DESIGN, OPERATION, AND MAINTENANCE CONSIDERATIONS FOR IMPROVED DRY GAS SEAL RELIABILITY IN CENTRIFUGAL COMPRESSORS

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Abstract

The use of dry gas seals in process gas centrifugal compressors has increased dramatically over the last twenty years, replacing traditional oil film seals in most applications. Over 80% of centrifugal gas compressors manufactured today are equipped with dry gas seals. As dry gas seals have gained acceptance with users and centrifugal compressor original equipment manufacturers (OEMs), the operating envelope is continually being redefined.

Ever greater demands are being placed on dry gas seals and their support systems, requiring continual improvements in the design of the dry gas seal environment, both internal and external to the compressor proper.

Contamination is a leading cause of dry gas seal degradation and reduced reliability. This paper will examine the experiences of one centrifugal compressor OEM in this regard. Several potential sources of dry gas seal contamination will be analyzed, drawing from actual field experience, and various means of increasing dry gas seal reliability will be discussed.

Introduction

Centrifugal compressors in process gas service require shaft sealing to prevent the process gas from escaping the compressor case uncontrolled, into the atmosphere. Multi-stage, "beam" style compressors require two seals, one at each end of the shaft (Fig. 1). Single stage, "overhung" style compressors require a single shaft seal, directly behind the impeller. Dry gas seals can be applied to accomplish the required shaft sealing.

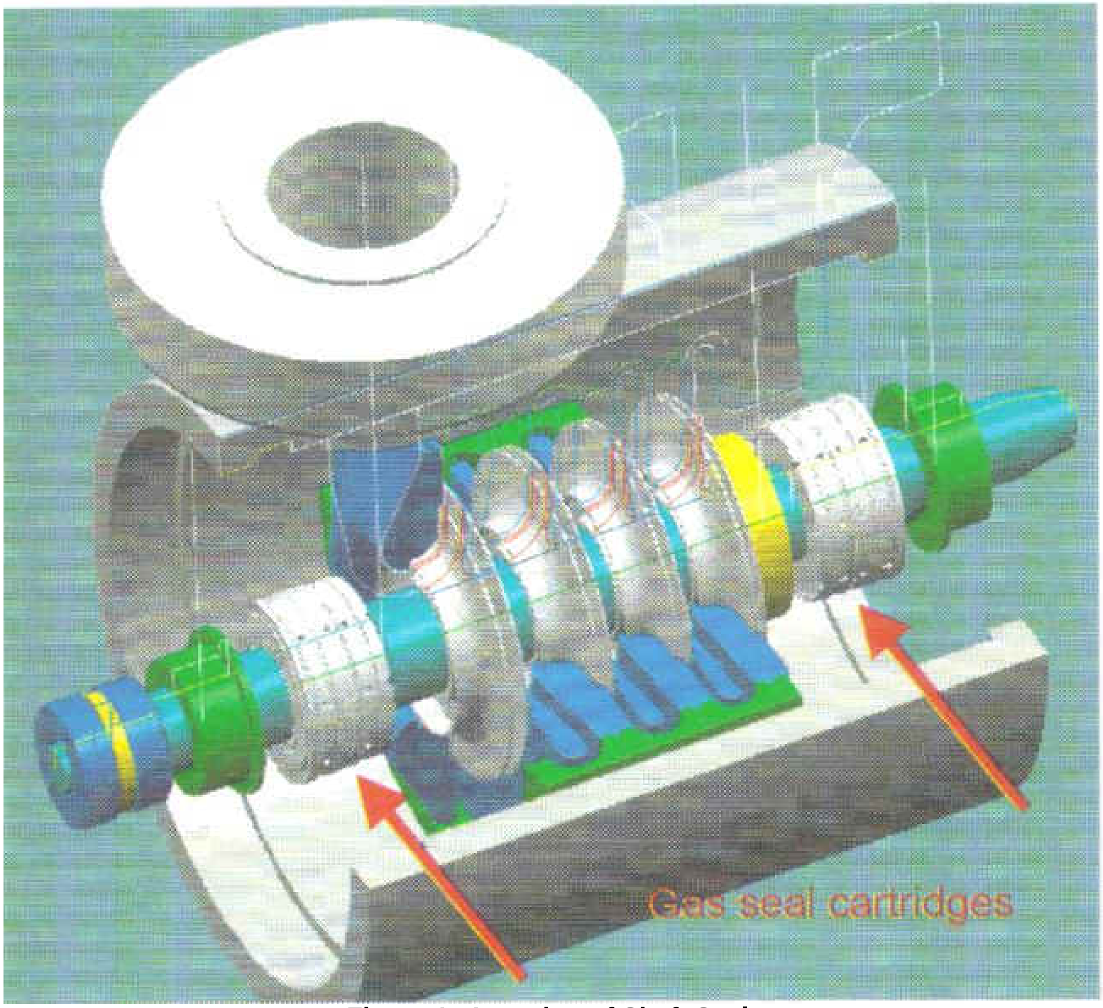

Figure 1 - Location of Shaft Seals

Dry Gas Seals

Dry gas seals are available in a variety of configurations, but the "tandem" style seal is typically applied in process gas service (Fig. 2). Tandem seals consist of a primary seal and a secondary seal, contained within a single cartridge. During normal operation, the primary seal absorbs the total pressure drop to a vent system, and the secondary seal acts as a backup should the primary seal fail.

