MECHANICAL SEALS FOR HIGH FACE SPEED APPLICATIONS

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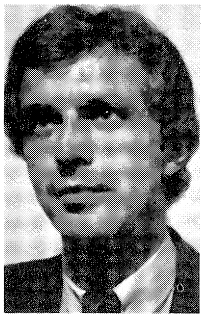


Wolfgang Schuebl received his Diploma in Mechanical Engineering from the University of Essen in 1975. The same year, he joined Burgmann Dichtungswerke in Germany as a Design Engineer. He operated the Munich Branch Office for Burgmann Dichtungswerke from 1976 to 1981.

In late 1981, he moved to the United States as Vice President and General Manager of Burgmann Seals America, Inc.

ABSTRACT

The rotating equipment industry's ever-increasing demand for efficient, safe and reliable seal elements has resulted in the application of mechanical seals in turbomachinery at very high face speeds. In addition to the commonly known installation of mechanical seals in fluid handling systems, such as heavy duty centrifugal pumps, applications have been extended in recent years to centrifugal compressors and gas turbines.

Mechanical seals in seawater and NGL injection pumps in oilfield applications are operating currently with face speeds up to 50 meters per second (m/s). In some cases, mechanical seals in centrifugal compressors used in the petrochemical industry have exceeded face speeds of 100 m/s.

The desirable features unique to mechanical seals—controlled axial forces with low leakage rates and acceptable power consumption—lead to state of the art development in order to meet the needs of efficient, reliable turbomachinery. It is the purpose of this paper to introduce the basic design considerations for high speed mechanical seals in this particular industry.

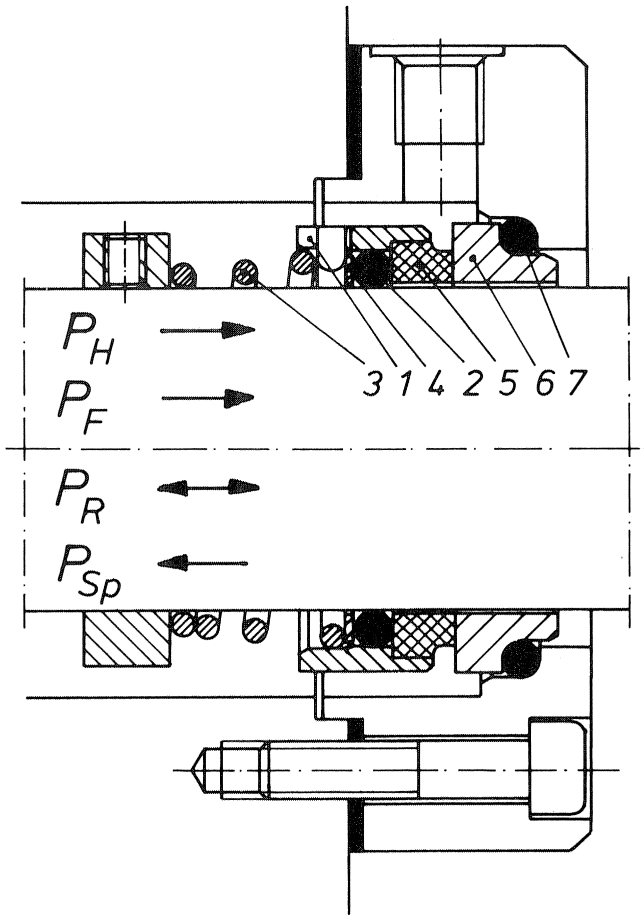
INTRODUCTION AND DEFINITION

In order to achieve a common understanding in this paper, it should be stated that an axial mechanical seal, or end face seal, is defined as a device which has forces acting in an axial direction on a set of seal faces in such a way that these forces control radial fluid flow between the faces. The fluid flow between the rotating and stationary faces is commonly known as "leakage" [1] (Figure 1). The term most often used for the defined arrangement is simply "mechanical seal." Consequently, this term will be used herein.

According to Mayer [1], a mechanical seal can be considered a high speed seal once a face speed of 20 m/s has been exceeded. It is the aim of this paper to introduce design features for mechanical seals operating well above this level.

