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ABSTRACT

Even though dry gas seals for centrifugal compressors are becoming more popular, knowledge and understanding of oil seals and their associated support systems are still very important. Many existing compressors still have oil seals; likewise, dry gas seals have pressure and surface velocity limitations that prevent them from being applied to all applications. The author describes the two most common types of API oil seals so that users can understand the fundamentals of each. Likewise, the strengths and weaknesses for each seal type are listed so users can have input into specification of their seals dependent upon their particular application. Examples are given of problematic seal designs and the changes made to improve them. The support systems for each type of oil seal are described, along with ways to specify the best system for different applications. A step-by-step method is given for troubleshooting seals and their support systems in the field. Case studies of problematic oil seal systems with solutions are presented.

INTRODUCTION

Compressor oil seals are designed around the principle of forcing oil into the compressor seals at a higher pressure than the process gas to prevent the gas from escaping from the pressure casing. In fact, compressor oil seals are not gas seals at all, but rather liquid seals designed to minimize the amount of seal oil that passes into the compressor. The seal oil is normally supplied from a combined lube and seal oil system, an example of one such system is shown in Figure 1. This system is designed to keep the seal oil pressure above the sealed gas pressure.

Oil seals are generally grouped into two different groups: face contact seals and floating ring seals. The two different types have different strengths and weaknesses that make them attractive and/or applicable to different applications. Likewise, their associated seal oil support systems are quite different.
face due to the high shaft rotational speeds encountered in compressors (i.e., high rotational speeds would cause springs to deflect and result in an unstable seal). Face seals have maximum surface velocity and sealing pressure limitations (Table 1).

Table 1. Approximate Pressure and Velocity Limitations of Face Contact Seals.

<table>
<thead>
<tr>
<th>Maximum Surface Velocity (fps)</th>
<th>300</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum Sealed Pressure (psia)</td>
<td>500-1500</td>
</tr>
</tbody>
</table>

A floating bushing seal (item 27, Figure 2) is normally used on the atmospheric or outer side of the seal to keep the seal housing pressurized. The seal oil that passes through the outer seal is drained directly back to the lube oil reservoir because it has not come in contact with the process gas. For this reason, it is called uncontaminated or “sweet” seal oil. The bushing seal clearance is sized to maintain the correct differential pressure on the process seal while at the same time providing enough flowrate through the seal housing to cool the seal (normally around five to 10 gpm). Alternatively, some seal designs use a second contact seal on the atmospheric side (Figure 3). This is usually required at high surface velocities, if the oil flow cannot be controlled adequately enough with a bushing seal. Depending upon the seal design and application, the sweet seal oil flow may require a separate drain to the reservoir. The flow through this separate drain is either restricted by an orifice or regulated by a control valve to maintain the seal oil differential pressure.

To lower the relative surface velocity between the rotating and stationary faces, some face type oil seals include a floating nonmetallic ring (usually carbon, item 1, Figure 4) between the two faces. This ring normally spins at approximately one-half the shaft rotational speed. This lowers the relative velocity between the seal faces to approximately half of the shaft speed. The carbon ring is normally scalloped on the inside diameter to make sure a pressure differential does not exist across it and pressed inside a steel ring (Figure 5). However, the lower velocity does not come without a cost, it does add an additional sealing face that can leak as well.

Face seals provide positive shutoff; i.e., they will contain the process gas if the seal oil differential is lost. This is very important in hazardous applications, even if buffer gas is used on the seals. Positive shutoff is maintained by the springs as well as a set of shutdown pistons (item 14, Figure 6). If the seal oil pressure differential drops to a few psid, the pistons will move in the outboard direction and push the seal retainer against the rotating seat. Alternatively, some seal designs incorporate a separate sealing face for shutdown protection (item 1, Figure 7). This design prevents process gas from escaping by closing the additional seal face on the outside of the contact seal. At the same time it pushes the contact seal together.

