

# RELIABILITY IMPROVEMENT IN MECHANICAL SEALS ANALYSIS OF DESIGN CONSIDERATIONS

by

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## ABSTRACT

The differences between mechanical and shaft seal devices intended for high reliability and seals which were designed without reliability as the foremost consideration are examined and explained. Highlighted are such important topics as flow distribution enhancement in the seal face region, the merits of narrow face configurations, pumping screws and their influence on the simplification of the buffer system, and other related topics.

Guidelines are included which enable the user to specify meaningful acceptance tests for mechanical seal systems. These guidelines allow the user to gauge the relevance of unusual material selections, face material impregnations, and seal geometries.

## INTRODUCTION

More than one manufacturer is able to design and produce dependable mechanical seals. Indeed, most mechanical seals installed in modern rotating equipment are rightly considered to be reliable, low maintenance components. Two facts are nevertheless casting aspersions on the reputation of mechanical seals: studies have shown that out of 100 pump failures in a typical petrochemical plant as many as 70 repair events are attributable to seal distress [1], and mechanical seal-related failures can account for 50 cents of every maintenance dollar spent on rotating equipment in a typical refinery. Another study showed that the average seal-induced pump failure in the petrochemical industry results in repair costs ranging from \$2500 in a plant with an average size of 29 hp per pump (400 pumps) to \$6000 in a plant with an average size of 73 hp per pump (2600 pumps).

Experience shows that the majority of these failures could have been prevented if the selection process for the seal had been approached with more attention. Also, it is quite evident that seals experiencing frequent failures in a representative time period should be closely examined for basic design or application errors.

Optimum seal selection practices rely, to a great extent, on the experience of qualified seal vendors. Since most applica-

tions of mechanical seals require seal systems, not just seals [2], it is important that the full scope of operating data, specific machinery data and related information be made available to the seal manufacturer. Equally important is the vendor-user interaction, in which the user must show more than superficial interest in product selection. Specifically, the seal user should be thoroughly familiar with the advantages and disadvantages of certain design features and should engage in a detailed comparison of physical configuration and material property alternatives presented by the various bidders.

These, then, are the key objectives covered herein:

- It should convince the user to take a far more active part in seal selection for vulnerable sealing services or applications where high reliability or fewer failures are to be achieved.
- The user should become more familiar with the design features and concepts which distinguish seals of high potential reliability from seals with low potential reliability in critically important or difficult sealing applications.
- It should explain to the user why reliability-oriented seal manufacturers have elected to incorporate certain features in their design.

Once these objectives have been reached, the user should feel motivated to solicit detailed proposals for mechanical seals from several capable manufacturers. The vendor should be required to submit dimensionally accurate layout drawings for the proposed product, and the user should engage in a feature-by-feature comparison of the competing offers. Keeping in mind the high cost of seal-induced pump failures, the user would then assign a monetary value to the probable failure avoidance and ask whether or not the added cost for a superior seal is justified. Experience shows that in the overwhelming majority of cases, a properly designed mechanical seal is worth many times its cost differential over the off-the-shelf "commodity" seal. The various reliability enhancement features will first be examined in detail. Thereafter, a good look will be taken at some recent application examples which incorporate one or more of these features.

## BASIC CONSIDERATIONS FOR USER AND MANUFACTURER

Since it is assumed that the reader has a good working knowledge of mechanical seals, concentration will be on relevant component considerations without first explaining how seals function. Here are some of these considerations:

*Design conditions* must be within the range of feasibility. To establish a seal design for optimum operation at the lowest long-term cost, the seal manufacturer must work within the following scope of requirements:

- high operating reliability
- appropriate interchangeability of components
- low leakage
- satisfactory life expectancy
- reasonable pricing

These requirements must stay within the present limitations for the application of mechanical face seals:

Shaft diameter	5 to 500 mm (0.2 to 20 inches)
Pressure	$10^{-5}$ to 250 bar (0.008 mm Hg to 3625 psig)
Temperature	-200°C to 450°C (-328°F to 842°F)
Face Speed	100 m/s and up (328 fps or 19620 fpm)

*Users please note: Don't expect seals to have high life expectancy outside of this application range.*

*Load conditions* should not fall outside the boundaries of vendors' experience. A convenient parameter for the load condition on a mechanical seal is the so-called p-v value, or pressure differential across one seal face multiplied by the face speed experienced by this sealing surface. It is the p-v value that determines if a mechanical seal design should consist of balanced or unbalanced seals, whether these seals will have a narrow or more traditional wide face configuration, or if the arrangement is stationary or rotating—just to name a few significant features linked to this parameter. In addition, the p-v value has a major influence on the selection of materials of construction, particularly seal face materials.

*Users please note: Stay within a given manufacturer's p-v experience. Verify how well a similar seal is doing elsewhere.*

*Materials of construction* utilized in mechanical seals usually must resist corrosion. Since austenitic and ferritic materials are characterized by low thermal distortion and high moduli of elasticity, they adapt quite well to this requirement [3]. However, more sophisticated alloys can be selected for services requiring improved corrosion resistance. The application of such alloys in mechanical seals requires additional design work, since physical properties can be significantly different as opposed to standard materials. Typical physical properties of construction materials are listed in Table 1.

As most seal users know, sealing difficulties more often occur on the secondary, elastomeric sealing elements generally found on pusher seals. Where O-rings serve as secondary sealing elements, many failure incidents can be traced to the application of O-ring materials, which are unsuitable for a given service. With O-ring manufacturers able to provide countless different compounds of O-ring materials, one or more will usually fit the requirements. Compromising by using the local distributor's standard selection may indeed be false economy.

Developments such as PTFE-wrapped elastomer materials combine necessary resilience and elasticity with good chemical and thermal resistance. PTFE wrapping may be the preferred solution in services with extensive temperature transients. The utilization of solid PTFE may introduce vulnerabilities, since the coefficient of thermal expansion of PTFE shows a rather extreme excursion at approximately 30°C (86°F). This leads to potential leakage at the surfaces contacted by the sealing element. The PTFE-wrapped O-ring material and the temperature-dependent expansion rates for PTFE are illustrated in Figures 1 and 2, respectively.

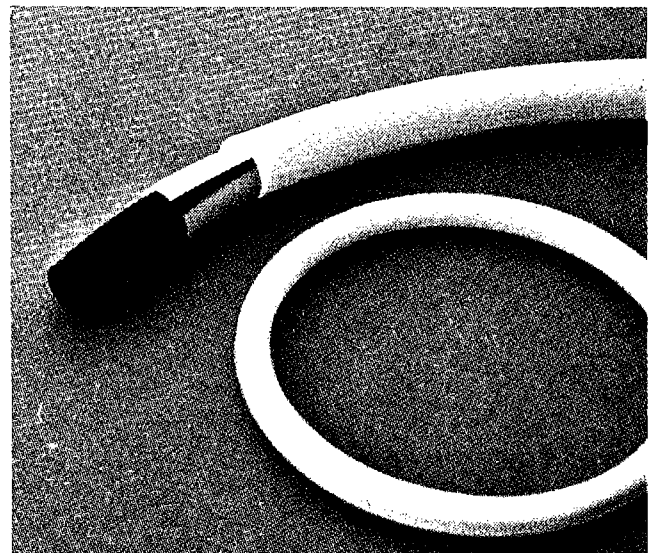


Figure 1. PTFE-Wrapped O-ring.

Table 1. Physical Properties of Seal Materials of Construction (Mean Values).

