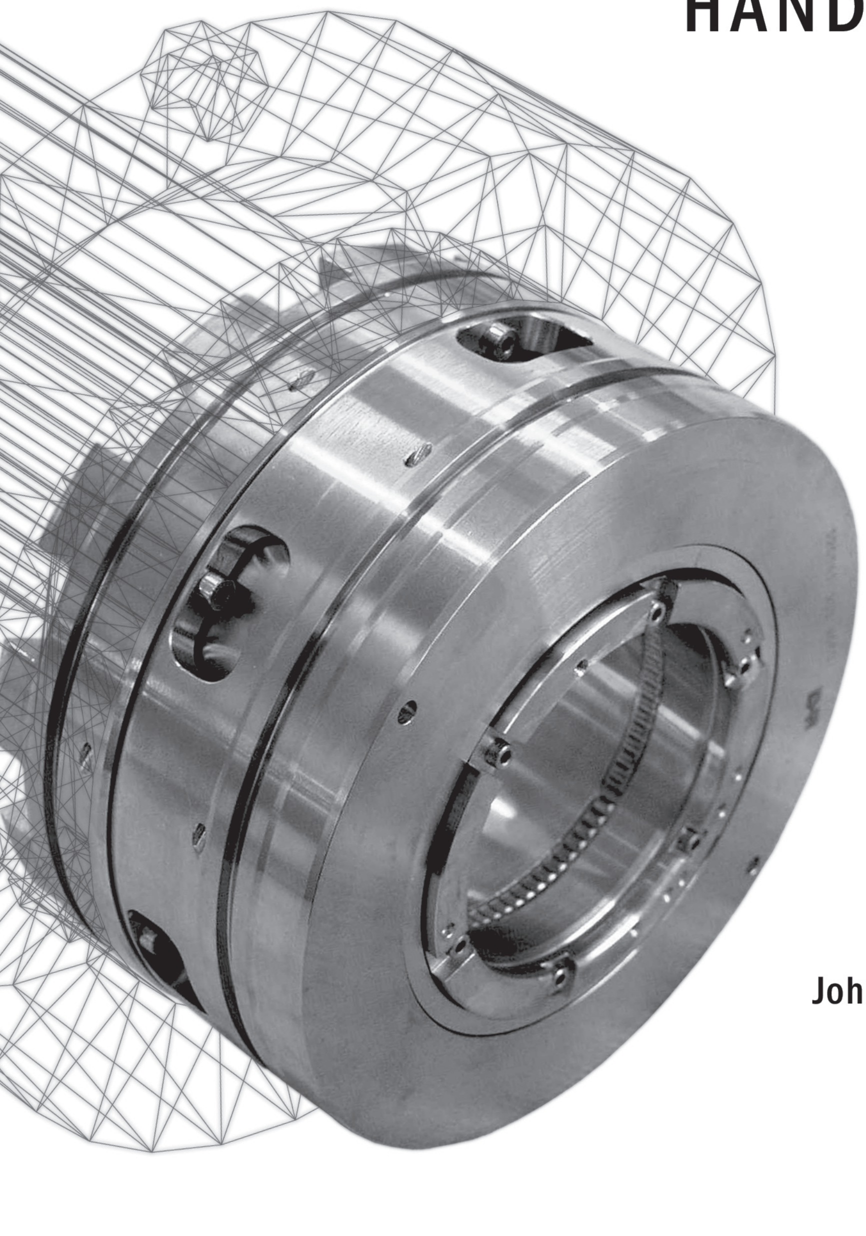
DRY GAS SEALS

HANDBOOK



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PREFACE

In 1999, I assumed responsibility for providing technical support for the installation, commissioning, and startup of centrifugal compressors manufactured by my employer. I quickly discovered that the requirements for reliable dry gas seal operation were not well understood by either EPCs (engineering, procurement, and construction contractors, who are usually involved in the plant and process designs in which centrifugal compressors are used) or compressor owner/operators. Despite the fact that the use of dry gas seals in process gas centrifugal compressors has increased dramatically over the last 20 years (more than 90% of new units are now equipped with dry gas seals), the technology remains misunderstood by many in the industry. At the time of my new assignment, I knew very little about dry gas seal technology (and let me be the first to say that I still have much to learn), so I began to research the subject. I was amazed and intrigued by what I learned. Dry gas seals operate with clearances measured in microns, much less than the thickness of a human hair! Such small clearances are unheard of in the rotating equipment industry. And these seals are being applied in rotating equipment destined for the oil patch, into services that are not known for their inherent cleanliness. I learned first hand that dry gas seals can be very problematic if not properly applied at the time of plant, process, and/or compressor design. Conversely, when properly applied, dry gas seals can operate trouble free, as can be attested by the many, many successful installations. Furthermore, I found very little published literature on the subject beyond gas seal manufacturers' sales and marketing brochures.

