No need to read this if all your turbocompressors have gas seals!

One day in the fall of 1988 my manager said to me: “You must have felt last night like one of the people launching the first space rocket - not knowing whether or not it would work”. He was referring to our self-directed retrofit of gas lubricated shaft seals[1] and a successful start-up on two critical motor driven 3,000hp ethylene refrigeration compressors. Our achievement was a first for our large multi-national petrochemical company, even though, admittedly, gas transmission companies in North America were applying gas seals in the early 80s well ahead of the HC process industries.

Things have changed since then. It seemed the ice was broken in those days and many compressor operators followed suit by replacing their cumbersome oil lubricated shaft seals with gas seals. Today gas seals have become to be known as a reliable and cost-effective alternative to oil type seals. Their key characteristics are low leakage rates, wear-free operation and an extremely low level of power consumption.

Gas seals are non-contacting, dry running mechanical seals. Figure 1 shows the principal design features of such a seal. They are usually furnished as a cartridge containing a spring-loaded stationary seal face or sliding ring (1) sealed by an O-ring and a rotating seat or mating ring (2). The sealing faces slide over each other without contact. This results in almost no wear and a long seal life.

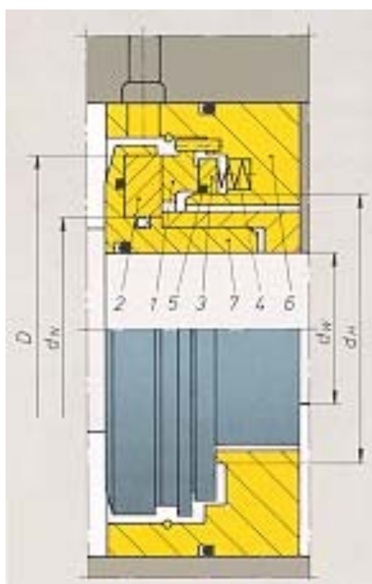

Figure 1. Cross section of a gas seal. 1 Stationary seal face (sliding ring); 2 Rotating seat (mating ring); 3 Thrust ring; 4 Compression spring; 5 O-ring; 6 Housing; 7 Shaft sleeve with cupped retainer. (*EagleBurgmann DGS*).

The seal face of the rotating mating ring is divided into a grooved area at the high pressure side and a dam-area at the low pressure side. See Fig. 2. The

[1] **API STD 692 - 1st Edition, June 2018**

stationary sliding ring is pressed axially against the mating ring by spring forces and sealing pressure.

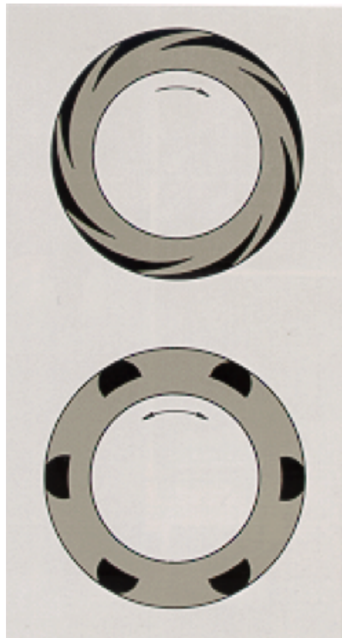

Figure 2. Mating ring V-grooves, U-grooves. (Note: Arrows indicate sense of rotation).

The sealing gap is located between the mating ring and the sliding ring. For proper non-contacting operation these two rings have to be separated by a gas film acting against the closing forces in the sealing gap. The gas film is achieved by the pumping action of the grooves and the throttling effect of the sealing dam. Groove geometry is critical for trouble-free operation of the seal.

Current general operating limits of gas seals are shown in Table 1:

Table 1. General operating limits for compressor gas seals.[2]

Criterion	Limits Customary Units	Limits SI Units	Remarks
Nominal Shaft Diameter	0.984 to 14.961″	25 to 380 mm	
Pressure, abs.	≤ 1,740 psi ≤ 6,500 psi	≤ 12 MPa ≤ 45 MPa	elastomeric components non-elastomeric comp.
Temperature	- 22°F...+392°F - 270°F...+446°F	- 20°C...+200°C - 170°C...+230°C	elastomeric components non-elastomeric comp.
Sliding Velocity Referred to Outer Dia. of Mating Ring	≤ 656 ft/sec	≤ 200 m/s	

[2]EagleBurgmann, *Gas-Lubricated Mechanical Seals*, EagleBurgmann Marketing Communication, Germany.

How can we justify retrofitting gas seals? It was easy to justify our 1988 gas seal conversion. At the time we arrived at a very favorable payback well worth taking the risk of a retrofit. The original floating ring type oil seals on the refrigeration compressors had generally worked satisfactory. However, when they failed they had invariably caused expensive clean-up work, as lost seal oil would permeate the entire refrigeration system and reduce process yield for a long time.

On pipeline compressors equipped with oil type seals we should look once more at seal oil losses. We met operators who would not admit to malfunctioning oil type seals indicated by excessive oil losses while routinely emptying oil pots and low point drains of compressor discharge lines. Table 2 shows a comparison of seal leakage rates.

Table 2. Comparison of leakage rates from various compressor shaft seals

Seal Type	**Geometry***	**Normal Recoverable Oil Leakage** gal/h (l/h)	**Gas Consumption**** CFH (m^3/h)
Floating Ring Oil Seal	2 rings, 0.787 in. (20 mm) each rad.cl. = 0.002 in. (0.05 mm)	gas side 1.9 (7.2)	S.O. trap vent 636 (18)
Mechanical Seal	gap = 0.04 mil (1 μm)	gas side 0.019 (0.072)	S.O. trap vent 636 (18)
Gas Seal	gap = 0.20 mil (5 μm)		12.7 (0.36)

* Nominal shaft diameter = 5.5 in. (140 mm); 5000 rpm; gas pressure p_1 = 87 psi (600 kPa); buffer (seal) pressure p_3 = 109 psi (750 kPa); gas: air; buffer fluid: oil.

** Frequently vented to atmosphere and not recovered due to the unavailability of containment at a lower pressure.

With natural gas prices wildly fluctuating during the past decade it would be difficult to base our justification of conversions from conventional seals to DGSs on product and oil loss savings alone.

We suggest to study the failure statistics of your oil type seals and compare them to this: Current quotes of failure rates of gas seals are in the neighborhood of 0.175 failures/year[3], meaning that we could expect a problem every six years or so. One seal manufacturer bases recommended maintenance intervals around gas seals on limits set by the elastomer aging process. He suggests the following preventive maintenance routine after 60 months of operation:

- Replace all elastomers
- Replace the springs
- Replace all seal faces and seats
- Carry out a static and dynamic test run on a test rig.

We cannot entirely agree because we adhere to an old saw applied in the Canadian Navy that goes: "Leave well enough alone!"

[3] $\lambda \sim 20 \times 10^{-6}$

Bibliography:

Bloch, H.P. and Geitner, F.K., *An Introduction to Machinery Reliability Assessment,* 2nd Edition, ISBN 0-88415-172-7, Gulf Publishing Company, Houston, TX, 1994, p.242-247.

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