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SPECIALREPORT

PLANT SAFETY AND ENVIRONMENT

Dry gas seals and support systems: benefits and options

Follow these guidelines for proper specification, installation and maintenance

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Since their introduction in the mid '70s, dry gas seals have revolutionized axial and centrifugal compressor sealing. Dry gas seals are installed in virtually every new compressor. Compared to traditional wet sealing, dry gas seals offer the following commercial advantages:

- On shutdown, pressurized containment of process gas is possible, reducing plant flaring
- Prevents oil contamination of compressed gases
- Reduced power consumption (conservatively >1%)
- Increased process train reliability/availability
- Predictable rotor characteristic with consistent stability
- Reduced space requirements
- Prevents reservoir oil contamination and associated bearing failures in sour or saturated gas applications.

Additionally, plant safety is improved by eliminating the following problems associated with wet seal systems:

▶ Seal oil pump failure with the consequent possibility of gas blowout

▶ Seal blowout caused by an oil trap float failure, a clogged coalescer or a sticky buffer gas regulator

▶ Thrust bearing failure resulting from excessive buffer gas differential pressure

▶ Possibility of a lube oil tank explosion. Gas breakout from the seal return oil can increase the flammable gas concentration above the lower explosive limit (LEL) when the oil reservoir breather is clogged or overwhelmed from excessive gas leak at the seals. One compressor supplier has reported 26 such reservoir explosions.

Life cycle cost comparison: dry gas vs. wet seal systems. If technical review confirms the suitability of dry gas seals for a given application, then consider the following factors for a comparative life cycle cost analysis between the dry gas and wet seal systems:

- First cost
- Plant availability and its impact on throughput
- Cost of lost process gas
- Operating utilities and maintenance costs.

First cost: Wet seal systems comprise seal cartridges, seal oil pump, blower, degassing unit, seal oil reservoir, overhead seal oil vessel, connecting piping, and a control and monitoring panel. A dry gas seal system requires only seal cartridges, filtration and a control and monitoring panel.

For a land-based plant, a wet seal system would cost on the order of \$500 K compared with \$250 K for a dry gas seal system in a single-casing compressor with a 2–3 in. (50–75 mm) diameter shaft. The high cost of real estate and maintenance in an offshore application would automatically justify a dry gas seal application.

Availability and plant throughput: In process plants with 330 to 340 days of planned availability per year, any main process compressor trip (usually unspared) translates into a loss of production. In general, wet seal systems are attributed as the root cause of 75% of motor, and 30% of gas/steam turbine-driven compressor downtime. With a realistic availability of 98% and 95% for motors and turbine-driven compressors respectively, this equates to a 1.5% production downtime due to wet seal systems. Historically, dry gas seals have exhibited some infant mortality, with a subsequent 99.9% availability. Assuming a 1:2 plant variable vs fixed production cost ratio and 15% rate of return on investment, the benefit from five days of extra production due to increased reliability from dry gas seals equates to 0.66 × 0.15 × 5 × daily production value.

Compared with a negligible 3–4 hp (2–3 kW) frictional loss in dry gas seals, up to 1% of compressor shaft power is lost in wet gas seals. In the main process compressor application, 1% extra power equates to 3.65 days of additional production.

Lost gas: Dry gas seals typically leak 5–10 scfm/seal (8–16 Nm^3/hr/seal) compared with an aggregate leak of 100 scfm/seal (161 Nm^3/hr per seal) to flare for a wet seal. For natural gas service at the current price of \$4/MMBtu* (1,000 Btu/scf*), 180 scfm of saved gas per compressor body results in a saving of \$1,100 per day. Similar calculations can be carried out for other gases.

Operating and maintenance costs: Dry gas seals use pressurized nitrogen/air as buffer and separation gases. Wet seal systems employ a seal oil pump and degassing blower, and in sour gas applications

* 1 MMBtu = 1.055 Giga Joules; 1 Btu/scf = 37.25 kJ/m^3.