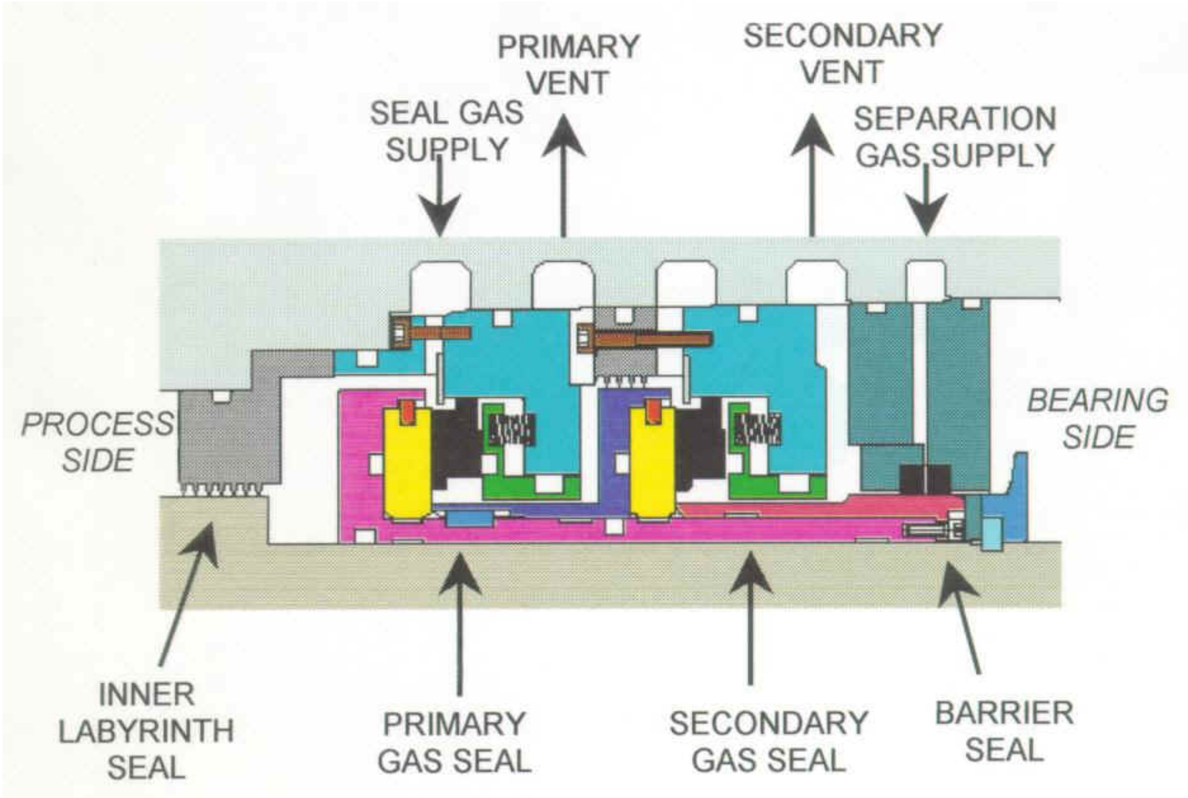

Figure 2 - Typical Tandem Gas Seal / Barrier Seal Configuration

Dry gas seals are basically mechanical face seals, consisting of a mating (rotating) ring and a primary (stationary) ring (Fig. 3). During operation, grooves in the mating ring (Fig. 4) generate a fluid-dynamic force causing the primary ring to separate from the mating ring creating a "running gap" between the two rings. A sealing gas is injected into the seal, providing the working fluid for the running gap and the seal between the atmosphere or flare system and the compressor internal process gas. For a more complete description of dry gas seal principles of operation, see Shah (1988).

Inboard of the dry gas seal is an inner labyrinth seal, which separates the process gas from the gas seal. Outboard of the dry gas seal is a barrier seal, which separates the gas seal from the compressor shaft bearings (Fig. 2).

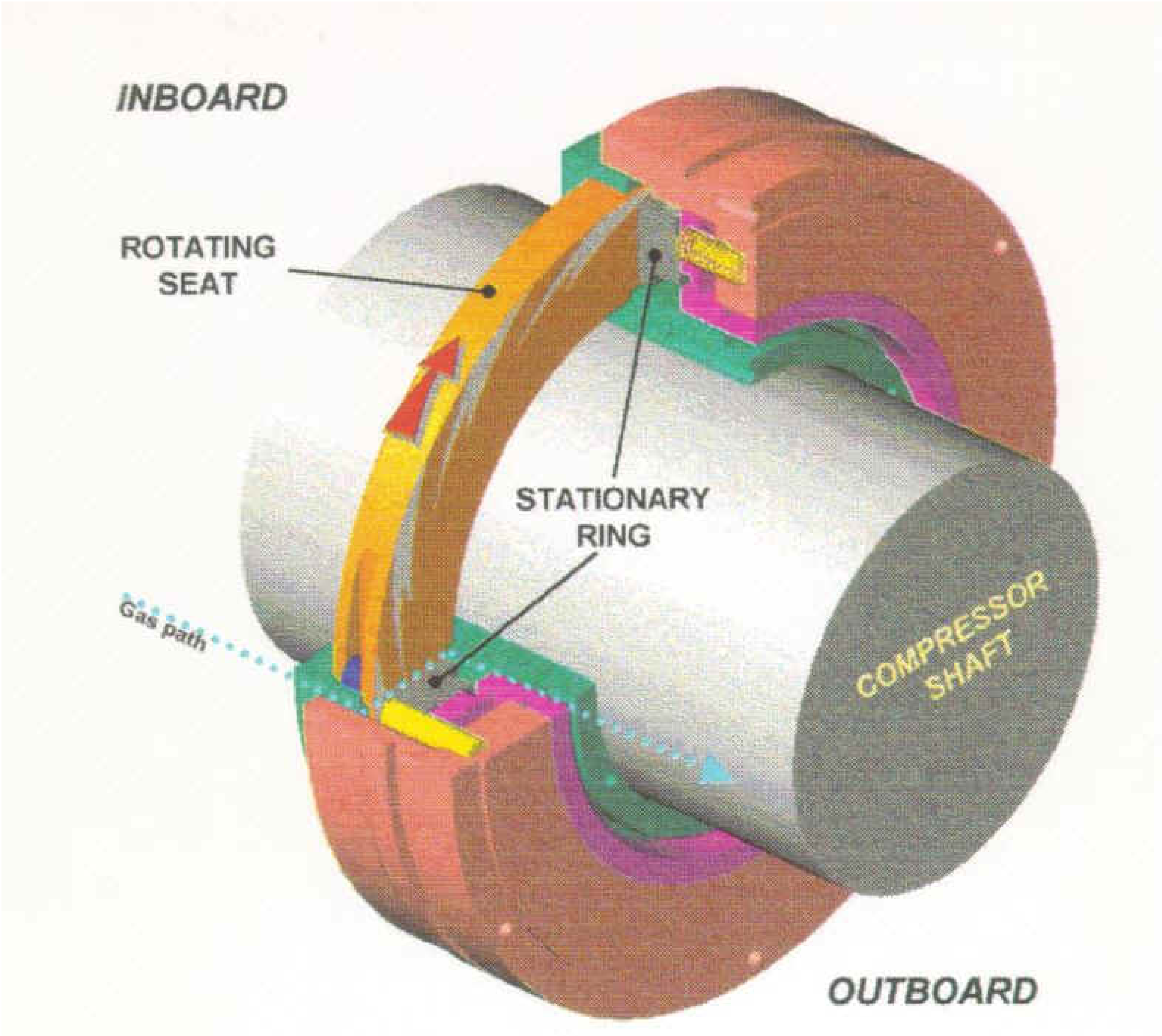

Figure 3 - Gas Seal Components

Figure 4 - Grooves in Seal Mating Ring

Dry Gas Seal Systems
The use of dry gas seals requires a system designed to supply sealing gas to the seal as a sealing and working fluid for the running gap. These gas seal systems are normally supplied by the compressor OEM mounted adjacent to the compressor. There are two basic types of gas seal systems - differential pressure (D P) control and flow control. D P systems (Fig. 5) control the supply of seal gas to the seal by regulating the seal gas pressure to a predetermined value (typically 10 PSI) above a reference (sealing) pressure. This is accomplished through the use of a differential pressure control valve.