BASIC MECHANICAL SEAL ARRANGEMENTS IN TURBOMACHINERY

In addition to labyrinth type seals and floating bushings, the mechanical seal represents another main group of seals in



1 Rotating Housing
2 O-Ring
3 Spring
4 Retainer
5 Rotating Face
6 Stationary Face
7 O-Ring

P_H Hydraulic Force
P_F Spring Force
P_R Friction Force
P_{Sp} Interface Film pressure

Figure 1. Axial Forces on Mechanical Seals.

turbomachinery (Figure 2). Single-acting mechanical seals require a flush system for lubrication and cooling. A mechanical seal is installed on the product side of the arrangement, while a floating ring seals the buffer fluid to the atmosphere side. In the simplest arrangement, the seal flush comes directly from the compressor bearing lubrication system, and the leakage through the floating ring is combined with bearing discharge and returned directly to the lube oil reservoir.

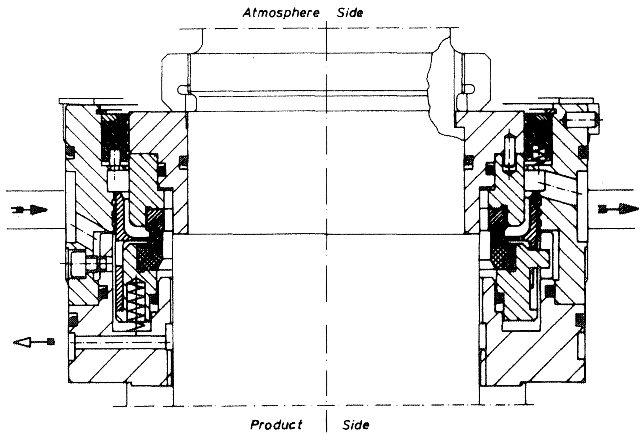


Figure 2. Single Mechanical Seal With Floating Ring.

A combined arrangement with basic similarity in design might be used in heavy duty pumps, where the mechanical seal is working on the atmosphere side, and the required cooling and lubrication flush is returned through a floating ring or throat bushing into the pump.

In cases where the product medium temperature exceeds the operating limitations of the seal, a throat bushing is used to separate a cooling loop in which a certain amount of liquid is circulated through a heat exchanger to maintain a feasible temperature within the seal cavity. Boiler feed pump seals are typically cooled in this manner.

In more severe applications, double-acting, face-to-face mechanical seals should be used (Figure 3). The arrangement requires a pressurized buffer fluid for cooling and lubrication and for the necessary hydraulic load. An arrangement such as this could be installed in any type of turbomachinery. Because the double-acting mechanical seal represents an independent operating system, the influences of the product, which might be either gas or liquid, are negated to some degree. By using the double-acting seal arrangement, the designer is enabled to adjust the main design parameters of the seal arrangement to the performance of the machine; as opposed to the installation of a single-acting seal, in which the characteristics of the product have a heavy impact on design.

It should be noted that mechanical seals in turbomachinery have a large advantage over any other seal arrangement. According to Mayer [1], it is recognized that floating ring type seals in boiler feed pumps with cool condensate injection have leakage rates which are exponentially higher than the leakage rates of mechanical seals. Leakage rates of mechanical seals operating in feed pumps under severe conditions, such as alternating speed, evaporation, and heavy temperature influence, can be controlled within a few liters per hour (l/hr), thereby offering significant savings in pump operation.

DESIGN PARAMETERS

Mechanical seals for turbomachinery should be carefully selected. The first consideration should be the product medium.

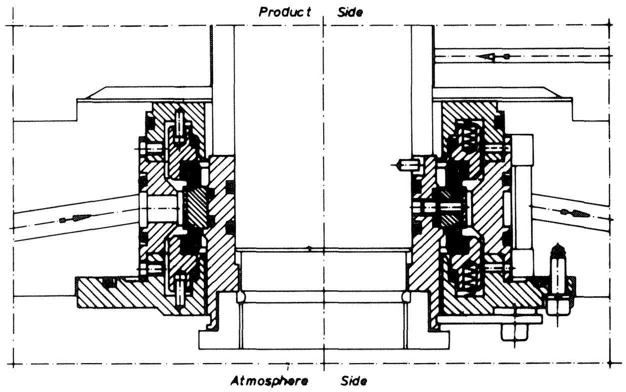


Figure 3. Double Mechanical Seal (Face to Face).

