

# THE GAS LUBRICATED SPIRAL GROOVED FACE SEAL IN THE PROCESS INDUSTRY

by

Hans Pennink

Director, Research and Development

Atlas Copco Comptec, Incorporated

Voorheesville, New York



Hans Pennink holds a B.S.M.E. degree from the University of Delft in Holland, and an M.S.M.E. degree from Rensselaer Polytechnic Institute (R.P.I.) of Troy, New York. He has over 28 years of experience in high speed rotating machinery, including compressors, expanders and gas turbines. Most of that experience has been in aerodynamics, controls and machine design and development.

Mr. Pennink is Director of Research and Development for Atlas Copco Comptec, Incorporated, in Voorheesville, New York, and has been in engineering management for over ten years. He has been technically involved with the development of high speed centrifugal compressors and expanders. His prior experience has been with Alco Products, the Boeing Company and Mechanical Technology, Incorporated (M.T.I.). He is the author of papers presented earlier at Texas A&M University and I-MECH E. Mr. Pennink has been a member of ASME and API for many years.

## ABSTRACT

The gas lubricated spiral grooved face seal and its application on integrally geared centrifugal compressors in the process industry is discussed. A brief description is given of the seal configuration and its operating principles. Advantages and disadvantages are discussed. The operating envelope and the range of applications are outlined; a number of installations and operating records of this seal on integrally geared centrifugal compressors, as manufactured for the process industry, are also discussed.

## INTRODUCTION

Attention is focused herein on the gas lubricated face seal (sometimes called a dry face seal) for application in the process industry. Further technical information about the gas lubricated dry face seal is not provided, but rather the experience of a manufacturer of integrally geared centrifugal compressors with this type of seal, operating in the process industry, is described.

There are several manufacturers of gas lubricated face seals and all of the seals look somewhat alike. This manufacturer uses only the spiral grooved (whipple groove) face type of seal. Briefly explained are the seal configuration, operating principles, operating envelope, advantages, disadvantages, range of applications, problems and current track record of this type of seal.

## CONFIGURATION OF THE GAS LUBRICATED FACE SEAL

A cross-sectional view of a gas lubricated spiral grooved face seal is shown in Figure 1 with the following parts:

Item	Part Name	Material
1.	Primary seal ring (stationary)	Silicon-carbide or carbon-graphite
2.	Mating ring (rotating)	Titanium-carbide or tungsten-carbide or S.S.-316
3.	O-ring	Elastomer
4.	O-ring	Elastomer
5.	Shaft sleeve	17-4PH (H 1150)
6.	O-ring	Elastomer (Kalrez)
7.	Coil springs	S.S.-316 (harddrawn)
8.	Disk	S.S.-316
9.	Retainer assembly	S.S.-316
10.	O-ring	Elastomer

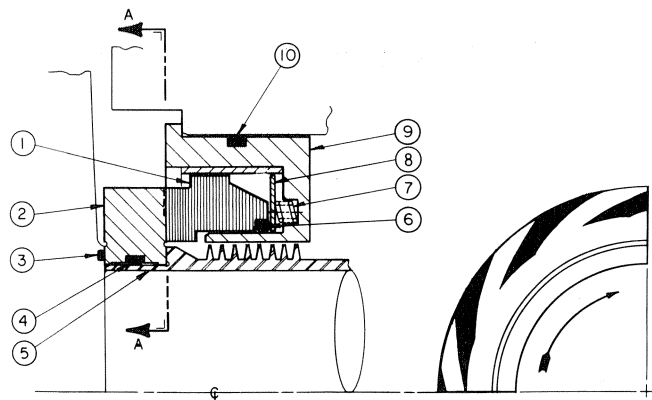


Figure 1. Gas Lubricated Face Seal Cross-Sectional View. 1) primary seal ring (stationary), 2) mating ring (rotating), 3) O-ring, 4) O-ring, 5) shaft sleeve, 6) O-ring, 7) coil springs, 8) disk, 9) retainer assembly, 10) O-ring.

The rotating mating ring has shallow grooves etched into the surface facing the primary sealing ring, (as shown in View A-A of Figure 1), from the outside diameter to a certain intermediate diameter, which is called the groove diameter (DG). The spiral groove (whipple) design is a characteristic feature and contributes a great deal to the successful operation of the seal. The mating ring is trapped on the shaft, between a labyrinth sleeve and the impeller, by a precisely controlled interference.

