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INTRODUCTION TO DRY GAS SEALS AND SYSTEMS

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ABSTRACT

This tutorial presents an introduction to dry gas seals (DGSs) and their associated systems. DGSs are used as a shaft end seals for compressors and turbines, and they are installed on virtually all centrifugal compressors produced today. A DGS is able to provide very low leakage flow rates due to the small running clearance between the rotating and stationary components. This tutorial discusses the basic principles of DGS operation, terminology, applications, and API Standard 692 design requirements for DGS systems. The last section covers causes of DGS failures and practices to prevent them.

INTRODUCTION

Seals are used to prevent the flow of a fluid from areas of high-pressure to areas of low-pressure, and compressors, turbines, etc., utilize many types of seals throughout the machines. Static seals prevent flow, or leakage, between stationary components, while dynamic seals prevent leakage between components that have relative motion. Dynamic seals can be contacting, where the seal element maintains contact with the moving component, but surface velocity and pressure limit these applications, which contribute to heat generation and wear. Non-contacting dynamic seals maintain a small clearance between the moving and stationary components to prevent solid-solid contact, avoiding wear issues, but some amount of leakage is inevitable.

The prominent example of an important dynamic seal in a turbomachine is the shaft end seal that prevents pressurized process fluid from leaking to atmosphere at the location where the rotating shaft penetrates the casing. The focus of this tutorial will be on the DGS. DGSs have been in use since the 1980s and are installed on the vast majority of new compressors today [1-3]. It is also common for older compressors with oil seals to be retrofitted with DGSs due to the improved performance and lower operational costs [4]. DGS systems were addressed in the 1999 edition of API Standard 614 [5], which also covers other lubrication and seal systems. However, API Standard 692 [6], first released in 2018, is dedicated to the design and application of DGS sealing systems.

BASIC OPERATION

A DGS is a face seal, meaning that the operating clearance between the rotating and stationary elements exists in the axial direction. Figure 1 depicts a cross-section of a DGS, and the rotating and stationary seal faces, or rings, identified as elements 1 and 2, respectively. More detail on the flows in and out of a DGS are presented in a later section, so we simply focus on the basic operating principles here. High-pressure gas exists on the outer diameter of the rings, and leakage flows radially inward to the inner diameter, which is at a lower pressure. Grooves exist on the rotating ring to help generate hydrodynamic lift force to separate the stationary ring from the rotating ring. The grooves start at the outer diameter and do not communicate to the inner diameter, creating a pressure dam effect as the gas flows inward due to pressure and the scooping action of the rotating ring. As shown in Figure 2, the groove geometry can be unidirectional, meaning that the seal will only operate as intended if the shaft is rotating in the correct direction, or the geometry can be bidirectional to prevent seal damage if the shaft happens to rotate backwards (e.g., compressor back-flow scenario). Other groove geometries exist beyond those shown in Figure 2, and their three-dimensional details (i.e., including variable depth) are highly engineered for optimal performance. In general, unidirectional grooves generate larger lift force, film thickness, and stiffness with lower lift-off speed than bidirectional grooves. In addition to reverse-rotation tolerance, bidirectional grooves also allow users to have the same DGS part number on either side of a machine, reducing spare inventory requirements.

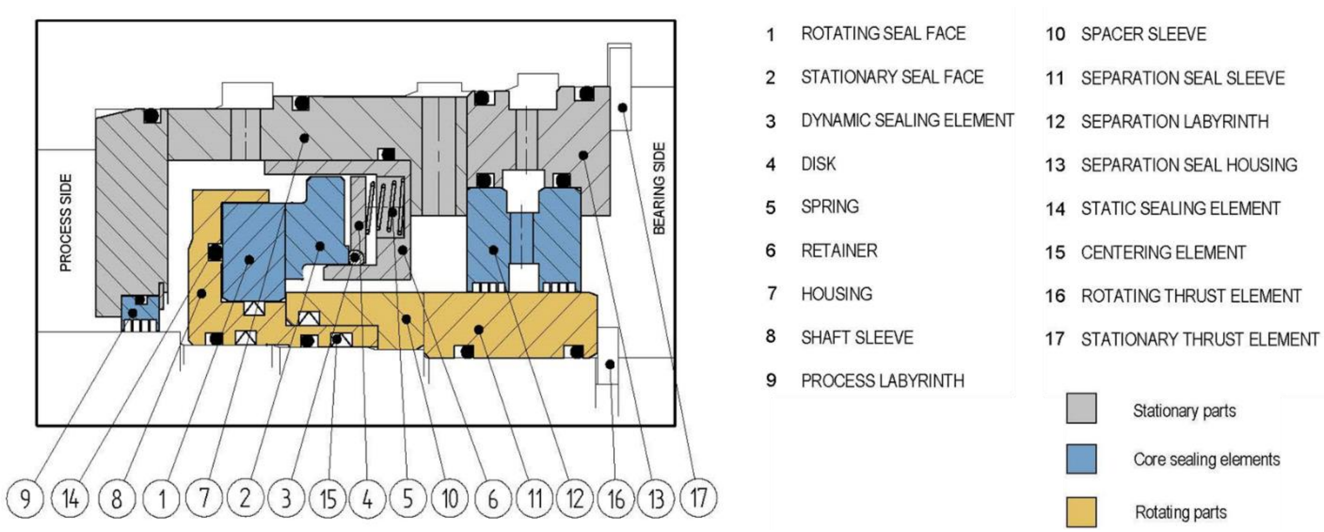

Figure 1. DGS Nomenclature for a Single Seal [6]

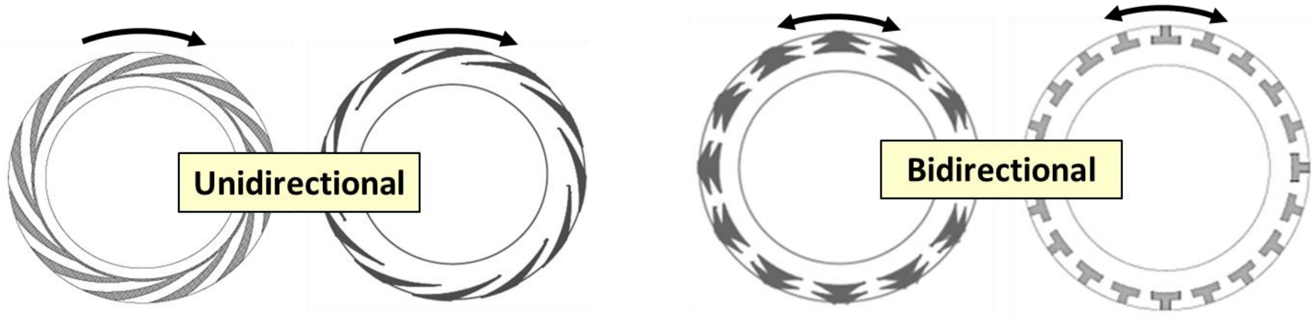

Figure 2. DGS Groove Patterns on Rotating Rings (images from [7])

An illustration of the pressure dam operating principle and the physical scale of the grooves is shown in Figure 3. Typical operating clearances for a DGS are 3-5 micron (0.1-0.2 mil), and a typical groove depth is 10 micron (0.4 mil) – this is about ten times smaller than the thickness of a human hair. The pressure distribution within the film or clearance region is represented in Figure 4. Peak pressure exists at the inner diameter of the grooves. The pressure force in the film region acts equally on the rotating ring and the stationary ring, but in opposite direction. The thrust force acting on the rotating ring must be reacted through the shaft. The hydrodynamic force on the stationary ring (opening force) is balanced by the static pressure force on the back side (closing force)[1]. This is illustrated in Figure 4, though it is greatly simplified. The extent of the closing force pressure is dictated by the diameter of the "dynamic sealing element" (see Figure 1), which is designed to generate the desired film thickness or clearance for the application. Note that the hydrodynamic opening force is a strong function of the clearance (approximately inversely proportional to clearance squared), while the closing force is essentially constant (Figure 4). The slope of the opening force vs. clearance curve represents the film stiffness (i.e., $k_{\text{film}} = -\partial F_{\text{open}}/\partial c$). An important feature of the DGS is that the stationary ring is permitted to move axially, and the pressure forces always act to maintain the pressure balanced condition. This provides the necessary characteristic that the stationary ring can follow axial excursions of the rotor (e.g., vibrations, thermal growth) and maintain the desired operating clearance.

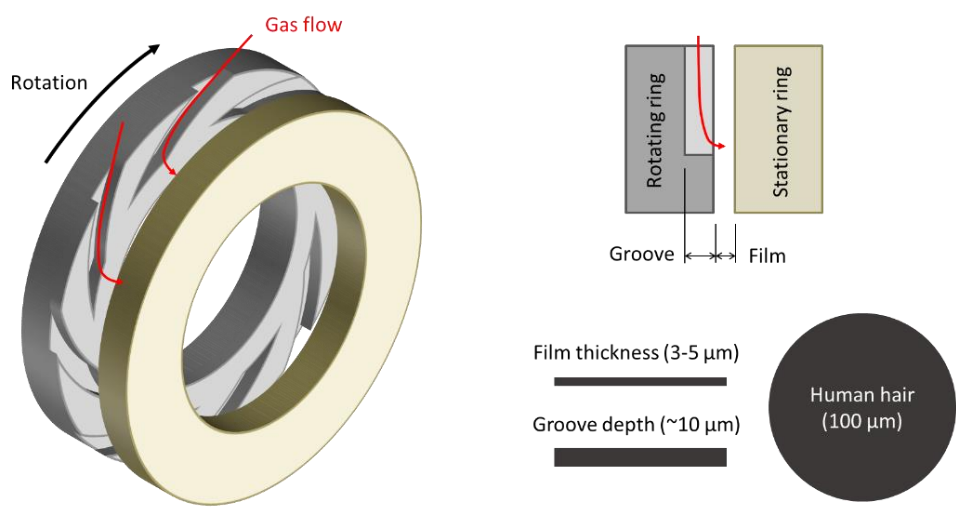

Figure 3. Grooves in Rotating Ring Create a Pressure Dam that Generates Hydrodynamic Lift

1 There is also a spring force acting to "close" the seal clearance (see Figure 1), but this is negligible compared to the magnitude of the pressure force in most cases.

