Carbon Ring Seal

Carbon ring seals are a good compromise between low leakage and simplicity for nonhazardous gases like air, nitrogen, or carbon dioxide.

From: Compression Machinery for Oil and Gas, 2019

Related terms:

Differential Pressure, Compressors, Turbines, Nitrogen, Dry Gas Seal, Gas Seal, Process Gas, Ring Seal, Separation Seal

View all Topics

Dynamic compressors

Maurice Stewart, in Surface Production Operations, 2019

8.2.6.1.3 Liquid film seals

Liquid film seals are often used in combination with the labyrinth seal. The liquid film seal consists of inner and outer floating carbon seal rings that do not touch each other or the shaft (Fig. 8.42). Sealing is accomplished by injecting oil between the rings at a pressure slightly higher than the gas pressure in the case (Fig. 8.43). Sealing oil is fed to the seal from an overhead tank located at an elevation above the compressor set to maintain a fixed, typically 5 psi, differential above "seal reference" pressure (seal reference pressure is very close to suction pressure).

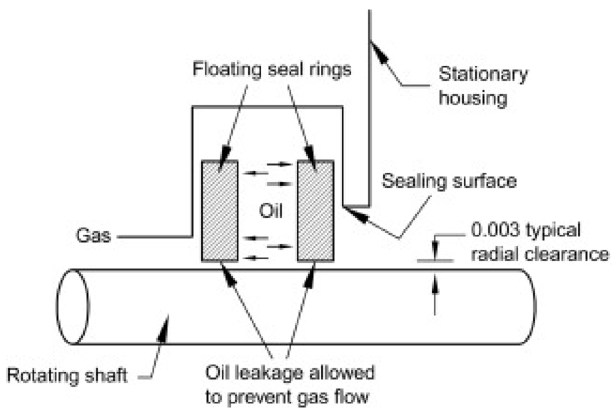

Fig. 8.42. Cutaway illustrating a liquid film seal incorporating carbon seal rings.

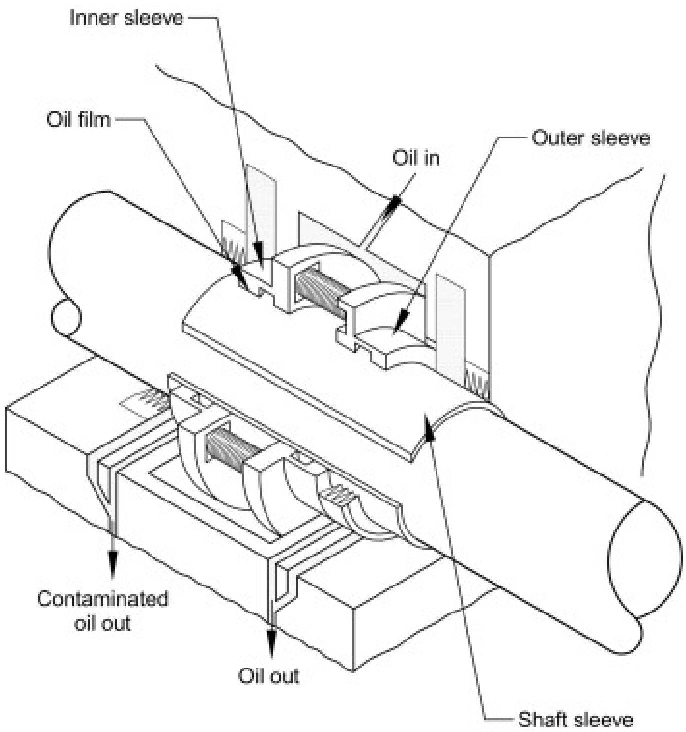

Fig. 8.43. Three-dimensional view illustrating a liquid film assembly.

The oil enters between the seal rings and flows in both directions to prevent inward leakage to the process gas or outward leakage of the gas to the atmosphere. “Buffer ports” are often available for injection of an inert gas to further ensure separation of the process from the sealing medium. The liquid film seal requires a continuous supply of high-pressure oil and a system for circulating the oil and maintaining pressure. They are typically suitable for sealing pressures in excess of 3000 psi.

> Read full chapter

Dry Gas Seal Best Practices

W.E. Forsthoffer, in Forsthoffer's Best Practice Handbook for Rotating Machinery, 2011

Separation systems

Regardless of the type of seal configuration (double or tandem), the function of the separation system is to prevent process gas from entering the bearing housing in the event of a seal failure, and oil from entering the seal cartridge. Entrance of process gas into the bearing housing exposes the plant to catastrophic consequences and extended downtime.

There are several types of separation seals. The choice depends on the availability of the separation gas (usually N_2). The alternatives, arranged in order of highest usage of separation gas, are:

- Labyrinth seals
- Abradable labyrinth seals
- Non-contact carbon seals
- Segmented carbon contact seals

The best practice is to use labyrinth or abradable labyrinth separation seals, if sufficient N_2 is available. This recommendation is based on the reliability of labyrinth-type seals compared to carbon seals, and the fact that the differential pressure across labyrinth seals is not limited, as is the case for most carbon ring seals.

If carbon ring seals are used, the control system must limit the differential pressure to the design maximum. In addition, if carbon contact seals use cryogenic N_2, the best practice is to condition the N_2 to ensure sufficient moisture is present for optimum carbon life (see B.P: 9.5).

Experience shows that, in the case of a catastrophic seal failure, there is a possibility that process gas could enter the bearing housing through the separation seal. For this reason, the best practice is to individually vent each of the bearing housings to a safe location.

The method of separation gas control depends on the type of seal selected. For labyrinth and abradable labyrinth seals, the best practice is to use differential pressure control – seal supply pressure minus secondary vent pressure – to each seal. For carbon ring seals, pressure control could limit the maximum differential pressure across the carbon rings.

The condition of each separation seal can be determined by monitoring and alarming on low differential pressure for labyrinth and abradable labyrinth seals. For carbon ring seals, monitoring and alarming on low pressure is recommended. These parameters should be used as permissive signals to prevent starting the oil system if N_2 gas is not being supplied to the separation seals.

Best Practice 9.5

Ensure that N_2 dew points are above –30°C (–22°F) to optimize the life of carbon used in seal faces and separation seals.

This is achieved by:•Using small air separation units that produce moist N_2 (dew point > – 30°C [–22°F])•Using a N_2 bubbling system to condition 'bone dry' N_2 so that the dew point > – 30°C (–22°F)

Currently, air separation units produce nitrogen with dew points below –50°C (–58°F).

The life of carbon seals (radial and face seal) is significantly reduced in dry gas applications where the N_2 dew points are below –30°C (–22°F).

Lessons Learned

The use of 'bone dry' N_2 (dew points below –30°C) for intermediate and separation sealing duties has resulted in low seal MTBFs (below 12 months).

In some cases, floating carbon seal wear was observed during the factory acceptance tests (FATs).

Benchmarks

This best practice was first used in 2008. Since that time, specifications that require the dew point of supplied N_2 to be above –30°C (–22°F) have been produced. It should be noted that small, dedicated N_2 generators can produce N_2 above –30°C (–22°F).

> Read full chapter

The AP-X® Process

Robin Pearsall, William P. Schmidt, in Proceedings of the 3rd Gas Processing Symposium, 2012

4.2.5 Large N_2 compander design

A compander is a turbo expander which uses the expander work to directly drive a process compressor. The expander extracts energy from the expanding refrigerant gas, reducing the gas temperature. It uses this energy to compress warm refrigerant gas upstream of the economizer. Over the past decades, over one thousand cryogenic turbo expanders and companders have been installed and run for Air Separation Unit (ASU) service, and there is extensive experience in hydrocarbon applications.

