INTRODUCTION TO DRY GAS SEALS AND SYSTEMS

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ABSTRACT

Dry gas seals are used as low-leakage shaft end seals for many centrifugal compressors and other turbomachines. This short course provides a comprehensive overview of sealing system and dry gas seals in various turbomachinery applications, addressing multiple topics ranging from fundamentals to detailed design considerations for reliable operation. A course attendee can expect a greater understanding of technologies, failure modes, and requirements for components in dry gas seals and seal supply/vent systems, with perspectives from an end user, a seal manufacturer, and a research organization.

This short course will give listeners a thorough understanding of dry gas seals, including design, operation, and maintenance. Starting with the background of how dry gas seals were developed as a response to issues with wet seals, the course will then move into a detailed discussion on seal design. The instructors will explain how each component of the seal contributes to its operation and issues that can arise if parts are selected incorrectly. Next, seal selection for various applications (pipeline, process, advanced applications) will be discussed. Methods for seal testing to ensure that design conditions are met will be described, including test rigs studying off-design conditions, such as transients or contaminant injection.

The gas conditioning process can be critical to successful seal operation, so seal gas panels and their components will be discussed in great detail. Operation during transients can be particularly challenging, so panel considerations specific to transient operation will be discussed. The recently-released API 692 will be discussed as it pertains to dry gas seal panel design, seal requirements, and seal testing.

Understanding common failure modes is an important step to improving dry gas seal reliability. Recent research on dry gas seal failures will be presented, including failure statistics and failure modes. Insight on failure modes specific to heat generation from liquid contamination will be discussed, and recommendations will be provided to reduce failures.

This short course is aimed primarily at end users, but the multifaceted approach (end user, OEM, research) will provide a valuable perspective on dry gas seals to anyone in the rotating equipment industry. By the end of the course, attendees will have a detailed understanding of dry gas seals and their associated systems.

INTRODUCTION

The purpose of dry gas seals is to provide reliable shaft sealing on turbomachinery to reduce the amount of process gas escaping from the machine's primary flowpath to an acceptable amount. According to Stahley [1], dry gas seals first emerged around 1970, in response to an industry need for a less complex and hazardous system than floating ring oil seals. By comparison, dry gas seals are simpler and require fewer support systems that wet seal systems.

There are a large variety of dry gas seal configurations, including single seals, tandem seals, and tandem seals with an intermediate labyrinth seal. While most of the seals examined were tandem seals or tandem seals with intermediate labyrinth seals, single seals will be first explained due to their simplicity. Figure 1 shows a single dry gas seal configuration. The rotating face (shown in red) and the stator ring (shown in orange) are held in place by a spring (shown in grey). The rotating face has a grooved pattern on the face intended to optimize seal performance [1]. The stator ring is held in contact with the mating ring by the spring retainer assembly when the machine is not spinning or during pressurization. When the faces are in contact and not rotating, there is very little or no leakage. As shown in Figure 1, O-rings (labeled as "dynamic sealing element") are also used in the seal assembly to provide seal face alignment.

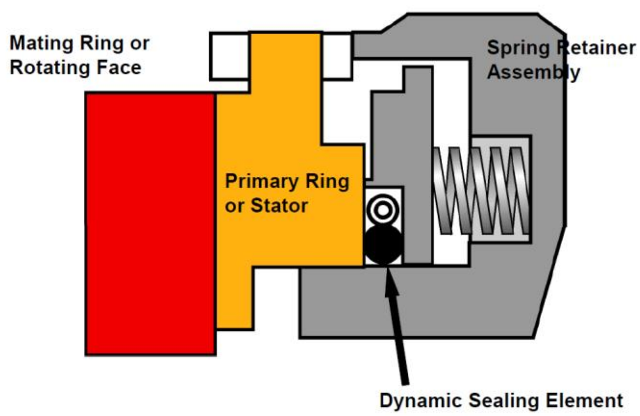


Figure 1 – A Single Dry Gas Seal [2]

There are two ways for the rotating face and primary ring to lift-off: static and dynamic. Static lift-off may occur when hydraulic balancing within the seal assembly causes the rings to separate slightly. Dynamic lift-off occurs during rotation when flow enters the grooves on the rotating face. The volume reduction at the tips of the grooves causes the gas to compress and form a pressure dam (an area of slightly higher pressure). The pressure dam causes lift-off between the faces to a running gap of 3 to 10 µm (0.1 to 0.4 mils) [1]. There is a wide variety of groove patterns intended to accomplish this; a few of these are shown in Figure 4 and Figure 5.

During operation, distance between the rotating face and primary ring is self-regulating to accommodate axial movement during operation. If axial movement causes the gap to increase, the pressure dam will be reduced due to the increased volume, and the faces will be drawn back together to their intended running gap [1]. If the gap decreases, the pressure dam will increase and the faces will be pushed apart. Figure 2 graphically explains this behavior on a simple level, while Figure 3 demonstrates the effect of the changing pressure dam on running gap. The spring force is only noticeable at lower pressures. At higher pressures, the spring force is negligible compared to forces due to pressure.

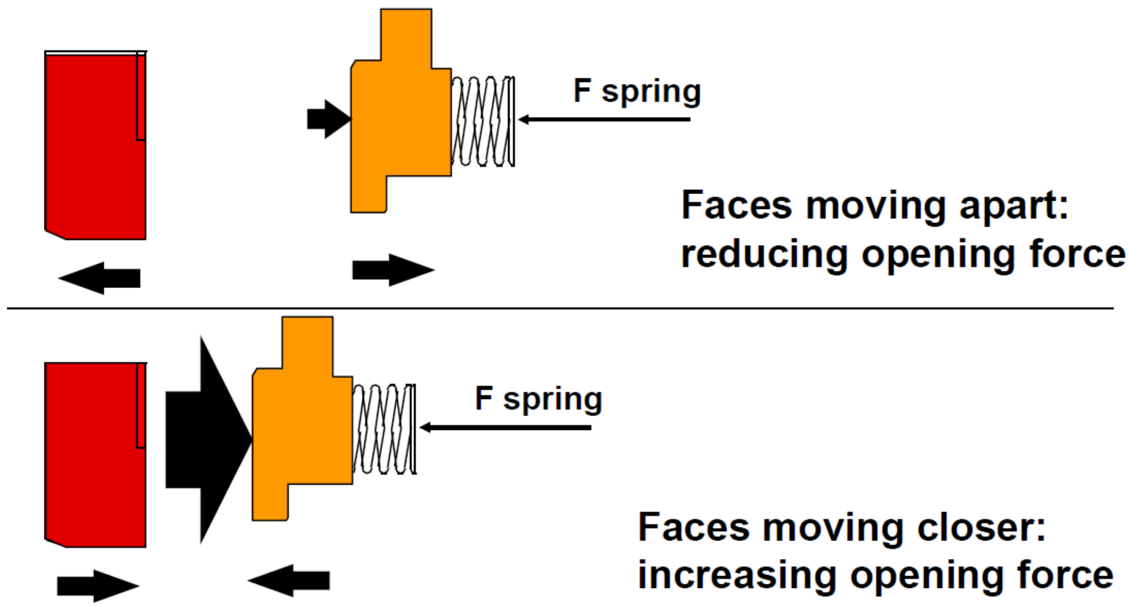


Figure 2 – Representation of Self-Regulating Gap [2]

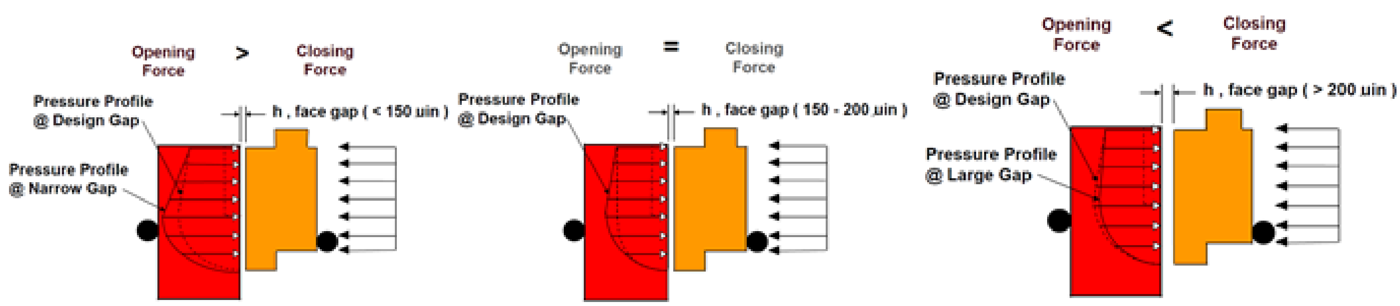


Figure 3 – Forces with Running Gap Narrower than Usual, Usual, and Wider than Usual, Respectively [2]

The rotating face and primary ring should not come into contact when spinning at high speeds, as this will rapidly degrade the faces and lead to seal failure. Seal leakage is proportional to the cube of the running gap and directly proportional to the sealing pressure and seal diameter. Film stiffness is equal to the derivative of opening force with respect to the running gap and increases nonlinearly as the running gap decreases. Thus to minimize leakage, ideally the gap would be as narrow as possible, and the film would have a high gas stiffness [1].

The rotating face can be designed for unidirectional rotation or bidirectional rotation.

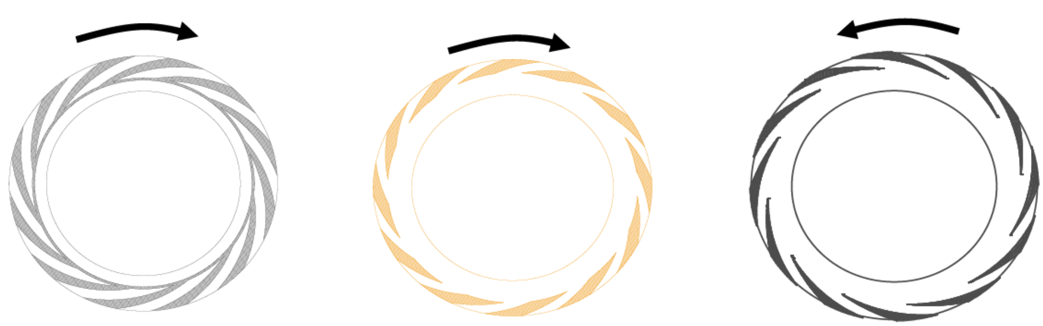


Figure 4 – Drawing of Groove Patterns of Unidirectional Seal Designs [2]

Unidirectional seals (several examples shown in Figure 4) are generally better suited for applications with axial movement since they have a wider running gap that reduces the risk of contact and a stiffer gas film making the seal able to accommodate some disturbance [1].

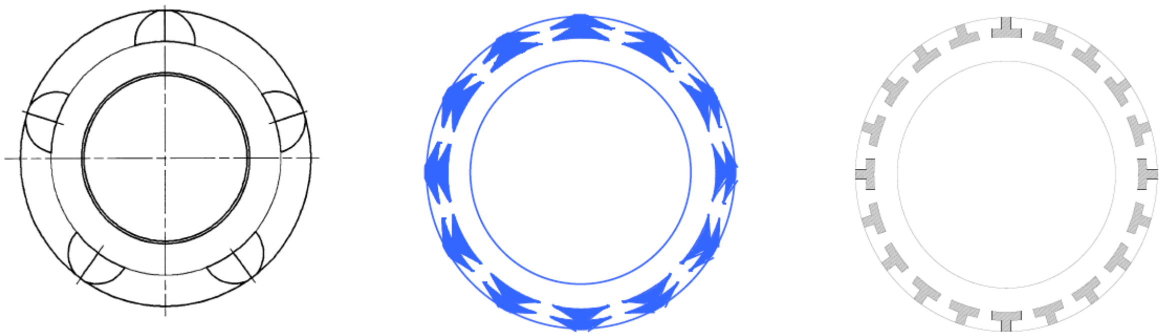


Figure 5 – Drawing of Groove Patterns or Bidirectional Seals [2]

Bidirectional seals, shown in Figure 5, are generally only recommended for applications where the ability to withstand counter-rotation is necessary [1]. However, maintaining a stock of bidirectional replacement seals is a common end user practice because it reduces the required inventory of spare seals, since a bidirectional seal can be used on either end of a rotor. Each seal is very application-specific; therefore it may be possible that conditions exist where either a unidirectional or bidirectional seal will function equally well. It is recommended that the original equipment manufacturer be contacted to ensure that the seal being selected matches the application and system.

OVERVIEW OF DRY GAS SEAL FUNDAMENTAL OPERATING PRICIPLES AND BASIC DESIGN CONCEPTS

As highlighted in the introduction, dry gas seals provide a solution for low leakage end seals. By running on a thin film, the gap is much smaller than what would exist for standard labyrinth or hole pattern seal. With a reduced gap, the amount of flow through the seal will decrease significantly, limiting process gas injection into the environment and also improving overall machine efficiency. In addition, this very stiff film between the mating and primary ring can tolerate transient conditions from speed and pressure changes. They do require a minimum speed to ensure that the seals lift off and will not rub during slow roll operation. Because of this thin film and no contact between the stationary and rotating rings, there is no wear between the seals during high speed operation. They only contact during start up and shutdown and at very low speeds. In addition, by not contacting, there is little to no torque acting on the seals that would increase power consumption. Only additional windage has to be considered. When compared to an oil seal, a dry gas seal leads to no oil migration into the process gas and also vice versa. An example seal is shown in Figure 6.

