribological mating faces) can operate with a smaller PTM than seals without these good characteristics. The fixed minimum margins stated in API 682/ISO 21049 are values that general field experience has proven to give reliable operation. Some seal vendors may claim success at lower margins; this is possible, but must be judged in the context of the specific fluid characteristics and pump service conditions.

The use of a fixed ratio (at least 1.3) between the seal chamber pressure and maximum fluid vapor pressure is a criterion appropriate for hydrocarbons with a steep saturation pressure versus temperature curve and lower pressure applications, but reaches a practical limit at very high pressures. Ratios around 1.3 are usually acceptable for seals using premium materials, having good heat transfer characteristics and having good flush designs with adequate flush rates, like API 682/ISO 21049 Type A seals.

The use of product temperature margin or a 30 percent pressure margin between seal chamber pressure and maximum vapor pressure are reasonable alternate methods for determining that a seal will achieve three years of uninterrupted service, but specific fluid characteristics required with this method may not be readily available. The new draft of API 682 and ISO 21049 will propose reverting back to the 0.35 MPa (3.5 bar) (50 psi) vapor pressure margin in API 682, First Edition. This simpler performance evaluation strategy is adequate for most hydrocarbon services, but may be inadequate on high vapor pressure services.

SEAL PERFORMANCE: PART 3—SEAL CHAMBER TEMPERATURE RISE AND FLUSH

The steady-state temperature of the fluid in the seal chamber is a function of a simple thermodynamic balance. The heat flow into the seal chamber fluid minus the heat flow out of the seal chamber yields a zero net heat flow. This is deceptively simple. In actual applications, the heat flows into and out of the seal chamber fluids are extremely complex.

There are several sources of heat flow into the fluid. These include heat generated due to friction and fluid shear at the seal faces, heat generated due to turbulence caused by the rotating seal components, and heat conducted from the pump through the seal chamber and shaft (or positive heat soak). There are also several sources of heat flow out of the seal chamber. These include heat conducted back into the pump through the seal chamber or shaft (or negative heat soak) and heat lost to the atmosphere through convection and radiation.

When seal face generated heat, heat soak, balance ratio, fluid properties and other factors are combined, required flush flow rates or temperature rise in the seal chamber can be calculated. While operating margin between fluid vapor pressure and flush fluid temperature can determine the correct piping plan and flow rate, a flush flow rate that results in the recommended temperature rise are generally considered adequate to meet seal life expectations. Achieving the required buffer and barrier liquid flow rates with seal Piping Plans 52 or 53 A/B/C that utilize an internal circulating device requires special attention to the piping system curves for these systems. Starting torque, seal power and seal generated heat can be significant issues for small pump drivers, seals at or above the balance diameter and pressure boundaries of API 682/ISO 21049, and for Arrangement 3 seals. Certain seal chamber arrangements such as dead-ended and taper bore boxes have other considerations.

Seal Face Generated Heat

While the calculation of the heat generated by a mechanical seal appears to be a simple matter, several assumptions must be made that introduce potentially large variations in the results. Two variables that are particularly influential are K, the pressure drop coefficient, and f, the effective coefficient of friction.

K is a number between 0.0 and 1.0 that represents the pressure drop as the sealed fluid migrates across the seal faces. For flat seal faces (parallel fluid film) and a nonflashing fluid, K is approximately equal to 0.5. For convex seal faces (converging fluid film) or flashing fluids, K is greater than 0.5. For concave seal faces (diverging fluid film), K is less than 0.5. Physically, K is the coefficient that is used to quantify the amount of differential pressure across the seal faces that is transmitted into the hydraulic component of the fluid film support forces, referred to as the opening force. The opening force is expressed by the following equation:

$$F_{\text{opening}} = A \times \Delta p \times K \tag{1}$$

where:

$F_{opening}$	is the opening force
A	is the area of the seal face
Δp	is the differential pressure
K	is the pressure drop coefficient, dimensionless

For practical purposes, K varies between 0.5 and 0.8. As a standard practice for nonflashing fluids though, a value of 0.5 is selected for K. Although K is known to vary depending upon seal fluid properties (including multiphase properties) and film characteristics (including thickness and the convergent or divergent radial shape of the fluid film, referred to as coning), this value is selected as a benchmark for consistent calculation. The engineer must be aware that this assumption has been made.

The effective coefficient of dynamic friction, f, is a number that is similar to the standard coefficient term that most engineers are familiar with. The standard coefficient of friction term is used to represent the ratio of parallel forces to normal forces. This is normally applied to the interaction between two surfaces moving relatively. These surfaces may be of the same material or different materials.

In a mechanical seal, the two relatively-moving surfaces are the seal faces. If the seal faces were operating dry, it would be a simple matter to determine the coefficient of friction. In actual operation, the seal faces operate under various lubrication regimes, and various types of friction are present.

If there is significant asperity contact, *f* is highly dependent on the materials and less dependent on the fluid viscosity. If there is a very thin fluid film (only a few molecules thick), friction may depend upon interaction between the fluid and the seal faces. With a full fluid film, there is no mechanical contact between the faces and *f* is solely a function of viscous shear in the fluid film. All of these types of friction can be present at the same time on the same seal face.

An effective coefficient of friction is used to represent the gross effects of the interaction between the two sliding faces and the fluid film. Actual testing has shown that normal seals operate with *f* between about 0.01 and 0.18. For normal seal applications, API 682/ISO 21049 has selected a value of 0.07 for *f*. This is reasonably accurate for most water and medium hydrocarbon applications. Viscous fluids (such as oils) will have a higher value, while less viscous fluids (such as liquefied petroleum gas [LPG] or light hydrocarbons) can have a lower value.

The combination of the assumption of *K* and the assumption of *f* can lead to a significant deviation between calculated heat generation results and actual results. Therefore, the engineer must keep in mind that these calculations are useful only as an order-of-magnitude approximation of the expected results. These results shall never be stated as a guarantee of performance.

Calculation of the effective frictional face generated heat first requires an evaluation of the normal forces on the seal face. The opening force has already been discussed but the opposing closing force (normally the higher value) is a sum of the seal spring force and a hydraulic force determined by the seal ring design (refer to section *Balance Ratio* below). The seal face generated heat is the normal force (difference between the closing and opening forces) multiplied by the effective coefficient of friction and translated into a heat rate by adjusting for diameter and shaft speed (refer to section on formulae below).

Balance Ratio

Seal vendors design seal faces with a balance ratio to minimize seal face generated heat consistent with optimum seal life expectations and emission limits. The balance ratio impacts the face closing force, heat generated and the pressure rating of the seal. A balanced seal design will have a balance ratio less than 1, typically in the range of 0.6 to 0.9. The balance ratio can be interpreted as the proportion of the seal chamber pressure that is helping to create the closing force on the seal face. For example, the typical range of 0.6 to 0.9 balance ratio means that there is a 10 to 40 percent reduction in the hydraulic pressure load on the faces. Type A pusher seal designs will often require a step in the shaft sleeve as shown in Figure 5. The step in the shaft sleeve increases the area of the seal face on which seal chamber pressure is offset or balanced resulting in a reduction in face load and face generated heat.

Balance diameter varies with seal design, but for Type A seals it is normally the diameter of the sliding contact surface of the dynamic O-ring. For the inner Type A seal of a dual seal configuration the sliding surface can vary depending on whether the pressure is internal or external. For Type B and C seals, the balance diameter is normally the mean diameter of the bellows, but this will vary with the pressure. Contact the seal vendor for determination of the balance diameter under varying pressure conditions.

An example of the seal balance ratio measurement points shall be as shown in Figure 5. There are other methods of achieving pressure balance under pressure reversals. Contact the seal vendor if the sliding contact surface of the dynamic O-ring is not readily apparent.

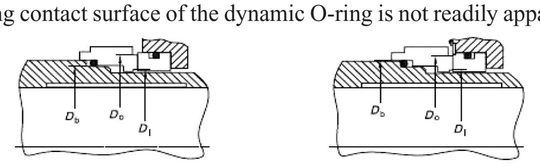

Figure 5. Illustration of Balance Ratio Measurement Points.