**Balance**

Seal balance or balance ratio is the ratio of the seal face closing area to seal face contact area (Figures 8 and 9, and Equation (1)).
Balance ratios for compressor oil seals typically range from 60 to 80 percent, which means that there is a slight closing bias. Leakage rates for compressors are much higher than pumps (typically gallons per day for compressors compared to parts per million (ppm) for pumps) because the faces require more lubrication and cooling. Typically, most compressor seals are designed for a complete liquid film across the entire seal face, in comparison to pumps where it is common for the sealed liquid to vaporize at the inside diameter of the seal faces. The liquid film provides for a much longer operating life.

\[ Balance = B = \frac{\text{closing area}}{\text{contact area}} = \frac{r_0^2 - r_b^2}{r_o^2 - r_i^2} \quad (1) \]

Seal Face Loading

The hydraulic axial load on the seal faces is a function of the seal balance, face orientation, differential pressure, and pressure profile across the seal faces (Figure 10). The shape of the pressure profile across the seal faces is a function of the seal face orientation. The seal faces are never exactly parallel. A divergent face profile is unstable and results in face wear and short run-life. The parallel or slightly convergent face profile results in lower leakage rates and is the normal design case (Table 2).

A nonparallel face orientation is not due solely to manufacturing tolerances or installation errors. The seal face runs warmer on the inside diameter because the oil film is thinner on the inside than the outside (Figure 11). Additionally, there is a large temperature gradient in the axial direction as well. These temperature gradients tend to deflect the seal faces in a convergent direction (Figure 12). Likewise, pressure induces moments on the stationary seal faces that "tips" or deflects the seal faces (Figure 13). Since the stationary face pivots on the secondary seal, the distance from the seal face to the secondary seal affects face orientation and as a result the seal face loading (SFL) (i.e., as the rotor shifts axially, the pressure effects
Table 2. Seal Face Orientations.

<table>
<thead>
<tr>
<th>Face Orientation</th>
<th>Stability</th>
<th>Leakage</th>
<th>Life</th>
</tr>
</thead>
<tbody>
<tr>
<td>Parallel</td>
<td>Stable – but very</td>
<td>Low</td>
<td>Long</td>
</tr>
<tr>
<td></td>
<td>difficult to obtain</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>exactly</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Hydrodynamic lift</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>only produced by</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>surface roughness.</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Converging</td>
<td>Stable – produces</td>
<td>Low/Moderate</td>
<td>Long,</td>
</tr>
<tr>
<td></td>
<td>hydrodynamic lift</td>
<td></td>
<td>assuming</td>
</tr>
<tr>
<td></td>
<td>Maybe the result</td>
<td></td>
<td>faces do</td>
</tr>
<tr>
<td></td>
<td>of temperature</td>
<td></td>
<td>not touch.</td>
</tr>
<tr>
<td></td>
<td>effects in seal</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Diverging</td>
<td>Unstable – no</td>
<td>Very Low</td>
<td>Short</td>
</tr>
<tr>
<td></td>
<td>hydrodynamic lift</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>Will cause wear on</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>OD and result in</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>failure or parallel</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

on the stationary face can change). This sensitivity to axial location can be eliminated by fixing the secondary seal in the stationary face instead of on the seal carrier (Figure 14). The temperature and pressure effects are normally designed to oppose each other such that the face orientation is parallel or slightly convergent.

The hydraulic load along with the spring load are added together to provide the seal face loading (Equation (2)). The pressure profile across the seal faces is commonly approximated as linear, which results in $K = 0.5$.

$$SFL = P_{SPRING} + \Delta P_{OIL} (B - K)$$  \hspace{1cm} (2)

**Leakage**

Sour oil leakage for mechanical contact type seals is normally five to 10 gpd per seal. Typical axial clearances between seal faces are only 0.0001 to 0.0005 inch. Therefore, only small increases in this clearance can cause high leakage rates. Equation (3) gives an estimate of the leakage rate between two radial faces, assuming the oil film thickness is uniform across the seal (i.e., the faces are parallel). Note: the flowrate is proportional to the cube of the axial clearance ($h$).

$$Q = \frac{\pi r h^3 \Delta p}{6 \mu \Delta r}$$  \hspace{1cm} (3)

As an example, a cracked gas compressor was recently consuming approximately 750 gpd (from one end only). To determine if the problem could be caused by the face seals “hanging up,” a calculation was made to determine the amount of axial clearance required for this leakage rate. Equation (3) can be rearranged to give the clearance for a given leakage rate (Equation (4)).

$$h = 3 \left( \frac{6 Q \mu \Delta}{r} \right)$$  \hspace{1cm} (4)

For a 6.75 inch seal with 45 psid seal oil differential and using 100 SSU oil at an estimated inlet temperature of 110°F, the axial clearance ($h$) required for the actual leakage of 750 gpd is only 0.0015 inch! Considering that this particular compressor floated 2 to 3 mils axially every day because of load swings, it seemed very plausible that the leakage was a result of a stationary face hung up on the seal carrier. This estimate was further backed up by the fact that the seal balance was very low, approximately 50 percent. If the pressure gradient across the seal face is assumed linear (i.e., $K = 0.5$), a 50 percent balance results in the SFL to be provided only by the spring load (i.e., $B - K = 0$ in Equation (2)). In other words, if the seal faces are disturbed from their equilibrium condition, the only restoring force is that applied by the springs. This makes the seal very susceptible to build up between the stationary face and the seal carrier, which might keep the seal faces further apart when the rotor shifts axially.