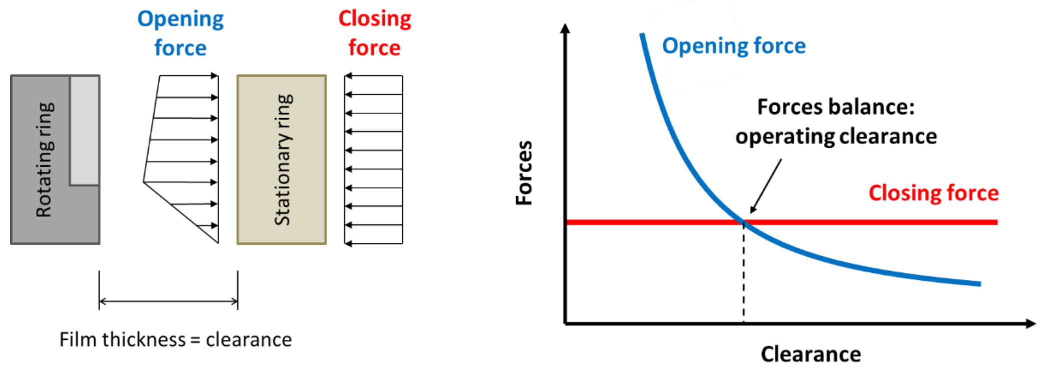

Figure 4. Illustration of Pressure Forces Acting on the Stationary Ring

Ideally, operating clearances would be as small as possible to reduce leakage. However, practical limitations, including manufacturing tolerance and dynamic operating effects, limit typical operating clearances 3-5 micron (0.1-0.2 mil). Mass flow rates through the DGS faces are approximately proportional to $c^3(p_1^2 - p_2^2)/\ln(R_1/R_2)$, where c is the axial clearance, p_1 is the high pressure at the outer radius R_1, and p_2 is the low pressure at the inner radius R_2. Typical DGS leakage is around 28-425 slpm (1-15 scfm), depending on seal size and pressure conditions [1, 4]. For a sense of relative magnitude, a 90 mm (3.5 in) diameter labyrinth seal with a radial clearance of 0.001 times the diameter would leak over 7,200 slpm (250 scfm) of methane with a 3.5 MPa (500 psi) pressure differential.

SEAL ARRANGEMENTS

Single Seal

The simplest DGS arrangement is the single seal (Figure 5). Seal gas enters between the process seal and the DGS, where the majority of the flow leaks to the process side. The rest of the flow leaks past the DGS seal faces and flows out of the vent along with part of the separation gas supply. The seal gas is conditioned process gas cleaned of contaminants and is at slightly higher pressure than the inboard process. This prevents the process gas in the compressor, which could contain solid or liquid contamination, from flowing into the DGS. Since the seal gas goes to the vent, which typically goes to atmosphere, single seals are commonly used in compressors for nonflammable gases that are not harmful to the environment (e.g., nitrogen or air). Compared to the other arrangements described below, the single seal is the lowest cost, has the smallest overall dimensions, and has the smallest added mass to the rotor.

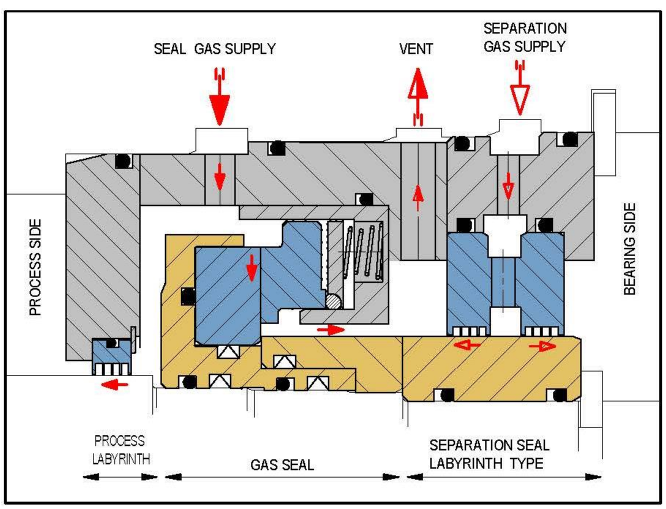

Figure 5. Single Seal Schematic [6]

Double Seal

Figure 6 shows a double seal, which has two sets of seal faces oriented in opposite directions, like two single seals in a back-to-back arrangement. The supply gas is introduced between the primary (inboard) and secondary (outboard) DGS. On the inboard side, seal gas and buffer gas leak into the process. On the outboard side, supply gas and separation gas are vented to atmosphere. In case of primary seal failure, the secondary seal prevents uncontrolled leakage of the process to atmosphere. Double seals typically use nitrogen (or other inert gas) for seal gas so that no process gas leaks to atmosphere. This requires that nitrogen is permitted to mix with the process gas.

Due to low seal gas consumption (leakage through the primary and secondary DGS faces), double seals have less inherent self-cooling capacity, and sealing pressures tend to be lower than the other arrangements.

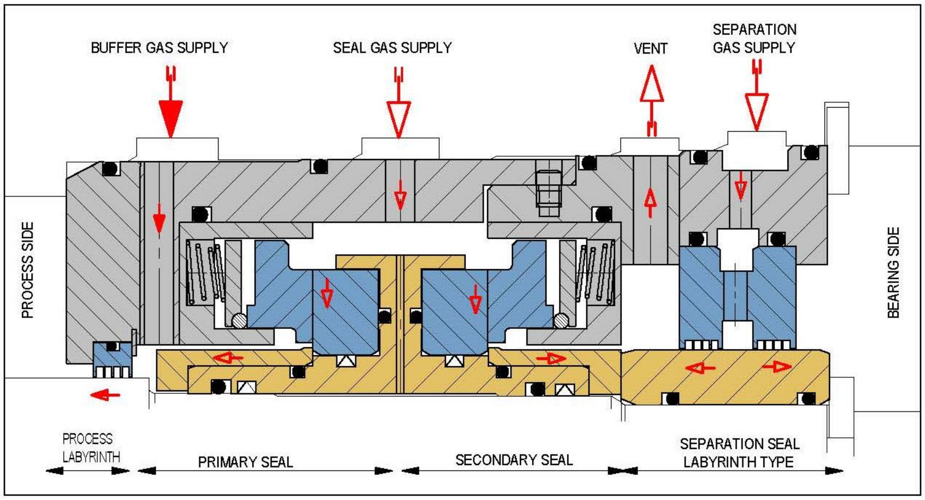

Figure 6. Double Seal Schematic [6]

Tandem Seal

Figure 7 shows a tandem seal arrangement, which is effectively two single seals in series. The inboard DGS face is the primary seal, which operates in the same way as the single seal described above. Part of the primary seal gas goes to a vent system, and the remainder leaks through the secondary seal. The secondary seal normally sees significantly lower pressure than the primary seal, but it is designed to be a redundant seal for the full pressure in case the primary seal fails. The seal gas that leaks through the secondary seal is vented with the separation gas to atmosphere. Lower pressure leads to an overall reduction in process gas leakage to atmosphere.

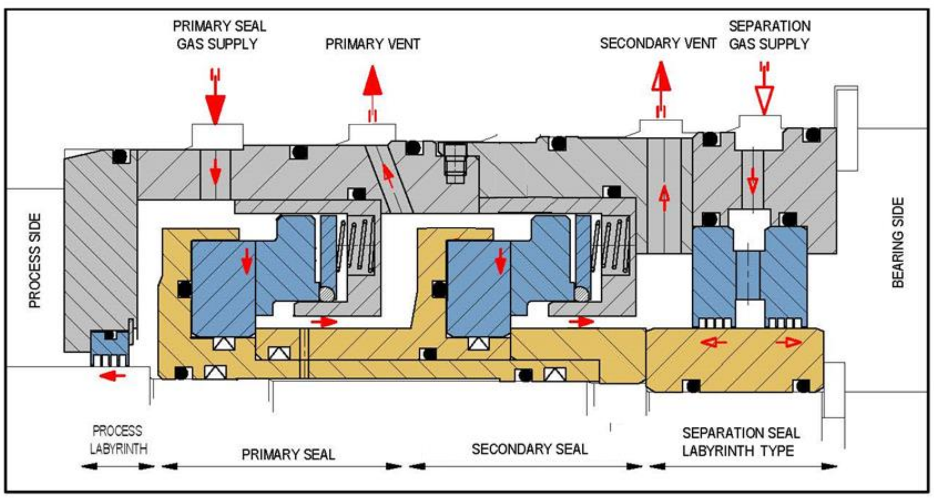

Figure 7. Tandem Seal Schematic [6]

Tandem Seal with Intermediate Labyrinth

Figure 8 shows the tandem seal with an intermediate labyrinth. This is similar to the tandem seal arrangement with the addition of a labyrinth seal between the primary and secondary seals and a secondary seal supply gas between the intermediate labyrinth and the secondary seal. This prevents primary seal gas from reaching the secondary seal. Most primary seal gas flows into the process, and the rest is diluted with secondary seal gas and passes through the primary vent. Secondary seal gas that leaks through the secondary seal vents to atmosphere with separation gas.