The spring loaded primary sealing ring is pushed against the mating ring by the spring forces, forming a zero leakage seal at rest. The profile of the cross-section of the primary sealing ring is extremely important and is carefully designed to provide restoring moments, in order to minimize distortion due to pressure forces. The proper position of the mating ring is with the sealing face lining up with the face of the cartridge flange. The springs are then compressed to a desired preload.

The key to the success of the gas lubricated spiral grooved face seal is the result of in depth theoretical analyses of pressure and temperature distortion, heat transfer, and seal dynamics [1,2,3,4].

A sectional view into the seal cartridge to illustrate the internals of the seal is shown in Figure 2. The assembly of a typical combination of a gas lubricated spiral grooved face seal and labyrinth seal, as used on high speed integrally geared centrifugal compressors, is illustrated in Figure 3.

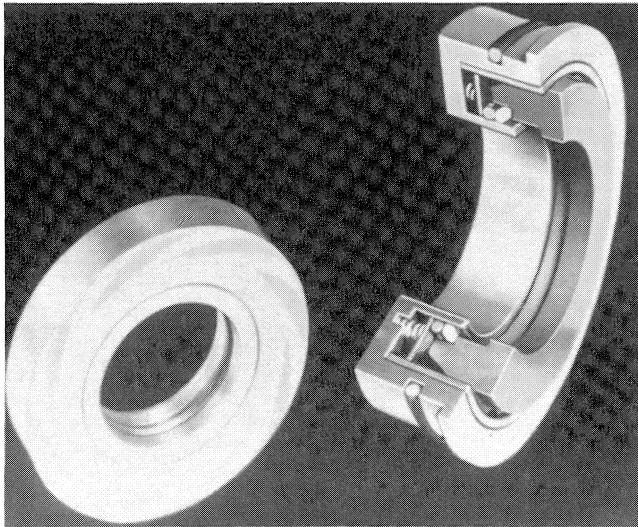


Figure 2. Seal Cartridge Assembly.

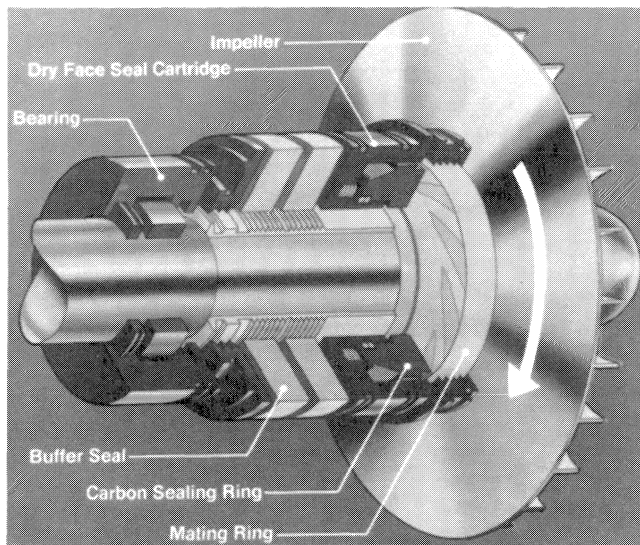


Figure 3. Seal Assembly on Shaft.

## OPERATION

The operation of the seal may be described as follows:

- The mating ring has logarithmic spiral grooves of 10 degrees to 30 degrees etched in its surface, which have a depth

of 100 to 400 micro inches (100-400  $\mu\text{in}$ ). The grooves do not extend all the way to the inside diameter of the primary sealing ring, so that there is an annular area of full contact between the mating ring and the primary sealing ring at rest. This annular area is called the sealing dam and provides tight sealing at rest.

- Upon rotation the spiral grooves pump fluid inwards, creating a pressure rise in the grooves. This pressure field causes the primary sealing ring to "lift off" against the spring forces, creating a narrow gap over the dam area. Fluid spills over the dam towards the low pressure side of the seal, resulting in a controlled leakage, and the seal is now a film riding seal.