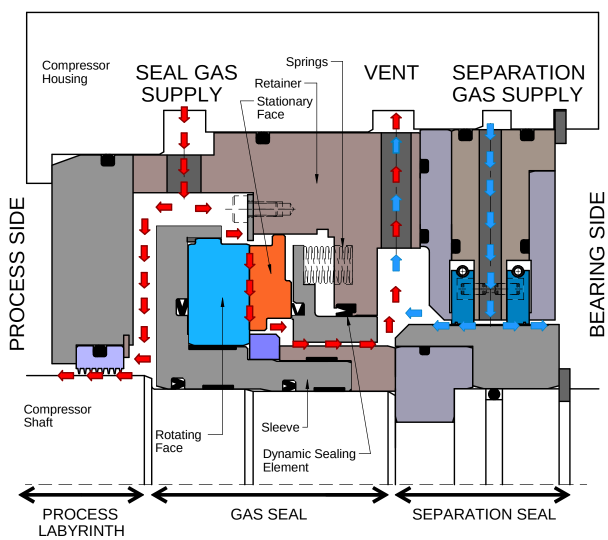


Figure 6 - Dry Gas Seal Layout Example

Figure 6 shows a cross section of a single dry gas seal and labyrinth separation seal installed within a compressor cavity. The rotor interfaces with the compressor shaft and comprises of: Sleeve assembly and Rotating sealing face. The stator interfaces with the compressor housing and is comprised of: retainer, carrier, springs, and a stationary sealing face. Secondary seals like O-rings and spring energized polymers are used in other locations around the seal to contain the process gas and ensure that it is forced only through the rotating and stationary sealing face.

There are a few seal layouts that can be considered when selecting a dry gas seal. A few of the choices are shown in Figure 7 through Figure 9.

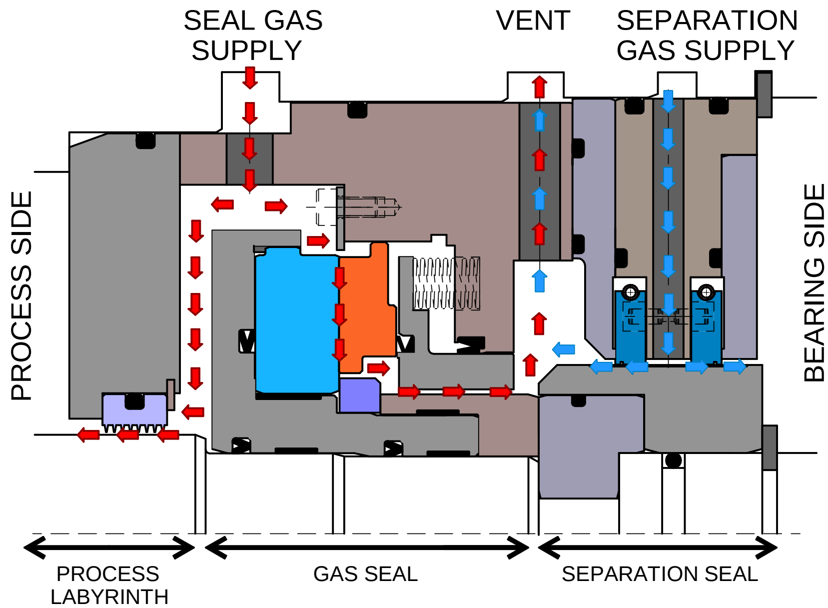


Figure 7 – Single Seal

A single seal consists of a single set of sealing faces separating process gas from the atmosphere. These seal types are generally used for applications where the gas is neither flammable nor harmful to the environment e.g. Air, Nitrogen, CO_2 etc.

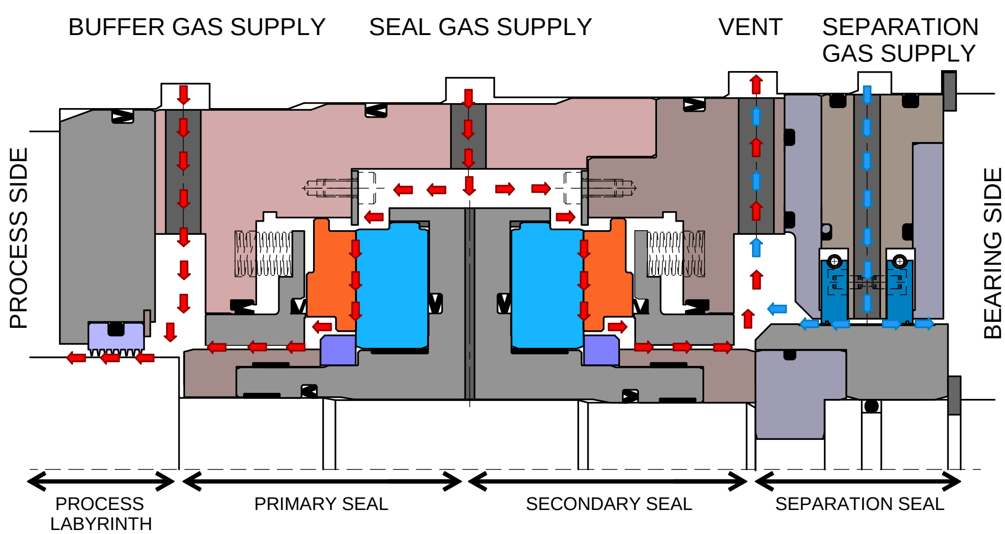


Figure 8 – Double Seal

Double seals are used where leakage of process gas to the atmosphere is undesirable under normal operating conditions. Double seals consist of two sets of sealing faces arranged in an opposed configuration. Seal gas is supplied at pressures higher than the process gas. This ensures that the higher pressure gas is expanded across the primary seal. Seal gas leaks across the primary seal, along with any buffer gas that is introduced into the compressor. Double seals suffer from very high heat generation because both the primary and secondary seals are operating at high pressure. Double seals are typically used for lower pressure duties due to the heat generation. In case of primary seal failure, the secondary seal is designed to operate at primary seal conditions and therefore prevent uncontrolled leakage into the atmosphere.

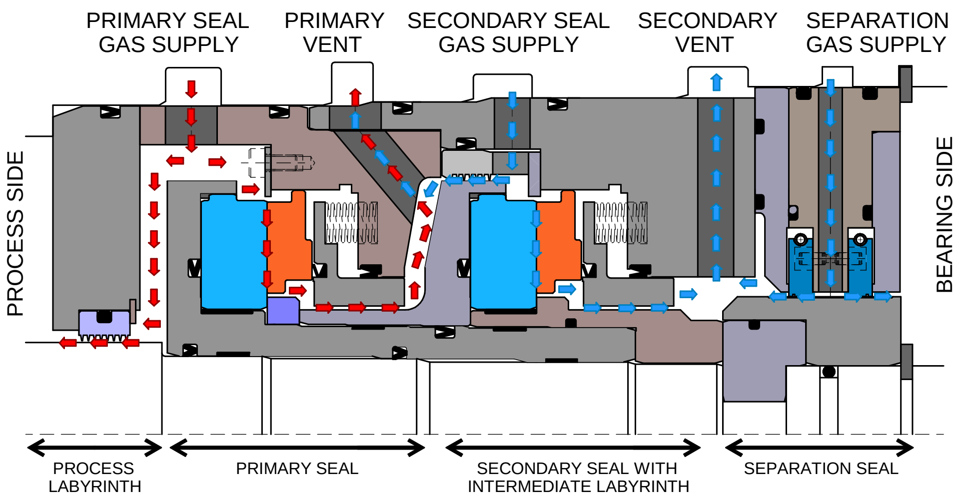


Figure 9 – Tandem Seal

Tandem seals consist of 2 single seals coupled in series. They are suitable for all pressures where some leakage to atmosphere is acceptable. Intermediate labyrinth seals are used where leakage to atmosphere is unacceptable. Buffer gas is used to direct all process gas to the primary vent.

Balancing Forces

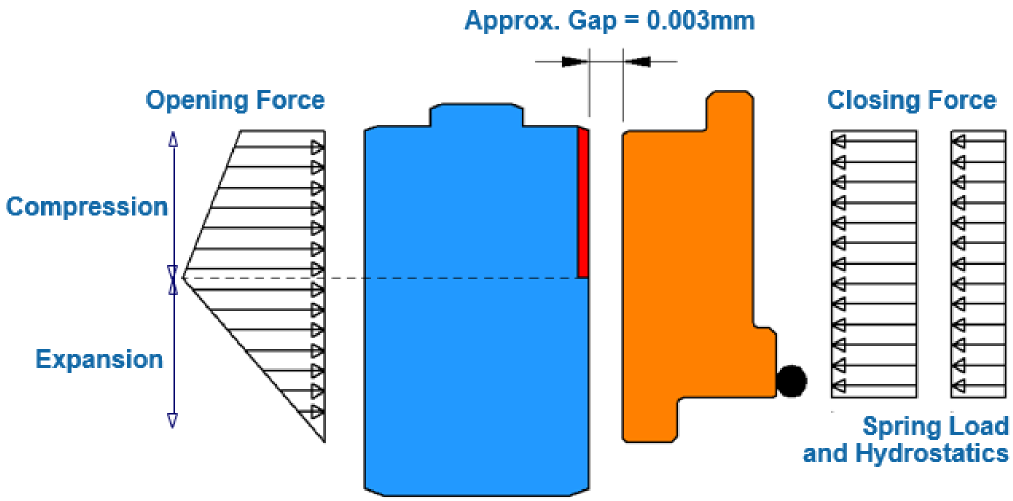


Figure 10 – Seal Force Balance Diagram

The primary ring (stationary) floats on a thin film generated by the optimized groove. The groove features allow the sealing faces to create hydrodynamic and hydrostatic lift. Hydrostatic lift is when seal faces separate when pressure is applied across the sealing faces. Hydrodynamic lift is when seal faces separate when the shaft is rotated irrelevant of pressure. Gas is drawn towards root of the groove, forcing the two faces apart. The seal operates in an equilibrium state in which the closing and opening forces are equal. If the gap reduces, the opening force increases considerably to maintain the gap (film stiffness effect). If the gap increases, the opening force decreases considerably to maintain the gap (film stiffness effect).

Groove Designs

A few groove designs were looked at in the introduction in Figure 4 and Figure 5 in showing some of the unidirectional and bi-directional designs. Depending on the application, each seal design will have its own advantage. In terms of performance, uni-directional seal will typically have stiffer films and have more freedom in overall operation range, especially during transient events. However, in various applications, there are possibilities of reverse rotation of the compressor or turbine shaft. This will be due to reverse flow during a trip event. In that case, a bi-directional seal will have to be considered. Figure 11 and Figure 12 show more details on the differences between the two designs. Both operate on the same principles overall.

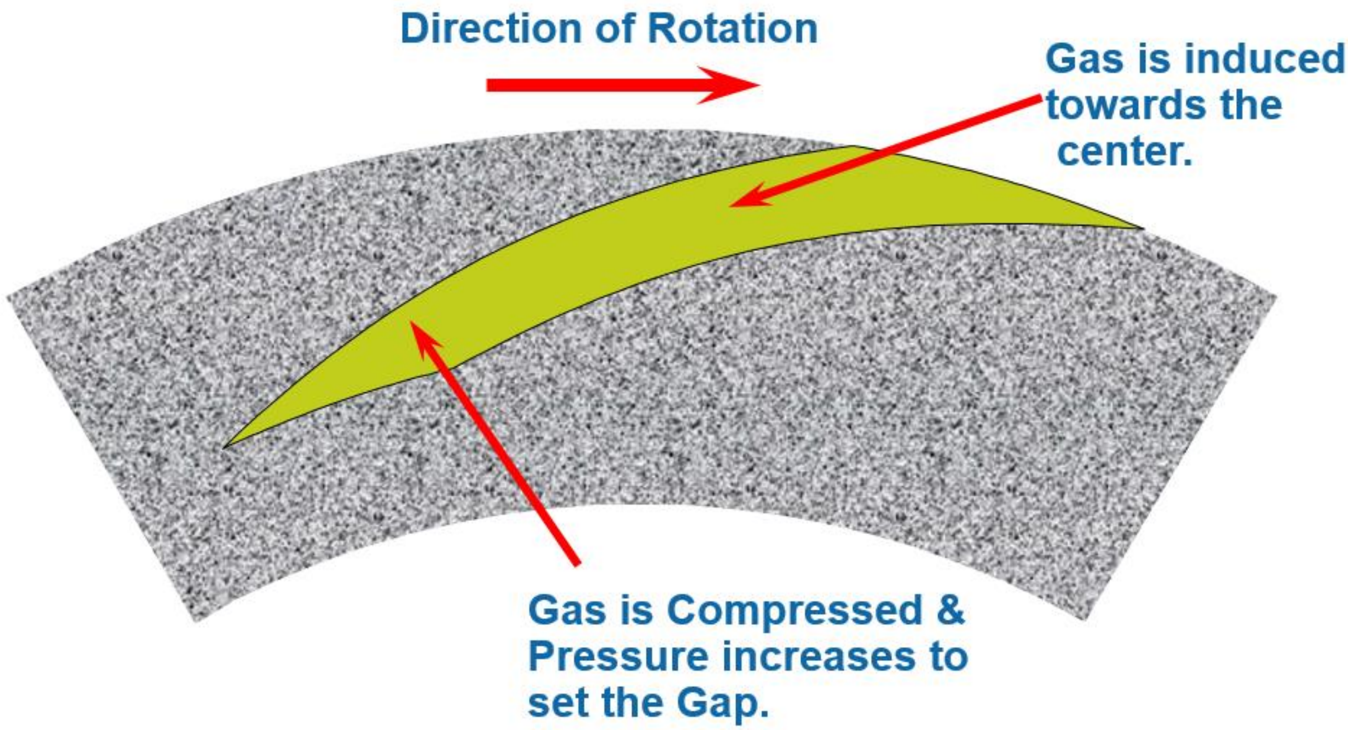


Figure 11 – Unidirectional Spiral Groove

For a uni-directional seal, the spiral groove design is optimized to provide proper gas compression and allow for needed lift off on the seal to set the operating gap. The groove works by pulling gas towards the root, during this process, the gas is compressed. The leads to an operating gap 1µm (0.04 mil) static and 3 µm (0.1 mil) dynamic approximately. They create an impeller dynamic effect, but the uni-directional is more effective than a bi-directional seal.

