PRESSURE RATINGS OF MECHANICAL SEALS

by
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ABSTRACT

Although mechanical seals are widely used in many machines, there is no universally accepted system for establishing limits of pressure, temperature, and speed. As a result, manufacturers of mechanical seals may use different methods for establishing these limits. In turn, users of mechanical seals sometimes have their own rating system. Presented here is a general discussion of the difficulties involved in establishing limits along with some specific issues related to seals used in centrifugal pumps. Recommended definitions are given for terms to be used in establishing pressure ratings of mechanical seals in centrifugal pumps. Some simplified methods for evaluating the suitability of commercially available mechanical seals in particular services are presented and discussed.

INTRODUCTION

"What is the rating for that seal?" A simple question deserves a simple answer; however, when the object of the question is a complex machine like a mechanical seal, the answer is not likely to be a simple one. In fact, there is no single answer to this question. The purpose herein is to describe the various pressure limits of contacting mechanical seals for liquid services and to explain how these limits are sometimes said to be the "rating" of the seal.

When the question is asked, the desired answer is a pressure (or temperature) that is less than some maximum value established for that service. The question is often based on an analogy to piping or pressure vessel components. For example, an ANSI Class 300 carbon steel pipe flange might be rated at 720 psig at 100°F; at 600°F, the rating decreases to 555 psig. Naturally, someone purchasing a mechanical seal for a service using such flanges would like assurance that the seal had a similar rating. Unfortunately, there is no standardized rating procedure for mechanical seals.

Whereas piping and pressure vessels are fixed equipment with ratings that can be well defined by structural limits, mechanical seals are dynamic with both structural limits and performance limits. In addition, the performance of the mechanical seal is dependent on the fluid that is being sealed and the operation of the equipment. Moreover, the overall performance of the seal may even be time dependent as the physical properties of various components, especially elastomers, change during operation. Finally, performance of a seal is often path dependent; that is, the steady state performance depends on the startup conditions and procedure. Again, rating a mechanical seal is not a simple process.

The technical literature contains much useful information for rating mechanical seals. Early efforts at rating mechanical seals emphasized wear and employed the pressure velocity (PV) value, i.e., the product of face pressure and speed. Schoenherr [1] describes the use of PV as a design parameter for mechanical seals. Schoenherr [2] gives further information about the relationship between the PV value and wear. Buck [3] showed that the PV value did not provide a particularly good correlation to seal life, especially for volatile fluids. Lebeck [4] describes how the PV limit depends on not only the mating pair of face materials, but also on the tribological and heat transfer design of the seal.

Abar [5] showed how the cyclic pressure limits of a low balance ratio seal were governed by pressure induced distortion of the seal faces. In his tests, it was possible to have satisfactory operation at high pressure but excessive leakage when a worn-in seal was slowly repressurized from a lower pressure.

Abar [5] also showed how the PV limit for a particular seal design and set of materials decreased with increasing temperature when operating in water. According to Johnson and Schoenherr [6], wear rate increases with increasing temperature and is roughly inversely proportional to the temperature. Paxton [7] describes how the wear rate of carbon graphite materials typically increases slowly with temperature until, at some upper limit, the wear rate increases dramatically. He noted that there could be a tenfold difference between various carbon grades. Flitney and Nau [8] reported that, in general, seal life was reduced at higher temperatures.

Although seal manufacturers undoubtedly have their own proprietary data, there is little information in the open literature regarding high pressure static tests of mechanical seals. The carbon graphite pusher seals used by Abar [5] were operated up to 2100 psig. Commercially available carbon graphite pusher seals are available for pressures up to 3000 psig [9]. For welded metal bellows, Datta, et al. [10], mention tests of one and two inch, single ply, cores with burst pressures of 8,800 and 6,600 psi respectively.

Previously available information will be combined with some recent test data and simple equations to illustrate the various parameters that affect the overall pressure ratings of contacting mechanical seals in liquid service.

PERFORMANCE REQUIREMENTS

Before service related ratings can be provided for the mechanical seal, it is first necessary to define expected performance. Without a specification, a default should be provided by the seal manufacturer.

The only available standard for mechanical seals, API 682 [11, paragraph 1.1.3] requires that, seals must ". . . have a high probability of meeting the objective of at least three years of uninterrupted service while complying with emission regulations."

The performance of a mechanical seal can be thought of as a compromise between leakage and wear. It is well known that the wear rate (and friction) of a leaky seal can be very low. In general, when leakage rates are reduced, wear rates increase. Within this relationship, and using modern designs incorporating high performance material, nearly any level of performance can be obtained through selection of the seal arrangement.

For purposes of illustration, the performance of a single mechanical seal is presented, although the concepts explained herein are also applicable to multiple sealing arrangements.

PRESSURE LIMITS

Although API 682 [11] does not apply to all seals, it is useful to note that it defines three pressure terms that relate to the mechanical seal:

- *Maximum static sealing pressure* is "The highest pressure, excluding pressures encountered during hydrostatic testing, to which the seal (or seals) can be subjected while the pump is shut down."

- *Maximum dynamic sealing pressure* is "The highest pressure expected at the seal (or seals) during any specified operating condition and during startup and shutdown. In determining this pressure, consideration should be given to the maximum suction pressure, the flush pressure, and the effect of clearance changes with the pump."