- The position of the primary sealing ring is controlled by hydrostatic and hydrodynamic forces as shown in Figure 4. The closing force ( $F_c$ ) consists of the static pressure force ( $P_c$ ) on the primary sealing ring on the high pressure side of the seal, the spring force ( $P_s$ ) and the static pressure force ( $P_i$ ) on the primary sealing ring on the low pressure side of the seal, all pushing to the left in Figure 4. The opening force ( $F_o$ ) is created by the hydrodynamic action of the spiral grooves and dam pushing to the right. The opening forces and the closing forces reach equilibrium at a certain gap opening.

The static balance of the seal is expressed by the equation:

$$b = \frac{(D_C)^2 - (D_B)^2}{(D_C)^2 - (D_I)^2}$$

where:

$D_C$  = groove diameter

$D_B$  = balance diameter

$D_I$  = inside diameter

- The value  $b$ , which expresses the pressure balance on the primary seal ring, should be larger than 0.60, otherwise the operation may be unstable. A good value is  $b = 0.75$ , but the choice depends on the desired operating conditions. A high value of static balance "retards" the lift off, but gives a more stable operation and less leakage. It also seals better at rest.

- At the operating speed, the film gap is approximately 200  $\mu\text{in}$  to 500  $\mu\text{in}$ , depending on the design and operating conditions.

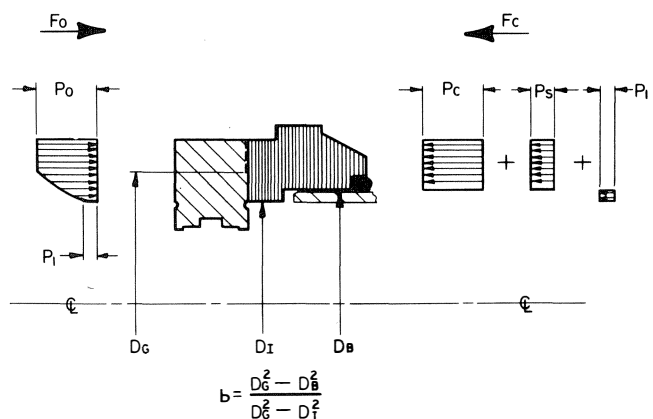


Figure 4. Hydrostatic Pressure Distribution.

## OPERATING ENVELOPE

Conservative limits of operation may be represented by the values in Table 1. Higher values are possible but should be confirmed by the machinery manufacturer.

Table 1. Conservative Limits of Operation.

Molecular Weight	Helium through Freon
Differential Pressure	600 psi (1200 psi maximum)
Temperature	500° F (750° F maximum)
Average Face Velocity	500 ft/sec
Face Rim Velocity	625 ft/sec
Operating Speed	60,000 cpm (90,000 cpm)
Any Clean or Filtered Dry Gas	

A dirty gas must be filtered and dried before allowing it to be pumped through the seal. A few gases do require special materials to avoid problems with prolonged use in a difficult environment. The user should discuss the intended operation with the machinery manufacturer, who should have the specific experience necessary to avoid these problems.

SEAL LEAKAGE

An important characteristic is the rate of leakage of this type of seal. At rest there is no leakage, provided the seal is designed with the proper static balance of  $b=0.60$  or higher. Leakage rate as a function of pressure for various shaft diameters is shown in Figure 5.

As an example, a 4.0 in shaft rotating at 10,000 cpm with a 600 psi differential pressure across the seal, operating on air at ambient temperature, gives a leakage rate of only 1.5 standard cubic feet per minute (scfm).

Notice that the leakage curves do not go through zero, at zero differential pressure. This is the nature of this seal. It is a film riding seal and it must pump fluid through the dam area or it would overheat. Only at rest will the leakage be zero at any pressure, which makes this an automatic shutdown seal.

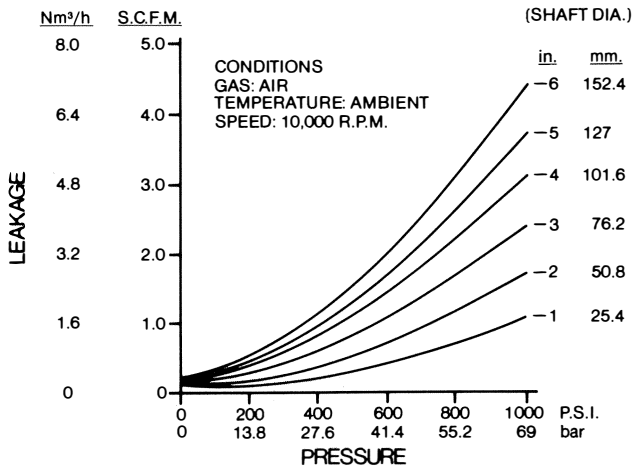


Figure 5. Seal Leakage Diagram.