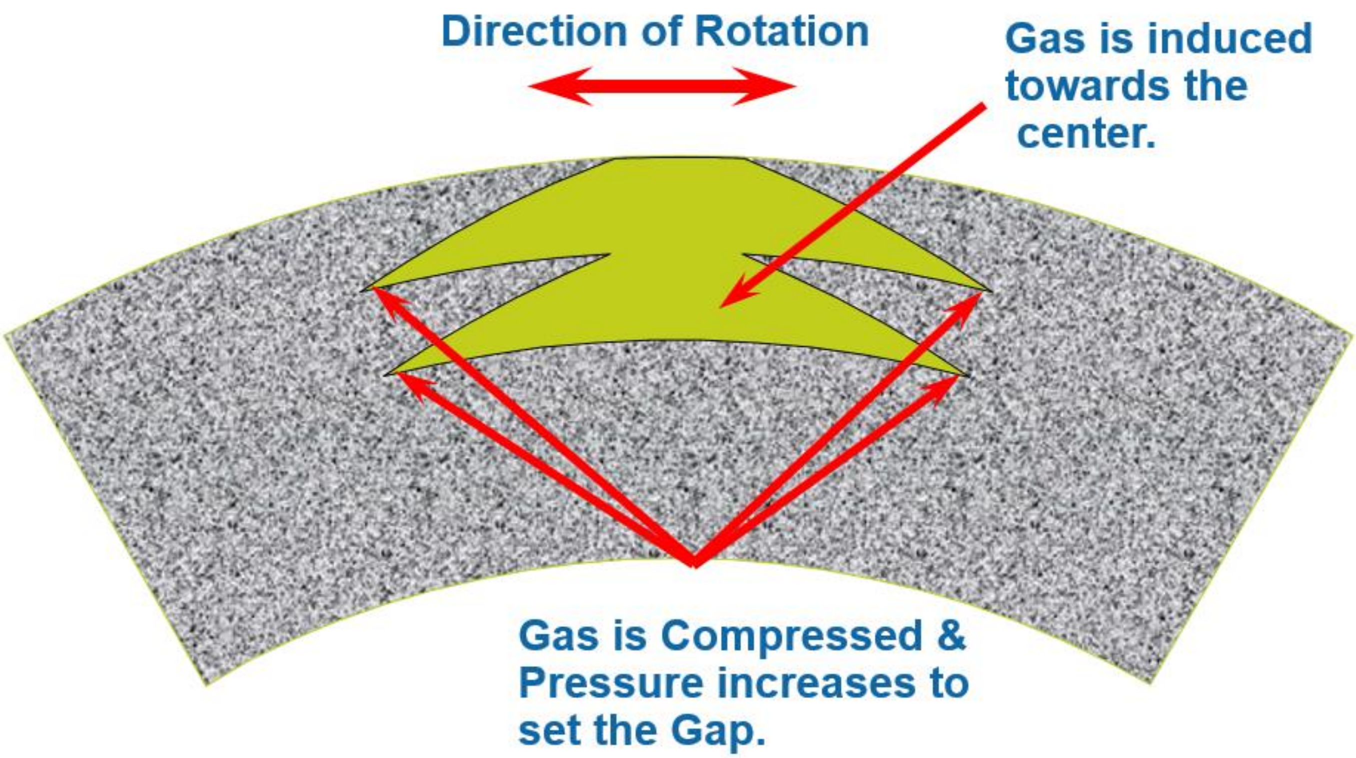


Figure 12 – Bi-idirectional Spiral Groove

Dynamic Sealing Element

One of the other critical parts of the dry gas seal assembly, is the dynamic sealing element as shown in Figure 13. This particular seal is important because it provides the other main seal between high pressure process gas and atmosphere. In addition, this seal has to also be able to slide freely (with minimal resistance) to allow for axial movement of the mating ring. One of the key features of the proper operation of a dry gas seal is that it can move axially and not act like a thrust bearing. This movement is critical when it comes to seal lift off and axial growth of a rotor as it heats up and cool down. Without a properly functioning dynamic sealing element, the stator can lock up and cause a forced rub between the mating ring and rotating seal. The stator can also lock up and cause a seal to open up (stop sealing).

In addition, this seal also provides the balance pressure to the mating ring which ensures proper lift off of the seal. In the overall seal assembly, this seal is the 2nd most important element. Being able to properly seal while also being able to slide freely provides many challenges in terms of material selection and seal type. This will be described in more detail in the material section.

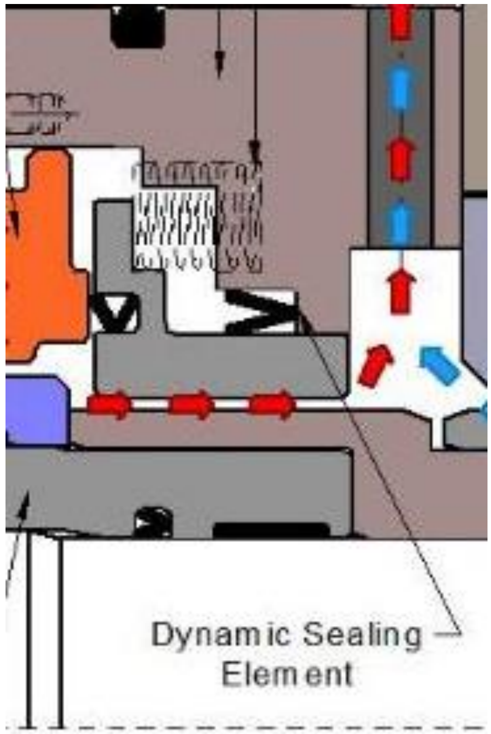


Figure 13 – Dynamic Sealing Element

Materials

In regards to the many internal seal components, it is important to understand the various material options and the advantages of each. In addition, it is also important to understand the limitations of each. While seals will usually operate on the suction side of the system, there are operating scenarios where higher pressures or temperatures will be seen and it is critical to understand what operation limits the seals can handle to make sure the proper configuration is chosen. It will also help in determining necessary controls and measurement features that will be required to ensure safe and successful operation of the machine and seals.

For the main process seal, there are two main seals that have been discussed, the mating ring (rotating seat) and the primary ring (stationary face). The mating rings are generally manufactured from Silicon Carbide or Tungsten Carbide. Silicon carbide has some clear advantages in that it is inert and does not have any chemical compatibility issues with acidic gases. There is no binder that can be damaged from the process gases. Also, it is a low density material with lower inertia which leads to less power consumption to turn the seals and also limits rotating weight for rotordynamic considerations. Tungsten carbide is a much harder material but has some main disadvantages. Unlike silicon carbide, acidic gases will attach to the nickel binder and can lead to seal damage. Also, it is a higher density material that will have higher inertia. Typically the tradeoff for seal materials is pressure capability vs forgiveness. A harder seal will be able to handle higher pressures but will not be as forgiving if there is a seal rub or the seals touch. For a high pressure application, there can be an issue with two hard materials running at tight clearance.

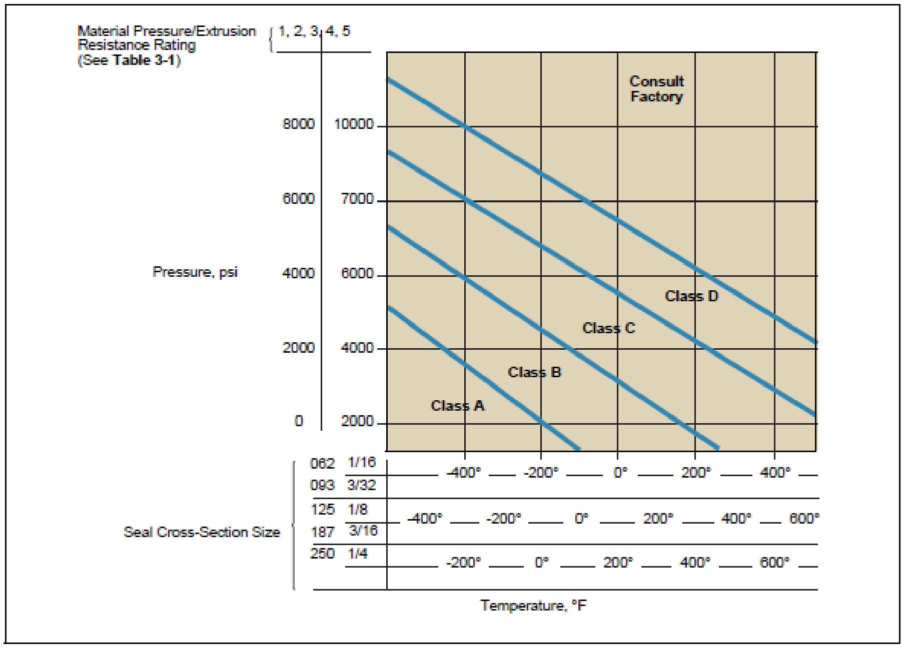
For the primary rings, the two materials of choice are usually carbon or silicon carbide. Carbon has the advantage of being a softer material and being more forgiving in the case of a rub or seal touching during operation. As with tungsten carbide vs silicon carbide, the same comparison is for silicon carbide vs carbon for the primary ring. Because of this, it is important to really investigate operating scenarios to determine if the harder seal material will be required for operation or if operational protection can be implemented to allow for more forgiving lower pressure seal faces. A breakdown of operating pressures based on material is described later in this section.

In terms of the seal cartridge and internal seals, there are a few materials that are considered. Typical materials are 410 stainless steel (SS), 17-4 SS, and Hastelloy. All of these materials have the advantage of being corrosion resistant steels that will limit potential particles and debris that can form inside the seals and potentially lead to damaging the mating and primary rings. Because of the tight clearance, preventing debris is critical. Filtration systems and requirements will be discussed later. In terms of these materials, each has its advantage. 410 SS is the most conventionally used material. It is a high strength, corrosion resistant alloy. It has higher strength than 300 SS and similar thermal growth as many nickel based materials and alloy steels. If higher strength materials are required for higher pressure applications, then the next best choice would be 17-4 SS. If chemical capability is a concern, then Hastelloy is the next common choice. Out of the three, 410 SS is the cheapest and easiest to machine and is what it is most commonly chosen.

The next materials of concern are the secondary seals. These are the dynamic sealing elements described earlier. The various materials that are used based on the design pressure of the seals are shown in Figure 15 through Figure 16. In terms of the dynamic sealing element and also other secondary seals through the seal, two main seals types are looked at: standard O-ring with a V90 shore hardness and PTFE. Both of these types have seals have high hardness that will limit extrusion and also prevent explosive decompression. For lower pressure applications, less than 10.0 MPa (1,450 psi), O-rings will typically be used. There are many material options for O-rings that can be chosen based on chemical compatibility. They are also very compliant to sealing surfaces and require a lower seating force to ensure proper sealing. For higher pressure applications, PTFE is typically chosen. These will also most likely be spring energized to ensure a proper sealing force at low pressure operation and then become pressure energized when pressure is increased in the system. All of these seals do have temperature limitations and extrusion gaps need to be looked at based on those operating conditions. Like O-rings, PTFE has many material fill options, Table 1, that can help its compatibility with various chemicals and also how well they perform at elevated temperatures and pressures. PTFE also has the advantage of not seeing explosive decompression. To sum up material choices based on pressure, reference Table 2.

Table 1 – PTFE Material Options and Operating Limits [3]

Material	Color	Typical Applications & Description	Service Temperature Range (°F)	Tensile Strength in psi at Break	Elongation in %	Hardness-Shore D
Virgin PTFE	White	Excellent for cryogenic applications. Good for gases.	-425 to +450	4575	400	60
Modified PTFE	Turquoise	Lower creep, reduced permeability and good wear resistance.	-320 to +450	4600	390	60
Fiberglass Filled PTFE	Gold	Excellent compressive strength and good wear resistance.	-200 to +575	3480	190	67
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
Carbon-Graphite Filled PTFE	Black	Excellent wear resistance and reduced creep.	-250 to +575	2250	100	64
Graphite Filled PTFE	Black	Excellent for corrosive service. Low abrasion to soft shafts. Good in unlubricated service.	-250 to +550	3200	260	60
Carbon Fiber Filled PTFE	Brown	Good for strong alkali and hydrofluoric acid. Good in water service.	-200 to +550	3200	150	60
Mineral Filled PTFE	White	Excellent low abrasion to soft surfaces & improved upper temperature performance.	-250 to +550	4070	270	65
Aromatic Polyester Filled PTFE	Tan	Excellent high temperature capabilities & excellent wear resistance.	-250 to +550	2500	200	61
Stainless Steel Filled PTFE	Gray	Excellent extrusion resistance at high temperatures and pressures.	-250 to +600	2200	190	72
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
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
Proprietary Low Wear PTFE	Purple	Excellent low wearing material. Kind to soft mating surfaces in the Rb range.	-250 to +550	3470	200	63
Mineral Filled PTFE — FDA compliant for rotary applications	White	FDA compliant materials for sanitary food and pharmaceutical processing.	-250 to +550	2800	250	66
Mineral Filled PTFE — Antimicrobial	White	FDA material with an antimicrobial agent added to prevent bacterial growth.	-250 to +550	2800	250	66



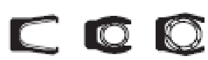
Heel Type	Cross-Section	Class A	Class B	Class C	Class D
Standard		0.008"	0.006"	0.004"	0.002"
Extended		0.012"	0.009"	0.006"	0.002"

Figure 14 – Spring Energized Seal – Extrusion Gaps [3]