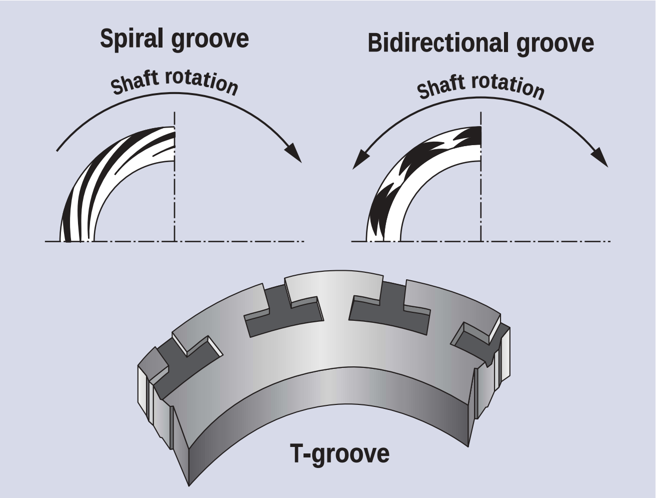

FIG. 1. Some of the many seal face patterns.

seal oil changeout is required every 4,000–8,000 hr. Considering 67 hp (50 kW) of power consumption by the seal oil pump and blower as the major operating cost difference per compressor casing, at 4.5 cent/hp-hr (6 cent/kW-hr) a dry gas seal would save $24 K/yr in utility cost. Labor and material savings, based on five days of reduced maintenance support per year by a team of six at $1,000/day (including materials) equates to $30 K annually.

Other savings: If applicable, also evaluate the following savings:

▶ Impact of a 6-yr operating cycle between overhauls for a dry gas seal compared with a 3-yr cycle for a wet seal

▶ Savings from a process loop inventory not flared on the spurious compressor trips (say, one trip per year) due to the pressurized hold capability of dry gas seal.

Assuming 20-yr equipment life and a 15% internal rate of return, the present value of benefits from a dry gas seal installation is the total annual savings × 6.26. For new installations, add the first cost difference to arrive at total life cycle benefit from ownership of dry gas seals. For a retrofit, deduct the first cost to arrive at the life cycle cost savings and justification.

Dry gas seal principles and construction features. The sealing mechanism is comprised of two rings: a stationary (plain face) and a shaft-mounted rotating face with "etched in" grooves. On rotation, the groove continuously pumps the sealed gas into the reducing cavity. The high pressure generated creates a thin gap between faces allowing minute leakage of 5–10 scfm (8–16 Nm^3/hr). Separation of the stationary and rotating seal faces, also called "lift off," occurs between 150 and 350 rpm. Manufacturers have developed numerous rotating seal face patterns: radial, spiral, wavy face, V groove, T slot, etc., to achieve the desired sealing properties. Some of these are shown in Fig. 1.

A number of factors determine the lift off speed, such as seal tip speed, groove depth, rotating face pattern design, dynamic and static sealing pressure, stationary face spring closing force, compressed gas molecular weight, slow- or high-speed application, slow roll after machine shutdown, etc. Normally, seal tip speed is 90 fps (27.5 m/s), though seals up to 650 fps (198 m/s) are operating successfully.

Based on the seal groove geometry, the working gas film thickness (or gap) between rotating and stationary seal faces varies from 0.0001 in. to 0.0003 in. (3–8 microns). Research continues to improve seal face geometry to maintain film stiffness and prevent face contact under transient operating conditions. Ideally, rotating faces are perpendicular to the axis of rotation. On upset, if the axial gap reduces, pressure between the two seal faces rises rapidly. The hydrostatic and hydrodynamic forces stabilize, thereby restoring the seal face gap. Similarly, film stiffness restores the gap if there is radial misalignment or thermal deformation. Seals generally tolerate ±0.12 in. (±3 mm) of axial and ±0.024 in. (±0.6 mm) of radial movement.

Normal dynamic seal leakage rate varies from 5 to 10 scfm (guaranteed at 25 scfm) depending on shaft diameter, operating speed and sealing pressure. Gas turbulence and viscosity increases with temperature and reduce leakage rate. However, cooler operation is preferred to ensure long life of the seal elastomers. Static leakage is less than the dynamic leakage rate. On shutdown, good seals with automated compressor isolation can hold system pressure for up to 24 hr, thus allowing operators to investigate any problem before depressurization. The consequent reduction in flaring is environmentally and economically desirable.

