2.2.3 CENTRIFUGAL PUMP MECHANICAL SEALS

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Mechanical seals have been used for many years to seal any number of liquids at various speeds, pressures, and temperatures. Today plant operators are benefiting from improved seal technologies driven by the U.S. Clean Air Act of 1990, and the American Petroleum Institute (API) Standard 682. These new seal technologies are based on advanced computer programs used to optimize seal designs, which are then verified through performance testing at simulated refinery conditions required by the API. The results to date indicate not only an improvement in emissions control, but also a major increase in equipment reliability.

CLASSES OF SEAL TECHNOLOGY

Emerging seal technologies are providing clear choices for sealing. Various plant services require the application of these new technologies for emissions control, safety, and reliability. Sealing systems are now available that are based on the preferred method of lubrication to be used. These classes of seals are as follows:

1. Contacting liquid lubricated seals:
   - Normally, a single seal arrangement is cooled and lubricated by the liquid being sealed. This is the most cost-effective seal installation available to the industry.
   - Dual seals are arranged to contain a pressurized or non-pressurized barrier or buffer liquid. Normally, this arrangement will be used on applications where the liquid being sealed is not a good lubricating fluid for a seal and for emissions containment. These arrangements require a lubrication system for the circulation of barrier or buffer liquids.

2. Non-contacting gas lubricated seals:
   - Dual non-contacting, gas-lubricated seals are pressurized with an inert gas such as nitrogen.
   - Dual non-contacting, gas-lubricated seals are used in a tandem arrangement and pressurized by the process liquid being sealed, which is allowed to flash to a gas at the seal. A tandem seal arrangement is used on those liquids that represent a danger to the plant environment. For non-hazardous liquids, a single seal can be used.

Each of these solutions has been used on difficult applications to increase the *mean time between maintenance* (MTBM).

SEAL DESIGN

Advancing the state-of-the-art sealing systems are new suites of computer programs such as C'Stedy[SM(1)] used to analyze the performance of both contacting and non-contacting seal designs during steady-state and transient conditions. This type of finite analysis considers all of the operating conditions, the fluid sealed, the materials of construction, and seal geometry. The outputs from the program are seal distortion, temperature distribution, friction power, actual *PV* (pressure × velocity), leakage, the percentage of face in liquid or vapor, and fluid film stability (see Figure 1). This type of analysis requires accurate fluid and material properties. The results from the program can predict the success or failure of a given installation.

For example, a mixture of liquid hydrocarbon made up of ethane, propane, butane, and hexane has to be sealed. This is a new application. The operating condition is 1,300 psig at 70°F. The shaft speed is 3,600 rpm. To determine the performance of this seal prior to installation, an analysis must be made. The results of this study indicate stable operations for a contacting seal, as shown in Figure 2. This study was used to predict seal performance. Actual field results from this difficult service were excellent at startup and during equipment operation.

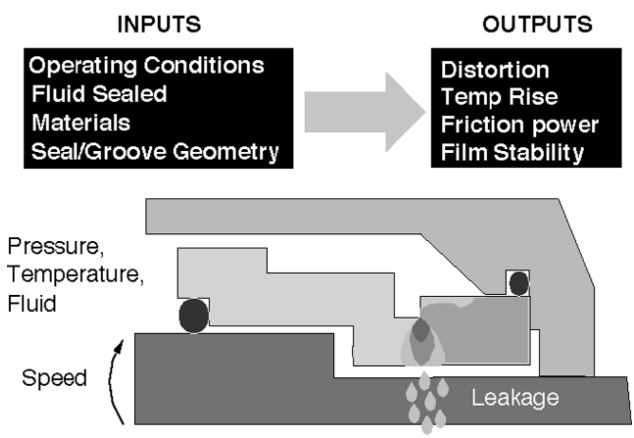


FIGURE 1 C'Stedy[SM] fluid film model for a mechanical seal (John Crane Inc.)

[1]Service Mark of John Crane Inc.

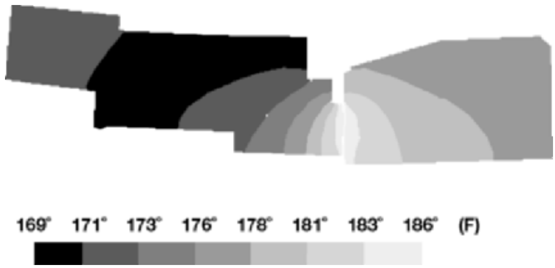


FIGURE 2 C'Stedy output for a successful seal on high pressure light hydrocarbon service (John Crane Inc.)

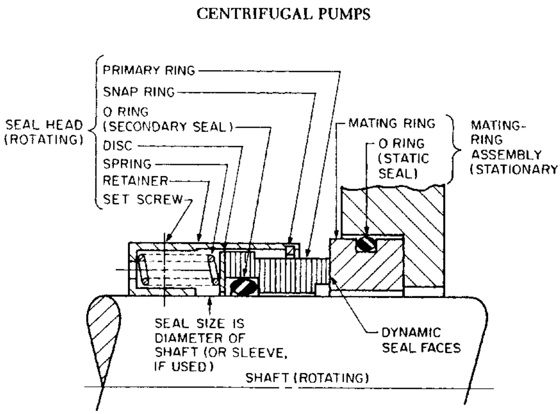


FIGURE 3 The basic components of a mechanical seal

These state-of-the-art computer tools not only predict performance, but they also can be used to determine any short seal life. By using a series of calculations per second, this type of analysis can be used to create an animation that will visibly show changes to a seal at startup and during fluctuations in operating conditions. A seal can be examined for stable and unstable operations. This is a useful analysis tool for critical applications.

DESIGN FUNDAMENTALS

Contacting Liquid Lubricated Seals The basic components of a mechanical seal are the primary and mating rings. Together they form the dynamic sealing surfaces, which are perpendicular to the shaft. The primary ring is part of the seal head assembly, while the mating ring and static seal form a second assembly, making a complete installation for a pump. These basic seal parts are shown in Figure 3. For slower and normal shaft speeds, the seal head assembly will rotate with the shaft, while on high shaft speeds, the seal head assembly will be held stationary to the equipment.

The only difference between contacting and non-contacting seal technologies is found in the design of the seal faces. Each system has the same type and number of parts. Each has its own area of application for maximum sealing efficiency. Non-contacting seal technology will be discussed later.

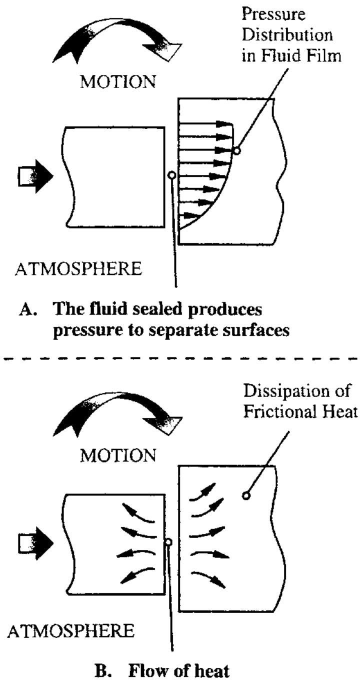


FIGURE 4 Processes involved at contacting seal faces

In a contacting seal, as the shaft begins to rotate, a small fluid film develops, along with frictional heat from the surfaces in sliding contact. These processes occurring at the seal faces are shown in Figure 4. The amount of heat developed at the seal faces must be removed to prevent the liquid being sealed from flashing or beginning to carbonize. Seal heat can be removed with a seal flush located at the seal faces. To analyze the performance of a seal and determine amount of cooling, the following calculations can be made.

Seal Balance The greatest concern to the seal user is the dynamic contact between the mating seal surfaces. The performance of this contact determines the effectiveness of the seal. If the load at the seal faces is too high, the liquid at the seal faces will vaporize or carbonize and the seal faces can wear out. Damage to the seal faces can occur due to unstable conditions. A high wear rate from solid contact and leakage can occur if the bearing limits of the materials are exceeded. Seal balancing is a feature that is used to avoid these conditions and provide for a more efficient installation.

The pressure in any seal chamber acts equally in all directions and forces the primary ring against the mating ring. Pressure acts only on the annular area a_c (see Figure 5a), so that the force in pounds (Newtons) on the seal face is as follows:

$$F_c = pa_c$$

where p = seal chamber pressure, lb/in^2 (N/m^2) and
a_c = hydraulic closing area, in^2 (m^2)

The pressure in lb/in^2 (N/m^2) between the primary ring and mating ring is

$$P_f' = \frac{F_c}{a_o} = \frac{pa_c}{a_o}$$

where a_o = hydraulic opening area (seal face area), in²(m²).

To relieve the pressure at the seal faces, the relationship between the opening and closing forces can be controlled. If a_o is held constant and a_c is decreased by a shoulder on a sleeve or seal hardware, the seal face pressure can be lowered (see Figure 5b). This is called *seal balancing*. A seal without a shoulder in the design is referred to as an *unbalanced seal*. A balanced seal is designed to operate with a shoulder.

The ratio of the hydraulic closing area to the face area is defined as seal balance b:

$$b = \frac{a_c}{a_o}$$

Seals can be balanced for pressure at the outside diameter of the seal faces, as shown in Figure 5b. This is typical for a seal mounted inside the seal chamber. Seals installed outside the seal chamber can be balanced for pressure at the inside diameter of the seal faces. In special cases, seals can be double-balanced for pressure at both the outside and inside diameters of the seal. Seal balances can range from 0.65 to 1.35, depending on operating conditions.

Face Pressure As relative motion takes place between the seal planes, a liquid film develops. The generation of this film is believed to be the result of surface waviness in the individual sealing planes. Pressure and thermal distortion, as well as anti-rotation devices such as drive pins, keys, or dents used in the seal design, have an influence on surface waviness and on how the film develops between the sliding surfaces. Hydraulic pressure develops in the seal face, which tends to separate the sealing planes. The pressure distribution, referred to as a pressure wedge, shown in Figure 6, can be considered as linear, concave, or convex. The actual face pressure p_f in lb/in² (N/m²) is the sum of the hydraulic pressure p_h and the spring pressure P_{sp} designed into the mechanical seal. The face pressure P_f is a further refinement of P_f', which does not take into account the liquid film pressure or the mechanical load of the seal:

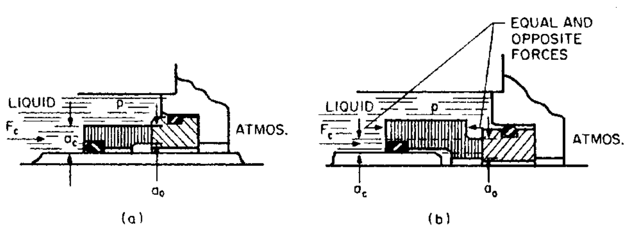


FIGURE 5 Hydraulic pressure acting on the primary ring: a) unbalanced, b) balanced

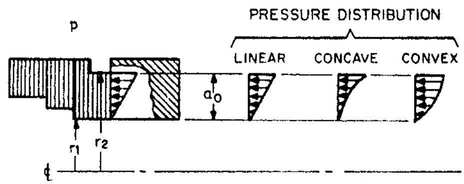


FIGURE 6 The pressure distribution can be considered linear, concave, or convex.

$$P_f = P_h + P_{sp}$$

where $P_h = \Delta p(b—k)$, lb/in²(N/m²) and
Δp = pressure differential across seal face, lb/in² (N/m²)
k = pressure gradient factor
b = seal balance

The mechanical pressure for a seal design is

$$P_{\mathrm{sp}} = \frac{F_{\mathrm{sp}}}{a_o}, \ \mathrm{lb/in^2\ (N/m^2)}$$

where F_{sp} = seal spring load, lb (N) and
a_o = seal face area, in² (m²)

Then the actual face pressure can be expressed as

$$P_f = \Delta p(b - k) + P_{\mathrm{sp}}$$

The actual face pressure is used in the estimate of the operating pressure and velocity for a given seal installation.

Pressure-Velocity As the sealing planes move relative to each other, they are affected by the actual face pressure and rotational speed. The product of the two, pressure times velocity, is referred to as *PV* and is defined as the power N_f per unit area with a coefficient of friction of unity:

$$PV = \frac{N_f}{a_o}$$

For seals, the equation for *PV* can be written as follows:

$$PV = P_f V_m = [\Delta p(b - k) + P_{\mathrm{sp}}]V_m$$

where V_m = velocity at the mean face diameter dm, ft/min (m/s).

The *PV* for a given seal installation can be compared with values developed by seal manufacturers as a measure of adhesive wear.