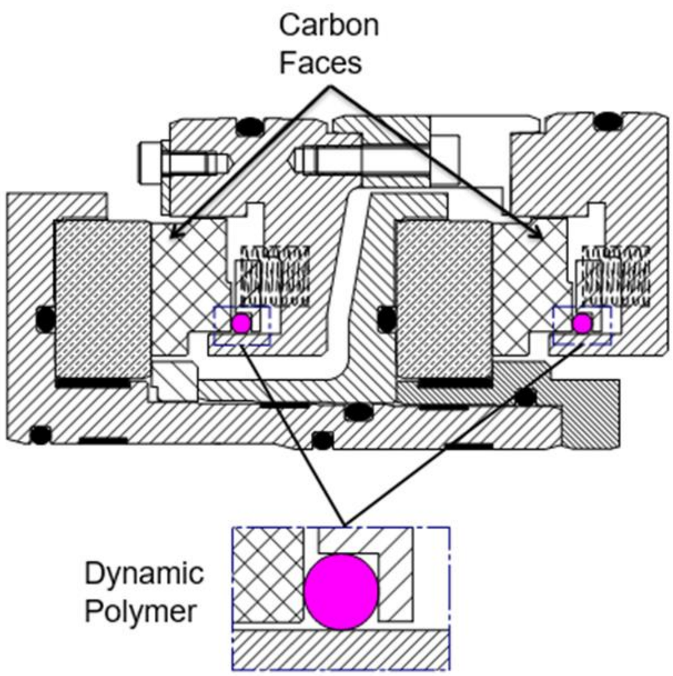


Figure 15 – Low Pressure Design Configuration up to 10.0 MPa (1,450 psi)

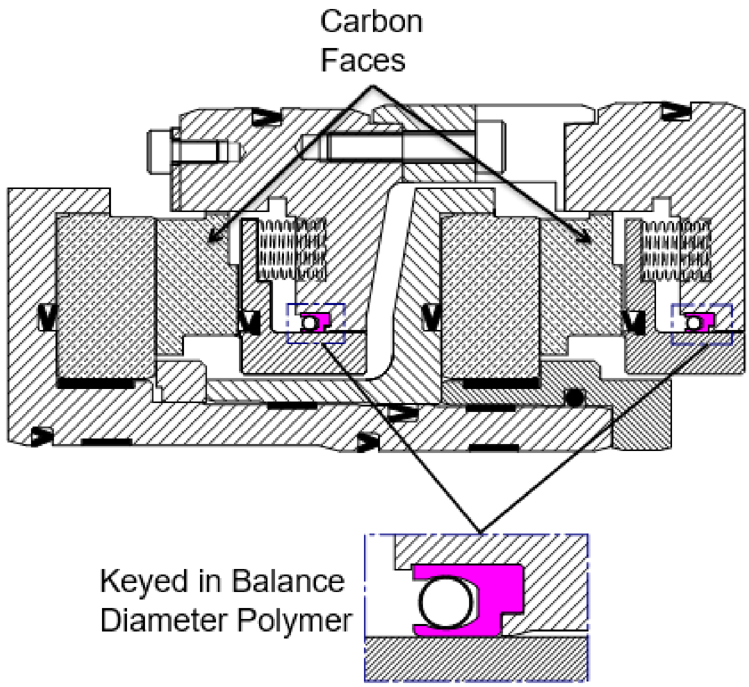


Figure 16 – Medium Pressure Seal Configuration up to 18.0 MPa (2,611 psi)

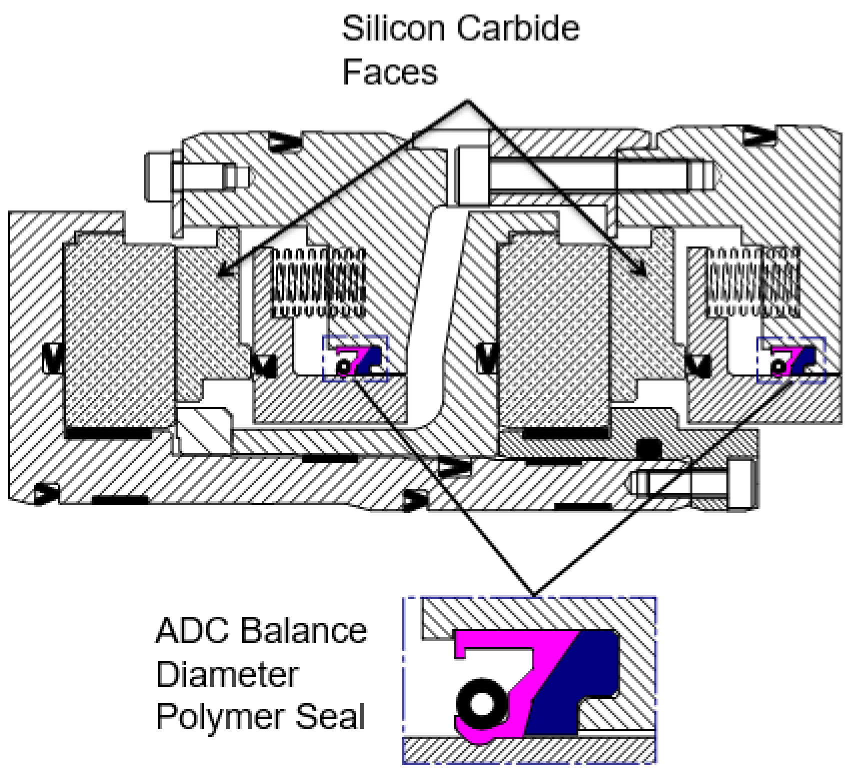


Figure 17 – High Pressure Seal Configuration greater than 18.0 MPa (2,611 psi)

Table 2 - Material Breakdown Based on Operating Pressure

Pressure	Primary Ring	Mating Ring	Secondary Seals	Cartridge
< 10.0 MPa	Carbon	Silicon Carbide	O-Rings	410 SS
10.0-18.0 MPa	Carbon	Silicon Carbide	PTFE	410 SS
>18.0 MPa	Silicon Carbide	Tungsten / Silicon Carbide	PTFE w/ back up	17-4 SS

Leakage Calculation

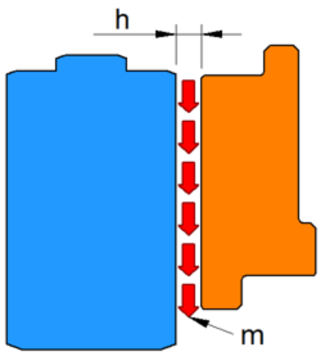


Figure 18 - Leakage Calculation Diagram

$$m \propto \frac{h^3(P_1{}^2 - P_2{}^2)}{\ln\frac{R_1}{R_2}} \tag{1}$$

Where:

m is leakage rate, m^3/sec (in^3/sec)
h is separation gap height, µm (µin)
P_1 is outer pressure, Pa (psi)
P_2 is inner pressure, Pa (psi)
R_1 is inner radius, mm (in)
R_2 is inner radius, mm (in)

Unlike stationary seals, dry gas seals are a controlled leakage seal. Like hole pattern seals, dry gas seals require leakage to properly operate. With no leakage, the seals would not be able to lift off and separate which will lead to seal damage. However, with the ability to run at tight clearances, the leakage is very low compared to other standard rotating seals and shows the clear benefit for limiting leakage and emissions to the atmosphere.

Current Size, Speed, Temperature, and Pressure Limits

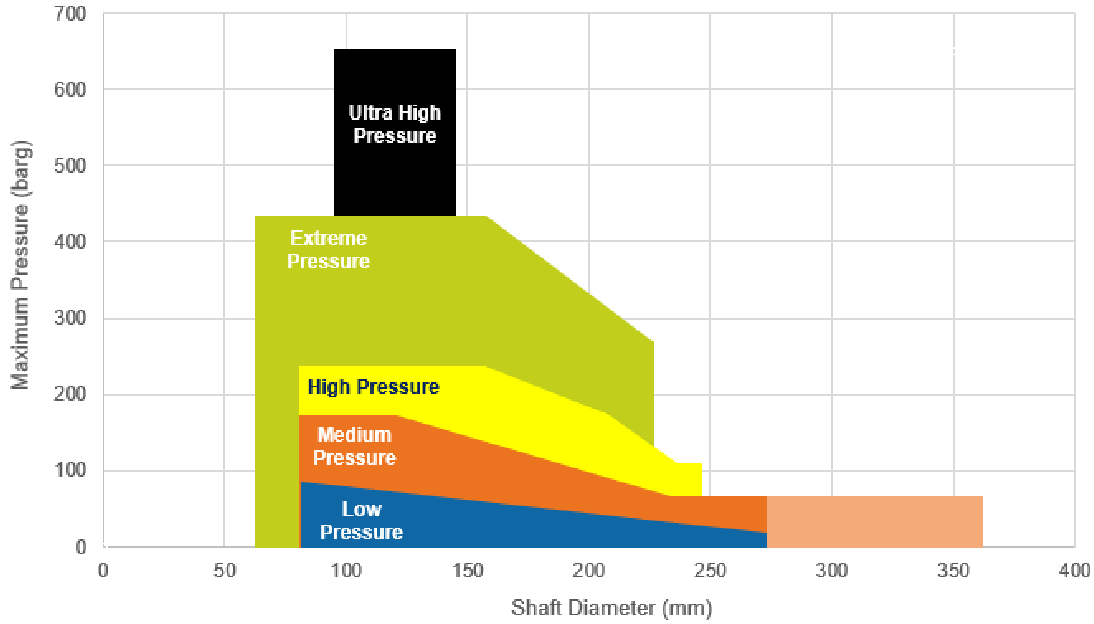


Figure 19 – Pressure Limitations based on Shaft Diameter

Seal performance limits are generally dependant on size. Lower pressure seals cover operating diameters from 40mm ((1.6") to 350mm (13.8"). As pressure requirements increase, seals are generally limited in size to approximately 175mm (6.9") shaft diameter. Small shaft diameter compressors typically run to higher speeds up to 140m/s (460 ft/s). As compressors get larger, the speed requirement is lower.

Table 3 - Design Limits of Various Seal Options

	Extreme Pressure	High Pressure	Medium Pressure	Low Pressure
Max Static Pressure	425 Bar (6164 PSI)	220 Bar (3190 PSI)	180 Bar (2610 PSI)	95 Bar (1378 PSI)
Max Dynamic Pressure	425 Bar (6164 PSI)	200 Bar (2900 PSI)	160 Bar (2320 PSI)	95 Bar (1378 PSI)
Min and Max Temperature	-50°C (-58°F)/ 120°C (248°F)	-50°C (-58°F)/ 180°C (356°F)	-50°C (-58°F)/ 180°C (356°F)	-20°C (-4°F)/ 180°C (356°F)
Max Speed @ Balance Diameter	140m/s (459ft/s)	140m/s (459ft/s)	Typically less than 140m/s (459ft/s)	Typically less than 140m/s (459ft/s)
Uni/BD	Yes with the same performance envelope	Yes with the same performance envelope	BD Max pressure can be lower. Largest BD sizes can have further limitation on speed	BD Max pressure can be lower. Largest BD sizes can have further limitation on speed
Size Range	50mm (2") to 210mm (8.26") Shaft	70mm (2.75") to 230mm (9.05") Shaft	70mm (2.75") to 260mm (10.24") Shaft	70mm (2.75") to 260mm (10.24") Shaft

Separation Seals

Separation seals are outboard of the dry gas seals and are intended to prevent bearing lube oil from migrating into the dry gas seal. Separation seals can be labyrinth seals, contacting segmented carbon ring seals, and non-contacting carbon ring seals. Each type of separation seal has its own set of advantages and disadvantages.

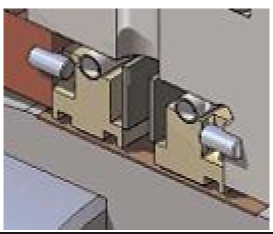
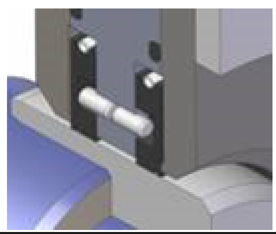
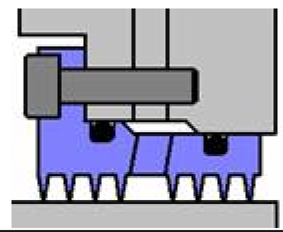
Contacting Carbon Bush	Non-contacting Carbon Bush	Labyrinth
Advantages: Cartridge Arrangement Lowest Flow Rates Positive restriction to oil when not energised	Advantages: Cartridge design Lower dynamic (hot) flow rates than labyrinths	Advantages: Simple design Simple installation
Disadvantages: Wearing parts	Disadvantages: High cold/static flow rates Limited restriction to Oil when not energised	Disadvantages: Very high flow rates No restriction to Oil when not energised
		

Figure 20 - Separation Seal Options

Labyrinth seals were the original separation seals used with dry gas seals. They are inexpensive and highly reliable [1]. However, they have the highest leakage rate of all the separation seals, which will increase cost of separation gas required.

Segmented carbon ring seals consist of two sets of circumferential rings within a housing. The rings are held together with springs, and separation gas is injected between the rings. Segmented carbon rings seals use significantly less separation gas due to the smaller shaft clearance, which can reduce the cost of separation seal gas. Segmented carbon ring seals can be contacting or non-contacting. The contacting shaft seal has no clearance between the seal inner diameter and shaft, while the non-contacting seal has a small clearance. Contacting segmented carbon ring seals have a service life of 3-5 years due to wear, while non-contacting segmented carbon have a nearly indefinite service life [1]. Figure 21 shows a schematic of a non-contacting segmented carbon ring seal, and Figure 22 shows a contacting segmented ring seal.