- *Maximum allowable working pressure* is "The greatest discharge pressure at the specified pumping temperature for which the pump casing is designed."

API 682 also defines the pressure casing as including the seal chamber, but excluding the stationary and rotating members of the mechanical seal. This means that there is no requirement that the seal have the same maximum allowable working pressure as the pump. Obviously, but not stated, the seal is expected to have a static pressure rating equal to, or exceeding, the maximum static sealing pressure for that service. In addition, the seal obviously is expected to have a dynamic pressure rating equal to, or exceeding, the maximum dynamic sealing pressure for that service; again, this requirement is not stated in API 682.

The *Seal Ratings* section in the data sheets of API 682 includes places to record the *maximum static sealing pressure* and *maximum dynamic sealing pressure* at the pump temperature. There is some confusion about whether the intention is to supply the maximum pressures that might occur in that service (as noted in the definitions) or the rated values for the seal. Because of the location in the Seal Ratings section, the requested information is assumed to be the rated values for the seal. Unfortunately, API 682 does not provide guidance about how these ratings might be consistently determined. That subject is addressed herein.

STATIC SEALING PRESSURE RATING

The *static sealing pressure rating* should be defined as:

- The highest pressure that the seal can continuously withstand at the pumping temperature while the shaft is not rotating. Thereafter, the seal must maintain its dynamic sealing pressure rating.

Along with the static sealing pressure rating, there should be a hydrostatic test pressure rating for the seal that should be defined as:

- The highest pressure that the seal can withstand for 30 minutes at ambient temperature on water while the shaft is not rotating. Thereafter, the seal must maintain its static and dynamic sealing pressure ratings.

These definitions establish the difference between static and hydrotest ratings. They also ensure that performance is not affected by static conditions within the limits of the static sealing pressure rating.

In many cases, the static rating will be equal to the hydrotest rating. This is not the normal practice for pressure vessels in which a ratio of 1.5 is employed between maximum allowable working pressure and hydrotest pressure. However, as will be shown later, the static pressure rating can be significantly below pressures that result in structure damage or rupture of the seal.

With terminology established, the static sealing pressure rating can be considered quantitatively.

Pusher Seals

A single, rotating pusher seal using multiple springs is illustrated in Figure 1. For simplicity, an unbalanced seal is shown; however, the balance ratio does not necessarily affect the static pressure rating. The primary ring is manufactured from a single piece of carbon graphite. The mating ring might typically be manufactured from silicon carbide or tungsten carbide. Some physical properties of typical seal materials are shown in Table 1.

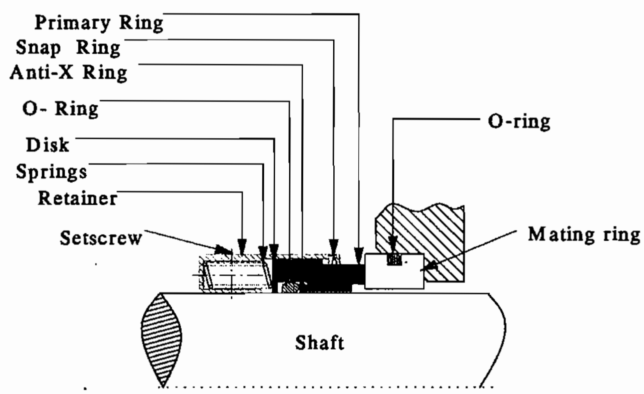

Figure 1. Typical Pusher Seal.

Table 1. Typical Properties of Materials.

Typical Property	316SS	Carbon Graphite	Tungsten Carbide	Silicon Carbide
Modulus of Elasticity, Mpsi	28	3	85	60
Compressive Strength, Kpsi	30 [1]	30	650	67
Tensile Strength, Kpsi	30 [1]	7	200	45
Coefficient of Expansion, in/in °F x 10^6	8.9	2.5	3.4	2.5
Thermal Conductivity, Btu/hr ft °F	9.4	5	50	80

Note 1. Based on yield strength.

When the seal is pressurized, stresses are produced within the primary ring and mating ring. These stresses may be estimated by using the equations for stress in a cylinder. Equation (1) is the relationship for a thin wall cylinder and is used herein for purposes of illustration; however, using the relationships for a thick wall cylinder is more accurate.

$$\sigma = (P_i - P_o)\frac{D_m}{2t} \tag{1}$$

The use of Equation (1) produces positive values, that is, tensile stresses, for internal pressurization and negative values, that is, compressive stresses, for external pressurization. These stresses must be compared to the tensile or compressive strength of the material. Examination of Table 1 shows that the carbon graphite is much weaker than either tungsten carbide or silicon carbide. Since these components are often roughly the same size, the carbon graphite primary ring is usually, but not necessarily, the limiting component with respect to pressure induced stresses.

As an example, suppose that the primary ring has a four inch mean diameter with 1/4 inch cross sectional wall thickness. Using Equation (1), the stress at, say, 1000 psi external pressure is 8,000 lbf/in^2 in compression. With reference to Table 1, this level of stress is well within the compressive strength of carbon graphite.

