# DRY GAS COMPRESSOR SEALS

by

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## ABSTRACT

Dry gas seals can eliminate some of the conventional problems associated with seal oil systems. Increased safety, lowered maintenance and improved reliability are key considerations for retrofitting with gas seals. Understanding the dry gas sealing technology and optimizing the seal selection for a given set of operating parameters is crucial for the successful application on rotating equipment. Control systems incorporating filtration, leakage, and pressure monitoring can provide real time diagnosis of seal performance.

## INTRODUCTION

The gas seal design discussed herein is the spiral groove type. Various types of face configurations have been proposed by several manufacturers over the past fifteen years. The spiral groove face configuration design is one which has been successfully developed and applied on centrifugal compression equipment.

An attempt has been made to provide some seal design criteria, the different types of seal arrangements and to emphasize the need for optimum seal and control system selection. The process conditions, namely pressure, temperature, gas composition, and contaminants in the gas stream are an integral part of the seal design process. Of equal importance are the key characteristics of the compression equipment involved, namely the rise in temperature from suction to discharge, the surface speed involved, whether an overhung or beam unit, rotordynamics, etc.

Review of the location of the compression equipment in the overall process loop can provide important information on the contaminants in the gas stream. P & I D reviews can provide possible alternative choices of buffer gas supply for the seal. Above all, a change from the existing wet seal system to dry seals can possibly impact plant efficiencies positively by eliminating oil contamination of downstream equipment, catalysts, etc.

## SPIRAL GROOVE GAS SEAL DESIGN

A cross-sectional view of a spiral grooved gas seal is shown in Figure 1. The rotating assembly consists of the mating ring (with spiral grooves) mounted on a shaft sleeve held in place axially with a clamp sleeve and a locknut. It is typically pin or key driven. The stationary assembly consists of the primary ring mounted in a retainer assembly held stationary within the compressor housing. Under static conditions, the primary and mating rings are held in contact due to the spring load on the primary ring.
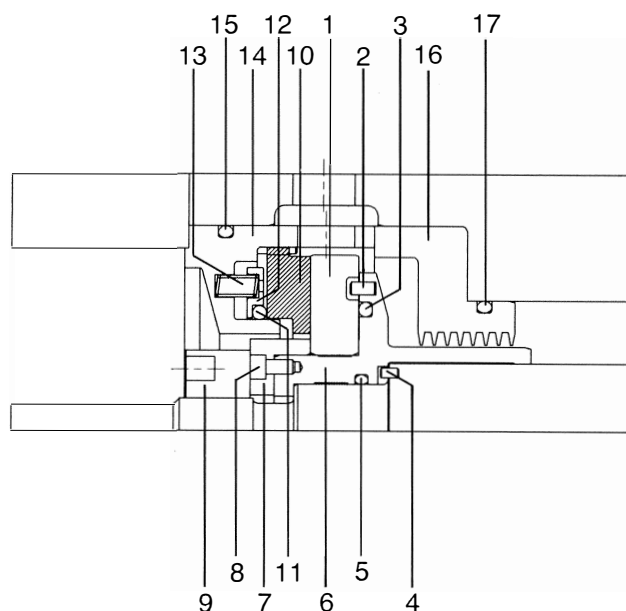


*Figure 1. Dry Seal Configuration. 1) mating ring, 2) pin (rotating), 3) O-ring, 4) pin, 5) O-ring, 6) shaft sleeve, 7) clamp sleeve, 8) cap screw, 9) locknut, 10) primary ring, 11) O-ring, 12) disc, 13) spring, 14) retainer, 15) O-ring, 16) labyrinth, 17) O-ring. Items 1-9 represent rotating assembly. Items 10-17 represent stationary assembly.*

## OPERATING PRINCIPLE

The operating principle of the spiral grooved gas seal is that of a hydrostatic and hydrodynamic force balance. Under pressurization, the forces exerted on the seal are hydrostatic and are present whether the mating ring is stationary or rotating. Hydrodynamic forces are generated only upon rotation. The mating ring consisting of the logarithmic spiral grooves is the key to generating these hydrodynamic forces.