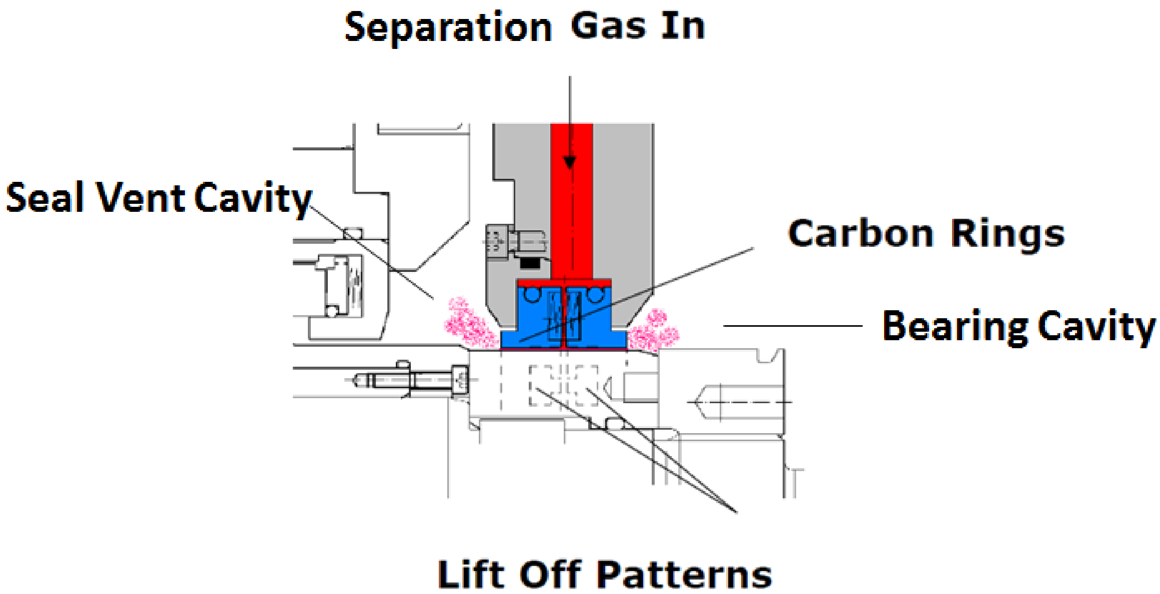


Figure 21. Representation of Non-contacting Segmented Carbon Ring Seal [2]

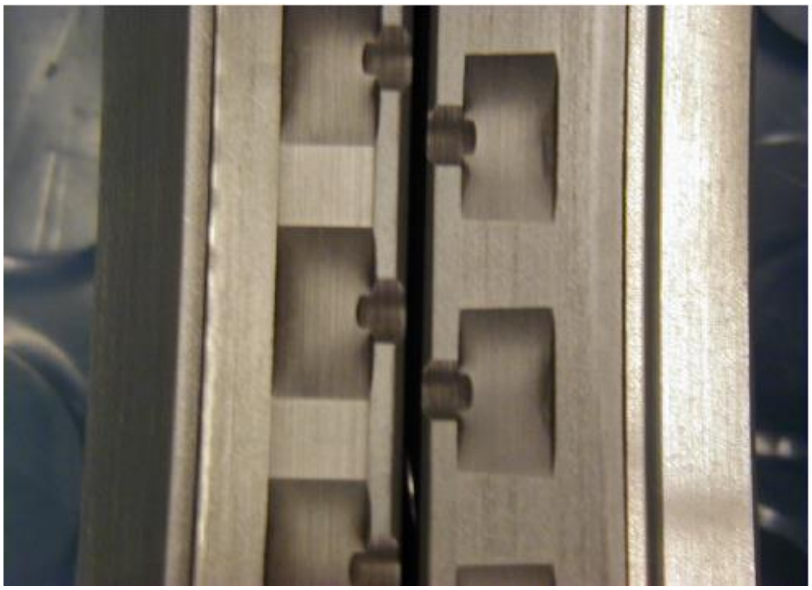


Figure 22. Contacting Segmented Carbon Rings [2]

Separation Seal Supply

Separation seals should have a supply source that, like the seal gas supply, is clean and is pressurized throughout all operating conditions. Nitrogen can be used, but its high cost may be prohibitive. Instrument air is usually used with generally good results [1]. This air is generally fairly clean to start out with, so the filter only really needs to remove particles of 5 µm (0.2 mils) (absolute) and larger [1]. It is important that separation seal supply filters are also coalescing filters to avoid lube oil from the source compressor from entering the separation seal. Like seal gas supply filters, duplex assemblies are recommended for continuous services. Alternately, if a sufficiently high quality instrument air can be used, the need for a filtration system is reduced. To monitor the supply source and filter level, upstream pressure and filter differential pressure should be monitored and have alarms. Separation gas supply pressure should be a start permissive for the lube oil system and should have a shutdown on very low or no separation gas supply pressure. Running the lube oil system without separation gas contaminates the dry gas seals with lube oil, which could lead to failure. In flammable gas applications, the mixture of air and flammable gas in the secondary vent should be reviewed to avoid flammable mixture ranges. Generally, more air is preferred.

Separation gas supply should be controlled via a differential pressure control scheme [1], i.e. the separation gas supply pressure should be a certain amount above the secondary vent pressure. The pressure requirement varies by type of seal, and should be specified by the separation seal manufacturer.

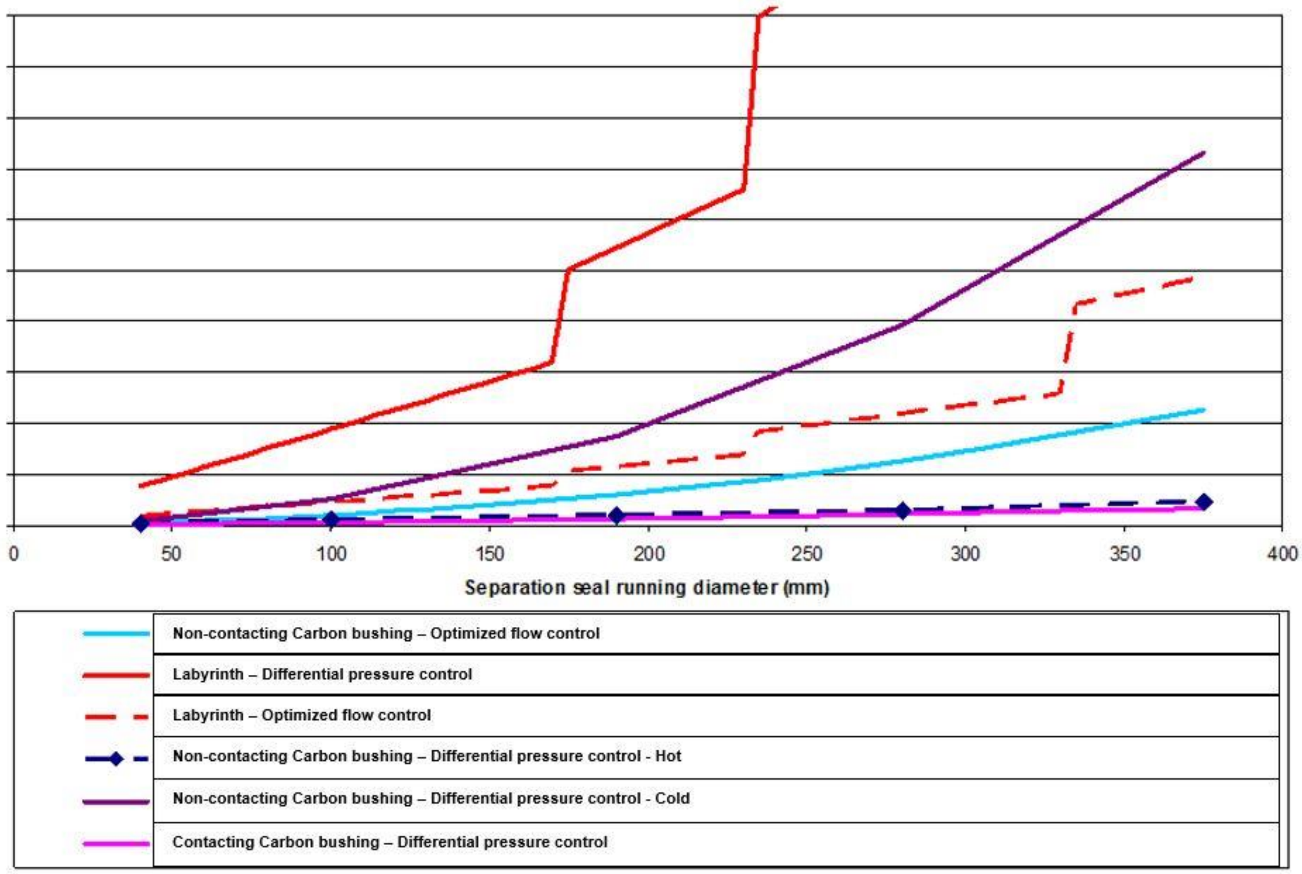


Figure 23 - Separation Seal Consumption Comparison

Seal Testing

Dry gas seals are required to go through various testing to prove out the function of the seal. This includes performance testing, minimum continuous speed testing, and start / stop testing. Each test proves out a difference function of the seal as highlighted in Table 4. Performance testing looks at a few key aspects on the operation of the seal: static test across the pressure range; low pressure dynamic running at ambient and at elevated temperatures; high pressure and high speed; high pressure start/ stops; static hold point at high temperature; and repeat ambient temperature static test. Minimum continuous speed testing consists of continuous running with the IB seals at max pressure and the OB seals unpressurised. This test ensures that the seals can run consistently and reliably at a given minimum speed. Start / Stop testing. Simulates a typical compressor run up and run down cycle. This cycle is very vigorous and can have a long shut down tail where the seal is operating below the lift off speed for a number of minutes before the seal comes to a stop. The test is designed to simulate a typical seal lifetime.

Table 4 - Dry Gas Seal Test Summary

Name	Type of test
Performance Test	Comprehensive testing across the full performance envelope of the seal including: Max Speed, Pressure and Temperature, IB and OB start stops etc.
Minimum Continuous Speed	To confirm the minimum non-contacting running speed
Stop / Start test	Pressurised stop starts to simulate a typical seal lifetime following the stop / start profile of known demanding installations

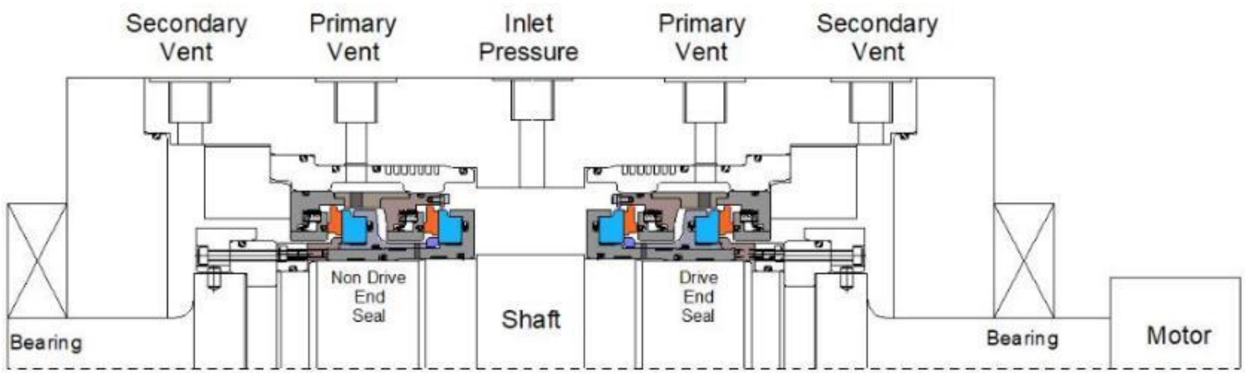


Figure 24 - Seal Test Rig Rotor Layout

Figure 25 - Seal Test Rig (External View)

SEAL GAS SYSTEM COMPONENTS AND REQUIREMENTS

In addition to the seal itself, one of the most critical aspects of successfully running with dry gas seals is the seal gas system upstream of the seals. This system is required to properly supply "Dry" gas to the seals at specified pressure and temperatures to ensure there is proper seal flow and that the seals are not seeing temperatures that are too cold and too warm depending on the process gas and temperature limits. During operation, dry gas seals are typically supplied with pump / compressor discharge. Most of the times, this pressure is much higher than the design pressure of the seals, since the seals are always going to be on the low pressure end of a compressor since high pressure sections are usually in the middle of the shaft to limit blow out forces on the casing. This also allows for more compliant materials as described in the material breakdown of a dry gas seal. This section will describe in the detail the various components of the process diagram shown in Figure 26 and how each component is critical to the process. Typically, a regulator that is set at the design pressure of the supply system will be at the inlet to system to drop pressure from compressor discharge. In addition, feed lines can also be tied to other compression stages other than the max pressure in the system.

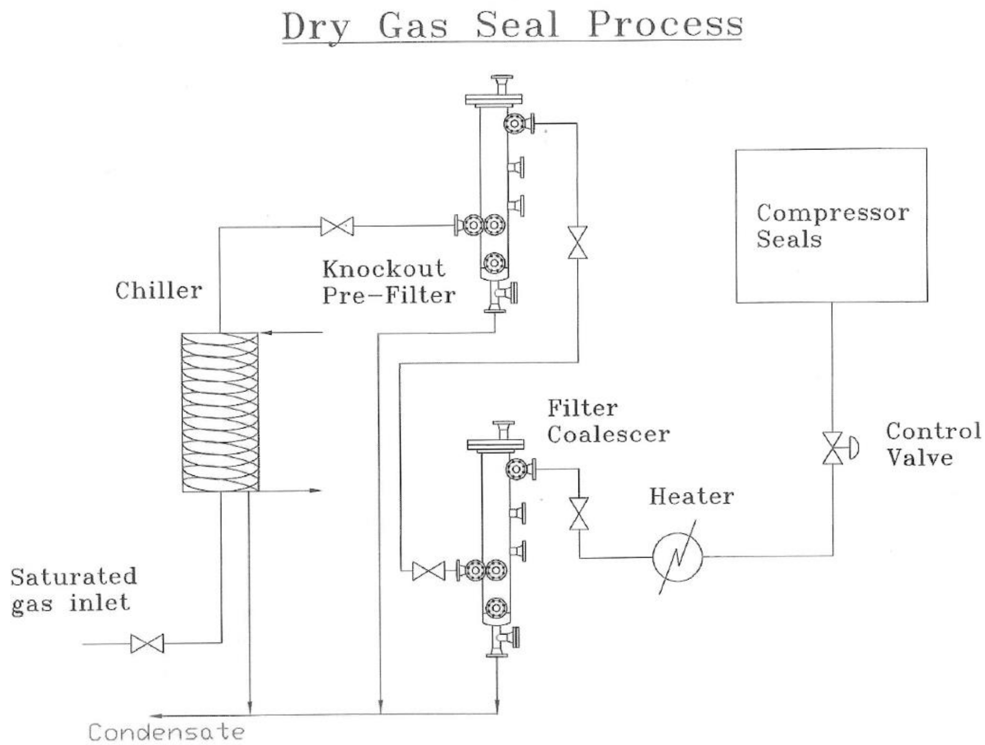


Figure 26. Representative Dry Gas Seal Supply Process Diagram (Instrumentation Locations Not Shown) [2]

Cooler

For certain applications, coolers are required upstream of the seal supply to reduce temperature to 16.7ºC (30ºF) below saturation temperature for subsequent separation and heating. Because of the process gases that are running through the compression systems are not always dry gas, it is important that potential liquids are removed from the supply gas before entering the seal system. By cooling the flow below the saturation temperature, the gas and liquid can be more easily separated. These coolers can either be water or air cooled, depending on necessary 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 dealt with to ensure proper effectiveness of the coolers. A control system will typically be designed to measure temperature downstream of the cooler to ensure the flow is below the saturations temperature before entering the filters and separators. Flow of the cooling water / air or the process flow can be adjusted to bring the temperature down or raised as needed.