Material no.	Material code to DIN	AISI (similar)	Hardness (N/mm <sup>2</sup> )	0.2%-yield-strength (N/mm <sup>2</sup> )	Modulus of Elasticity (10 <sup>9</sup> N/mm <sup>2</sup> )	Coefficient of Expansion (10 <sup>-6</sup> /K)	Coeff. of Thermal Cond (W/mK)		
Chrome Steel	1.4122	X35CrMo17	440C	2.250-2.750	600	21.3	10.5	29.3	High strength, good properties for shrink fitting of SIC and hard metals. High thermal conductivity, limited use because of chemical resistance.
Cr-N-Mo-Steel	1.4460	XBCrNiMo27.5	329	1.900-2.300	500	21.0	11.5	14.6	Good properties for shrink fitting, usable for high resistance is required. In most cases sufficient resistance against seawater.
Cr-Ni-Mo-Steel	1.4571	X10CrNiMoTi810	316 Ti	1.300-1.900	230	20.3	16.5	14.6	Standard material for higher chemical resistance, e.g., for aggressive atmosphere, standard material for springs.
Hastelloy C	2.4602	NiMo16Cr	---	1.850	320	20	11.3	12.6	High corrosion resistant, also in use for springs for aggressive media.
Monel K500	2.4375	NiCu30Al	---	1.300-3.150	280-420	18.25	13.6	17.6	Seawater resistant also at higher salt contents and temperatures.
Carpenter 20 Cb-3	---	X6NiCrMoCuNb34 20	---	1.600-1.750	300	19.7	15.0	14.7	Highest corrosion resistance, e.g., employed with "formation water." 20-30% salt content
Inconel 625	2.4B56	NiCr22Mo9Nb	---	1.300-2.400	410-655	21.0	11.0	10.0	Highest safety against gap corrosion and extreme chemical attack.

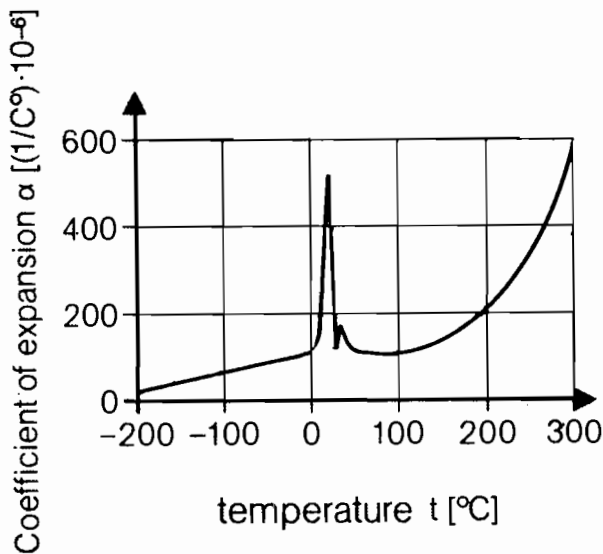


Figure 2. Coefficients of Thermal Expansion for PTFE at Various Temperatures.

O-rings in dynamic applications are expected to perform while undergoing movement in the axial and/or radial direction. They are typically installed with 7.0 to 8.5 percent radial compression. However, the compression set experienced by certain O-ring materials can have a significant negative impact on the performance of the O-rings.

Clearly then, O-ring selection influences the performance of a mechanical seal. This selection process should also be entrusted to an experienced seal manufacturer, since only he will know whether or not certain materials adapt to existing design features. It can be inferred that on difficult sealing services, proper material selection must precede finalizing the seal geometries. The seal must be engineered for the applica-

tion, as opposed to the application being forced to utilize a certain standardized seal geometry.

*Users please note: Ask the vendor for relevant application charts. Be aware of the widely differing properties seemingly similar elastomeric compounds may exhibit. Insist on the right one for vulnerable services.*

Temperature limits and Shore hardness values of frequently applied secondary seal elements are shown in Table 2.

Metal bellows can be used instead of O-rings in mechanical seals. Metal bellows, in fact, combine the function of a second-

Table 2. Temperature Limits and Hardness of Secondary Seal Components.

Temperature limits and hardness of secondary seal components			
Material	Temperature limit (°C)	Hardness (Shore)	
Nitril-Butadien-rubber (Buna N)	-55... +110	70..80	High resistance against swelling with oil, fuel and process gases, small permeability to gas, resistance to aging and fatigue
Fluorosilicone	-70... +230	70	Excellent resistance against hydrocarbons, oil. Preferred in use if resistance against hydrocarbons and low temperature is required at the same time.
Fluoro carbon rubber (Viton)	-25... +190	75	High resistance against mineral oil, salt water, aliphatic and aromatic hydrocarbons, water <250°F and atmospheric conditions (e.g. ozone-solar irradiation).
TTV, TTE, TTS (double-PTFE-coated)	-70... +250	-	Best chemical and thermal resistance against all aggressive liquids, elasticity ensured by use of Viton-, Silicone- or Ethylen-Propylen-core

ary sealing element with that of the necessary spring load. Traditionally, metal bellows seals were mainly used in high and low temperature services or where the limitations of O-ring materials would not allow their application in a mechanical seal. In recent years, metal bellows seals have also been widely used in service where O-rings might cause seal hang-up or shaft fretting. Aside from these advantageous features, metal bellows seals, nevertheless, have some significant limitations. Reliable long-term operation of bellows seals may be difficult to guarantee in high pressure, low specific gravity services, because the hydraulic diameter on a metal bellows often varies as the pressure differential changes. Slip-stick motion due to partial dry running of the seal faces in hydrocarbons with low specific gravity has caused bellows failure due to lateral offset, thus posing a potential safety hazard. Prior to installation, the resonant frequency of the metal bellows should be compared to the frequencies induced by the resonance speeds of the rotating equipment.

*Users please note: Verify the manufacturer's experience before accepting metal bellows seals, if offered for high pressure, low specific gravity services.*

Advanced face material combinations, such as carbon graphite running against silicon carbide, are mandatory for mechanical seals operating at elevated p-v values. Corrosion and wear resistance, dry-running behavior, and temperature resistance determine face material selection. Impregnations of carbon graphite (artificial carbon) have significant influence on the coefficient of thermal conductivity. Carbon graphite with antimony impregnation has a coefficient of thermal conductivity which is approximately 1.5 times that of resin impregnated carbon graphite. This could determine whether or not there will be partial dry running between two seal faces.

Hardness of the face material translates directly into wear resistance. Reaction bonded silicon carbide with approximately 12 percent free silicon provides high hardness, good thermal conductivity, resistance to heat checking, and excellent dry-running behavior.

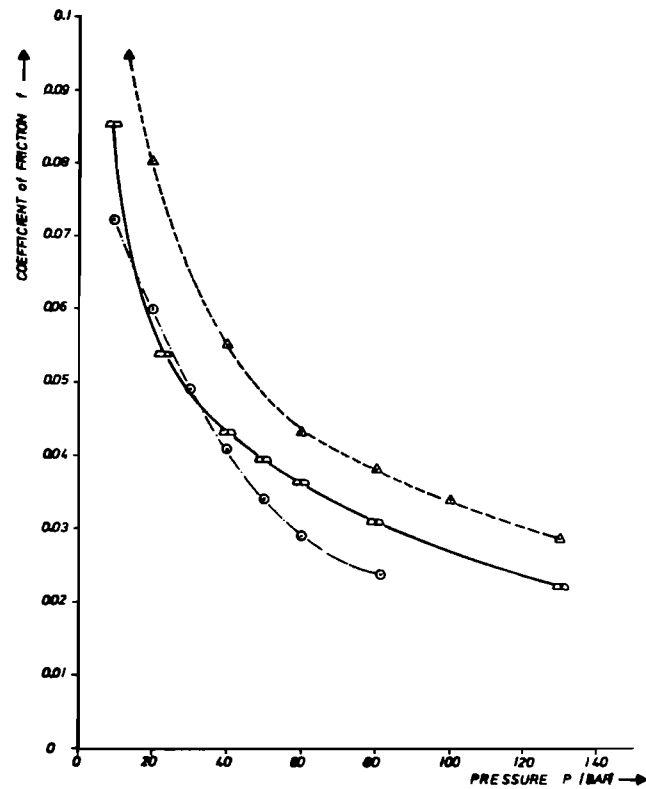
Direct sintered silicon carbide has improved chemical resistance even if exposed to liquids with a pH value above 11, but also possesses an increased coefficient of friction which will have a significant influence on the total power consumption of a seal system.