The leakage through the seal at a certain speed can be represented by the following relationship:

$$W = K_1 \frac{h^3 (P_o^2 - P_i^2)}{\mu R T_i 1\pi (D_o/D_i)}$$

where

- W = leakage (weightflow)
- K<sub>1</sub> = constant

- h = seal gap
- P<sub>o</sub> = pressure at the outside diameter
- P<sub>i</sub> = discharge pressure at the inside diameter
- μ = dynamic viscosity
- R = gas constant
- T<sub>i</sub> = absolute temperature at the inside diameter
- D<sub>o</sub> = outside diameter
- D<sub>i</sub> = inner diameter

The exact quantitative relationship depends on the groove geometry, speed, and static balance b. Calculation of the film thickness and power loss requires the use of a computer program involving many variables.

POWER LOSS

The heat generated by the seal in the film gap can be expressed by the relationship:

$$Q = K_2 \frac{\mu [N^2 (D_o^4 - D_i^4)]}{h}$$

where:

- Q = generated heat
- K<sub>2</sub> = constant
- N = operating speed

Since the seal is essentially a gas thrust bearing, the power loss or generated heat is very low.

Cooling of the seal is normally not required. Most of the heat is dissipated to the primary sealing ring and mating ring and a small amount of heat is carried away by the fluid as it is pumped through the seal. The primary seal ring and mating ring will reach an equilibrium temperature.

The following represents typical conditions on a high speed spiral grooved face seal:

Shaft diameter	1.25 in
Speed	52,000 cpm
Pressure	400 psig
Temperature	300°F
Axial temperature gradient in the primary sealing ring	53°F
Face temperature of the primary sealing ring	410°F
Leakage rate	0.25 scfm

ADVANTAGES

The advantages of the gas lubricated face seal are:

- Simple design
- No seal oil, overhead tanks, or other support or control system required
- Very low leakage
- Very low power loss
- Zero leakage at rest (shut down seal)
- Very high operating speeds are possible (above criticals)
- No maintenance
- Relatively low cost
- Low installation cost

DISADVANTAGES

There are some disadvantages to the gas lubricated face seal, but they are not serious and they are far outweighed by the advantages. The disadvantages include:

- Sensitivity to dirt, eliminated by using filtered gas.
- Leakage at zero differential pressure when rotating.
- High machining tolerances for mating ring seating on the shaft. (The mating ring must run true.)

- Partial disassembly of the rotor is required to install the seal. This can be greatly facilitated by an appropriate impeller to shaft mounting system.

### CLEAN-UP LOOP

The sensitivity to dirty gas can be eliminated by means of buffer gas injection or a "clean-up loop," similar to the one shown in Figure 6.

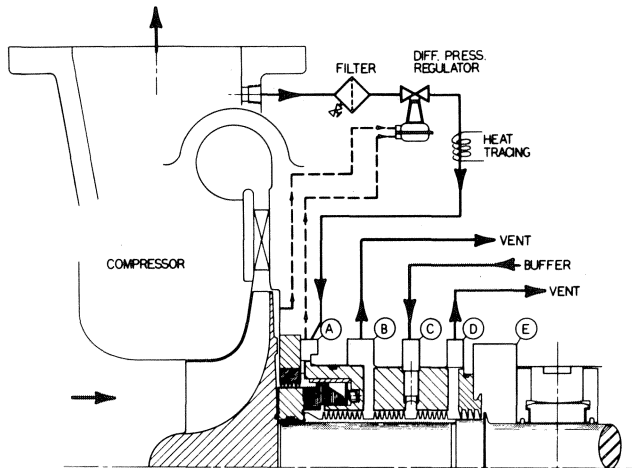


Figure 6. Clean-Up Loop. (A-E represent ports).

To operate with a clean-up loop, it is necessary to provide a labyrinth seal on the high pressure side of the seal. High pressure gas from the compressor discharge is filtered through less than ten micron filters, heated or cooled, if necessary, and fed to the cavity (A) between the labyrinth seal and the face seal. The high pressure cleaned gas leaks back to a space of lower pressure behind the impeller of the compressor and only clean gas will be pumped through the seal.