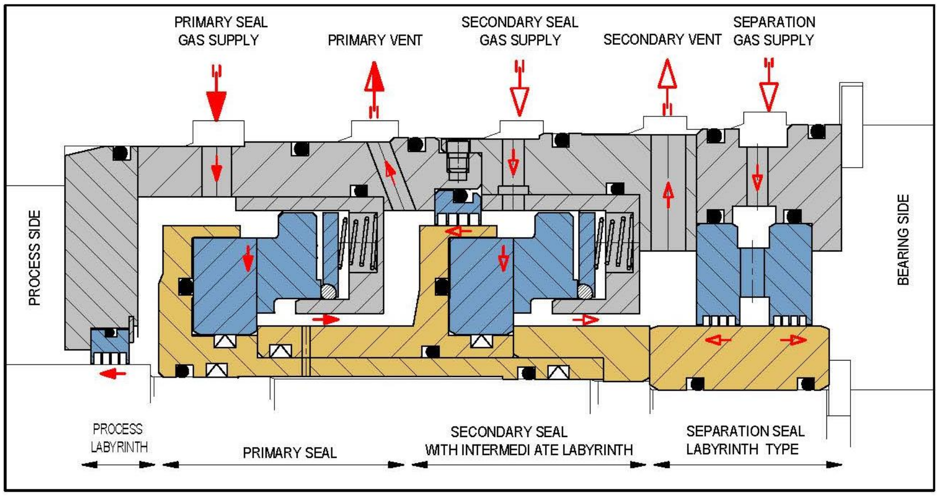

Figure 8. Tandem Seal with Intermediate Labyrinth Schematic [6]

PROCESS AND SEPARATION SEALS

Process Seal

A process seal is incorporated inboard of all of the DGS arrangements shown in Figures 5-8. Seal gas is supplied at a higher pressure than the process to ensure that process gas, which may contain solid or liquid contaminants, cannot flow into the DGS faces and lead to failure. The process seal serves as a flow restriction between the seal gas supply and the process to limit the amount of flow that leaks into the process. Since this is a parasitic loss on the system, the leakage through the process seal should be minimized. In the case of DGS failure, the process seal can also restrict the flow of process gas to atmosphere.

Figure 9 shows different seals types that are used as process seals. The simplest design is the labyrinth seal. Labyrinth seal teeth can be on the stator or rotor, and clearance usually exceeds the bearing clearance and is designed to avoid contact during normal operation. For tooth on rotor labyrinths, an abradable material can be used to allow a smaller clearance (lower leakage) and be tolerant of minor rubbing. For either type of labyrinth, the rotor is usually steel, while the stator can be a non-metallic material, such as PTFE or engineered thermoplastics. The non-contacting bushing, comprising one or more segmented carbon rings and held together with a garter spring, can float radially in the housing to follow shaft motion, though it is constrained to prevent rotation. As a result, these seals can have smaller radial clearance than the labyrinth (reduced gas consumption), though they are more complex and costly. Labyrinth seal clearance (and leakage) is typically fairly constant at different temperature operating conditions, but this is not the case with the non-contacting bushing due to the greater difference in thermal growth coefficient and smaller nominal clearance.

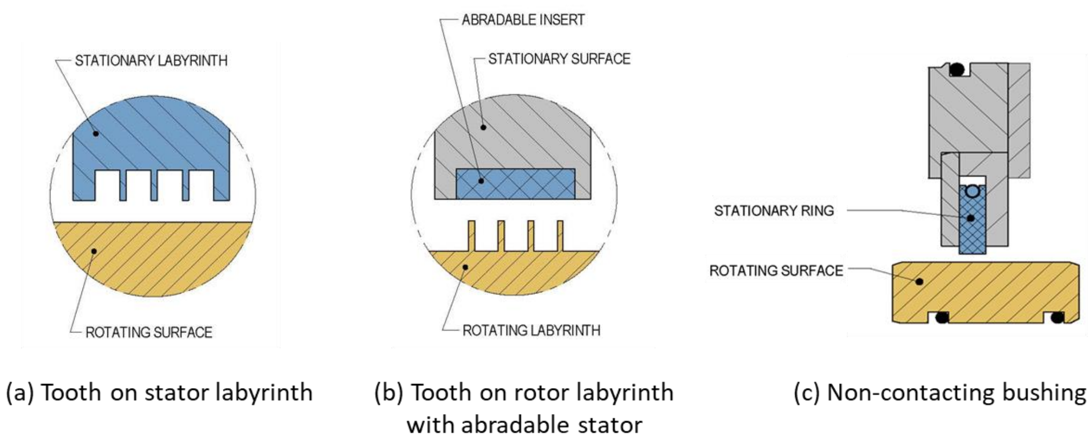

Figure 9. Types of Process Seals [6]

Separation Seal

A separation seal is incorporated outboard of all of the DGS arrangements shown in Figures 5-8, and it serves to prevent bearing oil from contaminating the dry gas seal. Separation or buffer gas, typically nitrogen or air, is supplied at a higher pressure than the outboard vent of the DGS and the bearing cavity to ensure bearing oil does not leak into the DGS and DGS gas does not leak into the bearing.

Separation gas exits the machine via the DGS and bearing vents. Figure 10 shows different types of separation seals, which are similar to the process seals described above. In addition to the non-contacting bushing, there is also a contacting bushing (Figure 10c), which is also composed of segmented carbon rings. This variant maintains rotor-stator contact at all operating conditions, so it has the lowest gas consumption rate. However, this means that this seal wears at a faster rate than the non-contacting bushing – typical life is 3-5 years.

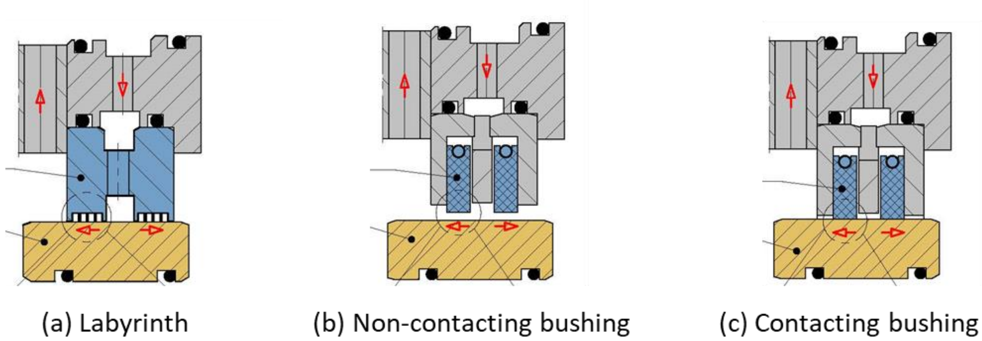

Figure 10. Types of Separation Seals [6]

SEAL MATERIALS

A DGS is constructed from numerous components and various materials. This section briefly discusses material selection for a few: the rotating and stationary rings, housings, and secondary seals. The rotating and stationary rings contact each other during starts and stops, so material selection should resist abrasion. Typically, hard-soft material combinations are chosen, with the stationary ring being the softer of the two rings. Carbon graphite is commonly chosen for this purpose in medium to lower pressure applications. Due to lower relative material elastic modulus, a carbon graphite stationary ring can conform to the stronger rotating ring to ensure good performance. For high-pressure applications, the rotating ring can be made from silicon carbide with a hard coating, such as diamond-like coating (DLC), to improve tribological performance.

The rotating ring has two key requirements that are different than the stationary ring, namely that the rotating ring contains the grooves and that it has to withstand the centrifugal loading from high speed operation. Tungsten carbide, silicon carbide, and silicon nitride are typical materials for the rotating ring. Less commonly, alloy steel can be used if higher fracture toughness is required. Material selection depends on several criteria. High hardness is desired for abrasion resistance from intermittent ring contact during starts and stops. High material elastic modulus is required to resist deformation of the grooves, particularly for high-pressure applications. Due to centrifugal loading, high material strength-to-weight ratio is important for high-speed applications. High thermal conductivity and low thermal expansion coefficient are desirable to mitigate thermal distortion due to viscous heat generation. The performance of different rotating ring materials relative to the properties of alloy steel is summarized in Figure 11. Table 1 and Table 2 highlight seal material options based on operating pressure and surface speed.