The latest face material development is an electrographite, impregnated with silicon. Both hardness and compressive strength of this face material are approximately three times higher than a comparable resin impregnated carbon graphite material. The thermal conductivity, usually the weakest link in a face/housing combination, is five times higher than in an antimony impregnated carbon graphite material. These unique physical values make the material a prime contender for application in high p-v and high temperature services. An additional benefit is derived from the low coefficient of friction of this material. The resulting power savings will often compensate for the added cost. This is illustrated in Figure 3, where the coefficient of friction of three face material combinations is plotted against a range of differential pressures. Finally, the characteristic properties of the various materials are listed in Table 3.

*Users please note: Ask if your sealing service would benefit from some of these features. If yes, choose from a source which can demonstrate proven experience and is willing to incorporate advantageous materials in your seals.*

## QUALITY CONTROL AND ACCEPTANCE TESTS

Detailed specifications for quality control and testing of seals are very complex and involve many parameters. Since it is evident that mechanical seal components are manufactured to



Conditions: Mean face dia: 59.5 mm (2.34 in.)  
 Face width: 2.7 mm (.106 in.)  
 Rotational speed: 3250 rpm  
 Face speed: 10 m/s (32.8 fps)  
 Balance ratio: 0.79  
 Spring load: 30.2 N/cm<sup>2</sup> (43.8 psi)  
 Service: desalinated water

Coefficient of Friction of Various Face Material Combinations at Certain Pressure Differentials

- Δ Tungsten carbide/resin impregnated carbon
- ◻ Reaction bonded silicon carbide/resin impregnated carbon
- Silicon carbide impregnated carbon/resin impregnated carbon

Figure 3. Coefficient of Friction of Various Face Material Combinations at Certain Differential Pressures.

very close tolerances, an extended quality assurance program that controls in-house manufacturing processes at strategically integrated locations will go far towards ensuring that quality is maintained at the established standards. Statistical and numerical quality control results should be compared against specifications given to the sub-supplier.

Generally, any mechanical seal design should be tested prior to release for an industrial application. Ideally, the seal vendor should be able to provide the seal user with data derived from operation of the seal on a test stand. This testing should have been done under conditions similar to those at the final installation site. Where this is not possible, seal face behavior under operating conditions should be simulated and studied by finite element analysis. Thus, it can be determined whether thermally and pressure-induced deflections will remain within acceptable limits to ensure long-term trouble-free seal operation.

Where acceptance tests are called for, it would be prudent for the seal purchaser and the seal vendor to pre-define expectations or allowable deviations from norms in most or all of the following areas:

- leakage behavior of the mechanical seal during the hydraulic test

Table 3. Physical Properties of Mechanical Seal Face Materials.

	Compressive strength (N/mm <sup>2</sup> )	Density (g/cm <sup>3</sup> )	Modulus of Elasticity (10 <sup>4</sup> N/mm <sup>2</sup> )	Coefficient of Expansion (10 <sup>-6</sup> /K)	Coefficient of Thermal Conductivity (W/mK)	Hardness (N/mm <sup>2</sup> )	Heat stress crack resistance factor (W/m)	Special suitability
I. Artificial carbons Hard carbon, antimony impregnated	310	2.5	1.8	4.5	14	1.280	5.500	Good emergency running, properties, high temperature resistance and compressive strength, therefore especially suitable for high pressure mechanical seals
II. Metals Chromium casting	1.200	7.5	20.5	9.5	20	3.200	3.900	Operational limits, p<230 PSI (16 bar) v <sub>g</sub> <50 ft/sec., e.g., for secondary mechanical seals in tandem arrangement
III. Metal carbides Tungsten carbide	5.000	15.0	60.0	5.2	79.82	15.000	12.650	High hardness and wear resistance Harder than tungsten carbide, excellent chemical resistance, good emergency running properties and thermal conductivity
Silicon carbide	3.500	3.1	41.0	4.3	84	25.000	8.000	
IV. Silicon impregnated artificial carbons SiC-C, carbon silicon impregnated	800	2.65	14	4.5	70	3.100	12.000	Better hardness and wear resistance than carbon used in high-temperature and high-pressure mechanical seals

- leakage behavior of the mechanical seal during the hydrodynamic test
- surface roughness and flatness of faces before and after test
- wear measurement of seal face after test
- power consumption of seal arrangement at various pressures
- temperatures, absolute, inlet versus outlet
- cooling water flow
- cooling water temperature, inlet versus outlet
- buffer fluid flow
- pumping device performance
- start/stop operation to evaluate torque transmission devices
- plotting radial and axial vibrations

Users please note: When in doubt, specify testing, negotiate cost, and monitor test results.

## DESIGNING FOR DECREASED FAILURE RISK

A few examples of difficult, or failure-prone sealing applications will be examined. The intent is to see how and where some of the reliability improvement measures described have recently been applied.

Based on the frequency of seal related failures, the cost of seal-induced pump repairs in 1983 averaged over \$5000 for a centrifugal pump in U.S. Gulf Coast petrochemical plants [4]. If, therefore, a better mechanical seal costs \$1000 more than the next choice and avoids even one repair, the additional investment is recovered in the first few months of operation.

Thus, there are compelling reasons to analyze the design features of the various seals in an effort to identify the vulnerable executions and to give preference to seal designs which avoid potential problems. The user must perform this analysis

by comparing and evaluating the responses of several vendors. The review process should never be a mere comparison of initial costs; instead, the bids must be conditioned with a view towards the probable failure frequency, maintenance cost, or overall reliability of the machine train in which the competing seals would serve over an anticipated time period. This was done for many of the seals which are described next.

### Mechanical Seal for Boiler Feed Pump (Figure 4)

#### Operating Conditions:

Pump main feed pump at fossil fired power plant  
 Shaft size 200 mm (7.874 in) nominal seal size  
 Shaft speed 6700 rpm  
 Face speed 70 m/s (229 fps)  
 Service boiler feedwater  
 Pressure p<sub>1</sub> 70 bar (1015 psig)

Modern boiler feed pump seals are typically single, stationary type, cartridge mechanical seals (Figure 4). A portion of the feedwater passes from the pump cavity through an independently cooled throat bushing with a diametral clearance of 0.5 mm (0.02 in) and a length approximately equal to the shaft diameter. This water is circulated from the seal cavity through magnetic filters and a heat exchanger, utilizing a pumping screw system (Items 6 and 8 on Figure 4) back to the seal cavity.

Effective cooling of the entire arrangement and seal faces assures that the temperature between the seal face is kept below the boiling point of the feedwater at atmospheric pressure. It has been observed on a mechanical seal of similar design, operated under comparable conditions, that insufficient face cooling created a vapor ring around the seal faces. This caused dry-running conditions and severe damage to the seal. This is why a flow guide (Item 12 on Figure 4) which directs cool, clean water toward the seal faces should be incorporated in feedwater seals.