Power Consumption The *PV* value also enables the seal user to estimate the power loss at the seal with the following equation:

$$N_f = (PV)fa_o, \ \mathrm{ft \cdot lb/min\ (N \cdot m/s)}$$

where f is the coefficient of friction.

As a rule of thumb, the power to start a seal is generally five times the running value. The coefficients of friction for various common seal face materials are given in Table 1. These coefficients were developed with water as a lubricant at an operating *PV* value of 100,000 lb/in² · ft/min (35.03 bar · m/s). The coefficient of friction is a function of the tribological properties of the mating pairs of seal face materials and the fluid being sealed. Values in oil would be slightly higher because of the viscous shear of the fluid film at the seal faces. For a double or tandem seal, the barrier/buffer oil should have a low viscosity and be a good lubricant. The values given are suitable for estimating the power loss in a seal.

For example, let's say we have a pump having a 2-in (50.8-mm) diameter sleeve at the seal chamber is fitted with a balanced seal of this size and mean diameter. The seal operates in water at 300 lb/in² (20.68 bar), 3,600 rpm, and ambient temperatures. The materials of construction are carbon and tungsten carbide. Determine the *PV* value and power loss of the seal, given the following:

TABLE 1 Coefficient of friction for various seal face materials (John Crane Inc.)

Sliding Materials		
Rotating	Stationary	Coefficient of friction
Carbon-graphite (resin filled)	Cast iron	0.07
	Ceramic	0.07
	Tungsten carbide	0.07
	Silicon carbide	0.02
	Silicon carbide converted carbon	0.015
Silicon carbide	Tungsten carbide	0.05
Silicon carbide	Silicon carbide converted carbon	0.04
	Silicon carbide converted carbon	0.05
	Silicon carbide	0.05
	Tungsten carbide	0.01

$$\Delta p = 300 \text{ lb/in}^2 \text{ (20.68 bar)}$$

$$b = 0.75$$
$$k = 0.5$$
$$d_m = 2 \text{ in (50.8 min)}$$
$$P_{sp} = 25 \text{ lb/in}^2\text{(1.72 bar)}$$
$$V_m = \frac{\pi}{12} \times 2 \times 3600 = 1885 \text{ ft/min} \left(\frac{\pi \times 50.8 \times 3600}{1000 \times 60} = 9.57 \text{ m/s} \right)$$
$$a_o = 0.4 \text{ in}^2 \text{ (0.000258 m}^2\text{)}$$
$$f = 0.07 \text{ (Table 1)}$$

In USCS units:

$$PV = [300(0.75 - 0.5) + 25](1885) = 188{,}400 \text{ lb/in}^2 \cdot \text{ft/min}$$
$$N_f = (188{,}400)(0.07)(0.4) = 5275 \text{ ft} \cdot \text{lb/min} = 0.16 \text{ hp}$$

In SI units:

$$PV = [20{,}68(0.75 - 0.5) + 1.72](9.57) = 66 \text{ bar} \cdot \text{m/s} = 66 \times 10^5 \text{ N/m}^2 \cdot \text{m/s}$$
$$N_f = (66 \times 10^5)(0.07)(2.58 \times 10^{-4}) = 119 \text{ N} \cdot \text{m/s} = 119 \text{ W}$$

Temperature Control Controlling the temperature at the seal faces is desirable because wear is a direct function of temperature. Heat at the seal faces also causes thermal distortion, which will contribute to increased seal leakage. Many applications require some type of cooling.

The temperature of the sealing surfaces is a function of the heat generated by the seal, plus the heat gained or lost to the pumpage. The heat generated at the faces from sliding contact is the mechanical power consumption of the seal being transferred into heat. Therefore,

$$Q_s = C_a N_f = C_1(PVfa_o)$$

where Q_s = heat input from the seal, Btu/h(W) and
C_1 = 0.077 for USCS units and 1 for SI units

If the heat is removed at the same rate it is produced, the temperature will not increase. If the amount of heat removed is less than that generated, the seal face temperature will

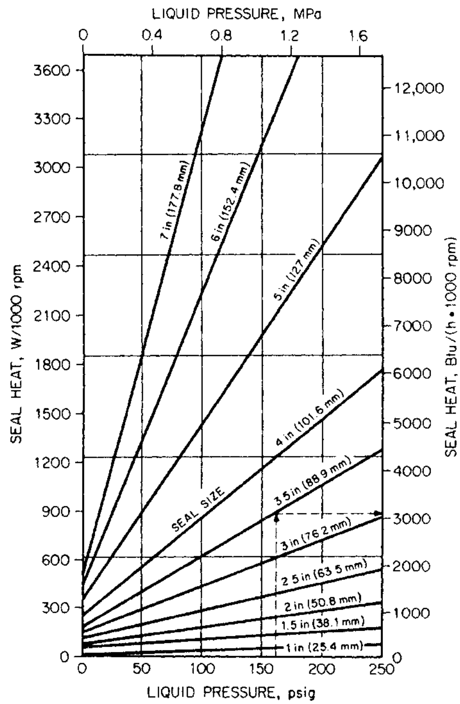


FIGURE 7 Unbalanced seal heat generation (John Crane Inc.)

increase to a point where seal face damage will occur. The estimated values for heat input are given in Figures 7 and 8.

Heat removal from a single seal is accomplished by a seal flush. The seal flush is usually a bypass from the discharge line on the pump or an injection from an external source. The flow rate for cooling can be found by calculating the following:

$$\text{gpm (m}^3\text{/h)} = \frac{Q_s}{C_2(\text{sp. ht.})(\text{sp. gr.})\Delta T}$$

where Q_s = seal heat, Btu/h(W)

C_2 = 500 in USCS units and 1,000 in SI units

sp. ht. = specific heat of coolant, Btu/lb. · °F (J/kg · K)

sp. gr. = specific gravity of coolant

ΔT = temperature rise, °F (K)

When handling liquids at elevated temperatures, the heat input from the process must be considered in the calculation of coolant flow. Thus,

$$Q_{net} = Q_p + Q_s$$

The heat load Q_p from the process can be determined from Figure 9.

As an example, let's determine the net heat input for a 4-in (102-mm) diameter balanced seal in water at 1,800 rpm. The pressure and temperature are 400 lb/in^2 (27.6 bar) and 170°F (76.7°C). From Figure 8, we have the following:

In USCS units:

$$Q_s = (3500 \text{ Btu/h/1000 rpm}) \times (1800) = 6300 \text{ Btu/h}$$

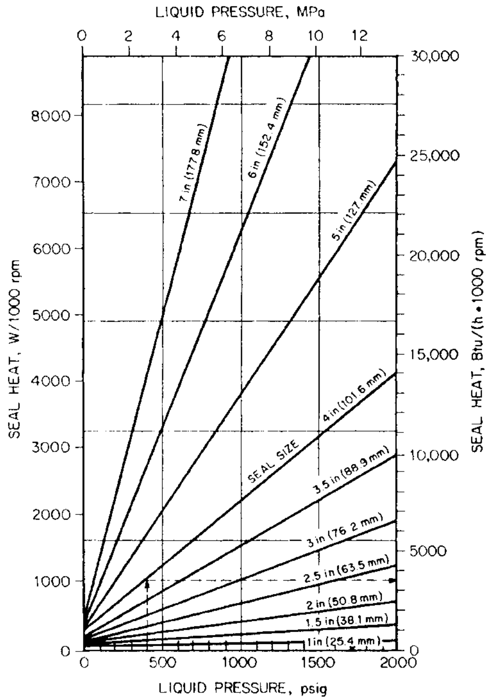


FIGURE 8 Balanced seal heat generation (John Crane Inc.)

In SI units:

$$Q_s = (1025 \text{ W}/1000 \text{ rpm}) \times (1800) = 1845 \text{ W}$$

From Figure 9, assuming that the seal chamber will be cooled to 70°F (21°C) and that the temperature difference between the seal chamber and pumpage is 100°F (37.8°C), we have the following:

In USCS units: $Q_p = 255$ Btu/h

In USCS units: $Q_p = 75$ W

$\therefore$ $Q_{net} = 6555$ Btu/h (1920 W)

The total heat input can be used to estimate the required flow to the seal. When multiple seals are used in a pump seal chamber, the heat load from each seal must be considered as well as any heat soak from the process.

Different methods are used to supply cool liquid to the seal chamber (see Figure 10). When the liquid is clean, an internal flush connection at (A) can be used to cool the seal. When the liquid is dirty, an external flush at (B) can be used. This will allow the flush, a bypass from the discharge line, to pass through a filter or centrifugal separator. The seal faces will be flushed with clean, cool liquid. Increased pressure from the flush provides positive circulation and prevents flashing at the seal faces caused by the heat generation.

When a pump handles liquids near their boiling point, additional cooling of the seal chamber is required. A typical arrangement to accomplish this is shown in Figure 11. This seal is equipped with a pumping ring and a heat exchanger. The pumping ring acts as a miniature pump, causing the liquid to flow through the outlet piping at the top of the seal

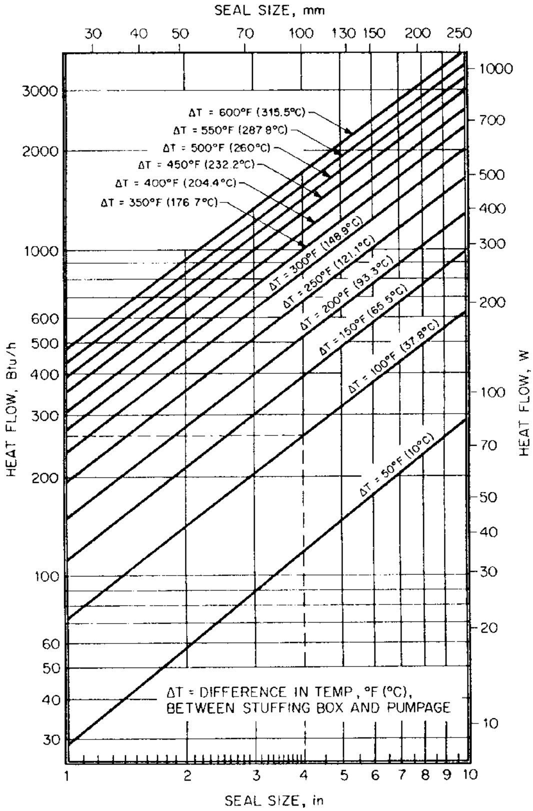


FIGURE 9 Heat soak from process when water is used for lubrication (John Crane Inc.)

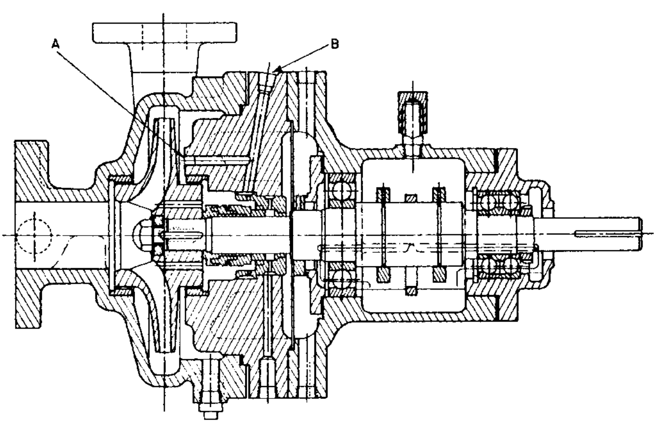


FIGURE 10 Cooling circulation to mechanical seal: (A) internal circulation plug port, (B) external circulation plug port

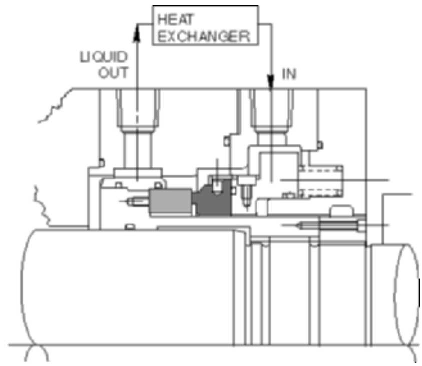


FIGURE 11 High performance boiler feed pump seal with external cooling

chamber. The liquid passes through the heat exchanger and returns directly to the faces at the bottom inlet in the end plate. As the liquid is circulated, heat is removed from the seal and seal chamber. A closed loop system is commonly used on hot water pumps. This method is extremely efficient since the coolant is circulated only in the seal chamber and does not reduce the temperature of the liquid in the pump.

It should be noted that during periods of shutdown different wear problems might exist because the seal faces may be too cold. The product being pumped may be a solution that can crystallize or solidify at ambient temperatures. For these applications, the seal faces may have to be preheated before starting to avoid damage to the seal.