The spiral groove pattern on the mating ring is shown in Figure 2 rotating in a clockwise direction. As gas enters the grooves, it is sheared towards the center. The sealing dam acts as a restriction to the gas outflow, thereby raising the pressure upstream of

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the dam. This increased pressure causes the flexibly mounted primary ring to separate from the mating ring [1, 2]. The mating ring with spiral grooves and the primary ring held within the retainer assembly are shown in Figure 3.
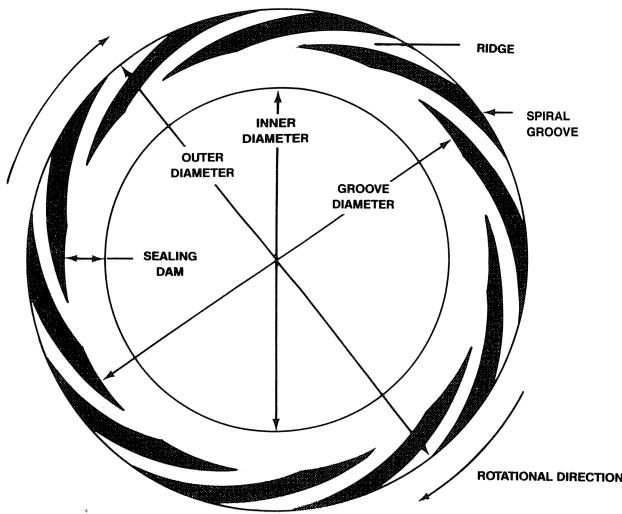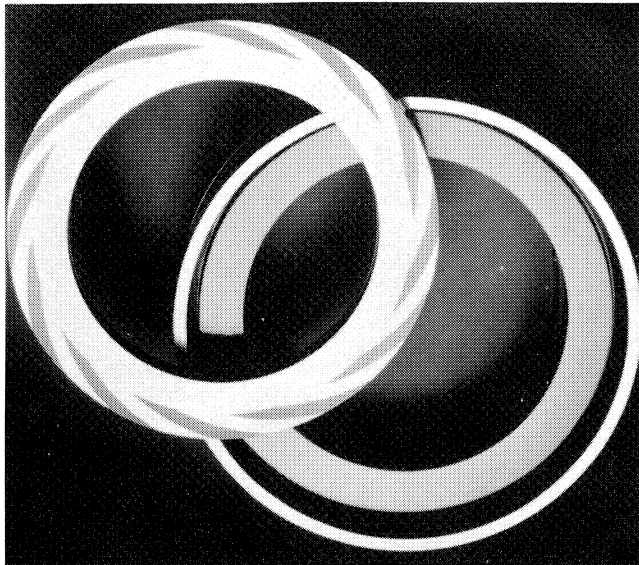


Figure 2. Spiral Grooved Mating Ring.



Figure 3. Dry Seal Faces.

## OPERATING RANGE

Gases ranging from inert gases such as nitrogen to highly toxic gaseous mixtures of natural gas and hydrogen sulfide can be sealed utilizing the optimum seal arrangements. The operating range of the spiral grooved gas seals is as follows:

Sealed Pressure:    2,400 psi
Temperature:        500°F
Surface Speed:      500 ft./sec.
M.W.:               2-60

The materials of construction are as follows:

| | |
|---|---|
| Mating Ring: | Tungsten Carbide, Silicon Carbide |
| Primary Ring: | Carbon, Silicon Carbide |
| O-Rings: | Elastomers (Viton™, Kalrez™) |
| Hardware: (Sleeves, discs, etc.) | 300 or 400 series ss |
| Coil Springs: | 316 ss, Hastelloy |

The gas composition, contaminants in the gas stream, operating temperatures and process conditions dictate the choice of materials.

## SEAL COMPRESSOR INTERFACE

The characteristics of seals incorporating the hydrostatic and hydrodynamic force balance are listed in Table 1.

Table 1. Seal-Compressor Interface.

- Dry seals can withstand rapid speed and pressure changes.
- Allowance for axial shaft shifts is normally ± .100″; larger shifts can be accommodated.
- Radial shaft movements allowed are ± .025″; larger allowance is possible.
- All rotating seal components are balanced.
- Shaft tilts, well in excess of those encountered on overhung and beam compressors, are routinely handled.
- Dry seal is unaffected by shaft vibrations.
- Accidental contamination causing wetting of sealing surfaces by bearing oil migration during running or idle periods, has not been found harmful to seal performance.
- Seal is capable of starts and stops at full pressure conditions on the seal.