The same objective can be obtained by injecting a buffer gas to the cavity (A). This is preferred when operating with hot, contaminated, toxic or oil vapor containing gases.

This arrangement solves the problem of the sensitivity to dirt.

### INSTRUMENTATION

Normally, instrumentation is not required for the proper operation of the seal. During endurance testing of the seal, some temperature sensors have been installed in the primary sealing ring in order to establish temperature gradients. An interesting result was that the moment of lift off could be observed by monitoring a thermocouple imbedded in the primary seal ring just under the surface. A typical temperature-time diagram observed is shown in Figure 7, which clearly shows the moment of lift off.

As a result of these tests, the recommendation is made for installation of two thermocouples in the carbon ring just under the sealing surface, for two reasons:

- To monitor lift off.
- By keeping a log of the temperature readings, early warning can be obtained that the grooves are filling up with dirt—if such should occur—and a shutdown can be scheduled to clean the seal.

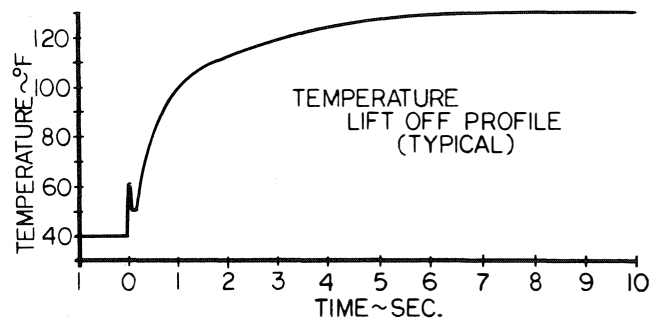


Figure 7. Temperature/Time History.

### TYPICAL APPLICATIONS

The spiral grooved face seal can be applied in a number of ways (Figures 8, 9, and 10).

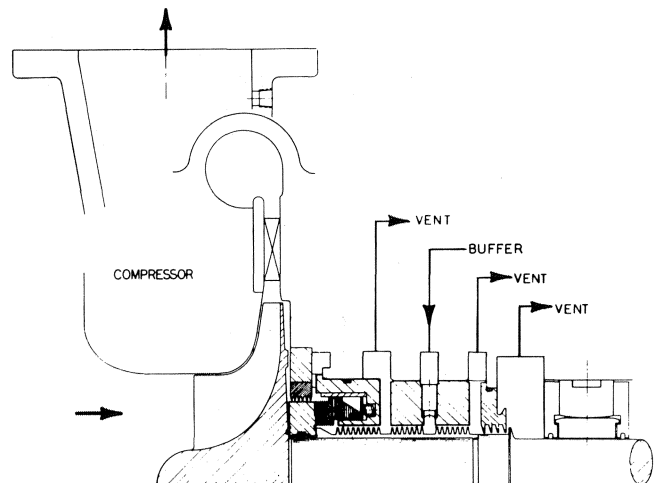


Figure 8. Single Seal Arrangement.

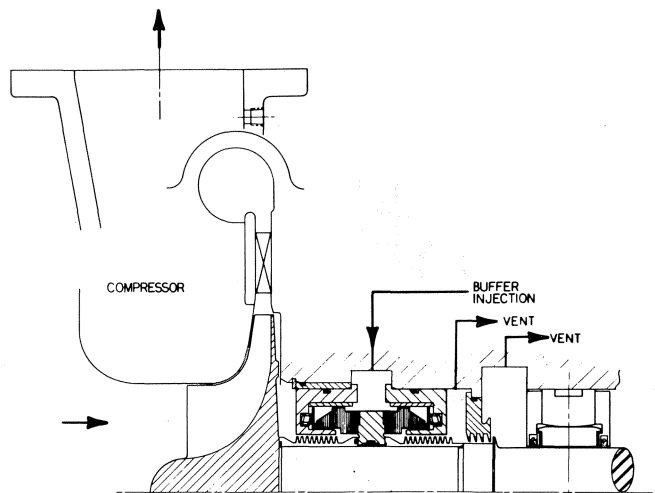


Figure 9. Double Seal Arrangement.