Dry gas seal technology is quite amazing. A totally dry seal operates at high speeds and pressures at a relatively miniscule clearance and gas leakage rate. Some gas seal configurations can even operate with zero hydrocarbon gas emissions, which provides an environmental benefit to us all. As dry gas seals achieve more and more successful run time in the field, the operating envelope is continually being redefined. Advances in technology and materials have made it possible for dry gas seals to operate at sealing pressures greater than 6,000 psi.

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THE EVOLUTION OF DRY GAS SEALS

Centrifugal compressors in process gas service require shaft sealing to prevent the process gas from traveling down the shaft and escaping the compressor case into the atmosphere. Multistage, beam-style compressors (with impellers located between two bearings) require two seals, one at each end of the shaft (fig. 1–1). Single-stage, -overhung style compressors (with a single impeller located outboard of the bearings) require a single shaft seal, directly behind the impeller.

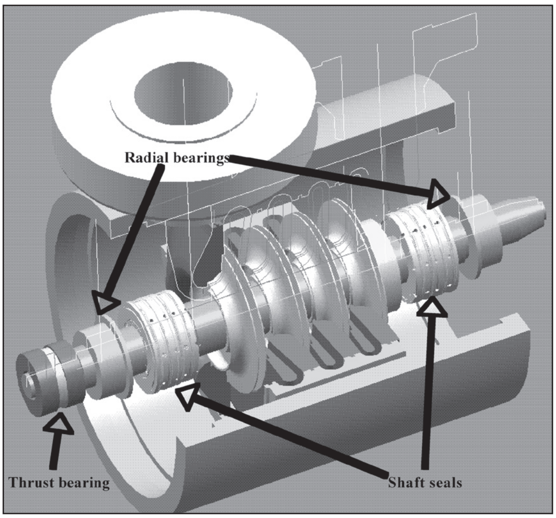

Fig. 1–1. Cross section of a centrifugal compressor

deformation (and therefore potential for the seal rings to make contact) of the parts. Therefore, the gas seal design must incorporate optimum balance of forces, pressure profiles, and gas film stiffness.

Balance of forces

The hydrostatic and hydrodynamic forces acting on the seal (fig. 2–4) control the position of the primary seal ring. *Hydrostatic forces* are present whenever the seal is pressurized. *Hydrodynamic forces* are present only during rotation of the mating ring. The *closing force* (F_C) is a function of the gas pressure, spring force, and friction acting on the primary ring. The closing force can be calculated as follows:

$$F_c = S_1(P_{in}) + S_2(P_{out}) + F_s - F_f \tag{2.1}$$

where:

F_c = closing force

S_1 = surface area of back side of primary ring (upper, high-pressure side)

S_2 = surface area of back side of primary ring (lower, low-pressure side)

P_{in} = gas pressure at outside diameter of primary ring (high)

P_{out} = gas pressure at inside diameter of primary ring (low)