It is essential to look at the specific properties of the medium first: whether it be liquid or gaseous, the chemistry of the product, the content of abrasives, and the characteristics of the abrasives. From these, a preliminary selection of seal types can be made which will consider chemical resistant materials on contaminated parts, and the basic decision whether to use single or double acting mechanical seals may be made.

To develop the basics for selection even further, the temperature of the product must be considered. With regard to the temperature limitations of the components, a primary concern is the selection of secondary sealing elements. It is also necessary to know what distortions a seal arrangement might see, once temperature changes over a wide range occur. For example, a seawater injection pump at Prudhoe Bay in Alaska operates in an environment of $-50°C$, but it has a power consumption of approximately 28 kW [2]. In this case, the effects of temperature differentials become obvious.

For further determination of the design layout, the face speed, v_g, is a major parameter.

$$v_g = \frac{n \cdot \pi \cdot D}{60{,}000} \quad \left(\frac{m}{s}\right) \tag{1}$$

where

v_g = face speed ($\frac{m}{s}$)

n = rotational speed (rpm)

D = mean face diameter (mm)

As stated above, high face speed mechanical seals operate in the range of 20 m/s to well above 100 m/s.

Usually, it can be assumed that the pressure in the stuffing box of a turbomachine will be high, thus leading to the need for a hydraulically balanced mechanical seal. The face load caused by the pressure differential from the stuffing box to the atmosphere is determined as follows:

$$p_g = k \cdot p_1 + p_f \quad \left(\frac{N}{cm^2}\right) \tag{2}$$

where

p_g = face load ($\frac{N}{cm^2}$)

k = balance ratio

p_f = spring force ($\frac{N}{cm^2}$)

On double-acting seals, the face load, p_g, is determined for each seal face by the pressure differential that will apply, which is between buffer fluid pressure, p_3, and atmospheric pressure, and between the buffer fluid pressure, p_3, and stuffing box pressure, p_1.

The balance ratio, k, is used to reduce the face load, p_g, to a feasible amount by eliminating a possible overload on the seal face when pressure, p_1 (p_3), exceeds a certain limit. The balance ratio, k, is determined as shown below:

$$k = \frac{AH}{A} \tag{3}$$

where

k = balance ratio
AH = hydraulic loaded face area (cm^2)
A = contact face area (cm^2)

AH and A are explained in the comprehensive drawing of Figure 4. Balance ratio, k, values used for high speed mechanical seals are in the range of 0.7 to 0.8. The spring force, p_f, does not have a major influence on the equation. The average value used in designs for mechanical seals at elevated face speeds is in the range of 20 N/cm^2, with a tendency to lower values at larger diameters.

Construction Materials

Metal parts used in mechanical seals as described should be exclusively corrosion resistant. Stainless steel grades with high strength values and low thermal expansion coefficients are preferred. Ferritic materials, such as chrome steel, with at least 17% chrome content, adapt very well to specific requirements, such as minimized thermal distortion and high moduli of elasticity [3]. Austenitic materials should be considered if corrosion resistance is important. Table 1 shows some characteristic property values for the materials most commonly used.

Face Materials

The proven excellent performance of the face material combination carbon against carbide at high p_1 (p_3) × v_g values has resulted in the almost exclusive use of this combination in high speed seals.

Carbons used are synthetic ceramic carbons, with a variety of impregnations which should be selected based upon mechanical and thermal properties. Antimony impregnated carbons have increased thermal conductivity and therefore should be applied whenever possible. Table 2 shows typical material properties of carbons commonly used.

Carbides might be divided into the two main groups tungsten carbide, with nickel and cobalt impregnation, and silicon carbide. Nickel impregnated tungsten carbide has proven its reliability for many years. Compared with other face materials, tungsten carbide can be finished to very low face roughness values, which have a positive influence on the wear rate of carbon.

Lately, silicon carbide is more often applied in heavy duty seals. Due to its hardness, chemical resistance, and increased thermal conductivity, it has proven its superiority in high speed applications. Considering its excellent resistance to heat checks, silicon carbide, with its ongoing development, seems to be the best face material available yet (Table 3).