Each compander produces up to 10 MW of refrigeration. These were the biggest companders ever built by Air Products. Key features of the companders are dry gas face seals and split circumferential carbon ring seals to limit nitrogen leakage to atmosphere. The companders have passive thrust control, which requires no control valves to regulate the axial thrust. This simplifies the design and improves reliability. A robust and proven nozzle assembly design was used to efficiently and reliably adjust flow through each compander.

The heart of the compander is the design of the impellers. A high performance aerodynamic shape for each expander and compressor impeller is defined using Computational Fluid Dynamics (CFD). Finite Element Analysis (FEA) verified acceptable stresses due to speed and blade loading, and defined an operating zone free from destructive natural frequencies. This work was verified by outside consultants.

Lastly, each compander underwent rigorous testing. Several components were tested at much more severe conditions than will be seen in operation, to ensure reliable and robust operation. These components include the thrust bearings, pin drive and dry gas process seals. The aerodynamic performance was demonstrated by building and testing with 1/3 scale expander stages for the main and balance companders. This included the nozzle cascade, impeller, and diffuser. The final compander assemblies were tested at over a wide range of speeds and flows. All testing was successful, confirming the calculations and predictions.

> Read full chapter

Turbomachinery

T.C. Allison, ... B. Ertas, in Fundamentals and Applications of Supercritical Carbon Dioxide (sCO_2) Based Power Cycles, 2017

7.3.2.6 Hanwha Techwin/SwRI integrally geared compander

Wilkes et al. (2016b) present a novel machinery architecture for 5–25 MWe modular power blocks for concentrating solar power plants. The concept combines all radial turbine and compressor stages for a recompression cycle on a single integrally geared unit as shown in Fig. 7.27, where 705°C inlet turbine stages are shown with orange volutes and compressor stages with blue volutes. Although only two compressor and turbine stages are shown, the other side of the gearbox has four additional overhung stages resulting in a two-stage main compressor, two-stage recompressor, and four-stage turbine with reheat in between stages 2 and 3.

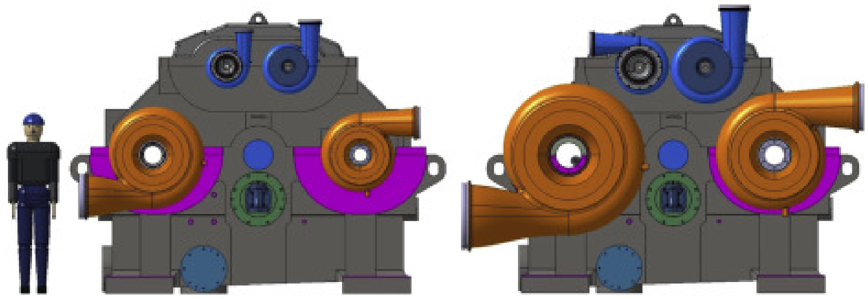

Figure 7.27. Integrally geared supercritical CO_2 compander layout for 5-MWe (left) and 25-MWe (right) recompression cycles (Wilkes et al., 2016a).

The predicted efficiencies and diameter of the largest stage for turbines and compressors as a function of scale are shown in Fig. 7.28, where the efficiency increases with scale as a result of improved aerodynamics at the larger sizes. The nondimensional aerodynamic performance parameters for all stages are provided and are reproduced in Table 7.8.

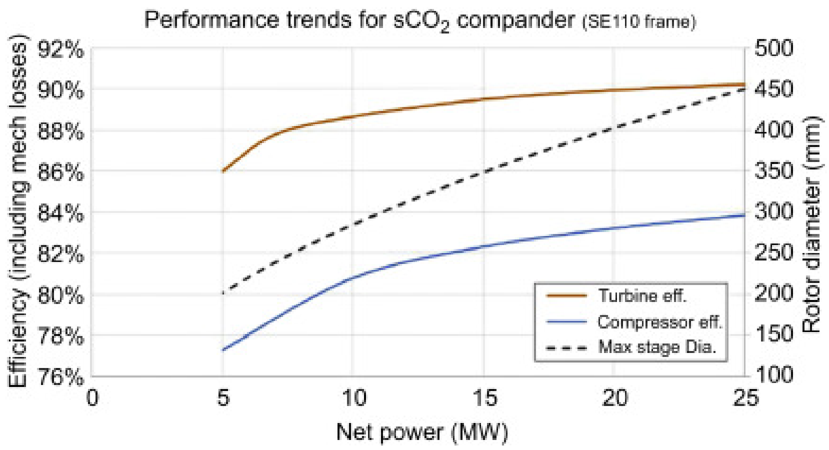

Figure 7.28. Integrally geared supercritical CO_2 compander efficiency and size dependency on scale (Wilkes et al., 2016a).

Table 7.8. Integrally geared compander stage performance parameters recompression cycle (Wilkes et al., 2016a)

Several design details are highlighted in a cross-section of one of the turbine pinions, shown in Fig. 7.29. Between each turbine casing and the main gearbox, an insulating plate is used to separate the hot casing from the gearbox and protect the bearings. The indicated high-strength casings will be designed following American Society of Mechanical Engineers (ASME) pressure vessel guidelines to give robust management of the high-pressure gas. The gearbox architecture can also accommodate either dry gas or carbon ring seals, although the authors state that “[d]ry gas seals will most likely be required for the turbine stages where the temperature or pressure is greatest. Carbon ring seals could be considered as an option for the lower temperature and pressure compressor stages” (Wilkes et al., 2016a).

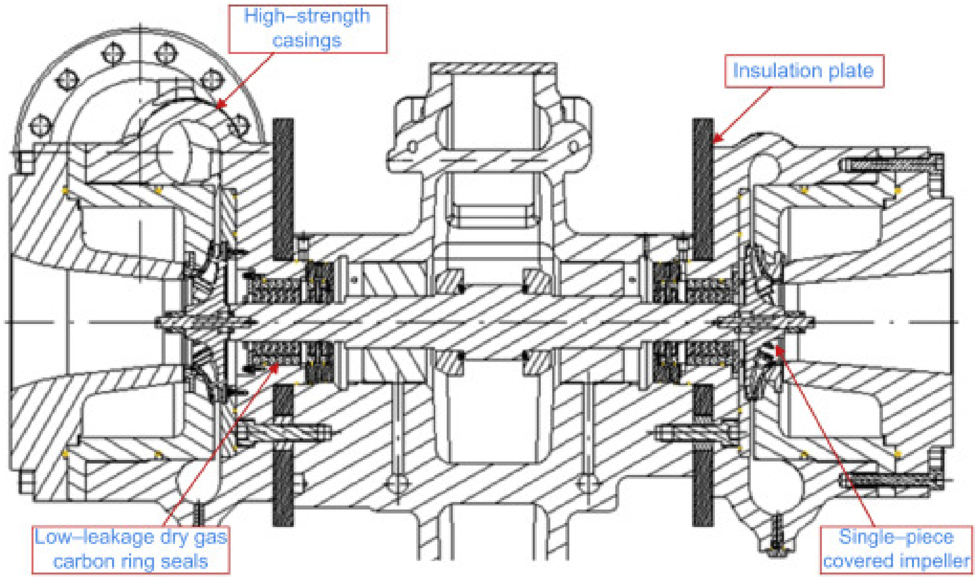

Figure 7.29. Integrally geared supercritical CO_2 compander cross-section of turbine pinion (Wilkes et al., 2016a).

The compressor impeller designs are expected to utilize 3D metal printing technology with one-piece covered impellers to maximize efficiency and operating range. The 3D printed impellers will not have a shroud joint, which reduces mechanical stresses. The manufacturing process also has minimal flow path geometry limitations, and the detailed blade design will leverage this technology to create a flowpath that is efficient over a wide operating range. A review of range extension technologies is performed by Allison et al. (2016) for incorporation/adaptation into the main compressor to accommodate wide range variations near the critical point.