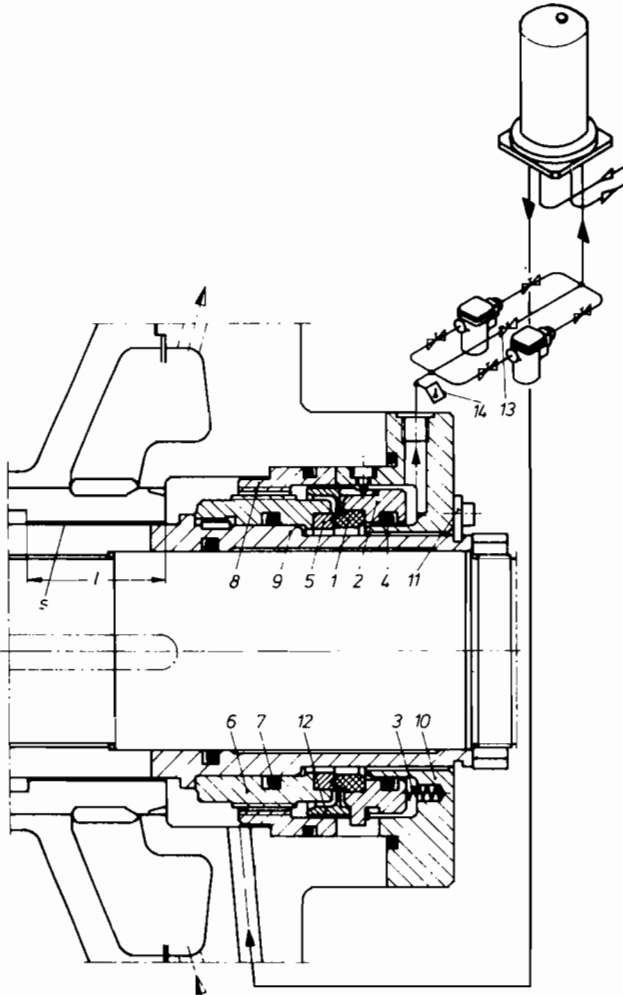


Figure 4. Mechanical Seal for Boiler Feedwater Service. 1) Carbon ring, 2) Face housing, 3) Compression spring, 4) O-ring, 5) SiC ring, 6) Seat housing with pumping screw, 7) O-ring, 8) Stationary pumping screw, 9) Shaft sleeve, 10) Housing flange, 11) Assembly fixture, 12) Fluid flow guide, 13) Bypass, 14) Double temperature probe.

Generously dimensioned seal components, properly sized heat exchangers, and small throat bushing clearances provide outlet temperatures for these seals below 60°C (140°F). This is a well proven system which is in service in numerous installations.

A significant improvement in the reliability of boiler feed pump seals can be accomplished with the utilization of silicon carbide faces. Certain feedwater conditioning additives will often initiate a selective corrosion/erosion process on the binder component of previously used tungsten carbide faces. This process can lead to excessive face wear and premature seal failure. Since reaction-bonded silicon carbide is chemically inert against any conventionally used feedwater conditioners, it is a more appropriate face material for these critically important services.

Antimony impregnated carbon is selected for hot water applications because of its improved thermal conductivity and acceptable wear behavior. Since partial vaporization of feedwater in the seal gap cannot be totally eliminated, a face material with acceptable dry-running properties is mandatory.

Ethylene-propylene O-rings are installed in boiler feed service because of their advantageous performance in hot water. Although Viton does have somewhat similar temperature

constraints, it is limited to a temperature environment of only about 90°C (194°F) in hot water services.

Considerable thought must be given to adequate cooling of the seal system when the feed pump is in the stand-by mode. Slow-roll operation substantially reduces the efficiency of the pumping screw. Therefore, the system must rely on thermosyphon circulation for cooling, which will be activated by opening the bypass valve (Item 13 on Figure 4) prior to stand-by operation of the pump. The capacity of the cooling circuit at stand-by conditions is designed to limit seal face deflection caused by temperature increases to an acceptable value. Thus, excessive leakage at start-up of the feed pump will be prevented.

A performance diagram of two mechanical seals similar to the one described is shown in Figure 5. The power consumption of the seals is plotted against various pressures, increased in steps up to the maximum design pressure. The measured values shown on this diagram should closely match initial calculations to prove the feasibility of this design.

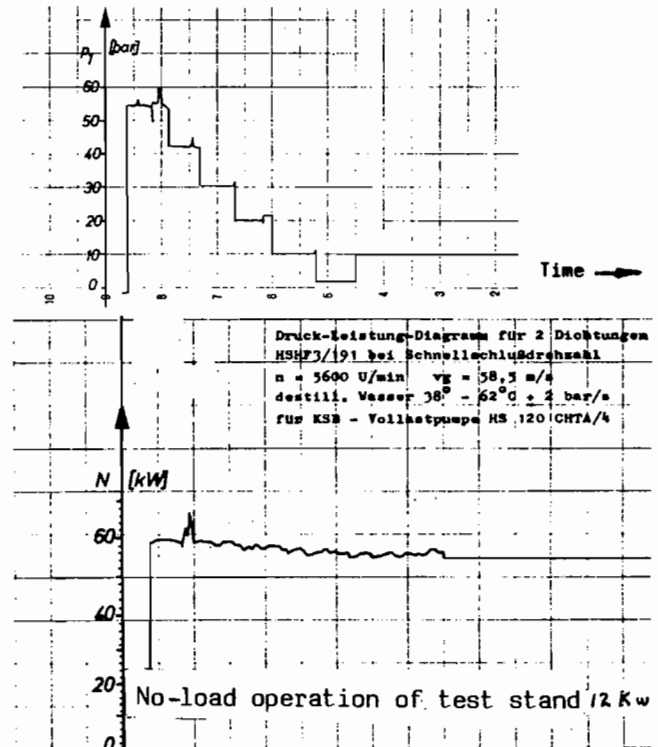


Figure 5. Performance Diagram for Two Similar Mechanical Seals in a Feed Pump.

#### Mechanical Seal for Horizontal Pipeline Pump (Figure 6)

##### Operating Conditions:

Pump	single stage, double volute pipeline pump
Shaft size	220 mm (8.661 in) (nominal seal size)
Shaft speed	1150-1740 rpm
Face Speed	20 m/s (65.5 fps)
Pressure $p_1$	100 bar (1450 psig) maximum
P-V value	2000 bar m/s (95000 psi-ft/sec)
Temperature $T_1$	15°-52°C (59°-126°F) operating, 80°C (176°F) maximum
Service	potable water
Specific gravity	0.999-0.987 kg/dm <sup>3</sup>

The mechanical seal shown in Figure 6 is operated with a seal flush arrangement per API 610, Plan 31, i.e., the seal flush

line is connected to the pump discharge. A cyclone separator is installed in this line (Item 11 on Figure 6).

For safety reasons, the seal is equipped with an external throttle bushing (disaster bushing) to control excessive seal leakage. Leakage is collected in a trap (Item 10 on Figure 6) and flows past a leakage detector (Item 12 on Figure 6) into a common return line. The leakage detector will trigger an alarm if the amount of leakage exceeds a preset value.

The seal is a single acting, stationary type, high pressure seal. To prevent seal face distortion due to the high pressure differential, a stainless steel ring is shrunk onto the stationary carbon to support the face. Hydrodynamic grooves are cut into the rotating seal face to introduce hydrodynamic lubrication between the seal faces. The hydrodynamic forces will essentially counterbalance the hydraulic face load.

Face materials are antimony impregnated carbon against reaction bonded silicon carbide. Secondary sealing elements are Buna-N O-rings with a Shore hardness increased to a value of 80. In addition, a support ring (Item 4 on Figure 6) is installed in front of the dynamic O-ring to counteract extrusion.