If, on the other hand, the same seal is pressurized internally, the stress is 8,000 lbf/in^2 in tension. The particular carbon graphite shown, with a tensile strength of only 7,000 lbf/in^2, is not adequate for this pressure. A stronger carbon graphite (or perhaps another type of material) would be required. Alternatively, the wall thickness could be increased.

Figure 2 was constructed using Equation (1) with carbon graphite properties from Table 1 and a safety factor of two for comparing computed stresses to strength. The maximum allowable pressure is shown as a function of seal size and wall thickness. Both external and internal pressure limits are shown.

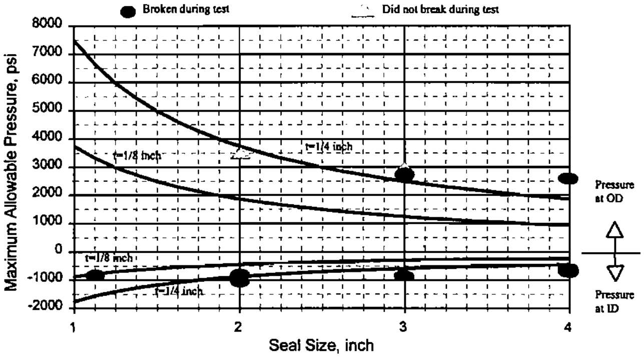

Figure 2. Pressure Limits of Seal-Like Cylindrical Shapes Based on Stresses.

The results of some static burst tests on carbon graphite primary rings are superimposed on Figure 2. The primary rings that were tested were of the same general design as shown in Figure 1; however, they were balanced seals. Typical wall thicknesses were on the order of 1/8 in to 1/4 in. Where breakage occurred, the typical mode of failure was a crack near a stress concentration factor such as a change in cross section. In general, a safety factor of two is more than adequate when stresses are computed using the thick wall equations.

A good illustration is shown in Figure 2 of how the static pressure rating of a seal decreases with increasing seal size for a given cross section thickness.

Welded Metal Bellows Seals

The computation of stresses in a welded metal bellows is much more complex than the computation for a monolithic pusher seal. Just as the stress in the simple pusher seal was estimated by using the equations for a thin shell, the stress in a welded metal bellows can be estimated by using the equations for a simple beam.

The total stress in a bellows can be thought of as consisting of two components: the stress from compressing the bellows and the stress from pressurization. To simplify computations, the stress due to compressing the bellows will not be computed here; however, it can be significant. The stress resulting from pressurization can be estimated as:

$$\sigma_p = \frac{1}{2} C_1 C_2 \left(\frac{S_e}{t}\right)^2 \Delta P \tag{2}$$

In Equation (2), the coefficients, C_1 and C_2, are employed to improve the accuracy (indeed, to render the equation useful!) of the correlation. In comparing Equation (2) to various finite element results and to older empirical equations, C_1 appears to range from 0.2 to 0.6. The shape coefficient C_2 is around 0.7 for tilt edge bellows and 1.0 for straight edge bellows. In any case, this equation produces very rough, but illustrative, estimates for bellows stress.

Figure 3 was produced from Equation (2) under the following assumptions:

- The stress due to compressing the bellows was 25 percent of the yield stress for a one inch seal and 10 percent of the yield stress for a four inch seal. These values are somewhat representative, but not necessarily a requirement, of typical bellows designs. Intermediate values were determined by interpolation.
- The equivalent span of the bellows was assumed to be 0.1 in for a one in seal and 0.3 in for a four in seal. Again, these values are somewhat representative. Intermediate values were determined by interpolation.
- Values of 0.4 and 0.7 were used for C_1 and C_2, respectively.

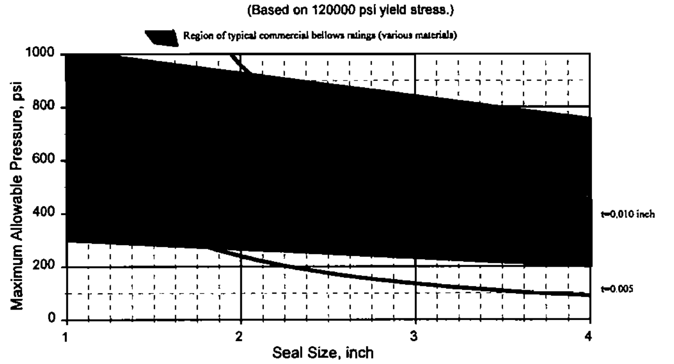

Figure 3. Pressure Limits of Welded Metal Bellows.

The shaded area in Figure 3 represents the broad range of maximum pressures that are in commercially available seals using various designs and materials. In examining Figure 3, it is important to realize that the smaller sizes usually have a thinner bellows than the larger sizes. This is usually necessary to get the required spring rates within available dimensional constraints.

The allowable pressure for metal bellows seals decreases with increasing seal size–just as the pusher ratings decrease; this is shown in Figure 3. A comparison of Figures 2 and 3 shows that the allowable pressure for a welded metal bellows seal is typically less than the rated value for a pusher seal; however, there is an important difference: Figure 2 is based on bursting as the failure mode, whereas Figure 3 is based on yielding.

The burst pressure of welded metal bellows is significantly higher than the pressure which causes yielding. This relationship is somewhat analogous to yielding versus bursting of pipe and pressure vessels. Tests by the author's company show that bursting occurs at pressures four to five times greater than the predicted yield. Even so, welded metal bellows seals should be limited to pressures that do not cause yielding because the performance can change after yielding occurs.