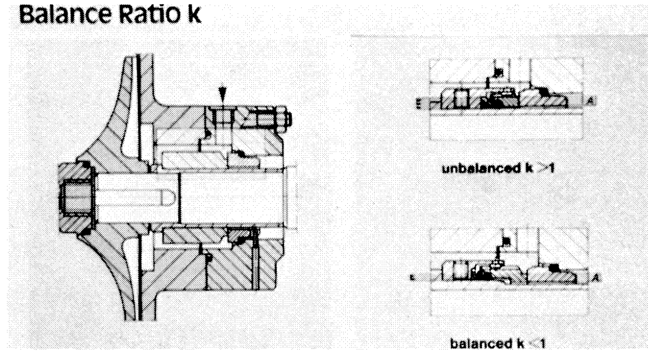


Figure 4. Hydraulic Balance on Mechanical Seals.

Secondary Sealing Elements

Of all components for a mechanical seal, the O-ring seems to be the one with the most limitations. It should be pointed out that the variety of elastomers available on the market must be carefully evaluated and tested for application in mechanical seals. Elastomer properties such as excessive compression set, embrittlement, softening, and temperature limitations in a particular type of liquid can result in the malfunction of a mechanical seal.

The use of polytetrafluorethylene (PTFE), which has very good resistance to chemical attack, is not the solution to all O-ring problems. Unsteady thermal expansion, a tendency to extrude, relatively poor thermal conductivity, and high stiffness do not allow the application of PTFE as solid material. In many applications, PTFE foil-wrapped elastomer cores have shown sufficient elasticity and resilience and therefore may be preferred [4]. Table 4 and Figure 5 show some of the characteristics stated above.

OPERATING PARAMETERS

Once all items stated so far have been evaluated, the basic design idea can be developed into a final stage. It should be noted at this point that any mechanical seal design can only be as good as the common understanding between the turbomachine designer, the seal designer, and the turbomachinery user. Entire knowledge about the elementary parameters of each technique involved will certainly help to provide larger stuffing boxes in the future, so that space restrictions do not affect mechanical seal design. Another example is that the knowledge of pressure and temperature gradients in machine housings where a mechanical seal will be adapted, and an understanding of how the equipment will be operated and maintained might be very helpful in avoiding the malfunction of mechanical seals.

Table 1. Physical Data of Seal Construction Metals (Mean Value).

Metal	DIN-number	Hardness HB (N/mm^2)	0.2% Tensile Strength (N/mm^2)	Modulus of Elasticity (10^4 N/mm^2)	Coefficient of Expansion (10^{-6} K)	Heat Conductivity (W/mK)
Cr-steel	1.4122 × 35 CrMo17	2250-2750	600	21.3	10.5	29.3
Cr-Ni-steel	1.4057 × 22 CrNi17	2250-2750	600	21.0	10.0	25.2
Cr-Ni-steel	1.4313G-X5 CrNi13 4	2300-3000	650		12.0	25.2
Cr-Ni-Mo-steel	1.4460 × 8 CrNiMo27 5	1900-2300	500	21.0	11.5	14.6
Cr-Ni-Mo-steel	1.4571 × 10 CrNiMoTi18 10	1300-1900	230	20.3	16.5	14.6

Table 2. Physical Data of Carbon Face Materials (Mean Value).

Carbon Material	Compressive Strength (N/mm^2)	Density (g/cm^3)	Modulus of Elasticity ($10^4 N/mm^2$)	Coefficient of Expansion ($10^{-6}/K$)	Heat Conductivity (W/mK)
Synthetic carbon resin impregnated	230	1.8	2.1	4.3	10
Synthetic carbon antimony impregnated	310	2.5	2.7	4.7	13
Electro graphite resin impregnated	150	1.8	1.1	4.3	65
Electro graphite antimony impregnated	170	2.5	1.6	4.5	65

Table 3. Physical Data of Carbide Face Materials (Mean Value).