Case study: 20-MWe recompression cycle

For discussion purposes, an approximate turbomachinery sizing was performed by the authors for a 20-MWe-scale recompression cycle implementation. Cycle conditions were taken from an example recompression cycle (U.S. Department of Energy, 2015) and are shown in Fig. 7.30, but the mass flow was doubled for a 20-MWe application to make it unique from other machines that are currently under development. Based on the scale, radial turbomachinery was selected for all stages. The turbine was specified for reheating between the first and second stages up to the same inlet temperature. Conditions and sizing calculations for each stage resulted in three turbine stages, two main compressor stages, and three recompressor stages, as shown in Table 7.9 for the compressors and Table 7.10 for the turbine.

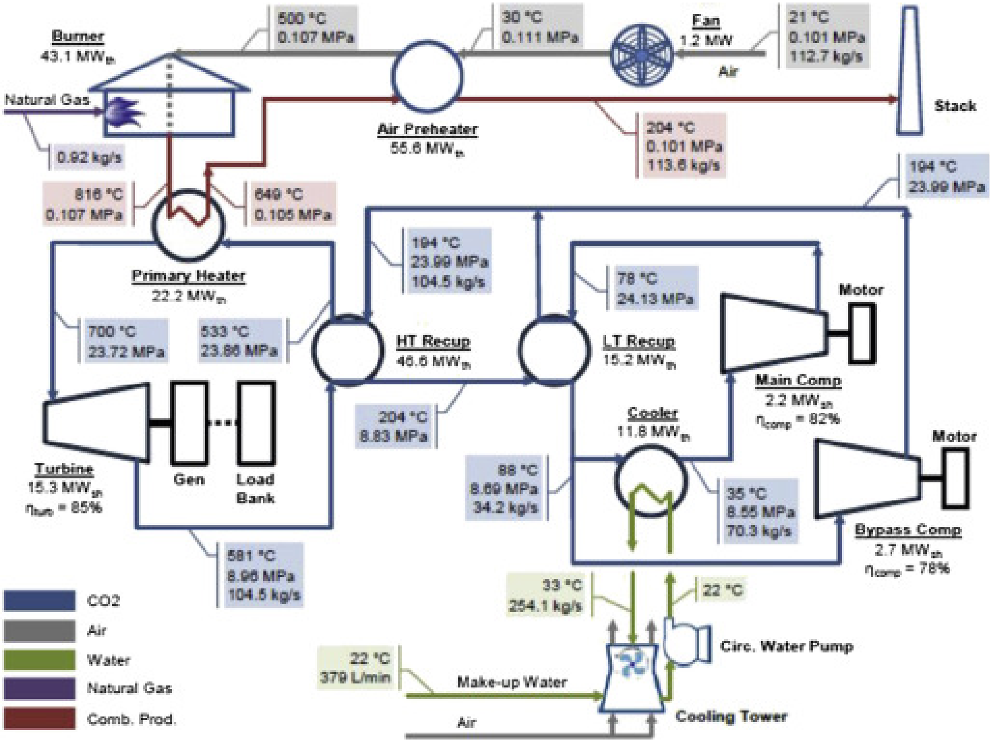

Figure 7.30. Cycle conditions for 20-MWe turbomachinery case study (U.S. Department of Energy, 2015).

Table 7.9. Compressor design details for 20-MWe case study

					Main comp.	Recompressor	
Parameter	**Symbol**	**Units**	**Stage 1**	**Stage 2**	**Stage 1**	**Stage 2**	**Stage 3**
Mass flow rate	*m*	kg/s	140.6	140.6	68.4	68.4	68.4
Inlet total pressure	*P00*	kPa	8550	16,033	8690	12,731	17,823
Inlet total temperature	*T00*	K	308.15	326.97	361.15	398.70	433.85
Pinion speed	*N*	rpm	14,400	14,400	14,400	14,400	14,400
Stage pressure ratio (PR)	*PR*	–	1.875	1.505	1.465	1.400	1.346
Flow coefficient	□	–	0.049	0.043	0.035	0.027	0.022
Specific speed	*Ns*	–	0.66	0.62	0.56	0.49	0.44
Machine Machine number	*MU2*	–	0.63	0.41	0.78	0.71	0.64
Isentropic efficiency	□*s*	–	83.37%	83.48%	78.98%	78.57%	78.55%
Inlet viscosity	□*00*	Pa-s	4.66E-05	5.57E-05	2.07E-05	2.40E-05	2.76E-05
Inlet density	□*00*	kg/m^3	619.1	691.8	169.6	219.2	270.8
Discharge viscosity	□*00*	Pa-s	5.57E-05	6.20E-05	2.40E-05	2.76E-05	3.14E-05

Discharge density	*▯08*	kg/m3	691.8	735.8	219.2	270.8	321.9
Discharge total enthalpy	*H08*	kJ/kg	321.0	334.5	523.3	549.5	575.8
Discharge total tempera-ture	*T08*	K	327.0	341.3	398.7	433.9	466.4
Discharge total pressure	*P08*	kPa	16,033	24,130	12,731	17,823	23,990
Impeller exit tip speed	*U2*	m/s	150.2	150.2	202.9	202.9	203.1
Impeller tip diameter	*D2*	mm	199.2	199.2	269.1	269.1	269.4
Impeller exit width	*B2*	mm	7.25	6.48	7.05	5.46	4.41
Axial length blades	*Zblade*	mm	25.0	23.1	27.0	23.1	20.5
Hub di-ameter	*D1h*	mm	45.0	42.6	51.6	45.4	40.8
Aero power	*Paero*	kW	1903	1900	1783	1792	1796
Windage	*PDF*	kW	9.0	10.2	10.5	13.4	16.5
Mechani-cal losses	*PMech*	kW	0.0	0.0	0.0	0.0	0.0
Stage power	*PStage*	kW	1911	1910	1793	1805	1812
Overall power	*PShaft*	kW			9233		
Volume	*Vimp*	mm^3	2.03 E+05	2.03 E+05	4.99 E+05	4.95 E+05	4.98 E+05
Moment of inertia	*IP*	ton▯mm^-2	11.22	11.60	55.44	59.31	62.87
	IT	ton▯mm^-2	5.92	6.12	29.28	31.36	33.26

Table 7.10. Turbine design details for 20-MWe case study

Parameter	**Symbol**	**Units**	**Stage 1**	**Stage 2**	**Stage 3**
Inlet mass flow rate	*M*	kg/s	210	210	210
Inlet total pressure	*P00*	kPa	23,720	17,370.0	12,548.8
Inlet total tem-perature	*T00*	K	973.15	932.32	890.62
Discharge total pressure	*P08*	kPa	17,370	12,549	8960
Specific speed	*Ns*	–	0.395	0.453	0.526
Pinion speed	*N*	rpm	10,800	10,800	10,800
	▯	–	0.263	0.286	0.313

Exit flow coefficient (Cm6/U4)					
U/C	*U4/Cs*	–	0.695	0.701	0.709
Stage pressure ratio	*PR*	–	1.366	1.384	1.401
Isentropic eff.	*□TT,s*	–	84.22%	85.85%	86.04%
Inlet absolute viscosity	*µ00*	Pa-s	4.21E-05	4.03E-05	3.87E-05
Inlet density	*□00*	kg/m^3	122.4	95.2	72.9
Inlet enthalpy	*H00*	kJ/kg	1222.0	1172.9	1123.6
Inlet entropy	*S00*	kJ/kg-K	2.92	2.93	2.93
Rotor tip speed	*U4*	m/s	251.1	251.1	251.1
Rotor inlet temperature	*T4*	K	950.3	909.5	867.9
Rotor inlet pressure	*P4*	kPa	20,774	15,102	10,832
Rotor inlet density	*□4*	kg/m^3	109.14	84.20	64.07
Rotor tip diameter	*D4*	mm	444.07	444.09	444.11
Rotor inlet width	*B4*	mm	19.67	23.83	28.99
Rotor exit temperature	*T06*	K	932.3	890.6	849.1
Rotor exit tip blade angle	*Bb6t*	Deg	–57.9	–57.7	–57.4
Rotor exit hub diameter	*D6h*	Mm	128.8	128.8	128.8
Aero power	*Paero*	kW	10,317	10,348	10,143
Mechanical loss	*Pmech*	%	0.0%	0.0%	0.0%
Total power	*Ptotal*	kW		30,807	
Volume	*Vimp*	mm^3	2.24 E+06	2.24 E + 06	2.24 E + 06
Moment of inertia	*IP*	ton□mm^2	486.4	461.0	433.4
	IT	ton□mm^2	255.4	241.7	226.9