Flow control systems (Fig. 6) control the supply of seal gas to the seal by regulating the seal gas flow through an orifice upstream of the seal. This can be accomplished with a simple needle valve, or through the use of a differential pressure control valve monitoring pressures on either side of the orifice.

Since D P systems basically control the flow of seal gas through a labyrinth seal, this can sometimes result in high seal gas flows through the inner seal labyrinth. Variations in the inner seal labyrinth clearance can result in high variations of seal gas flow. The majority of the seal gas flows across the inner seal labyrinth and back into the compressor, and very little flow is actually required for the gas seal. This "recycled" flow is inefficient, and uses more energy. This situation becomes more critical in high pressure applications due to the higher mass flows of gas involved. Since flow control systems control the flow of seal gas through an orifice, the flow rate is constant and does not vary with labyrinth clearance. Flow can control systems can be applied when reduced seal gas flows are desirable.

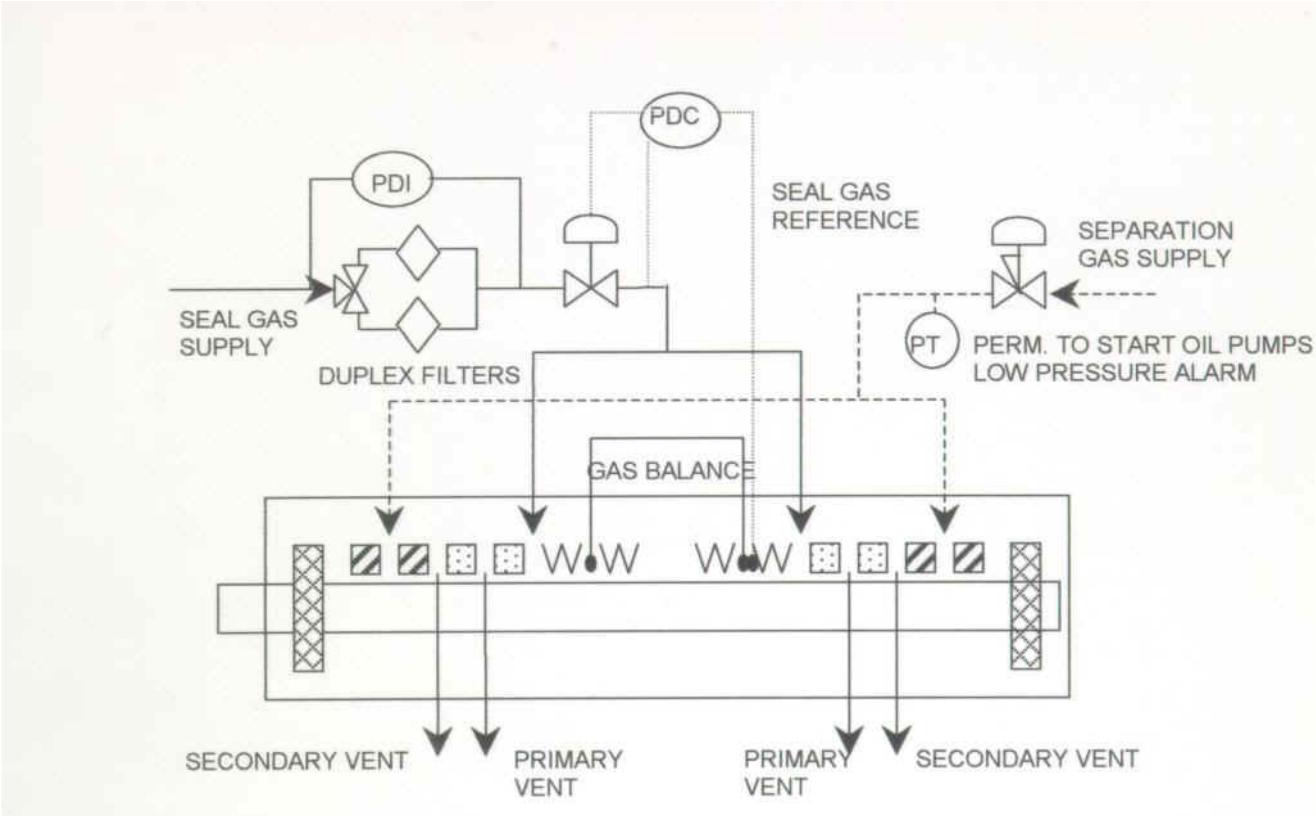

Figure 5 - Differential Pressure Control System

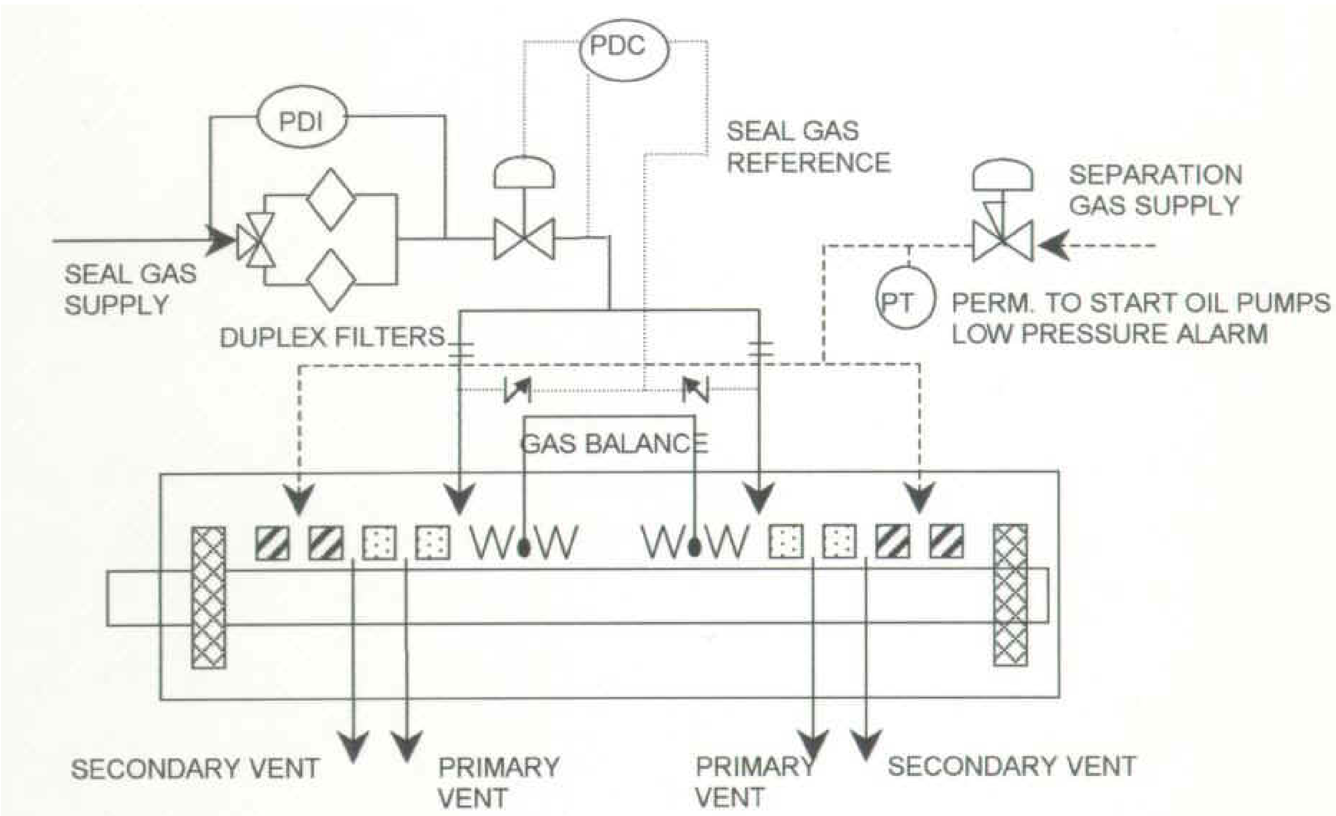

Figure 6 - Flow Control System

GAS SEAL Contamination

The running gap between the primary and mating gas seal rings is typically around 3 to 4 microns, about 1/20th the diameter of a human hair. Ingress of foreign material (solid or liquid) into this very narrow seal running gap can cause degradation of seal performance (excessive gas leakage to the vent) and eventual failure of the seal. Foreign material within the running gap of the seal results in increased shearing forces between the primary and mating rings. The seal components then overheat, leading to some form of mechanical seal failure such as o-ring deterioration, mating ring cracking, etc.).

There are three primary sources of gas seal contamination:

- Process gas (from the inboard or high-pressure side of the seal)
- Bearing lubrication oil (from the outboard or low-pressure side of the seal)
- Seal gas supply itself (injected into the seal)

Contamination from Process Gas

Contamination from process gas can occur when there is insufficient sealing gas pressure to overcome the reference pressure, allowing process gas to come into direct contact with the seal ring faces. Contaminants existing within the process gas can then damage the seal.

Contamination from Bearing Lubrication Oil

A barrier seal is required on the outboard side of the dry gas seal, between the gas seal and the compressor bearing housing area (Fig. 2). This seal is typically buffered with air or nitrogen. The primary function of the barrier seal is to prohibit the flow of bearing lubrication oil or oil mist into the gas seal. Contamination of the dry gas seal from lube oil can occur when the barrier seal fails to function as intended.

Contamination from Seal Gas Supply

Contamination from the seal gas supply occurs when the sealing gas is not properly treated upstream of the dry gas seal. Gas seal manufacturers have stringent requirements for seal gas quality. Typically, the sealing gas must be dry and filtered of particles 3 micron (absolute) and larger. Filters are normally provided in the gas seal system to address this requirement. While typical systems are supplied with coalescing-type filters, such devices may be inadequate depending on the source of seal gas supply.

Measures to improve gas seal reliability