Figure 27 - Example Coolers for Dry Gas Seal Supply Systems

Filter / Separator

The next components after the cooler are the filters and separators. As mentioned previously, removing particles and liquids is critical in ensuring successful operating of the dry gas seals. A separator is typically used when a system has a higher chance of liquid. For any filters and separator used in a DGS system, they must be sized adequately for range of flows. A liquid level indicator should be used for monitoring and drain control. To prevent damage to the filter element, a Pressure Differential (dP) sensor should be installed. 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 filet is installed without the need to remove the entire body.

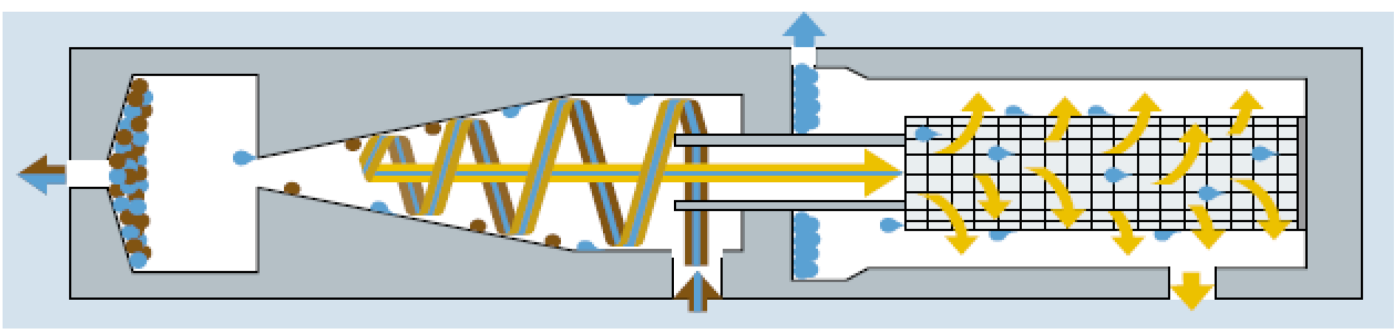


Figure 28 – Example Cyclone Filter Cross Section

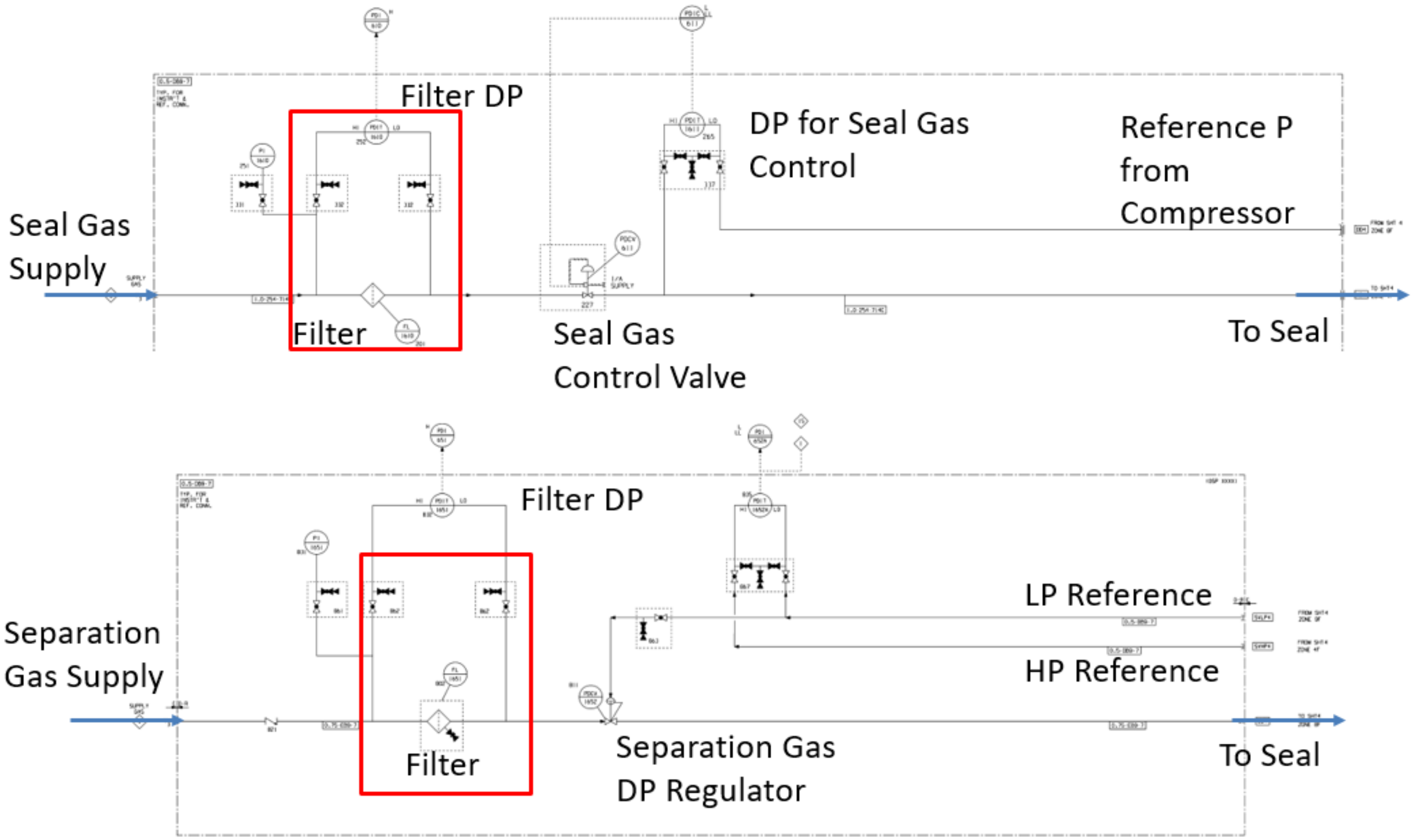


Figure 29 – P&ID with Filter Locations

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
- Must have drain valves w/automatic control or regular inspection/draining
- Level transmitter may be needed
- Monitor DP (alarm recommended)
- Prefilter 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 & Heat Trace

Depending on the process gas, heaters could be required to ensure minimum supply temperatures. It is important to note that as the supply gas drops pressure across the seal face, it will go through isenthalpic expansion and can potentially cool down significantly. This has to be considered when looking at various materials to ensure there won't be any ice formation.

Booster

Figure 30 – Example Booster Pump

Another important feature in any DGS system is the booster pump. While operating, DGS are supplied from the compressor / pump discharge which leads to the lowest leakage / mass into the system. 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 buffer gas to prevent potential particle contamination from the loop as it vents through the seals. Booster pumps should be sized for seal gas normal flow, and in some cases, multiple pumps could be required depending on gas source or needed supply pressures. Cycle life needs to be considered since these pumps will most likely see many quick transients due to the need for quick supply during trips. When choosing pumps, it is important to consider potential lube oil contamination from the pump into the supply gas.

Instrumentation

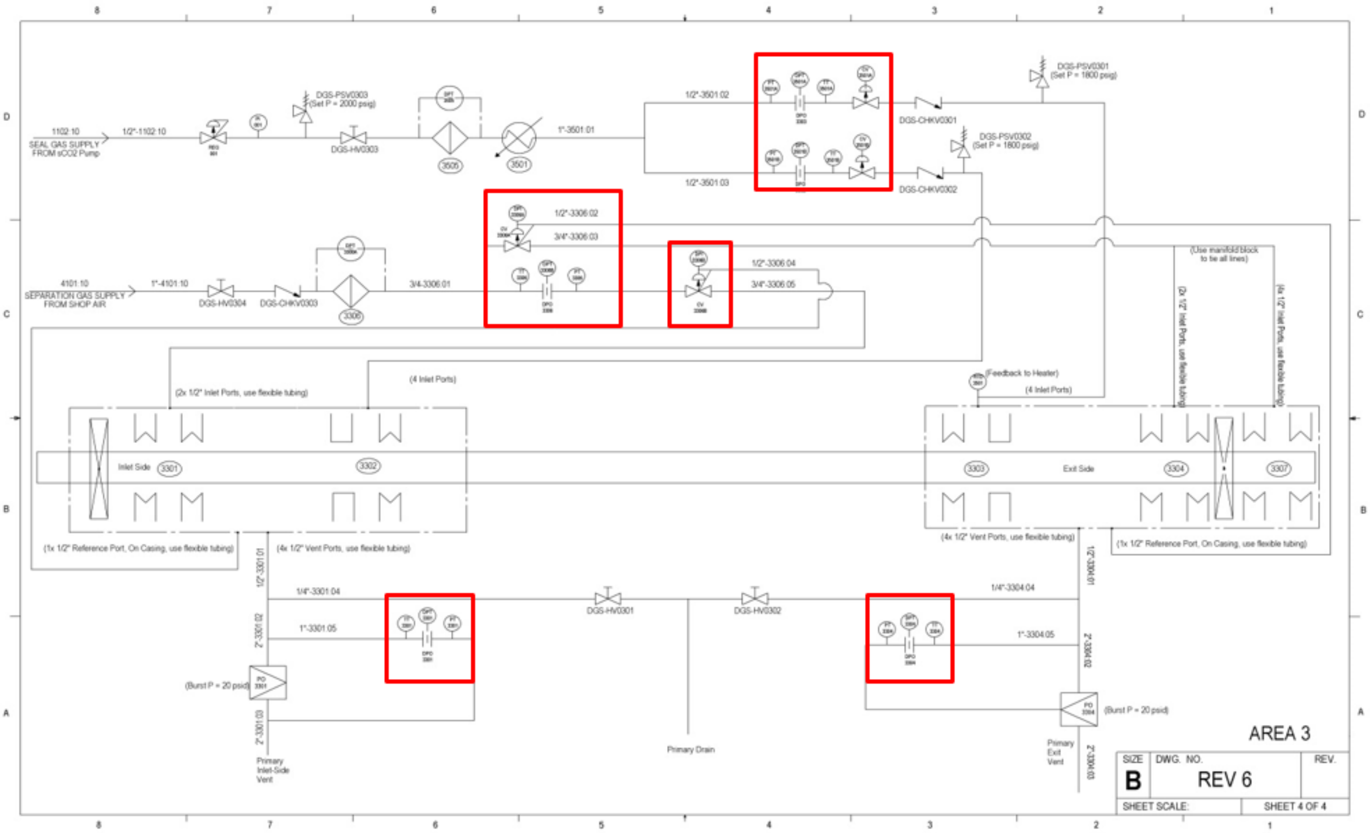


Figure 31 – Instrumentation Layout for a DGS Panel

Vents (Sizing and Safety)

Since most of the supply flow runs across the labyrinth seal into the process, the seal vent has relatively low flow. Vent lines should have a low point drain that needs to be checked regularly to ensure that liquid isn’t backing 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 [1].

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 [1]. 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. Typically, this burst disk is set to burst at a 0.14 MPa (20 psi) differential pressure, but this may vary based on the flare system used [1]. 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.