O-Ring Extrusion

Many pusher seals and some welded metal bellows seals use an O-ring or other elastomeric gasket as a secondary sealing element. Under extremes of high pressure, the elastomer can extrude into the surrounding clearances. Particularly for the dynamic secondary sealing element of the pusher seal, this extrusion can be a limiting factor. Guidelines for avoiding O-ring extrusion are available [12], but guidelines for other shapes are not so readily available. Some of the guidelines and limitations are shown in Table 2 where O-ring extrusion is considered.

Table 2. Pressure Limits to Avoid O-Ring Extrusion.

Durometer, Shore A	Diametral Clearance, Inch		
	0.005	0.01	0.02
70	1300	800	350
80	2000	1300	550
90	3400	2000	900

With reference to Table 2, it is important to recognize that the extrusion of the O-ring would result in a degradation of seal performance and not a rupture. As such, consideration of O-ring extrusion should be included in the static and hydrotest ratings along with the dynamic rating. It should also be noted that high durometer O-rings usually have more frictional drag and less flexibility than low durometer O-rings.

Diametral clearances may change with pressure according to the design, material, and construction of the seal. For example, for the seal shown in Figure 1, the diametral clearance for the dynamic O-ring (inside the primary ring) would decrease with external pressurization. On the other hand, the diametral clearance for the mating ring O-ring would increase with external pressurization. For high modulus materials such as silicon carbide, this change is very small. On the other hand, because carbon graphite has a relatively low modulus of elasticity, changes in clearance with pressure must be considered. Many high pressure seals and some general purpose seals use anti-extrusion rings to prevent extrusion while maintaining adequate clearance.

Sleeves

Seal sleeves are relatively thick and, when treated as a part of the pressure casing, do not usually affect the pressure rating of the seal. Cartridge sleeves using set screws for positioning, thrust, and torque transmission have a definite pressure limit that is a function of the number, size, type, and material of the setscrew. This topic is mentioned herein for completeness and as a cautionary reminder.

Glands and Gasketing

Pressure ratings of glands and gasketing associated with the complete seal package may be developed using conventional methods. As noted previously, these components are considered part of the pressure casing. This topic is mentioned only for completeness and as a cautionary reminder.

DYNAMIC SEALING PRESSURE RATING

The *Dynamic Sealing Pressure Rating* should be defined as:

- The highest pressure that the seal can continuously withstand at the pumping temperature while the shaft is rotating. Thereafter, the seal must maintain its static sealing pressure rating.

When discussing dynamic ratings, the relationship between leakage and wear becomes particularly apparent. In mechanical seal technology, the classical, but simplistic approach to analysis of wear is by using the PV value.

PV Value

The PV value is widely used as a guideline for mechanical seal design and application. The standard calculation is described in Schoenherr [1] as:

$$PV = [\Delta P(b - k) + P_{sp}]V \tag{3}$$

The PV value can be important because it represents both wear and heat generation. Even so, it has only a very rough correlation with overall seal performance. In Equation (3), the pressure gradient factor, k, is taken as $1/2$ according to Schoenherr [1]. In actual service, the value for k may vary from zero to one. Nevertheless, it is convenient and the usual practice is to use $k = 1/2$ when dealing with PV calculations.

Industry practice has been to limit the PV value based on seal face material combinations and to attempt to couple this limit with fluid properties. For example, a typical limit is 500 kpsi ft/min for resin filled carbons *vs* tungsten carbide for seals in "nonlubricating" liquids in typical pump services. Higher values (usually by a factor of 1.6) are allowed for "lubricating" liquids. Some examples are given in Table 3 of the recommended maximum PV values for various material combinations.

Table 3. Maximum PV Based on Two Year Wear Life in Nonlubricating Service for Plain Faced Seals.

Material Pair	Maximum PV Value psi ft per minute
Carbon vs Silicon Carbide (or Tungsten Carbide)	500,000
Carbon vs Alumina	100,000
Silicon Carbide vs Tungsten Carbide	350,000

Table 3 is not meant to be an all-inclusive reference; it merely illustrates the concept and general order of magnitude of PV limits for a material pair. In working with PV limits, it is important to recognize the basis and limits of this concept. Much of the data is based on tests of a commercially available 3-$3/8$ in unbalanced seal at 1800 rpm in warm water. The tests are run for 100 hr and the wear is measured. The PV limit is defined by either excessive wear for the target design life or by damage due to some sort of "overload"–usually thermal in nature. Some historical data at both 1800 and 3600 rpm indicate that the maximum recommended PV is not constant with respect to speed. It even appears that the allowable PV value increases with speed! For example, to obtain a projected two year wear life, a particular combination of carbon and tungsten carbide was rated for a PV of 300 kpsi ft/min at a velocity of 1820 ft/min but 525 kpsi ft/min at 3600 ft/min. The absolute maximums, based on face damage, were 467 and 587 kpsi ft/min, respectively. This type of relationship is consistent with hydrodynamic load support during the PV test.