Balance Ratio Calculation Inputs

- D_o is the seal face outside diameter
- D_i is the seal face inside diameter
- D_b is the balance diameter of the seal

Balance Ratio Formulas

For seals externally pressurized, the seal balance ratio, *B*, is defined by the equation:

$$B = (D_o^2 - D_b^2) / (D_o^2 - D_i^2) \tag{2}$$

For seals internally pressurized, the seal balance ratio, *B*, is defined by the equation:

$$B = (D_b^2 - D_i^2) / (D_o^2 - D_i^2) \tag{3}$$

Seal Face Generated Heat Calculation Inputs

Required inputs:

- D_o is the seal face contact outer diameter, expressed in millimeters
- D_i is the seal face contact inner diameter, expressed in millimeters
- D_b is the effective seal balance diameter, expressed in millimeters
- F_{sp} is the spring force at working length, expressed in Newtons
- Δ_p is the pressure differential across the seal face, expressed in megapascals
- N is the face rotational speed, expressed in revolutions per minute
- f is the coefficient of friction (assume 0.07)
- K is the pressure drop coefficient (assume 0.5)

Seal Face Generated Heat Calculation Formulas

- Face area, *A*, (mm^2)

$$A = \frac{\pi(D_o^2 - D_i^2)}{4} \tag{4}$$

- Seal balance ratio, *B*

$$B = \left(\frac{D_o^2 - D_b^2}{D_o^2 - D_i^2}\right) \tag{5}$$

- Spring pressure, p_{sp}, (MPa)

$$p_{sp} = \frac{F_{sp}}{A} \tag{6}$$

- Total face pressure, p_{tot}, (MPa)

$$p_{tot} = \Delta p\left(B - K\right) + p_{sp} \tag{7}$$

- Mean face diameter, D_m, (mm)

$$D_m = \frac{\left(D_o + D_i\right)}{2} \tag{8}$$

- Running torque, *T*, (N-m)

$$T_r = p_{tot} \times A \times f \left(\frac{D_m}{2\,000}\right) \tag{9}$$

- Starting torque, T_s, (N-m) estimated at three to five times running torque

$$T_s = T_r \times 4 \tag{10}$$

- Seal face generated heat, H, (kW)

$$H = \frac{\left(T_r \times N\right)}{9548} \tag{11}$$

Seal Face Generated Heat Example Calculation

Fluid: Water
Pressure: A gauge pressure of 2 MPa (20 bar)
Speed: 3000 r/min

Inputs:
D_o = 61.6 mm
D_i = 48.9 mm
D_b = 52.4 mm
F_{sp} = 190 N
Δ_p = 2 MPa (20 bar)
N = 3 000 r/min
f = 0.07
K = 0.5

- Calculate face area:

$$A = \left(\frac{\pi}{4}\right) \times \left(61{,}6^2 - 48{,}9^2\right) = 1\ 102\ \text{mm}^2 \tag{12}$$

- Calculate seal balance ratio:

$$B = \frac{\left(61{,}6^2 - 52{,}4^2\right)}{\left(61{,}6^2 - 48{,}9^2\right)} = 0{,}746 \tag{13}$$

- Calculate spring pressure:

$$p_{\text{sp}} = \left(\frac{190}{1\ 102}\right) = 0{,}172\ \text{N/mm}^2\ \text{(MPa)} \tag{14}$$

- Calculate total face pressure:

$$p_{\text{tot}} = 2\left(0{,}746 - 0{,}5\right) + 0{,}172 = 0{,}664\ \text{N/mm}^2\ \text{(MPa)} \tag{15}$$

- Calculate mean face diameter:

$$D_{\text{m}} = \frac{\left(61{,}6 + 48{,}9\right)}{2} = 55{,}25\ \text{mm} \tag{16}$$

- Calculate running torque:

$$T_{\text{r}} = 0{,}664 \times 1\ 102 \times 0{,}07 \left(\frac{55{,}25}{2\ 000}\right) = 1{,}42\ \text{N}\cdot\text{m} \tag{17}$$

- Calculate starting torque:

$$T_{\text{s}} = 1{,}42 \times 4 = 5{,}68\ \text{N}\cdot\text{m} \tag{18}$$

- Calculate seal face generated heat:

$$H = \frac{\left(1.42 \times 3000\right)}{9548} = 0.446\ \text{kW} \tag{19}$$

SEAL PERFORMANCE: PART 4—HEAT SOAK

Heat soak is the heat transferred from the pump and pumped fluid to fluid in the seal chamber. The pump and pumped fluid heat are transferred into and out of the seal chamber in amounts dependent on the service conditions and pump design.

In some cases, assumptions can be made that simplify the model. For example, consider a single seal with Piping Plan 11, 12, 13, or 31. With these piping plans, the fluid injected into the seal chamber will be at pump process temperature so heat soak and heat loss to the atmosphere can be ignored. Except in the case of large seals at high speeds, heat generation due to liquid turbulence is usually insignificant and can also be ignored.

In applications that use a Piping Plan 21, 22, 23, 32, or 41, the fluid injected into the seal chamber may be at a significantly lower temperature than the pump temperature. If this is the case, there can be a significant heat flow or heat soak into the seal chamber from the pump. The calculation of heat soak is a complex matter, requiring detailed analysis or testing and a thorough knowledge of the specific pump construction, materials, and process liquid properties. Experience has shown in hydrocarbon processing industries that efforts to minimize heat soak with the use of cooling water in seal chamber jackets have been largely unsuccessful due to fouling and the limited cross sectional thickness of the pump parts.

It is necessary for the seal vendor to make an estimation of the rate of heat soak and the empirical formula below can be used to provide an estimation of the level. It is unable to consider all the differences in equipment design and hence the prediction is usually greater than may be experienced in the field.

- Heat soak calculation inputs

U is the material property coefficient
A is the effective heat transfer area
D_b is the seal balance diameter, expressed in millimeters
ΔT is the difference between pump process temperature and the desired seal chamber temperature, expressed in Kelvin

- Heat soak formula

If specific knowledge of the pump construction and pumped product properties is not available, the heat soak ($Q_{heatsoak}$ [kW]) can be estimated by the equation:

$$Q_{\text{heatsoak}}\ \text{(kW)} = U \times A \times D_{\text{b}} \times \Delta T \tag{20}$$

A typical value for (U × A) that can be used for estimating purposes with stainless steel sleeve and gland construction and steel pump construction is 0.000 25. This value will generally provide a conservative estimate of heat soak.

- Heat soak example calculation

U × A = 0.000 25
D_b = 55 mm (seal balance diameter)
Pump process temperature = 175°C
Desired seal chamber temperature = 65°C
ΔT = 175 − 65 = 110 K
$Q_{heatsoak}$ = 0.000 25 × 55 × 110 = 1.5 kW

SEAL PERFORMANCE: PART 5—SEAL FLUSH FLUID TEMPERATURE RISE AND FLOW RATE

Seal Flush Fluid Temperature Rise

Temperature rise of the flush fluid as it travels through the seal chamber is a function of a thermodynamic balance applied to a liquid flow rate. The seal face generated heat is added to the heat soak, if relevant to the piping plan, and applying this to a known flow rate using a thermodynamic formula, a temperature rise can be predicted.

The temperature rise calculated using the following formulas results in the average temperature rise of the flush fluid in the seal chamber. However, within the seal chamber, there are areas that are hotter and cooler than the mean fluid temperature. An effective flush design and flow rate are required to ensure that the area around the seal face is effectively cooled.

- Seal flush fluid temperature rise calculation inputs

Q is the heat generation at the seal faces, expressed in kilowatts

$Q_{heatsoak}$ is the heat transferred from the pump and pumped fluid to fluid in the seal chamber, expressed in kilowatts
q_{inj} is the injection flow rate, expressed in liters per minute
d is the relative density (specific gravity) of the injected fluid at the pump process temperature
c_p is the specific heat capacity of the injected fluid at the pump process temperature, expressed in joules per kilogram Kelvin

- Seal flush fluid temperature rise formula—without heat soak

The differential temperature, ΔT (in Kelvin), can be calculated by the following equation:

$$\Delta T = \frac{(60\,000 \times Q)}{(d \times q_{\text{inj}} \times c_p)} \tag{21}$$

- Seal flush fluid temperature rise formula—with heat soak

The differential temperature, ΔT (in Kelvin), including the effects of heat soak can be calculated using the inputs described above and the following equation:

$$\Delta T = 60\,000 \times \frac{(Q + Q_{\text{heatsoak}})}{(d \times q_{\text{inj}} \times c_p)} \tag{22}$$

- Seal flush fluid temperature rise example calculation (without heat soak)

Q = 0.9 kW
q_{inj} = 11 l/min
d = 0.75
c_p = 2 300 J/kg·K

- Calculate the seal flush fluid temperature:

$$\Delta T = \frac{(60000 \times 0.9)}{(0.75 \times 11 \times 2300)} = 2.8\ K \tag{23}$$

Seal Flush Flow Rate

In some applications, it is necessary to specify the flush rate required to maintain the seal chamber temperature below a certain level. In this case, the maximum allowable temperature rise would be calculated by subtracting the flush liquid inlet temperature from the maximum allowable temperature in the seal chamber (or buffer/barrier seal chamber). For good seal performance, the maximum temperature rise should be limited to 5.6 K (10°R) for Arrangement 1 and Arrangement 2 inner seal flush flow rates and 8 K (14.5°R) to 16 K (29°R) for buffer/barrier flow rates depending on the properties of the liquid. It is then a simple matter of rearranging equations to solve for the required flush flow rate.