Since these mechanical seals are operated in a remote desert region, special consideration was given to reliability and simplicity of the design. Only well proven components are utilized. Rapid seal replacement is ensured by using a cartridge configuration.

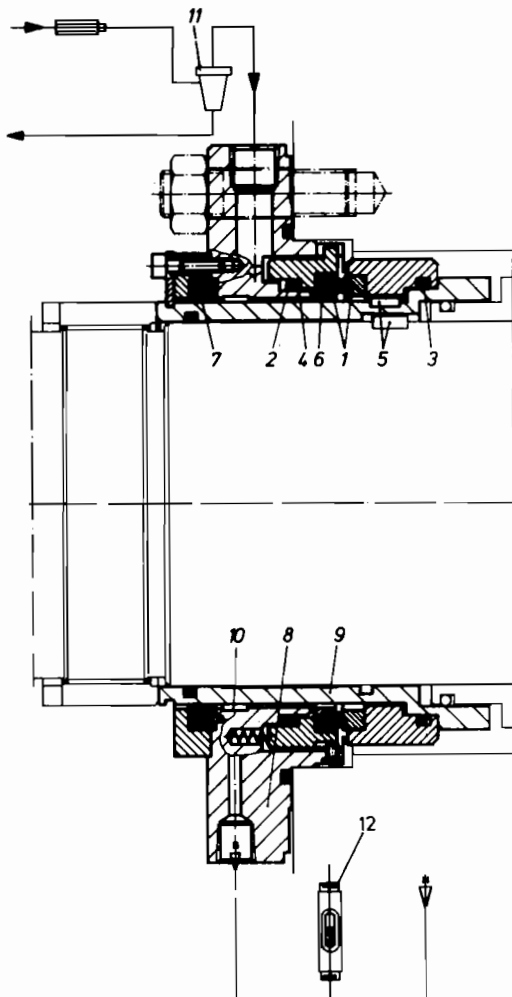


Figure 6. Mechanical Seal for Potable Water Pipeline Pump. 1) Seal faces, 2) O-ring, 3) O-ring, 4) Anti-extrusion ring, 5) Key, 6) Support ring, 7) Floating ring, 8) Gland plate, 9) Sleeve, 10) Relief groove, 11) Cyclone separator, 12) Flow meter.

Some interesting performance curves for the seal are shown in Figure 7. While the power consumption increased with the pressure  $p_1$ , the coefficient of friction underwent a proportionate decrease. The two diagrams indicate the influence of the hydrodynamic lubrication, which improves with increasing pressure. As expected, the maximum rate of seal leakage coincides with the highest pressure level [5].

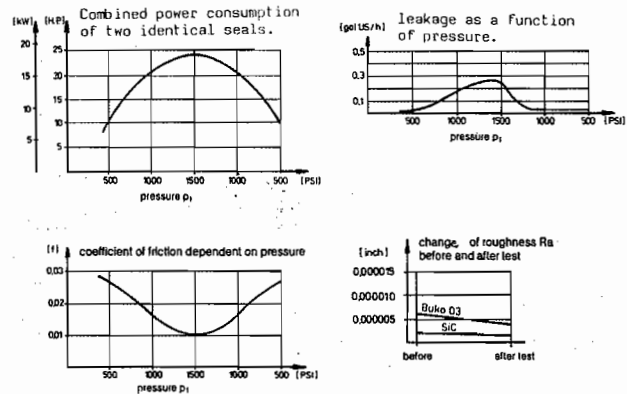


Figure 7. Test Parameters and Results for Mechanical Seals.

Mechanical Seal for Horizontal Anhydrous Ammonia Booster Pump (Figure 8)

Operating Conditions:

Pump	12 stage booster pump, barrel type
Shaft size	102 mm (4.015 in) (nominal size)
Shaft speed	7200 rpm
Face speed	42 m/s (138 fps)
Pressure $p_1$	42 bar (609) psi
P-V value	1764 bar m/s (83860 psi-ft/sec)
Temperature $T_1$	62°C
Service	anhydrous ammonia
Specific gravity	0.55-0.66 kg/dm <sup>3</sup>
Barrier fluid	water
Pressure $p_3$	26 bar (377 psig)
Temperature $T_3$	30°C (86°F)

The centrifugal pump utilized in this application embodies unique design features and was manufactured specifically for a large U.S. Gulf Coast chemical plant. Since this pump represents the most important machine in a rather sensitive process, mechanical seal reliability was given primary attention.

The two-stage tandem mechanical seal operating in anhydrous ammonia on the primary side and water on the secondary side is depicted in Figure 8. A tandem arrangement was selected for the various reasons listed below:

- Leakage into the product is not acceptable.
- Anhydrous ammonia is extremely volatile and the vapors are poisonous. A liquid barrier is required to prevent access of the fluid to the atmosphere.
- Pressure  $p_1$  can be broken down into two steps, reducing the p-v value for the inboard seal to an acceptable amount. Thus, the hydraulic load for the inboard seal is reduced, which is an advantage when considering the poor lubricity of anhydrous ammonia.
- Low specific gravity liquids tend to evaporate in the seal gap when reduced to atmospheric pressure. By pressurizing and cooling the secondary side of the seal, sufficient margin is provided away from the critical evaporation point.

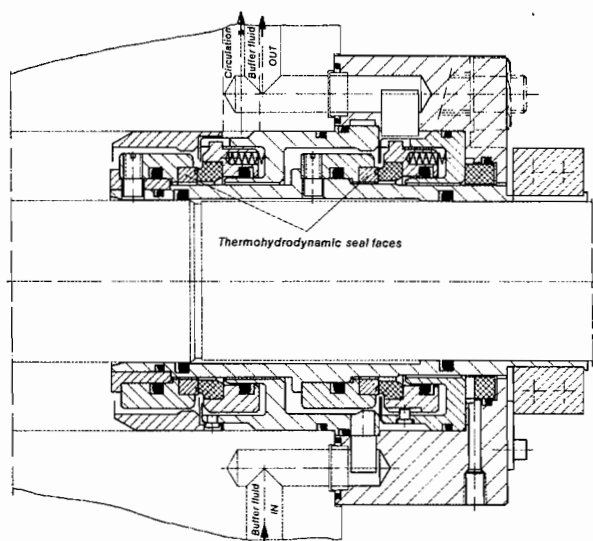


Figure 8. Mechanical Seal for Anhydrous Ammonia Pump.

Both inboard and outboard (primary and secondary) seal wear parts were made dimensionally identical. This would prevent potential assembly errors. Direct-sintered silicon carbide faces were utilized because of their resistance to leaching out in this highly alkaline environment.

The seal was mounted on a straight shaft. Unbalanced hydraulic forces acting on the seal would cause a significant axial thrust load. These forces had to be transmitted by the outboard clamping device, which was originally designed as two half shells connected by radial clamping bolts. This design repeatedly malfunctioned, due to insufficient transmission of forces, which caused an axial offset of the entire seal arrangement towards the outboard side. Consequently, the seal faces were overloaded and the seal was destroyed within a short period of time.

A new design now utilizes a shrink disc, in place of the clamping rings. Because of their design and construction, shrink discs allow the transmission of both axial and radial forces. Values for the maximum allowable forces are provided by the shrink disc manufacturer and enable the seal designer to select an adequately sized element for the purpose.

Quick interchangeability of the seal cartridges is an important criterion in this application, since the plant design tolerates only a few hours of pump downtime.