When PV limits are used as the basis for wear, it is important to recognize that the maximum recommended values are often based on a *two year* life *with a $1/8$ in wear length.* This is 0.00071 in wear during the 100 hr test–a difficult measurement to obtain accurately! If a longer wear life is required, then the PV limit must be lowered. If the available seal wear length is less than $1/8$ in, then either the PV limit must be lowered or a lesser wear life is projected. These are extreme extrapolations–a 17,520 hr wear life is projected from a 100 hr test! In the 100 hr test, most of the wear undoubtedly occurs during the first few hours of the test. *In actual long term service, mechanical seals rarely wear out.*

It is interesting that the qualification tests of API 682 for flashing services are based on a two in and a four in seal at 3600 rpm in propane at 250 psig and 90°F. These tests include 100 hr of steady state operation followed by five simulated upsets. Assuming a typical balance ratio of 80 percent and 30 psi unit spring load, the nominal PV values are around 210,000 and 420,000 psi ft/min. Gabriel [13] reported on successful qualification with wear rates typically less than 0.0005 in for seals with a plain face and still less wear for seals using a hydropadded face. Gabriel also reported on wear rates for partial tests. He found that most of the total wear apparently occurred during the early stages of testing.

Massaro [14] compared the results of a 500 hr hot water test to a 100 hr hot water test. The materials were resin filled carbon graphite *vs* reaction bonded silicon carbide. The nominal PV value was approximately 490,000 psi ft/min. For the 100 hr test, the average wear (two tests) was 0.0047 in; for the 500 hr test, the average (also two tests) was 0.0058 in. This means that (in very rough terms) the average wear rate after the first 100 hr was only about six percent of the wear rate during the first 100 hr.

Although data are incomplete, it seems entirely reasonable to assume that the PV limits previously used to design for a two year wear life actually result in a much longer wear life. These same values can be used to design for a three year wear life–and *still be conservative.* For example, suppose that tests at a PV of 500,000 psi ft/min resulted in 0.00071 in wear in 100 hr. If the wear length were 0.125 in, the linearly extrapolated time to wear out would be two years. However, if the wear rate is only 0.0005 in per 100 hr after the first 100 hr, the time to wear out would be three years.

On the other hand, PV limits and generalized tables (such as Table 3) must be used with great care because sometimes these limits are based on damage rather than wear rates. This is particularly true for data on two hard faces.

With the above arguments in mind, Figure 4 was constructed to illustrate how PV limits affect the dynamic pressure rating. Figure 4 is presented in terms of limiting pressure *vs* seal size for various balance ratios at a shaft speed of 1800 rpm. Such charts are used in the following manner: Suppose premium materials are used such that the allowable PV is 500,000 psi/ft min. The maximum allowable pressure for a two in seal with 70 percent balance ratio can be read directly as 2500 psig at 1800 rpm. The same seal at 3600 rpm (twice the speed) would be rated for 1250 psig (half the pressure). This procedure neglects the unit spring load but yields a good approximation when pressure is high compared to unit spring load.

The pressure rating of an unbalanced seal is much less than that of a balanced seal; this is also shown in Figure 4.

It is important to realize that materials are available with higher (and lower) PV ratings than shown in Table 3 and illustrated in Figure 4. Seals with very good heat transfer designs may operate at higher PV–just as seals with poor heat transfer designs may be useful only at low PV. Also, seals with special face designs, such as hydropads, may operate at higher PV than indicated by the nominal PV calculation.

Stability Factor

Because the PV value appears in relationships for heat generation and wear and combines many seal parameters, it is widely used to evaluate seal designs and applications. Unfortunately, emphasis on the PV value encourages lightly loaded seal designs. In 1979, this author [3] proposed a dimensionless ratio, the *Stability Factor*, for evaluating the potential for lightly loaded seal designs to become unstable. The Stability Factor is:

$$SF = \left(\frac{P_{sp}}{\Delta P}\right)\frac{1}{(1-b)} \tag{4}$$

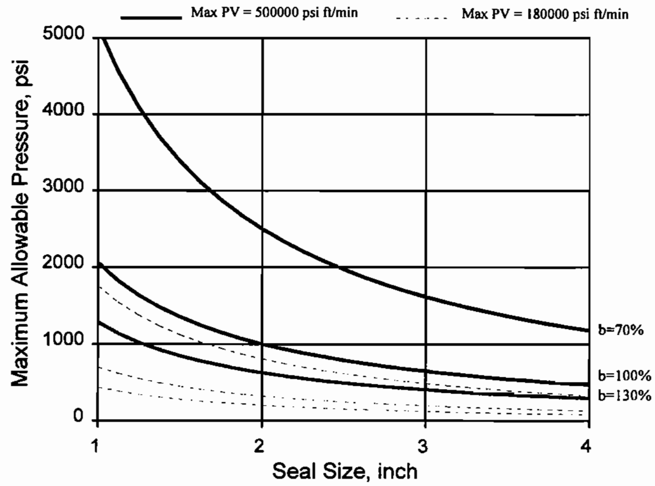

Figure 4. Maximum Allowable Pressure Based on PV Limits for Plain Face Seals at 1800 RPM.

In Equation (4), the quantity (1 - b) is derived by assuming that the pressure gradient factor, k, is at the maximum possible value of one.