Carbide Material	Hardness (K/mm^2)	Density (g/cm^3)	Modulus of Elasticity ($10^4 N/mm^2$)	Compressive Strength (N/mm^2)	Coefficient of Expansion ($10^{-6}/K$)	Heat Conductivity (W/mK)
Tungsten carbide	15,000	14.8	60	5000	4.5	82
Silicon carbide	25,000	3.1	41	3500	4.3	84
Comparison:						
Aluminum oxide	22,000	3.9	35	4000	7.0	26

Power Consumption

The power consumption on a mechanical seal consists of two major portions.

$$P_{ges} = P_r + P_T \quad \left(\frac{Nm}{s}\right) \tag{4}$$

where

P_r = power consumption related to friction losses $\left(\frac{Nm}{s}\right)$

P_T = power consumption related to turbulence losses $\left(\frac{Nm}{s}\right)$

$$P_r = p_g \cdot v_g \cdot f \cdot A \tag{5}$$

where

p_g = face load (see Equation 2)

v_g = face speed (see Equation 1)

f = coefficient of friction

A = contact area (see Equation 3)

The coefficient of friction, f, is usually empirically determined by a sequence of tests for each face material used. For the high speed mechanical seal, mixed lubrication between the faces should be attained which will lead to a value of 0.05. This value can be reduced drastically by introducing a hydrodynamic lubrication.

$$P_T = 1.02 \cdot 10^{-6} \cdot D^{3.6} \cdot n^{2.8} \cdot e \cdot 980.6 \quad \left(\frac{Nm}{s}\right) \tag{6}$$

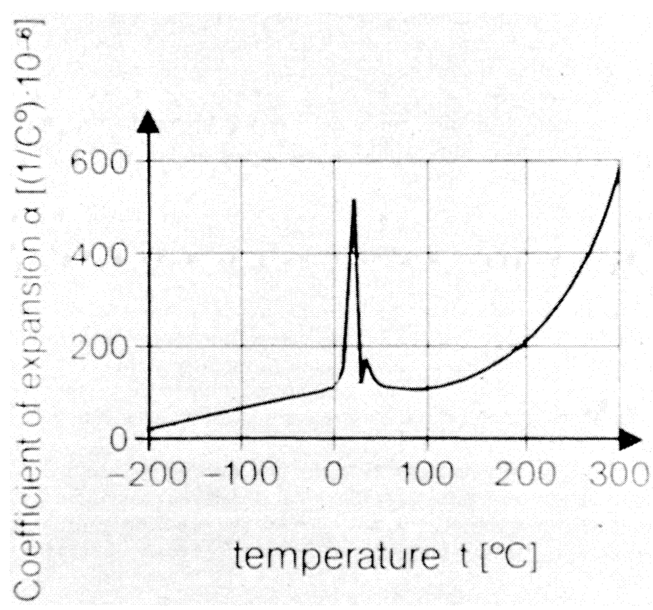


Figure 5. Coefficient of Thermal Expansion of PTFE.

where

D = outside diameter of rotating member (mm)

n = rotational speed (rpm) (see Equation 1)

e = length of rotating member (mm)

According to Mayer [1], the turbulence losses can exceed the friction losses on a mechanical seal at face speeds above 50 m/s. To reduce turbulence losses, considering Equation 6, it becomes obvious that a major reduction can be achieved with

Table 4. Temperature Ranges for the Use of Conventional O-Rings, Elastomeric and Non-Elastomeric Materials.

Material	Temperature Limit (°C)	Comments
Nitril rubber (Buna N)	−30 . . . + 90	
Chloroprene (Neoprene)	−30 . . . +120	
EP rubber	−40 . . . +190	Not resistant to mineral grease and oils
Silicon rubber	−100 . . . +180	
Fluorine rubber (Viton)	−50 . . . +190	With hot water only to a maximum of 120 °C
Hypalon	−30 . . . +120	
Butyl rubber	−40 . . . +140	Not resistant to mineral grease and oils
Kalrez®	. . . +270	Fluorine-containing solvents may result in swelling
PTFE	−200 . . . +250	As a solid ring very hard and non-resilient

optimized diameter and length of the rotating member of the mechanical seal. This, consequently, leads to face to face seals, as already introduced, where the feature of a minimized rotating mass can be used. Besides the installation of the springs in the stationary members of the seal, which should be considered at speeds above 25 m/s, and a dynamically balanced rotating ring, this arrangement represents an optimum layout.