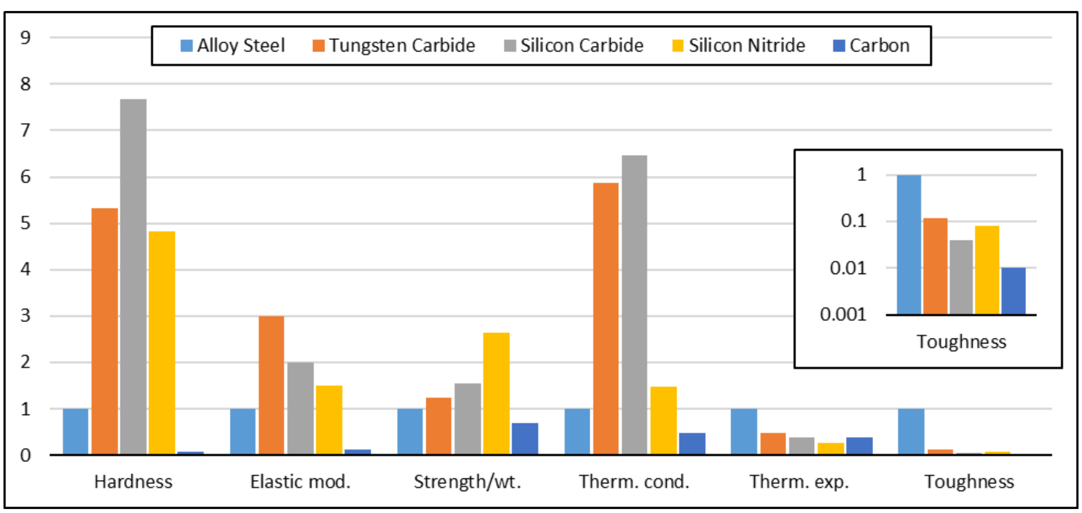

Figure 11. Ring Material Properties Relative to Alloy Steel

Typical housing materials and materials for other structural metallic components can range from stainless steels to nickel alloys, depending on service requirements. Materials must be corrosion resistant to limit potential particles and debris that can form inside the seals and potentially lead to damaging the rotating and stationary rings. 410 stainless steel (SS) is a common choice because it is economical and easy to machine. It also has higher strength than 316 SS and similar thermal growth to many nickel-based materials and alloy steels. 17-4 PH SS is a common choice if higher strength is required for higher-pressure applications. Hastelloy is an option for extreme chemical resistance capability, and other nickel alloys can be used for applications requiring particular chemical resistance or higher temperature service.

Table 1. Common DGS Material Selections as Function of Sealing Pressure

Pressure	Stationary Ring	Rotating Ring	Secondary Seals	Housings
< 10 MPa	Carbon	Silicon Carbide	O-Rings	410 SS
10-18 MPa	Carbon	Silicon Carbide	PTFE	410 SS
> 18 MPa	Silicon Carbide	Tungsten / Silicon Carbide	PTFE w/ backup ring	17-4 PH SS

Table 2. Common DGS Material Selection as Function of Speed

Speed	Stationary Ring	Rotating Ring	Housings
< 100 m/s	Carbon	Tungsten / Silicon Carbide	410 SS
100-120 m/s	Silicon Carbide	Tungsten / Silicon Carbide	410 SS
> 120 m/s	Higher grades of Silicon Carbide, such a Liquid Phase SiC	Tungsten / Silicon Carbide	17-4 PH SS

The secondary seals, and the dynamic seal in particular, also have multiple possibilities for material selection. Exact material choice depends on temperature, pressure, and chemical compatibility, but O-rings with V90 shore hardness and PTFE are common, though other elastomers and polymers can also be used. These materials have high hardness that limit extrusion and prevent explosive decompression. O-rings are typical for lower pressures (separation gas), while pressure and spring-energized PTFE sealing rings are used for higher pressure (process gas). Elastomer (O-rings) temperature limits are highlighted in Figure 12 and PTFE temperature limits are shown in Figure 13. For most seal configurations, it is the O-rings that will dictate the max operating temperature of the seal cartridge, not the housing or ring materials. To design seals for higher temperature operation (>600ºF), further research and development is required for secondary seals, primarily the spring energized Polymer seals in the Process gas section of the seal.

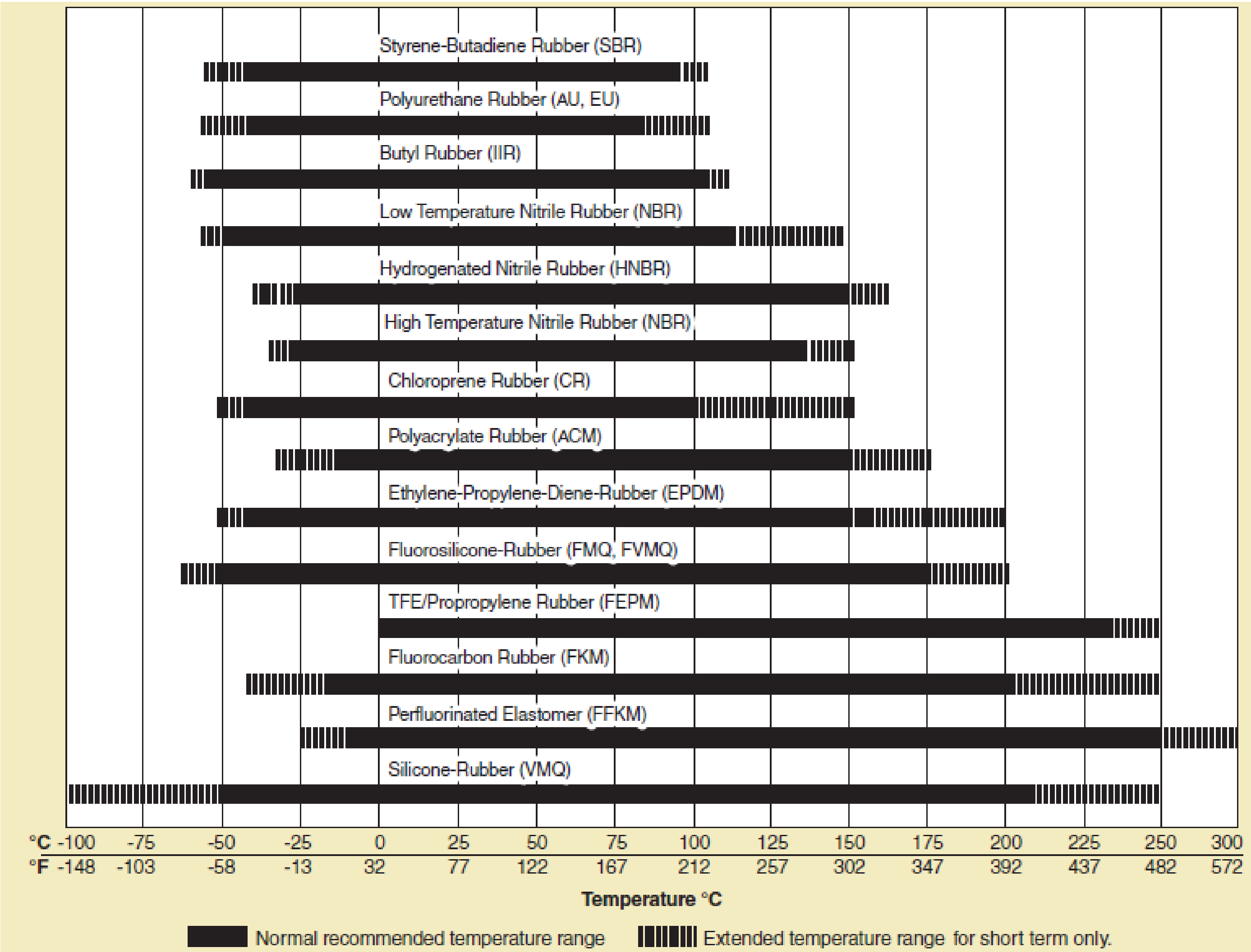

Figure 12. Temperature Range for Common Elastomeric Materials [8]

Parker Material Code	Material	Color	Typical Applications & Description	Service Temperature Range (°F)	Tensile Strength in psi at Break	Elongation in %	Hardness-Shore D
0100	Virgin PTFE	White	Excellent for cryogenic applications. Good for gases.	-425 to +450	4575	400	60
0102	Modified PTFE	Turquoise	Lower creep, reduced permeability and good wear resistance.	-320 to +450	4600	390	60
0203	Fiberglass Filled PTFE	Gold	Excellent compressive strength and good wear resistance.	-200 to +575	3480	190	67
0204	Fiberglass & Moly Filled PTFE	Gray	Excellent for extreme conditions such as high pressure & temperature and for longer wear life on hardened dynamic surfaces.	-200 to +575	3100	245	62
0307	Carbon-Graphite Filled PTFE	Black	Excellent wear resistance and reduced creep.	-250 to +575	2250	100	64
0301	Graphite Filled PTFE	Black	Excellent for corrosive service. Low abrasion to soft shafts. Good in unlubricated service.	-250 to +550	3200	260	60
0502	Carbon Fiber Filled PTFE	Brown	Good for strong alkali and hydrofluoric acid. Good in water service.	-200 to +550	3200	150	60
0120	Mineral Filled PTFE	White	Excellent low abrasion to soft surfaces & improved upper temperature performance.	-250 to +550	4070	270	65
0601	Aromatic Polyester Filled PTFE	Tan	Excellent high temperature capabilities & excellent wear resistance.	-250 to +550	2500	200	61
0405	Stainless Steel Filled PTFE	Gray	Excellent extrusion resistance at high temperatures and pressures.	-250 to +600	2200	190	72
0913	Hytrel®* Unlubricated Thermoplastic Elastomer	Natural	Excellent in gases and most hydraulic fluids. Good abrasion resistance with high wear properties.	-80 to +275	5800	500	55
0901	UHMW Polyethylene	Translucent	High wearing plastic for use in abrasive medias. Excellent in water-based medias, but restricted chemical and heat resistance.	-320 to +200	6000	325	67
0615	Proprietary Low Wear PTFE	Purple	Excellent low wearing material. Kind to soft mating surfaces in the Rb range.	-250 to +550	3470	200	63
0127	Mineral Filled PTFE — FDA compliant for rotary applications	White	FDA compliant materials for sanitary food and pharmaceutical processing.	-250 to +550	2800	250	66
0128	Mineral Filled PTFE — Antimicrobial	White	FDA material with an antimicrobial agent added to prevent bacterial growth.	-250 to +550	2800	250	66

Figure 13. Standard PTFE Seal Materials with Temperature Limits [9]

SEAL GAS SYSTEM COMPONENTS AND REQUIREMENTS

In addition to the seal itself, one of the most critical aspects of successfully running with DGSs is the seal gas supply system. This system is required to supply "dry" gas to the seals at specified pressures and temperatures to ensure proper buffering to prevent contamination. During operation, a stream from compressor discharge (or turbine inlet) supplies the DGSs with pressure that exceeds process sealing pressure. Almost all shaft end seals will see compressor suction (or turbine exhaust) to reduce leakage out of the system while also reducing blow out force on the external casing. Figure 14 shows a representative supply system schematic, for a tandem seal with intermediate labyrinth, whose elements will be discussed in the following sections. API 692 Part 3 specifies seal gas conditioning requirements.