Post shutdown, "slow roll" of the rotor in gas turbine-driven compressors should take into account the seal vendor's recommended minimum speed requirement; typically >100 rpm for a 4 in. seal. A stationary carbon ring can accommodate slow roll at speeds of 5–15 rpm on steam turbine shutdown.

Table 1 and Figs. 2 and 3 show different categories of dry gas seals, their principal features and applications.

Dry gas seals are installed with the labyrinth on the gas side and the carbon ring/labyrinth on the outer atmospheric side. Quad seals with four seal stages have also been successfully installed and operated.

Silicon carbide/tungsten carbide are commonly used materials for rotating seal faces. Their good thermal conductivity and low deformation at high temperature assist in maintaining the seal face gap. Silicon carbide or special grade carbon is used for the stationary seal faces. Sleeves and other cartridge parts are generally made of stainless steel. Review and minimize screwed multiple sleeve construction for improved reliability. If H_2S is present, seal materials

TABLE 1. Seal types and applications

Seal type	Feature/application
Single	• With inboard labyrinth usually applied in air, N_2, CO_2 or other nonhazardous services
Tandem	• Two seals in line with the same orientation • The second back-up seal is designed for full sealing pressure
Tandem with intermediate labyrinth	• Two seals in line with an intermediate labyrinth to safeguard and reduce sealing pressure of the secondary back-up seal • Most widely used in hydrocarbon applications
Triple seal in tandem	• Three seals in line with the same orientation • The primary and intermediate seals share the high-pressure (usually above >1,750 psig or 12,000 kPa) sealing load
Double, opposite faced	• Two opposite, face-to-face seals with clean gas injected in between at higher than the sealing pressure • Used in applications where absolutely no leakage to atmosphere is acceptable (e.g., toxic gases) • Preferred in screw compressor applications due to the limited axial sealing length

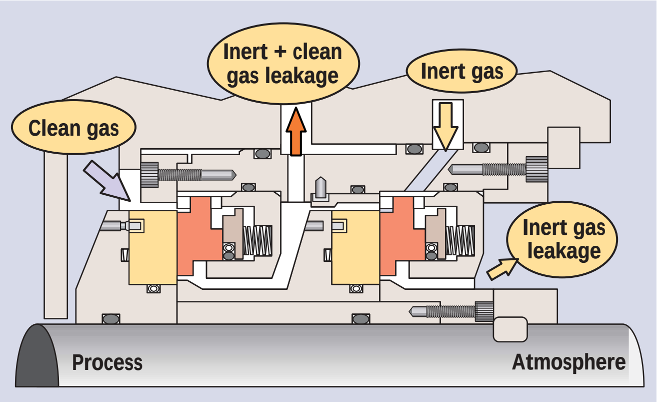

FIG. 2. Tandem dry gas seal.

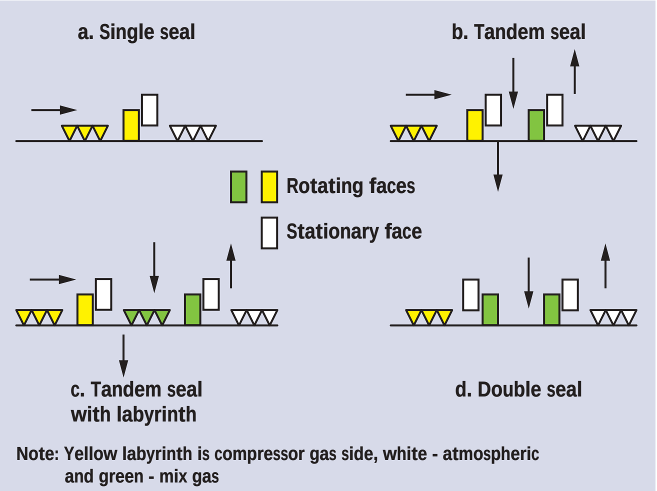

FIG. 3. Different categories of dry gas seals.

should be NACE compliant. Consider duplex stainless steel parts, hastelloy springs and silicon carbide faces if wet chlorides are present in the process gas.