- A single seal without buffer gas may be used for applications with non-polluting clean gases which may be vented to the atmosphere.
- A single seal with a clean-up loop, but without buffer gas may be used for applications with non-polluting dirty gases, which may be vented to the atmosphere.

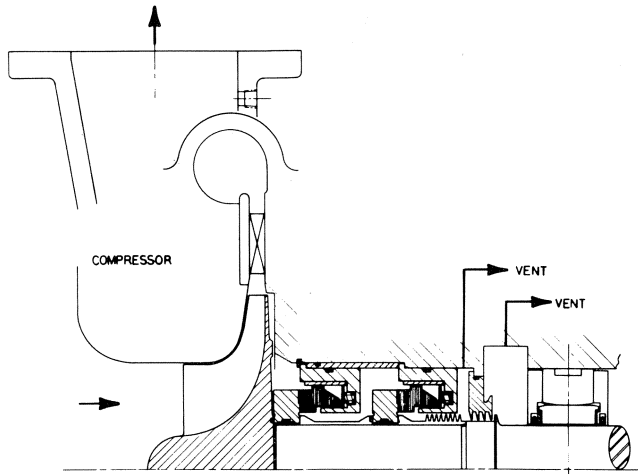


Figure 10. Tandem Seal Arrangement.

- A single seal with buffer gas may be used for applications with hot, contaminated, toxic, flammable gases, or gases which can plug up the spiral grooves of the mating ring.
- A double seal with buffer gas may be used for applications with contaminated, toxic or flammable gases. By controlling the pressure drop across the inboard seal, leakage to the process can be minimized.
- A tandem seal may be used for high pressure applications and/or low leakage to the atmosphere.

RECORD OF INSTALLATIONS

A record of installations, including the process gas, shaft diameter, operating speed, process gas pressure and temperature, test hours and operating hours, as well as the number of starts and stops, is shown in Table 2. The various operating conditions and test experience with this seal by the manufacturer and a customer is shown in Table 3. Total Accumulated numbers as of April 1, 1985, indicated in Table 2, are:

Total testing time	2000 hr
Total operating time	66,000 hr
Average operating time per seal	2400 hr

Table 2. Record of Installations.

Ref.	Qty.	Gas Consumption	Shaft Dia."	Speed kcpm	P (psia)	T (°F)	Test (hr)	Oper. (hr)	Comments	
1	1	CO <sub>2</sub>	1.25	44.0	340	210	4	12,000	Intermittent Duty	
2	1	Hydrocarbons	1.75	11.0	270	270	8	24,000	Variable Speed	
3	1	Ammonia	1.75	21.0	35	250	50	-	Project Terminated	
4	1	Ammonia	1.25	33.0	70	260	50	-		
5	1	Ammonia	1.25	52.0	140	245	50	-		
6	1	He, N <sub>2</sub> , CO <sub>2</sub>	1.25	26.0	20	280	4	8,000	Laser Research	
7	1	Hydrocarbons	1.75	15.0	160	150	20	12,000	Continuous Operation	
8	4	Hydrocarbons	1.25	3.6	50	375	40	50/1000	260 Starts/Stops	
9	4	Hydrocarbons	1.25	3.6	50	375	40	50/1000	160 Starts/Stops	
10	1	Methanol	1.75	10.0	95	350	8	200	Experimental Process	
11	1	Methanol	1.75	26.0	230	410	8	200		
12	2	Methyl Chloride	1.25	38.0	65	220	200	4,000	Continuous Operation	
13	2	Methyl Chloride	1.75	23.0	130	220	120	4,000		
14	1	CO <sub>2</sub>	1.75	21.0	52	309	200	-	Ready for Startup	
15	1	CO <sub>2</sub>	1.75	38.0	219	370	200	-		
16	1	H <sub>2</sub> , CO <sub>2</sub> , CO	1.75	27.0	146	213	20	-		
17	1	N <sub>2</sub>	1.75	44.0	200	255	20	-		
18	1	N <sub>2</sub>	1.75	44.0	367	215	20	-		
							26	1,062	66,400	

Table 3. Seal Test Experience.