All three types of contamination described previously are influenced by the approach taken in the design of the gas seal environment, the availability of seal gas sources of supply to cover all operating conditions, and the operation and maintenance of the compressor or gas seal system. There are various design, operational, and maintenance techniques that can be applied to mitigate gas seal contamination and increase gas seal reliability / availability.

Compressor Design Considerations

Various features of a compressor's design have an impact on gas seal performance and reliability. Of particular concern is the contamination of the dry gas seal with bearing lubrication oil.

The design of the bearing and seal cavities should consider prevention of lube oil migration into the gas seals. Lube oil will have a natural tendency to migrate along the shaft from the bearing to the barrier seal, requiring the bearing cavity to be designed to minimize the exposure of the barrier seal to lube oil. Flooded bearing drains place greater demands on barrier seals, so it is also imperative to size and locate bearing cavity drain porting and piping to allow for the maximum rate of drainage of lube oil from the cavity. To facilitate drainage of the oil from the shaft, adequate axial shaft spacing and/or oil "slingers" between the bearing and the barrier seal should be considered. It is also important that bearing cavity venting is "free flowing" to avoid the buildup of a backpressure within the bearing housing.

The compressor bearing and seal cavity design should also be such that lube oil migration beyond the barrier seal can be prevented from contaminating the gas seal. Recent field experience involving an offshore installation of high-pressure gas injection compressors with flooded bearing designs has shown that most carbon ring barrier seals cannot totally prevent lube oil migration through the barrier seal (although, as will be discussed later, different barrier seal designs can influence the results). Gas seal system design requires a "secondary vent" be located in the area between the barrier seal and the secondary gas seal (Fig. 5). This secondary vent should be located in the lower half of the seal cavity, allowing a drainage path for lube oil that migrates through the barrier seal. This design requires a low-point drain in the secondary vent piping external to the compressor to prevent blockage of the vent piping with lube oil. Barrier seals will be discussed in more detail later.

Recent experience indicates that special design precautions are warranted for high-pressure back-to-back configuration compressors. Back-to-back designs have two separate sections, each with their own inlet and discharge nozzles (Fig. 7). Back-to-back compressors are common in high-pressure applications. Since the impellers in each section oppose each other, the rotor thrust is balanced, and there is no need for a balance piston. As is the case with most beam-style compressors, the seals are pressure balanced by means of a gas balance line connecting the seal cavities (Fig. 5 and 6). This allows both seals to work against essentially the same sealing pressure. In a back-to-back compressor, the second section inlet end seal is sealing against first section inlet pressure as a result of the gas balance line. This results in a high differential pressure across the inner labyrinth seal, which is located between the reference port and the first impeller of the second section. The high-pressure side of the inner labyrinth seal is equal to second section inlet pressure, while the low-pressure side is essentially at first section inlet pressure, thus the high differential pressure across the labyrinth.