Leakage Leakage is affected by the parallelism of the sealing planes, angular misalignment, coning (negative face rotation), thermal distortion (positive face rotation), shaft runout, axial vibration, and fluctuating pressure. For parallel faces only, which take into account seal geometry only, the theoretical leakage in cubic centimeters per hour can be estimated from the following:

$$Q_3 = -C_3 \times h^3(P_2 - P_1)/u \; ln \; (R_2/R_1)$$

where C_3 = 2.13×10^{10} in USCS units and 1.88×10^9 in SI units
h = face gap, in (m)
P_2 = pressure at face ID, lb/in^2 (N/m^2)
P_1 = pressure at face OD, lb/in^2 (N/m^2)
u = dynamic viscosity, C_p (N · s/m^2)
R_2 = outer face radius, in (m)
R_1 = inner face radius, in (m)

Negative leakage indicates flow from the face's outer diameter to the inner diameter. The effect of centrifugal force from one of the rotating sealing planes is very small and can be neglected in normal pump applications. The gap between the seal face is a function of the materials of construction, flatness, and the liquid being sealed. The face gap can range from 20×10^{-6} to 50×10^{-6} in (0.508×10^{-6} to 1.27×10^{-6} m).

Contacting Seal Operating Envelope Every seal has an operating envelope. The basic envelope for a contacting seal is shown in Figure 12. The upper limits are defined by wear of the seal faces, usually defined by a pressure-velocity limit. The fluid being sealed should be cooled so the liquid at the seal faces does not flash. Operating within the envelope will result in excellent seal performance.

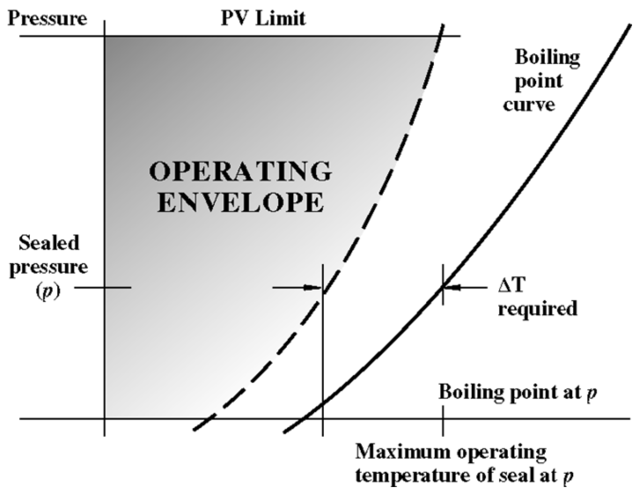


FIGURE 12 Operating envelope for a contacting seal

CLASSIFICATION OF SEALS BY ARRANGEMENT

Sealing arrangements can be classified into two groups:

1. Single seal installations
   a. Internally mounted
   b. Externally mounted
2. Multiple seal installations
   a. Double seals
   b. Tandem seals

Single seals are used in most applications. This is the simplest seal arrangement with the least number of parts. An installation can be referred to as *inside-mounted* or *outside-mounted*, depending on whether the seal is mounted inside or outside the seal chamber (see Figure 13). The most common installation is an inside-mounted seal. Here the liquid under pressure acts with the spring load to keep the seal faces in contact.

Outside-mounted seals are considered to be used for low-pressure applications since both seal faces, the primary ring and mating ring, are put in tension. This limits the pressure capability of the seal. An external seal installation is used to minimize corrosion that might occur if the metal parts of the seal were directly exposed to the liquid being sealed.

Multiple seals are used in applications requiring

- A neutral liquid for lubrication
- Improved corrosion resistance
- A buffered area for plant safety

Double seals consist of two single seals back to back, with the primary rings facing in opposite directions in the seal chamber. The neutral liquid, at a pressure higher than that of the liquid being pumped, lubricates the seal faces (see Figure 14). The inboard seal keeps the liquid being pumped from entering the seal chamber. Both inboard and outboard seals prevent the loss of neutral lubricating liquid.

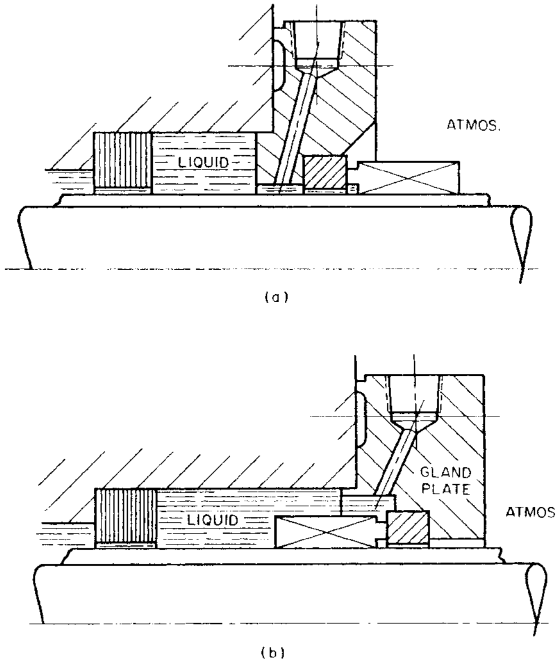


FIGURE 13 Single seal installations: a) outside mounted, b) inside mounted

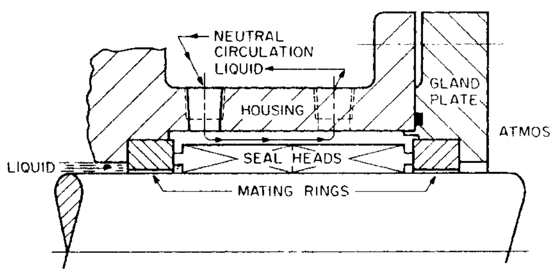


FIGURE 14 Double Seals

Double seals can be used in an opposed arrangement. Two seals are mounted face to face, with the primary sealing rings rotating on a common mating ring (see Figure 15). In this case, the neutral liquid is circulated between the seals at a pressure lower than that of the process fluid. This pressure is limited since the outboard seal faces are in tension. The inboard seal is similar to a single inside-mounted seal and carries the full differential pressure of the seal chamber to the neutral liquid. The outboard seal carries only the pressure of the neutral liquid to the atmosphere. The purpose of this arrangement is to fit a seal installation having a shorter axial length than is possible with back-to-back double seals and still form a buffered area for plant safety.

Tandem seals are arranged with two single seals mounted in the same direction (see Figure 16). The outboard seal and neutral liquid create a buffer zone between the liquid being pumped and the atmosphere. Normally, the pressure differential from the liquid

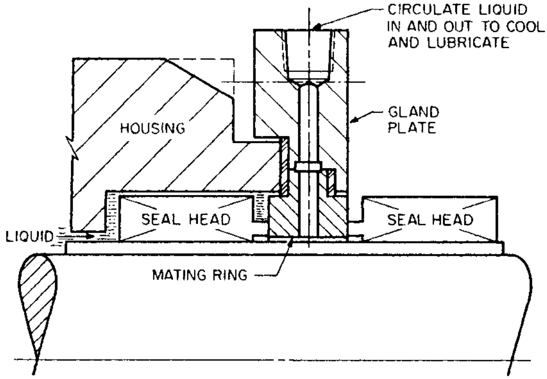


FIGURE 15 Opposed double seals

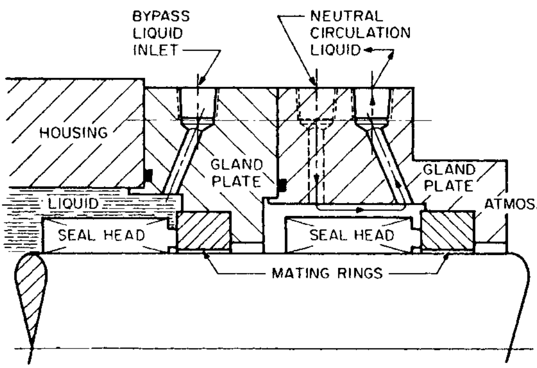


FIGURE 16 Tandem seals

being sealed and atmosphere is taken across the inboard seal, while the neutral lubricating liquid is at atmosphere pressure. This arrangement can also be used as a method to break down the pressure on high-pressure applications. For example, the pressure difference across each seal can be half the fluid pressure being sealed. The liquid in the outboard seal chamber may be circulated to remove seal heat. Tandem seals are used on toxic or flammable liquids, which require a buffered or safety zone.

Package or cartridge seals are an extension of other seal arrangements. A package seal requires no special measurements prior to seal installation. For a single seal, the seal package consists of the gland plate, sleeve, and drive collar (see Figure 17). A spacer is provided on most package seals to properly set the seal faces. The spacer is removed after the drive collar has been locked to the shaft and the gland plate bolted to the pump.

CLASSIFICATION OF SEALS BY DESIGN

There are four seal classification groups:

- Unbalanced or balanced
- Rotating or stationary seal head

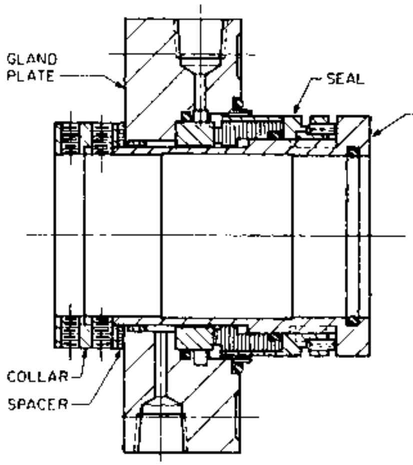


FIGURE 17 A single package (cartridge) seal assembly (John Crane Inc.)

- Single-spring or multiple-spring construction
- Pusher or nonpusher secondary seal design

The selection of an unbalanced or balanced seal is determined by the pressure in the seal chamber. Balance is a way of controlling the contact pressure between the seal faces and power generated by the seal. When the percentage of balance *b* (the ratio of hydraulic closing area to seal face area) is 100 percent or greater, the seal is referred to as unbalanced. When the percentage of balance for a seal is less than 100 (1.0), the seal is balanced. Figure 18 illustrates common unbalanced and balanced seals.

The selection of a rotating or stationary seal is determined by the speed of the pump shaft. A seal that rotates with the shaft is a rotating seal assembly. Typical rotating seals are shown in Figures 17, 21 and 22. When the mating ring rotates with the shaft, the seal is stationary (see Figure 19). Rotating seal heads are common in the industry for normal pump shaft speeds. As a rule of thumb, when the shaft speed exceeds 5,000 ft/min (25.4 m/s), stationary seals are required. Higher speed applications require a rotating mating ring to keep unbalanced forces, which may result in seal vibration, to a minimum. A stationary seal should be considered for all split case pumps. This will eliminate seal problems that occur when the top and bottom halves of the pump casing do not line up. The pressure in the pump can cause a misalignment of these parts that creates an out-of-square condition at the seal faces.

The selection of a single-spring or multiple-spring seal head construction is determined by the space limits and the liquid sealed. Single-spring seals are most often used with bellows seals to load the seal faces (see Figure 20a). The advantage of this type of construction is that the openness of design makes the spring a nonclogging component of the seal assembly. The coils are made of a large diameter spring wire and therefore can withstand a great deal of corrosion.

Multiple-spring seals require a shorter axial space. Face loading is accomplished by a combination of springs placed about the circumference of the shaft (refer to Figure 1 and see Figure 20b). Most multiple-spring designs are used with assemblies having O-rings or wedges as secondary seals.

Pusher-type seals are defined as seal assemblies in which the secondary seal is moved along the shaft by the mechanical load of the seal and the hydraulic pressure in the seal chamber. The designation applies to seals that use an O-ring, wedge, or V-ring. A typical construction is illustrated in Figure 21.