## PERFORMANCE CHARACTERISTICS

### Seal Leakage

Since the typical operating gaps between the two sealing surfaces range from 0.0001 in to 0.0003 in, the resultant leakage is very small in magnitude. The size and speed effect on leakage are shown in Figure 4. Under conditions of static pressurization beyond 50-75 psi, the seal leaks a very small amount. This leakage increases with increasing pressure. For example, four inch shaft seal on a natural gas compressor statically pressurized to 1,000 psi will leak about one scfm. Under dynamic condition, due to the pumping effect of the spiral grooves, the leakage increases as well.
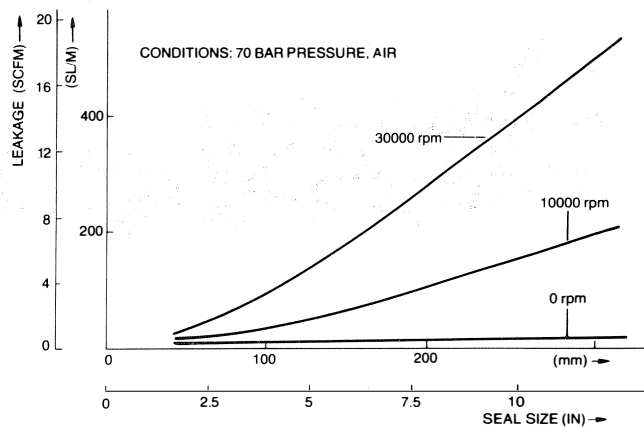


Figure 4. Size and Speed Effect on Leakage.

The pressure, temperature, and gas effect on leakage is shown in Figure 5. Increased viscosity of gases at higher temperatures reduces the amount of seal leakage.
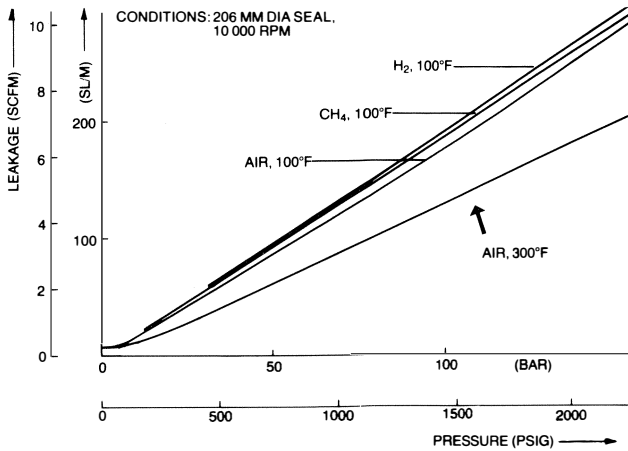


Figure 5. Pressure, Temperature and Gas Effect on Leakage.

## Power Loss

Since the sealing surfaces are noncontacting under dynamic conditions, the power loss associated with spiral grooved seals is very low. The pressure and thermal distortions of the sealing surfaces are computed and compensated for in the seal design. The power loss associated with increasing shaft speeds is shown in Figure 6. The comparative economic evaluation of the wet seal and dry seal systems is listed in Table 2.

## SEAL ARRANGEMENTS

Analysis of the operating conditions and process parameters must be carefully undertaken in order to determine the optimum seal arrangement requirements. Sealing pressure, temperature, speed, gas composition and contaminants are factors governing the seal selection procedure. Since there is a small leakage from the seal, venting, flaring, or primary leakage recovery systems should be considered as part of the seal arrangement and control system design.

Table 2. Comparative Economic Evaluation of Wet Seal vs Dry Gas Seal.

| Comparative Economic Evaluation of Wet Seal vs. Dry Gas Seal | | |
|---|---|---|
| | WET OIL SEALS | DRY GAS SEAL |
| Seal oil support systems costs | Pumps, reservoirs, filters, traps, coolers, consoles. | None |
| Seal oil consumption | 1-100 gallons/day | No seal oil |
| Maintenance cost | A major expenditure over equipment life. | Negligible |
| Energy costs | Seal Power loss: 10-30 HP Unit driven pumps: 20-100 HP | 1-2 HP |
| Process gas leakage | Gas Leakage: 25SCFM & higher | Less than 2 SCFM |
| Oil contamination | Of Pipelines: High clean up costs Of Process: Catalyst Poisoning | None |
| Toxic and corrosive Applications | Buffer Gas Consumption (egN$_2$) 40-70SCFM | 2-4SCFM |
| Unscheduled shutdowns | High downtime costs | Very reliable |
| Aborted startups | Frequent | Rare |
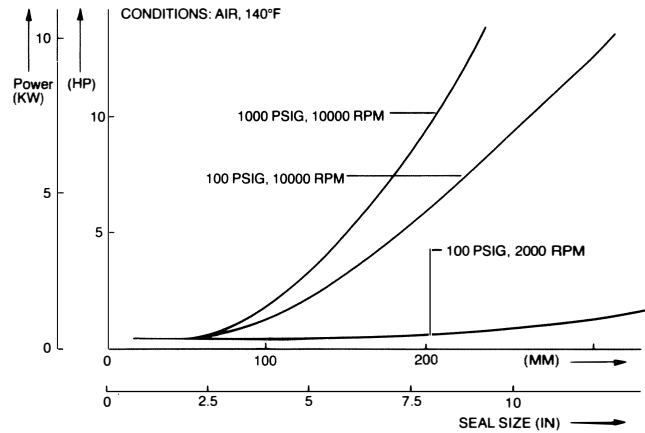


Figure 6. Gas Seal Power Loss.