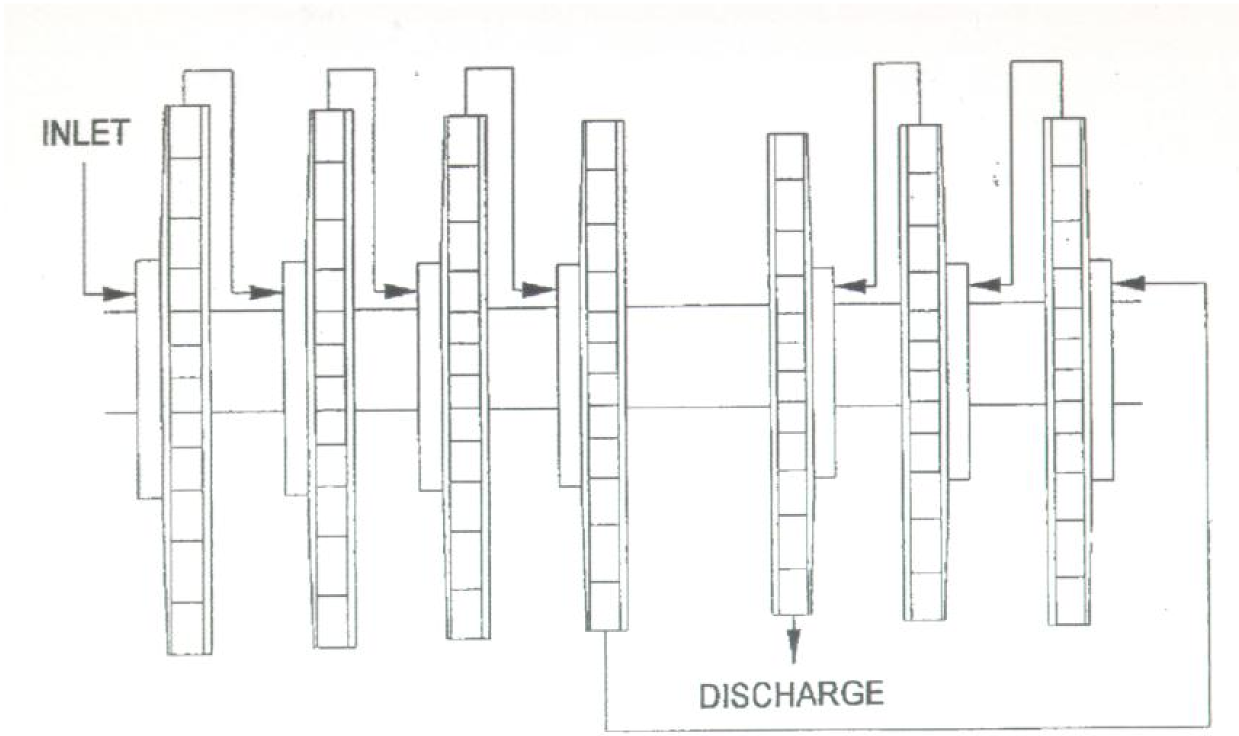

Figure 7 - Back-to-Back Compressor Rotor

The inner labyrinth seal actually consists of two separate sets of labyrinth teeth. The inboard set of teeth forms the seal balance labyrinth, which is subject to the high differential pressure explained previously. The outboard set of labyrinth teeth forms the seal gas labyrinth, which is between the process gas and the sealing gas.

Several gas seal failures were recently experienced on a back-to-back compressor in the Middle East. Seal contamination by process gas appeared to be facilitated by the drive end (second section inlet end) seal cavity design. The initial drive end seal cavity design used a straight inner labyrinth seal, i.e. the seal balance labyrinth and the seal gas labyrinth teeth had equal inside diameters (Fig. 8). Compressor designers theorized that the high differential pressure across the seal balance labyrinth caused the process gas to "jet" across the labyrinth at such high velocities that instead of flowing into the seal reference pressure port, the process gas flowed directly into the gas seal.

The solution to this problem involved more generous sizing of the annulus between the two sets of labyrinth teeth, and the use of a "stepped" inner seal labyrinth. With a stepped inner seal labyrinth, the seal gas labyrinth has a smaller inside diameter than the seal balance labyrinth (Fig. 9). The stepped labyrinth design would encourage the flow of process gas into the seal reference pressure port rather than into the gas seal. This design change was proven to be successful during both high-pressure shop testing and in field operation. This design was employed in the problem compressor in the Middle East but, as will be discussed later, did not fully resolve all of the seal reliability issues.

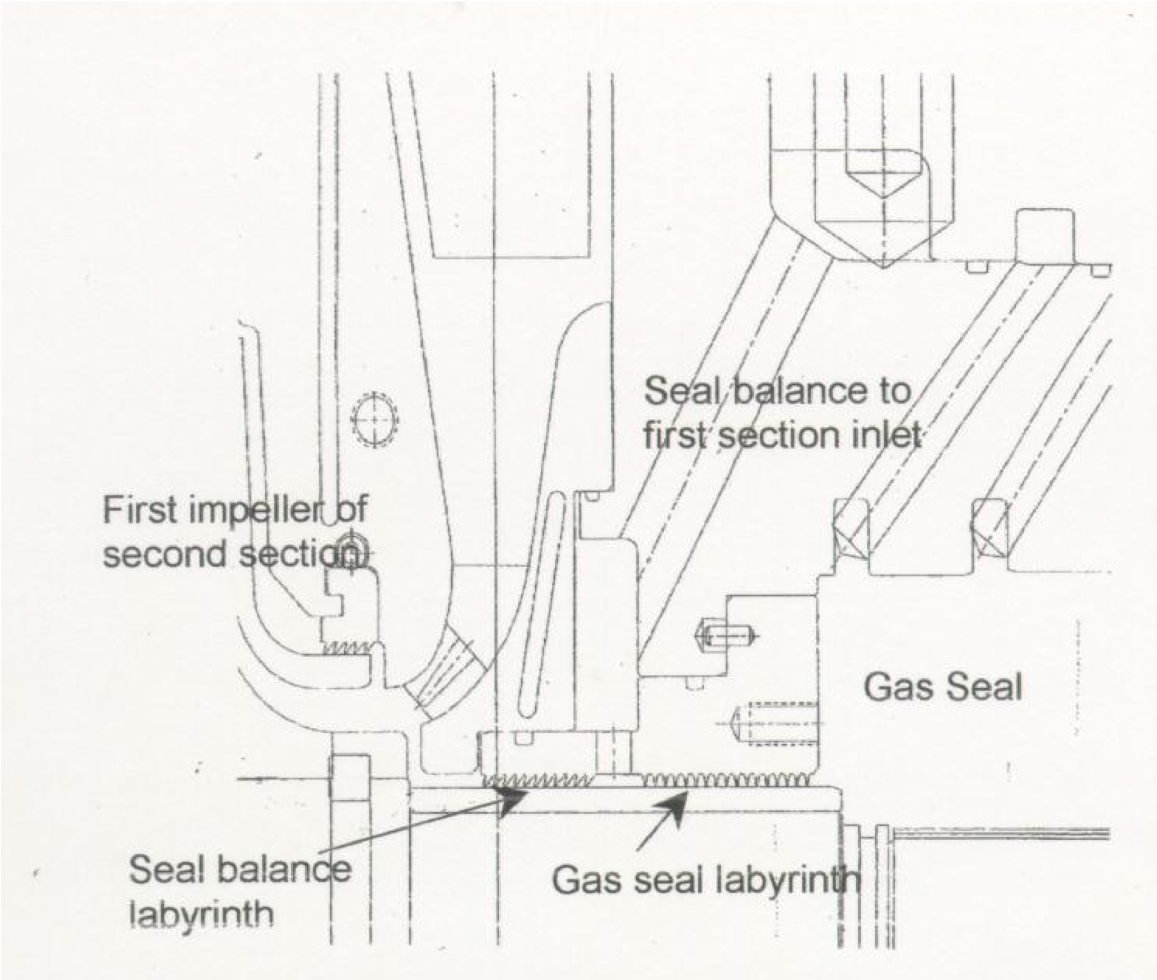

Figure 8 - Back-to-Back Compressor Seal Cavity (Initial Design)