A geared inline layout was selected, with the turbine driving the compressors through a speed-increasing gearbox and driving the generator through a speed-decreasing gearbox. The compressors would likely be arranged in BTB fashion to minimize net thrust and decrease the size of the balance piston. Although operating the compressor and recompressor located on the same shaft does not allow for independent speed optimization of each component, this configuration strikes a good balance between cost, flexibility, and technology readiness. The rotordynamic implications of this layout and other layouts are described in the rotordynamics portion of Section 7.4, and the aerodynamic sizing and predicted performance for the compressor and turbine stages are discussed in more detail in Sections 7.5 and 7.6, respectively.

> Read full chapter

Integrally Geared Compressors

Aaron M. Rimpel, ... Kolja Metz, in Compression Machinery for Oil and Gas, 2019

Seals

Theoretically, any type of shaft seal can be used in IGCs. Labyrinths seals, floating carbon rings, and dry gas seals are most common, depending on the pressure and the process gas, and brush seals can be used for high-temperature applications. Labyrinth seals are used for many low-pressure air compressors, especially when robustness is more important than low leakage. They are normally made of metal, mainly aluminum, but thermoplastics can be used for higher corrosion resistance or better rubbing behavior. With special labyrinth teeth design, smaller clearances can be achieved for improved leakage performance. Carbon ring seals are a good compromise between low leakage and simplicity for nonhazardous gases like air, nitrogen, or carbon dioxide. They allow smaller clearances than labyrinths because the floating carbon rings can follow the shaft motion when traversing critical speeds. Due to the longer axial sealing gap, fewer carbon rings are needed than labyrinth teeth for the same pressure difference.

The lowest leakage can be achieved with dry gas seals, which are the most popular seals for explosive and toxic gases. They require the most complex seal gas system to maintain operation free of particles, liquids, and back pressure. The complexity of the seal gas systems increases with higher numbers of compressor stages and can turn unfavorable compared to single-shaft compressors, which only have two dry gas seals per casing regardless of the number of stages. Dry gas seals also add mass to the pinion outboard of the bearings, which lowers the natural frequencies and must be considered in the rotordynamics analysis.

Besides the shaft seals, internal sealing between discharge and suction side of the impeller is needed in every compressor stage. For closed impellers, normally labyrinths are used as cover disc or eye seals, while brush seals are less common. Open impellers have no specific seal, but a stator contour matching the blade geometry maintains a small gap, which is limited by the amount of radial and axial movement of the rotor relative to the stator.

> Read full chapter

Major Process Equipment Maintenance and Repair

In Practical Machinery Management for Process Plants, 1997

Special Purpose Steam Turbine Operation and Maintenance

As petrochemical process machinery increases in complexity, proper coordination of the operating and maintenance functions becomes an important aspect of machinery management. Someone once observed in an exaggerated way that if one could see the “gray line” between machinery operation and maintenance functions one would be in trouble. Large steam turbines are no exception. A good example would be the running in and startup of a special purpose turbine after an extensive overhaul. A good machinery maintenance person will not walk away after the overhaul—his job seems never done. For instance careful carbon ring break-in is often ignored or bypassed based on the justification of getting the turbine on line a few hours sooner.

Proper Break-In of Carbon Rings*

□□□□□□

Incentives are:

□□□□□□□

- Long runs
- Higher turbine efficiency
- Protection of bearings and journals by keeping water out of the oil due to blowing steam past the seals into the bearing housing.
- In the winter the machine results in happy operators and a safer unit.
- Lower vibration levels

A common method of breaking in carbon rings involves mounting dial thermometers on the gland housing and observing its temperature rise at incremental speeds for about three hours. Stuffing box temperature rise is a function of carbon ring wear rate, heat transfer rate from the carbon rings through the gland housing, and steam conditions. Surface temperature monitoring procedures are highly questionable due to their poor time response to events happening at the sealing zone between the carbon rings and turbine shaft. Directly observing shaft vibration gives real time knowledge of the condition of the seals.

Factors affecting break-in.

Figure 8-3 is a typical carbon ring gland housing assembly for a small steam turbine. The carbon rings that actually do the steam sealing are made of a special form of graphite that is self-lubricating. The seal is usually constructed of three or more segments bound together and against the rotor shaft by a garter spring. The carbon rings are prevented from rotating by a tang.

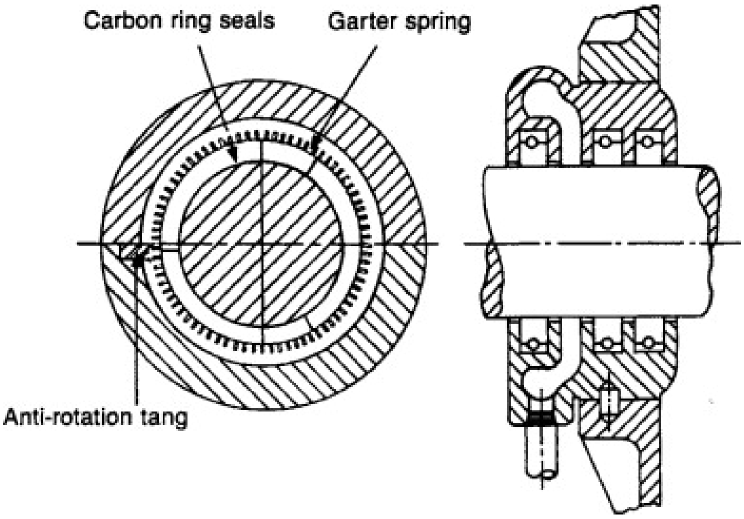

Figure 8-3. Carbon ring gland assembly.

Mechanism of break-In.

Assuming that the carbon ring packing clearances are within design specifications, the carbon rings are “broken in” when they acquire a slick glaze due to controlled rubbing action. Time required for the packings to wear in varies as a function of: steam temperature and pressure, clearances, pressure drop across the seals, sealing steam flow, shaft surface smoothness, shaft surface speed, seal casing configuration and carbon ring composition and design.

Break-in may take from 3 to 12 hours and occurs at about 2,500–3,500 rpm for 3–4 in. internal diameter carbon rings. Cold carbon ring to shaft clearance for 3–4 in. internal diameter rings is about 15-16 mils. Hot running clearance should be about 1–2 mils. Following wear-in, the carbons fit rather loosely in the stuffing box and in the cold condition have a large clearance from the rotor shaft. In this condition, the seals are relatively immune to the sudden thermal changes that a turbine goes through during its normal duty cycle.

The differential coefficient of expansion between steel and carbon is .000004 in. per in. of diameter for each degree Fahrenheit increase in temperature. Since the thermal expansion of carbon is less than that of steel, too rapid of a wear-in often will result in broken rings, or excessive carbon ring to shaft clearances. Large clearances produce poor sealing and destructive steam “wire drawing” across the carbon ring faces. High levels of vibration, high gland box surface temperatures, noise and a big cloud of steam occurring shortly after turbine startup to full speed are sure signs that the carbon rings were inadequately broken in and are grabbing the shaft. If this happens, don't even ask if the carbon seal rings are “broken”—they are.

Warm up.