The temperature rise used in these calculations is the sealing chamber temperature rise. The temperature rise at the seal faces will be greater than the chamber temperature rise. If the seal flush flow rate calculations (below) are used to calculate a minimum flow rate based on sealing chamber temperature, the seal faces may overheat and perform poorly. Depending on the application, a design factor of at least two may need to be applied to the calculated required minimum flow rate. The injection must also be directed at the seal interface to ensure proper cooling.

- Seal flush flow rate calculation inputs for Arrangement 1 and 2

Q is the heat generation at the seal faces, expressed in kilowatts
$Q_{heatsoak}$ is the heat transferred from the pump and pumped process fluid to fluid in the seal chamber, expressed in kilowatts
ΔT (in Kelvin) is the desired maximum differential temperature
d is the relative density (specific gravity) of the injected fluid at the temperature of the seal chamber inlet
c_p is the specific heat capacity of the injected fluid at the temperature of the seal chamber inlet, expressed in joules per kilogram Kelvin

- Seal flush flow rate formula

For flush flow in liters per minute without heat soak typical for seals with Piping Plan 11, 12, 13, or 31, the equation would be:

$$q_{\text{inj}} = \frac{(60\,000 \times Q)}{(d \times \Delta T \times c_p)} \tag{24}$$

For flush flow in liters per minute with heat soak typical for seals with Piping Plan 21, 22, 23, 32, or 41, the equation would be:

$$q_{\text{inj}} = 60\,000 \times \frac{(Q + Q_{\text{heatsoak}})}{(d \times \Delta T \times c_p)} \tag{25}$$

- Seal flush flow rate example calculation (Arrangement 1 without heat soak)

Q = 0.9 kW
ΔT_{max} = 5.6 K
d = 0.90
c_p = 2593 J/kg·K

- Calculate the minimum seal flush flow rate:

$$q_{inj} = \frac{(60000 \times 0.9)}{(0.9 \times 5.6 \times 2593)} = 4.1\ \text{l/min} \tag{26}$$

SEAL PERFORMANCE: PART 6—PIPING PLAN 52AND 53A SYSTEMS FLOW RATE CALCULATIONS AND THE IMPACT OF PIPING SIZE

Introduction

Buffer/barrier seal chamber generated heat and the appropriate flush flow for Piping Plan 52 and 53A seal systems are particularly unique because they usually utilize an internal circulating device, the buffer/barrier fluid circulates through the reservoir/accumulator, and the exchanger would be internal to the reservoir/accumulator. Estimated system friction curves are included in this section for 52 and 53A Piping Plans. These system curves represent piping losses and do not include losses through porting in the gland plate or other components.

Unlike Piping Plans 52 and 53A, Piping Plans 53B and 53C may utilize an external exchanger and the circulating flow does not pass through the accumulator. There would be a significant increase in system friction if losses through an external exchanger are added to the interconnecting piping losses.

Performance curves for the internal circulating devices used with any 52 or 53 Piping Plan will vary depending on the type and design of device, the operating clearance, the gland plate design, fluid properties, and the peripheral velocity. As a result, the specific device performance curve should be overlaid on the Plan 52 or 53A/B/C system curve to determine the appropriate interconnecting pipe/tube size so the desired flow will be achieved. API 682/ISO 21049 currently advise a change to 20 mm (0.75 inches) diameter pipe or tubing on shaft sizes > 63.5 mm (2.5 inches). The new clauses in Section 8 of the draft of API 682/ISO 21049 require a change to a larger pipe or tube size also based on a flush flow rate > 8 l/min and/or a total of > 2.5 m of interconnecting pipework length. When there is any doubt about these parameters, 20 mm (0.75 inch) pipe or tubing should be used because, as can be seen in the systems curves below, friction losses are significantly minimized. An analysis of the parameters would determine that an increase to a 25 mm (1 inch) pipe offered little benefit. While not modeled, whenever possible, the purchaser should consider tangential oriented buffer/barrier fluid gland plate connections to improve flush flow rates.

While selected less frequently than internal circulating devices, seal vendors can also offer an external circulating pump to ensure that the desired flush flow is achieved.

This section provides the background behind the pipe size recommendations in API 682/ISO 21049 and describes how a seal vendor might analyze and check the performance of a Piping Plan 52 or 53A system. Illustrative diagrams are shown.

System Resistance Curve for a Piping Plan 52 and 53A

Piping Plan 52 and 53A seal systems have been modeled with standardized stub pipes with lap joints to the gland plate. The length of the stub pipe has been assumed at 150 mm (6 inches), as shown in Figure 6. The stub pipe material has been assumed as ½ inch schedule 80 pipe irrespective of whether the main circuit is constructed of pipe or tube.

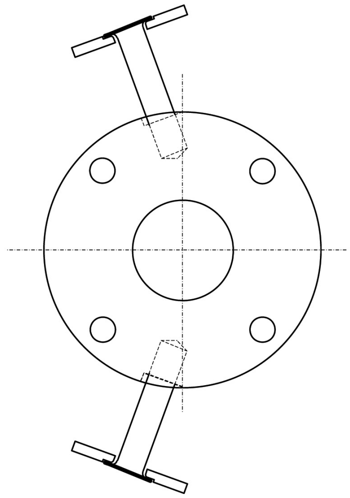

Figure 6. Gland Plate Model.

Fluid properties used to generate the system curves are:

- Water with a specific gravity (SG) of 0.9983 at 20°C and viscosity of 1 cP.
- Oil with an SG of 0.85 at 20°C and viscosity of 10 cP.
- Maximum flush flow rate is assumed to be 20 l/min (1.2 m^3/h) (5 gpm).

The general model used for the barrier fluid system is as shown in Figure 7. The interconnecting piping to and from the reservoir have been assumed to be of equal length, and this has been set at 2.5 m per leg. The inlet to the gland plate is assumed to be from the lower pipe leg with an exit from the gland plate as the upper pipe leg (refer to Figure 6).

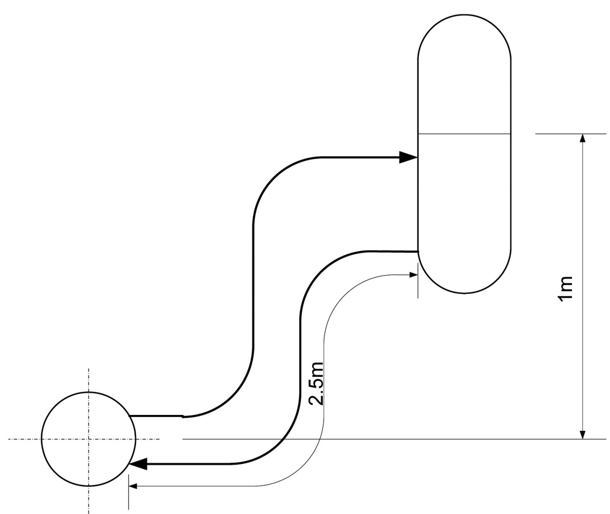

Figure 7. System Model.

The piping materials are either schedule 80 pipe or tube and the diameter and bore used to calculate the system losses are shown in Table 1.

Table 1. Pipe and Tube Sizes.

Diameter	Metric Bore (mm)	Imperial Bore (inch)
15 mm (0,5 inch) Pipe	13,84	0,546
20 mm (0,75 inch) Pipe	18,88	0,742
12.7 mm (0,5 inch) Tube	9 (OD 12 x 1,5 wall)	0,37 (0,5 x 0,065 wall)
20 mm (0,75 inch) Tube	16 (OD 20 x 2 wall)	0,543 (0,75 x 0,095 wall)

Estimated system curves for the piping sizes shown in Table 1 are illustrated in Figures 8 and 9 for mineral oil and water. Tubing sizes and wall thickness can vary and the layout and length of piping will also vary between installations, so the curves in Figure 8 and 9 should be used as a guideline rather than an exact reflection of a specific field installation.

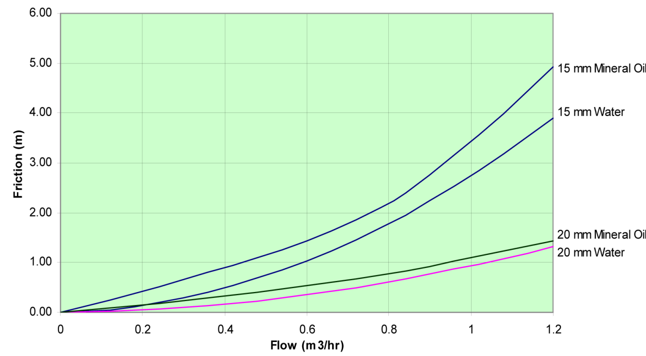

Figure 8. Pipe System Friction Curves.