Total power consumption on seal arrangements, as explained, might approach 20 kW. The efficiency of the buffer fluid flow, and the thermal conductivity of the parts generating heat are therefore very important to the design. Besides the already explained material properties, such as thermal conductivity, it is necessary also to control and direct buffer fluid flow to areas with the highest temperatures. It has been experienced, while operating in water under elevated load conditions, that inefficient face cooling subsequently created a gas ring circumferential to the seal faces. This led to dry running conditions and severe damage to the seals. Therefore, flow guides which force cool, clean liquid directly to the seal faces are imperative on high speed mechanical seals (Figure 2). In some cases the flow guides are even designed to follow the wear of the carbon member of the seal, so that the injection is strictly directed to the face area during seal operation. Due to the turbulent flow in the seal cavity, no separation of hot and cold liquid will occur. This eliminates hot spots in some undetected areas of the seal, which would lead to distortions that would have a negative influence on the seal's performance.

Leakage

As stated earlier, the leakage can be controlled on mechanical seals.

$$Q = \frac{\pi \cdot d_E \cdot \Delta p \cdot h^2 \cdot s}{p_g^2} \tag{7}$$

where

Q = leakage volume (cm^3/hr)

d_E = entering diameter (cm)

Δp = pressure differential ($\frac{N}{cm^2}$)

h = interface clearance (10^{-4} cm)

s = gap factor ($N/cm^2 \cdot sec$)

p_g = face load ($\frac{N}{cm^2}$) (see Equation 2)

For a theoretically parallel running sealing gap, the leakage path between the faces depends on the roughness of the surfaces. With a surface roughness, Ra, of 0.05 μm for silicon carbide, and 0.03 μm for carbons (standard quality values), the interface clearance, h, is assumed to be 1 μm.

Gap factor, s, is based on the discharge cross section of the faces and the centrifugal pressure as a function of the face speed. Given a set of operating conditions, and using knowledge gained from previous testing experiences, the parameters in the equation can be adjusted. In high speed applications, for instance, the amount of leakage is greater, thus leading to improved face cooling and the elimination of excessive face contact.

Wear and Life Expectation

The wear height of a mechanical seal will be in the range of approximately 3 mm. This represents the optimum, considering possible thermal and mechanical distortions on the rather sensitive carbon. Assuming that a parallel seal gap can be maintained in all operating situations, wear rates will be at a minimum, leading to life expectation of several thousand hours.

A key to the effects of wear on a mechanical seal is the seal face combination with major respect to thermal conductivity. It is a rule of thumb to install the seal face with the higher thermal conductivity on the heat sink side of the arrangement.

Wear on mechanical seals can be divided into five main groups:

1. adhesive wear
2. grinding wear
3. corrosive wear
4. surface wear
5. erosion wear

With the exception of Item 1, adhesive wear, it is possible to adjust all other main groups to the specific requirements of the seal operation. On high speed seals, grinding and surface wear have a major impact. Due to the high flow speed for cooling purposes, erosion wear should also be taken into consideration.

Additional Features

It has been proven in many field installations that easy to handle units are preferred for quick maintenance. For mechanical seals in turbomachinery, cartridge units are almost imperative (Figure 6). Mechanical seals which are preassembled at the manufacturer's site provide proper alignment of all internal parts. Such cartridge units might be tested on the manufacturer's test rig, where the performance of the mechanical seal can be recorded prior to delivery.

On the rig, the seal is operated at its nominal speed and at a constant buffer fluid inlet temperature. The pressure p_1 (p_3) is then adjusted in steps to maximum pressure. Power consumption of the test rig drive is plotted according to the pressure changes. If the idle power consumption of the test rig motor is deducted, the total power consumption of the seal can be recognized (Figure 7). During the test operation of the seal, the buffer fluid outlet temperature, as well as the amount of leakage, will be recorded.

A mechanical seal, especially in critical applications, requires manufacturing of parts with very close tolerances. An extended quality assurance program that controls in-house manufacturing processes at strategically integrated locations will assure that the quality is maintained at the prescribed standards. Statistical and numerical quality control should be checked against specifications given to the subsupplier. It is understood that a precision mechanism, such as a mechanical seal, does not allow any deviation from the highest standards available for this line of product.