If the Stability Factor is greater than one, the seal faces will remain closed even if there is significant distortion or flashing between the seal faces. If the Stability Factor is less than one, there is potential for the faces to separate and excessive leakage might occur. Unfortunately, just as an emphasis on the PV value encourages low balance ratios, the Stability Factor encourages high spring loads and high balance ratios. Both PV and Stability Factor should be considered during the rating process for a seal.

When PV is a limiting parameter and the Stability Factor is less than one, special attention must be given to control of seal face distortion and vaporization between the seal faces. Doing so can prevent the pressure gradient factor from reaching its maximum value. In addition, the operating conditions, including startup procedures, should be carefully reviewed.

In the tests by Abar [5], the cyclic pressure limits for a three in and 4-3/8 in seal with 65 percent balance ratio were 1500 and 900 psig, respectively. Abar noted that, at those pressures, the seal faces were worn to approximately the same angle–about six min of a degree.

EFFECT OF TEMPERATURE ON PRESSURE LIMITS

The previous discussions lead to the general conclusion that pressure limits decrease with increasing temperature. This decrease is because higher temperatures result in reduced material strength and increased wear rate. Also, the viscosity of liquids decreases with increasing temperature; therefore, the lubricity of the fluid being sealed decreases with increasing temperature.

In the tests conducted by Abar [5], the allowable PV limit was reduced by a factor of nearly 1/2 for a particular seal using carbon *vs* tungsten carbide when the water temperature was increased from 80°F to 160°F. Interestingly, the viscosity of water at these temperatures is reduced by approximately this same ratio.

The vapor pressure of fluids increases as temperature increases. A popular guideline is that, for reliable sealing, the sealing pressure should be at least 50 psi above the vapor pressure. An odd, but interesting aspect of this guideline is that it establishes a *lower* limit to the rated seal pressure. For example, suppose propane at 90°F is to be sealed. The vapor pressure of propane at 90°F is about 170 psia. This means that the seal should not be rated for *less* than 220 psia according to the 50 psi rule of thumb margin!

Elastomers usually soften as temperature increases; therefore, according to Table 2, the limiting pressure based on O-ring extrusion would also decrease. This effect may be reduced or increased by the differential thermal expansion between carbon and 316SS components.

HIGH PRESSURE SEALING

Sealing at high pressures can be particularly difficult. Applications are made even more difficult when a wide range of pressures is expected. In such applications, the end user naturally wants and expects a high "rating" for the seal. It is very easy, perhaps natural, to assume that the seal with the higher pressure rating is better than one with a lower rating. This is not necessarily so. In many cases, the end user might really need a seal with a high static pressure rating that is capable of operating at high pressures occasionally even though the normal operation is at modest pressures. Such a seal, designed for the true conditions, will probably perform better than a seal designed for the higher pressure.

As previously discussed, to have a high dynamic pressure rating and long theoretical life, a seal will always have a reduced balance ratio. This leads to problems with instability unless great care is taken to control distortion. Distortion from pressure and distortion from temperature are often in opposite directions. If the distortion control is targeted for an untrue high pressure condition, it includes the effects of heat generation on face temperature and thermal distortion. If the seal actually operates at a significantly lower pressure, the pressure and thermal distortions may not offset each other and the seal might leak excessively.

This author has seen such extremes as a normal pressure of 165 psig, a rated pressure of 950 psig, and a static rating of 3400 psig in specifications when the pump seal chamber was obviously at the suction pressure of 30 psig!

THE BASIC PRESSURE RATING

It has been shown that the dynamic rating can vary with many parameters, including size, temperature, fluid, speed, allowable leakage rate, and expected life. The various pressure limits that make up the dynamic rating can be shown on a single chart (Figure 5).

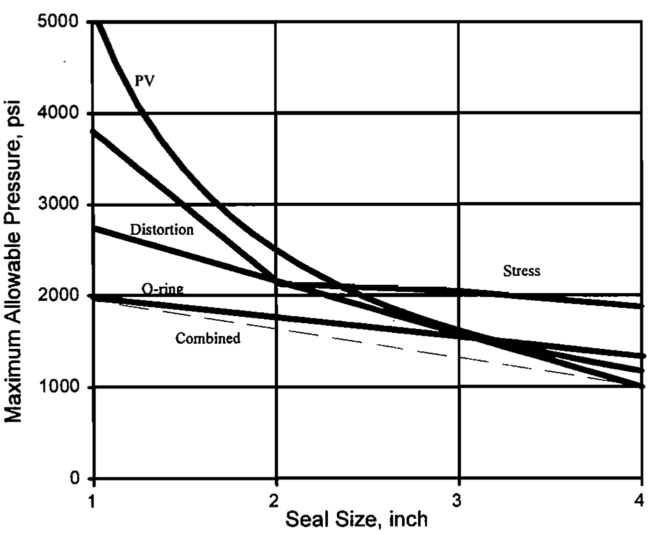

Figure 5. Quantitative Example of Superimposed Ratings at 1800 RPM.

Figure 5 was constructed using Table 2, Figure 2, and Figure 4 as the design bases of an imaginary seal family. In this seal family, the one in seal has 1/8 in minimum thickness and the four in seal has 1/4 in minimum thickness. The one in seal has a 0.005 in diametral clearance and the four in seal has 0.010 in clearance. Seals between one in and four in are proportioned according to size. The pressure limits based on O-ring extrusion in Figure 6 are for 80 durometer. The balance ratio is 70 percent and PV limits are 500,000 psi ft/min. The distortion limits are sketched in based on the tests by Abar [5]. The pressure limit based on PV in Figure 5 is for 1800 rpm shaft speed.