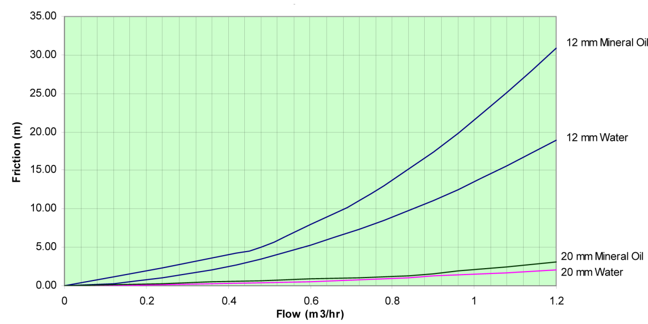

Figure 9. Tube System Friction Curves.

Internal Circulating Device Performance Verification

When an internal circulating device is used the seal vendor should evaluate its performance curve. The curve should illustrate head versus capacity and the vendor should also confirm that the NPSH(r) is satisfied over the entire flow range of the device. The device NPSH(r) may be represented by a curve or data. Users should carefully review applications using an internal circulating device, but especially when:

- The process fluid temperature exceeds 176°C (350°F),
- The shaft rotating speed is less than 3000 rpm,
- Variable speed drives are used,
- Shaft diameter is less than 50 mm (2 inches),
- The total length of interconnecting pipework exceeds 5 m (16.4 feet),
- A radial clearance smaller than that specified in the draft clause 6.1.2.6 of API 682 Fourth Edition/ISO 21049 is proposed to achieve the required flush flow rate.

Performance of the internal circulating device should exceed the required flush flow rate using the specified buffer/barrier fluid at all operating and start up conditions. The system resistance curve (based on the auxiliary components supplied, the specific buffer/barrier fluid, its mean settlement temperature, and the specific seal system layout) should be plotted over the circulating device performance curve. The typical system resistance curves for Piping Plan 52 and 53 tube and pipe systems should assume standard guidelines are followed for installation of these plans. Piping Plan 23 seal systems will likely have steeper system resistance curves compared to Piping Plan 52 or 53 systems because of the additional system resistance of the heat exchanger. Piping Plan 23 systems typically utilize heat exchangers with the process fluid inside the exchanger tubing.

To improve flush flow circulation rates, inlet and outlet connections for the internal circulating device should be tangential and oriented to facilitate thermosyphon. In addition, the seal chamber or gland plate inlet and outlet ports should properly align with the internal circulating device and their drill-through diameters designed as large as is practical.

Figures 10 and 11 illustrate the intersection points between a hypothetical circulating device performance curve(s) and the system curves. These intersection points indicate the estimated comparative flow that can be achieved with each combination of pipe and tube size and mineral oil or water buffer/barrier systems. Please note:

- Performance data for the circulating device is identical for the tubing and pipe plots.
- The values for the flow axis are identical.
- The values for the head axis are identical.
- Variations in the resulting intersection points are solely the result of differences in the system curves created by combinations of fluid with different size pipe or tubing.

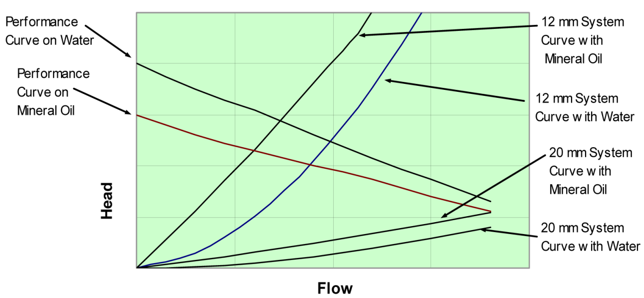

Figure 10. Tube System Curves and Circulating Device Performance Illustration.

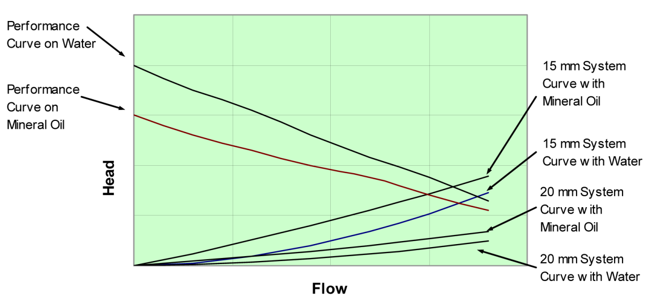

Figure 11. Pipe System Curves and Circulating Device Performance Illustration.

Figures 10 and 11 also show the system resistance in tubing systems is normally significantly higher than pipe systems for the same fluid, nominal size, and flow rate producing steeper tube system curves. As a result, the performance curves intersect the tubing system curves at a lower flow compared to same nominal size pipe. The user should be aware that the highest flush rate is achieved with an interconnecting pipework selection of pipe and with a size selection of 20 mm.

Typical Flush Flow Rates for Arrangement 3 CW Seals

The following are typical required flush flow rates for an Arrangement 3CW seal, pressurized dual contacting wet seals, graphically illustrated. The curves are based on:

- A barrier fluid specific heat C_p of 2093 J/KgK (0.5 Btu/lb°F)
- Shaft speed 3600 rpm
- Seal balance ratio of 0.75
- A flush flow temperature rise of 5.6 K (10°R)
- Seal chamber pressure of 1.034 MPa (10.034 bar) (150 psig)
- Barrier fluid pressure of 1.379 MPa (13.79 bar) (200 psig)
- A safety factor for flush flow of 1.0

Note: API 682/ISO 21049 require a maximum flush flow temperature rise of 8 K (15°R) or 16 K (30°R) depending on the barrier fluid type. The curves thus have an effective safety factor built into the output.

Note: For barrier fluids with a different specific heat, divide the predicted graph flow rate by the C_p ratio (actual barrier C_p divided by 2093 J/KgK [0.5 Btu/lb°F]).

While curves are provided for pumped fluid temperatures above 176°C (350°F), achieving an adequate flow using an internal circulating device for higher temperature applications becomes increasingly difficult and a Piping Plan 54 may be required for these services. This is especially true considering that the illustrated flush rates are based on a safety factor of 1.

Figures 12, 13, 14, and 15 show typical flush flow for Arrangement 3 CW Seals.

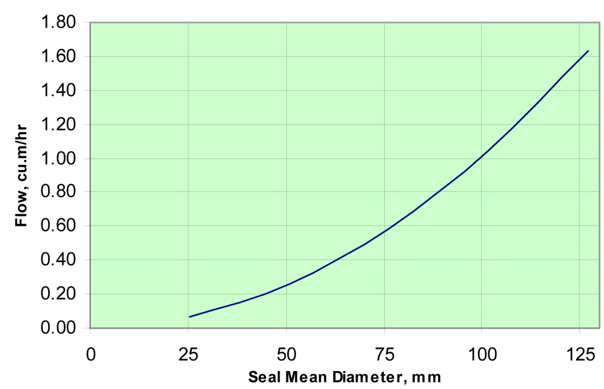

Figure 12. Typical Required Flush Flow for Arrangement 3 CW Seals Without Heat Soak Considered and a Pumped Process Temperature of 54°C (130°F).

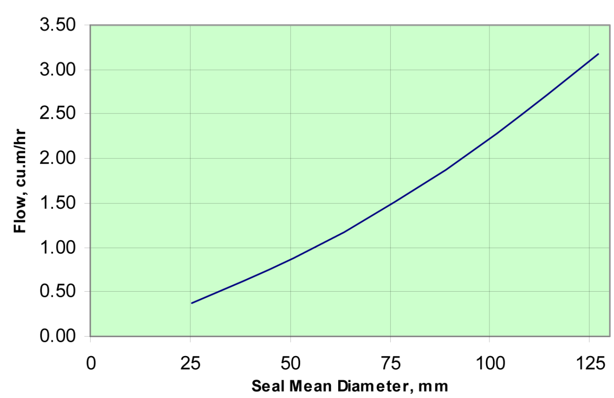

Figure 13. Typical Required Flush Flow for Arrangement 3 CW Seals with Heat Soak Considered and a Pumped Process Temperature of 176°C (350°F).

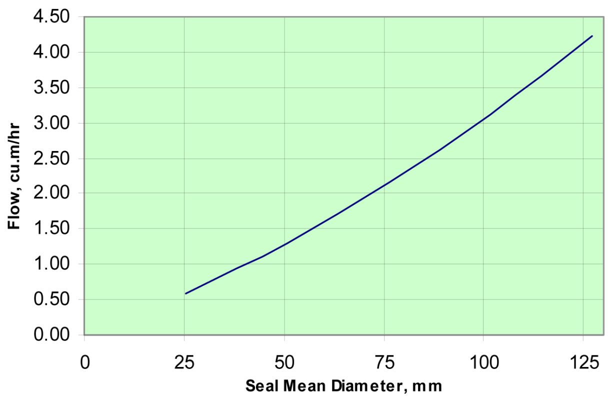

Figure 14. Typical Required Flush Flow for Arrangement 3 CW Seals with Heat Soak Considered and a Pumped Process Temperature of 260°C (500°F).