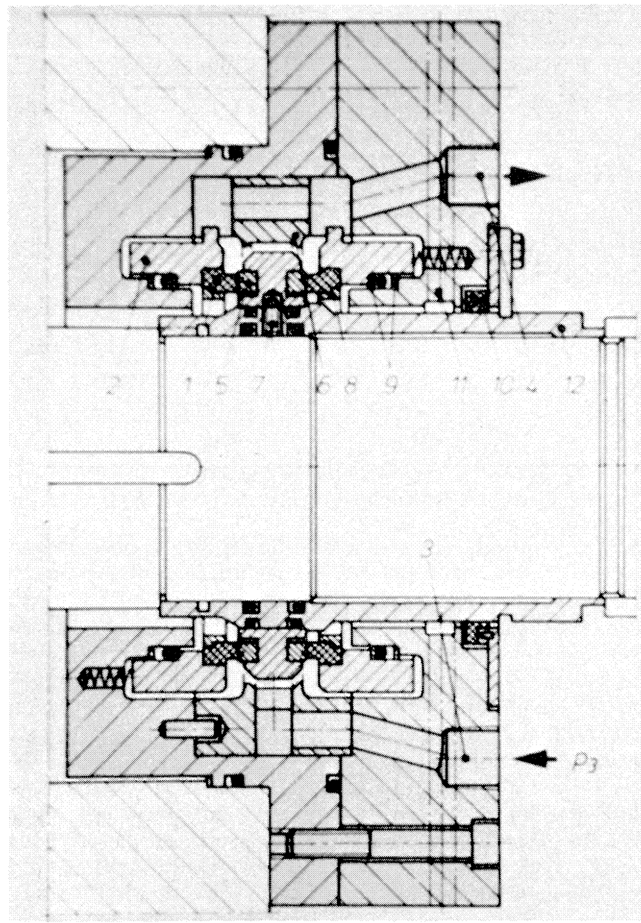


Figure 6. Cartridge Unit.

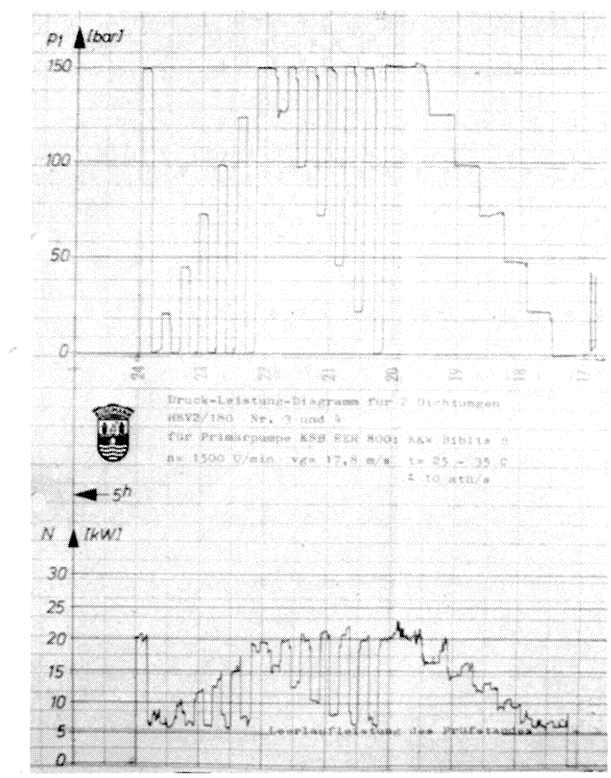


Figure 7. Performance Diagram.

CONCLUSION

From all items explained thus far, it is possible to define current limitations of experience for high speed mechanical seals. Referring to Figure 6, the seal is operating at the conditions listed below:

Product:	propane, butane, pentane
Pressure p_1:	80 bar (800 $\frac{N}{cm^2}$)
Temperature:	10 – 50 °C
Buffer fluid:	hydraulic oil, 4 °E at 50 °C
Pressure p_3:	100 bar (1000 $\frac{N}{cm^2}$)
Rotational speed:	6000 rpm

The seal is installed in a gas reinjection pump located in Saudi Arabia. It has a total power consumption of 19.5 kW. The $p_3 \cdot v_g$ value is 5000 bar $\cdot \frac{m}{s}$, which is the highest value yet for a mechanical seal installed in an industrial application.

Due to ongoing developments in the area of face materials (e.g., upgraded silicon carbide), it is possible to look at applications of $p_3 \cdot v_g = 6000$ bar $\cdot \frac{m}{s}$, and even extended values, in the not too distant future.

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