The primary ring, with a hardened metal surface, rotates with the shaft and is held against the stationary ring by the compression ring through loading of the O-ring. The

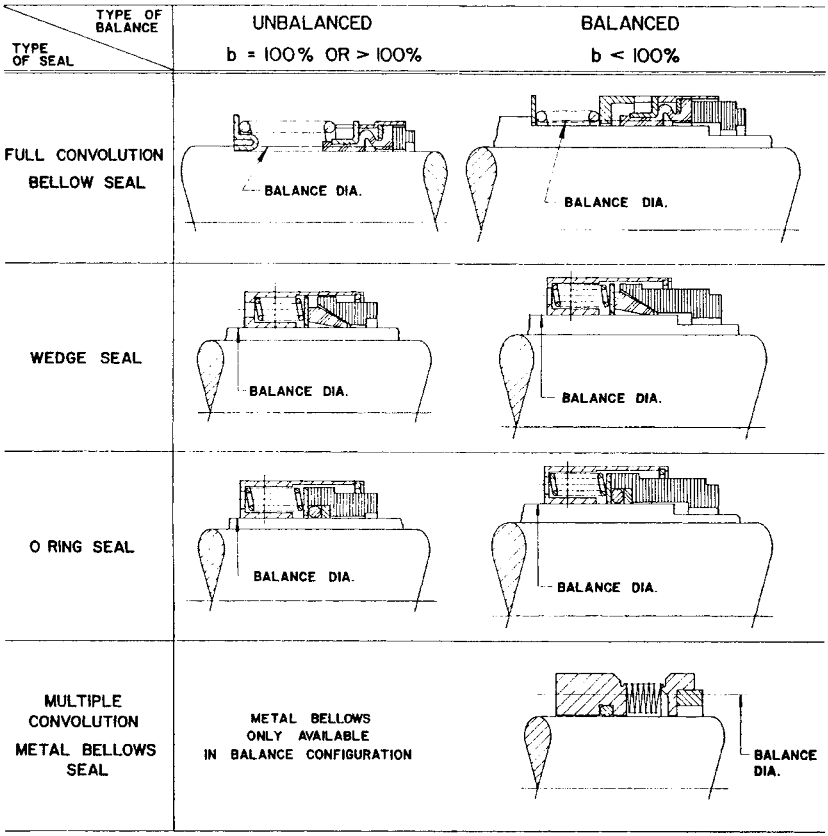


FIGURE 18 Common unbalanced and balanced seals

compression ring supports a nest of springs that is connected at the opposite end by a collar, which is fixed to the shaft. The primary ring is flexibly mounted to take up any shaft deflection or equipment vibration. The collar is fixed to the shaft by setscrews.

Another pusher-type seal is illustrated in Figure 22. When elastomers cannot be used in the product, a wedge made of TFE must be considered. A metal retainer locked to the shaft by (A) provides a positive drive through the shaft and to the primary ring (F) through drive dents (D), which fit corresponding grooves. The seal between the primary ring and shaft or sleeve is made by a wedge (E), which is preloaded by multiple springs (B). The spring load is distributed uniformly by a metal disc (C). The primary ring (G) contacts the mating ring (H) to form the dynamic seal.

Pusher seals also come in split designs. Illustrated in Figure 23 is a split seal design for an ANSI pump. This is a fully split seal design with all of the basic parts fitted outside the seal chamber. The gland plate is fully split and provides easy access to other seal components. A finger spring located on the atmospheric side of the seal provides an axial load and drive to the stationary primary ring. Since it is located on the atmospheric side of the seal, it will not be clogged from material in the pumpage. This is suited for a variety of applications, including paper stock, sewage, slurries, and river water. Two flush ports in the gland plate provide for a seal flush for cooling.

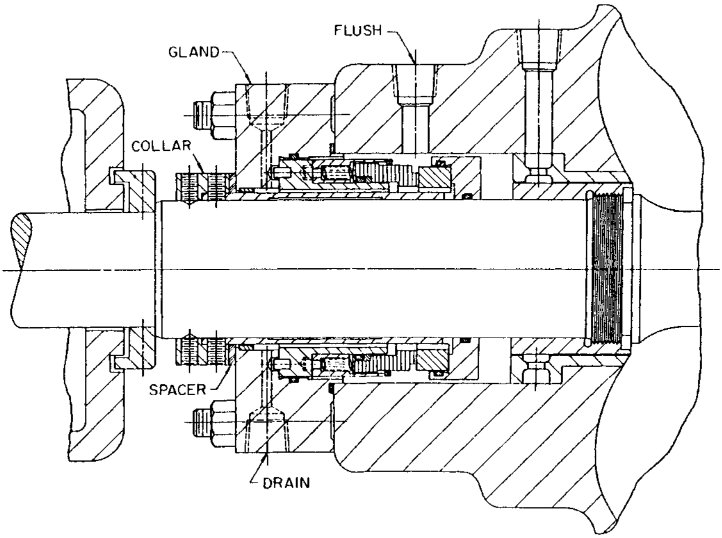


FIGURE 19 Stationary seal with a rotating mating ring (John Crane Inc.)

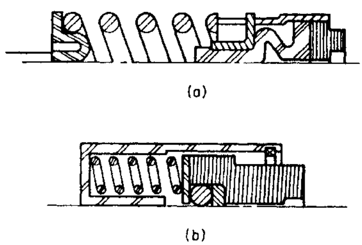


FIGURE 20 Comparison of a) single-spring and b) multiple-spring seals

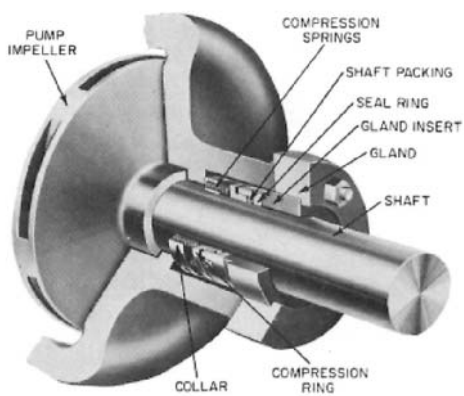


FIGURE 21 O-ring type mechanical seal (Flowserve Corp.)

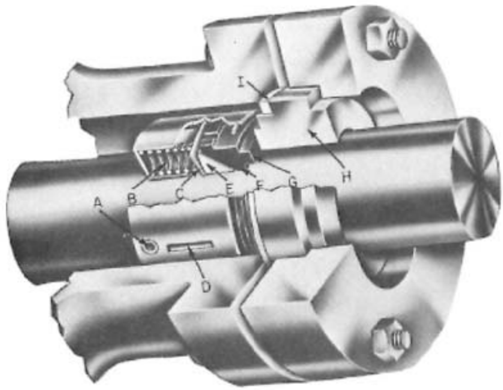


FIGURE 22 Wedge-type mechanical seal (John Crane Inc.)

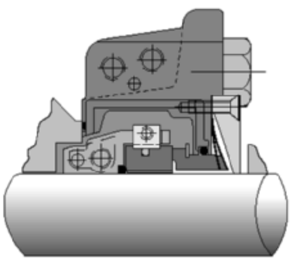


FIGURE 23 A split seal design for an ANSI pump (John Crane Inc.)

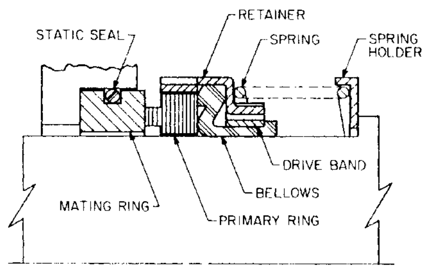


FIGURE 24 A half-convolution bellows seal (John Crane Inc.)

Nonpusher seals are defined as seal assemblies in which the secondary seal is not forced along the shaft by the mechanical load or hydraulic pressure in the seal chamber. Instead, all movement is taken up by the bellows convolution. This definition applies to those seals that use half-, full-, and multiple-convolution bellows as a secondary seal.

The half-convolution bellows seals are always made of an elastomer (see Figure 24). The tail of the bellows is held to the shaft by a drive band. This squeeze fit seals the shaft and enables the unit to rotate with the shaft. Positive drive is accomplished through the drive band, retainer, and primary ring by a series of slots and dents. A static seal is created at the back of the primary ring and at the front of the bellows. This type of seal is used for light-duty service conditions. The amount of axial travel along the shaft is half that of a full convolution bellows.

The full-convolution bellows seal is illustrated in Figure 25. The tail of the bellows is held to the shaft by a drive band. The squeeze fit seals the shaft and enables the unit to rotate with the shaft. The drive for the seal assembly is similar to that of the half-convolution seal. Static sealing is accomplished at the front of the bellows and the back of the primary ring. The heavier full-convolution bellows design can tolerate greater shaft motion and runout to pressures of 1200 lb/in^2 (8.3 bar).

Multiple-convolution bellows seals are necessary to add flexibility to those secondary seal materials that cannot be used in any other shape. The mechanical characteristics of TFE and metals require multiple-convolution designs.

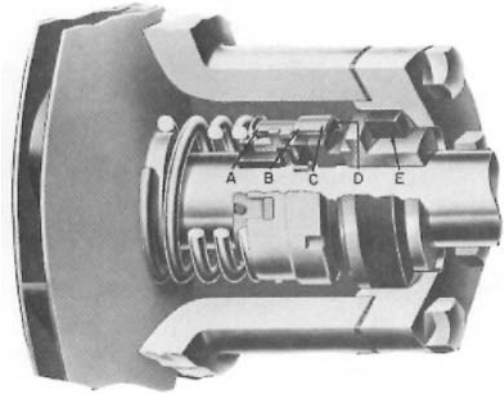


FIGURE 25 A full-convolution bellows seal (John Crane Inc.)

A TFE bellows assembly is illustrated in Figure 26. Because of the large cross-sectional area, these types of seals are mounted outside the seal chamber. Pressure at the inside diameter of the seal helps keep the faces closed. Small springs on the atmosphere side of the seal supply the mechanical load to keep the seal faces closed initially.

Multiple convolution metal bellows seals come in various designs and are discussed in the following section.

SEALING REQUIREMENTS IN THE PETROLEUM REFINING INDUSTRY

The API Standard 682 is an industrial standard developed by users with input from equipment and seal manufacturers. The goal of the standard was to create a specification for seals that would have a good probability of meeting emission standards defined by the U.S. Clean Air Act of 1990 and have a life of at least three years. The implementation of this specification indicates not only an increase in emissions control, but also a major increase in equipment reliability.

The following contacting, liquid-lubricated seal designs were identified by API 682, 1st edition (October 1994) as solutions to sealing refinery services. These have been verified by seal manufacturer tests under simulated refinery conditions. These are as follows:

- **Type A** A single, pusher-type seal mounted inside the seal chamber with a rotating flexible element. This is a balanced cartridge design with multiple springs and an O-ring as a secondary seal (see Figure 27). This seal is preferred for all refinery services except non-flashing hydrocarbons above 300°F (150°C). It is considered to be the standard for temperatures up to 500°F (265°C).
- **Type B** A single, low-temperature, non-pusher, inside-mounted seal, with a rotating metal bellows flexible element. The secondary static seals for this nickel allow metal bellows design are fluorocarbon elastomer O-rings. This low-temperature seal design is a standard optional selection for non-flashing hydrocarbon services up to 300°F (150°C).
- **Type C** A single, high-temperature, non-pusher, inside-mounted seal with a stationary metal bellows flexible element. The secondary static seals for this high-temperature bellows design are flexible graphite. This seal is the standard selection for non-flashing hydrocarbon applications when temperatures are above 300°F (150°C) and pressures are less than 250 lb/in^2 absolute (17 bar).

Each of the previous seal types is also available as a dual seal arrangement (see Figure 30). When the space between the inboard and outboard seals is pressurized with a barrier fluid, the seal arrangement is referred to as a pressurized dual seal. When the

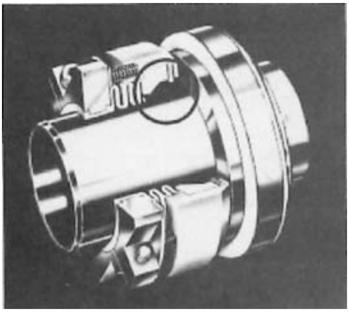


FIGURE 26 A Teflon bellows seal (John Crane Inc.)

FIGURE 27 Type A Single mounted pusher seal (John Crane Inc.)

FIGURE 28 Type B Single inside mounted non-pusher seal (John Crane Inc.)

FIGURE 29 Type C Single inside mounted high temperature seal (John Crane Inc.)

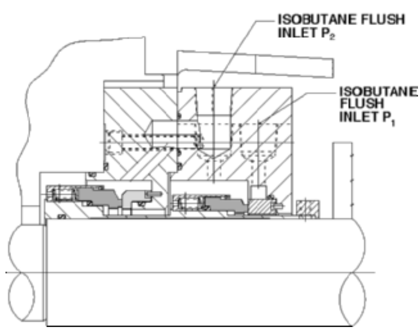


FIGURE 30 Pressurized dual API 682 seal for HF alkylatin service (John Crane Inc.)

space between the inboard and outboard seals is unpressurized with a buffer fluid, the seal arrangement is referred to as a nonpressurized dual seal. This is the only seal cartridge that can function as a double or tandem seal with the same individual seal parts. The terms *barrier fluid* and *buffer fluid* refer to the same fluid lubricating the seal. When the fluid is pressurized, it is a barrier fluid. When the fluid is non-pressurized, it is a buffer fluid.