One of the largest weaknesses of the face contact seal is the dynamic gasket on the stationary face. The gasket is required to slide back and forth on the seal carrier or seal ring as the rotor moves axially. This motion can wear a groove in the seal carrier,
which can cause the stationary face to “hang up.” Likewise, dirty seal oil or other contaminants can build up in the area between the stationary face and the seal housing and hang up the stationary face (Figure 16). Either of these situations can cause the seal leakage rate to increase dramatically. This weakness is especially acute on compressors that have large swings in load (which results in large amounts of axial travel).

Figure 15. Stationary Face with Build Up on Sour Side (Inside Diameter).

Materials

The seal face materials are very similar to those seen in pump seals. Nickel or antimony bonded carbon is commonly used as stationary face material. Likewise, silicon carbide, tungsten carbide, or surface-hardened steel is used for rotating seats. The faces are lapped to a surface flatness of approximately $10^{-6}$ inches. Secondary seals are normally made from Teflon® or Viton® depending on the process. The metal parts are normally 316 SS, Inconel® or Hastelloy® depending upon the corrosive nature of the gas.

Rotordynamic Effects

The process side of contact seals (with or without a center carbon ring) has very little rotodynamic effect on the compressor rotor. Because the differential pressure across the process side is typically low (< 100 psid), there is very little frictional radial load carrying capability in the contact seal. Without the frictional radial load, the seal cannot develop stiffnesses that contribute to the rotordynamics of the system. In contrast, the atmospheric side, which is normally a bushing style, can affect the rotordynamics of the compressor. Depending upon the differential pressure, balance, L/D ratio, and shaft clearance, the atmospheric ring can “lock up” in an eccentric position and produce marginal amounts of destabilizing cross-coupled stiffness. However, this can be avoided by designing the atmospheric ring to reduce its frictional load (i.e., lapping faces, decreasing balance ratio) and by reducing the L/D ratio. In contrast, if the axial load on the atmospheric ring is too low, fretting can occur (Figure 16). The atmospheric ring also produces a great amount of direct damping that helps to stabilize the rotor.

Face Contact Seal Oil Systems

The most common seal oil system for face contact oil seals is forward pressure control (Figure 17). This system uses a differential pressure controller on the seal oil supply to maintain the required seal oil supply pressure. The reference gas pressure is normally supplied from the process gas side of the labyrinth seal shown in Figure 2, note the port labeled “control gas.” This gas reference pressure should always be located between the seal and the buffer gas port. Likewise it should always be obtained from the end of the compressor with the highest pressure (i.e., discharge or interstage suction) to prevent balance piston problems from starving one side of the compressor for oil. The sweet oil is drained back to the reservoir through either the atmospheric seal and/or a separate orificed drain. The sour oil is drained away to the seal traps.

Figure 16. Wear on Load Face of a Bushing Atmospheric Seal.

Figure 17. Contact Seal Oil System, Forward Pressure Control.

An alternate seal oil system for face contact oil seals is the backpressure control shown in Figure 18. The system uses a differential pressure controller as well, but is placed on the sweet oil drain back to the reservoir (i.e., the “control oil” port on Figure 2). Normally an adjustable flow controller is required on the supply oil line as well. This is normally set at a fixed amount determined by the seal vendor.
FLOATING RING TYPE OIL SEALS

Floating ring oil seals typically comprise two to four stationary rings in a seal housing that are allowed to float radially with the shaft (Figure 19). There is normally only one ring on the process side of the seal, with varying numbers of rings on the atmospheric side (but normally not more than three). Unlike face seals, floating ring seals have no pressure or speed limitations. There are two sealing surfaces on floating rings: the radial face that slides on the seal housing and the clearance between the shaft sleeve and the seal ring. The radial face is usually lapped to an 8 to 15 rms finish. Likewise, the inside diameter of the rings is normally lined with a thin layer of babbit (approximately 0.010 to 0.060 inch deep). The seal sleeves (or shaft) that the seal rings run on are normally hard-surfaced by weld overlay (typical hardness of 50 to 60 Rockwell C). Axial clearance between the rings and the seal housing is normally 0.010 to 0.020 inch per ring.