Ref.	Qty.	Gas Composition	Shaft Dia."	Speed (kcpm)	P (psia)	T (°F)	Test (hr)	
19	1	N <sub>2</sub>	1.75	31.	52	207	20	
20	1	N <sub>2</sub>	1.75	34.	95	220	20	
21	1	N <sub>2</sub>	1.25	44.	244	203	20	
22	1	N <sub>2</sub>	1.25	48.	428	207	20	
23	1	N <sub>2</sub>	1.75	13.	200	550	-	
24	1	H <sub>2</sub> , C <sub>1</sub> , C <sub>3</sub> , CO	1.75	11.	270	136	-	
25	2	N <sub>2</sub> , CO <sub>2</sub> , H <sub>2</sub> S	3.50	7.8	18	177	-	
26	1	C <sub>2</sub> , C <sub>3</sub>	1.75	25.	56	219	-	
27	1	C <sub>2</sub> , C <sub>3</sub>	1.25	52.	127	315	-	
							10	80
Seal Test Rig								
28*	1	H <sub>2</sub> /N <sub>2</sub> /CO <sub>2</sub>	1.25	52.	450	400	120	
29**	1	NH <sub>3</sub> /CO <sub>2</sub>	1.25	44.	450	400	1000	
								1120

\*Tested by Manufacturer.  
 \*\*Tested by Customer.

The end users are at first reluctant to use the gas lubricated face seal, because they are not familiar with it; however, once they have tried it, they are quite pleased with its performance and reliability. Field experience has shown that, if properly installed and operated, the seal will run with minimum maintenance for several years with many starts and stops.

Personnel involved with these seals have been impressed by this seal in comparison with the oil bushing seal, which tends to be rather temperamental and requires a substantial and expensive support system and control system. The "low maintenance" feature has been of particular interest. Some customers are retrofitting machines with the gas lubricated face seal, which is possible because the seal cavity can be made dimensionally identical to the one required for a labyrinth seal.

TYPICAL CUSTOMER CONCERNS

Typical customer concerns pertaining to seals are safety, reliability, leakage rate, maintenance and costs. The gas lubricated face seal can address these concerns as follows:

*Safety.* If the gas is toxic or flammable, and since the leakage rate is very low for any kind of gas, it takes only a small amount of buffer gas to drive the leakage into the seal vent manifold, where it can be recovered or flared. A second face seal used as an end seal would reduce the buffer gas consumption to a minimum. Thus, this seal can function in a very safe manner, with the second seal as a back-up seal.

*Reliability.* For a clean gas application, reliability is extremely high without any special arrangements. If the process gas is dirty, it can be easily filtered and cleaned before it is pumped through the seal, as shown in Figure 6. If desired, thermocouples can be installed in the primary seal ring for additional peace of mind. The temperature of the primary seal ring can be monitored and the machine safely shut down for seal cleaning, if the temperature exceeds a preset limit, indicating reduced film thickness.

*Leakage rate.* The leakage rate depends on the gas, the hydrostatic balance, the speed, the shaft diameter and the differential pressure across the seal. Typically for a two inch seal at 10,000 cpm, and 200 psi differential pressure, the leakage rate is 0.2 scfm or 300 scfd or 100,000 standard cubic feet per year, assuming continuous operation. If a small amount of buffer gas is acceptable, the leakage can be returned to the inlet of the compressor.

*Maintenance.* Maintenance is minimal. There are no controls, overhead tanks, traps, pumps or seal oil contamination to worry about. The problem associated with contaminated seal oil disposal is eliminated. If the seal should need cleaning, a simple impeller mounting, such as a friction drive and polygon, allows easy access to the seal.

With proper installation, using clean or filtered gas, the seal will operate for many years without cleaning.

*Cost.* Even without the ongoing savings of minimal maintenance, initial cost is another positive attribute of the gas lubricated face seal. Savings depend on the installation, but typical savings may be conservatively estimated as follows:

• Elimination of an API 614 seal console	\$35,000
• Elimination of overhead tanks, etc.	\$10,000
• Elimination of seal oil traps	<u>\$ 3,000</u>
Savings:	\$48,000

In addition, real estate in the installation area may be at a premium and the space saved could easily cover the capital cost.

## CONCLUSIONS

It has been established through testing and field experience that the gas lubricated spiral grooved face seal is a very desirable seal to use in the process industry, which has a number of advantages as listed earlier.

If properly manufactured, applied, installed and operated, there is basically nothing that can go wrong with the seal and it should run indefinitely.

So far, the gas lubricated face seal has an excellent operating record with a large range of applications and wide operating envelope.

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