Of the utmost importance for any turbine operation, including carbon ring break in, is proper warmup. The entire rotor case assembly must be allowed to heat up to its equilibrium temperature prior to starting slow roll. Even heating is required to avoid rotor rubs, high thermal stresses and unequal expansion of the seal rings. During heatup, if the steam plume starts at the case drain pipe outlet and is noisy, this means that water is flashing even if no liquid appears. Dry steam travels through a foot or more of clear space before a wet plume develops and there is much less noise. Again: do not start rolling until all case drains are blowing dry, without puffing. This heat-up may take several hours.

Sealing steam.

For a condensing turbine, if possible, have the condenser vented and the sealing steam initially turned off. Sealing steam should not be turned on unless the rotor is turning. Cold air sucked across seals into a hot rotating or nonrotating turbine can distort the shaft as severely as hot sealing steam entering the seal area of a cold nonrotating turbine. Shaft distortion will cause a rotor bend or “bow” to form which can result in destructively high vibration levels. If the shaft develops or has a thermally induced rotor bow, a 1-hour 300-600 revolutions per minute slow roll usually will allow most of it to relax out.

The normal sealing steam pressure of a condensing turbine is about 3–4 psig. A higher sealing steam pressure is recommended at the outset to assist the outboard seals to begin break-in. This is important, for if the low pressure end is primarily sealed with the high pressure end seal leaking off steam, exhaust end packings may not get much steam until the unit actually is coupled up and running at normal speed. If this is the case, the low pressure seal area suddenly may get its first dose of hot steam preceded by a slug of water at full speed. The result may be a sudden seal “grab,” carbon shattering, and violent failure. This is the cause of the mysterious severe turbine vibration that occurs shortly after the operator walks away from a machine that was just put on line.

Surface condenser use.

The turbine is heated up and brought to minimum speed as a reduced back pressure machine. This maximizes heat input into the seal areas. Caution must be exercised to avoid overpressurizing the surface condenser expansion joint and the steam turbine exhaust casing. Steam flow to the condenser is minimal during an uncoupled slow speed run. As such, the exhaust pressure of the turbine, either positive or negative, can be controlled by balanced use of the surface condenser vent valve, cooling water flow to the condenser exchanger, proper hogging jet operation, and turbine case drain valve positions.

Use of vibration probes.

For monitoring carbon ring break in, one temporary probe holder bracket mounted on the inboard face of each bearing housing, with a reasonably clean and nick-free shaft surface for the probe to monitor, will work. If the machine is to be permanently monitored with vibration probes, see API-670, “Non-Contacting Vibration and Axial Position Monitoring Systems” for additional details.

Normally, carbon ring break-in is performed with the turbine uncoupled from the driven unit. Having the turbine uncoupled eliminates most sources of external vibration, and has the turbine ready for its overspeed trip check immediately following completion of the carbon seal ring break in. Note, however, that a solo run turbine is quite different thermally from a coupled fully-loaded turbine at the same speed. This difference must be accounted for during carbon packing break-in.

Carbon Break-In Procedure

1. Heat the lubrication oil to a minimum of 100°F before beginning slow roll. Running oil temperature target is usually 110-120°F. Mount dial thermometers on the gland seal housing, mid-turbine case and exhaust casing. These temperatures are used to determine the steady state temperature point of the turbine prior to slow roll.
2. Open all case drains, trip and throttle valve and steam line drains leading to the turbine and begin slowly admitting warm-up steam. *Do not start slow roll until the turbine is hot.* Larger condensing turbines, particularly partial admission turbines, may require a special manufacturer's recommended startup procedure to avoid localized rotor bowing.
3. Slow roll at 500 rpm at least one hour. Open sealing steam line and establish 5–8 psig pressure.As the turbine gets hotter or the vacuum increases, it will speed up rapidly using the same steam flow due to the increased availability of energy.
4. Close case drains as appropriate.
5. Record vibration readings at both ends of the turbine.

6. Raise the speed to 1,000 rpm and immediately record vibration levels. Stay at 1,000 rpm for one hour minimum. At about 1,000–1,200 rpm, the bearing's oil film is carrying the rotor and the shaft has established a reasonably stable orbit in the bearing. Assuming that the rotor has relaxed its thermal bow, the “first” reading you will get at 1,000 rpm is primarily residual rotor unbalance. After about 15–30 minutes, you will observe an increase in vibration (about .25 to 1 mil). Gradually the vibration will drop nearly back to the first reading you took at 1,000 rpm. This is what you've been looking for; a slight carbon seal ring rub followed by a return to steady state.
7. Raise the speed to 3,500 rpm in 500 rpm increments, repeating the sequence of immediately taking “new speed” steady state vibration levels and watching for the vibration increase and decrease cycle caused by the carbon rings breaking in.At about 2,500–3,500 rpm, the new carbons are fairly well glazed and nearly run in. This is also the point where most people destroy their packings by assuming that the job is complete.
8. From 3,500 rpm, raise the speed by increments of 1,000 rpm up to running speed going rapidly through the criticals.If running smoothly, and a sudden severe jump in vibration occurs, immediately drop the speed to 2,000 rpm or less for about ½ hour. The carbon ring seals were grabbing the shaft. After a ½ hour cool down, return the turbine to operating speed.
9. Run at normal maximum running speed for one hour prior to checking the overspeed trip and coupling up.This procedure has produced consistently positive results with a variety of machines, some of which were considered to be characteristically bad performers. The key to a successful and long life carbon ring break-in is patience and the continuous presence of an operator through the entire procedure to handle any contingency.

> Read full chapter

Bearings and Seals

Meherwan P. Boyce, in Gas Turbine Engineering Handbook (Fourth Edition), 2012

Ring (Bushing) Seals

The restrictive ring seal is essentially a series of sleeves in which the bores form a small clearance around the shaft. Thus, leakage is limited by the flow resistance in the restricted area and controlled by the laminar or turbulent friction. The API 617 codes characterize this type of seal. Most of the restrictive-type seals are of the floating type rather than the fixed type. The floating rings permit a much smaller leakage, and they can be of either the segmented type as shown in Figure 13-24 (a) or the rigid type as shown in Figure 13-24 (b).

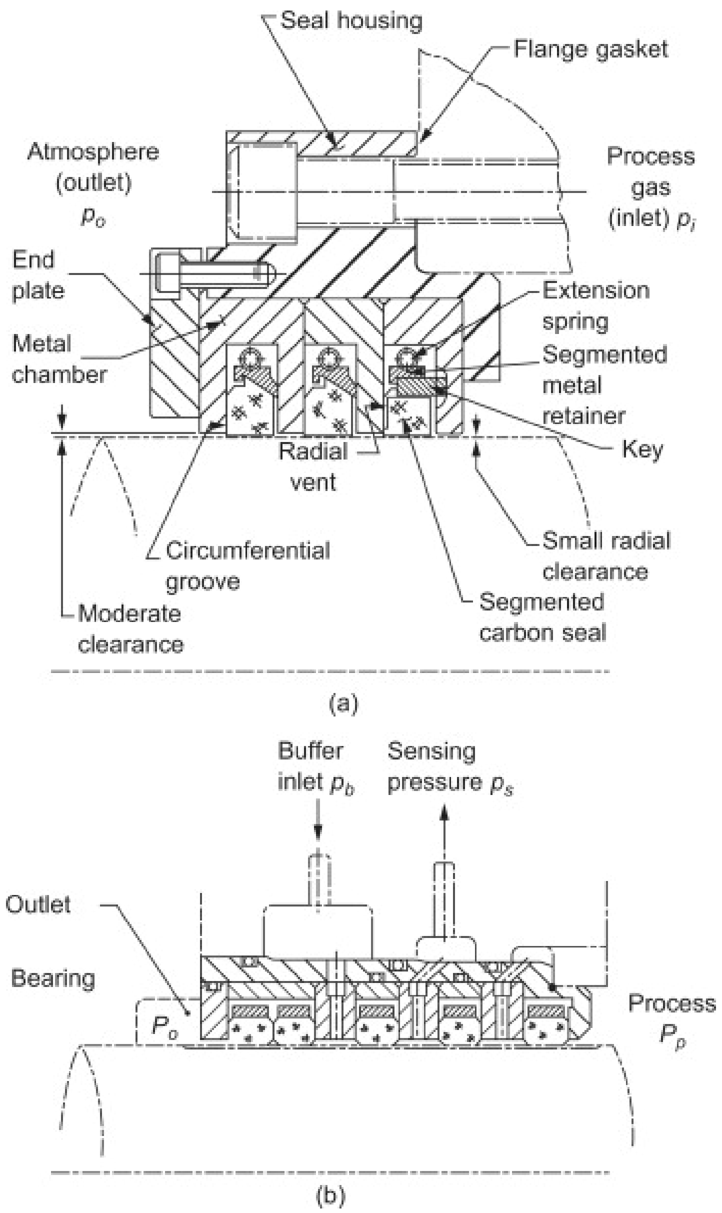

Figure 13-24. Floating-type restrictive ring seal.