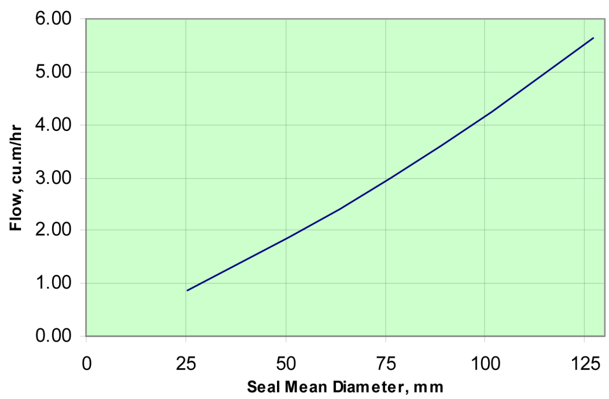

Figure 15. Typical Required Flush Flow for Arrangement 3 CW Seals with Heat Soak Considered and a Pumped Process Temperature of 371°C (700°F).

SEAL PERFORMANCE: PART 7—PIPING PLAN 53A AND 53B BARRIER PRESSURE OPERATION AND CALCULATIONS

Piping Plans 53A and 53B provide barrier liquid to Arrangement 3 dual seals at a pressure above the maximum (process pumped fluid) seal chamber pressure by using a gas charged reservoir or accumulator. Piping Plan 53C also provides a pressure margin above the maximum seal chamber pressure, but it is achieved by using a reference line from the seal chamber and a piston accumulator rather than a gas charged accumulator. Pressure variations in Piping Plans 53A and 53B can be significant due to the use of a gas charged accumulator so these piping plans are covered in detail by this section. Piping Plan 53C pressure fluctuations are minimal and are not covered in this tutorial.

The maximum process fluid seal chamber pressure may vary for a variety of reasons such as pump design, static liquid level, and pressure relief setting on the suction vessel. It is important that the maximum suction pressure be reviewed and confirmed prior to starting the calculation of the gas charge pressure for either Piping Plan 53A or 53B.

The minimum barrier liquid pressure will normally include a pressure margin above the maximum seal chamber pressure to avoid a pressure reversal across the inner seal. A typical pressure margin may be 0.14 MPa (1.4 bar) (20 psi), but can be higher or lower in some circumstances.

When properly selected the Piping Plan 53A barrier reservoir pressure or the Piping Plan 53B gas charge pressure will avoid a pressure reversal at the inner seal and also avoid overpressurizing the seal or seal flush system components due to seasonal or diurnal fluctuations in ambient temperature or solar radiation exposure.

Category 1 seal and seal flush system components are rated for a minimum gauge pressure of 2 MPa (20 bar) (300 psi). Category 2 and 3 seal and seal flush system components are rated for a minimum gauge pressure of 4 MPa (40 bar) (600 psi). Some seals may have a pressure rating lower than their associated flush system components. It is important to verify the pressure rating of seals and associated flush system components and confirm that pressure fluctuations do not exceed these ratings. For example, Type B or C seals typically have lower differential pressure rating than Type A seals. Some dual seal configurations may utilize the pump seal chamber as part of the barrier liquid system so the pump seal chamber would need to be considered in the pressure evaluation.

With both Piping Plans 53A and 53B, as barrier fluid pressure increases seal face related friction also increases (refer to PART 3). Users should be aware that it may become difficult or impossible to rotate some pumps prior to startup when the seal is pressurized. In small pumps, seal face friction may also contribute significantly to the motor load and it is possible to experience an overload condition (high amps) causing shutdown of a marginally sized motor.

Circulation of barrier liquid at required flow rates is important for seal reliability (refer to PART 6).

Piping Plan 53A Operation

Figure G.15 in Annex G of API 682/ISO 21049 illustrates a typical Piping Plan 53A system. The barrier liquid reservoir is pressurized by an outside source, typically the plant nitrogen system, another plant gas source or bottled gas. A pressure regulator should be installed upstream of the gas supply isolation valve, but the pressure regulator is not normally in the scope of supply of the pump or seal vendor and hence is not shown in Figure G.15. However, to avoid a release of potentially hazardous gas, the pressure regulator is not normally self relieving.

When the source of gas for pressurizing the reservoir is bottled gas, the user may want to consider the use of a low pressure alarm on the gas bottle, upstream of the pressure regulator, for early indication of the need to replace the gas bottle. This low pressure alarm is not normally in the scope of supply of the seal or pump vendor.

The minimum barrier liquid pressure and the set point for the pressure regulator are the maximum seal chamber pressure plus a pressure margin. However, the reservoir pressure may vary due to diurnal and seasonal ambient temperature changes, changes in barrier liquid temperature, and/or solar radiation exposure (if applicable).

The barrier liquid in a Piping Plan 53A system circulates through the reservoir and the reservoir usually incorporates a cooler. Since the gas in the reservoir is exposed to the circulating barrier liquid, reservoir pressure variations are complicated by the influence of the barrier liquid on the gas temperature and gas solubility. During stable operation, it is reasonable to expect the barrier liquid temperature to reach equilibrium at a temperature above average ambient temperatures because of:

- Heat soak into the circulating barrier liquid due to an elevated process pumped liquid temperature (refer to PART 4).
- Seal face generated heat from both seals (refer to PART 3) which require a temperature difference above the cooling water in the reservoir to be removed from the barrier liquid flow.

It is unlikely that barrier liquid pressure will exceed the rated pressure of Category 2 or 3 systems if the charge gas supply is a plant nitrogen system that normally operates at or below a gauge pressure of 1MPa (10 bar) (150 psi). This pressure is significantly below the minimum rating of Category 2 and 3 seals and flush system components. Pressure fluctuations due to diurnal or seasonal ambient temperature variations, barrier liquid temperature changes, or solar exposure will also likely not exceed the pressure rating of Category 2 or 3 seals or flush system components.

However, Category 1 seals and support systems are rated for lower pressure so it is important to verify that pressure fluctuations do not exceed component or support system ratings.

For charge gas supply systems that operate at a pressure above 1MPa (10 bar) (150 psi) it is important to verify that pressure fluctuations do not exceed component or support system ratings for all seal categories.

If the gas supply isolation valve is closed between the barrier liquid reservoir and the pressure regulator/gas supply system, it is possible to experience a drop in reservoir pressure caused by either a drop in ambient temperature, a drop in barrier liquid temperature, or a drop in reservoir level. With the gas supply isolation valve closed, users should consider the impact of ambient temperature extremes and changes in barrier liquid temperature on reservoir pressure. Failure to do so may result in a pressure reversal across the inner seal. Figure G.15 in Annex G of API 682/ISO21049 shows this valve as normally open to avoid this scenario.

Barrier liquid level will drop due to seal leakage. The need to add barrier liquid to the reservoir occurs when the operating volume of barrier liquid is used. A level indicator and level transmitter with a low level alarm are provided on Piping Plan 53A systems to indicate the need to add barrier liquid. Filling frequencies are similar to those required by 53B systems; the new draft of API 682/ISO 21049 recommends a minimum of 28 days.

In addition to a level transmitter, Piping Plan 53A systems are also provided with a pressure transmitter. As a minimum, a low alarm set point is required for level and pressure, however a high alarm set point for each is optional.

Figure 16 illustrates a Piping Plan 53A system for a reservoir continuously connected to the gas supply, typically through a pressure regulator that is not self relieving. The associated calculations are consistent with the figure. It is reasonable to expect an increase in reservoir pressure caused by exposure to maximum ambient temperature, an elevated barrier liquid temperature, and/or solar radiation. The graph and associated calculations assume the reservoir gas temperature reaches the maximum ambient, maximum barrier liquid temperature, and solar radiation temperature. During stable operation, the gas temperature fluctuations may be minimized because of exposure to the barrier fluid as it flows through the reservoir. Also, any unsafe pressure rise may be limited if the pressure regulator is self relieving or if a relief valve is installed; however, neither of these is included in a typical Piping Plan 53A system.

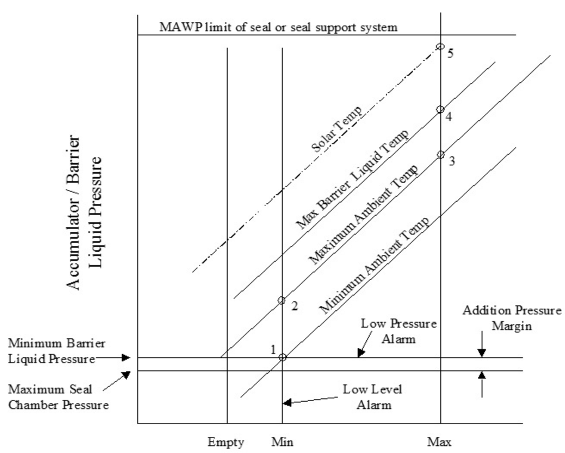

Figure 16. Flush Plan 53A Barrier Liquid Level.

