THE ROLE OF HYDRAULIC BALANCE IN MECHANICAL PUMP SEALS

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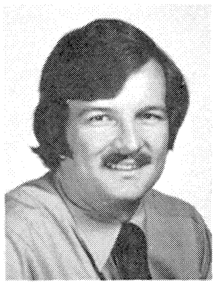

Gordon S. Buck received his B.S.M.E. from Mississippi State University in 1970, and his M.S.M.E. from Louisiana State University in 1978. He has ten years of experience in engineering, research and development, and maintenance for major petrochemical processing companies. At Engineered Repair Services, Mr. Buck is responsible for failure analysis and redesign of machinery. He is a member of ASME, several honor societies, and is a registered professional engineer.

ABSTRACT

The mechanical seal is the least reliable component in centrifugal pumps. Most seals fail prematurely, i.e., before the seal face wears out. One cause of premature failure is face instability when the seal geometric balance ratio is too low. In this study, forces on the seal are reviewed to define seal balance ratio and pressure gradient factor. These terms are arranged to show that balanced seal faces may separate before wearing out. Unbalanced seals are not subject to this type of separation. This potential instability of balanced seals is a significant factor in reducing seal reliability. The limitations for seal face loading show that unbalanced seals are suitable for most services.

INTRODUCTION

In many machines a shaft penetrates a housing which is under a pressure differential relative to the ambient environment. A desirable and sometimes essential design feature of the machine is minimum leakage around the shaft. In a petroleum refinery the most common example of this machine is a centrifugal pump. The two most common devices for restricting leakage around rotating shafts are packing and the end face mechanical seal (often abbreviated "mechanical seal" or simply "seal"), see Figures 1 and 2.

Packing is the older of the two sealing methods and utilizes resilient material arranged around the shaft to form a relatively long axial seal. As shown in Figure 1, an adjustable gland is used to compress the packing rings inside a chamber called the stuffing box. The primary leakage path is between the packing rings and the shaft. A certain amount of leakage is necessary to lubricate and cool the shaft-packing interface. Gland adjustments may be required frequently. Packing in centrifugal pumps has largely been replaced by the mechanical seal because of the higher leakage and maintenance associated with packing.

The mechanical seal is structured of materials that are less resilient than packing; the parts are arranged to effect a short but effective radial sealing path as shown in Figure 2. Because the materials are not resilient, a spring is used to provide resiliency and automatic compensation for wear. The spring and rotating face are driven by a locking collar which is fastened to the shaft. The spring and hydraulic forces are designed to maintain a small gap between the rotating and stationary faces which are the primary leakage path. O-rings provide sealing between the rotating face and shaft and the stationary face and seal plate. These locations are secondary leakage paths. Since rubbing and viscous shear between the faces generates heat, the seal plate contains a port for injection of liquid, the flush, to provide cooling. Mechanical seals come in different configurations, but all have rotating and stationary faces, secondary sealing elements and drive elements.

At first glance a mechanical seal appears to be relatively simple. Leakage is prevented by a rubbing contact between the rotating and stationary faces. In reality, the mechanical seal is a complex combination of thrust bearing, heat generator, heat exchanger and sealing device. The seal may be operated under boundary, mixed or full film lubrication, cavitation and liquid-vapor conditions. The success or failure of the seal depends upon dimensions on the order of micro-inches. Many theories on various aspects of mechanical seal design have been published, but there is presently no concise engineering aid which will guide the user or designer to a reliable mechanical seal.

Because no concise methodology exists for evaluating seal designs, mechanical seal technology appears to be more art than science. Successful seal users have learned that reliable sealing is obtained when a clean lubricating liquid is between the seal faces. This has become well known as a primary requirement for long seal life. In the absence of clean, lubricating liquids, exotic face materials are required. These rugged materials may also improve reliability in less demanding services. Most importantly, to minimize leakage, the seal face gap must be very small and stable with no tendency to increase or "pop open." The objective of this study is to examine the requirements necessary for a stable seal face gap and relate those requirements to existing standards and commercially available seals.