Because of the minimal contact between the stationary ring and the rotor, these seals, when properly designed, are ideal for high-speed rotating machinery.

When adequate lubrication and cooling fluid is available, the seal ring, manufactured from babbitt-lined steel, bronze, or carbon, will function satisfactorily. When the medium to be sealed is air or gas, carbon seal rings must be used. Carbon has self-lubricating properties. Cooling of the seal is provided by the leakage flow through the seal. Depending on the operating temperature and environment, aluminum alloys and silver are also used in the manufacture of the seal rings. Leakage limitation depends upon the type of flow and type of bushing. There are four types of flow: compressible and incompressible, each of which may be either laminar or turbulent. Ring seals are divided into two categories: fixed breakdown rings and floating breakdown rings, according to whether or not they are fixed with respect to the stationary housing.

Fixed seal rings

The fixed seal ring consists of a long sleeve affixed to a housing in which the shaft rotates with small clearance. It is an inexpensive assembly. However, since it is fixed, the seal behaves like a redundant bearing when rubbing occurs and, like the labyrinth, requires large clearances. Therefore, long assemblies must be used to keep leakage within reasonable limits. Since long seal assemblies aggravate alignment and rubbing problems, sturdier shafts are required to keep operating speeds in a subcritical region. The fixed-bushing seal almost always operates with appreciable eccentricity. This, plus the combination of a large clearance and a large eccentricity ratio, produces large leakages per unit length. Fixed-seal rings are therefore impractical where leakage is undesirable.

Floating seal rings

Clearance seals, which are free to move in a radial direction relative to the shaft and machine housing, are known as floating seals. These seals have advantages that very close annular clearance-type seals do not possess. The floating characteristic permits them to move freely with shaft motions and deflections, thereby avoiding the effects of severe rubbing.

Differential thermal expansion is a problem at high temperatures where the shaft and bushing are of dissimilar materials, or where there is any substantial temperature gradient between them. For example, the grades of carbon used commonly have a linear thermal expansion coefficient of one-third to one-fifth that of steel, necessitating the design of thermal expansion control into the carbon bushing. This is achieved by shrinking the carbon into a metallic retaining ring with a coefficient of expansion that equals or exceeds that of the shaft material.

It is good practice in critical applications to use bushings of a material with a slightly higher coefficient of thermal expansion than that of the shaft. Here, incipient seizure causes the bushings to grow away from the shaft. The large torque associated with high shearing intensity may necessitate locking the bushings against rotation if the unbalanced pressure forces seating them against the housing walls are insufficient to prevent rotation.

Build up of dirt or other foreign material between the seal ring and seat will result in damage to the journal and excessive seal spin on a floating seal ring unit. Soft materials, such as babbitt and silver, are notorious for trapping contaminants and causing shaft damage.

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Novel Machinery

Natalie R. Smith, ... Timothy C. Allison, in Compression Machinery for Oil and Gas, 2019

Hermetically Sealed and Oil-Free Compression

Hermetically sealed compressors have emerged into more markets over the past several years. Historically, hermetically sealed compression systems have been operating in the so-called clean gas applications for over three decades. More recently, further applications seek to take full advantage of the hermetically sealed design. In the offshore oil and gas business sector, for example, the very compact footprint of hermetically sealed compressor skids is advantageous given the space and weight restrictions on production platforms. These electric motor-driven compressors are packaged such that the entire rotating shaft from driver side to driven equipment, including motor, coupling, all impellers, diaphragms, and bearings are enclosed inside a pressure containment vessel. By doing this, all rotor-to-ambient-air interfaces are eliminated and the need for seals, for example, dry gas seals (DGSs), oils seals, or carbon ring seals, between the process gas and the environment is removed.

Process gas leakage is completely eliminated which is attractive from a performance and an environmental perspective. This arrangement typically utilizes a high-speed motor driving the compressor directly without gearboxes. A lube oil supply system is not needed because radial and axial thrust AMBs or process medium lubricated gas bearings are employed to levitate the motor and the compressor shaft. Internal casing thermal management and cooling of the motor utilizes the process gas. The concept of this assembly and packaging is certainly advantageous over more conventional arrangements with separate driver and compressor casings. Fig. 15.10 illustrates the reduced complexity and smaller footprint of the hermetically sealed design compared to a conventional drivetrain. With all these characteristic design features the hermetically sealed compressor becomes an all-electric system (drive, bearings, instrumentation) providing all the necessary preconditions for remote control and operational flexibility, reduced maintenance activities, and furthermore, given the right instrumentation, remote monitoring, and diagnostics can be easily realized.

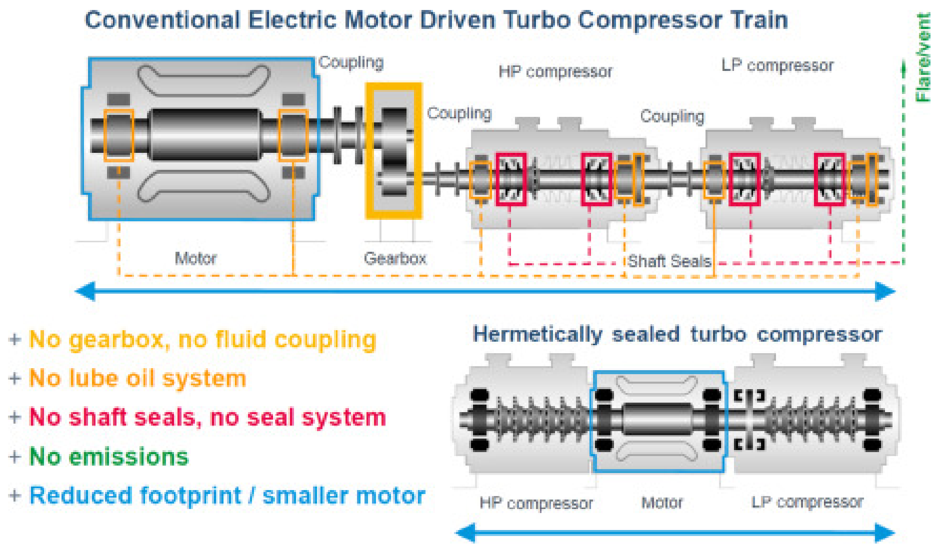

Fig. 15.10. Size comparison of conventional EMD and hermetically sealed compressor trains [1].

While there are clear advantages to the hermetically sealed design, several disadvantages of these systems must be considered. The cooling systems for these machines can have complex control systems as well as result in substantial efficiency penalty due to large windage losses. Furthermore, if the process gas used for cooling is not substantially dry and clean, the motor internals will foul and degrade rapidly. Unit costs are high due to a combination of engineering design complexity and high costs of components like high-speed motors and magnetic bearings. Although lube oil systems and gearboxes do not need to be maintained, the enclosed and highly compact design makes removing and replacing compressor internals more difficult. Finally, hermetically sealed compressors still require a VFD, thus often eliminating the advantages of a smaller compressor package.