The combination of the pressure limits (the dashed line, shown for illustration just beneath the O-ring line) coincides with the O-ring limit for the small seals and with the distortion limit for the larger seals. The PV limit actually does not come into consideration. Using the concepts discussed herein, for operation at 3600 rpm, this combined limit could be multiplied by (1800/3600) to get the new pressure limit. However, the real pressure limit at 3600 rpm should be based on the superimposed pressure limits at 3600 rpm. This is illustrated in Figure 6.

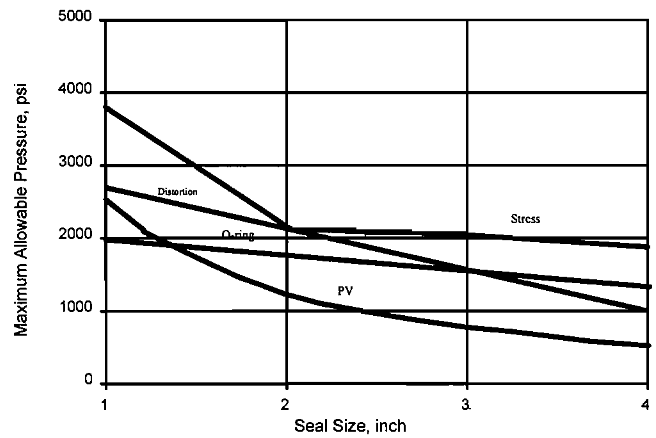

Figure 6. Quantitative Example of Superimposed Ratings at 3600 RPM.

Figure 6 is the same as Figure 5 except that the PV limits are drawn for 3600 rpm; that is, the pressures are approximately half the values shown in Figure 5. The O-ring, stress and distortion limits remain the same because these limits are not a function of speed. However, in Figure 6, the controlling factor is the PV limit (except for small seals). This PV limit in Figure 6 is generally higher than half the combined limit used in Figure 5.

If an O-ring with higher durometer elastomer were used, the limits based on O-ring extrusion would be higher; if antiextrusion rings are used, this limit essentially disappears. In fluids with better lubricity, the PV limits might be increased. By using a high strength carbon, metal, silicon carbide, or tungsten carbide, the stress limit can be raised significantly. These examples illustrate how seals can be used at higher pressures than given by the combined curves, but the application engineer must review each of the individual pressure limits: stress, extrusion, wear, heat generation, distortion, etc.

Combined limits shown in charts like Figures 5 and 6 are actually based on particular designs, including materials and configuration. Such charts are sometimes called PV charts which, unfortunately, implies that the rating is based solely on the engineering parameter PV. This author prefers to call these charts the *Basic Pressure Rating*. In Figure 7, the Basic Pressure Rating is drawn beneath the limits for stress, O-ring extrusion, and face distortion as well as the PV parameter. Even though the Basic Pressure Rating is drawn beneath other limiting factors, it can be taken as the limit of the individual factors and then manipulated accordingly so long as care is used. In particular, a conservatively low first estimate for the dynamic pressure rating can be *reduced* from the Basic Pressure Rating to account for conditions different from those assumed. Because only reduction factors are used, it is best to base the PV component of the Basic Pressure Rating curve on a lubricating fluid and a relatively low shaft speed.

As discussed previously, an increase in temperature often results in reduced pressure ratings. Temperature reduction factors can be developed to account for changes in strength and wear rate as well to reflect the general difficulty of operation at high temperatures.

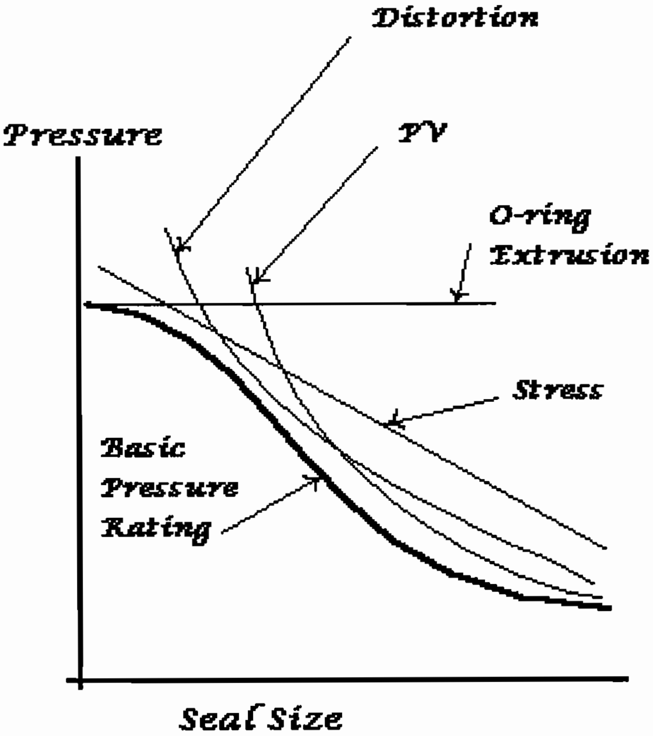

Figure 7. Concept of Basic Pressure Rating.