#### Mechanical Seal for Horizontal Liquid Ethylene Pump (Figure 9)

##### Operating Conditions:

Pump	final product transfer to cavern
Shaft size	110 mm (4.33 in) (nominal seal size)
Shaft speed	3600 rpm
Face speed	23 m/s (75.4 fps)
Pressure $p_1$	26 bar (377 psig)
P-V value	598 bar m/s (28425 psi-ft/sec)
Service	liquid ethylene
Specific gravity	0.4 kg/dm <sup>3</sup>
Temperature	-22°C (-6°F)
Barrier fluid	ethylene glycol/water
Pressure $p_3$	ambient

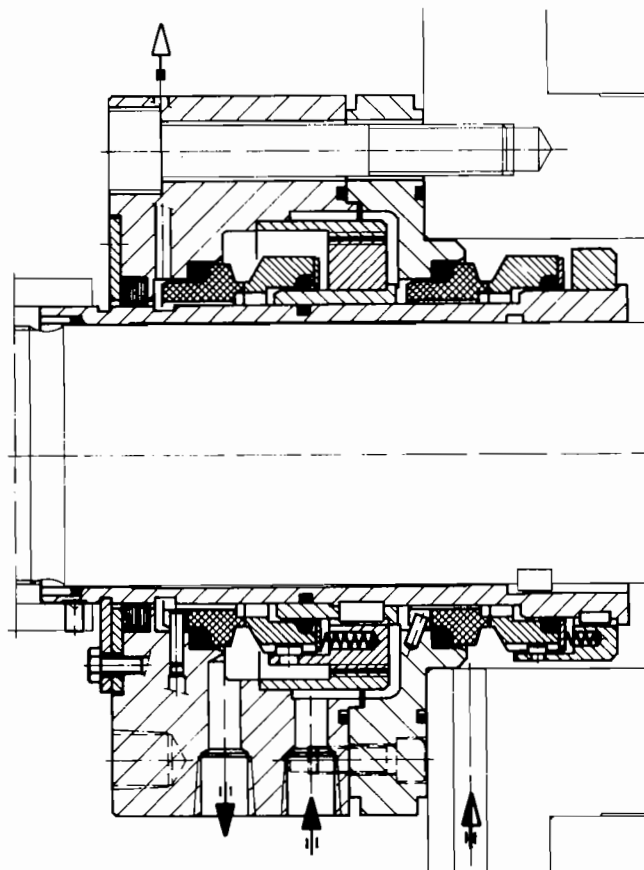


Figure 9. Mechanical Seal for Liquid Ethylene Service.

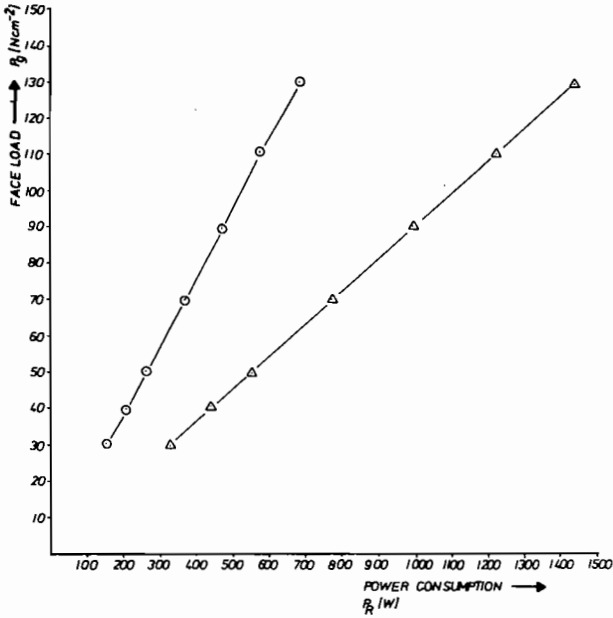
A cross-sectional view of a tandem mechanical seal arrangement operating in ethylene service is shown in Figure 9. Propane and ethylene are categorized as light-end hydrocarbons with very little temperature margin to the fluid boiling point. Narrow face configurations as depicted in Figure 9 provide reduced seal face environmental temperatures since they generate considerably less heat than standard wide face seals. This lower heat generation leaves sufficient temperature margin to the boiling point of ethylene at atmospheric pressure. Vaporization of the sealing fluid between the seal faces, which would lead to dry-running conditions and excessive face wear, is thus prevented [1]. The power consumption for two demonstration seals operating at identical conditions and with geometries identical, except for face width, is graphically presented in Figure 10.

At the given nominal diameter of the seal, it becomes obvious that the narrow faces weaken the rotating face considerably from a standpoint of strength. A finite element analysis, provided for both thermal and mechanical distortion forces, resulted in the seal face geometry shown, which will provide optimum strength at given load conditions. The desired seal face flatness is obtained in this manner.

The outboard seal has seal faces identical to the inboard seal. It is equipped with a pumping screw that circulates the barrier fluid through a pressure vessel which is vented to the flare. This improved auxiliary quench circuit rapidly transfers potential ethylene leakage away from the inboard seal and into the buffer fluid contained in the vessel [6].

Face materials selected for this service are again antimony impregnated carbon versus reaction bonded silicon carbide. This combination runs with a low coefficient of friction, which aids in keeping heat generation to a minimum.





Power Consumption Narrow Face (○) versus Wide Face (△) (Simplified)

Narrow Face:		Wide Face:	
Mean face dia:	76.7 mm	Mean face dia:	79 mm
Face width:	3 mm	Face width:	6 mm
Rotational speed:	3600 rpm	Rotational speed:	3600 rpm
Face speed:	14.45 m/s	Face speed:	14.88 m/s
Balance ratio:	1.0	Balance ratio:	1.0
Coeff. of friction:	0.05 (constant)	Coeff. of friction:	0.05 (constant)
Spring load:	30 N/cm <sup>2</sup>	Spring load:	30 N/cm <sup>2</sup>

Figure 10. Power Consumption of Narrow and Wide Face Seals.

Special consideration was given to the O-rings for this mechanical seal. Buna-N compound, specifically designed for low temperature applications, proved to be of sufficient quality.

The entire seal system relies on the fact that only minor deviations from the calculated seal face load will be allowed. This prompted the utilization of a cartridge seal, which allows proper calibration at the point of manufacture.

**Mechanical Seal for Vertical Acetylene Compressor (Figure 11)**

**Operating Conditions:**

Compressor	single stage, low flow, high-speed compressor
Shaft size	34 mm (1.338 in) (nominal seal size)
Shaft speed	28427 rpm
Face speed	64.45 m/s (211.3 fps)
Pressure $p_1$	0.83 bar (12 psig)
Pressure $p_3$	5 bar (72.5 psig)
P-V value	322.25 bar m/s (15320 psi-ft/sec)
Service	acetylene gas
Temperature $T_1$	110°C 230°F
Barrier fluid	hydraulic oil at 32 cSt

This design features a double face-to-face seal as the main seal, and an additional single seal on the atmospheric side that prevents the gear box oil from entering the product cavity. The design is the result of a mutual development endeavor between user, original equipment manufacturer, and seal manufacturer with the intent of increasing the mean-time-between failure (MTBF) of this high-speed, low flow compressor. Historical failure records indicate that the equipment suffered from frequent outages which were most often attributed to seal failures.