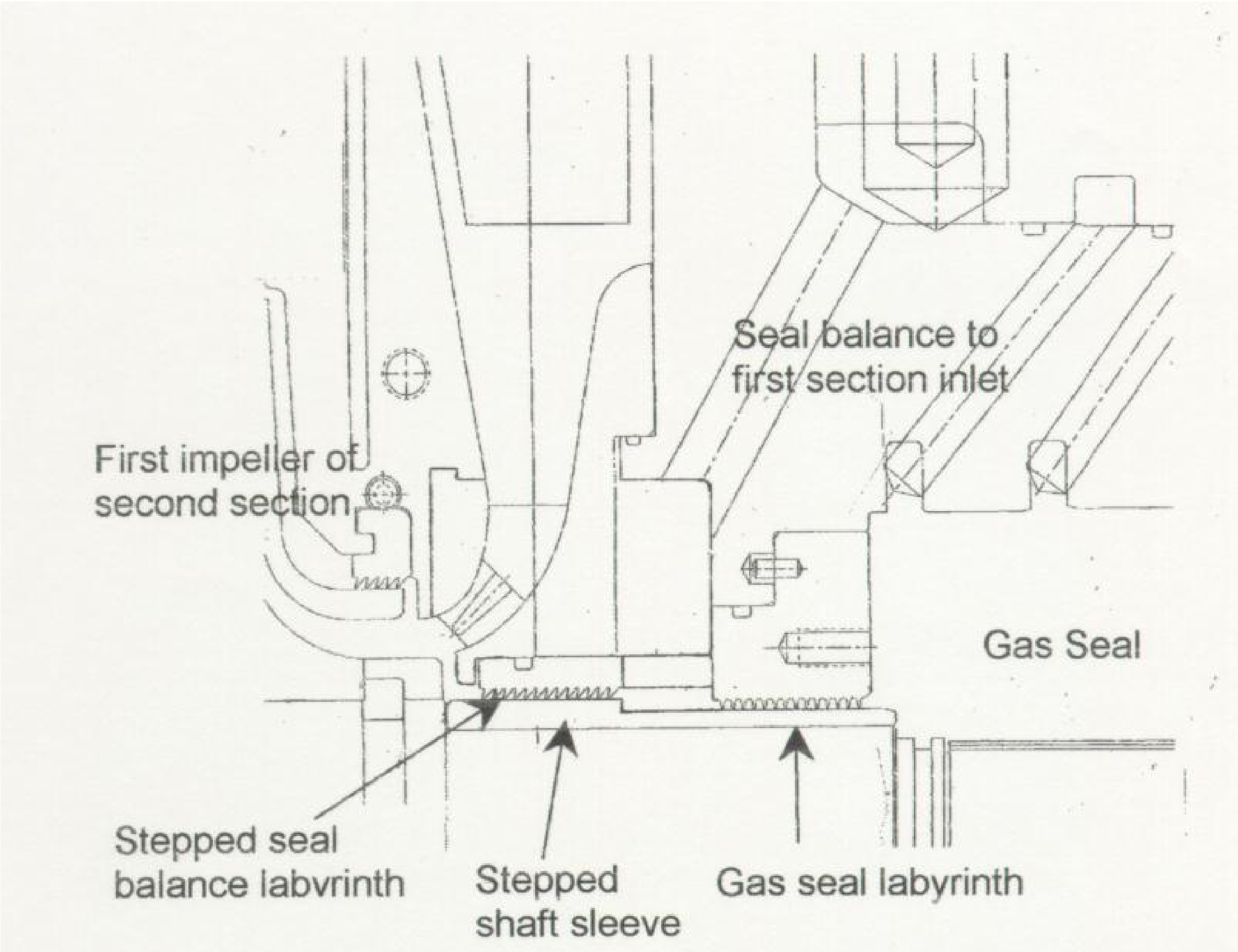

Figure 9 - Back-to-Back Compressor Seal Cavity (Revised Design)

Gas Seal System Design Considerations

Gas seal system design philosophy has an obvious impact on gas seal reliability. The gas seal system defines how the gas seals will be operated and controlled, and therefore significantly impacts reliability of the seals.

Of primary concern is the source of the seal gas supply. It is imperative that the seal gas source be available at sufficient pressure to cover the entire operating range of the compressor. It is very common in the industry to source the seal gas directly from the compressor discharge. On the surface, this may appear to be a very sound practice, since compressor discharge pressure is obviously higher than compressor inlet pressure (normal sealing pressure). However, this design does not address transient conditions such as startup, shutdown, or idle. During these transient conditions, there can be insufficient pressure rise across the compressor to allow for a continuous positive flow of seal gas into the seals. A good rule of thumb is to require a minimum pressure rise of about 50 PSI across the compressor to allow for pressure drops throughout the gas seal system and still have sufficient seal gas pressure at the seals. This may not be the case during some transient conditions. A gas plant in Asia, where several duplicate gas turbine driven compressor trains were operating in parallel, had experienced multiple gas seal failures. Compressor discharge gas was used as the source of seal gas for each of these compressors. An investigation into the operating history of the plant revealed that when one of the compressors was not needed by the process, it was brought down to an idle speed and operated in a "hot standby" mode rather than being shut down. While in hot standby mode, a very low pressure rise existed across the compressor, leading to insufficient seal gas supply pressure. This allowed the process gas to come into direct contact with the gas seals, resulting in contamination of the primary seal and eventual seal failure.

Another concern in gas seal system design is the quality and composition of the seal gas. As mentioned previously, gas seal manufacturers typically require the sealing gas to be dry and filtered of particles 3 micron and larger. Filters are normally provided in the gas seal system to meet this requirement, but "pre-filtering" may be desirable depending on the specific application.

It is important that the composition of the seal gas be well defined at the time of the system design. Of particular concern are "heavy end" hydrocarbons (C6 and higher) and water vapor in the gas. These components will have a tendency to condense as a result of pressure (and therefore temperature) drops throughout the gas seal system.