Secondary sealants O/V ring: Under pressure, permeable O-rings (fluoroelastomer) absorb gas. On shutdown, a high rate of depressurization can damage these O-rings (explosive decompression) as absorbed gas exits. To prevent explosive decompression, suppliers recommend controlled depressurization. Depressurization rate reduces as sealing pressure increases. For sealing pressures above 900 psig (6,000 kPa), consider an O-ring of harder (but lesser compression/sealing ability) material like Aflas. Alternatively, consider using polymer (e.g., teflon) seals with a position retaining step. The O- or V-ring material is also affected by gas composition and should be discussed with the seal supplier. Generally, a water saturated gas with 5% CO_2, 1 % H_2S or the presence of methanol/ethanol will require review of the secondary seal material.

Face contact on *reverse pressurization* results in instantaneous seal failure. Reverse pressurization can occur in low-pressure service due to high plant flare pressure on upset/depressurization. To prevent such incidents, the seal can be protected by suitable instrumentation or installing a check valve in the leakage line to flare. For extended periods of high flare pressure, other solutions may be needed. During engineering advise the maximum possible flare pressure and time period to the seal vendor.

Reverse rotation on compressor shutdown and consequent seal damage is possible under one or more of the following conditions:

- Suction valve does not close or
- Recycle valve fails to open or
- Check valve malfunctions.

A T-slot or other bidirectional groove seal face will maintain the seal gap on reverse rotation. Reverse rotation is rare due to large machine inertia and, hence, not a prime criteria for seal face selection.

Seal design is a continuously evolving technology; as manufacturers resolve field problems, new construction features are introduced. Engineers should not hesitate to evaluate new developments and select the features most suitable for their application. Some significant new features are:

▶ Provision of a rotating face shroud that contains seat material in case of seal face fracture

▶ Provision of an oil slinger or carbon rings between the seal and the bearing to prevent oil migration from the bearing to the seal

▶ Larger return oil passage to allow its free flow from the bearings to reservoir. Recently developed, directed lubrication bearings with decreased oil requirement may assist further in reducing this problem

▶ Centering collar on main seal sleeve

▶ Prevent gas leak along the shaft by providing relief path to flare in the seal sleeve.

Successful operating envelope of dry gas seals can be defined as:

Operating parameter	Conventional unit	SI unit
Maximum dynamic sealing pressure:	6,160 psig	42,500 kPag
Maximum static sealing pressure:	6,160 psig	42,500 kPag
Minimum operating temperature:	(–) 242°F	(–) 152°C
Maximum shaft size:	13.75 in.	350 mm
Maximum compressor operating speed:	70,000+ rpm	
Normal allowable radial movement:	±0.024 in.	±0.6 mm
Maximum allowable axial movement: (size dependent)	0.24 in.	6 mm
Maximum surface speed at seal balance diameter:	590 fps	180 m/s

Any one seal or manufacturer has not achieved the above parameters. Every supplier's experience should be individually evaluated.

Support system. Typically, on compressor trip/shutdown, the automated suction and discharge valves close and the recycle control valve opens. Consider blow down of the compressed gas in this loop if the trip is due to the dry gas seal (usually primary) failure. However, on shutdown from other causes, consider keeping the system under pressurized hold. On pressurized hold, the dry gas seals are subjected to an intermediate "settle out" pressure that depends on the volume of the loop.

Estimated settle out pressure should be provided as a minimum design condition for the seal and support system. In a multibody compressor train with interstage check valves, the transient settle out pressure should be considered in the seal design. Some end users and suppliers prefer/recommend that the dry gas seal and support system be suitable for the maximum discharge pressure.

The seal support system is required to carry out one or all of the following functions:

- Maintain seal, separation and buffer gas supply at the required

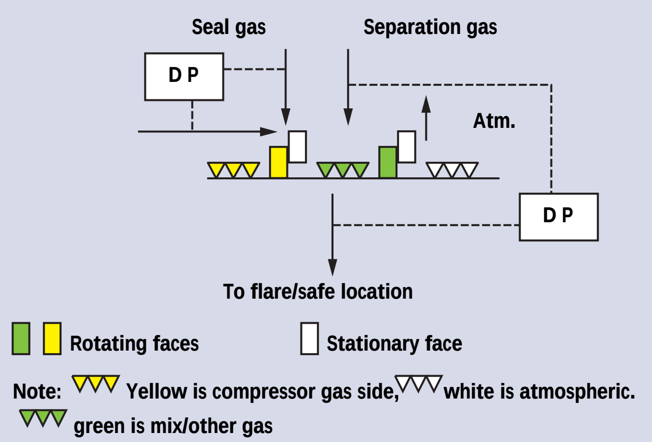

FIG. 4. Process control on tandem seal with intermediate labyrinth.