There are five basic arrangements that can be prescribed for a diverse group of applications.

## Single Seal Arrangement

### Operating Limit
Pressure:        400 psi max.
Temperature:     500°F
Speed:           500 ft./sec.
Gases:           Air, Nitrogen, etc.
Service:         Wet Gas Compressors, Plant Air & Nitrogen Service

Gases that are inert or nontoxic are typically sealed by this seal arrangement (Figure 7). The leakage from the seal is either vented or flared. On wet gas compressors in refinery services the leakage can be educted by a steam eductor and flared (Figure 8). The control system requirements are shown in Figure 9 for a single seal arrangement.
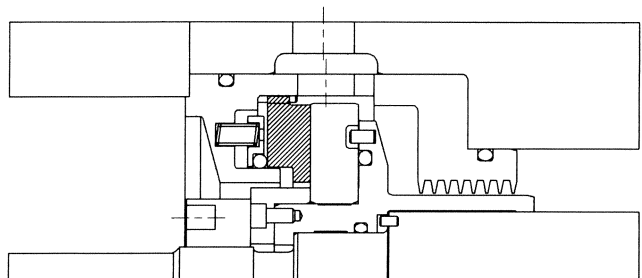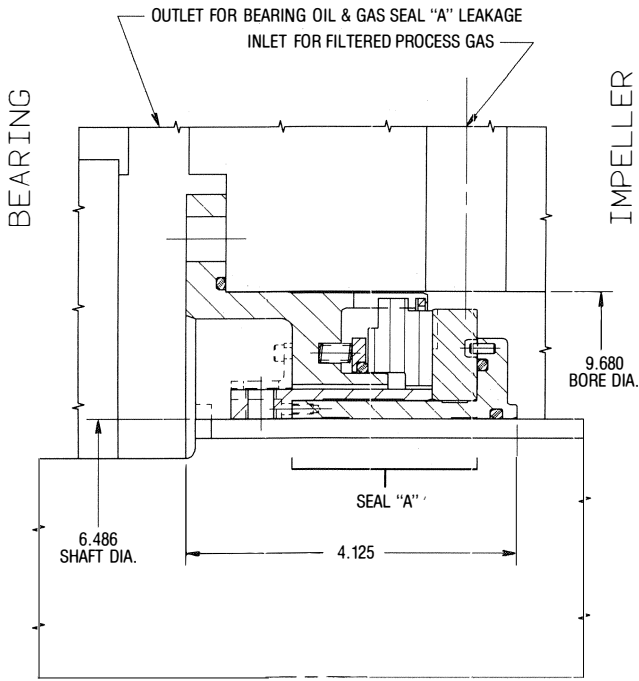


Figure 7. Single Seal Arrangement.

## Tandem Seal Arrangement

### Operating Limit
Pressure:        1,200 psi
Temperature:     −135°F to 500°F
Speed:           500 ft./sec.
Gases:           All
Service:         Natural Gas Pipeline Compressors and Process Gas Compressors

A majority of the hydrocarbon mixtures, chemical and petrochemical process gases and gases having toxic and corrosive contaminants such as hydrogen sulphide have to be sealed from

*Figure 8. Single Seal Arrangement—Wet Gas Compressor.*

GAS HANDLED: *****
SHAFT SPEED: 8,100 RPM
DISCHARGE PRESSURE: 33.7 PSIA
DISCHARGE TEMPERATURE: 100°F

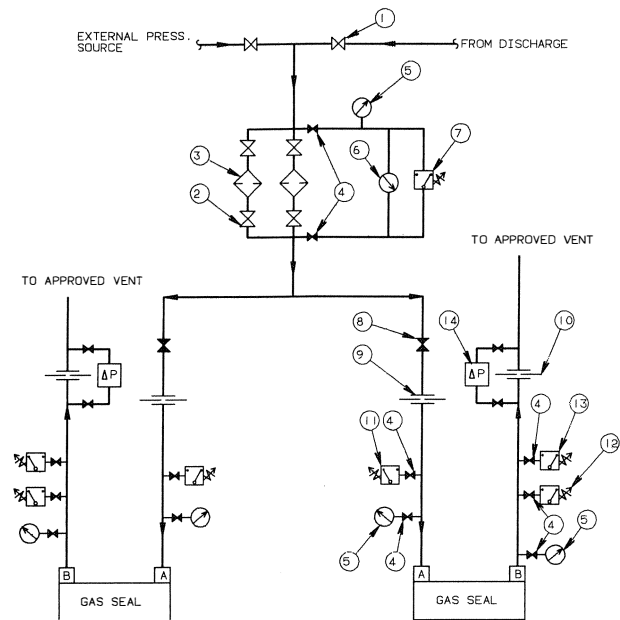| *****GAS | % BY WEIGHT |
|---|---|
| PROPYLENE | 16.30 |
| METHANE | 13.52 |
| BUTYLENES | 12.81 |
| PENTANES | 10.16 |
| ISOBUTANE | 7.65 |
| ETHANE | 6.20 |
| ETHYLENE | 6.14 |
| PROPANE | 5.31 |
| HYDROGEN SULFIDE | 4.95 |
| INERTS | 4.55 |
| HYDROGEN | 4.47 |
| HEXANES | 3.13 |
| WATER VAPOR | 2.74 |
| NORMAL BUTANE | 2.07 |



