ABSTRACT

This discussion briefly summarizes some of the devices, seal systems, and operating parameters encountered with turbines and compressors, but because of the magnitude of technical information that would be required, the writer has omitted any comments related to some of the more recent developments, such as welded bellows and aspirated circumferential liquid seals. It is hoped that the topic description presented herein serves as a basis for further technical study and, therefore, understanding that will facilitate the utilization of seal theory in applications.

INTRODUCTION

This presentation concerns itself with general sealing devices and systems as well as the basic related theory together with comments on apparent deviations from theory. There are many applications in which mechanical seals or clearance devices of one type or another fail to behave as expected and these applications, at times, shade the overall validity of analytic approaches to sealing problems with uncertainty. The analytic statements are made to expunge the notion that seal design and seal system selection is a secretive, occult art. Understanding sealing devices is rather straightforward, requiring only ordinary common sense and recognition that a fair number of interrelated parameters are involved. Of primary importance is the fact that in sealing devices, behavior is governed by certain small dimensions defining the interface separation of the leakage paths. Exacting considerations must be given to the influences affecting this interface distance. In many conventional engineering applications, mill-inch deflections, whether they be thermally or structurally induced, may have significance, while with automatic, motions of inches or micro-inches may be important. As a consequence, discrimination is required in selecting a seal system and component designs so that adequate precision is engineered into those dimensions which may affect the interface dimensions which controls the leakage.

This discussion will attempt to take a rather straightforward approach to seal theory when such is required and avoid wherever possible duplication of the discussion points to be made by the other authors participating in this tutorial.

LABYRINTH SEALS

The labyrinth is one of the simplest of the clearance type sealing devices, consisting of a series of circumferential strips of metal extending from the body of the shaft housing to form a cascade of annular orifices. Labyrinths have a leakage that is greater than that of clearance bushings, contact seals or film riding seals. Consequently, labyrinths are utilized when a small loss in efficiency can be tolerated, when the conditions are so severe that other types are impractical, or when the geometry is such that more efficient designs cannot be incorporated. They are sometimes a valuable adjunct to the primary seal.

In large gas turbines, labyrinth seals are used in static as well as dynamic applications. The essentially static function occurs where the casing parts must remain unjoined to allow for difference in thermal expansion. At this junction location the labyrinth minimizes leakage. Dynamic labyrinth applications for both turbines and compressors are interstage seals, shroud seals, balance pistons, and end seals.

Advantages of labyrinths are their simplicity, reliability, tolerance to dirt, system adaptability, very low shaft power consumption, material selection flexibility, minimal effect on rotor dynamics, back diffusion reduction, integration of pressure, lack of pressure limitations, and tolerance to gross thermal variations. The major disadvantages are the high leakage, lack of machine efficiency, increased buffering costs, tolerance to ingestion of particulates with the resulting damage to other critical items such as bearings, the possibility of the cavity clogging due to low gas velocities or back diffusion, and the inability to provide a simple seal system that meets OSHA or EPA standards. Because of some of the foregoing disadvantages, many machines are being converted to other types of seals.

The different typical labyrinth types are the straight, the staggered and the stepped, as illustrated by Figures 1, 2, and 3 respectively. Often different types will be incorporated in the same design to form a combination labyrinth such as depicted by Figure 4. The straight and stepped as well as the combination designs facilitate assembly since they permit insertion of the shaft, whereas the staggered type must be assembled around the shaft. Within a specified space the staggered labyrinth permits the least leakage of the four. Unlimited axial movement is allowed with the straight type while the others restrict the axial displacement. Many versions of these basic types exist and they normally can be related to those just listed. One special embodiment is to make the coating surface a spring-loaded segmented sleeve. Should a rapid radial thermal growth cause knives to make a severe rub, the segments can be pushed away while the grooves are being cut.

With respect to labyrinth seal systems, the first three figures illustrate typical direct seals. Flow can be in either direction depending on the pressure conditions. These are generally
must be maintained above the process gas pressure and the outlet pressure. The latter can be greater than, equal to, or less than the atmospheric pressure.

Utilizing concepts similar to the foregoing as well as those that are described in later sections of this discussion, other complex labyrinth systems are sometimes employed. A self-contained aspirating system can be devised by withdrawing system gas from a place along the labyrinth near the inlet end and using it as the motivating fluid in an eductor which then sucks gas from a vent near the atmospheric end. The locations should be placed so as to achieve best overall efficiency.