flow/pressure to ensure safe/reliable operation under all operating, transient and shutdown conditions

- Safely dispose of the normal seal leakage
- Contain or eliminate hazardous/toxic leakage
- Monitor seal performance
- Raise alarm or shutdown on malfunction/failure of the seal or the support system.

Each dry gas seal will need one or more of the following utilities:

▶ In a "between bearing" compressor, the pressure balance line reduces the bearing thrust load and the sealing pressure to compressor suction pressure. The *seal gas,* which is injected in a cavity before the primary seal faces, should be noncondensing and compatible with the compressed gas. By using a PDCV or control valve, the seal gas is controlled at 25 to 35 psi (175 to 250 kPa) above the sealing (reference) pressure to ensure an inward velocity of 20 ft/sec (6 m/s) when rotating and 10 ft/sec (3 m/s) at static conditions. Some suppliers recommend seal gas flow control.

Although labyrinth clearances at the top end of the shaft are larger when the machine is stationary, the closing force of the seal spring reduces leakage and, hence, a lower inward velocity is acceptable. Typically, total dynamic seal gas flowrates range from 50 to 150 scfm (internal + external leakage). Solids or magnetically charged particles larger than two microns should be removed from the seal gas, preferably by using coalescing filters.

Condensed liquids (water or C_6+ hydrocarbon) in the sealing gas produce a sticky substance that adversely affects elasticity of O-rings and springs, resulting in seal hang-up and machine trip from increased leakage rates. The effect is exacerbated under the stationary/pressurized hold condition when no windage heat is generated between the seal faces. In high-pressure application, the traditional 20°F (11°C) margin of the seal gas temperature above dew point is insufficient to prevent condensation of water/heavy hydrocarbons as the leakage gas pressure decreases to near atmospheric before discharge to a safe location or the flare. Check the seal gas phase envelope for the entire operating range before determining the super heat margin. If required, consider heat tracing of the seal piping.

Seal vendors recommend that machine start/stops are minimized. With turbine-driven compressors, inform the seal vendor if the compressor is required to idle in the hot standby condition. When seal gas is taken from the compressor discharge, on hot standby insufficient discharge pressure may result in dirty compressed gas reaching the seal faces with resultant seal failure. Selecting suitable seal gas source(s) to meet all operating/transient conditions is of prime importance for reliable operation.

▶ **Buffer gas** is filtered to 2–5 microns prior to injection into tandem/triple seals between the primary and secondary seals. Nitrogen at 50 to 100 psig (350 to 700 kPa) is commonly used as buffer gas and regulated above the back pressure in the primary leakage line (generally connected to plant flare). Though not preferable, in offshore applications, pressurized air is used when it's the only available utility. If air is used as buffer gas, use sufficient volume to ensure that the air and hydrocarbon mix does not reach the LEL. The LEL depends on the gas composition and should be evaluated on a case-by-case basis. Minimize length of the atmospheric vent line to keep the back pressure below 5 psig (35 kPag).

▶ **Separation gas** is injected between the seal and bearing to prevent lube oil migration toward the seals. Carbon bushes and/or an oil slinger assist in preventing this oil migration. A generously sized bearing lube oil return line and tank breather minimizes backpressure and consequently lube oil migration toward the seals.

Consider locating the seal support panel at similar or higher level than the compressor shaft center line to assist in draining liquids from the low point. The seal monitoring/control panel provides the following principal functions:

- Measures seal gas/sealing differential pressure to ensure a greater seal gas pressure
- Measures/routes/monitors primary seal leakage to flare or a safe location
- Routes/monitors secondary seal leakage to atmosphere or a safe location
- Monitors separation gas injection.