*Figure 9. Control Panel for Single Seal. 1) ball valve, 2) ball valve, 3) filter, 4) needle valve, 5) pressure gauge, 6) regulating valve, 9) orifice union, 10) orifice union, 11) pressure switch, 12) pressure switch, 13) pressure switch, 14) differential pressure gauge.*

the environment and the lubrication systems. In a tandem seal arrangement (Figures 10 and 11), two seal modules are oriented in the same direction behind each other. The first seal (inboard) handles full pressure while second seal (outboard) would run as standby or backup seal with zero pressure differential. The backup seal then functions as an additional barrier between the process gas and atmosphere. The primary leakage from the first seal can be safely vented or flared. The control system for a tandem seal arrangement is shown in Figure 12.
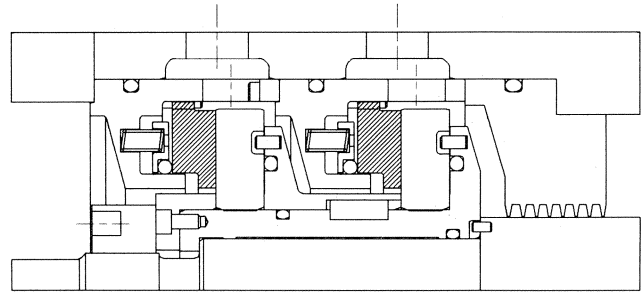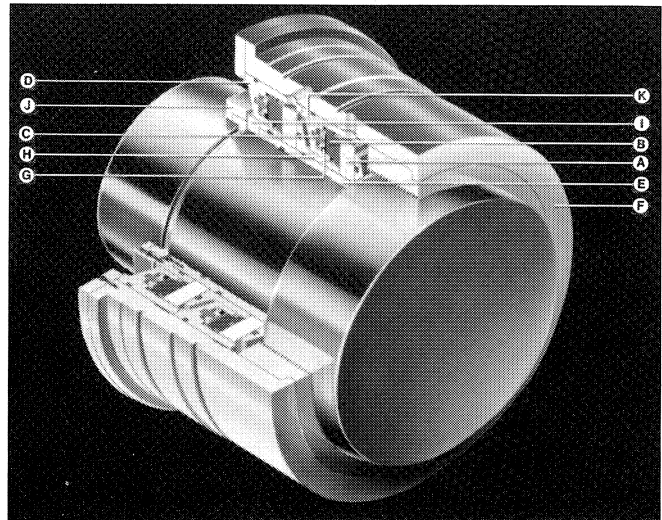


*Figure 10. Tandem Seal Arrangement.*



*Figure 11. Isometric View: Tandem Seal Arrangement. A) mating ring, B) primary ring, C) spring, D) retainer, E) shaft sleeve, F) housing sleeve, G) secondary O-ring, H) stub sleeve, I) clamp sleeve, J) lock sleeve, K) disc.*

For natural gas and hydrogen sulfide mixtures or ammonia, the second seal (seal B) in the Tandem seal arrangement should be buffered with an inert gas such as nitrogen or sweet natural gas (if nitrogen is not available). This prevents the hydrogen sulfide or ammonia from mixing with the bearing oil. The buffer arrangement on the second seal (seal B) is represented in Figures 13 and 14.