Figure 16 also illustrates important calculation points for Piping Plan 53A systems. Refer to the calculation section below for a detailed description of each plotted point.

The initial charge of barrier liquid is normally added prior to pressurizing the reservoir. Most systems have a pressure regulator that will be set at the minimum barrier system pressure so the barrier system pressure will not fall below this value, but the barrier system pressure may increase due to diurnal variations in ambient temperature or changes in barrier liquid temperature. If ambient temperature drops causing a drop in reservoir pressure the gas supply regulator will add gas to maintain the specified pressure. Assuming the regulator is not self venting and there is no relief valve in the barrier liquid system, then it is possible for the reservoir pressure to increase with increasing ambient or barrier liquid temperature.

Piping Plan 53A Calculation Tutorial and Formula

The following discussion refers to the illustrated "numbered" points in Figure 16.

- Point #1—Minimum barrier liquid pressure at minimum liquid level—This pressure is the basis for all subsequent calculations and is the sum of the maximum seal chamber pressure and a pressure margin; it is the set point for the pressure regulator. For the purposes of the following calculations, this pressure is assumed to be at the minimum ambient temperature because it is normally maintained by a pressure regulator. It is also the recommended alarm pressure.
- Point #2—Calculates the reservoir pressure using the value of Point #1, but applies a ratio of maximum ambient temperature and minimum ambient temperature.

  Point #2 pressure = Pressure at Point #1 × (maximum ambient temp [°C + 273] [or °F + 460]/(minimum ambient temp [°C + 273] [or °F + 460]).
- Point #3—Calculates the reservoir pressure using the value of Point #2, but applies a ratio of maximum gas volume (at minimum barrier liquid level) and minimum gas volume (maximum barrier liquid level)

  Point #3 pressure = Pressure at Point #2 × (maximum gas volume/minimum gas volume).
- Point #4—Calculates the reservoir pressure using the value at Point #3, but applies a ratio of maximum barrier liquid temperature and maximum ambient temperature.

  Point #4 pressure = Pressure at Point #3 × (maximum barrier liquid temp [°C + 273] [or °F + 460]/(maximum ambient temp [°C + 273] [or °F + 460]).
- Point #5—Calculates the reservoir pressure using the value of Point #3, but applies a ratio of solar radiation temperature and maximum ambient temperature.

  Point #5 pressure = Pressure at Point #3 × (solar radiation temp [°C + 273] [or °F + 460]/(maximum ambient temp [°C + 273] [or °F + 460]).

Piping Plan 53A Example Calculation

The example calculation is for a Piping Plan 53A application showing the effects of solar radiation. The example seal support system is designed for a gauge pressure of 4 MPa (40 bar) (600 psi) typical of Category 2 and 3 seal systems.

- Assumptions include site conditions:
  40°C Maximum site temperature
  −10°C Minimum site temperature
  68°C Maximum barrier liquid temperature
  80°C Maximum solar radiation temperature
- Seal system assumptions:
  20 liter total reservoir volume
  10 liter reservoir gas volume at minimum barrier liquid level (10 liter barrier liquid volume in reservoir)
  6 liter reservoir gas volume at maximum barrier liquid level (14 liter barrier liquid volume in reservoir)
  0.7 MPa (7 bar) maximum seal chamber gauge pressure (0.8 MPa absolute pressure)
  0.14 MPa (1.4 bar) pressure margin above maximum seal chamber pressure

- 1. Determine the minimum operating reservoir pressure at minimum liquid level assuming minimum ambient temperature.
  Point #1 = 0.8 + 0.14 = 0.94 MPa (absolute pressure)
  (a gauge pressure of 0.84 MPa) (8.4 bar) (122 psig)

Note: This value represents the low pressure alarm pressure.

- 2. Calculate the corresponding reservoir pressure at maximum ambient temperature and minimum barrier liquid level.
  Point #2 = 0.94 × (40 + 273)/(−10 + 273) = 1.119 MPa (absolute pressure)
  (a gauge pressure of 1.019 MPa) (10.19 bar) (148 psig)
- 3. Calculate the corresponding reservoir pressure at the maximum barrier liquid level and maximum ambient temperature.
  Point #3 = 1.119 × 10/6 = 1.865 MPa (absolute pressure)
  (a gauge pressure of 1.765 MPa) (17.65 bar) (256 psig)
- 4. Calculate the corresponding reservoir pressure at the maximum barrier liquid level and temperature.
  Point #4 = 1.865 × (68 + 273)/(40 + 273) = 2.032 MPa (absolute pressure)
  (a gauge pressure of 1.932 MPa) (19.32 bar) (280 psig)
- 5. Calculate the corresponding reservoir pressure at the solar radiation temperature.
  Point #5 = 1.865 × (80 + 273)/(40 + 273) = 2.103 MPa
  (a gauge pressure of 2.003 MPa) (20.03 bar) (290 psig)

Piping Plan 53B Operation

Figure G.16 in Annex G of API 682/ISO 21049 illustrates a typical Piping Plan 53B system. The barrier liquid is pressurized using a gas charge inside a bladder within the accumulator. Unlike a typical Piping Plan 53A system, after a Piping Plan 53B accumulator is charged to a predetermined gas pressure, the accumulator is then isolated from the gas source during operation.

Accumulator pressure will drop due to seal leakage and reduced barrier liquid volume. Knowing the expected rate of seal leakage (determined by empirical data or estimated by the seal vendor) and the operating volume of barrier liquid the frequency of refilling the accumulator with barrier liquid can be determined. It is reasonable to expect a filling frequency of 28 days or more, but this is dependent on the volume of barrier liquid, the leakage rate and the alarm strategy employed.

Accumulator pressure will also be affected by the gas temperature in the bladder. The barrier liquid does not flow through the accumulator, so the bladder gas temperature will change with ambient temperature (and solar exposure if not shaded). Accumulator pressure variations can be significant. Accumulator gas charge pressure should consider the extremes of ambient temperature and the temperature during commissioning the system in the same way as has been discussed with Piping Plan 53B. Failure to do so may result in a pressure reversal across the inner seal or over pressurizing the seal or seal support system components.

The calculations that follow illustrate a method to determine the initial gas charge pressure to avoid problems associated with variations in barrier liquid pressure. If the accumulator pressure at minimum liquid volume and minimum ambient conditions is equal to or greater than the maximum seal chamber pressure plus the pressure margin (Point #1 in Figure 18 and Figure 19), then it is assumed that the accumulator pressure will only increase at higher ambient temperatures and liquid volumes.

While most accumulators are exposed to atmospheric conditions, the affect of solar radiation can be eliminated by the use of a sun screen or shade. The impact of ambient temperature variations may be reduced if the accumulator is insulated or temperature controlled (i.e., heat traced). The user should verify that the seal and seal support system is suitable for all system pressures by following the calculation sequence illustrated in this tutorial.

The MAWP of Category 2 and 3 installations is significantly higher than Category 1; it is therefore reasonable to expect Category 1 installations may be more vulnerable to expected fluctuations in barrier liquid pressure.

Possible ways to limit the impact of local ambient temperature variations on accumulator pressure include:

- Use of a larger accumulator.
- Use of an engineered auxiliary system design that has an MAWP above standard Category 1, 2, or 3 systems.
- Use of an engineered seal rated for higher pressure than standard Type A, B or C seals.
- Pressure relief valve in the barrier liquid piping.
- Shade the accumulator to eliminate solar radiation effects.
- Limit the impact of the ambient temperature range on the gas inside the accumulator by insulating and/or temperature control (heat tracing for example) of the accumulator.

Three descriptive phrases listed below are used to identify illustrated points in Figures 16, 17, and 18, and are referred to in the example calculations that follow.

- Accumulator minimum barrier pressure—The lowest operating barrier pressure equal to the sum of the maximum seal chamber pressure and a pressure margin, which is recommended to be a minimum of 0.14 MPa (1.4 bar) (20 psi). This establishes Point #1 in the Figures 18 and 19. The value is used as a starting point for the example calculations in this tutorial. The pressure is temperature specific and the accumulator minimum barrier pressure will increase (between Point #1 and Point #7 in Figures 18 and 19) with increasing gas temperature in the bladder.
- Accumulator pressure range—The pressure range between the maximum and minimum barrier pressure and is specific to a temperature value. It is illustrated between Point #1 and Point #5 if a floating pressure alarm is utilized, but will be reduced to the pressure between Point #7 and Point #5 when a fixed pressure alarm strategy is utilized.
- Accumulator working liquid volume—The liquid volume in the accumulator released between the maximum barrier pressure and the alarm pressure. This is dependent on the alarm strategy applied. It is the liquid volume difference between maximum and minimum liquid barrier liquid volumes if a floating pressure alarm strategy is employed, but could be significantly less if a fixed pressure alarm strategy is used (Figure 17). The selection of the accumulator sizes in the draft of API 682 Fourth Edition/ISO 21049 have been made to optimize the working liquid volume to be roughly equal to the working liquid volume for reservoir systems provided with Piping Plan 52 and 53A systems.