SUMMARY

Based on the arguments presented herein, it is apparent that a mechanical seal might have as many pressure limits as it has components. Even so, these various limits can be combined into three pressure ratings: static, hydrotest, and dynamic.

The hydrotest rating can be treated as a constant for a given seal design, size and material. The static pressure rating will also vary with temperature. A conservative first estimate for the dynamic pressure rating can be reduced from a Basic Pressure Rating that has been assembled from limits imposed by stress, extrusion, distortion, wear, lubricity, and leakage. However, when high dynamic pressure ratings are required, a more detailed engineering review is necessary. When multiple operating conditions are specified, or when the dynamic and static operating conditions differ significantly, still more engineering analysis is necessary.

Although reducing a dynamic pressure rating from a Basic Pressure Rating is a great convenience, it is not standardized throughout the sealing industry. By offering this topic in the open literature, the author hopes to open dialogues that might lead to standardization.

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APPENDIX

Stresses in Thick Walled Cylinders

The equation for stress in a thick walled cylinder is not particularly complex but the results are not so easy to visualize as the results from Equation (1). Shigley [15] gives the relationship for tangential stress as:

$$\sigma_t = \frac{P_i a^2 - P_o b^2 - a^2 b^2 (P_o - P_i)/r^2}{b^2 - a^2} \qquad \text{(A-1)}$$

The radial stress is given by:

$$\sigma_t = \frac{P_i a^2 - P_o b^2 + a^2 b^2 (P_o - P_i)/r^2}{b^2 - a^2} \qquad \text{(A-2)}$$

Positive stresses indicate tension and negative stresses indicate compression. When the external pressure is zero, the maximum stresses occur at the inner radius, a. When the internal pressure is zero, the maximum stresses occur at the outer radius, b.

Strictly speaking, Equations (A-1) and (A-2) should be used in all cases; however, the thin wall equation is accurate when the wall thickness is less than 1/10 of the diameter. In that case, the radial stress is negligible.

NOMENCLATURE

a	Inside radius of cylinder, in (APPENDIX)
b	Outside radius of cylinder, in (APPENDIX)
b	Geometric balance ratio, dimensionless
C_1	An empirical coefficient used in Equation (2)
C_2	An empirical coefficient used in Equation (2)
D_m	Mean diameter, in
k	Pressure gradient factor, dimensionless
P	Pressure
PV	A parameter combining seal face contact pressure and velocity, psi ft/min
P_i	Internal pressure, psi
P_o	External pressure, psi
P_{sp}	Unit spring load ("spring pressure"), psi
ΔP	Differential pressure, psi
r	Radius of cylinder, in
S_e	Equivalent span of a metal bellows, in
SF	Stability factor
t	Wall thickness, in; bellows thickness, in
V	Velocity at the mean face diameter, ft/min
σ	Stress, lb/in^2
σ_p	Stress in bellows as a result of pressure, lb/in^2
σ_r	Radial stress, lb/in^2
σ_t	Tangential stress, lb/in^2

REFERENCES

1. Schoenherr, K. S., "Design Terminology for Mechanical End Face Seals," Society of Automotive Engineers Transactions, *74*, Paper Number 650301 (1966).

2. Schoenherr, K. S., "Life and Wear of Mechanical Seals," American Society of Metals Wear Conference (1969).

3. Buck, G. S., "A Methodology for Design and Application of Mechanical Seals," American Society of Lubrication Engineers (1979).

4. Lebeck, A. O., *Principles and Design of Mechanical Face Seals*, New York: Wiley-Interscience (1991).

5. Abar, J. W., "Performance Limitations of High Pressure End Face Seals," National Conference on Fluid Power (1971).

6. *Wear Control Handbook*, Eds. M. B. Peterson and W. O. Winer, American Society of Mechanical Engineers (1980).

7. Paxton, R. R., *Manufactured Carbon: A Self-Lubricating Material for Mechanical Devices*, CRC Press (1979).

8. Flitney, R. K. and Nau, B. S., "Seal Survey: Part 1–Rotary Mechanical Face Seals," BHRA Report CR1386 (1976).

9. John Crane Inc., Bulletin No. S-3017, "Type 38B."

10. Datta, A., Gardner, J. F., Derby, J. U., and Casucci, D. P., "Metal Bellows Mechanical Face Seals for High Performance Pump Applications," *Proceedings of the Fifth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1988).

11. API Standard 682, "Shaft Sealing Systems for Centrifugal and Rotary Pumps," First Edition, American Petroleum Institute (1994).

12. *Parker O-ring Handbook*, Parker Seal Group, Parker Hannifin Corporation, Lexington, Kentucky (1991).

13. Gabriel, R P. and Niamathullah, S. K., "Design and Testing of Seals to Meet API 682 Requirements," *Proceedings of the Thirteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas (1996).

14. Massaro, A. J., "The "Mating Pairs" Concept for Mechanical Face Seals," Society of Tribologists and Lubrication Engineers, 42nd Annual Meeting (May 1987).

15. Shigley, J. E., *Mechanical Engineering Design*, McGraw-Hill, 3rd Edition (1977).