*Triple Seal Arrangement*

*Operating Limit*

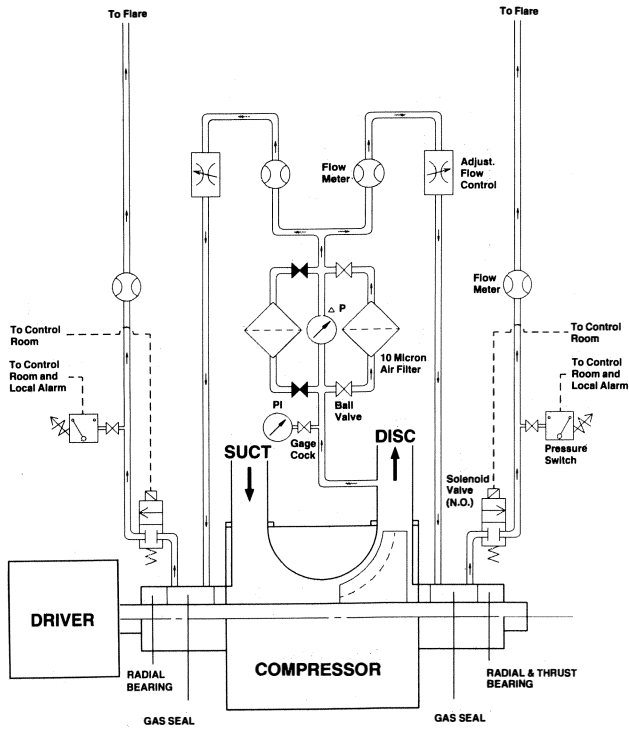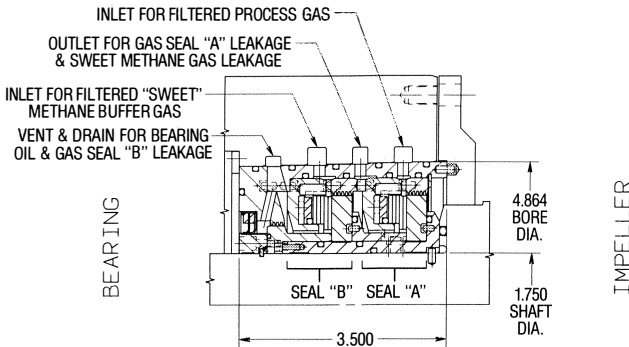| | |
|---|---|
| Pressure: | 2,400 psi |
| Temperature: | 500°F |
| Speed: | 500 ft./sec. |
| Gases: | Natural, Hydrogen, etc. |
| Service: | Hydrogen Recycle Compressors and Gas Reinjection Compressors |

Figure 12. Control System: Tandem Seal Arrangement.



GAS HANDLED: AMMONIA
SHAFT SPEED: 10,550 RPM
DISCHARGE PRESSURE: 204 PSIG
DISCHARGE TEMPERATURE: 268°F

Figure 14. Tandem Seal Arrangement: Ammonia Service.



GAS HANDLED: METHANE & HYDROGEN SULFIDE
SHAFT SPEED: 23,550 RPM
DISCHARGE PRESSURE: 1,500 PSIG
DISCHARGE TEMPERATURE: 300°F

Figure 13. Tandem Seal Arrangement: Sour and Toxic Service.



GAS HANDLED: HYDROGEN
SHAFT SPEED: 10,500 RPM
DISCHARGE PRESSURE: 1,800 PSIG
DISCHARGE TEMPERATURE: 150°F

Figure 15. Triple Seal Arrangement.

In this arrangement (Figure 15), the sealing pressure is broken down across the first (seal A) and second (seal B) seal. The primary leakage is after the second seal. The seal design permits a natural breakdown (approximately 50 percent across each seal) of pressure across the two seals without any external controls. The control system for a triple seal arrangement is shown in Figure 16.