The seal oil is supplied between the inner and outer rings (Figure 19). The differential pressure across the inner ring (which is on the process gas side) is normally very low, around 5 psid. This differential is normally maintained by elevating a seal oil tank approximately 15 ft above the shaft centerline and connecting the sealed gas reference pressure to the top of the tank (Figure 20). The seal oil supply is either pumped up to the seal oil tank and then gravity fed down into the compressor or connected to a drain off the bottom of the seal oil tank such that the supply pressure is set by the tank. The inner ring normally has the tightest clearance (approximately 0.0005 to 0.001 inch/inch of shaft diameter) to reduce the amount of seal oil that is exposed to the process. To further reduce the sour leakage, a windback groove may be cut into the inside diameter of the inner ring (Figure 21). This causes the inner ring to pump the seal oil back toward the outer ring. Likewise, for applications where the process gas may corrode the babbit, aluminum may be sprayed on the inside diameter of the inner ring. Typical sour oil leakage rates are approximately 10 to 25 gpd.

The atmospheric side of the seal has a differential pressure equal to the sealed gas pressure plus the seal oil differential pressure. It is not uncommon for the atmospheric side of a floating ring seal to have differential pressures in excess of several thousand psid. It just depends upon the process conditions. The clearance between the outer ring(s) and the seal sleeve is sized to provide enough oil flow to cool the seal. The clearance is normally higher than the process side (normally 0.001 to 0.003 inch/inch of shaft diameter). Higher
pressure applications are normally designed with two or three outer rings (or one ring with multiple grooves) to reduce the L/D ratio of the rings. Normally, a compressed O-ring on the face of the inner ring loads the inner and outer rings in the axial direction (item 5, Figure 22). This axial loading is provided to help keep the inner ring centered since it has very little differential pressure to hold it in place. Because the compression of the O-ring is very important, using the correct durometer is critical. Alternative seal designs use a spring between the inner and outer rings to provide axial loading. The spring is located between items 6 and 7 in Figure 23. This prevents changes in axial loading due to O-ring degradation and also prevents the O-ring from causing the inner ring to hang up in an eccentric position. Normally the outer rings are loaded in the axial direction by a substantial differential pressure. The radial sealing surface on the outboard of the outer rings can easily wear in high-pressure applications. For this reason, it is common for these contact surfaces to be hard-surface coated (i.e., HVOF) (Figure 23). This produces hardness values that are approximately 70 to 75 Rockwell C.

$\frac{\partial}{\partial z} \left( h^3 \frac{\partial P}{\partial z} \right) + \frac{1}{R^2} \frac{\partial}{\partial \theta} \left( h^3 \frac{\partial P}{\partial \theta} \right) = \omega \frac{\partial h}{\partial \theta} + 2 \frac{\partial h}{\partial \theta} \left( \frac{3 h^2}{2} \right)$ (Equation 5)

$Q = \frac{\pi D h^3 \Delta P}{\mu L}$ (centered) (Equation 6)

It is important for the floating ring seals to stay centered to reduce the amount of leakage. The leakage rate for eccentric rings is approximated by Equation (7).

$Q = \frac{\pi D h^3 \Delta P}{\mu L} \left( 1 + \frac{3 h^2}{2} \right)$ (eccentric) (Equation 7)

Neither floating ring nor pumping bushing seals provide positive shutoff. If the seal oil pressure drops below the process gas pressure, the process gas will escape down the shaft. For this reason, they are not normally used for hazardous applications (i.e., H2S, acid gas).

**Pumping Bushing Seal**

A slight variation of the floating ring seal is the pumping bushing seal. The straight inner ring is replaced with a conical sleeve (Figure 24) or a radially stepped arrangement (Figure 25). The pumping bushing design has larger radial clearances between the stationary inner ring and the rotating member (typically 0.008 to 0.012 inch). The flow of sour oil into the compressor is retarded by the dynamic pumping action of the cone or radial step, which acts like an impeller. The seal leakage rate of the pumping bushing seal is less than standard floating ring seals but more than contact seals. Because of the larger radial clearance, the pumping bushing seal may be considered less susceptible to contaminants in the seal oil than standard floating ring seals. However, it is much more sensitive to changes in rotational speed and axial misalignment due to thermal growth. The pumping bushing seal requires flow through seal oil, so its seal oil system is slightly more complicated because it has an overhead seal oil tank and a back-pressure controller (Figure 26). The flow-through oil is required to remove the additional heat generated by the pumping bushing.