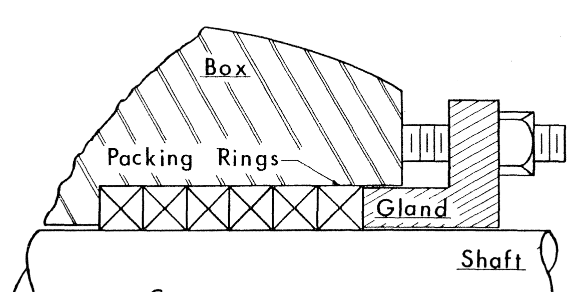

Figure 1. Packing Used to Effect a Shaft Seal.

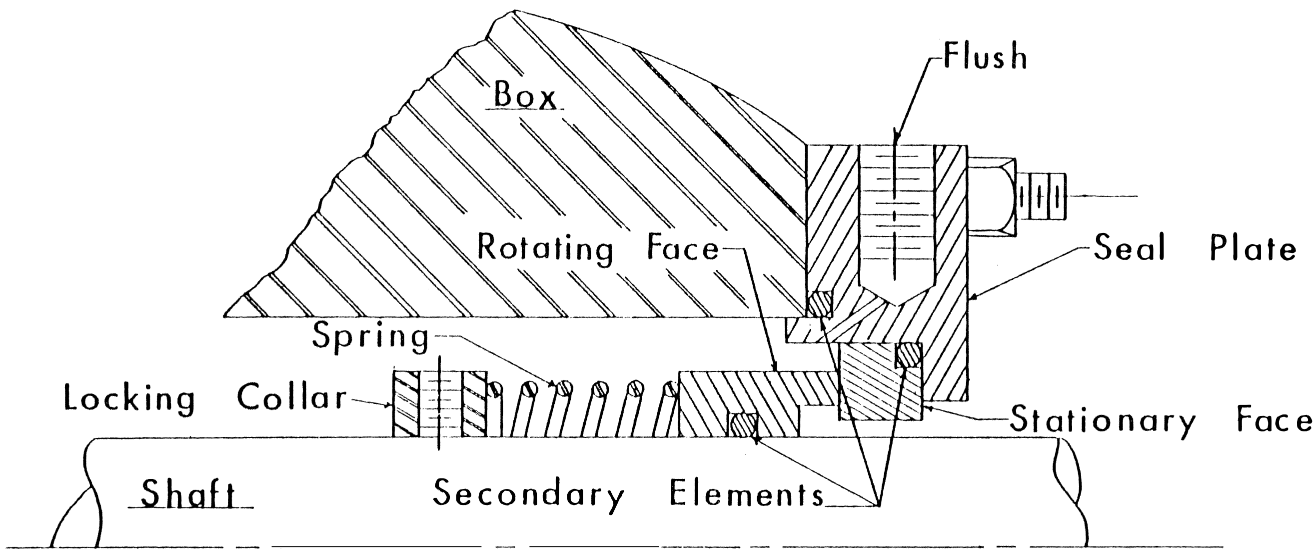

Figure 2. Basic Concepts in a Mechanical Seal Design.

In the author's experience and from discussions with other seal users, the mechanical seal is the least reliable component in centrifugal pumps. According to [1, 5] the user should expect mechanical seal life to exceed two years if the seal is properly selected, installed and operated. The failure mode should be by face wearout. However, in refineries and chemical plants, seal life is frequently less than two years and face wearout hardly ever occurs. When face wearout does occur, the seal is most often the so called "unbalanced" design.

Since mechanical seals are designed to fail by wearout, any failure before wearout is a premature failure. In [1], many types of premature failures are described. Some of these causes of premature failures are given in Table 1. However, even after doing his best to satisfy these criteria, the frustrated seal user must live with premature seal failures. Apparently, there is a basic problem in the application of mechanical seals.

In the author's opinion, a basic cause of premature seal failure is the unnecessary use of the balanced seal. The relationship between seal balance ratio and reliability can best be appreciated by examining the variables affecting seal performance.