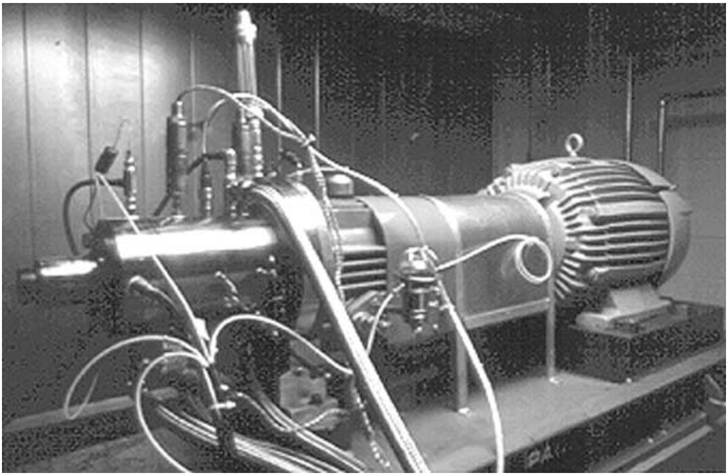


FIGURE 31 API 682 qualification test rig (John Crane Inc.)

The dual seal design shown in Figure 30 is without a pumping ring on the outside diameter of the outboard seal. This figure also represents a successful installation on an HF alkylation unit. In this design, isobutane is circulated at a pressure greater than at the pressure at the outside diameter of the inboard seal. The isobutane is then flushed over the inboard seal to keep the hydrofluoric acid away from the inboard seal. Seal life has been significantly increased with this improved sealing technology.

API 682 requires qualfication testing for all seal designs by the seal manufacturer. To meet these requirements, seal manufacturers constructed new testing facilities that allow testing at simulated refinery conditions for common process fluids. Figure 31 shows an API qualification test rig with instrumentation installed. Each seal type from each seal application group is required to be tested in four different test fluids that model fluids that model fluids from the application groups. These fluids include water, propane, 20 percent NaOH solution, and mineral oil. Each qualification test for each test fluid consists of three phases:

a. the dynamic phase at constant temperature, pressure, and speed
b. the static phase at 0 rpm using the same temperature and pressure as the dynamic phase
c. the cyclic phase at varying temperatures and pressures, including start-ups and shut-downs. For flashing hydrocarbons, the cyclic test phase includes excursions into vapor and back to liquid.

The seal is expected to perform within the regulated emissions limits after being exposed to qualification testing and upset conditions and demonstrate a capability of at least three years life in service. The result of this effort is not only an improvement in emssions control but also a major improvement in seal reliability. This naturally results in substantially lower life-cycle cost for the user.

The success or failure of a seal installation can often be traced to the selection of the proper piping arrangement. A piping arrangement or plan defines how a seal installation will be cooled or, in some cases, heated. Commonly used systems have been defined by API and are shown in Figure 32.

Performance testing to qualify seal designs to API 682 has resulted in an improved seal flush required for cooling. The mating ring, chamfered at the outside diameter, enables the flow of the flush liquid not only around the circumferential groove, but also directly to the

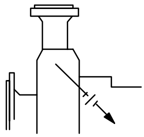
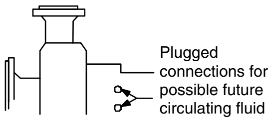
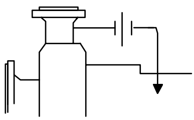
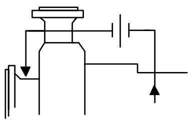
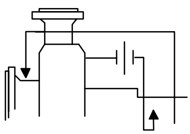
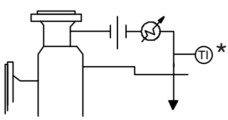
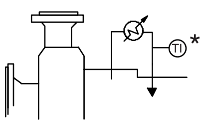
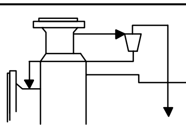
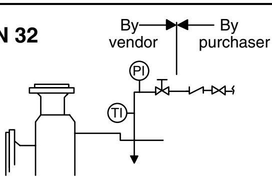
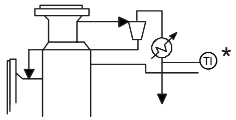
PLAN 01 	Fluid being pumped is circulated internally from discharge to seal chamber. Internal recirculation must be sufficient to maintain stable conditions at the seal face. Recommended for clean pumpage only and horizontal pumps. Not recommended for vertical pumps.
PLAN 02 	Dead-ended seal chamber with no circulation of a seal flush fluid. Used on special applications with horizontal pumps. Not recommended for vertical pumps.
PLAN 11 	Fluid pumped is circulated externally from discharge to seal chamber. An orifice may be used to control flow. The flow enters the seal chamber adjacent to the mechanical seal faces. This flow must be sufficient to maintain stable conditions at the seal faces. Not recommended on vertical pumps.
PLAN 13 	Fluid pumped is circulated from the seal chamber back to pump suction. An orifice may be used to control flow.
PLAN 14 	Fluid pumped is circulated from discharge to the seal chamber and back to the suction nozzle. An orifice, as shown, may be used to control flow, and must be sized in accordance with the throat bushing and the return line. Similar to Plan 11, flow back to suction side will evacuate vapor that may collect in the seal chamber. Recommended for vaporizing liquid such as those found in light hydrocarbon services.
PLAN 21 	Fluid pumped is circulated from discharge through a heat exchanger and into the seal chamber. An orifice, as shown, may be used to control flow. A dial thermometer (*) may be used in the recirculation line.
PLAN 23 	Fluid pumped is moved from the seal chamber by a pumping ring through a heat exchanger and back to the seal chamber. This plan can be used on hot applications to minimize heat load on the heat exchanger by cooling only the small amount of liquid that is recirculated. a dial thermometer (*) may be used in the recirculation line. Plan 21 Fluid pumped is circulated from discharge through a heat exchanger and into the seal chamber. An orifice, as shown, may be used to control flow. A dial thermometer (*) may be used in the recirculation line.
PLAN 31 	Fluid pumped is circulated from discharge through a heat exchanger and into the seal chamber. An orifice, as shown, may be used to control flow. A dial thermometer (*) may be used in the recirculation line.
PLAN 32 	A fluid separate from the pumpage is injected into the seal chamber from an external source. Care must be exercised in selecting an external fluid for injection to provide good lubrication to the seal and eliminate the potential for vaporization and also to avoid contamination of the pumpage with the injected flush. A dial thermometer (*) and flow indicator (*) are optional.

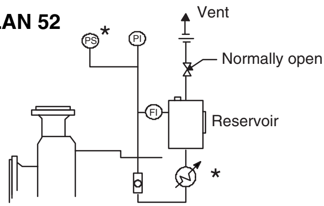
FIGURE 32 Piping plans for mechanical seals

PLAN 41



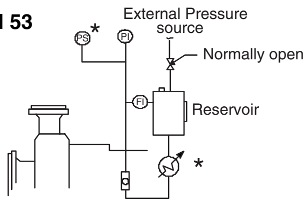
When a hot fluid is pumped which contains suspended abrasive particles, flow from the discharge to a cyclone separator delivers clean flow to the seal chamber through a heat exchanger. An orifice, as shown, may be used to control flow. Solids are delivered to pump suction. Clean discharge to the seal chamber and dirty discharge to pump suction must be at equal pressures. A dial thermometer (*) may be used in the flush line to seals.

PLAN 52



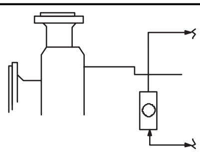
Applies to an outer seal of an unpressurized dual seal arrangement. An external reservoir provides a buffer fluid which is circulated by an internal pumping ring in the outboard seal cavity during normal operation. The reservoir is usually continuously vented to a vapor recovery system which is maintained at a pressure less than the pressure in the seal chamber. A pressure switch (*) and heat exchanger (*) are optional.

PLAN 53



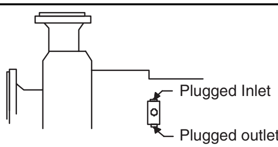
Applies to an outer seal of a pressurized dual seal arrangement. An external reservoir provides a barrier fluid under pressure which is circulated by an internal pumping ring in the outboard seal cavity during normal operation. Reservoir pressure is greater than the process pressure. A pressure switch (*) and heat exchanger (*) are optional.

PLAN 54



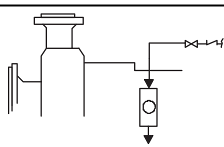
An outboard seal chamber is pressurized by a barrier fluid from an external reservoir. Circulation is by an external pressure system or pump. Reservoir pressure is greater than the process pressure being sealed.

PLAN 61



Tapped connections are plugged. When used, the purchaser provides quench fluid (steam, gas, water, etc.) to an auxiliary sealing device.

PLAN 62



An external source is used to provide a quench which is required to prevent solids from building up on the atmospheric side of the seal. Typically used with a close clearance throttle bushing.

NOTE: This table provides a quick reference to piping plans described in API Standard 610, 8th edition August 1995, for centrifugal pumps for petroleum, heavy duty chemical and gas industry. The reader is encouraged to consult this specific standard for more detailed information.

LEGEND

	Heat exchanger		Cyclone separator		Check valve
PI	Pressure gauge with block valve	FI	Flow indicator		Orifice
TI	Dial thermometer		Flow regulating valve	LI	Level indicator
PS	Pressure switch with block valve		Block valve		

FIGURE 32 Continued.

seal faces for cooling. This design also helps to eliminate any trapped vapor at the seal faces (see Figure 33).

GLAND PLATE CONSTRUCTION

An essential component of any seal installation is the *gland plate*. The purpose of this part is to hold either the mating ring assembly or the seal head assembly, depending on whether the seal head is rotating with the shaft or stationary to the pump casing. It is also a pressure-containing component of the installation. The alignment of one of the sealing surfaces, particularly the mating ring used with a rotating seal assembly and a gland plate bushing, is dependent on the fit of the gland plate to the pump. To ensure the proper installation, the API specification requires a register fit with the inside or outside diameter of the seal chamber. The static seal on the face of the seal chamber must be completely confined. Three basic gland plate constructions are shown in Figure 34:

- A *plain* gland plate is used where seal cooling is provided internally through the pump stuffing box and where the liquid to be sealed is not considered hazardous to the plant environment and will not crystallize or carbonize at the atmospheric side of the seal.
- A *flush* gland plate is used where internal cooling is not available. Here coolant (liquid sealed or liquid from an external source) is directed to the seal faces where the seal heat is generated.
- A *flush-and-quench* gland plate is required on those applications that need direct cooling as well as a quench fluid at the atmospheric side of the seal. The purpose of the quench fluid, which may be a liquid, gas, or steam, is to prevent the buildup of any carbonized or crystallized material along the shaft. When properly applied, a seal quench can increase the life of a seal installation by eliminating the loss of seal flexibility due to hangup. This gland plate can also be used for flush, vent, and drain where seal leakage needs to be controlled. Flammable vapors leaking from the seal can be vented to a flare and burned off, while nonflammable liquid leakage can be directed to a safe sump.

Figure 35 illustrates some restrictive devices used in the gland plate when quench or vent-and-drain connections are used. These bushings can be pressed in place, as in Figure 35a, or allowed to float as in Figures 35b, c, and d. Floating bushings enable closer running clearances with the shaft because such bushings are not restricted at their outside diameter. The bushing shown in Figure 35d is also split to enable the thermal expansion of the shaft. This restrictive bushing is preferred on refinery applications. Small packing rings can also be used for a seal quench, as shown in Figure 35e.

SEAL CHAMBER DESIGN

A critical part of the sealing system is the seal chamber design. The selection of the proper seal chamber can increase the life of the mechanical seal. Changes in pump design have resulted in three chambers, as shown in Figure 36. These are the conventional seal chambers, referred to as the standard bore, the enlarged bore, and the tapered bore. Seal chambers can influence the seal environment through pressure, solids handling, vapor removal, and temperature. A standard bore seal chamber was originally designed for packing and has a restriction at the bottom of the chamber that limits the interchange of fluids between the chamber and pumpage. This seal chamber is dependent on the application of the proper piping plan to remove heat or abrasives.

The enlarged bore is similar to a standard bore, enabling the installation of large seal cross-sections and dual seals. The increased cross-section increases the volume of the liquid for cooling. This seal is also dependent on the proper piping plan to remove heat.