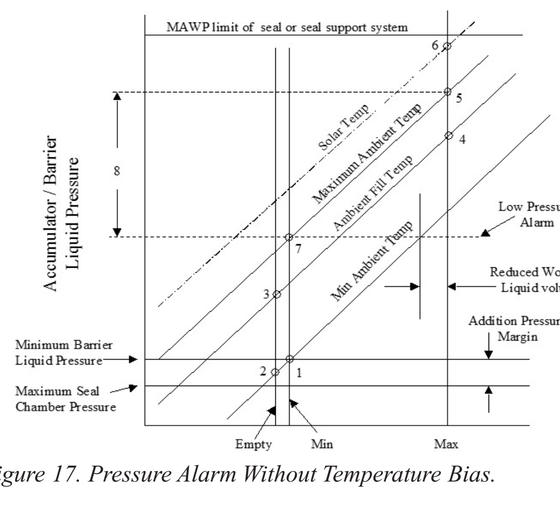

Figure 17. Pressure Alarm Without Temperature Bias.

Alarm Strategy and
Accumulator Working Liquid Volume

The recommended pressure alarm for refilling Piping Plan 53B requires the use of a floating alarm set point (a pressure alarm with a temperature bias). The alarm set point is calculated continuously by the plant's distributed control system (DCS) to actuate when barrier liquid volume reaches minimum liquid volume based on the temperature of the gas in the bladder. As can be seen in Figures 18 and 19, the alarm pressure can vary between Points #1 and #7 at minimum liquid volume. The use of a pressure alarm with a temperature bias also maximizes the accumulator working liquid volume.

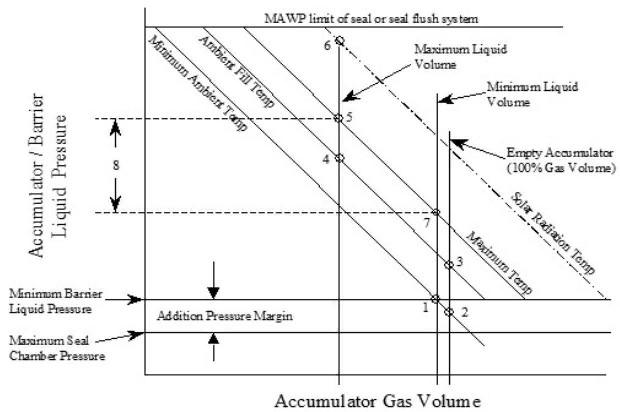

Figure 18. Plan 53B Gas Volume Versus Pressure.

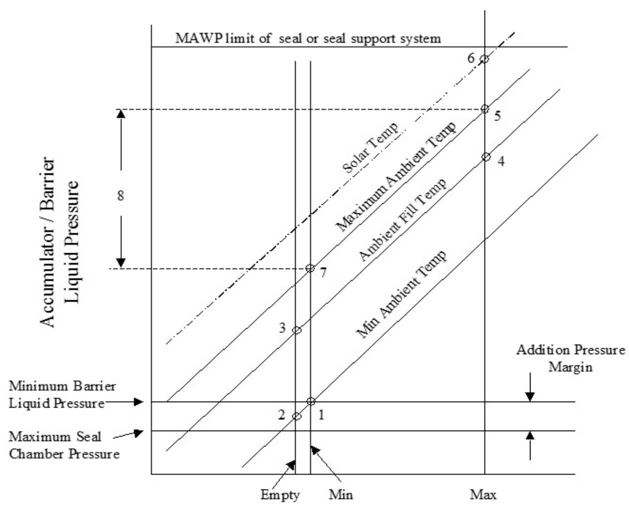

Figure 19. Plan 53B Barrier Liquid Volume Versus Pressure.

A pressure alarm with a temperature bias provides a floating set point that is recommended because it will maximize the working liquid volume at all local ambient temperatures. It is accomplished by the use of a pressure and temperature transmitter in the seal support system. These signals would be integrated into a plant DCS system to provide an accurate temperature adjusted pressure alarm set point. While using the plant DCS system is the least costly approach for installations where a DCS is available, a local programmable logic controller (PLC) or single loop controller could also be used with this same alarm strategy.

Specific DCS input required for a floating alarm algorithm will include the minimum and maximum barrier liquid volume, the accumulator volume, and the accumulator minimum barrier pressure calculated at minimum ambient temperature. The vendor will use these data and the site ambient temperature data to optimize system design, minimize the frequency of refilling, and verify that the system design is suitable for the local installation.

A fixed pressure alarm (without a temperature bias) utilizes a pressure transmitter or pressure switch with a low pressure setting at Point #7. This choice will under most operating conditions result in a significantly reduced accumulator working liquid volume. Figure 17 illustrates the alarm strategy for a pressure alarm without a temperature bias. While this alarm strategy will work, it is operationally more restrictive than a floating pressure alarm.

The accumulator working liquid volume is dependent on many variables, but should be optimized by the vendor to balance the accumulator working pressure range with the performance limits of the seal system, the frequency between refilling and the alarm strategy. The accumulator working liquid volume is typically 15 to 25 percent of the total accumulator volume.

Assuming a sunshade is fitted and the solar temperature need not be considered, the illustrated accumulator working pressure range (#8 in Figures 18 and 19) represents the minimum pressure range, but may rise to the difference in pressure between Point #1 and Point #5 with a maximum ambient temperature change when a floating alarm strategy is utilized.

Fixed and Floating Alarm Strategies

The graphs (Figures 17, 18, and 19) illustrate important calculation points for Piping Plan 53B systems. In addition, Figure 17 shows the impact a single point alarm strategy has on the working liquid volume. When a single alarm strategy is employed, a fixed pressure value at Point #7 is required to provide an alarm corresponding to the minimum liquid volume at maximum ambient temperature. The choice of a lower fixed pressure value may risk the accumulator minimum liquid volume being reached at high ambient temperature without a warning alarm. When the barrier pressure value at maximum ambient temperature (Point #7) is considered at lower ambient (gas bladder) temperatures, the result is a reduced accumulator working barrier liquid volume. The operating pressure range is also reduced, between Point #7 and the maximum barrier pressure.

Unlike a fixed alarm set point, a floating alarm set point (pressure alarm with a temperature bias) will utilize the full potential liquid volume between minimum and maximum in the accumulator. The accumulator pressure range is also maximized, between Point #1 to Point #5 depending on the local ambient temperature change over the barrier pressure drop.

Refer to the calculations that follow for a detailed description of each plotted point. All figures assume solar radiation effects are eliminated by the use of a sunshade above the accumulator.

Figures 17 and 19 show the same basic information, but the focus of Figure 17 is the reduced working liquid volume associated with a fixed alarm strategy. The information presented in Figures 18 and 19 is the same, but presented from two different perspectives. Figure 18 graphs barrier liquid pressure against accumulator gas volume. Figure 19 graphs barrier liquid pressure against barrier liquid volume. The calculations that follow refer to the points identified in these figures.

Piping Plan 53B Calculation Tutorial and Formula

The following discussion refers to the illustrated "numbered" points in Figures 18 and 19. It assumes the accumulator bladder gas temperature corresponds to the local ambient temperature. To simplify the explanation the calculation also assumes the bladder precharge pressure is applied at the same ambient temperature prevailing when the system is initially filled with barrier liquid.

- Point #1—Minimum barrier pressure at minimum barrier liquid volume and minimum ambient temperature—This pressure is the basis for all subsequent calculations and is the sum of the maximum seal chamber pressure and a pressure margin to avoid pressure reversals across the inner seal.

- Point #2—Piping Plan 53B accumulator bladders are precharged with gas (usually nitrogen) when completely empty; Point #2 uses the value of Point #1 to determine the equivalent gas precharge pressure with an empty accumulator (zero liquid volume) if the local ambient temperature is also at a minimum.

Point #2 Pressure = Pressure at Point #1 × (gas volume at minimum liquid volume/total empty accumulator volume)

- Point #3—Calculates the gas precharge pressure based on actual ambient temperature at the time of charging the accumulator bladder. Point #3 uses the value of Point #2, but applies a ratio of temperatures; ambient at the time of filling and minimum ambient temperature.

Point #3 pressure = Pressure at Point #2 × (ambient temp [°C + 273] [or °F + 460] at time of filling/minimum ambient temp [°C + 273] [or °F + 460])

Note: The pressure at Point #3 is the value used to precharge the accumulator. When the gas charge reaches the prescribed pressure, it should be isolated and then the system should be prepared for adding barrier liquid. When the barrier liquid reaches the maximum liquid volume the pressure in the accumulator would reach the pressure at Point #4.