DESIGN VARIABLES

Leakage

The simpler theories describe the seal as a controlled leakage device represented by two stationary, non-porous, plane, parallel walls separated by a distance h, the seal face gap. Assuming constant physical properties and laminar, incompressible flow, the leakage rate is proportional to the pressure and may be calculated from Equation (1) by Abar [2].

$$\dot{m}_1 = \frac{\pi h^3 \rho \Delta P}{6\mu \ln(R_1/R_2)} \tag{1}$$

When the sealed fluid is a compressible ideal gas in laminar, isothermal flow the leakage is proportional to the square of the pressure and Equation (2) is used.

$$\dot{m}_v = \frac{\pi h^3 (P_1^2 - P_2^2)}{12 \mu R T \ln(R_1/R_2)} \tag{2}$$

The seal face gap, h, cannot be calculated with the simpler theories, but is of the same magnitude as the surface roughness. In practice the gap is strongly affected by face flatness and face loading. Specifications for commercially available seal faces call for flatness within two helium light bands (22 microinches) but distortions of the faces may easily produce deviations from flatness of up to 10 light bands. Face loads are composed of hydraulic and spring forces. Since loading is not constant, the gap and even flatness may vary.

Forces on the Seal Face

As shown by Equations 1 and 2, the leakage rate and therefore the success of the seal, depend on maintaining a very small seal face gap, h, between the rotating and stationary faces. In order to assure a small gap, the net closing forces acting on the seal must have a positive gradient with respect to the face gap.

Figure 3 shows the rotating face of the seal shown in Figure 2. For this type of seal, the face area, A_f, lies between the outer radius R_1 and the inner radius R_2. The pressure to be sealed, P_1, is at the outer radius, R_1. The pressure to be protected, P_2, is at the inner radius, R_2. For most seals, the pressure to be protected is the atmospheric pressure. The radii and pressures are numbered to indicate that leakage flow goes from point 1 to point 2 and the fluid is at state points 1 and 2.

Any pressure forces on the face area, A_f, tend to open the faces. This opening force, F_o, has three components as shown in Equation 3. The pressure P_2 exerts a force $P_2(A_c - A_f)$ on the area below the seal face. The pressure gradient between the faces exerts an opening force, $2\pi\int_{R_2}^{R_1} P\, r\, dr$ on the face. Since most seals seem to operate with some direct rubbing contact, the average direct contact pressure, P_f, which exerts a force $P_f A_f$ must also be considered.

$$F_o = P_2(A_c - A_f) + P_f A_f + 2\pi\int_{R_2}^{R_1} P\, r\, dr \tag{3}$$

The closing force, F_c, has two components as shown in Equation 4. The sealed pressure, P_1, acts on the end of the seal opposite the face, A_c, to exert a force $P_1 A_c$. For most seals a mechanical spring force, F_{sp}, also acts on A_c to insure a positive closing force at low pressures as well as during assembly.

$$F_c = P_1 A_c + F_{sp} \tag{4}$$

TABLE 1. MODES AND CAUSES OF MECHANICAL FACE SEAL FAILURES (Adapted from Reference 1).

Failure Mode	*Cause*
Primary Seal Faces	
Over-all corrosion	Improper materials
Seal face Distortion	Excessive fluid pressure on seal Swell of confined secondary seal Improper assembly of seal Excessive P_fV value Improper equipment operation
Fracture	Improper assembly Excessive thermal stress Excessive P_fV value
Edge chipping	Excessive shaft deflection Seal face vibration
Uniform adhesive wear	P_fV value too high Failure of axial holding hardware Poor environment
Nonuniform wear	Abrasive contaminants P_fV value too high
Secondary Seals	
Chemical attack	Improper material
Extrusion	Excessive pressure
Mechanical Drive	
Wear	Excessive torque Excessive shaft end play
Seal hang-up	Deposition of seal fluid decomposition

For steady state operation the opening and closing forces are equal, therefore the average direct contact pressure, P_f, may be found by equating Equations 3 and 4.

$$P_f = P_1\left(\frac{A_c}{A_f}\right) - P_2\left(\frac{A_c}{A_f} - 1\right) - \frac{2\pi}{A_f}\int_{R_2}^{R_1} P\,r\,dr \quad (5)$$

If P_f is positive there is some indirect rubbing between the rotating and stationary faces; this is a condition of low leakage. For P_f exactly zero the faces would not be rubbing but the resulting hydrodynamic lubrication conditions would allow increased leakage. For P_f to be negative the opening forces have overcome the closing forces and the seal has failed.