Because of the low differential pressure across the inner ring, the correct operation of floating ring seals is very dependent upon the condition of the balance piston seal (Figure 20). Besides counteracting the thrust in the compressor, the balance piston seal reduces the pressure on the discharge end seal to approximately suction pressure. Normally, the differential pressure across the balance line is less than 2 psid (even for high-pressure applications). If the balance piston seal fails, the seal reference pressure will rise as the balance cavity pressure rises. Because the seals are supplied off a common header, this increases the differential pressure across the inner ring in the suction end seal. This will increase the sour oil leakage on the suction end. Note: failure of the balance piston seal has much less effect on the oil seals if the compressor is equipped with a seal equalizing line.

**Leakage**

The leakage through both the inner and outer rings is approximately laminar due to the very small clearances. This flow regime is modeled with the Reynolds equation, which is a simplified combination of the conservation of mass and momentum equations for a fluid (Equation (5)). For a centered ring with constant viscosity and linear pressure drop, the leakage between the ring and the rotating shaft can be approximated by Equation (6). The net result is that the leakage between the floating rings and shaft sleeve is proportional to the cube of the clearance.

$Q = \frac{\pi D h^3 \Delta P}{\mu L}$ (centered) (Equation 6)

$Q = \frac{\pi D h^3 \Delta P}{\mu L} \left( 1 + \frac{3 h^2}{2} \right)$ (eccentric) (Equation 7)

Neither floating ring nor pumping bushing seals provide positive shutoff. If the seal oil pressure drops below the process gas pressure, the process gas will escape down the shaft. For this reason, they are not normally used for hazardous applications (i.e., H2S, acid gas).
Floating ring seals normally require a minimum suction pressure to ensure proper lubrication before starting the compressor (Figure 27). Obviously, this is more of a concern for high-pressure seals designed to operate with high differentials across the outer rings. Note that the minimum pressure is a function of the shaft rotational speed.

**Rotodynamic Effects**

The rotodynamic effects of the floating ring seals are similar to those of face seals, though floating ring seals tend to have more rotodynamic problems due to their application in high-pressure applications. The inner seal ring has very little effect on the rotodynamics of the compressor because there is very little differential pressure across the ring (< 5 psid). This normally prevents the development of large frictional loads between the seal ring and the seal housing. In contrast, the outer rings can greatly affect the rotodynamics of the system, especially if it is a high-pressure application. The high differential pressure can cause large radial frictional forces between the floating rings and the housing. This can cause the rings to lock up in an eccentric radial position and act like a straight journal bearing. This can produce large amounts of destabilizing cross-coupled stiffness. This can be reduced by various methods:

- **Lower the balance**—Adjusting the balance so that the normal load produced by the differential pressure is lower, resulting in lower radial frictional loads. However, a minimum load is required to keep the seal ring from fretting.
- **Hard surface radial faces**—Coat the radial sealing faces to reduce the coefficient of friction between the floating rings and seal housing, thus reducing the tendency of the seal rings to lock up in an eccentric position.
- **Increase the number of segments in the seal land**—This increases the turbulence in the seals that reduces the cross-coupled stiffness produced, but may reduce the direct damping as well. A full rotodynamic study must be done to evaluate the effects of this change.
- **Replace the inner ring of the outer ring group with a tilt-pad ring/bearing** (Figure 22)—This tilt-pad seal produces less cross-coupled stiffness than the one-piece ring. The centered leakage rate may be slightly higher, but since the seal tends to stay more centered than conventional ring seals, the actual leakage may be less.

A comparison of contact and floating ring seals is shown in Table 3.