- Point #4—Calculates the maximum barrier accumulator pressure with the maximum barrier liquid volume in the accumulator at the prevailing ambient temperature at the time of barrier liquid filling (assumes the same temperature as that used when precharging the bladder (refer to Point #3)). Point #4 uses the value of Point #3, but applies a ratio of volumes; empty accumulator and the gas volume with barrier liquid at the maximum volume.

Point #4 Pressure = Pressure at Point #3 × (total empty accumulator volume/bladder gas volume at maximum barrier liquid volume)

Note: The bladder gas volume at maximum barrier liquid volume is a result of removing the volume between maximum and minimum barrier liquid volume values plus the minimum liquid volume from the empty accumulator volume. The volume between maximum and minimum barrier liquid volume is normally estimated by the system design engineer and is an iterative value resulting from optimizing a compromise between the maximum barrier pressure and accumulator working liquid volume (an initial value needs to be assumed and subsequently adjusted as appropriate).

- Point #5—Calculates the maximum barrier pressure at maximum barrier liquid volume, but at the maximum ambient temperature. Point #5 uses the value of Point #4, but applies a ratio of temperatures; maximum ambient and ambient temperature at the time of precharging the bladder.

Point #5 Pressure = Pressure at Point #4 × (maximum ambient temp [°C + 273] [or °F + 460]/ambient temp [°C + 273] [or °F + 460] at time of filling)

Note: It is important the maximum barrier pressure at maximum ambient temperature does not exceed the dynamic sealing pressure rating (DSPR) of the seal or the MAWP of the system. The system designer, when considering the level of accumulator working volume may use the criteria below to ensure these limits are not exceeded.

Maximum barrier liquid volume must be <=

$$V_{tot} - ((V_{tot} - V_{min}) \times (T_{max}/T_{min}) \times ((P_{cmax} + 0.14) / (\text{Minimum of the DSPR or MAWP}))) \tag{27}$$

where:
V_{tot} = Total empty accumulator volume
V_{min} = Minimum liquid volume
T_{max} = Maximum absolute ambient temperature (K or °R)
T_{min} = Minimum absolute ambient temperature (K or °R)
P_{cmax} = Maximum absolute seal chamber pressure (MPa)
DSPR or MAWP in absolute pressure (MPa)

- Point #6—Calculates the barrier pressure at maximum barrier liquid volume, but at the solar radiation temperature. Point #6 uses the value of Point #5, but applies a ratio of temperatures; solar radiation and maximum ambient temperature.

Point #6 Pressure = Pressure at Point #5 × (solar radiation temp [°C + 273] [or °F + 460]/maximum ambient temp [°C + 273] [or °F + 460]).

Note: If the accumulator is shaded, insulated, or other means are used to limit the bladder gas temperature fluctuations this calculation step is not needed.

- Point #7—This represents an alarm pressure set point. It corresponds to the barrier pressure at minimum liquid volume, but at maximum ambient temperature. Point #7 uses the value of Point #1, but applies a ratio of temperatures; maximum ambient and minimum ambient temperature.

Alarm pressure at Point #7 = Pressure at Point #1 × (maximum ambient temp [°C + 273] [or °F + 460]/minimum ambient temp [°C + 273] [or °F + 460])

Note 1: If a fixed alarm strategy is chosen, the value calculated for Point #7 will be the recommended alarm pressure. If a floating alarm strategy is chosen then the value calculated for Point #7 represents the highest alarm pressure based on a calculated algorithm, but the alarm pressure will vary between Point #1 and Point #7 depending on the bladder gas temperature.

Note 2: It is important with a single alarm strategy that there is adequate accumulator working liquid volume to satisfy the refill frequency of 28 days or longer and the system designer may use the criteria below, combined with the criteria described in the Note specific to Point #5 to assist in selecting the performance limits of the system.

Maximum barrier liquid volume (with a fixed alarm strategy) must be >=

$$\text{Vtot} - ((\text{Vtot} - \text{Vmin}) \times (\text{Tmin}/\text{Tmax})) + (\text{minimum level of accumulator working liquid volume to achieve 28 days of operation}) \tag{28}$$

where:
V_{tot} = total empty accumulator volume
V_{min} = minimum liquid volume
T_{max} = Maximum absolute ambient temperature (K or °R)
T_{min} = Minimum absolute ambient temperature (K or °R)

Piping Plan 53B Example Calculation

The example calculation is for a Piping Plan 53B application showing the effects of local ambient temperature range and solar radiation. It is assumed that the auxiliary seal support system is designed for a MAWP gauge pressure of 4 MPa (40 bar) (600 psi) typical of Category 2 and 3 seal systems and the dynamic sealing pressure rating exceeds this limit.

- Assumptions include site conditions:
  40°C maximum site temperature
  −10°C minimum site temperature
  20°C ambient temperature at time of precharging and filling
  60°C maximum solar radiation temperature

- Seal system assumptions:
  20 liter total accumulator volume (no barrier liquid)
  0.2 liter minimum barrier liquid volume
  3 liter maximum barrier liquid volume (this includes the minimum barrier liquid volume)
  1.5 liter minimum acceptable accumulator working liquid volume to achieve 28 day operation
  2 MPa (20 bar) maximum seal chamber pressure (gauge pressure; 2.1 MPa absolute pressure)
  0.14 MPa (1.4 bar) pressure margin above maximum seal chamber pressure

- 1. Calculate the minimum barrier pressure at minimum liquid volume and minimum ambient temperature.
  Point #1 = 2.1 + 0.14 = 2.24 MPa (absolute pressure)
  (a gauge pressure of 2.14) (21.4 bar) (310 psig)

Note 1: The following calculations will be utilizing absolute pressure values. The user should recognize that precharging and filling maintenance activities will normally utilize local gauge readings.

Note 2: If a fixed alarm strategy is chosen, this value calculated for Point #1 is only used as a basis for other calculations and is not an alarm pressure. If a floating alarm strategy is chosen then the value calculated for Point #1 represents the lowest alarm pressure based on a calculated algorithm, but the alarm pressure may vary between Point #1 and Point #7 depending on the bladder gas temperature.

- 2. Calculate the corresponding accumulator bladder pressure assuming an empty accumulator (100 percent gas volume) and minimum ambient temperature.

Point #2 = 2.24 × ((20 – 0.2)/20) = 2.218 MPa (absolute pressure)

- 3. Calculate the corresponding accumulator bladder pressure assuming the ambient temperature at the time of filling and an empty accumulator (100 percent gas volume). This is the field precharge gas (usually nitrogen) pressure.

Point #3 = 2.218 × (20°C + 273)/(−10 + 273) = 2.471 MPa
(a gauge pressure of 2.371 MPa) (23.71 bar) (344 psig)

- 4. Calculate the corresponding maximum barrier pressure after the barrier liquid is added to the gas charged accumulator assuming both are completed at the same ambient temperature.

Point #4 = 2.471 × (20/(20 – 3)) = 2.907 MPa
(a gauge pressure of 2.807 MPa) (28.07 bar) (407 psig)

- 5. Calculate the corresponding maximum barrier pressure at the maximum ambient temperature and at maximum barrier liquid volume.

Point #5 = 2.907 × (40°C + 273)/(20 + 273) = 3.105 MPa (absolute pressure)

Note: Check the flexibility for increasing the maximum barrier liquid volume.

Maximum barrier liquid volume <= 20 – ((20 – 0.2) × ((40 + 273) / (10 + 273)) × ((2.1 + 0.14) / 4.1))
<= 7.13 liters

The selected 3 liter maximum barrier liquid volume successfully meets the criteria and is in excess of the 1.5 liter minimum acceptable accumulator working liquid volume.

- 6. Calculate the corresponding barrier pressure at maximum barrier liquid volume, but at the solar radiation temperature.

Point #6 = 3.105 × (60°C + 273)/(40 + 273) = 3.303 MPa (absolute pressure)

- 7. Calculate the barrier alarm pressure setting corresponding to the pressure at minimum liquid volume and maximum ambient temperature.

Point #7 = 2.24 × (40 + 273)/(−10 + 273) = 2.666 MPa (absolute pressure)
(a gauge pressure of 2.566 MPa) (25.66 bar) (372 psig)

Note 1: If a fixed alarm strategy is chosen, the value calculated for Point #7 will be the recommended alarm pressure. If a floating alarm strategy is chosen then the value calculated for Point #7 represents the highest alarm pressure based on a calculated algorithm, but the alarm pressure may vary between Point #1 and Point #7 depending on the bladder gas temperature.

Note 2: Check the accumulator working liquid volume is suitable for a single alarm strategy.

Maximum barrier liquid volume >= 20 – ((20 – 0.2) × (10 + 273) / (40 + 273)) + 1.5
>= 4.86 liters

The selected 3 liter maximum barrier liquid volume does not meet the criteria. If a single alarm strategy is required the maximum liquid volume needs to be between 4.86 and 7.13 liters. This will change the calculation on Point #4 and Point #5 above and raise the maximum barrier pressures.