The primary function of most labyrinths is to restrict the leakage of compressible fluids. High fluid velocities are generated at the throats of the constrictions and the kinetic energy is then dissipated by turbulence in the chamber beyond each throat. Thus, the labyrinth is a device wherein there is a multiple loss of velocity head. With a straight labyrinth there is some velocity carry-over, resulting in a loss of effectiveness, especially if the throttles are closely spaced. To minimize this carry-over, the diameters can be stepped or staggered to cause impingement of the expanding orifice jet on a solid, transverse surface. If one modifies the Egli [1] leakage formula one obtains equation 1 which is sufficiently accurate for most staggered labyrinths or four or more strips.

$$G = 0.75A \left[ \frac{1 - \frac{p_t}{p_0}^2}{n + \ln \frac{n}{p_t/p_0}} \right]^{1/2} \cdot \left[ \frac{p_0}{g V_0} \right]^{1/2}$$

Figure 1. Direct Sealing Straight Labyrinth.

Figure 2. Direct Sealing Staggered Labyrinth.

Figure 3. Direct Sealing Stepped Labyrinth.

Figure 4. Buffered Combination Labyrinth.

Figure 5. Buffered-Vented Straight Labyrinth.
where:
\[ G = \text{leakage, lb per sec} \]
\[ A = \text{leakage area of single throttling, sq ft} \]
\[ P_0 = \text{absolute pressure before the labyrinth, lb per sq ft} \]
\[ V_0 = \text{specific volume before the labyrinth, cu ft per lb} \]
\[ P_n = \text{absolute pressure after the labyrinth, lb per sq ft} \]
\[ n = \text{number of throttlings} \]

We have separated the exponential terms in equation 1 simply to expedite analysis and facilitate a graphical solution. Egli presents the first bracketed term graphically which is especially useful with a small number of throttles. It is worth pointing out that leakage is approximately inversely proportional to the square root of the number of knives. For example, if the leakage of a five touch-point labyrinth must be cut in half, one would have to provide 20 such restrictions. Longer labyrinths, of course, require longer shaft length with consequent lowering of critical speed. With many machines this can be an important design factor and could force consideration of an alternate seal type. If straight labyrinths are being considered, multiply the result obtained by equation 1 by a factor of 1.2 to 1.1 to allow for the carry-over effect. Since the last throttle causes the greatest pressure breakdown, the stepped labyrinth is more efficient if the flow is towards the smallest diameter. With flow toward the largest diameter, the efficiency might be equivalent to that of a straight labyrinth operating on a radius between the largest and the smallest but somewhat near the largest.

Labyrinth clearances must allow for bearing clearance, shaft vibrations, deflections, thermal growth of the shaft and similar factors; otherwise, the touch-points will rub and the machine will make its own clearances, often with attendant damage.

When labyrinths must be used and leakage requirements prohibit adequate running clearances, tighter labyrinths can be devised with shaft knives inside closely fitted sleeves which melt or wear rapidly, such as babbitt, soft carbon, phenolic compositions, abradable fibermetal, metal honeycomb structures, etc. With these, grooves can be cut in the sleeve without serious wear or mushrooming of knives. The V-shaped cut in the sleeve does not greatly increase the effective clearance area of the labyrinth. Obviously, damage will result if the labyrinth strips are imbedded in the grooves while large shaft travel occurs. Therefore, the shaft should have little axial travel at the seal or else axial travel should have a regular relation to radial motion so that any ramp type cut made during run-in will be duplicated during subsequent operation.

Most problems of labyrinths can be attributed to misapplication, poor design, failure of a critical machine component such as a bearing, excessive machine vibration, poor handling, dirty environment or faulty assembly. Many times, misapplication or faulty labyrinth design can alter the basic turbo-machinery performance, as for example in an expander where excessive external labyrinth leakage prevents adequate internal cooling. A severe rub with a labyrinth can cause excessive heating, severe machine vibration and even bowing of the machine shaft due to uneven thermal effects. Almost any rub will cause mushrooming of the touch points and some growing or crazing of the coating surface. A light rub will not necessarily cause any extensive damage and, often, the labyrinth can be repaired by simply deburring the knives and smoothing the grooves in the coating surface.

A device which has close structural resemblance to the labyrinth but operates in an entirely different principle is the windback, as shown by Figure 6. When oil at a low rate enters a confined interspace between a shaft with moderately high surface speed and a housing bore, it is carried continuously around the bore by the windage of the shaft. This effect is utilized by a screw thread device which winds the oil into an internal drain for return to the system. This device can be very effective in restraining leakage of oil splash from unpressurized sumps and gear cases. There must be no outward blow-through of air and gas. Thread depth and thickness, pitch, clearance and length as well as shape and location of drain outlet have a pronounced effect on oil capacity. Windbacks are also used as adjuncts to other types of seals, as shown by Figure 7. With circumferential seals they can be used to keep oil splash from reaching the seal carbons when coking problems exist. In oil buffered seals for compressors they are used to direct the small internal leakage into a pressurized drain, effecting practically complete recovery of the leakage.