**Table 3. Comparison of Contact and Floating Ring Seals.**

<table>
<thead>
<tr>
<th></th>
<th>Mechanical</th>
<th>Contact</th>
<th>Standard</th>
<th>Pumping Bushing</th>
</tr>
</thead>
<tbody>
<tr>
<td>Construction</td>
<td></td>
<td>Very simple</td>
<td>Simple</td>
<td></td>
</tr>
<tr>
<td>Seal oil dp</td>
<td>75-40 psid</td>
<td>5-10 psid</td>
<td>5-10 psid</td>
<td></td>
</tr>
<tr>
<td>Leakage rates</td>
<td>5-10 gpd</td>
<td>10-25 gpd</td>
<td>10-15 gpd</td>
<td></td>
</tr>
<tr>
<td>Positive Shutoff</td>
<td>Yes</td>
<td>No</td>
<td>No</td>
<td></td>
</tr>
<tr>
<td>Rotodynamic Effects</td>
<td>Negligible</td>
<td>May be a problem with high pressure applications</td>
<td>May be a problem with high pressure applications</td>
<td></td>
</tr>
<tr>
<td>Pressure Limits</td>
<td>500-1500 psig</td>
<td>None</td>
<td>5000 psig</td>
<td></td>
</tr>
<tr>
<td>Speed Limits</td>
<td>300 ft/s</td>
<td>None</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Clearance</td>
<td>0.0001-0.0005 inch</td>
<td>0.0015-0.005 inch</td>
<td>0.0008-0.0014 inch</td>
<td></td>
</tr>
<tr>
<td>Seal oil system</td>
<td>DP control</td>
<td>Ovhd Tank</td>
<td>Ovhd Tank w/ DP</td>
<td></td>
</tr>
<tr>
<td>Sensitive to axial growth</td>
<td>Yes</td>
<td>No</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Sensitive to speed changes</td>
<td>Slightly</td>
<td>No</td>
<td>Yes</td>
<td></td>
</tr>
<tr>
<td>Sensitive to balance piston problems</td>
<td>Very Little</td>
<td>Yes</td>
<td>Yes</td>
<td></td>
</tr>
</tbody>
</table>
TROUBLESHOOTING GUIDELINES/STEPS

The first step to solving a seal problem is identifying the problem, which is usually one of the following:

- Cannot maintain seal oil differential pressure or seal oil tank level
- Excessive sour seal oil consumption
- Excessive sweet seal oil consumption

Performing the following guidelines/steps can help identify the seal or seal system problem and provide valuable information for design modifications that may be needed to improve the reliability of the seal system.

1. **Monitor sour oil consumption regularly** (i.e., how fast is the level dropping in the reservoir)—If a degassing tank is used, shut it off periodically to measure sour oil consumption. Even if the degassing tank is working correctly, the amount of sour leakage should be monitored.

2. **Check seal oil differential pressure reading**—A differential pressure gauge or transmitter should always be used, not two separate pressure gauges. For contact seals verify that the seal oil differential pressure is maintained at a constant value as the process pressure changes. This is very important on startup, when the process pressure may change rapidly. Likewise, for floating ring seals, verify that the seal oil level in the overhead tank is not fluctuating. A common problem is a plugged gas reference line, which results in the seal oil differential pressure being either too high or too low as the process changes.

3. **Check oil quality, water content, viscosity, metal additives.**
   - Improper seal oil (too light or too heavy) can cause the seals to run hotter than design, resulting in higher wear and leakage. OEMs usually recommend a certain viscosity range, typically 100 to 150 SSUs at the seal oil supply temperature. This may require blending in a heavier oil during higher ambient temperatures if seal oil cooling is a problem.
   - Water content should be kept below 50 ppm (below 10 ppm if possible). Water will greatly reduce the viscosity of the oil. Likewise, water may have metals or other contaminants that will foul the seals (Figures 28 and 29).
   - Generally, oil additives are bad for seals. Metallic additives can react with process gases and cause fouling at the seal faces. Turbine oils, which normally contain only a small amount of antifoam and antioxidation additives, are normally recommended. Seal oil should contain less than 5 ppm zinc.

4. **Measure seal oil supply flowrates**— Normally, seal oil systems are designed to control the seal oil differential pressure, not the oil flowrate. Likewise, the seal oil supply to the compressor is normally common for both ends of the compressor (i.e., the seal oil differential pressure is only controlled at one seal). Measuring the flowrate to the seals at key locations can reveal problems in the seal oil system.
   - Check seal oil control valve output. The output of the DPC control valves in Figures 17 and 18 as well as the level control valve in Figure 20 can be used to calculate the seal oil flowrate to the seals, because the seal oil flowrate is not normally measured. The flowrate through a control valve can be approximated by Equation (8). Note: the value of $C_v$ is dependent upon the valve position as well. Figure 30 shows a typical $C_v$ chart for different types of control valves (this can be supplied by the valve manufacturer). Bench testing the control valve (in a shop) and measuring the flowrate and valve position can achieve the most accuracy.

$$Q = C_v \sqrt{\Delta P} \quad \text{(\% Open)} \quad (8)$$

Figure 28. Deposits on Shaft Sleeve Caused by Contaminated Seal Oil (Amine and Water).