The tapered bore seal chamber has increased radial clearance and the bottom of the chamber is exposed to the impeller. The walls of the chamber promote self-venting during shutdown and self-draining during disassembly. Internal flow within the chamber elimi-

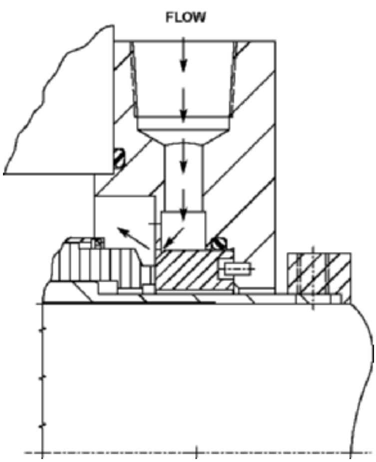


FIGURE 33 Improved seal flush for refinery services (John Crane Inc.)

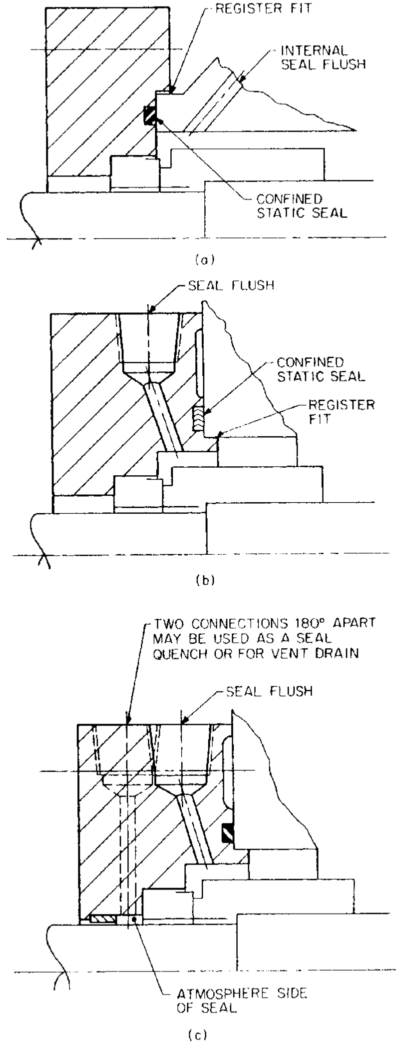


FIGURE 34 Basic gland plate designs: (a) plain gland plate, (b) flush gland plate, (c) flush and quench, or flush vent and drain gland plate

nates the need for external piping for cooling. For some applications, operating at a higher pressure will require an external flush.

NON-CONTACTING GAS LUBRICATED SEALS

The evolution of this sealing concept for pumps has its origin in gas sealing technology developed for gas compressors in the mid-1970s. The development of this technology for

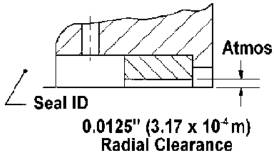
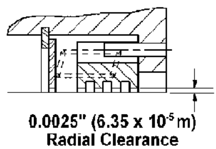
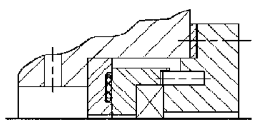
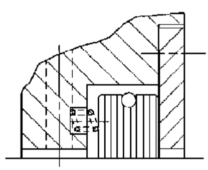
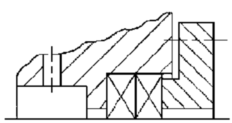
DESCRIPTION		COMMENTS
(a) **Throttle Bushing**		Used on vent and drain designs. Made of non-sparking materials. Meets API specifications.
(b) **Floating Throttle Bushing**		May be used on quench, vent and drain glands. Spring-loaded to float with shaft. Made of non-sparking materials. Meets API specification. Requires more space than fixed bushing.
(c) **Floating Bushing**		Soft packing sized to shaft diameter used on quench. Vent and drain glands. Spring loaded. No adjustments required. Excellent dry run.
(d) **Split Floating Bushing**		May be used on quench, vent and drain glands. Spring-loaded to float with shaft. Split carbon bushing to take into account shaft expansion due to temperature preferred for API application.
(e) **Packing Rings**		Creates positive seal on quench designs. Requires some adjustment during operation.

FIGURE 35 Common restrictive devices used with quench, or vent and drain gland plates.

pumps was accelerated in the early 1990s as a way to control emissions. Eliminating the contact between the seal faces while the pump shaft is turning also eliminates the tribological problems of frictional heat and wear. This is no easy task with a liquid present in the pump. Deflections of the seal faces from temperature and pressure must be controlled to very precise levels. Gas, rather than a liquid, must be used as a barrier fluid and one of the seal faces must be designed with a lift mechanism.

A lift mechanism can take the shape of a spiral groove, an L-shaped slot, or a controlled wavy surface. When the shaft turns, pressure builds up in the seal faces, which causes the face separation. The basic construction of a spiral groove face and the pressure buildup is shown in Figure 37. The separation of the face is very small and could be measured in nanometers, which enables a small amount of gas to flow across the seal face. Since this type of seal is non-contacting, the only heat that is developed is from the shearing of gas at the seal faces. The small amount of gas flow helps cool the seal faces. The temperature rise at the seal faces is just a few degrees, making this a preferred seal for heat-sensitive liquids. The processes occurring at the seal faces are shown in Figure 38.

The operating envelope for a non-contacting, gas-lubricated seal for liquid pumping services is shown in Figure 39. Since the rubbing contact at the seal faces has been elim-

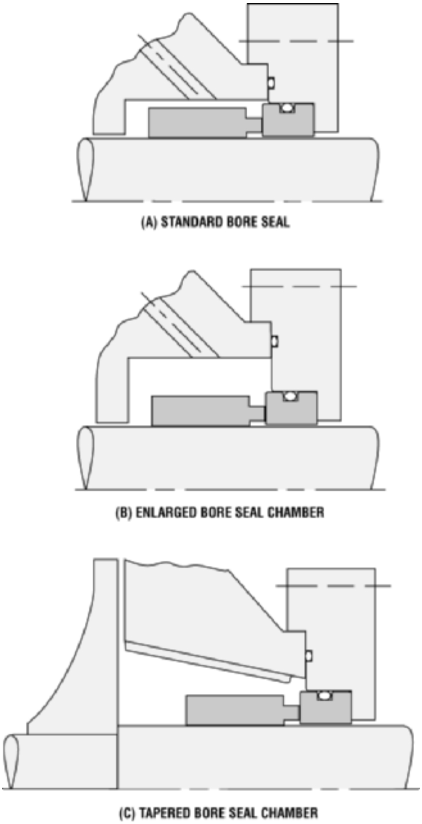


FIGURE 36 Standard bore, enlarged bore, and tapered bore seal chamber arrangements.

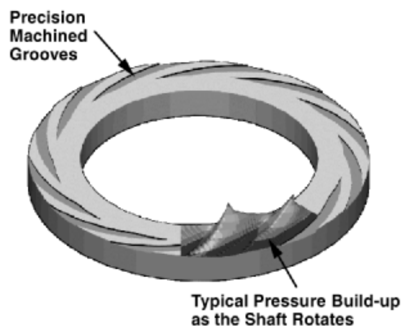


FIGURE 37 Spiral groove seal face and pressure build up in the grooves (John Crane Inc.)

inated, the seal can be operated at the vapor pressure of the liquid being sealed. In addition, no limiting factor exists due to the pressure-velocity relationship and wear at the seal faces. The limiting factor in the application of the seal is the pressure that has an effect on seal face deflection.

Dual pressurized, gas-lubricated seals have been designed to fit oversized and small bore seal chambers. An oversized seal chamber that enables a larger cross-section seal and that can handle pressures up to 600 psig (40 bar) is illustrated in Figure 40.Many existing pumps in the field have small bore seal chambers and do not require the same pressure capability as a larger cross-section seal. The seal to fit these units is illustrated in Figure 41. This type of seal is being used to pressures of 230 psig (16 bar).

Dual seals are pressurized with an inert gas. Gas pressure is normally 20 to 30 psig (1.4 to 2 bar) above the process liquid being sealed. The amount of gas consumed through a dual seal can be estimated from Figure 42. The gas consumption is the sum of the flow across the inboard and outboard seals. The amount of gas consumed on an annual basis is very small. This makes this type of installation very economical when compared to a fully pressurized liquid barrier/buffer system. Gas pressure and flow are monitored by a safety gas panel like that shown in Figure 43.

This technology represents a solution for those sealing problems that affect the cooling and lubrication of a contacting seal by the liquid being sealed. These problems, identified

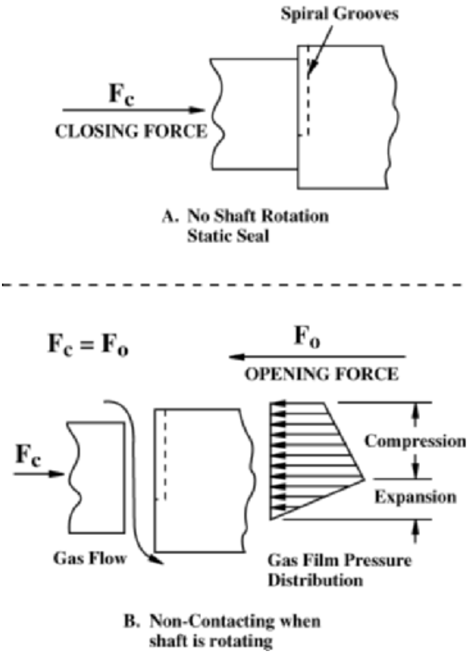


FIGURE 38 The processes occurring at the seal faces

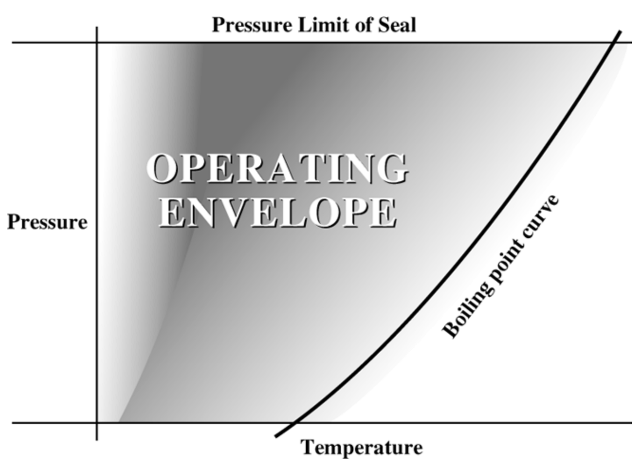


FIGURE 39 The operating envelope for a non-contacting, gas-lubricated seal for liquid pumping services. (John Crane Inc.)

by users as reasons for a short seal life, include a loss of seal flush, dry running, startup without venting, *low net positive suction head* (NPSH), and cavitation. Cavitation may also result in vibration of the equipment. The vibration limit for a non-contacting gas lubricated seal is 0.4 in/s (10 mm/s). The benefits of this technology are increased performance, emissions control, safety, reliability, and efficiency for a conventional pump fitted

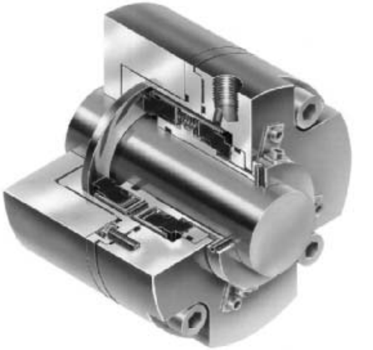


FIGURE 40 Non-contacting gas-lubricated seal for pumps with a large bore seal chamber (John Crane Inc.)

FIGURE 41 Non-contacting gas-lubricated seal for pumps with a small cross-section seal chamber (John Crane Inc.)

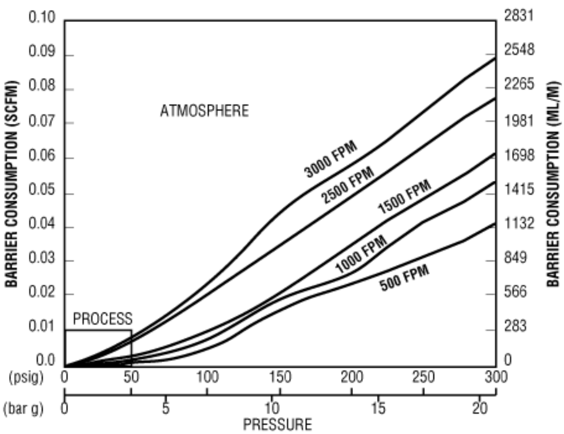
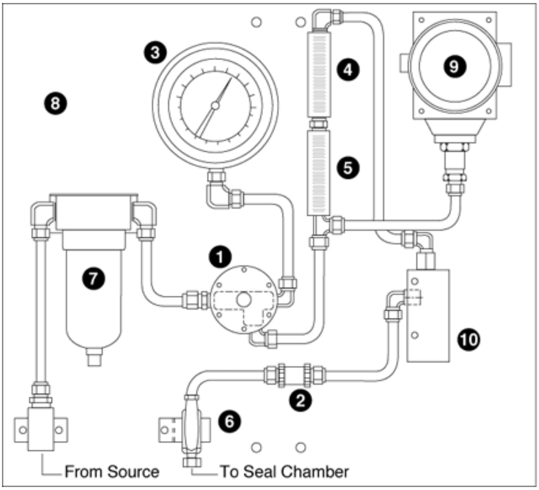


FIGURE 42 Gas consumption through a seal face. (John Crane Inc.)

with non-contacting gas lubricated seals. This translates into increased *mean time between maintenance* (MTBM) and reduced costs of ownership of the equipment. An inert gas, such as nitrogen, is a preferred barrier fluid in refinery and petrochemical industries, while purified air is used in the pharmaceutical and biotech industries. Protecting the environment is the main reason for the development of this technology, but, it is also a primary sealing system used to maintain product purity in the pharmaceutical and biotech industries.