Windback structures are extremely simple, clearances about the shaft are ample and the device has high reliability. When shaft speeds extend into the low regions where windage effects are inadequate for effective operation, augmentation of windage can be achieved by special proprietary configurations of the shaft surface.

![Figure 6. Windback.](image-url)

![Figure 7. Circumferential Seal with Windback.](image-url)
Figure 8 illustrates a face seal assembly which has both a windback with windage screen and a labyrinth appended to the static casing member. The windback function is to prohibit the oil from fouling the coexisting transverse primary sealing surface while the labyrinth minimizes heat transfer and serves as an emergency seal in the event of a gross seal failure.

![Face Seal with Windback and Labyrinth](image)

**Figure 8.** Face Seal with Windback and Labyrinth.

RESTRICTIVE RING SEALS

Restrictive ring gas seals such as characterized by API 617 [2] and, as shown by Figure 9, are essentially a series of sleeves, the bores of which form a small clearance about a shaft, leakage being limited by flow resistance in the restricted area. Leakage is inhibited mainly by laminar or turbulent friction, but velocity head loss is significant if the clearance is large. The majority of restrictive ring seals are of the floating type rather than the fixed because normally the latter permits a higher leakage rate. Floating rings may be of the segmented type, such as shown by Figure 9, or of the rigid type, such as shown by Figure 10. Even though we have illustrated the banded design, in certain applications a solid compatible material design would be suitable.

![Buffered Rigid Restrictive Ring Assembly](image)

**Figure 10.** Buffered Rigid Restrictive Ring Assembly.

One type of segmented floating ring seal is a proprietary design wherein the carbon segments are held in position with a segmented retainer, such as shown by Figure 9. This assembly behaves like a rigid, lightly loaded carbon ring of rugged construction. The method of supporting the assembly is illustrated by Figure 9. The segmented carbon ring has a conical outer surface engaging with the tapered bore of the segmented metal retainer. Under the action of the garter spring, the retainer segments slide down the taper until stopped by the side walls. The restraining force from the walls provides friction to suspend the ring assembly out of contact with the shaft. The small internal bend on the retainer keeps it from sliding off the carbon when there is no restraint from the walls of the housing. Installation of the ring in the annulus is simple.

The conventional plain carbon seal ring is subject to upstream pressure on one transverse face and to a gradient from upstream pressure to downstream pressure on the other transverse face. With a significant pressure differential, the ring can be plastered tightly against the housing wall so that a large transverse force is required to move it. Should the shaft touch the ring, high rubbing loads can result before the ring is pushed away. With a flexible shaft, serious whirl and vibration problems can result.

The transverse face of the proprietary ring is pressure relieved, as shown by Figure 9, to minimize the side loading of the ring on the housing wall. Because of the low loading, the ring can be moved easily should the shaft make contact with it. The degree to which the pressure loading is reduced by the proprietary ring is illustrated in Figure 11.

The left side of Figure 11 shows the cross section of a plain garter spring ring along with a diagram showing the pressure on the transverse faces of the ring. A similar cross section and pressure diagram for the proprietary ring is shown to the right of Figure 11. For both rings, the higher pressure is assumed to be on the left side. For the plain ring, the left face is subjected to the high pressure over its entirety as represented on the pressure diagram by the line a-b. The right face of the ring is in contact with the housing and is subjected to the high pressure at its outer diameter, the pressure diminishing to the downstream pressure at the inner diameter of the housing wall. The pressure along the contact face will follow a gradient such as that represented on the diagram by the line a-c. The shaded area between lines a-b and a-c, representing the difference in pressure on the upstream and downstream faces, corresponds to the pressure loading on the downstream wall of the housing. For a plain carbon ring on a shaft of 4 inch diameter, with a 100 psi pressure drop, the side loading would exceed 300 pounds.
by hydrodynamic forces generated in the fluid within the clearance by shaft rotation. The bushing clearance should be sufficient to avoid continuous agitation by shaft motions resulting from run-out, vibration and radial play in the shaft bearings. In general, diametral clearances for floating bushings will average about one mil-inch per inch of shaft diameter, but this is modified in accord with the design of other elements in the machine, such as shaft bearings, shaft stiffness, possible angular misalignment, etc.