Figure 29. Grooves in Shaft Sleeve Caused by Oil Rings Due to Contaminated Seal Oil.

Figure 30. Values of $C_v$ for Three Types of Typical Control Valves.

- Measure the seal oil flow to each end of the compressor using a nonintrusive flowmeter (ultrasonic is preferable). The higher seal oil supply flow may not be the problem seal. Since
both seals are normally supplied from a common header, if the atmospheric seal fails on one end, it can starve the entire seal on the other end. Knowing the supply flowrate can assist in determining whether the excessive leakage is on the process or atmospheric side. For example, the original seal oil flow diagram for a wet gas compressor with contact seals is shown in Figure 31. The seal oil flowrates were measured at various locations and are shown in Figure 32. As can be seen, nearly all the seal oil is going to one side. This one-sided flowrate is confirmed by observing the bearing oil drain on the end with a failed bushing seal. The sight glass in the bearing drain was completely full. Likewise, there was lube oil leaking from the coupling guard. This was caused by the flowrate through the bushing exceeding the drain capacity of the sweet oil drain and spilling over into the coupling guard (Figure 33). Inspection of the seals showed broken pieces of metal in the sweet oil drain (Figure 34). It was determined that the broken pieces shown in Figure 34 were pieces of the wavy spring used to provide a preload for the atmospheric bushing seal (item 12, Figure 35). The spring failed because it had spun with shaft (Figures 36 and 37). The broken spring damaged the bushing as it was pushed out the atmospheric bushing (Figure 38).

- Compare the sour seal oil flowrates (either through flow sight glasses or by monitoring how fast the levels rise in the seal oil traps).
• Compare the sour oil drain temperatures on both ends of the machine. Typically, the hottest sour drain indicates the highest sour oil flow. The flow through the sour oil drain is normally low enough that the line is relatively cool. Increased sour oil leakage will cause it to warm up.

• Compare the sweet oil drain temperatures on both ends of the compressor. In contrast to the sour drains mentioned above, the cooler sweet oil drain is many times the problem (higher flow dilutes temperature rise in seal). For example, the seal oil pump for a high-pressure barrel compressor was having trouble maintaining a level in the seal oil overhead tank. The measured seal oil supply rate indicated that the discharge end seal was consuming approximately three times as much seal oil as the suction end (Figure 39). Even though the gas on the discharge seal end is much hotter (there was no buffer gas), the sweet oil drain on the discharge end was 30°F cooler because of the higher flowrate.

5. Check buffer gas differential pressure

• Too low—If a buffer gas is used to isolate the seals from the process gas, the buffer pressure is normally maintained at 3 to 5 psid above the process gas pressure (which should always be the balance piston end of the compressor). Too little buffer gas differential pressure can allow the process gas to migrate into the seals, which may foul the sour seal oil. Adequate buffer gas differential can be verified by sampling the seal gas off the sour traps. If it contains process gas, the differential is too low.

• Too high—Too much buffer gas differential pressure ( > 5 psid) can cause a venturi-type effect in the compressor (Figure 40). This can result in sour oil being ingested into the compressor instead of draining to the sour traps.

6. Check differential pressure between seal cavities (Figure 41)—Most compressor OEMs design the balance piston for a differential of less than 2 psid. If the balance piston seal begins to wear, this differential can go up, which can be very detrimental for floating ring seals, which typically operate with only 5 psid across the process ring. As mentioned above, the gas reference pressure should always be connected to the high-pressure end of the compressor. Increasing the balance line differential causes the low-pressure seal to have a higher seal oil differential pressure, which will increase its sour oil leakage. Likewise, if for some reason the gas reference is on the low-pressure seal, increased balance line differential can cause the gas...
pressure on the high-pressure side to exceed the seal oil pressure. This will cause the seal to overheat and fail. As mentioned above, a seal equalization line will prevent this problem. Even though contact seals operate with a much higher seal oil differential pressure, the seal traps should still operate at the same pressure. If they do not it probably indicates a problem with the sealing system.

Figure 40. Effects of Excessive Buffer Gas on Sour Oil Leakage.