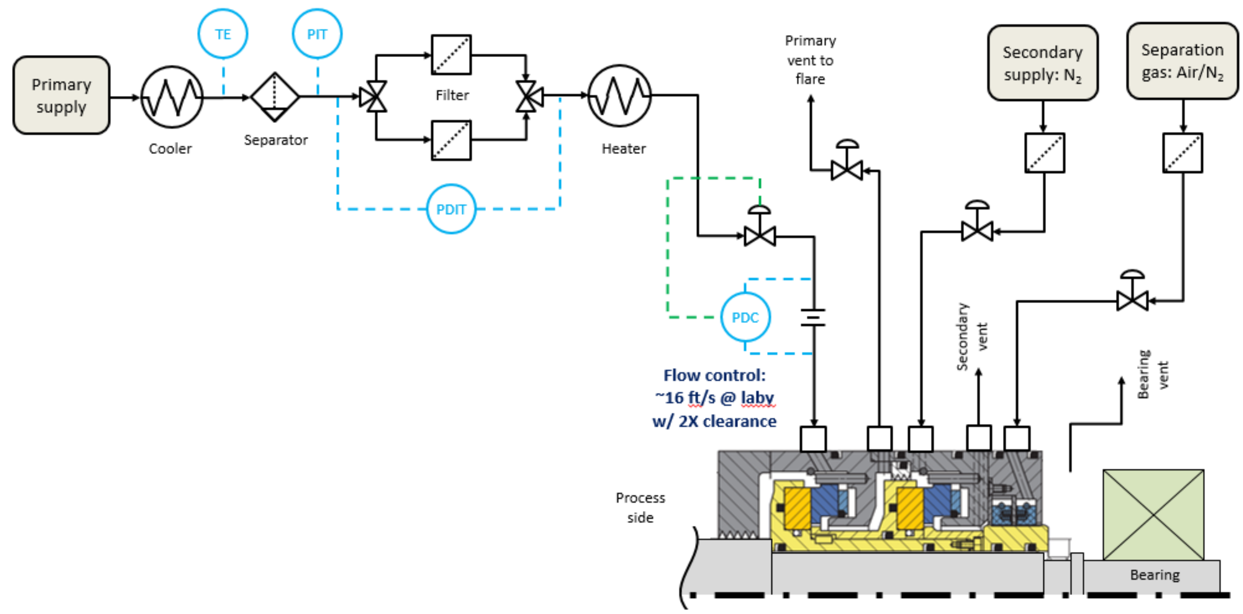

Figure 14. Representative DGS Supply Process Diagram (Tandem Seal with Intermediate Labyrinth)

Primary Supply

It is imperative that the seal gas supply to the seals has sufficient pressure, is clean, and is not at risk for condensation. The seal gas must be uninterrupted when the machinery casing is pressurized and during rotation in order to prevent reverse flow through the process labyrinth and contamination of the DGS. It must be higher in pressure than the required sealing pressure for the entire compressor operating range, including transients. API 692 specifies to maintain at least 16 ft/s (5 m/s) velocity across the process seal, assuming a worn clearance (2X design clearance). This can be achieved by differential pressure control or flow control strategy. Supply pressure should be monitored, and it is recommended that seal gas supply pressure be a start permissive, to have an alarm on low pressure, and have a shutdown on very low or no pressure. The supply gas source is supplied during all operating conditions (startup, shutdown, pressurized hold, normal operation, etc.), and seal gas boosters and/or alternate supply sources such as nitrogen may be necessary in order to maintain sufficient supply pressure. An example P&ID is shown in Figure 15.

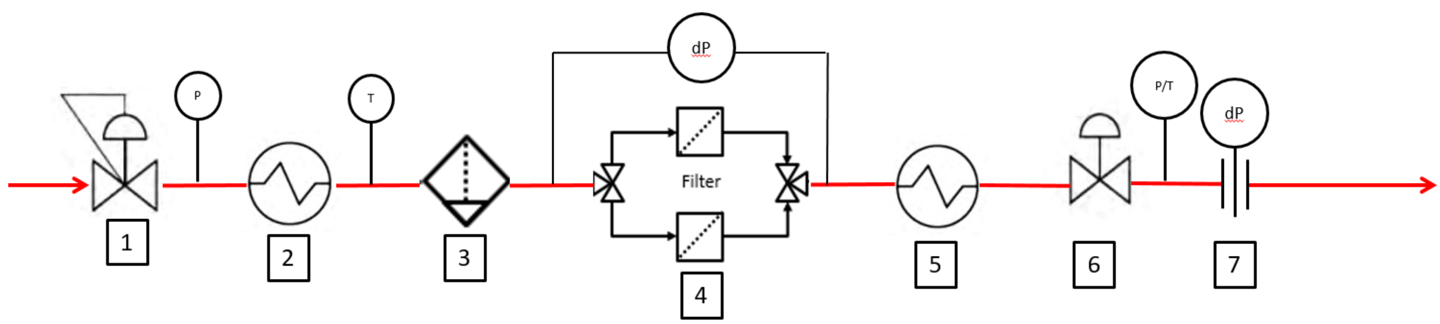

Figure 15. Primary Seal Supply & Seal Gas Conditioning System

Figure 15 component identification:

1. Pressure regulator to drop pressure between compressor discharge and seal conditioning system if components have a Max Allowable Working Pressure (MAWP) less than compressor discharge
2. Cooler (if required) to reduce temperature below saturation upstream of the separator
3. Separator to remove liquids from the system
4. Duplex filter to remove particulates with Pressure Differential to monitor blockage and a backup filter so that cartridges can be cleaned / replaced while operating
5. Heater (if required) to ensure that seal gas supply is warm enough (high enough enthalpy) to ensure that the fluid will not drop into a two-phase region across the primary seal face
6. Control valve to regulate flow (16 ft/s) through process seals
7. Orifice flowmeter for monitoring flow rates, pressure, and temperature entering the seal

Cooler (API 692, Part 3, Section 7.2)

Because process gases that are running through the compression systems are not always dry, or they may contain condensates, it is important that liquids are removed from the supply gas before entering the seal system. Coolers are used to reduce process gas temperature 30ºF (17ºC) below the saturation temperature so that the gas and liquid may be separated. These coolers can either be water or air cooled, depending on saturation temperatures. These components should have vents and drains for bleeding out air and also removing trapped water from the system. They should have removable bundles that can be removed and cleaned as needed. Because there is potential water in the system and also depending on the process gas, corrosion can occur and will need to be mitigated. A control system is designed to measure temperature downstream of the cooler to ensure the flow is below the saturations temperature before entering the separators and filters.

Table 3. API 692 Cooler Requirements (Unless Otherwise Specified by Purchaser) [6]

Water Velocity of HX Surface	1.5 m/s to 2.5 m/s	5 ft/s to 8 ft/s
MAWP	≥ 7 barg	≥ 100 psig
Test Pressure (1.5 MAWP)	≥ 10.5 barg	≥ 150 psig
MAX Pressure Drop	1 bar	15 psi
Max Inlet Temperature	30ºC	90ºF
Max Outlet Temperature	50ºC	120ºF
Max Temperature Rise	20ºC	30ºF
Min Temperature Rise	10ºC	10ºF
Water-side fouling factor	0.35 m^2K/kW	0.002 hr-ft^2-F/Btu
Corrosion Allowance (carbon steel)	3 mm	1/8 in

Separators (API 692, Part 3, Section 7.3)

The next components after the cooler are separators. A separator is used when there is potential liquid in the system. Key considerations for separators are centered on being adequately sized for the total flow (liquid + gas) with maximum downstream liquid. Separators ensure that gas entering the seal is "dry" to prevent seal damage.

A separator shall include: separation section, liquid collection section, level-indicating transmitter, drain, removable cover, and separate connections for individual devices. It shall be sized for 3X seal gas normal flow with the following separation requirements: 98% of 10 µm and larger solid particles and 50 ppmw of liquid particles. Removable bolted covers are required for top access for cleaning and maintenance. All separator materials in contact with the process gas shall be 316/316L stainless steel.