Components of the gas seal system such as filters, valves, orifices, and the seal faces themselves, will cause seal gas pressure drops during operation. As the seal gas expands across these components, the Joule-Thomson effect will result in a corresponding decrease in the gas temperature. For this reason, the pressure-temperature relationship of the seal gas must be considered. This can be done by simulating the seal gas pressure and temperature drops expected across the various components within the gas seal system. This data can then be plotted on a phase diagram of the seal gas (Fig. 10).

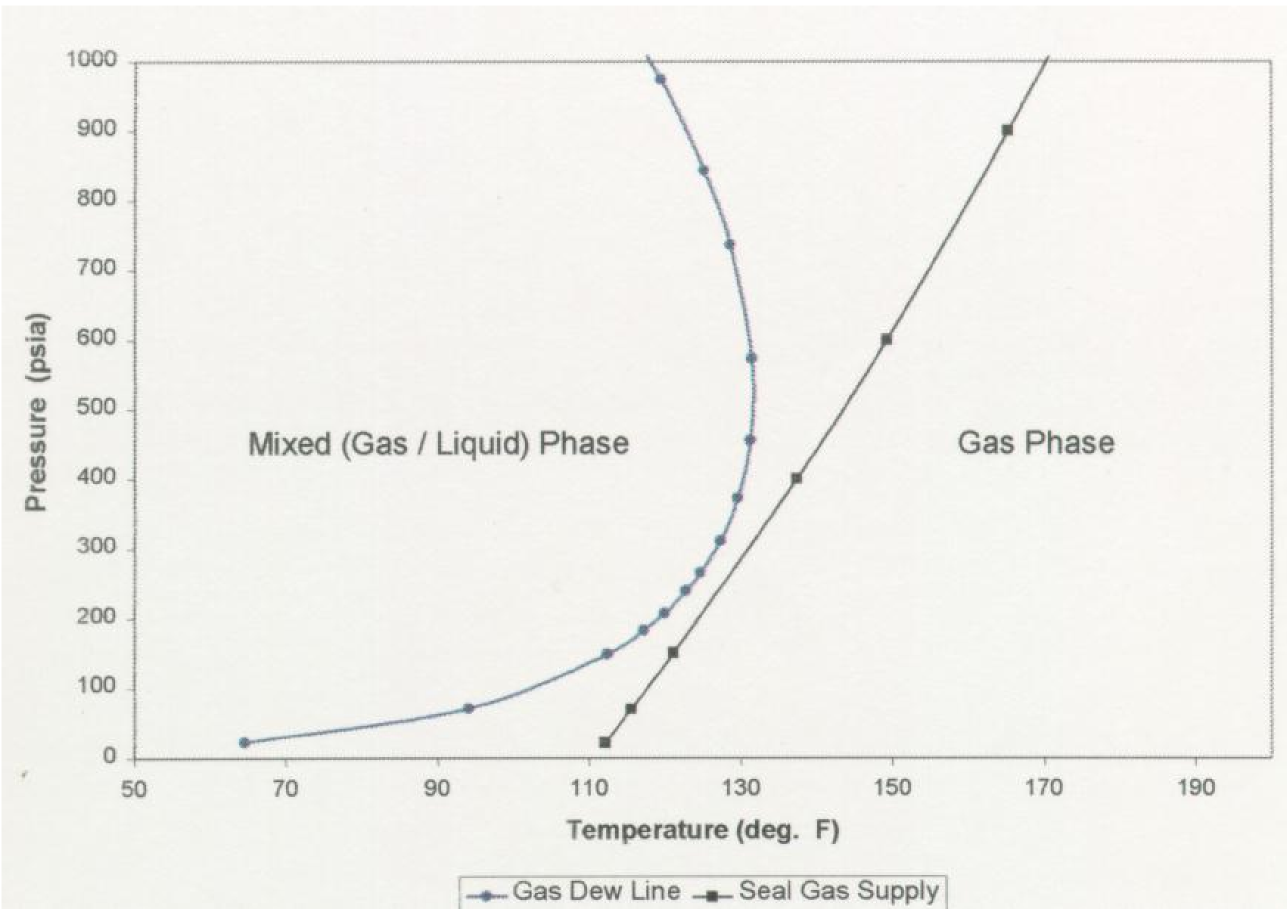

Figure 10 - Typical Phase Diagram

If the results of the gas seal system simulation indicate that the seal gas supply remains in the gaseous phase for all operating conditions, then no further seal protection is warranted. However, if the seal gas passes through a liquid phase, special liquid separation and filtration equipment, and possibly heating of the sealing gas may be required. It is often recommended that the seal gas be heated or otherwise maintained at least 20° F above its dew point. Seal gas lines should be heat traced if ambient temperatures can fall below the dew point of the seal gas.

Returning to the example of the back-to-back compressor in the Middle East, it was later discovered (using the process described above) that liquids were condensing at the second section inlet end inner labyrinth seal due to the Joule-Thomson effect as the gas pressure dropped across this labyrinth. When mixed with the high velocity gas jetting across the labyrinth, the liquid droplets were causing substantial erosion of the labyrinth teeth, which resulted in premature wear of the labyrinth and increased seal gas flow. Problems such as this can be addressed by increasing the gas temperature and/or through the use of alternate labyrinth material.

In order to avoid contamination of the gas seal from bearing lubrication oil, it is necessary to monitor the separation gas supply to the barrier seals. It is highly recommended that the compressor control logic be designed such that separation gas supply is required any time the lubrication oil pumps are operating. This is normally accomplished by requiring a certain separation gas pressure (measured by a pressure switch) before permitting the startup of the lubrication oil pump(s) (Fig. 5 and 6). It is also a normal practice to include an alarm if the separation gas pressure falls below a predetermined value, or ultimately a shutdown if separation supply is not recovered within a given period of time.

Finally, the gas seal system should be designed to handle the possibility of liquid presence within the system. At a minimum, coalescing type filters should be used to treat the seal gas and drains should be provided at all low points in the system piping.

Barrier Seal Considerations

Field experience has proven that not all barrier seal designs are created equal. In the early days of gas seal applications, labyrinth seals were used almost exclusively for the prevention of lube oil migration into the gas seal, and proved to be quite effective. In recent years, labyrinth seals have been replaced by segmented carbon ring seals in most barrier seal applications.

Segmented carbon ring barrier seals (Fig. 11) are available in two designs - shaft contacting and non-contacting. The basic design of the two types of seals is essentially the same - two sets of segmented carbon rings held together by a garter spring around their circumference - with the exception of the clearance between the seal inside diameter and the shaft outside diameter. The contacting barrier seal has zero clearance, riding tight on the shaft. The non-contacting, bushing type barrier seal has a small shaft clearance.