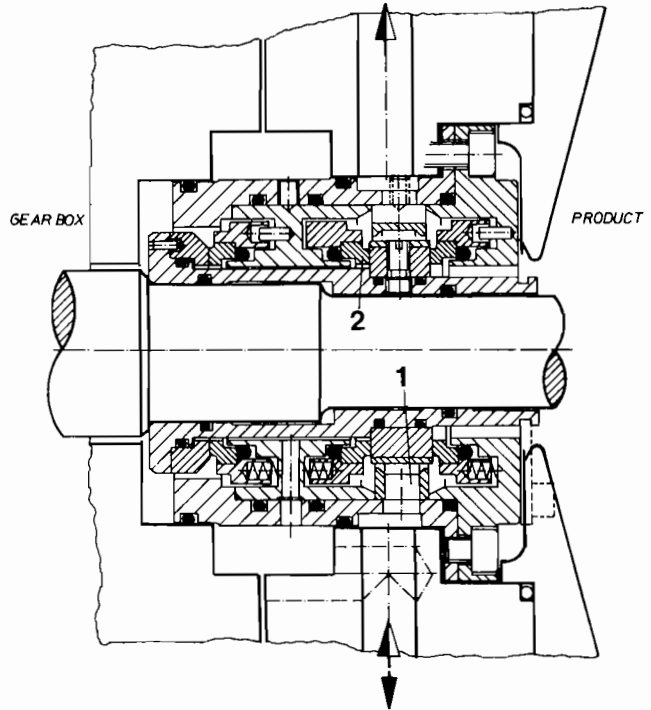


Figure 11. Mechanical Seal for Acetylene Compressor. 1) Circumferential flow guide and 2) Stationary seal faces.

The initially installed OEM seal consisted of a number of parts which had to be installed according to complicated instructions, if proper functioning of the seal was to be assured. The equipment owner felt this would be difficult to achieve in the field and made ease of installation a specific requirement. This resulted in a design that contains all required components in a single cartridge assembly, which eliminates field errors and confusion about seal parts stack-up.

The relatively high face speed on this mechanical seal was thought to lead to potentially troublesome heat generation. To counteract this possibility, the manufacturer introduced a circumferential flow guide (Item 1 on Figure 11) as an added feature for the buffer fluid cavity. Turbulent flow of hydraulic oil is directed towards the seal faces for optimum cooling.

The stationary seal faces (Item 2 on Figure 11) all have the same geometry to assure interchangeability. Face material is silicon impregnated electrographite with a compressive strength of 600 N/mm<sup>2</sup> (87,000 psi). This material was selected to compensate for the low cross-sectional profile of the face inserts. (Refer to Table 3 for comparison with other face materials).

In addition to the mechanical seal, a hydraulic buffer fluid system was designed and supplied by the seal manufacturer. System capacity, pressure, and back-up features were arranged to match the requirements of the mechanical seal (Figure 12). Fluid flow and instrumentation details are shown in Figure 13. Items 6 through 8 in Figure 12 represent components which serve as pressure retention elements. Should there be a power interruption or pump failure in the buffer fluid system, the accumulator (Item 7 on Figure 12) will lock the buffer fluid pressure at the desired level between the check valves (Items 3 and 8 on Figure 12).

Many elements of the seal system instrumentation are interconnected with the control room annunciators. A low switch point on the Pressure Indicating Alarm (PIA) triggers an alarm. A low-low switch point on the same PIA shuts down the compressor, but also acts as a permissive start switch.

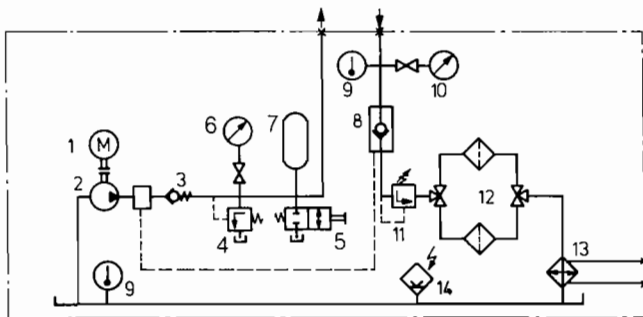


Figure 12. Flow Schematic Diagram for Buffer Fluid System. 1) Three-phase a-c motor, 2) Gear pump, 3) Check valve, 4) Pressure relief valve, 5) Discharge valve, 6) Pressure gauge (vessel pressure), 7) Pressure vessel, 8) Check valve (lockable), 9) Thermometer, 10) Pressure gauge (buffer fluid pressure), 11) Pressure valve (buffer fluid pressure), 12) Double filter, 13) Heat exchanger, 14) Level switch.

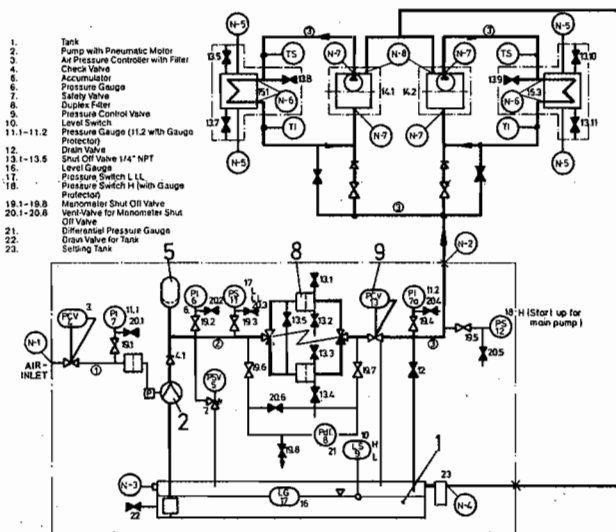


Figure 13. Flow Schematic Diagram for Simplified Buffer Fluid System.

Inlet and outlet temperatures of both buffer fluid and cooling water are controlled to register abnormal seal behavior which might precede serious seal failure.

The reliability of the compressor has been substantially improved since the installation of the seal system and considerable life expectancy increase is finally within reach.

#### Mechanical Seal for Vertical Transfer Pump (Figure 14)

##### Operating Conditions:

Pump	long shafted, single stage centrifugal pump
Shaft size	50.8 mm (2.00 in) (nominal seal size)
Shaft speed	2300 rpm
Face speed	8.9 m/s (29.2 fps)
Pressure $p_1$	4.2 bar (60.9 psig)
Pressure $p_3$	0.7 bar (10.15 psig)
P-V value	37.78 bar m/s (17777 psi-ft/sec)
Service	radioactive sludge, non-Newtonian
Temperature	100°C (212°F)

This mechanical seal was to be immersed in a tank containing non-Newtonian, radioactive and abrasive sludge. Its duty is to seal clean water inside the pump column on the inside

diameter of the seal against the sludge on the seal outside diameter. A leakage rate of no more than 20 ml/min (0.67 fl oz/min) at a given maximum water pressure of 2.75 bar (40 psig) on top of a 13.7 m (45 ft) column was specified. Radioactive material had to be positively prevented from entering the pump column.

Elastomers were not permitted in the design. Materials of construction specified were either Hastelloy-C or 316 SS. Secondary sealing elements had to be pure graphite gaskets. All these requirements were based on the properties of the radioactive sludge, which contains strontium-90 and cesium-137.

A proven wear-resistant face material, tungsten carbide with nickel binder, was selected for the application, because potential corrosion due to pumpage with a pH value of 13 had to be overcome. A Hastelloy-C bellows would serve as both the secondary sealing element and the spring component. It had to be designed to create a sufficient hydraulic force load when pressurized from either the ID or OD (Item 1 on Figure 14). The metal bellows is also designed to take a maximum axial shaft movement of 6.35 mm (0.25 in) compression and 1.02 mm (0.04 in) extension. This relatively wide axial float is due to axial movement in both operation and installation of the approximately 14 m (46 ft) long shaft.

An additional pumping screw arrangement (Item 2 on Figure 14) was incorporated in the mechanical seal to circulate water from the column through the seal cavity and a bushing that is installed above the seal to prevent dead-end operation.