Filters (API 692, Part 3, Section 7.6)

For any filters used in a DGS system, they must be sized adequately for a range of flows. To prevent damage to the filter element, a differential pressure (DP) sensor is required. Filter elements are not designed to handle a high DP. A high DP reading will indicate one of two things: high flow rates, which the filter was not adequately designed for, or the filter element being clogged. Because of this potential for clogging, filters should have a removable cover (for cleaning and maintenance) that can be removed and inspected while the filter is installed without the need to remove the entire body. In addition, a backup filter (duplex style) allows flow to be diverted while being properly filtered, and clogged elements can be removed and cleaned. Filter housings shall be 316/316L stainless steel. They should have a minimum efficiency of 99.9% on particles larger than 10 µm.

A few important things to note when choosing a filter / separator for a given application:

- Should be immediately upstream of the seal to catch contamination from other components
- Seal gap requires 3 µm filtration
- Use transfer valves for duplex setup to enable changing of filter elements while in service
- Monitor DP (alarm recommended) – Max clean pressure drop of 10 kPa
- Pre-filter may be needed to remove larger particles (e.g., a cyclone filter)
- Removable bolted cover for filter cartridge replacement
- Should not be able to bypass all filters

Heater (API 692, Part 3, Section 7.5)

Depending on the process gas, heaters could be required to ensure at least 20°C (35°F) margin above the dew point. Due to isenthalpic expansion across a seal face, there is potential to drop into a two-phase fluid region. As shown in Figure 16, it is important to ensure adequate superheat of the process gas mixture at all pressures so that liquids are avoided at all points of the process. The same principle applies if the formation of solids could also occur, e.g., with CO_2. To ensure proper control, heater set points are tied to a temperature reading downstream of flow control valves close to seal gas inlet.

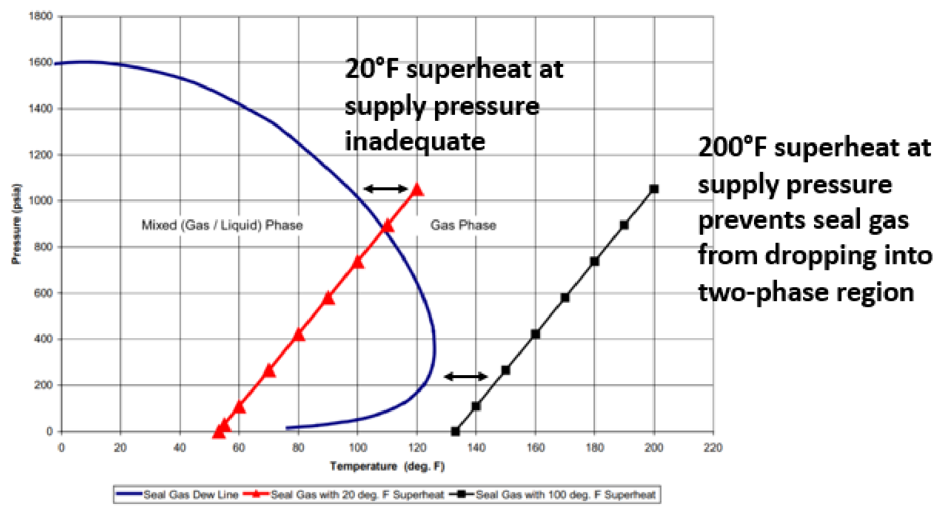

Figure 16. Pressure Temperature Plot of Typical Process Gas Mixture [1]

With dry gas seal supply flows, there will typically be a minimum supply enthalpy to ensure that two phase regions are not reached when dropping pressure across a seal face. It is important to note that there will be energy input into the leakage as it migrates across the seal, but there is a need to maintain minimum supply temperatures to prevent localized regions of two-phase flow that could lead to seal failure.

Some important considerations / requirements for DGS heaters:

- All heater materials in contact with process gas shall be 316/316L
- Heaters shall be electric type
- Sized for 3X normal seal gas flow
- Sized for minimum temperature at seal gas cooler outlet and maximum seal gas pressure (highest density)

Booster (API 692, Part 3, Section 7.4)

Another important feature in any DGS system is the gas booster. While operating, primary supplies are fed from compressor / pump discharge. However, off-design scenarios need to be considered, especially when the compressor is idle and pressurized, or when there is insufficient DP from the process compressor. When the loop is pressurized, the DGS must be supplied with filtered supply gas to prevent potential particle contamination from the loop as it vents through the seals. Boosters should be sized for seal gas normal flow, and in some cases, multiple boosters could be required depending on gas source or needed supply pressures. Cycle life needs to be considered since these boosters will most likely see many quick transients due to the need for quick supply during trips. When choosing boosters, it is important to consider potential lube oil contamination from the booster into the supply gas.

Vents

A typical vent arrangement is shown in Figure 17, with components identified below.

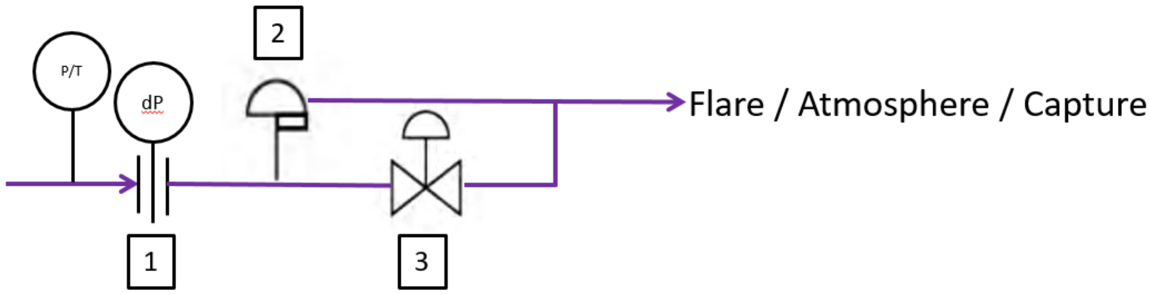

Figure 17. Vent Layout

1. Vent flow monitoring
2. Pressure-relief device
3. Back pressure control valve (if needed)
4. Secondary vents may require a gas analyzer to ensure that process gas is not leaking across the secondary seal

Since most of the supply flow runs across the process seal into the compressor, the seal vent sees relatively low flow. Vent lines should have a low point drain that needs to be checked regularly to ensure that liquid is not able to back up into the seal. If the vent is connected to a flare, a check valve should be provided to prevent the flare from back-pressuring the vent line and damaging the seal [2].

A useful monitor of seal health is flow through the primary vent using an orifice meter. Both the flow rate and differential pressure across the restriction orifice should be monitored [2]. Generally, dP measurement is a better indicator of secondary seal health. Since the seal gas leakage is directly proportional to the running gap cubed, unacceptable increases in running gap can be used to set alarm and trip values for the flow through the vent.

A catastrophic primary seal failure will result in reverse flow of the process gas across the process labyrinth seal, causing a sudden increase in vent line flow which will result in back pressure building upstream of the orifice. A pressure-relieving device, such as a burst disk, should be included in the vent line parallel to the orifice plate to remove the back-pressure. Pressure relief set points will vary based on seal design and flare system used [2]. It is recommended that this burst disk have a feedback signal to indicate when it has burst to trip the compressor and close isolation valves to minimize leakage.

Some important things to consider when choosing / sizing vents for an application:

- Block and full flow bypass valves to isolate each conditioning component
- Use resilient seat materials to reduce risk of explosive decompression
- If internal mechanisms fail, both paths should not be blocked
- PSVs as needed

Secondary Seal Supply and Separation Supply

For secondary seal supply and separation gas, the required upstream components are similar (Figure 18). Like the primary seal supply, they will require filters, flow control, and fluid monitoring for pressure, temperature, and flow. Unlike the primary seal supply, they will not require coolers, heaters, and separators as they are typically supplied with nitrogen or dry air. This simplifies the overall supply system and reduces necessary components.

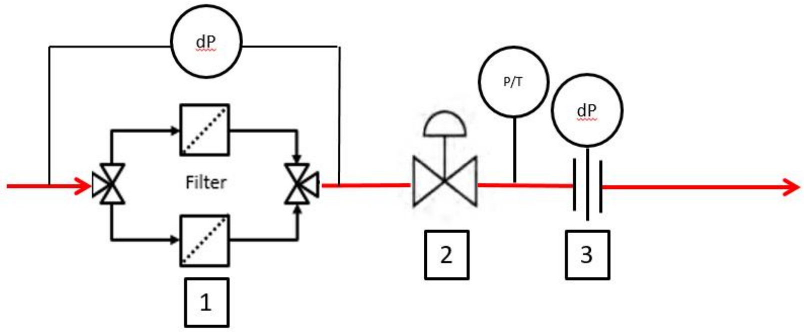

Figure 18. Secondary Seal Supply

1. Duplex Filter
2. Flow control valve
3. Flow monitoring

Unlike process gas, which is typically "dirty" when compared to required seal flow, the secondary supply and separation gas is typically "clean." However, because of the small clearances between the seal and mating ring, it is important to ensure that ALL gas entering the seal is filtered to prevent damage. As with the primary supply, a duplex filter with transfer / isolation valves, allows for filter cleaning / replacement without have to shut down the system. For any maintenance, it is important to consider down time and cleaning / repairs that can be performed while the system is operating. It is a direct tradeoff between capital and operating expenses.