F_s = spring force

F_f = friction force

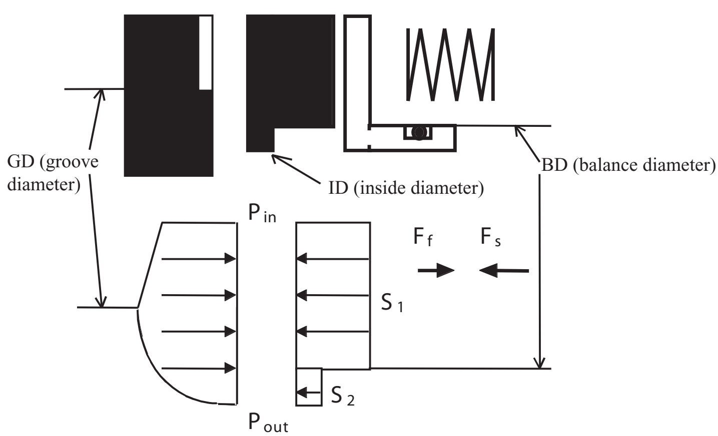

Fig. 2–4. Gas seal forces

7. Maintain this speed for a minimum of 60 minutes, recording data every 5 minutes. The average primary seal leakage must be less than the maximum specified primary seal leakage.
8. Increase the primary vent back pressure to the maximum value specified.
9. Maintain this primary vent back pressure for a minimum of 15 minutes, and record data.
10. Increase the primary vent back pressure to the maximum specified seal gas pressure.
11. Maintain this primary vent back pressure for a minimum of 15 minutes, recording data every 5 minutes.
12. While maintaining the pressure, shut down the test rig to a complete stop.
13. Restart and accelerate to maximum continuous compressor operating speed, and then to compressor trip speed, as quickly as possible.
14. Reduce speed back to maximum continuous compressor operating speed.
15. Maintain this speed for a minimum of 5 minutes, or until the primary seal leakage reaches a steady state, and record data.
16. Repeat steps 12 through 15.
17. Shut down the seal test rig while maintaining gas conditions.
18. Record two sets of data immediately after shutdown.

Visual inspection

1. Upon completion of the dynamic test, disassemble the gas seal and observe the condition of the parts, noting wear patterns or buildup of material.
2. Document the observed condition of the gas seal components.

Static test verification

1. Reassemble the gas seal.
2. Repeat the static testing procedure.

Unidirectional and Bidirectional Gas Seals

Dry gas seals are available in both *unidirectional* or *bidirectional* designs. As explained in chapter 2, the grooves in the mating ring create the hydrodynamic force that allows the gas seal to run without contact between the mating ring and primary ring faces (fig. 3–5). In a unidirectional gas seal, the groove pattern (see fig. 2–2) is such that the lifting effect occurs only when the seal is rotating in the normal direction. When rotating in the reverse direction, the grooves in the mating ring are ineffective and do not generate the forces required to separate the mating ring and primary ring faces. Hence, the seal rings maintain contact and sustain damage because of the heat generated by the friction between the ring faces.

A bidirectional seal employs a symmetrical groove pattern on the mating ring face. The symmetrical groove pattern is able to generate the forces required to separate the mating ring and primary ring faces in either direction of rotation.

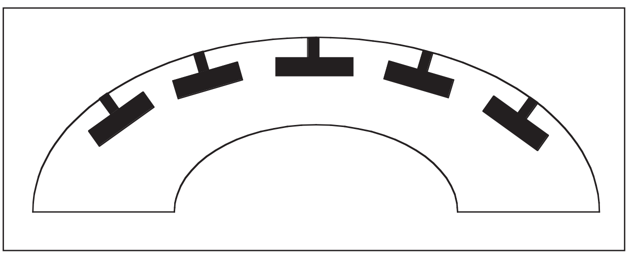

Fig. 3–5. Bidirectional grooves in seal mating ring

As explained in chapter 2, there are two vitally important characteristics of the gas film between the mating ring and primary ring faces:

1. Gas film thickness between the seal faces (the running gap)
2. Stiffness of the gas film

When the gas seal is rotating, the gas film thickness defines the distance between the seal ring faces. A wider running gap reduces the risk of contact between the seal ring faces, thus increasing gas seal reliability. However, increasing the running gap will increase the gas seal leakage rate. The challenge for the gas seal manufacturer is to design a film thickness that provides the best possible reliability, while minimizing the leakage rate.

the compressor discharge as the sole source of seal gas. On the surface, this may appear to be a very sound practice, because compressor discharge pressure is obviously higher than compressor inlet pressure (normal sealing pressure). However, this design does not address transient conditions such as startup, shutdown, or idle. During these transient conditions, there is usually insufficient pressure rise across the compressor to allow for a continuous positive flow of seal gas into the seals. An alternate seal gas source or seal gas pressure boosting equipment is required in this case. The availability of the seal gas source at sufficient pressure is one concern for all types of dry gas seals, including double opposed and tandem with intermediate labyrinth.

A second concern is the quality (cleanliness) of the seal gas. Gas seal manufacturers have stringent requirements for seal gas quality, typically requiring the seal gas to be dry and filtered of particles 3 µm (absolute) and larger. *Coalescing filters* (discussed later) are normally provided in the gas seal system to address this requirement, but a coalescing filter alone may be inadequate depending on the source of seal gas supply. To achieve the required level of filtration at the gas seal, most compressor manufacturer's require that the seal gas be free of solid particles 10 µm and larger, and 99.97% liquid free at the customer connection point on the gas seal system. In most cases, this will require a prefilter upstream of the customer connection point. Because double opposed gas seals normally use an inert gas such as plant nitrogen, which is generally quite clean and dry, seal gas quality is usually not an issue.