The hermetically sealed compressor is an excellent technical option for subsea compression applications. MAN Diesel Turbo introduced the first hermetically sealed compressor, the MOPICO, in 1990 [1]. Their motivation was to reduce the number of mechanical components which should increase the mechanical reliability and reduce the cost of the compressor. However, the inherent design advantages to hermetically sealed compressors, like lower weight, compact size, no DGSs, are less relevant for the majority of onshore compression applications, especially when most compression facilities have plenty of space available and are already equipped with secondary systems to support conventional sealing designs. In addition, there are questions regarding grid stability and reliability that need to be addressed when considering onshore electric motor-driven compressors.

Hermetically sealed compression systems are readily available from several OEM’s with minor design differences. Two main differences are explained in the following paragraphs:

Flexible Coupling Versus Solid Coupling

The compressor shaft(s) are directly coupled to the motor shaft. In case this connection is realized by means of flexible couplings the shaft vibrations of the motor and the compressor are largely decoupled and the components are mostly rotor dynamically independent. In this case each (shaft) component needs two radial AMB's. Because flexible couplings do not tolerate large axial loads, each compressor must have its own thrust bearing.

Connecting the shafts with solid couplings leads to one single train shaft where the rotor dynamic behavior of the entire system has to be analyzed. This system does not exhibit any large coupling overhung modes and the concurrent action of all bearings to all modes leads to better overall damping. With this arrangement one or two radial bearings can be spared and only one thrust bearing is needed. The train, on the other hand, needs more efforts to be properly aligned during manufacturing and assembly.

Stator Windings Exposed to the Process Gas Versus Canned Windings

As opposed to standard electric motors running in atmospheric conditions, a special situation is encountered for hermetically sealed compressors as the electric motor and magnetic actuators are operated under process gas conditions. In such cases, stator windings can be either directly exposed to the process gas or they can be protected from direct contact with the process gas by means of cans, which are a thin metallic, ceramic, or polymeric hermetic protective encapsulation of the winding and the electric laminations. Direct exposure not only has clear advantages in terms of design simplifications and electric performance, it also optimizes heat paths and thus clearly improves the cooling situation as well as motor torque. On the other hand, direct exposure of the winding and the electric laminations can pose very demanding requirements for the insulation system to withstand.

Furthermore, the structural integrity of the hermetic cans has to be assessed very carefully and the design can turn out to be very cumbersome for high-pressure applications, especially with rapidly changing operating and pressure conditions. Since the thickness of the can has to be minimized from the electrical point of view, one often is forced to implement special pressure equalizing design features or even to actively control the backpressure behind the can.

> Read full chapter

Centrifugal Compressors

Jason Wilkes, ... George Talabisco, in Compression Machinery for Oil and Gas, 2019

Shaft End

Shaft end seals are required to seal the gas inside the compressor at the point where the compressor or piston shaft penetrates the casing. This vital sealing function is necessary to prevent escape of process gas to the environment surrounding the compressor. With the exception of reciprocating compressors, all other types of compressors have rotating shafts. Accordingly, the applied type of seals differs. Reciprocating compressors mainly use serial-arranged packing rings for sealing the piston rod. For compressors with rotating shaft, however, dry-gas-lubricated mechanical seals (dry gas seals (DGSs)) are commonly used. The importance of oil-lubricated mechanical seals is reduced to applications where requirements in pressure, speed, and power consumption are low. Bushing-type seals as the shaft end seal are reduced to applications where leakage requirements play a minor role. Labyrinths can be considered technically obsolete. However, bushing-type carbon ring seals or labyrinth seals are commonly used as separation seal mounted between the DGS and the shaft bearing. Their function is both to protect the DGS from spray oil and to prevent uncontrolled process gas entry into the bearing cavity in the event of a total failure of the DGS.

The today's most common type of shaft end seal in the oil and gas industry is based on DGS technology. This triumphal procession goes back to the end of the 1980s of the last century. The use of DGSs in the place of oil lubricated mechanical seals improves mechanical efficiency as the shear and friction power of the seal is significant reduced. The application limit of a single oil-lubricated seal stage is about 100 m/s peripheral speed at the outer diameter of the seal faces and 50 bar differential pressure. By comparison, gas seals achieve more than 200 m/s and a pressure difference of more than 450 bar per single seal stage. The resulting degrees of freedom in the compressor design justify the success of gas sealing technology.

Fig. 3.12 shows the design of a DGS is basically similar to that of an oil-lubricated mechanical seal and consists of a stationary face (2) sealed by an axially displaceable seal element (4) and a rotational face (1). At standstill and depressurized condition, the stationary face is pressed by a set of circumferentially arranged springs (3) against the rotational face. A thrust ring is used to transmit the singular spring forces. The rotational face is centered by a shaft sleeve mounted on the compressor shaft. On its back side, another secondary seal element (4) is placed. The spring-loaded stationary seal face is centered by the seal housing mounted in compressor casing parts. In general, DGSs are pressurized from the outer side.

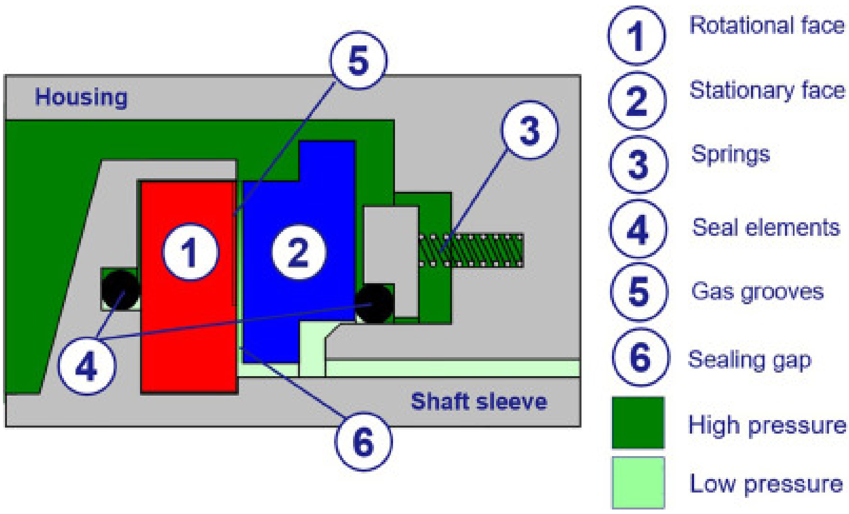

Fig. 3.12. Block representation of a dry-gas seal.

The diameters of the faces are designed to achieve hydrostatic liftoff at standstill with a certain pressure difference across the seal as shown in Fig. 3.13. The axially displaceable seal element on the back of the stationary face seals is the so-called hydraulic diameter, which is always larger than the inner diameter of the sealing surface. This means that the effective pressure area on the back of the stationary face is smaller than the sealing surface and the prerequisite for a stationary liftoff is given.