DRY GAS SEAL FAILURES

It is important to understand what leads to DGS failures. One study reviewed the literature and interviewed OEMs and end-users to gather data on 194 DGS failures and 144 root causes [10]. A follow-on study focused on liquid contamination [11]. The initial study highlighted four main categories that lead to DGS failures: process / supply contamination, installation / geometry, and lube oil contamination (Figure 19). Most failures are associated with contamination, which is most often due to improper gas supply (primary, secondary, or separation). That will lead to supply contamination and particles getting in between seal faces, non-adequate separation gas that will lead to lube oil getting into the seal, and improper backpressure in vents will lead to pressure imbalance inside the seal. Most of these issues can be corrected through proper control systems and monitoring, which is mostly impacted by the seal gas supply and vent system. Contamination issues can occur at any point throughout the life of the seal. Sometimes this can happen early during commissioning of a seal gas system or much later in the life of the seal due to improper maintenance. Geometry and installation failures typically occur soon after startup.

Figure 20 summarizes the results of the study by separating the cases into the four main categories. The total is greater than 100% due to some failures having multiple causes identified. Supply and process contamination are the two highest root causes. Supply is in reference to the primary supply line from the DGS seal conditioning system and process is in reference to unfiltered process gas not running through the seal conditioning system and entering the seal. Process contamination is usually a result of improper supply flow or reverse pressurization inside the seal cavity. Lube oil contamination is usually due to improper separation air buffering the seal from bearing lube oil. This could also be a result of improper vent / drain lines for oil leading to pressure building up inside the cavity, forcing oil to leak across the rotor-to-rotor seals, and in between the rotor-to-stator buffer seals.

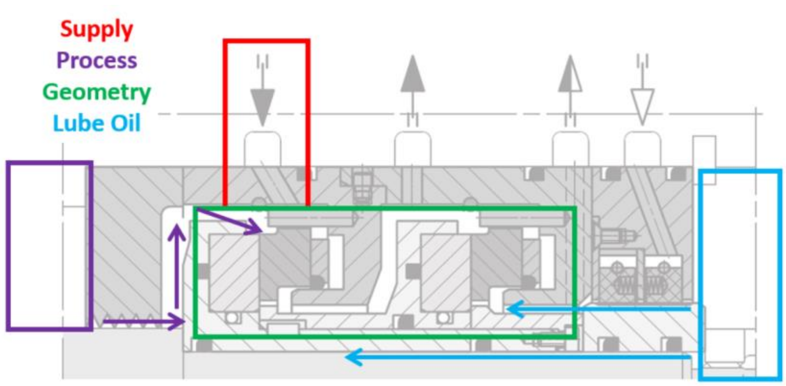

Figure 19. Contamination Locations

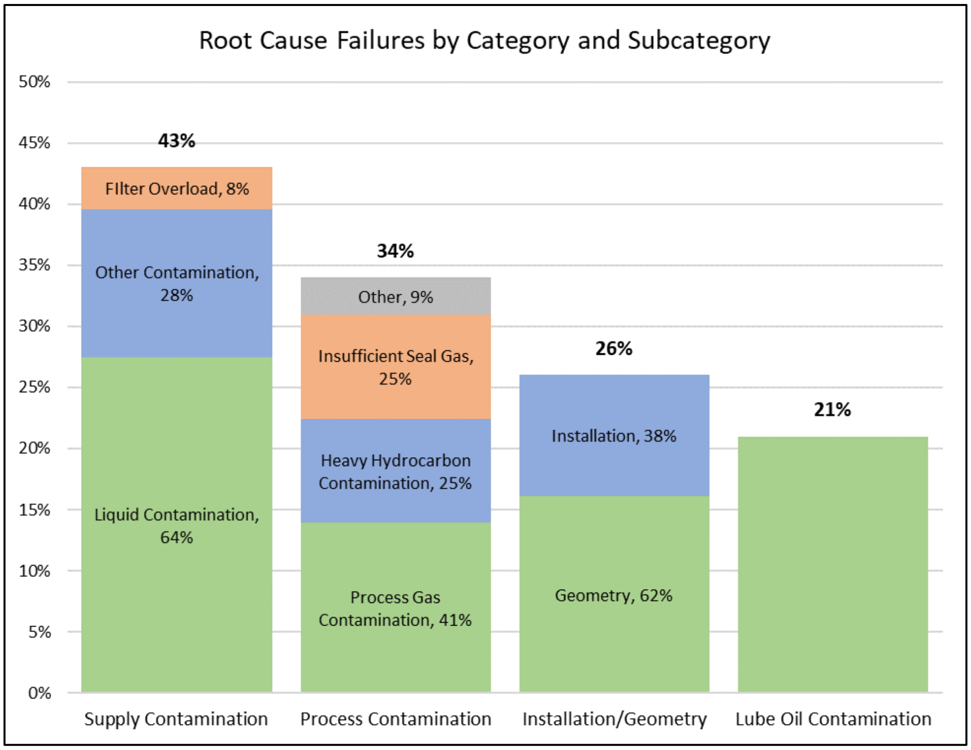

Figure 20. Identified Root Causes from Study of 194 DGS Failures [10]

Supply Contamination

The results from previous studies [10] show that supply contamination is by far the highest cause of failures in the dry gas seals studied – 43% of the failures that were reviewed. Supply contamination consists of several types of contaminants introduced to the seal through the supply system. Failures are broken down into subcategories for liquid contamination, other contamination, and filter overload, which will be discussed in detail below.

Liquid Contamination

Supply contamination includes liquid contamination, which is where liquid from the seal supply gas gets between the rotating face and mating ring, and eventually causes seal failure. Of the supply contamination failures, 64% of the failures were due to liquid contamination. Therefore, a subsequent study [11] focused on liquid contamination failures. Liquid can drop out of the gas due to the Joule-Thomson effect; as the seals gas expands in volume and drops in pressure over the seal face, the temperature will also drop. This causes condensation to drop out of the seal gas, which may result in seal failure due to the liquid between the rotating face and mating ring causing rubbing and possibly heat buildup.

To avoid condensation, API 692 recommends that seal gas supply must be at least 20°C (36ºF) above the dew point of the gas at all points throughout the seal (not simply at the supply conditions). It is possible that the 20°C (36ºF) requirement is more than necessary, which could lead to an oversized heater. Stahley [2] recommends using a detailed model of the temperature and pressure drops across the system to determine the required inlet temperature of the supply gas to both avoid liquid dropout and not oversize the heater, if one is needed. In cases where liquid dropout may occur, it is highly recommended to add a heater to the seal gas panel to ensure that the gas entering the seal is going to be warm enough. Many of the failures analyzed did not have a heater, which may have contributed to the condensation problem.

Visible signs of process gas on the seal face is a common indication of a failure due to liquid contamination. For sour gas, this could be a white sulfur powder build-up. Water lines may be visible on the seal if part of the seal was immersed in fluid for extended periods of time. Liquid on the seal faces can increase the necessary lift-off pressure, causing the seals to show signs of rubbing or contact. Figure 21 shows an example of liquid getting in between the seal and mating ring, sticking to a face, and leading to material removal.

To avoid liquid contamination in systems prone to liquid dropout, proper operation and maintenance of a seal gas heater may be recommended. It is particularly important to make sure that the seal gas is allowed to heat sufficiently before starting the compressor. A few reported cases suggested that due to a rush to restart the compressor and a lack of experience or training, compressors were started up prior to the seal gas reaching its required temperature. To avoid this, a minimum seal gas temperature could be specified as a start permissive. Additionally, alarms and trips at low temperatures could warn of a heater malfunction or insufficient heating.

Another cause of liquid contamination is from liquids in the vent line not being drained. These liquids would build up in the vent line and flow back into the seal. It is imperative that the vent line has a low point somewhere before the flare, and that the operator checks it regularly. The operator should also check the seal drains to ensure no liquid build up. Furthermore, if the seal gas supply comes from the compressor discharge, the take-off should not be from the bottom of the casing, as this could allow liquids into the supply gas. Figure 22 shows evidence of liquid contamination in the grooves of a rotating ring, which can prevent proper separation of the DGS faces.

It should be noted that some companies reported seals operating successfully for extended periods of time with liquid continuously between the rotating face and mating ring, covering most of the contact area. If the liquid is sufficiently lubricating, it can possibly help in seal operation. However, given that the chance for failure is high, it is recommended that seal supply systems be designed to consider eliminating the chance of liquid entering the seal.

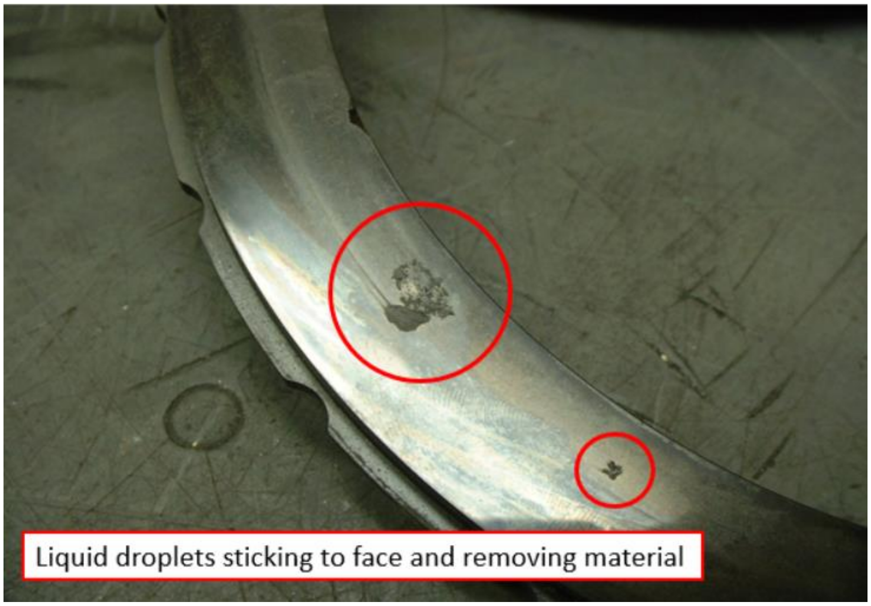

Figure 21. Liquid Build Up in a Seal

Figure 22. Grooves Filled with Liquid

Filter Overload

Another cause of supply contamination is filter overload or degradation, which accounted for 8% of reported supply contamination failures. Filter overload is a quick failure, where the filter becomes too full and collapses, sending a buildup of debris through the lines and into the seal. This can be seen as a large, rapid jump in vent pressure and vibrations. Filter degradation, on the other hand, is where the filter gradually becomes too full and starts to allow larger particles through. This will manifest itself as a slow increase in vent pressure over many days, leading to a gradual seal failure.