Third, the potential for liquid condensation within the gas seal system must be thoroughly reviewed. Heavy hydrocarbons (C6 and heavier) and/or water vapor contained in the sealing gas have a tendency to condense as the gas flows through the gas seal system. The various components of the gas seal system—filters, valves, and orifices—cause seal gas pressure drops during operation. In fact, the largest pressure drop in the entire system may be at the seal faces themselves. As the seal gas expands across these components, the Joule-Thomson effect will result in a corresponding decrease in the gas temperature, and heavy hydrocarbons or water can then condense. A seal gas heater can be provided as part of the gas seal system to superheat the seal gas above its dew point and thereby reduce the risk of liquid condensation. Again, this is usually not an issue with double opposed gas seals using plant nitrogen for seal gas.

The American Petroleum Institute's standard API614 (1999) requires that the seal gas temperature into the gas seal be at least 20°F above its dew point. This can be insufficient in many cases. To illustrate this point, replacement of the existing oil seal system with a new dry gas seal support system in addition to replacing the compressor seals. According to the Environmental Protection Agency's (EPA) Natural Gas STAR Program, a partnership between the EPA and the oil and gas industry focused on the implementation of cost-effective technologies and practices to reduce methane emissions, dry gas seals offer many advantages over oil seals. In addition to reducing fugitive gas emissions, dry gas seals offer significant reductions in operating costs. The STAR Program estimates that a dry gas seal conversion can save approximately $135,000 in annual operating expense with a project payback period of around 14 months.

Economic Advantages of Dry Gas Seals

The following sections discuss the cost savings after a dry gas seal retrofit identified by the EPA's Natural Gas STAR Program.

Seal oil leakage and disposal

It is inherent in oil film seal design that some of the injected seal oil will flow into the compressor casing and be lost into the process. The compressor operator must compensate for this lost oil by adding additional oil to the reservoir. The operator also incurs significant handling costs (particularly if they dispose of the sour oil rather than treating it for reclamation) and clean-up costs (if the process is contaminated with seal oil), in addition to the obvious cost of the oil itself. With dry gas seals, the costs associated with seal oil consumption and handling are totally eliminated.

Process gas emissions

The process gas, which leaks past the seal oil barrier, and the process gas absorbed by the sour oil, must be routed to flare. The process gas sent to flare is a direct operating loss, as this gas could have been used by the process and sold. Gas emissions from oil seal systems typically range from 40 to 200 SCFM. Because of the very narrow running gap of a dry gas seal, process gas emissions are much less, usually less than 10 SCFM, or even zero with some designs such as the double opposed gas seal. One Natural Gas STAR Partner reported a 97% reduction in gas emissions after a dry gas seal retrofit, from 75 Mcf per day to 2 Mcf per day, resulting in an annual

Liquids were also routinely drained from the compressor case after a shutdown of the gas treatment plant. Most of the gas seal failures occurred during or shortly after the gas treatment plant outages, and liquids were observed in the failed gas seals when inspected.

In at least one instance, the compressor control panel captured a gas seal failure event, and clearly revealed a liquid slug into the compressor as the root cause (fig. 8–10). As can be seen in bubble 1, the second and third lines up from the bottom of the graph indicate the compressor vibration is normal and constant. Bubble 2, the bottom line of the graph, indicates the gas seal leakage (as measured in the primary vent) is also normal and constant. Bubble 3 shows a vibration spike when the liquid slug enters the compressor inlet. Bubble 4 shows a return to normal vibration levels approximately four seconds later as the liquid slug exits the compressor discharge. Bubble 5 indicates excessive seal gas leakage to the primary vent as the liquid passes through the gas seal approximately eight seconds after it first entered the compressor. Bubble 6 indicates a compressor shutdown was initiated as the result of the high seal gas leakage, as the speed line (previously constant) suddenly drops off.