## Double Opposed Arrangement

### Operating Limit

Pressure:          250 psi
Temperature:       −80°F to 500°F
Speed:             500 ft./sec.
Gases:             HCl, Styrene, Methyl Chloride, etc.
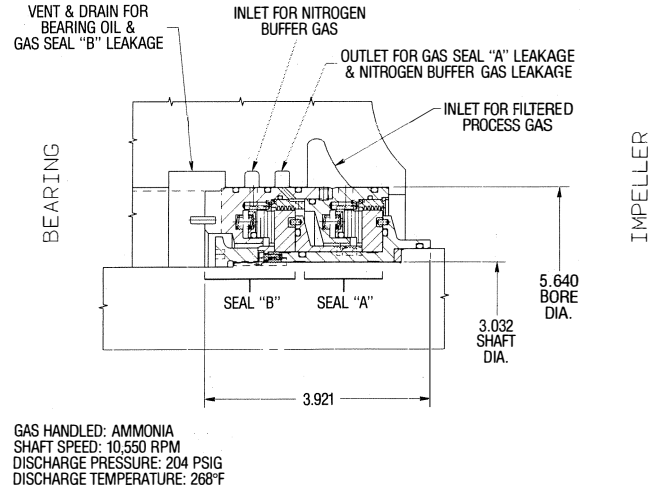Service:           Toxic and Corrosive Process Gas Compressors
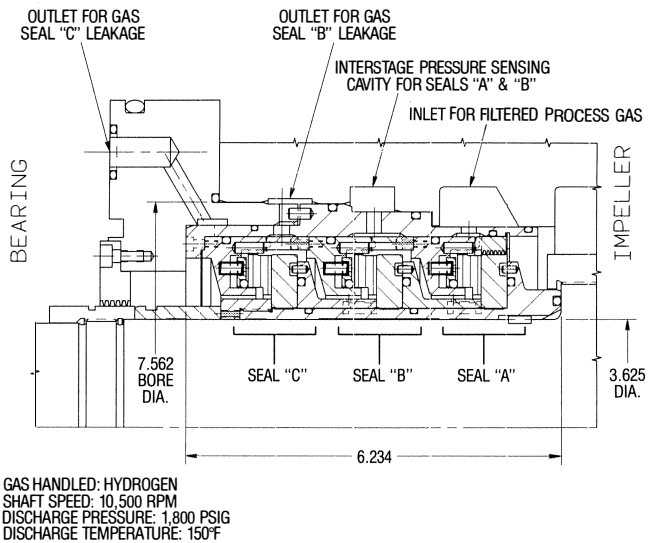
In processes where zero leakage to the atmosphere is the plant safety requirement, a double opposed seal arrangement (Figure 17) with plant nitrogen buffer should be utilized. The nitrogen buffer leaks into the process at a selective rate (< 1 scfm) and also leaks (< 1 scfm) to the atmosphere. This prevents atmospheric and bearing oil contamination. A thorough investigation of the plant safety procedures must be undertaken and should incorporate the review of control system requirements for the double opposed seals.

## Non-Elastomer Arrangement

### Operating Limit

Pressure:          1,000 psi
Temperature:       750°F
Speed:             500 ft./sec.
Gas:               Steam, Process Gas
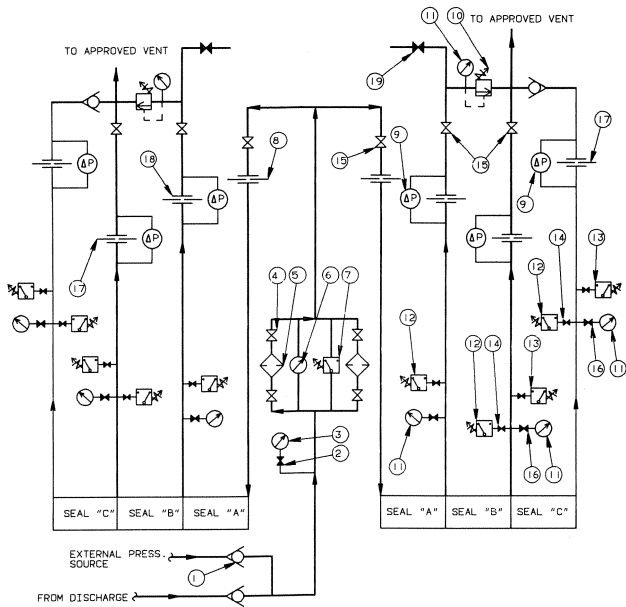Service:           Steam Turbines, Turbo-expanders

*Figure 16. Control System: Triple Seal Arrangement. 1) check valve, 2) block/bleed valve, 3) pressure gauge, 4) ball valve, 5) filter, 6) differential pressure gauge, 7) differential pressure switch, 8) orifice union, 9) differential pressure gauge, 10) pressure relief valve, 11) pressure gauge, 12) pressure switch, 13) pressure switch, 14) block/bleed valve, 15) regulating valve, 16) needle valve, 17) orifice union, 18) orifice union, 19) needle valve.*
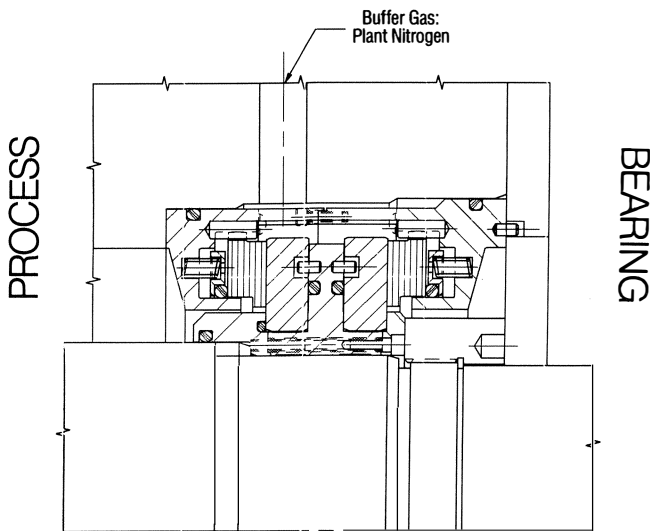


*Figure 18. Non Elastomer Seal Arrangement.*

*Special Arrangement*

   *Operating Limit*

   Pressure:          1,000 psi

   Temperature:    400°F

   Speed:               500 ft./sec.