One requirement of the specification was to design the seal for 2500 hour life expectancy. Within the limited scope of feasible materials, and due to the stringent dynamic conditions, the design had to be of superior quality. All materials had to be

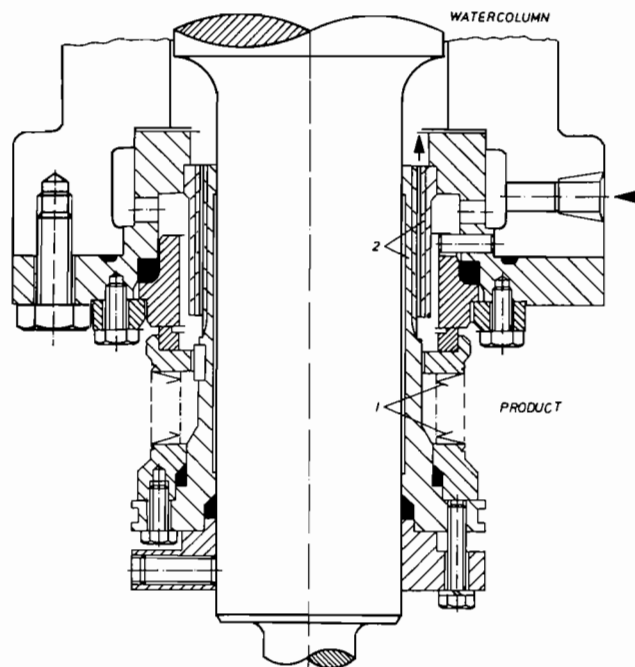


Figure 14. Mechanical Seal for Vertical Transfer Pump. 1) Inside/outside diameter and 2) Screw pump arrangement.

checked against explicit specifications, and certificates of compliance were issued for each individual part. In addition, each complete seal was run for an extended time period on a test rig at the seal manufacturer's laboratories. After successful completion of this test, a report was issued, certifying acceptable performance for the seal tested.

Certificates of compliance, a comprehensive test report and the manufacturer's willingness to let the user observe any test, assure the user that all manufacturing steps are executed in accordance with the specification. An experienced seal vendor will be able to supply such information without imposing any restrictions.

#### Mechanical Seal for Produced Water Reinjection (Figure 15)

##### Operating Conditions:

Pump	barrel type injection pump
Shaft size	106 mm (4.173 in) (nominal seal size)
Shaft speed	6000 rpm
Face speed	40 m/s (131.1 fps)
Pressure $p_1$	6.5 bar (94.3 psig)
Pressure $p_3$	10 bar (200 bar) [145 psig (2900 psig)]
P-V value	400 bar m/s (19015 psi-ft/sec)
Service	produced water (separated from crude oil)
Specific gravity	0.97 kg/dm <sup>3</sup>
Temperature $T_1$	71°-82° (160°-180°F)
Buffer fluid	ethylene glycol

A double acting seal with face-to-face configuration is illustrated in Figure 15. High content of abrasives and salt in the produced water, plus relatively high levels of H<sub>2</sub>S and CO<sub>2</sub> presented potential scale formation at the given temperatures. Thus, a double-acting seal was required for reliable service.

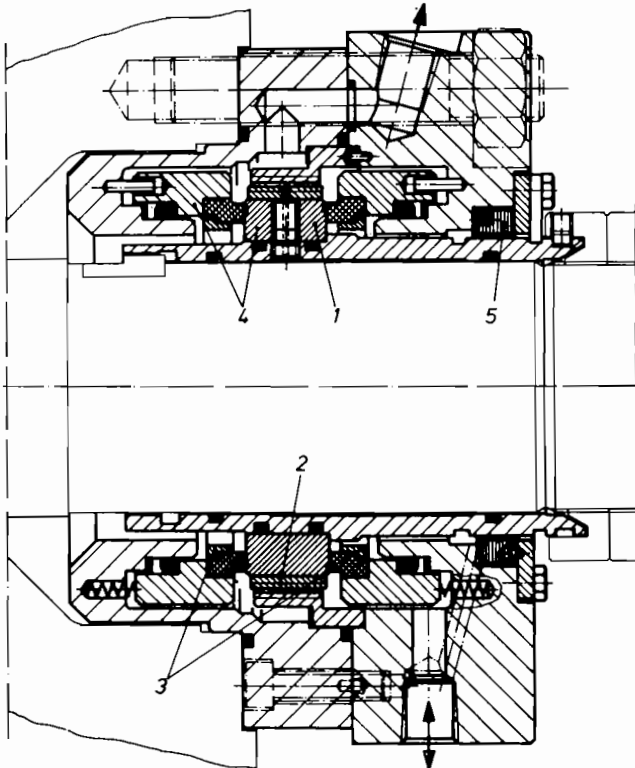


Figure 15. Mechanical Seal for Produced Water Injection Service. 1) Rotating seal faces, 2) Pumping screw, 3) Carbon faces, stationary, 4) Inboard seal faces, 5) Floating ring.

A face-to-face arrangement was selected, due to the high face speed. Reduced mass of the dynamically balanced rotating seal face led to controlled radial runout [7]. The seal face is manufactured from solid reaction bonded silicon carbide and has an optimized cross-section. The stainless steel radial support ring that is shrunk to the OD of the seal face is designed to

perform as a pumping screw as well (Item 2 on Figure 15). The pumping device creates enough head and flow to circulate buffer fluid from the seal cavity through a heat exchanger and back to the seal cavity. Both the seal and the cooling circuits are designed for a maximum static pressure of 200 bar (2900 psig), which might be approached under abnormal pump operating conditions. As described previously, stainless steel support rings prevent the carbon faces from distorting (Item 3 on Figure 15). The hydraulic pressure balance is so designed that, if buffer fluid pressure is lost, the seal faces will not separate. This feature protects not only the pump against unexpected leakage, but greatly reduces the risk of premature seal failure.

The material of construction for the seal is 1.4460 (AISI 329), which provides satisfactory corrosion resistance against seawater. Viton O-rings comprise the secondary sealing elements. Viton performs well in both seawater and ethylene glycol.

The axial pumping screw substantially simplifies the buffer fluid system in this application. Recall that buffer systems must perform four functions: circulation, pressurization, cooling, and leakage make-up. Referring back to Figure 13, circulation and cooling are accomplished by a pumping screw and a heat exchanger in a separate loop for each seal. An air-driven piston pump (Item 2 on Figure 13) provides pressurization and leakage make-up at a pressure of approximately 150 bar (2175 psig). An accumulator (Item 5 on Figure 13) and duplex filter (Item 8 on Figure 13) are installed downstream of the pump. The pressure control valve (Item 9 on Figure 13) reduces the piston pump discharge pressure to the seal buffer fluid loop pressure of 10 bar (145 psig). The piston pump is activated only when compensation for leakage loss is required in one of the cooling loops.

## CONCLUSION

Selecting and utilizing better mechanical seals is without doubt the most cost-effective method of extending the mean-time-between-failure of centrifugal pumps in virtually all industries where pumps are used. A properly engineered seal will distinguish itself from the average product by such elements as material selection, component configuration, seal auxiliaries, etc. While it could be claimed that for a given service a certain enhancement feature results in only marginal improvement, the combined influence of these elements has been demonstrated to increase equipment reliability and to decrease maintenance expenditures in the long run by very substantial margins.

Very few machinery component upgrade efforts will show higher, or more consistent, returns on investment than thoroughly engineered mechanical seals. It is hoped that users have an increased awareness of how more reliable mechanical seals can add to plant profitability, and how to spot the features which can make some seals more dependable than others.

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