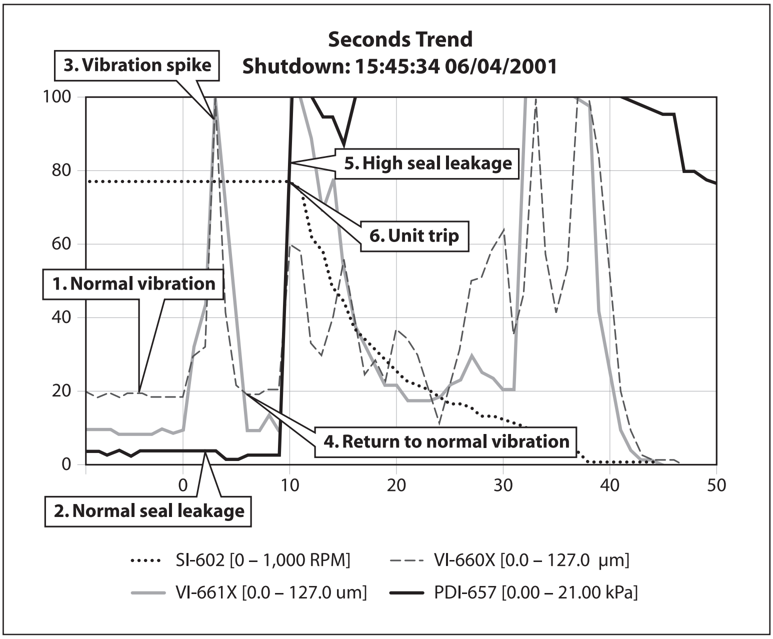

Fig. 8–10. Seal failure because of liquid slug

TEST LOG SHEET

DRESSER-RAND GAS SEAL PBU

Date: 27/05/04

Ref.: 6648 / CSS 192744

Gas Seal N°: 1545 /1546

PO N°: 64726

Time	SI134 (Speed)	PI104A (cavity pressure)	TIS109 (cavity temp.)	FI116B-CE (inboard flow)	TIS110-CE (inboard temp.)	FI115-CE (outboard flow)	.TIS111-CE (outboard temp.)	PI113B (inboard pressure)	FI116B-FE (inboard flow)	TIS110-FE (inboard temp)	FI115-FE (outboard flow)	TIS111-FE (outboard temp)
hh:mm:ss	*tr/min*	*bar G*	°C	*l/min*	°C	*l/min*	°C	*bar G*	*l/min*	°C	*l/min*	°C
15:37:00	10942	9,1	16,7	4,8	22,6	1,5	22,5	0,0	6,4	23,0	1,3	20,9
15:38:00	10942	8,9	16,7	4,8	22,6	1,5	22,5	0,0	6,4	23,0	1,3	20,9
15:39:00	10918	8,9	16,7	4,8	22,6	1,5	22,5	0,0	6,4	23,0	1,3	20,9
15:40:00	3350	8,9	16,4	1,6	23,4	0,2	22,5	0,0	2,0	23,0	0,3	20,9
15:41:00	0	8,9	17,2	0,3	23,4	0,2	22,5	0,0	0,0	23,0	0,0	20,9
15:42:00	0	9,1	17,2	0,3	22,4	0,2	22,2	0,0	0,0	23,0	0,0	20,9
15:43:00	0	8,3	17,2	0,3	22,4	0,2	22,2	0,0	0,0	22,0	0,0	20,9
15:44:00	0	2,0	17,2	0,3	22,4	0,2	22,5	0,0	0,0	22,7	0,0	20,9
15:45:00	0	0,4	17,2	0,3	22,4	0,2	22,7	0,0	0,0	22,7	0,0	21,7

TESTED BY (DRESSER -RAND)

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DRESSER RAND S.A.
Quality Engineering
WITNESS/REVIEW
By : P. LE GRIX

WITNESSED BY (CLIENT)

GLOSSARY

balance diameter: The diameter that separates the high (seal gas supply) pressure area from the low (seal vent) pressure area of the primary gas seal ring. This is typically sealed with an o-ring or polymer seal for high pressure applications.

barrier seal: A seal located between the bearing and the gas seal to prevent lube oil from contaminating the gas seal and process gas from migrating into the bearing housing. Usually a labyrinth or segmented carbon ring seal. Also known as a separation seal or tertiary seal.

bidirectional gas seal: A gas seal designed to rotate in either direction. A bidirectional seal employs a symmetrical groove pattern on the mating ring face that is able to generate the forces required to separate the mating-ring and primary-ring faces in either direction of rotation.

buffer gas: See *Seal Gas.*

closing force: The hydrostatic force acting on the gas seal mating ring in the direction of the primary ring. The closing force is a function of gas pressure, spring force, and friction.

coalescing filter: A filter designed to remove liquid aerosols from a gas.

differential pressure control: A system that controls the supply of seal gas to the seal by regulating the seal gas pressure to a predetermined value above a referenced sealing pressure.

double gas seal: See *double opposed gas seal.*

double opposed gas seal: A two-stage, back-to-back arrangement, dry gas seal. Also known as a double seal.

dry gas seal: A gas-lubricated, noncontacting, mechanical, end-face shaft end seal, consisting of a mating (rotating) ring and a primary (stationary) ring.

expected leakage rate: The primary gas seal leakage calculated by the gas manufacturer for normal operation at a specified operating condition.