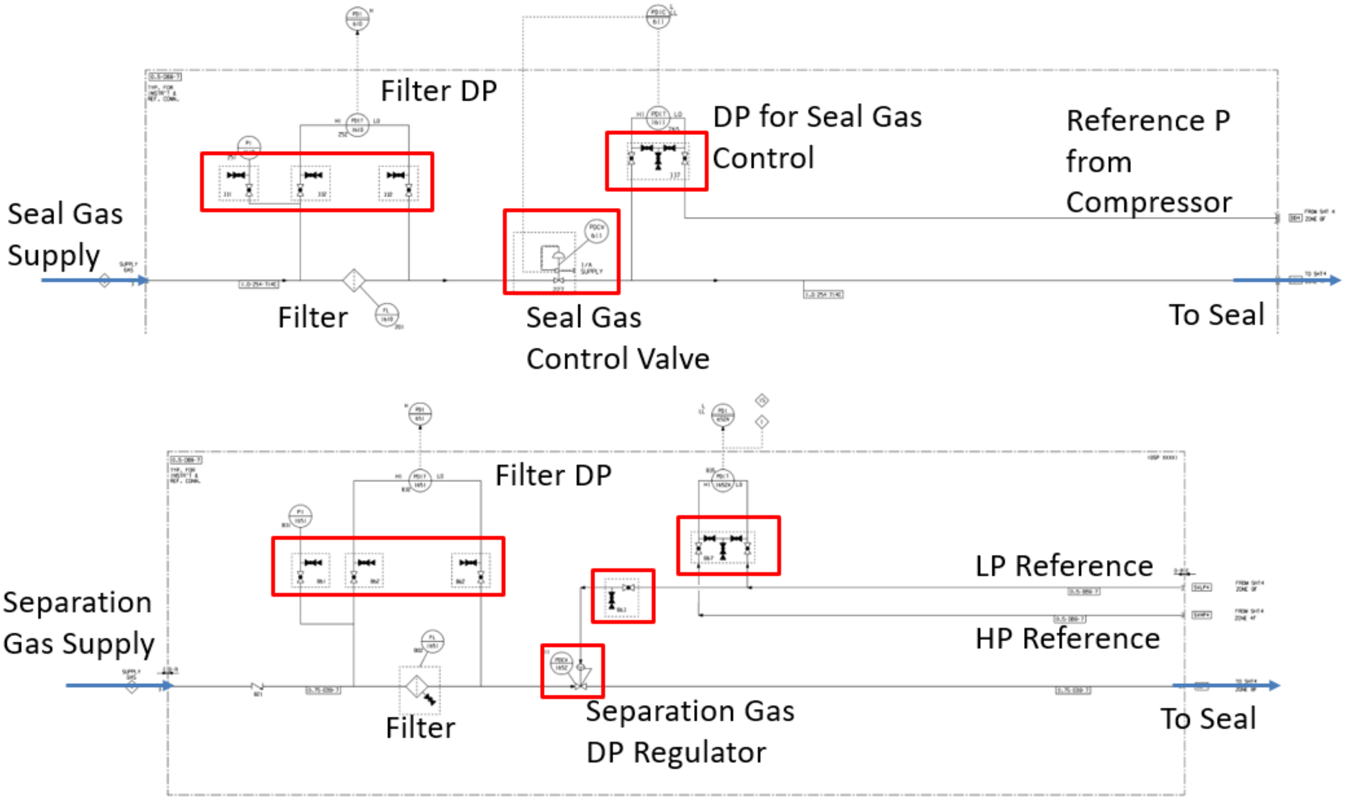


Figure 32 – Vent Locations

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

- Block and full flow bypass valves to isolate each conditioning component
- Check valves on supply line
- Control valve for supply control (appropriate size/controllability)
- Use resilient seat materials to reduce risk of ED
- If internal mechanisms fail, both paths should not be blocked
- Should be rated to highest pressure the entire system could potentially see (consider transients and failures)
- PSVs as needed
- No shutoff valves downstream of supply line instrumentation

Seal Gas Control

It is imperative that the seal gas supply feeding the seals has 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 dry gas seal, and it must be slightly higher in pressure than the required sealing pressure for the entire compressor operating range, including transients. There are no specific requirements for the value of pressure difference, but Stahley [1] recommends using 0.34 MPa (50 psi) in the absence of any other specifications. Another good rule of thumb is that the process labyrinth seal should have at least 9.8 m/s (32 ft/s) flowing across it. A pressure gage at the start of the seal gas supply line is recommended for monitoring supply pressure [1]. It is recommended that seal gas supply pressure be a start permissive, and also have an alarm on low pressure and a shutdown on very low or no pressure. The supply gas source 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.

As can be seen from the simple process diagram in Figure 26, after the take-off the seal should go through an optional knockout filter, a coalescing filter, a heater, and then have a control valve leading into the seal. A chiller may be necessary to dry out a saturated supply. The knockout filter is only required if the seal supply source is known to be particularly dirty. Otherwise, the coalescing filter should be able to handle the gas cleaning and drying. The heater is only required in some applications where the temperature across the seals may drop below the dew point of the gas or is likely to form hydrates.

To ensure supply gas is clean, a coalescing filter should be included. It is recommended that this filter can remove particles of 3 µm (0.1 mil) absolute and larger [1]. All large equipment and most fittings and instrumentation the supply line should be made of stainless steel. Filters, specifically, should be at least 316/316L stainless steel. For continuously operating equipment, filters should be duplex assemblies so that the filter can be changed without having to take the compressor offline. Coalescing filters should be used to remove any liquid mist that may be in the supply line. A drain on the filter should allow for removal of liquid build-up. It should be noted, however, that coalescing filters can only handle liquid mists, and a slug will not be caught. A liquid slug is almost guaranteed to cause seal failure. A filter should have a differential pressure gage with a high alarm it to alert the operator when the filter is full and needs to be replaced. This alarm should not be ignored.

The potential for liquid condensation should be minimized. The Joule-Thomson effect is when an expansion in gas volume induces a pressure drop, and the temperature decreases. The gas could potentially drop below its dew point, causing liquid dropout. Since the largest pressure drop in the seal gas supply systems is across the seal faces, there is a distinct chance that the gas could drop below its dew point and have liquid dropout. Although the seal faces heat up while running, at low speeds, the faces may not become hot enough to prevent liquid dropout. It is recommended that a thermodynamic study be completed to determine how hot the gas entering the seal needs to be. API 614 recommends a supply temperature sufficient to maintain a 20 °C (36°F) margin above the gas dew point throughout the seal [4].

Seal gas supply can be controlled via differential pressure control or flow control. Differential pressure control is where the seal gas pressure is maintained above a reference pressure by a certain amount, such as 0.07 MPa (10 psi). The reference pressure is typically the compressor suction pressure (the discharge side often has a balance piston seal inboard of the dry gas seal that is tied to suction pressure via a balance line). Flow control systems operate by regulating control through an upstream orifice. This should be an automatic control system in order to maintain sufficient flow during process upsets or compressor transients. In some cases, manual needle valves are used, but this configuration may be less robust if the supply gas source has varying pressures. The flow should be regulated in such a way that there always be at least 9.8 m/s (32 ft/s) flow velocity across the process labyrinth seal [1]. This number is used because it is twice the experience-based industry standard for labyrinth seal operation. Flow control schemes are better at maintaining the minimum flow velocity while using less seal gas than differential pressure controllers [1].

Tubing, Drains, Hand Valves, etc.

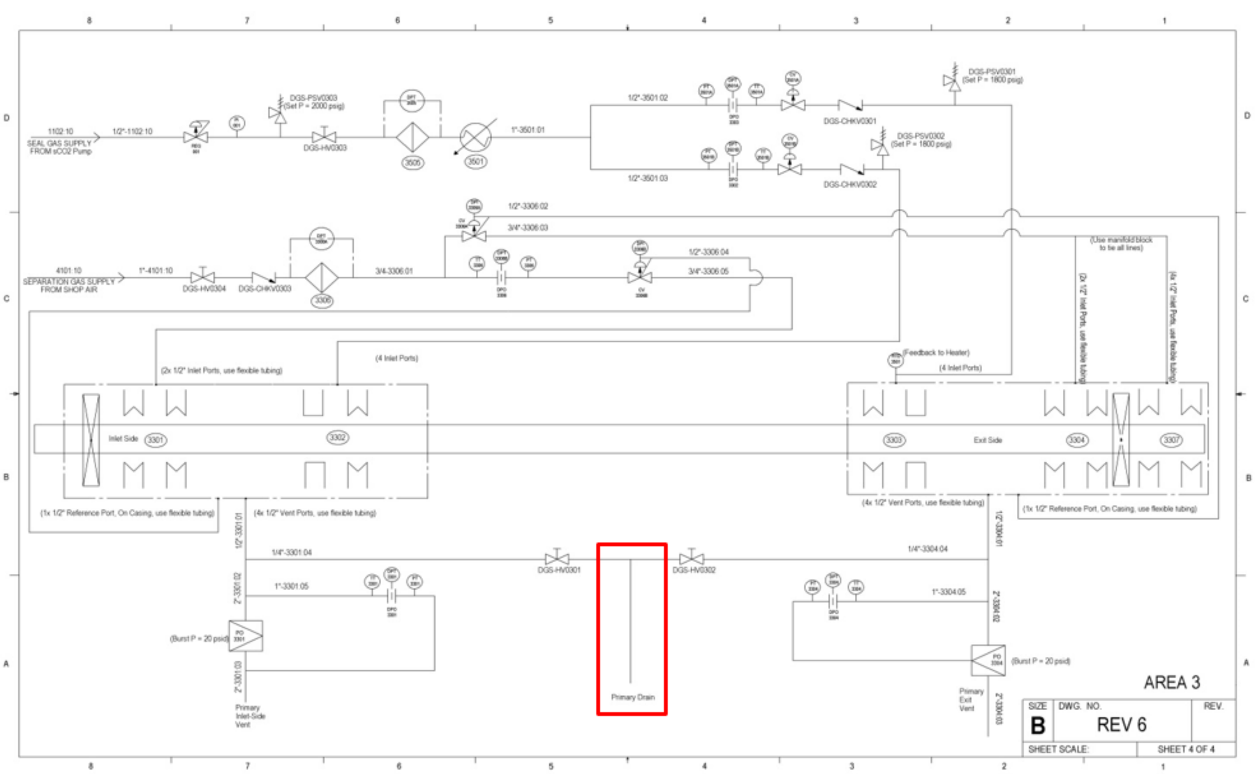


Figure 33 – Drain Location in DGS Panel

For any DGS system, drains should be placed at low points in the supply and vent lines. For most applications, a simple hand valve cab be installed that can be checked safely during operation or maintenance to either drain liquid from the loop or also to see if there is any liquid build up that might indicate an issue in the overall system. Drain capacity should exceed normal expected condensation rate or higher condensate collection rate (whichever is higher) with a sufficient safety margin. If continuous liquid is expected, a restriction orifice for continuous draining should be implemented. Manual drains may require a restriction orifice for high DP applications

FAILURE MODES AND STATISTICS OF DRY GAS SEALS

Analysis of dry gas seal failures was completed by approaching 11 companies and asking if they would be willing to contribute failure data for the project. The companies approached included both end users and OEM. Care was taken to cross check all data collected to ensure that failures were not double counted in cases where both the end user and OEM reported on the same failed seal. To protect the companies involved, all failure data was placed into an anonymous database for analysis. A few other reported cases of failures were taken from available literature, previous API surveys, and publications.

Eight companies agreed to participate in the study. Data was collected from these companies via phone calls, failure reports, and internal failure databases put together by the participating companies. The goal was to gather as much information as possible, while minimizing the work of the participating companies. Data for 194 failures was collected into the database, created a sample set many times larger than any other previously published study. Of this database, 144 provided root causes.

A few other sources provided information about failure case studies, and this information was incorporated into the database. This includes case studies from Stahley's book [1] and a paper by Stahley [5]. An article from CompressorTech2 [6] with a case study was included, as well as results from an API survey on dry gas seal reliability [7].

It should be noted that this data was analyzed "as-is." This means that root causes provided by companies were assumed to be correct. Care was taken to ensure that the root cause was reasonable given the diagnostic findings and conditions. Whenever possible, clarifying questions were asked of companies providing data to increase the fidelity of the database. In some cases, multiple root causes were provided by the company. Therefore, the percentages provided below will not necessarily add up to 100%.

Data Classification

The root causes of the failures were broken into the following categories: process contamination, supply contamination, lube oil contamination, and installation/geometry problems. Within each of these categories, further categories were identified based on the specific root cause. The resulting distribution of root causes and subcategories are shown in Table 5 and Figure 34. It should be re-emphasized that some failures had multiple root causes; therefore the percentages below do not add up to 100%. Details regarding failure data for each of these categories are discussed further in the following subsections. Percentages of subcategories are based on the data set of the subcategory only (i.e. 64% of failures in the supply contamination category were due to liquid contamination).

Table 5 – Breakdown of Root Cause Categories and Subcategories

Root Cause			
• Liquid Contamination (64%) • Other Contamination (28%) • Filter Overload (8%)	• Geometry 62%) • Installation (38%)	• Heavy Hydrocarbon Contamination (25%) • Process Gas Contamination (41%) • Insufficient Seal Gas (25%)	(not broken into subcategories)

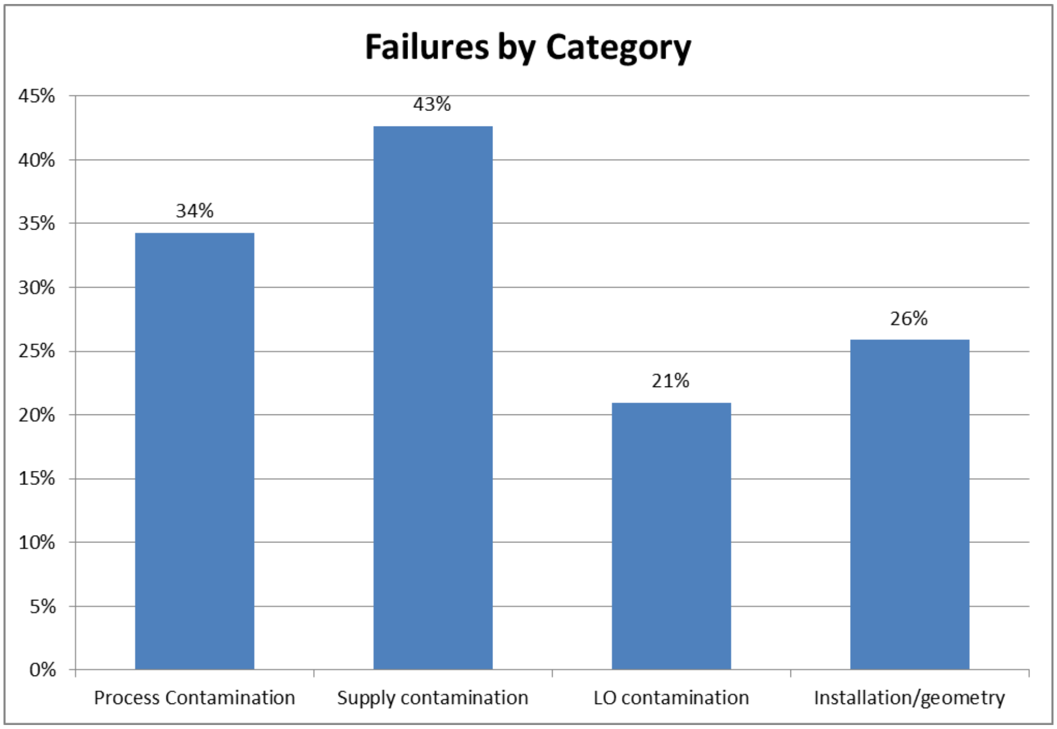


Figure 34 - Percent Failures by Category

The following sections will outline the findings of each category and subcategory, followed by a study of other parameters, such as application and state at time of failure.