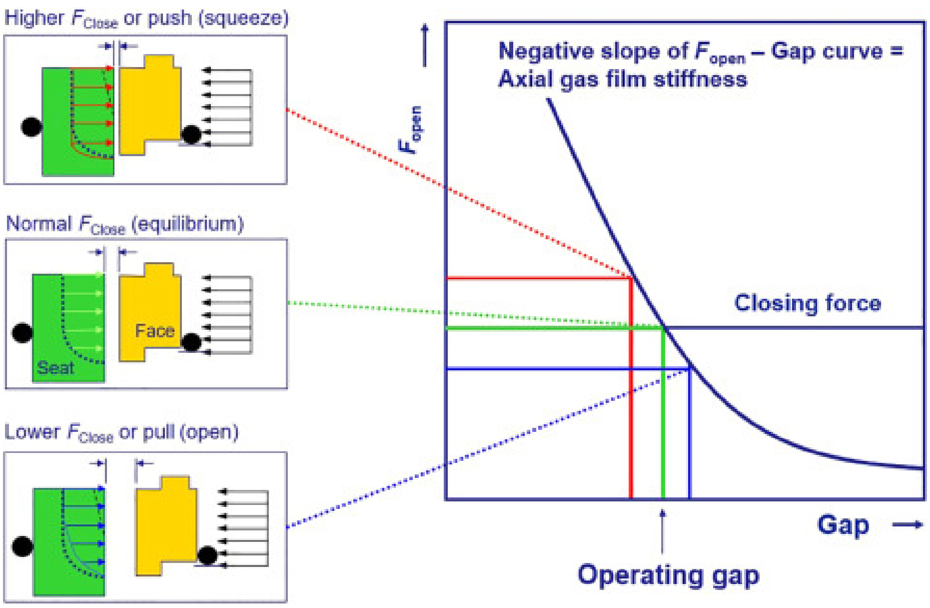

Fig. 3.13. Force balance on a dry-gas seal.

With appropriate evenness and parallelism of the facing surfaces, only the pressure pad in the sealing gap is able to keep the closing force on the back of balance. This gives the seal the ability to regulate the height of the sealing gap itself. Once any additional force tries to disturb the balance the gap height will change, and the pressure pad in the seal gap will create immediately an appropriate counter force. The height of the sealing gap is only a few micrometers.

The spring force is only dimensioned so that it can safely overcome the frictional force of the dynamic sealing element in the unpressurized state, and therefore, plays a minor role. The closing force is mainly represented by a rectangular pressure pad.

Starting from the outer diameter so-called gas grooves (5) are incorporated in one face. These are usually incorporated in the rotational face and reach into the middle of the sealing surface and are only a few micrometers deep. The main function of these grooves is to achieve an additional pressure pad and thus liftoff during dynamic operation. The rotation creates a shear flow in the grooves which in turn induces pressure generation (Fig. 3.14).

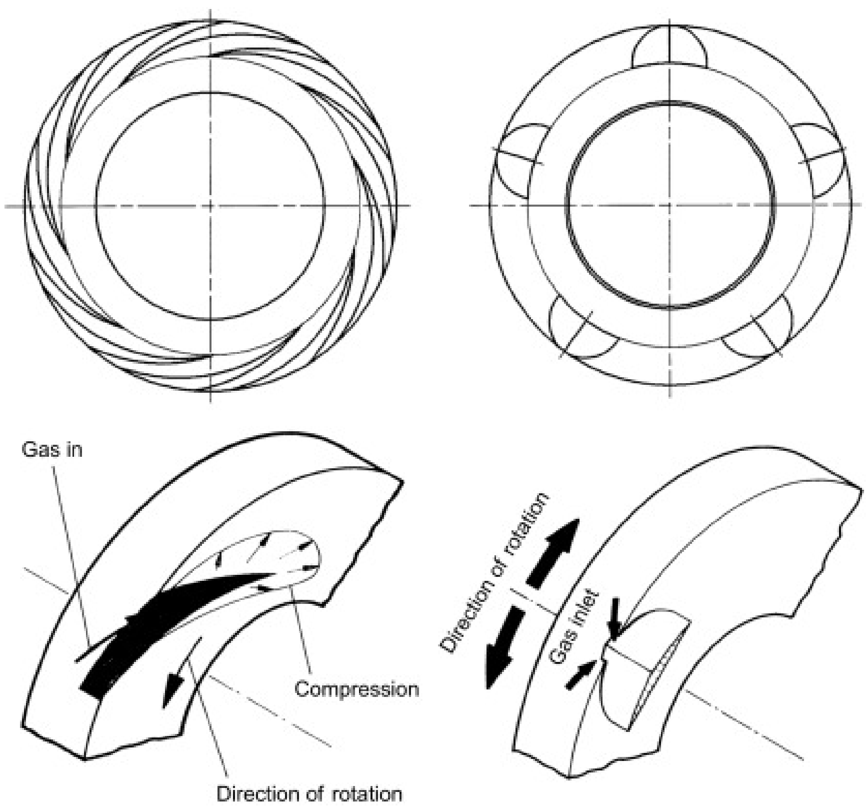

Fig. 3.14. Grooves in a DGS (top) and resulting pressure dam action (bottom).

The available groove geometries can be divided into unidirectional and bidirectional ones. Above all, due to the hardly avoidable possibility of reverse rotation of the compressor shaft and the smaller number of necessary spare seals, bidirectional grooves are gaining more and more popularity in the oil and gas industry.

Table 3.1 gives an overview of the typical built-in materials. At the beginning of the use of DGSs, especially hard-soft pairings with the rotational face made of tungsten carbide and the stationary face made of carbon were common. However, with increasing pressure and speed requirements, hard-hard pairings have proven to be more advantageous.

Table 3.1. DGS Materials

Part	**Material**

Housing, shaft sleeve	Stainless steel or nickel basis alloy
Rotational face	Silicon carbide, silicon nitride, or tungsten carbide
Stationary face	Silicon carbide or carbon
Secondary seals	Elastomer o-rings or spring energized PTFE lib-seals
Springs	Nickel basis alloy

DGSs require clean and dry seal gas for reliable operation in order to avoid particles or condensates between the seal faces. The seal gas is typically taken from the compressor discharge and then throttled, cooled, and filtered as part of a seal supporting system that will be discussed later. This conditioned seal gas is injected between the process-side seal and the seal faces. The process-side seal is typically a single labyrinth seal and is located between the DGS and the compressor internals as shown in Fig 3.15. When seal gas is injected, it prevents flow of process gas to the DGS. The majority of the seal gas flows into the compressor and a slight amount flows across the seal faces.

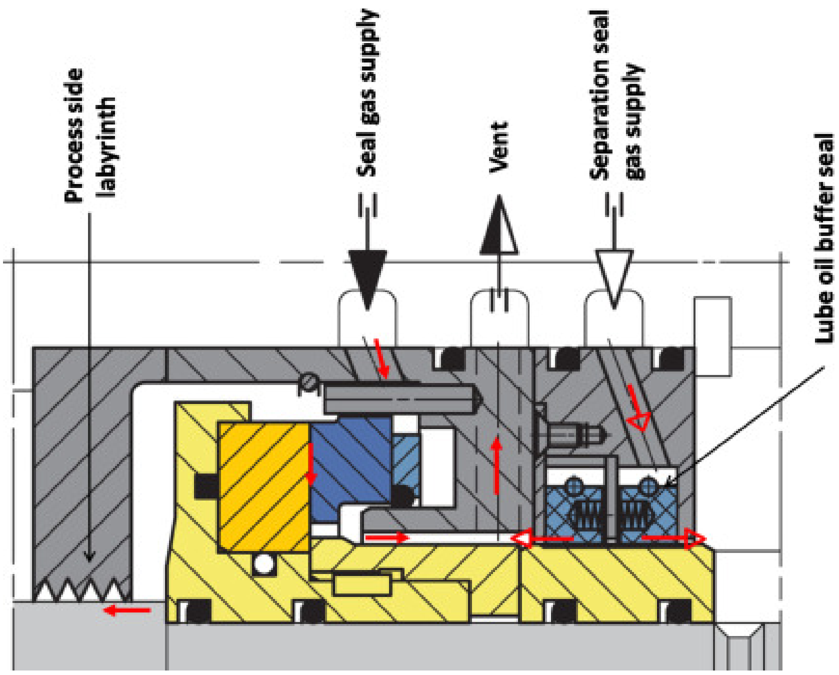

Fig. 3.15. Dry gas seal supply and vent ports.

The seal gas pressure is reduced across the seal faces to vent pressure. The gas leakage exits the compressor through piping, where it is then either sent to a flare system or to some other recovery system. Flow or pressure measurement in this line is used as an indicator of the health condition of the DGS.

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