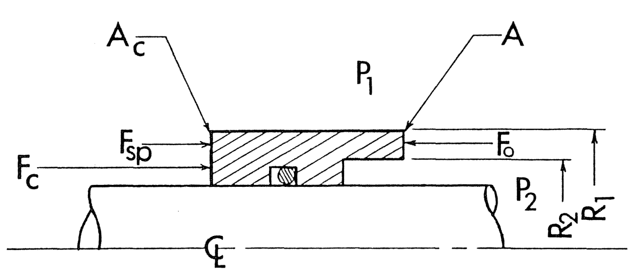

Figure 3. Forces on the Mechanical Seal.

Equation 5 is cumbersome to use but is basic to an understanding of mechanical seal design. The determination of the average direct contact pressure is simplified by defining a pressure gradient factor, k, and a balance ratio, b.

Pressure Gradient

Equation 5 can be made less formidable by replacing the quantity to be integrated with a simple algebraic quantity. By noting that $2\pi\int_{R_2}^{R_1} P\,r\,dr$ is simply the hydraulic force on the seal face, several cases can be examined which may indicate a suitable replacement. If the pressure is constant across the seal face, for example at P_1, then the hydraulic force on the face is P_1A_f. If the pressure is constant at P_2, the hydraulic force is P_2A_f. If the pressure drops linearly from P_1 at R_1 to P_2 at R_2, the hydraulic force is approximately the product of the average pressure and face area, $\frac{1}{2}(P_1+P_2)A_f$. This product may also be written as $[\frac{1}{2}(P_1-P_2) + P_2]A_f$.

If the pressure gradient factor, k, is defined such that

$$[k(P_1-P_2) + P_2]A_f = 2\pi\int_{R_2}^{R_1} P\,r\,dr \quad (6)$$

then $k = 0$ for the fluid pressure at P_2 over the entire face, $k = \frac{1}{2}$ for a linear pressure drop across the face and $k = 1$ for the fluid pressure at P_1 over the entire face. Figure 4 illustrates the relationship between pressure distribution and pressure gradient factor, k. According to Stein [8], for an incompressible fluid $k = \eta/(\eta+1)$ where $\eta = h_1/h_2$, the ratio of inlet to outlet face opening. This concept is also shown in Figure 4 where $k = 0$ for line contact at the seal outside diameter and $k = 1$ for line contact at the inside diameter when the pressure to be sealed is at the outside diameter.

For compressible fluids, Reference 8 shows that the pressure profile between two parallel plates is approximately parabolic and may be expressed as

$$P = P_1\left[1 - \left(1 - \frac{P_2^2}{P_1^2}\right)\frac{\ln(R_1/r)}{\ln(R_1/R_2)}\right]^{1/2} \quad (7)$$

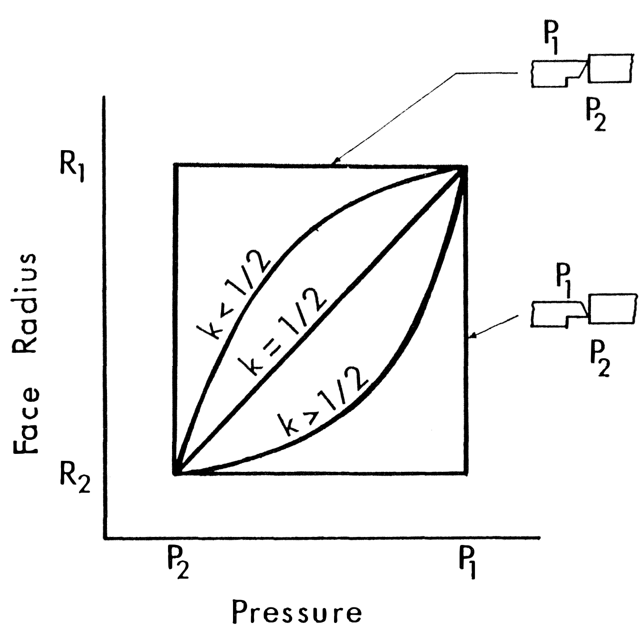

Figure 4. Relationship Between Pressure Gradient Factor and Pressure Profile Across the Seal Face.

Substitution of Equation 7 into 6 yields a limiting value of $k = 2/3$ as P_2/P_1 approaches zero, if supersonic choked flow is neglected. For low ratios of P_1/P_2, the effects of compressibility are small and k is approximately 1/2.