Filter failure can be caused by the operator not heeding alarms to change the filter. A differential pressure meter should be placed across the filter, and it should alarm on high DP. The operator must not ignore this alarm. Filters for continuous service applications should always have a duplex setup that allows the filter to be changed without taking the compressor offline, and the operator should be trained in filter replacement to ensure that debris does not enter the line during the filter change.

Coalescing filters are recommended to catch any liquid mist that may be present in the gas. However, it should be noted that a liquid slug coming through the lines will not be caught by a coalescing filter and will likely result in seal failure.

Other Contamination

Of the supply contamination failures reported, 28% were due to other contaminants not included in the above subcategories. This contamination could come from anything upstream of the seal getting blown into the seal, or debris being in the seal at time of installation. Contamination could result from a process where the system was opened for cleaning. During transients, contamination in the lines could be stirred up and forced into the seal. It is also possible to over-pressurize valves or other components upstream of the seal, causing parts to break off and flow into the seal.

Symptoms of contamination may include high primary vent pressure. Solid contamination may be visible upon seal removal and there may be pitting or signs of rubbing on the faces. If the seal gas is supplied by a separate reciprocating compressor, it is possible for lube oil to make its way into the seal. This type of contamination usually results in contact and visible marks on the seal faces. It can often lead to seal face break up.

To avoid failure by supply contamination, it is recommended that all upstream pipes are blown out after the loop is opened for cleaning. Also, using stainless steel pipes rather than carbon steel can minimize the risk of corrosion getting into the seal. Operating procedures should be checked to ensure that components are not over-pressurized at any time, particularly during transients. Valves and components upstream of the seal should be specified for the maximum pressure they could possibly experience to reduce the risk of them breaking during operation.

Process Contamination

Of the total failures, 34% were due to process contamination, which includes heavy hydrocarbon contamination (25%), process gas contamination (41%), and insufficient seal gas (25%). Heavy hydrocarbon contamination is when condensates from the process drop out and contaminate the seal. Process gas contamination is when solids drop out of the process gas, contaminating the seal. For sour gas applications, this usually results in a buildup of sulfur along the seal faces. Failures were only classified as having insufficient seal gas when it was explicitly stated as such. A surprising number of cases reported insufficient or no seal gas continuously or intermittently during operation. This is almost guaranteed to let the process gas flow into the seal and will likely result in failure. Indications of insufficient seal gas include sulfur residue on the seal face (for sour gas operations) or sticky black or clear liquids for heavy hydrocarbons. Seal face contact is nearly guaranteed, which may be severe enough to result in broken rotating faces.

To reduce the likelihood of failure from process contamination, it is recommended that alarms alert the operator when there is not enough supply gas pressure. A trip is advised for the supply gas to prevent operation without seal gas. If needed, a booster system can be added to ensure that that there is sufficient seal gas pressure during transients or if something causes the seal to have inadequate supply pressure. Furthermore, supply flow should be automatically regulated via a flow control or differential pressure control scheme, as it is more reliable than manual adjustment.

A highly important cause of seal failure due to insufficient seal gas is slow roll. It was found that 15% of process contamination occurred during slow roll. If the seal gas supply is taken off of the compressor discharge, during slow roll there may not be enough pressure rise across the compressor to supply adequate pressure to the seal gas supply. To avoid this, slow roll speed should be adjusted to be high enough to maintain sufficient seal gas supply pressure or a booster system should be used for seal gas supply.

Also of interest were several outer seal failures that failed due to insufficient supply gas. While this may seem counterintuitive, it was determined that moisture passed through the inner seal (or formed along the inner seal), and subsequently contaminated the outer seal. The outer seal has a smaller running gap due to the lower differential pressure, which may make it more susceptible to small contaminants that the primary seal is able to handle.

Geometry Problems

Of geometry and installation failures, 62% were due to geometry issues, most of which were related to the drive pin (shown in Figure 23). In many cases, the drive pin showed visible signs of being too long, which would restrict axial movement on the seal faces. This results in immediate failure on startup due to excessive swash (axial runout) and potential axial misalignment between the rotating face and mating ring. This swash will likely degrade the O-rings quickly and cause them to break. The mating ring retainer may show physical damage from the excessive movement.

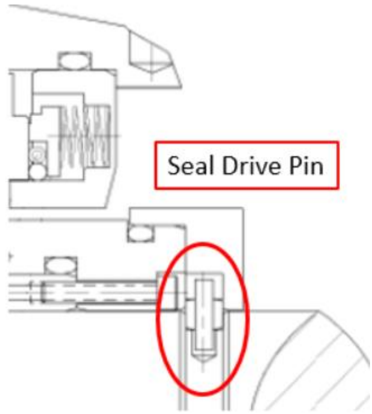

Figure 23. Drive Pin Alignment Feature [7]

To avoid drive pin issues, the drive pin dimensions can be checked prior to installation of the seal. It may be easier to perform a few checks during installation suggested by Stahley. These tests can verify that the seal is able rotate properly and move axially [2].

Several failures occurred when oil-trapping features in the compressor flow path allowed oil to drain intro the process labyrinth during pressurized hold and/or shutdown, even when a normal seal gas supply pressure was maintained. This proved to be an incredibly challenging problem to detect, since everything on the supply line appeared functional.

Other geometry issues are improper seal clearances (either too large or too tight) in which the assuming velocity is too low to properly buffer the seal (large clearances) or there is excessive rubbing (too small) leading to excessive heat generation inside the seal. Due to the critical function of the seal, it is necessary to verify critical dimensions or review inspection reports to ensure the supply systems are sized properly. In addition, it is important to communicate geometry and conditions outboard and inboard of the seal with the manufacturer to ensure that the chosen seal is properly designed for that specific machine

Installation

Of reported geometry and installation failures, 38% were due to installation issues. In some cases, the seal was not installed with all of the necessary components. This would result in immediate seal failure. Missing parts included bolts and labyrinth snap rings. There were a few cases where the seal was not aligned properly during installation and was therefore not able to rotate, or move axially. For this reason, Stahley [2] recommends checking the axial travel, and rotation if possible, of a seal manually prior to installation. A couple of cases reported sheared O-rings, or seals containing debris prior to startup.

Lube Oil Contamination

Twenty-one percent of reported failures were due to lube oil contamination, which is when lube oil from the bearings flows into the seal between the rotating face and stationary ring. This is caused by insufficient separation seal supply gas or excessive lube oil flow. Seals that failed due to lube oil contamination often had wet lube oil on the seal faces. Excessively dry nitrogen as the separation gas can increase the likelihood of lube oil contamination because the seal will become dried out and fail prematurely. A few reported failures were caused by the separation seal not being adequate for the lube oil conditions present.

A solution to avoid lube oil contamination is to increase the separation gas pressure to the upper design limit of the seal, as specified by the seal manufacturer. However, excessive differential pressure across the separation seal can accelerate the wear of the barrier seal and subsequently decrease its reliability. It is also recommended to have a minimum separation gas pressure as a start permissive for the lube oil system to ensure that the lube oil is not allowed into the system without separation gas present and shutdown on very low or no separation gas pressure. In the case of excessively dry nitrogen, experience suggests that adding humidity will reduce the likelihood of failure. Additionally, there are some specific seals designed for gasses with particularly low dew points.

CONCLUSIONS

DGSs are found on the majority of centrifugal compressor today due to their excellent leakage performance. A DGS is a face seal that relies on hydrodynamic pressure forces to maintain a small gap [3-5 micron (0.1-0.2 mil)] between the rotating and stationary faces, while being tolerant of shaft axial movement. Several DGS arrangements exist for various applications, including single seals, double seals, and tandem seals. Due to the small clearances involved, a DGS is sensitive to contamination, so they are designed such that the only gas that enters the seal is filtered and dry. Specifically, seal supply gas buffers the inboard side of the DGS to prevent process gas from leaking in through the process seal, and separation gas buffers the outboard side of the DGS to prevent bearing oil from leaking in. The systems that deliver these flows, defined in API Standard 692, include filters, separators, heaters, and coolers as well as instrumentation and control systems to ensure that only clean, dry gas is presented to the DGS for reliable operation. Some DGS failure modes and practices to prevent them were discussed. Overwhelmingly, contamination is the most-common cause of failures, according to recent surveys, and liquid contamination in seal supply gas was identified to be a significant single contributor to DGS failures. This highlights the importance of properly maintaining and monitoring seal supply and vent systems to ensure reliability. Other failures not related to contamination include geometry and installation issues, which can be mitigated by certain checks during assembly and installation.

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