Pumping liquid near its vapor pressure represents a challenge to equipment manufacturers and plant operators. Trying to seal this type of application with a contacting seal will result in an inefficient installation. The amount of heat generated would require too much cooling to prevent the liquid from flashing. The only solution is to eliminate the heat generated at the seal faces, allowing the liquid being sealed to flash to a gas and use a non-contacting gas lubricated seal.

For those liquids that are dangerous to the environment, a tandem seal arrangement would be used. The space between the seals would be vented to a flair or vapor disposal



System Components
1. Pressure Regulator
2. Check Valve
3. Pressure Gauge
4. Flow Meter - High
5. Flow Meter - Low
6. Isolating Valve
7. Coalescing Filter
8. Panel
9. Pressure Switch (Optional)
10. Flow Switch (Optional)

FIGURE 43 A safety gas panel for monitoring gas pressure and flow (John Crane Inc.)

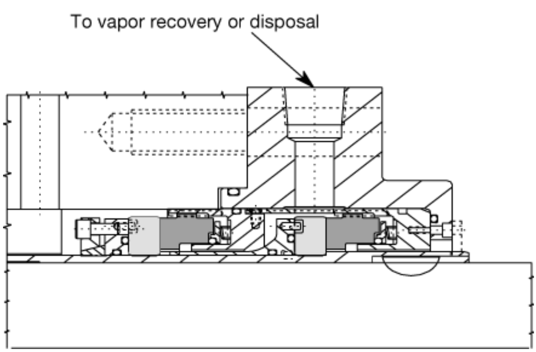


FIGURE 44 Early non-contacting gas-lubricated seal design for vaporizing hydrocarbon service (John Crane Inc.)

TABLE 2 Typical vaporizing hydrocarbon services (John Crane Inc.)

Operating Condition/ Seal Size (in)	Liquid	Pressure (lb/in^2)	Temperature (Degree F) Min/Max	Speed (rpm)
1.250	Ethane	554	+45 / +125	3,560
1.875	LNG	327	+37 / +125	3,560
2.875	LNG	400	+30 / +125	3,560
3.375	Ethane	392	−49 / +125	1,750
5.250	LNG	850	+30 / +125	3,560

area. Figure 44 represents an early tandem seal arrangement for this type of service. Applying this technology to pumps has resulted in a significant increase in equipment reliability. A list of typical applications for non-contacting gas seals to be applied to vaporizing hydrocarbon liquid services is shown in Table 2.

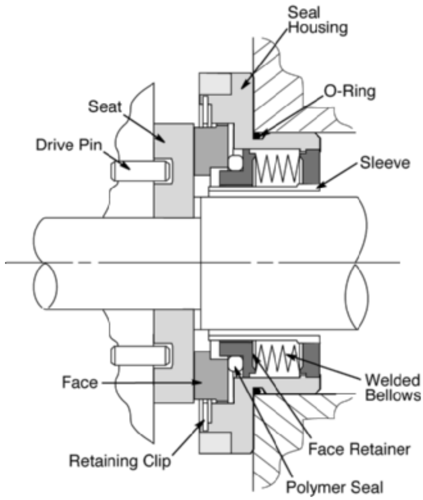


FIGURE 45 A non-contacting gas-lubricated seal for cryogenic service (John Crane Inc.)

Cryogenic liquids represent a similar design challenge in sealing technology. Traditionally, pumps that are used to pump these liquids relied on contacting seal designs. Although these fluids were at cryogenic temperatures, the seals were operating near the boiling point of the liquid. Frictional heat was enough to vaporize or flash the liquid to a gas. This resulted in short seal life. By allowing the liquid to flash to a gas and by using non-contacting gas lubricated seals, seal life has been extended from weeks to years. A non-contacting gas lubricated cryogenic seal is illustrated in Figure 45. Due to the low temperatures involved, a metal bellows is required in the seal design.

MATERIALS OF CONSTRUCTION

All component parts of a seal are selected based on their corrosion resistance to the liquid being sealed. The *National Association of Corrosion Engineers* (NACE) Corrosion Handbook provides corrosion rates for many materials of construction for mechanical seals used with a variety of liquids and gases. When the corrosion rate is greater than two mils (0.05 mm) per year, double seals that keep the hardware items of the seal in a neutral liquid should be selected to reduce corrosion. In this design, only the inside diameter of the mating ring, the primary ring, and the secondary seal are exposed to the corrosive liquid and should be constructed of corrosion-resistant materials, such as ceramic, carbon, and Teflon. Common materials of construction are given in Table 3. Table 4 lists the properties of common seal face materials.

The operating temperature is a primary consideration in the design of the secondary and static seals in the assembly. These parts must retain their flexibility throughout the life of the seal, as flexibility is necessary to retain the liquid at the secondary seal as well as to enable a degree of freedom for the primary ring to follow the mating ring. The usable temperature limits for common secondary and static seal materials are given in Table 5.

An additional consideration in the selection of the primary and mating ring materials in sliding contact is their *PV* limitation. This value is an indication of how well the material combination will resist adhesive wear, which is the dominant wear in mechanical seals. Limiting *PV* values for various face combinations are given in Table 6. Each limiting value has been developed for a wear rate that provides an equivalent seal life of two years. A *PV* value for an individual application can be compared with the limiting *PV*

TABLE 3 Common materials of construction for mechanical seals

Components	Materials of Construction
Secondary Seals:	
O-rings	Nitrile, Ethylene Propylene, Chloroprene, Fluoroelastomer, Perfluoroelastomer
Bellows	Nitrile, Ethylene Propylene, Chloroprene, Fluoroelastomer
Wedge or U Cups	Fluorocarbon
Metal Bellows	Stainless steel, Nickel-base Alloy
Primary Ring	Carbon, Metal-filled Carbon, Tungsten Carbide, Silicon Carbide, Siliconized Carbon, Bronze
Hardware (retainer, disc, snap rings, set screws, springs)	Stainless Steel, Nickel-base Alloy
Mating Ring	Ceramic, Cast Iron, Tungsten Carbide, Silicon Carbide

value for the materials used to determine satisfactory service. These values apply to aqueous solutions at 120°F (49°C). For lubricating liquids such as oil, values of 60 percent or higher can be used. Higher or lower values of *PV* may apply, depending on the seal face design.

INSTALLING THE SEAL AND IDENTIFYING CAUSES OF SEAL LEAKAGE

A successful seal installation requires operation of the pump within the manufacturer's specification. Relative movement between the seal parts or shaft sleeve usually indicates that mechanical motion has been transmitted to the seal parts from misalignment (angular or parallel), endplay, or radial runout of the pump (see Figure 46).

Angular misalignment results when the mating ring is not square with the shaft and will cause excessive movement of internal seal parts as the primary ring follows the out-of-square mating ring. This movement will fret the sleeve or seal hardware on pusher type seal designs. Angular misalignment may also occur from a seal chamber that has been distorted by piping strain developed at operating temperatures. Damage in the wearing rings can also be found here if the pump seal chamber has been distorted.

Parallel misalignment results when the seal chamber is not properly aligned with the rest of the pump. No seal problems will occur unless the shaft strikes the inside diameter of the mating ring. If damage has occurred, there will also be damage to the bushing at the bottom of the seal chamber at the same location as the mating ring.

Excessive axial endplay can damage the seal surfaces and cause fretting. If the seal is continually being loaded and unloaded, abrasives can penetrate the seal faces and cause premature wear of the primary and mating rings. Thermal damage in the form of heat checking in the seal faces because of excessive endplay can occur if the seal is operated below working height.

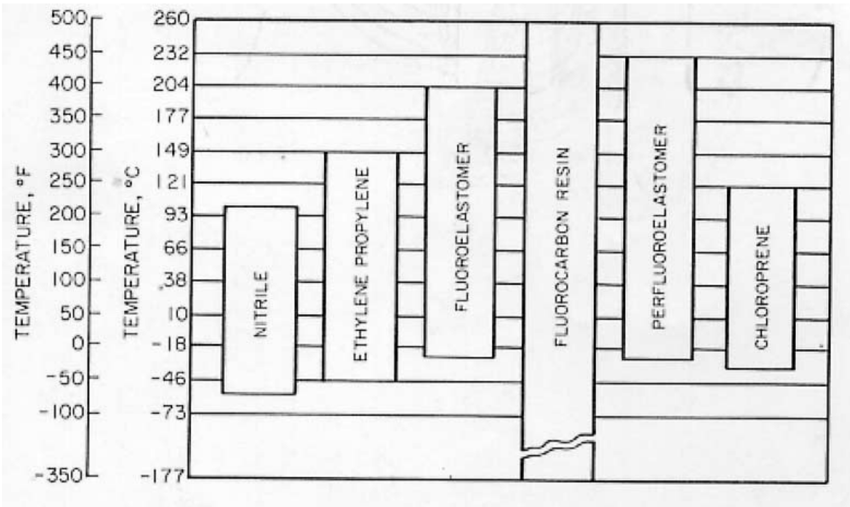
Radial runout in excess of limits established by the pump manufacturer could cause excessive vibration at the seal. This vibration, coupled with small amounts of the other types of motion that have been defined, will shorten seal life.

Instructions and seal drawings should be reviewed to determine the installation dimension or spacing required to ensure that the seal is at its proper working height (see Figure 47). The installation reference can be determined by locating the face of the seal chamber on the surface of the sleeve and then measuring along the sleeve after it has been

TABLE 4 Typical properties of common seal face material

			CERAMIC		CARBIDES		CARBON			
Property	**Cast Iron**	**Ni-resist**	**85% (AL_2O_3)**	**99% (AL_2O_3)**	**Tungsten (6% Co)**	**Silicon (SiC)**	**Resin**	**Antimony**	**Bronze**	**SiC Conv.**
Modulus of Elasticity $\times 10^6$ lb/in^2 ($\times 10^3$ Mpa)	13–15.95 (90–110)	10.5–16.9 72–117)	32 (221)	50 (245)	90 (621)	48–57 (331–393)	2.5–4.0 (17.2–27.6)	3.8–4.8 (26.2–33.1)	2.9–4.4 (20–30)	2-2. (13.8–15.9)
Tensile Strength $\times 10^3$ lb/in^2 (Mpa)	65–120 (448–827)	20–45 (138–310)	20 (138)	39 (269)	123.25 (8;50)	20.65 (142)	4.5–9 (31–62)	7.5–9.0 (52–62)	7.5–9 (52–62)	2 (14)
Coefficient of Thermal Expansion $\times 10^{-6}$ in/in F (cm/cm K)	6.6 (11.88)	6.5–6.8 (11.7–12.24)	3.9 (7.02)	4.3 (7.74)	2.53 (4.55)	1.88 (4.55)	2.3.–3.4 (4.14–6.12)	2.3.–4.7 (4.14–8.46)	2.4–3.1 (4.32–5.58)	2.4–3.2 (4.32–5.76)
Thermal Conductivity Btu · ft/h · ft^2 °F (w/m $\times$ k)	23–29 (39.79–50.17)	25–28 (43.25–48.44)	8.5 (14.70)	14.5 (25.08)	41–48 (70.93–83.04)	41–60 (70.93–103.8)	3.8–12 (6.57–20.76)	5.8–9.0 (10.0–15.6)	8–8.5 (13.84–14.70)	30 (51.9)
Density: lb/in^3 (kg/m^3)	0.259–0.268 (7169–7418)	0.264–0.268 (7307–7418)	0.123 (3405)	0.137 (3792)	0.50 (16.331)	0.104 (2879)	0.064–0.069 (1771–1910)	0.083–0.112 (2297–3100)	0.083–0.097 (2297–2685)	0.067–0.070 (1854–1938)
	Brinell		Rockwell A			Rockwell 45N	Shore			Rockwell 15T
Hardness	217–269	131–183	87	87	92	86–88	80–105	75–100	70–92	90

TABLE 5 Temperature limits of secondary seal materials



removed from the unit. It is not necessary to use this procedure if a step in the sleeve or collar has been designed into the assembly to provide for proper seal setting. Assembling other parts of the seal will bring the unit to its correct working height.