   Fluid:                Natural Gas Liquids (NGL) Supercritical $CO_2$

Service:               Pumps

The concept of dry seals can be extended to high vapor pressure liquids on pumps. In Figure 19, the first (seal A) seal supplies enough heat to vaporize the liquid at its periphery. The result is that the seal leaks gas (vapor) that can be safely vented. The second (seal B) seal acts as a safety backup and contains the process leakage. In continuous and intermittent pump services, control of the fluid film across mechanical contacting seals has been a sealing problem within the industry.



*Figure 17. Double Opposed Seal Arrangement.*



GAS HANDLE: "LIGHT" HYDROCARBONS
SPECIFIC GRAVITY: .5445
SHAFT SPEED: 3,565 RPM
DISCHARGE PRESSURE: 490 PSIA
DISCHARGE TEMPERATURE: 125°F

*Figure 19. Vaporizing Liquids Seal Arrangement.*

For optimal dry gas seal application an analysis of the fluid composition and the change in enthalpy of the fluid is essential. Heat balance ratios must be reviewed in order to maintain a constant film of gas (vapor) at the sealing surfaces.

## CONCLUSIONS

It has been demonstrated in operating applications that a thorough review of the service conditions, plant safety, and relia-

The nonelastomeric seal design (Figure 18) permits application of this dry seal arrangement at temperatures above 500°F. Low and intermediate pressure steam turbines natural gas, and tail gas expanders are applications where high temperatures are the norm.
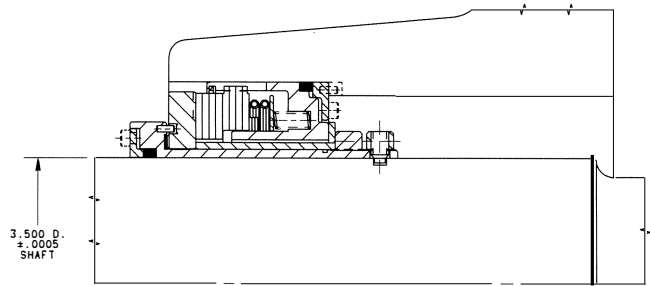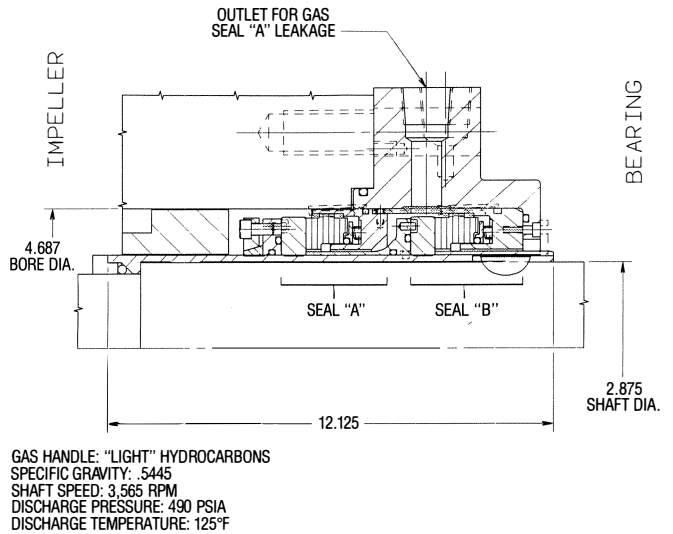
bility requirements is essential prior to consideration of gas seals.

Proper selection of seal arrangements and control systems incorporating plant operating philosophies is crucial to a successful retrofit and high availability of that piece of equipment. Whereas, with oil seals, little or no consideration was given to the process gas sealed, with dry seals the process gas itself is utilized in the seal as a buffer. Hence, it is essential to understand the seal dynamics, gas contaminants and process fluctuations.

## REFERENCES

1. Sedy, J., "A New Self Aligning Mechanism for Spiral Groove Gas Seal Stability," ASLE-79-LC-3B-3 (1979).

2. Sedy, J., "Improved Performance of film Riding Gas Seals through Enhancement of Hydrodynamic Effects," ASLE-78-LC-3B-1 (1978).