Substitution of Equation 6 into 5 yields

$$P_f = (P_1 - P_2)\left(\frac{A_c}{A_f} - k\right) + \frac{F_{sp}}{A_f} \tag{8}$$

which can be further simplified by defining the geometric balance ratio, b.

When the quantity A_c/A_f is defined as the seal geometric balance, b, Equation 8 is reduced to the frequently used form, Equation 9.

$$P_f = \Delta P(b - k) + P_{sp} \tag{9}$$

The geometry of the seal nose and shaft may be manipulated to obtain $b<1$ or $b>1$ to change face loading. Seals with $b>1$ are called unbalanced or over-balanced seals; seals with $b<1$ are called balanced seals. The phrase "balanced seal" is misleading and only indicates a lower face pressure than a similar "unbalanced" seal by virtue of geometric manipulations. Figure 5 shows a balanced seal.

A seal with $b = 1$ is impractical with a constant diameter shaft because there would be no clearance between the seal inside diameter and shaft. In practice unbalanced seals have balance ratios from 1.1 to 1.4. Balanced seals are used with a stepped shaft and are typically designed for $b = 0.6$ to 0.9. The spring generated face pressure is usually in the range of 10 to 40 psi. Because of the seemingly low value for spring pressure, it is often neglected in seal calculations. This may be an error because in many applications the spring pressure is the same magnitude as the hydraulic pressure.

The unit direct contact face load, Equation 9, must be positive for the faces to be in contact. A zero load would mean hydrodynamic lubrication. A negative load would mean the faces were separating. In order to have minimum leakage, the seal should be designed for a positive face load; however, an increase in face load may also mean an increase in heat generation.

The balance ratio may be calculated from the seal size and face dimensions.

$$b = \frac{R_1^2 - R_b^2}{R_1^2 - R_2^2} \tag{10}$$

For conventional spring type seals, the balance radius is at the secondary sealing element as shown in Figure 5. For metal bellows seals, the same equation for balance ratio may be used providing the balance radius is the mean effective radius. The mean effective radius at low pressure is approximately the mid point of the bellows.

The effect of pressure on balance ratio for metal bellows seals is opposite that of conventional seals. For conventional seals, pressure decreases the face diameters slightly so that balance ratio is decreased. For metal bellows, pressure also decreases the mean effective diameter; the net effect is an increase in balance ratio. Figure 6 shows the representative magnitudes of the effect of pressure on balance ratio for conventional and metal bellows seals.

Face Pressure-Velocity Product

The face pressure-velocity product, P_fV, is commonly used as a rule of thumb factor for describing the limits of materials in rubbing contact. The face pressure is determined for Equation 9 with the pressure gradient factor, k, equal to

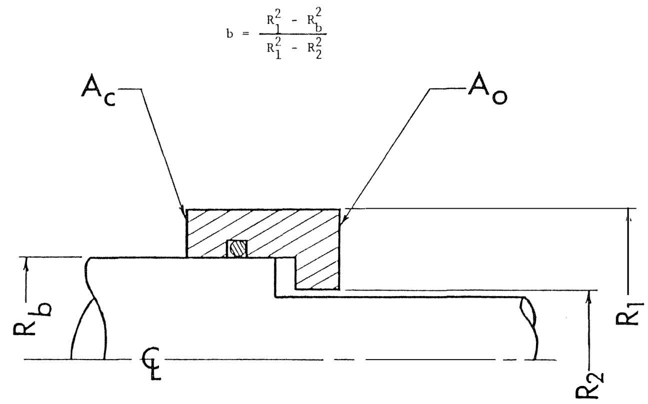

Figure 5. The Balanced Mechanical Seal.

1/2. The velocity V is the average face velocity and is calculated at the mean face diameter. The maximum P_fV value for a material is highly dependent upon the operating conditions. Some examples of limiting values are shown in Table 2.

In a typical refinery pump, a 2″ balanced seal, $b = 0.75$, might seal 100 psig while mounted on a 3600 rpm shaft. The resulting P_fV value is 100,000 psi ft./min. For the same size unbalanced seal, $b = 1.25$, $P_fV = 200,000$ psi ft./min. According to Table 2, either design would be acceptable for carbon versus tungsten carbide face materials, a common combination.