Verify that degassing tank is working properly—Many systems today have a seal oil degassing tank, which takes the sour oil from the seal oil traps, heats it to approximately 180°F, and vents off any light hydrocarbons or water (Figure 42). One word of caution, some hydrocarbons can collect in the drum that have a flashpoint above 180°F. Likewise, most amines (MDEA for sure) have a flashpoint over 250°F and will collect at the bottom of the degassing tank. After a period of time, the level of heavier contaminants will rise until they reach the level of the standpipe and are drained into the reservoir. For this reason, the drain off the bottom of the tank should be opened at least once a shift by operations to drain away any contaminants collecting there. A faulty or poorly designed degassing tank can contaminate a seal oil reservoir in a matter of days, which is much faster than the intervals plants sample their lube oil for analysis. Nitrogen sparging can be used to improve the efficiency of the degassing tank. It reduces the partial pressure of H2S in the tank, which lowers the H2S concentration in the oil. Degassing tanks should always be vented to atmosphere if at all possible.

Figure 41. Measuring Differential Pressure Between Seal Oil Traps.

7. Verify that degassing tank is working properly—Many systems today have a seal oil degassing tank, which takes the sour oil from the seal oil traps, heats it to approximately 180°F, and vents off any light hydrocarbons or water (Figure 42). One word of caution, some hydrocarbons can collect in the drum that have a flashpoint above 180°F. Likewise, most amines (MDEA for sure) have a flashpoint over 250°F and will collect at the bottom of the degassing tank. After a period of time, the level of heavier contaminants will rise until they reach the level of the standpipe and are drained into the reservoir. For this reason, the drain off the bottom of the tank should be opened at least once a shift by operations to drain away any contaminants collecting there. A faulty or poorly designed degassing tank can contaminate a seal oil reservoir in a matter of days, which is much faster than the intervals plants sample their lube oil for analysis. Nitrogen sparging can be used to improve the efficiency of the degassing tank. It reduces the partial pressure of H2S in the tank, which lowers the H2S concentration in the oil. Degassing tanks should always be vented to atmosphere if at all possible.

8. Verify that the seal oil traps are properly vented—If the seal oil traps are not vented properly, the sour seal oil will be ingested into the compressor. This can easily happen if the sour oil trap vents are not routed to a low-pressure destination such as the flare header. It is common for the vents off high-pressure sour oil traps to be routed to an alternate destination to recover the gas. If the destination pressure increases or the suction pressure of the compressor decreases, the traps will not work correctly. An example is a recycle hydrogen compressor in a Gulf Coast refinery that always consumed large amounts of seal oil on unit startup until the unit lined out and then it would quit. The problem only occurred during startup because the sour oil traps were vented to the plant fuel gas system. This was an adequate design as long as the process unit was operating properly and the suction pressure of the compressor was sufficiently above the fuel gas pressure to vent the sour oil traps. However, on startup the fuel gas system created enough backpressure on the oil traps to prevent the traps from working properly.

9. Determine if the compressor has been pressurized without the seal oil system in service—If the compressor has been pressurized before the seal oil system was in operation (i.e., the seal oil differential pressure was negative), the process gas may have pushed trash into the space between the seal faces (contact seals) or between the floating rings and the seal sleeves.

10. Check performance of seal oil pump—A lack of seal oil differential may be caused by insufficient seal oil pump performance. The easiest method to determine if the seal oil pump is performing well is to switch to the spare. Normally both pumps will not be deteriorated at the same time. Use the flowrate measured in step 4 above as well as the motor amps to determine if the pump is performing correctly. Likewise, verify that the seal oil temperature is below 110°F. Especially in high-pressure units, the slippage through a positive displacement pump is greatly affected by the oil temperature (i.e., viscosity).

11. Check discharge check valve on spare seal oil pump—A leaking discharge check valve on the spare seal oil pump may cause the seal oil flow to the compressor to be low and result in a lower than normal seal oil differential. This can be confirmed by blocking in the discharge of the spare pump and/or the flow measurements made in step 4.

12. Check all spillbacks—The system pressure controller on the lube and/or seal system will normally spillback to the reservoir. If this controller is flowing too much or too little, it can cause either inadequate flow or excessive seal oil differential pressure and cause the compressor seal leakage to increase.

13. Check that gas pressure gauges do not have liquid legs—All gas pressure instrumentation should be located above the shaft centerline of the compressor to prevent liquid build up in the tubing that could affect the readings.
NOMENCLATURE

B = Balance ratio
Cv = Control valve flow coefficient
e = Eccentricity ratio
h = Seal clearance
K = Seal face pressure gradient, dP/dh
L = Seal land length
P = Pressure, psi
Q = Leakage flowrate
r = Radius

Subscripts
b = Balance
i = Inside
o = Outside

Symbols
\mu = Viscosity

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