Supply Contamination

The results show that supply contamination is by far the highest cause of failures in the dry gas seals studied. 43% of the collected failures were a result of supply contamination. 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. 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 614 [4] 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 [1] recommends a computer model of the temperature and pressure drops across the system be used 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.

To avoid liquid contamination in systems prone to liquid dropout, the addition and 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.

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.

Filter Overload

Another cause of supply contamination is filter overload or degradation. 8% of reported supply contamination failures were due to filter overload or degradation. 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 and 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 impending to other 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.

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 35). 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.

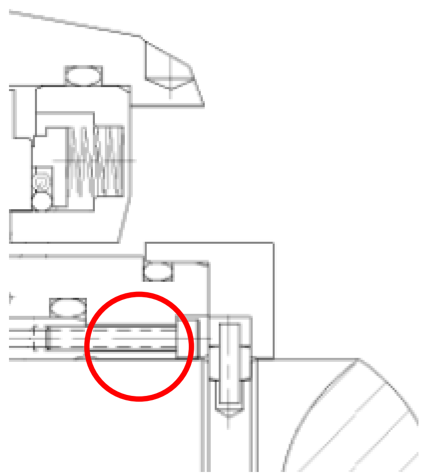


Figure 35 - Drive Pin Alignment Feature [2]

To avoid drive pin issues, the drive pin dimensions can be checked prior to installation of the seal. Since this may impractical, it may be easier to perform a few checks during installation suggested by Stahley instead. These test can verify that the seal is able rotate properly and move axially [1].

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.

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 [1] 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.

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 14.7% 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.

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 mating ring. This is caused by insufficient separation seal supply gas or lube oil flow that is excessively higher than design. Seal 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.

Seal Lifespan

Although not all reported failure cases have associated running hours, the breakdown of the 21% of cases with operating life data is shown in Figure 36. The failure data in the plot below are expressed as percentages of failure cases with operating hour information.

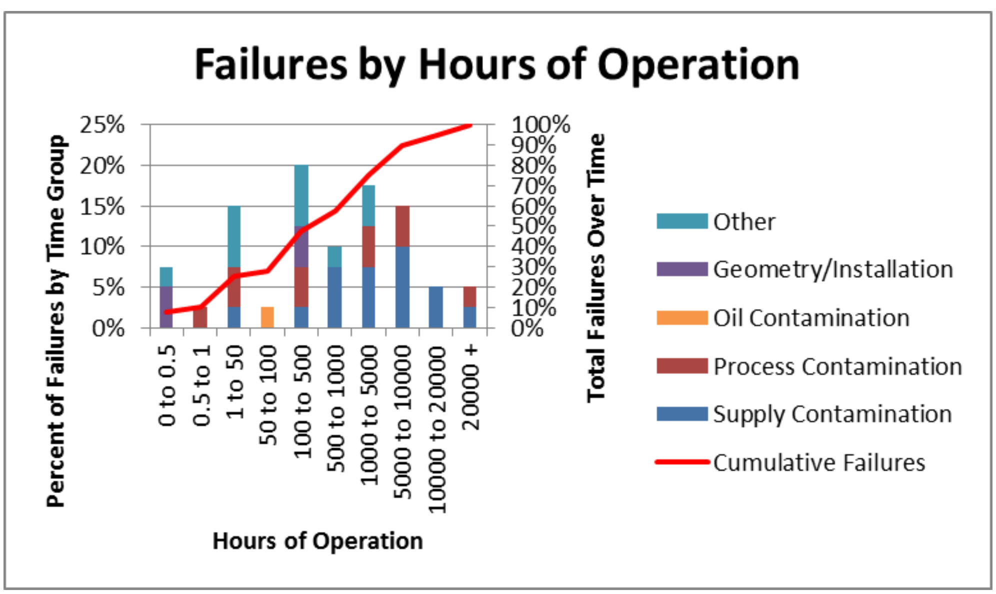


Figure 36 - Hours of Seal Operation Prior to Failure

Geometry and installation issues usually manifest themselves within half an hour of startup. While this 30 minute rule cannot be applied universally, it does appear to suggest that a failure within the first 30 minutes is very likely a result of geometry or installation.

Also of note is supply contamination. In the data set, supply contamination does not occur until after a span of running time. While seals are designed to last indefinitely when given perfect operating conditions, operating conditions in the field are almost never perfect. The data suggests that seals generally do not fail from supply contamination until after at least 50 hours of running time. This suggests that seals do not always fail the moment they become contaminated; rather, seals are sufficiently well-designed to handle a certain amount of contamination to account for the less-than-ideal situations encountered in the field.

In an effort to investigate failure causes, rather than failure rates, this study only collected data from cases where failures occurred, and therefore this dataset cannot be used to determine a mean time between failures for the entire dry gas seal population. Seals ope8rating in a properly designed and maintained system can exhibit exceptional reliability, as the O-ring is the only element of the seal subject to normal wear in perfect conditions. Because the data set collected consists entirely of failed seals, it is clear that the cases studied are representative of less than ideal operating conditions, as it is not representative of all seals collectively. To emphasize, these data are NOT stating, for example, that 15% of ALL seals will fail in 1-5 hours of operation.

Based on verbal feedback from participating companies, a good seal in normal conditions generally lasts 30,000 to 50,000 hours or more, or 5+ years. This is not an absolute, however, as there are the occasional anecdotes about a seal lasting for over 20 years.

Operational Status at Time of Failure

Since Figure 36 suggests that startup and commissioning are common times for seals to fail, it is worth investigating at what point during operation the seal failed. The seal can potentially fail at normal operating conditions, startup, shutdown, or commissioning. It should be noted that Figure 37 represents only failures where operation state was specified, which was 40% of the failures collected.

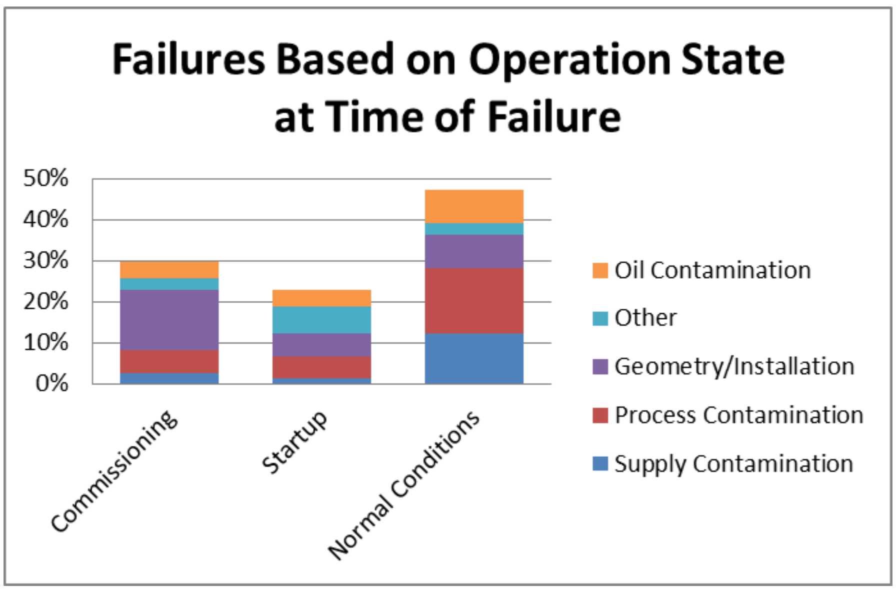


Figure 37 - Failures Sorted by Operation State at Time of Failure

In Figure 37, geometry and installation problems are generally seen during commissioning, which includes factory acceptance testing, and startup. Supply and process contamination is most commonly seen during normal operating conditions. Seals are unlikely to be reported as failing during shutdown. When combined, commissioning and startup make up a significant portion of the failures. This is likely due to off-design conditions during transients, which could provide inadequate seal supply flow. Most of the failures reported are from normal operating conditions, which may be due to seals spending more time in normal operation than in startup, shutdown, or commissioning.

Failures Grouped by Application

Communication with participants has indicated there is a wide variety of seal sizes (both in physical size and power of the associated machinery) in the data reported, but there is insufficient specific data on this to break failures down into size groups. However, 84% of the data provided included a specific application. These applications, as provided in the failure database, have been sorted into categories and associated failures.

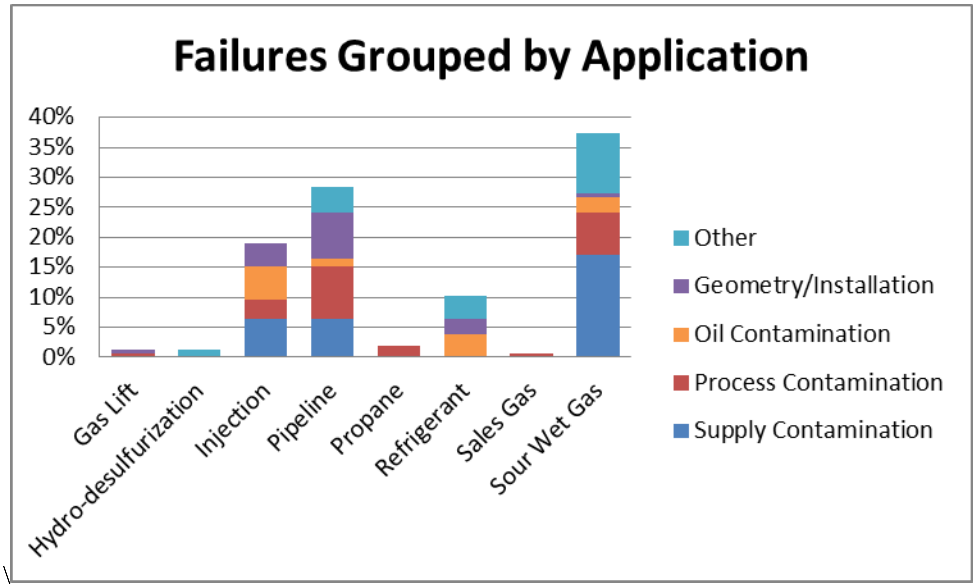


Figure 38 - Failures Grouped by Application

From Figure 38, supply contamination appears to be a large problem for injection, pipeline, and sour gas compressors. Sour wet gas applications may be dirtier than the other applications studied, which could explain why there are more reported supply gas contamination failures in that category.

Failures by Year

Of the data collected, 68% of the failures collected included information on the year the failure occurred.

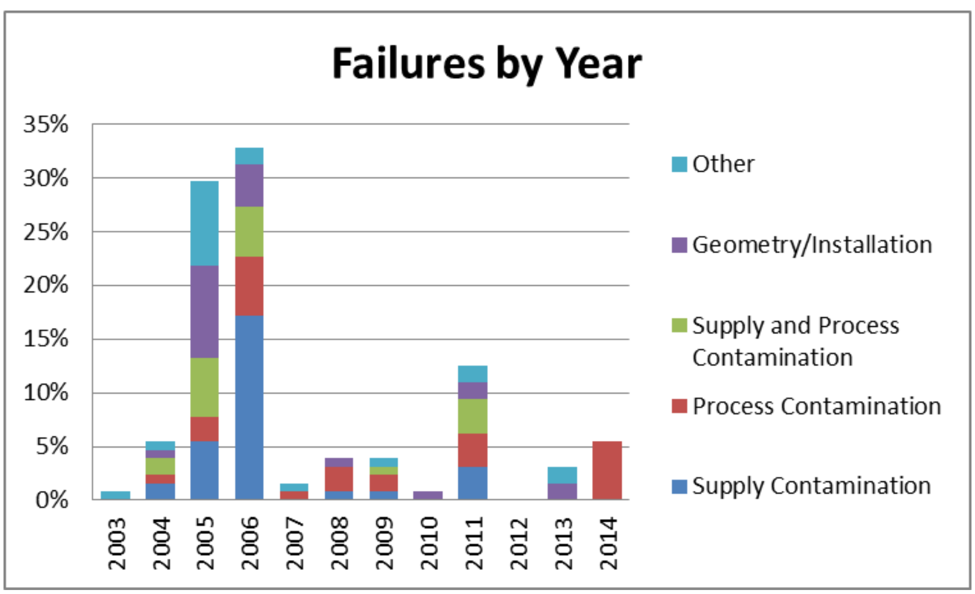


Figure 39 - Failures by Year

It must be noted that Figure 39 should not be interpreted as a reliability trend for the entire dry gas seal population; it is simply representative of data gathered (e.g. it cannot be assumed that there were no dry gas seal failures anywhere in 2012). However, some trends may still be identified. Supply contamination appears to be a smaller problem in 2007 and onwards, suggesting that strategies to avoid supply contamination were developed at that time. Reducing the chance of supply contamination is arguably one of the easiest ways of improving dry gas seal reliability. Subsequent years saw a significant decrease in seal failures.

It can also be seen that geometry and installation issues have become less common through the years. This suggests that dry gas seals are an evolving technology, meaning that continued collaboration between end users and original equipment manufacturer can provide valuable information to continue improving seal reliability. Studies such as this are invaluable to all interested parties in providing data-based guidance for improved reliability.

CONCLUSIONS

With years of proven success and experience, dry gas seals continue to be a good solution for advanced sealing technologies on rotating machinery to limit leakage. With a growing concern for emissions, being able to limit leakage to the atmosphere effectively is very important for both the environment and operators. By limiting leakage out of the end seals, there is less need to capture the emissions and pay for recompression and separation.

It is important to understand how a dry gas seal operates to know how a control and operating system needs to be designed and set up. This tutorial provides a summary of key aspects and features to consider when laying out the overall system. Because of their tight running clearances, just like bearings, DGS require features to prevent contamination that can lead to damage of the critical seal faces.

In addition to current applications, further research and development is being done to advance DGS design so that it can be used in more advanced machinery. This includes high temperature and higher pressure operation along with larger diameters. Extending the range of operation will be beneficial to all industries as there is a continuous need to increase power and reduce emissions.

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