Most seal users select a balanced seal because of its lower P_fV value. In fact, the American Petroleum Institute Standard 610, "Centrifugal Pumps for General Refinery Services", virtually forces the use of balanced seals in Section 24.1:

> *24.1 Hydraulically balanced seals shall be furnished when speeds and pressures are greater than those shown in Table 1. Balanced seals shall also be provided for pumps handling liquids with specific gravities less than 0.65 at stuffing box temperature, regardless of the sealing pressure requirement.*
>
> *API 610, Section 24.1*

TABLE 1. LIMITS FOR UNBALANCED SEALS.

Seal Inside Diameter (Inch)	*Shaft Speed (rpm)*	*Sealing Pressure (psig)*
1/2 to 2	*Up to 1800*	*100*
	1801 to 3600	*50*

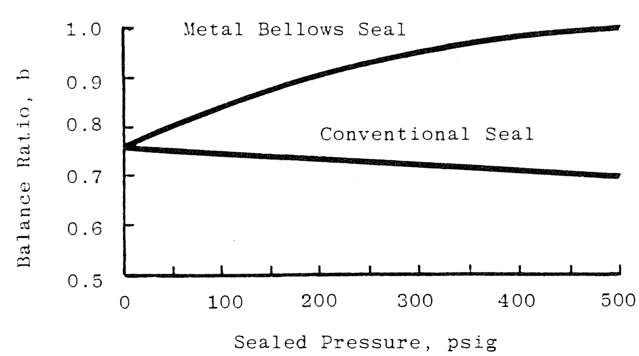

Figure 6. Variation of Balance Ratio with Pressure.

TABLE 2. LIMITING P_fV VALUE.

Material Combinations	Fluid	Maximum P_fV kpsi ft/min.
Tungsten Carbide/Tungsten Carbide	Not specified	200
Silicon Carbide/Silicon Carbide	Not specified	500
Tungsten Carbide/Silicon Carbide	Not specified	600
Tungsten Carbide/Carbon	Not specified	1000
Silicon Carbide/Carbon	Not specified	5000
Ni Resist/Babbited Carbon	125°F water	104
85% Al_2O_3/Babbited Carbon	125°F water	160
Tungsten Carbide/Babbited Carbon	125°F water	372
Tungsten Carbide/Babbited Carbon	80°F water	800
Tungsten Carbide/Carbon	Hydrocarbon	480
Tungsten Carbide/Carbon	Water	480

Over 2 to 4	*Up to 1800*	*50*
	1801 to 3600	*25*

Apparently, the intent of API 610 is to prevent excessive PfV by specifying balanced or unbalanced seals appropriately. There are several problems in this simple approach. For example, P_fV limits vary with seal type, seal face materials, liquid and operating conditions. However, the PfV value only considers pressure, balance ratio, spring forces, shaft speed and seal size. Of these variables, only balance ratio is readily and independently controllable. The net result of P_fV considerations is the virtual exclusive use of seals with a low (less than 80%) balance ratio.

In [4], Buck shows that dimensionless parameters which incorporate the P_fV value are more indicative of seal reliability than P_fV alone. One of the dimensionless parameters is the Stability Factor, SF.

STABILITY

Examination of hundreds of seal failures by the author has revealed that most failures are not caused by wearout. In fact, for many failures the amount of wear is on the order of thousandths of an inch whereas the seal is "designed" for about 1/8 inch wear before failure. One explanation for this premature failure is that the pressure gradient factor, k, becomes greater than the balance ratio, b. According to Equation 9, for $k \geqslant b$, the face contact pressure, P_f, may become zero if the spring is not powerful enough to overcome the pressure forces. The zero face pressure is the condition of increased leakage. If the leakage is unacceptable, the seal has failed.

Since k has a maximum value of 1, Equation 9 may be solved for a maximum ΔP which can be sealed before $P_f = 0$.

$$\Delta P_{max} = \frac{P_{sp}}{1-b} \tag{11}$$

The Stability Factor, SF, is a dimensionless number which shows the degree of stability.

$$SF = \frac{\Delta P_{max}}{P_1 - P_2} \tag{12}$$

An unbalanced seal (b>1) is always stable but balanced seals (b<1) may be unstable at certain pressures. For commercial balanced seals commonly used in refineries, $P_{sp} \sim 25$ psi and b $\sim$ 75%. Therefore ΔP_{max} is often only 100 psi.