Dry gas seal failures are generally instantaneous with little opportunity for operator intervention. Most failures occur early in the seal operating life. Once technical issues are resolved, failures are rare. Commonly used alarms/trips in the seal control panel are:

Initiating condition	Single seal	Tandem seal	Double seal
Primary seal leakage high	Alarm[1]	Alarm[1, 2]	Alarm[1]
Primary seal leakage high–high (failure)	Trip[1]	Trip[1, 2]	Trip[1]
Seal gas differential pressure low or high	Alarm[3]	Alarm[3]	Alarm[3]
Seal gas differential pressure low–low	Trip[3]	Trip[3]	Trip[3]
Seal/buffer gas filter differential pressure high	Alarm	Alarm	Alarm
Buffer gas flow high—secondary seal failure	Not applicable	Alarm	Not applicable
Buffer gas differential pressure low	Not applicable	Alarm	Not applicable
Separation gas pressure/flow low	Alarm	Alarm	Alarm

[1] Trip level flow is usually 5 to 8 times the normal leakage. Alternatively, differential pressure may be used to actuate a trip.
[2] For a tandem seal, normal flow is seal leakage + buffer gas flow.
[3] To prevent dirty compressed gas reaching seal faces.

Auxiliaries. The seal/buffer gas filter is generally stainless steel with a coalescing, duplex filter and continuous transfer valve. The filter is typically sized for a minimum of three times the normal flow, or the required flowrate for the seal failure case, whichever is greater. For a saturated or heavy condensing seal gas, consider installing the filter downstream of the pressure throttling device to maximize condensate removal. Consider designing the seal gas filter in accordance with the relevant pressure vessel code.

An orifice plate is commonly used for flow or differential pressure measurement of the primary seal leakage. Since leakage is small, a ¾ in. or 1 in. line will provide little differential pressure for accurate measurement. However, flow accuracy is not the prime concern since failure is detected when leakage rate increases to 5–8 times normal flow. Leakage from each seal should be measured separately. When the primary seal leakage is routed to atmosphere, consider installing a flame arrestor in the vent line.

Testing. Dry gas seal cartridges are tested for both dynamic and static seal leakage in the seal supplier's shop. The test is conducted using ambient air or nitrogen at various operating points up to the maximum sealing pressure and operating speed. In tandem construction, both primary and secondary seals are tested for the maximum sealing pressure. Dynamic and static seal leakage are measured at various pressures and speeds. The test procedure is jointly agreed upon. For cryogenic applications, it's possible to replicate low temperatures during testing. Manufacturers usually provide both normal and guaranteed maximum leakage rates.

Field installation. Most seal failures occur early. For critical or large applications, consider hiring the services of the seal vendor's representative for first startup in the field. Dry gas seals are customized and purchase of a spare cartridge set is recommended for quick maintenance turnarounds.

Retrofit. Consider the following issues when retrofitting dry gas seals on compressors with existing wet seals:

▶ Changes in piping/valve arrangement including blowdown to take advantage of pressurized holding

▶ Design pressure review of upstream equipment that may be subject to settle out pressure

▶ Impact on train rotor dynamics. Wet oil seals increase rotor critical speeds, reduce amplification factors and improve damping. However, as seals wear, they may cause subsynchronous vibration and adversely affect rotor stability. Installing dry gas seals will decrease rotor lateral critical speed and increase the amplification factor. However, rotor response is predictable and remains unchanged over time. Careful evaluation is required when retrofitting the dry gas seal on flexible rotors. If preliminary review warrants, then a third-party rotor dynamic analysis is a justifiable expense.

The full cost of a retrofit should be considered when carrying out the life cycle benefit analysis.

In addition to centrifugal compressors, some other common applications for dry gas seals are:

- Dry screw compressor (normally double pressurized due to space limitation)
- Steam turbines (limited success due to problems associated with steam condensation)
- Cryogenic pressure recovery service
- Agitators (low-speed dry gas seal).

Advanced computing methods including FEM/FEA/stress and torsional analysis continue to increase understanding of the hydraulics and mechanical rotor dynamics of large machines installed with dry gas seals and magnetic bearings. As experience increases, commercial considerations for remote/unmanned operations may force their application to be the norm.

Increased environmental regulations and energy cost mean design and operating engineers must make an effort to become conversant with dry gas seals and their support systems. HP

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