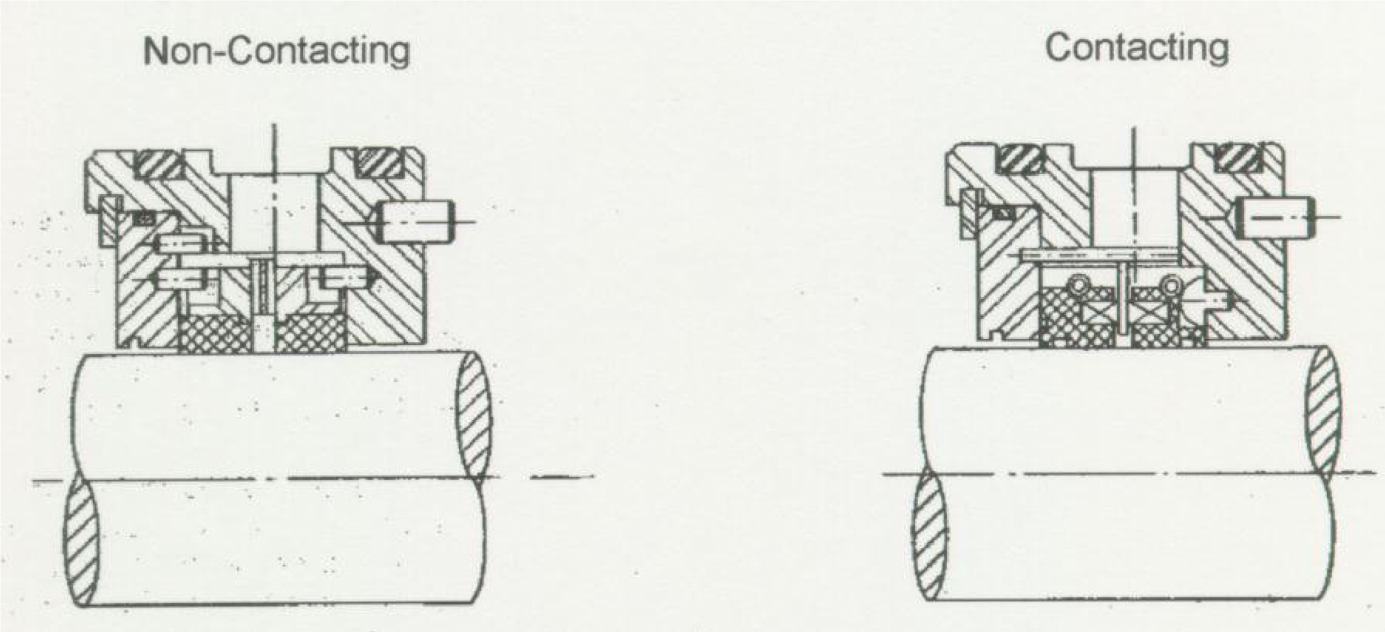

Figure 11 - Segmented Carbon Ring Barrier Seals

Field experience has shown the labyrinth seal to be the most reliable type of barrier seal. Labyrinth seals are also substantially less expensive than segmented carbon ring seals. However, due to the larger shaft clearance associated with labyrinth seals, the amount of injected separation gas required is much higher than the segmented carbon ring seals. Labyrinth seals typically require separation gas flows of about 30 SCFM (for two seals), while segmented carbon ring barrier seals require less than five SCFM.

Field experience with the same offshore high-pressure gas injection compressors mentioned previously has proven contacting (zero shaft clearance) segmented carbon ring barrier seals to be ineffective at preventing lube oil migration into the gas seal under semi-flooded bearing chamber conditions. This conclusion was later verified by shop testing conducted by the barrier seal manufacturer. Similar shop testing conducted by a manufacturer *of non-contacting* segmented carbon ring barrier seals, however, was successful.

While it is not fully clear why the shaft contacting segmented carbon ring barrier seals allowed lube oil migration into the gas seal area, one theory is that, under flooded bearing conditions, the very low separation gas flow is simply not enough to "hold back" the bearing lube oil. It should also be mentioned that shaft contacting segmented carbon ring barrier seals are more susceptible to damage during installation than the non-contacting style. It is not uncommon to damage the inboard edge of the inboard carbon ring seal when installing the seal assembly onto the compressor shaft. Damage usually occurs when the brittle carbon rings are nicked when sliding the seal over the various steps in diameter on the compressor shafts.

Whenever possible, labyrinth barrier seals should be applied for maximum protection of the gas seal from lube oil contamination. When the amount of separation gas is restricted, non-contacting bushing style segmented carbon ring barrier seals offer a reliable alternative.

Operation and Maintenance Considerations

Lastly, but certainly not of least importance, the operation and maintenance of the compressor and gas seal system can also directly influence gas seal reliability. As explained previously with the Asian gas plant example, operation of the compressor at transient conditions greatly increases the opportunity for gas seal contamination. For this reason, the number of starts and stops of the compressor should be held to a minimum. Operation of the compressor at idle conditions of low pressure rise should also be avoided (when using the compressor discharge as the source of seal gas supply). If it is absolutely necessary to run in a hot standby mode like the referenced gas plant, the operating parameters should be set such that enough pressure rise exists across the compressor to allow for a positive flow of seal gas. Another option is to employ another source of seal gas (other than compressor discharge) during the hot standby operation. The availability of such secondary sources of seal gas is highly dependent on the installation location, but some possibilities include gas turbine fuel gas, bottled nitrogen, or process gas from other compressors within the plant. If no such secondary seal gas sources exist, there are seal gas "boosters" on the market which can be used to increase the pressure of seal gas taken from the compressor discharge.

Finally, as with any mechanical system, good maintenance practices should be followed. Seal gas filter elements should be replaced as needed. A differential pressure switch is normally provided to indicate when a filter has become fouled. As discussed previously, a good gas seal system will include drains in the low points of system piping. These drains should be monitored on a regularly scheduled basis and any accumulated should be drained as required. The health of the gas seals themselves should also be monitored. Tracking and trending over time the primary seal vent leakage pressure or flow and/or the pressure differential between the seal gas supply and the seal reference pressure are examples of seal health monitoring methods. Increasing primary seal vent pressure or flow or decreasing seal gas supply to seal reference pressure differential are indicative of a deteriorating gas seal.

Conclusion

Dry gas seals operate under extremely tight tolerances which demands that special care be taken in the design of the gas seal environment, and in the operation of the compressor and gas seal system. While the threat of seal degradation and reduced seal life due to outside influences is real, the detrimental effects of these factors can be minimized. With proper communication between the compressor OEM and the user during the design phase, and with proper training of operators and maintenance personnel, gas seal reliability can be optimized.

References

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