Of course balanced seals with b = 75% and P_{sp} = 25 psi do seal pressures of over 100 psig so k is not always unity; however, the concept of stability explains premature seal failure, i.e., failure before the faces wear out.

Equation 9 may also be transposed to determine the required balance ratio for stability at any pressure.

$$b)_{SF=1} = 1 - (\frac{P_{sp}}{\Delta P}) \tag{13}$$

Figure 7 shows $b)_{SF=1}$ for various spring loads.

The spring load varies over the life of the seal. Some seals come with a spring preload; others rely only on the compression setting when installed. In either case, as the seal face wears away, the spring load decreases. For linear spring rates

$$P_{sp} = P_{spi}(1 - \frac{\Delta z_w}{\Delta z_i}) \tag{14}$$

where P_{spi} is the initial or as installed spring load, Δz_i is the total spring compression (including preload) and Δz_w is the length worn away during operation.

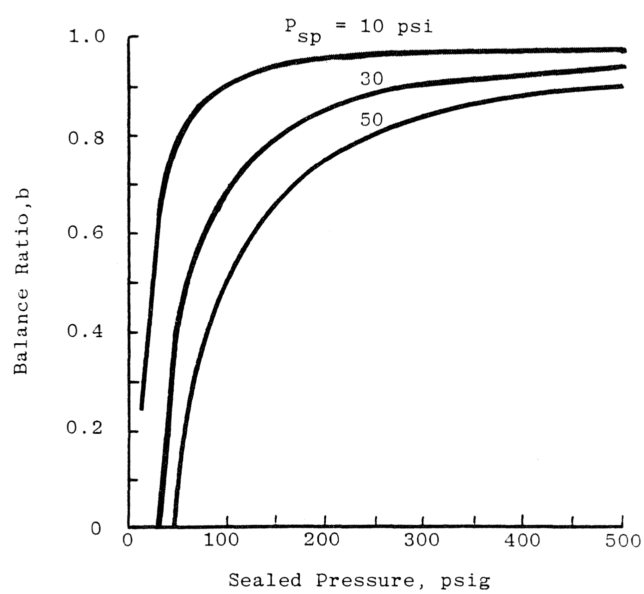

Figure 7. Required Balance Ratio for Stability.

Using Equations 13 and 14, the maximum wear length before failure, $\Delta z_w)_{max}$, may be determined.

$$\Delta z_w)_{max} = \left[1 - \frac{\Delta P}{P_{spi}}(1 - b)\right]\Delta z_i \tag{15}$$

As an example, if the balance ratio is 75% and the initial setting of 1/4″ gives a spring generated face pressure of 30 psi, failure will occur when 0.021″ wears off the seal nose.

Again, it is important to note that the concepts of stability, required balance ratios and wear lengths before failure only represent potential problem areas should the faces deform. However, these concepts do account for premature failures consistent with the author's experience.

CONCLUSIONS AND RECOMMENDATIONS

Although modern materials will permit heavy face loads, mechanical seal users frequently specify designs with reduced face loading. These low balance ratio seals are subject to premature failures due to face instability. Users who are concerned about premature failures should consider unbalanced or high balance ratio seals that are inherently stable.

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NOMENCLATURE

A	Area
b	Seal geometric balance ratio
d	Diameter
D_m	Mean diameter of seal face
F	Force on seal
h	Axial gap between seal faces
k	Pressure gradient factor
$\dot{m}$	Mass flowrate between faces
N	Rotational speed
P	Pressure
ΔP	Pressure difference $= P_1 - P_2$
ΔP_{max}	Maximum pressure rating for a stable seal
r	Radius
R_1	Radius at inlet to face
R_2	Radius at outlet of face
T	Temperature
V	Average face velocity
z	Axial coordinate
Δz_i	Total initial spring compression
Δz_w	Axial length worn from seal nose
$\Delta z_w)_{max}$	Axial wear length at point of instability
μ	Viscosity
ρ	Density

Subscripts

c	Closing
f	Face
i	Initial
l	Liquid
o	Opening
sp	Spring
v	Vapor
1	At flow inlet to face gap
2	At flow exit from face gap