All package or cartridge shaft seals can be assembled with relative ease because just the bolts at the gland plate and set screws on the drive collar need to be fastened to the seal chamber and shaft. After the seal spacer is removed, the unit is ready to operate.

To assemble a mechanical seal to a pump, a spacer coupling is required. If the pump is packed but may later be converted to mechanical seals, a spacer coupling should be included in the pump design.

Since a seal has precision-lapped faces and because secondary seal surfaces are critical in the assembly, installations to the equipment should be kept as clean as possible. All lead edges on sleeves and glands should have sufficient chamfers to facilitate installation.

When mechanical seals are properly applied, there should be no static leakage and, under normal conditions, the amount of dynamic leakage should range from none to just a few drops per minute. Under a full vacuum, a mechanical seal is used to prevent air from leaking into the pump. If excessive leakage occurs, the cause must be identified and corrected. Causes for seal leakage with possible corrections are listed in Table 7. In addition, Figure 48 illustrates the most common causes for mechanical seal leakage. Further information on seal leakage and the related condition of seal parts can be found in the works listed in the "Further Reading" section.

TABLE 6 Frequently used seal face materials and their *PV* limitations

Sliding Materials		PV limit,	
Rotating	Stationary	lb/in^2 · ft/min (bar · m/s)	Comments
Carbon-graphite	Ni-resist	100,000 (35.03)	Better thermal shock resistance than ceramic
	Ceramic (85% Al_2O_3)	100,000 (35.03)	Poor thermal shock resistance and much better corrosion resistance than Ni-resist
	Ceramic (99% Al_2O_3)	100,000 (35.03)	Better corrosion resistance than 85% Al_2O_3 Ceramic
	Tungsten Carbide (6% Co)	500,000 (175.15)	With bronze-filled carbon graphite, PV is up to 100,000 lb/in^2 ft/min (35.02 bar · m/s)
	Tungsten Carbide (6% Ni)	500,000 (175.15)	Ni binder for better corrosion resistance
	Silicon Carbide (converted Carbon)	500,000 (175.15)	Good wear resistance; thin layer of SiC makes relapping questionable
	Silicon Carbide (solid)	500,000 (175.15)	Better corrosion resistance than Tungsten Carbide but poorer thermal shock resistance
Carbon-graphite		500,000 (17.51)	Low PV, but very good against face blistering
Ceramic		10,000 (3.50)	Good service on sealing paint pigments
Tungsten Carbide		120,000 (42.04)	PV is up to 185,000 lb/in^2 ft/min (64.8 bar · m/s) with two grades that have different % of binder
Tungsten Carbide/ Silicon Carbide (solid)		300,000 (105.1)	Excellent abrasion resistance. Commonly used on high temperature applications
Silicon Carbide (converted carbon)		500,000 (175.15)	Excellent abrasion resistance, more economical than solid Silicon Carbide
Silicon Carbide (solid)		350,000 (122.6)	Excellent abrasion resistance, good corrosion resistance and moderate thermal shock resistance

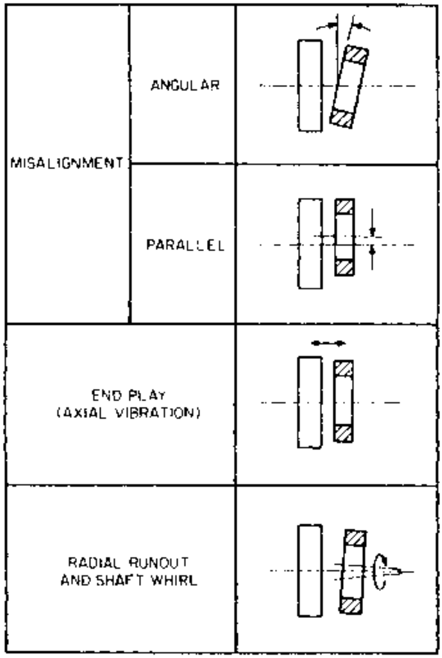


FIGURE 46 Common types of motion that influence seal performance

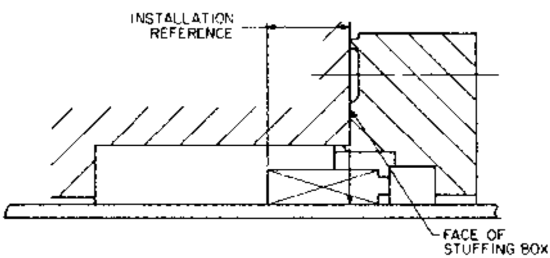


FIGURE 47 Typical installation reference dimensions

TABLE 7 Checklist for identifying causes of seal leakage

Symptom	Possible Causes	Corrective procedures
Seal spits and sputters ("face popping") in operation	Seal fluid vaporizing at seal interfaces	Increase cooling of seal faces. Check for proper seal balance with seal manufacturer Add bypass flush line if not in use Enlarge bypass flush line and/or orifices in gland plate Check for seal interface cooling with seal manufacturer
Seal drips steadily	Faces not flat Carbon graphite seal faces blistered Seal faces thermally distorted	Check for incorrect installation dimensions Check for improper materials or seals for the application Improve cooling flush lines Check for gland plate distortion due to overtorquing of gland bolts Check gland gasket for proper compression Clean out foreign particles between seal faces; relap faces if necessary Check for cracks and chips at seal faces; replace primary and mating rings
	Secondary seals nicked or scratched during installation O-rings overaged Secondary seals hard and brittle from compression set Secondary seals soft and sticky from chemical attack	Replace secondary seals Check for proper lead-in chamfers, burrs, and so on Check for proper seals with seal manufacturer Check with seal manufacturer for other material
	Spring failure Hardware damaged by erosion Drive mechanisms corroded	Replace parts Check with seal manufacturer for other material

(continues)

TABLE 7 Continued.

Symptom	Possible Causes	Corrective procedures
Seal squeals during operation	Amount of liquid inadequate to lubricate seal faces	Add bypass flush line if not in use Enlarge bypass flush line and/or orifices in gland plate
Carbon dust accumulates on outside of gland ring.	Amount of liquid inadequate to lubricate seal faces Liquid film evaporating between seal faces	Add bypass flush line if not in use Enlarge bypass flush line and/or orifices in gland plate Check for proper seal design with seal manufacturer if pressure in stuffing box is excessively high
Seal leaks	Nothing appears to be wrong	Refer to list under “Seal drips steadily” Check for squareness of stuffing box to shaft Align shaft, impeller, bearing, and so on to prevent shaft vibration and/ or distortion of gland plate and/or mating ring
Seal life is short.	Abrasive fluid	Prevent abrasives from accumulating at seal faces Add bypass flush line if not in use Use abrasive separator or filter
	Seal running too hot	Increase cooling of seal faces Increase bypass flush line flow Check for obstructed flow in cooling lines
	Equipment mechanically out of line	Align Check for rubbing of seal on shaft

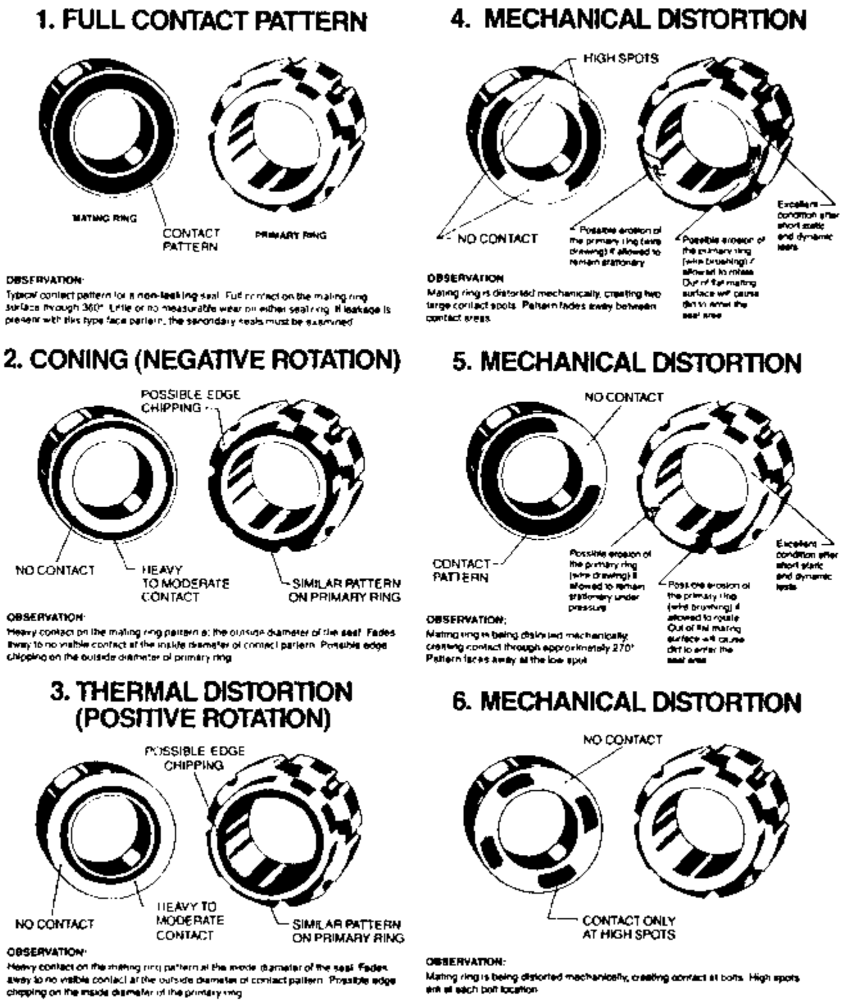


FIGURE 48 Identifying causes of seal leakage (John Crane Inc.)

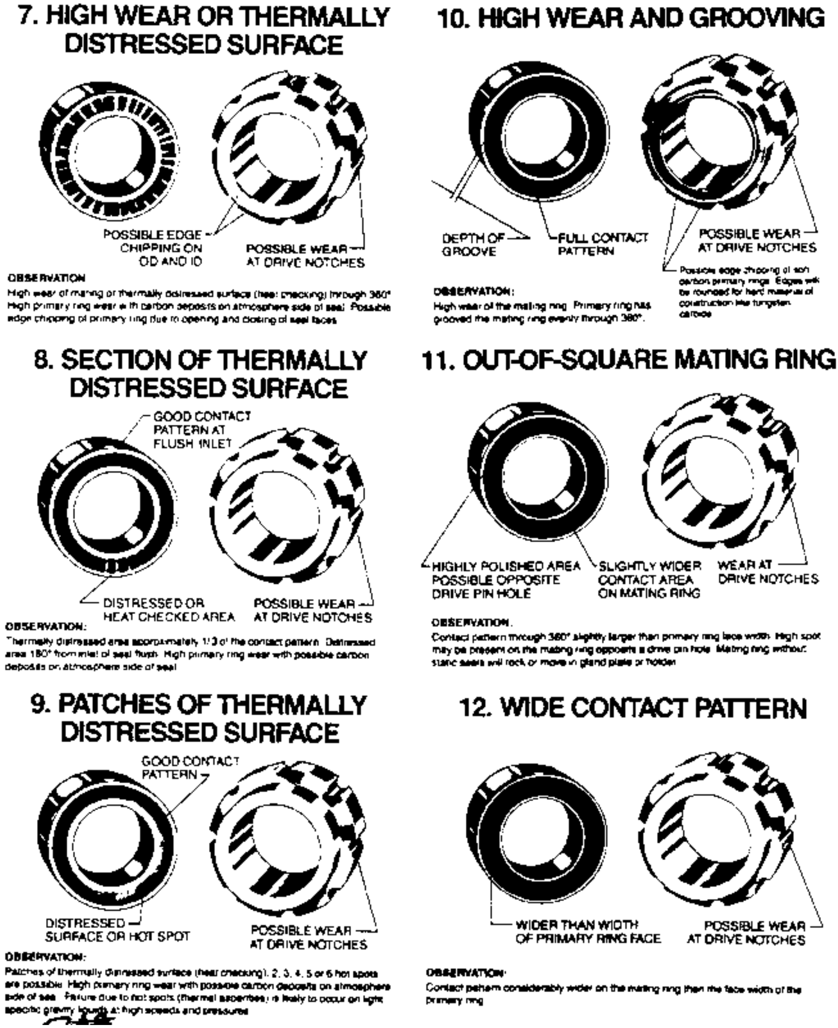


FIGURE 48 Continued.

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