Although the bushing seal is a simple structure, good design may involve complex requirements. The bushing length is determined by leakage limits, available space, need for hydrodynamic support, angular misalignment, etc. Long bushings provide better hydrodynamic support, but angular misalignment may restrict length and require use of several short bushings in series. High pressure differentials may be divided among several bushings in series to avoid excessive pressure loading on the transverse face on which the bushing must slide to adjust to the shaft. In general, bushing lengths should not greatly exceed the diameter. With viscous liquids, power consumption at high speeds demands very short bushings, since with fixed leakage, short, closer clearance bushings have less drag than long ones with large clearances. The collateral problems of heating effects and cooling requirements cannot be ignored.

In an effort to give you an appreciation of fluid film seal leakage and at the same time demonstrate how straightforward seal theory is, we briefly show how the leakage problem can be treated with an analysis which is based on the methods of civil engineering hydraulics rather than on the mathematical treatment of modern fluid mechanics.

In the region of laminar flow, the problem of flow between plane parallel walls is easily handled by equation 2.

\[ q = \frac{h^3}{12\mu} \frac{dp}{dl} \]  

(2)

wherein:

- \( q \) = leakage rate per unit width
- \( h \) = separation distance of the walls
- \( \mu \) = absolute viscosity
- \( p \) = fluid pressure
- \( l \) = axial distance

With an incompressible fluid, a concentric fluid film seal will have a total leakage rate as defined by equation 3.

\[ Q = 2\pi R \frac{h^3}{12\mu} \frac{\Delta P}{L} \]  

(3)

wherein:

- \( Q \) = total flow
- \( R \) = mean radius
- \( h \) = radial clearance
- \( \mu \) = absolute viscosity
- \( \Delta P \) = pressure difference
- \( L \) = length

By starting with the Darcy equation for friction for turbulent flow of incompressible fluids in pipes, we can derive the valid equation 4 for similar flow between parallel walls.
\[ P_f = \frac{1.33}{\left(\frac{Vh \mu}{\rho} \right)^{1/4}} \cdot \frac{L}{h} \cdot \rho \cdot \frac{V^2}{2} \]  

wherein,

- \( P_f \) = pressure drop due to friction
- \( L \) = length
- \( h \) = radial clearance
- \( \rho \) = density
- \( V \) = mean velocity
- \( \mu \) = absolute viscosity

Multiplying both sides of equations 2 and 4 by the terms \( h^3/\nu^3 \), we obtain the non-dimensional equations 5 and 6.

\[ \frac{h^3}{\nu^3} \frac{P_f}{L} = 12 R_e \]  

\[ \frac{h^3}{\nu^3} \frac{P_f}{L} = 0.0665 R_e^{7/4} \]

wherein the new terms are:

- \( \nu \) = kinematic viscosity
- \( R_e \) = Reynolds’s number

\[ R_e = \frac{Vh}{\nu} = \frac{\nu}{\nu} \]

Stein [3] has generated a convenient leakage chart, as shown by Figure 12. This figure shows the calculated curves for concentric and for fully eccentric bushings in both the laminar and the turbulent regions. Also plotted on the figure are experimental curves from the data of Cornish [4] [5], representing numerous tests, plotted points from Caldwell [6], and experiments by the writer’s firm. The foregoing evaluates only the friction loss terms and not the losses or the additional friction loss associated with development of the velocity profile in the transition after entry. Stein [3] utilizes these additional terms for more precise calculations.

Many of the modern high speed, high pressure compressors have as their primary seal a buffering liquid which is contained in its cavity by two fluid film seals, as shown by Figure 13. The sub-type classification of this buffered bushing assem-
conventional liquid face seal together with a free-floating bushing of the semibalanced type. This element could have a pressure breakdown sleeve, as shown in Figure 18, or the rotating equipment's sleeve bearing.

Essentially the Figure 19 primary element is the face seal which consists of two flat surfaced rings, one rotating with the shaft, the other nonrotating and carried by the housing. One must have freedom to contact the other; continuously, despite axial shaft movement, wobble and other deviations from pure rotational movement. The axially free member requires means for sealing it to its carrier. This secondary seal must be a resilient or elastic part which will not prevent axial freedom. At low
shaft speeds, either the static or the rotating member may have axial freedoms. With high speeds and large diameters, the shaft member should be rigidly attached and the axially free member should be the non-rotating one, otherwise, centrifugal forces on rotating, freely suspended parts would introduce design complications.

To achieve long wear life, the rubbing load must be minimal, yet adequate to overbalance axial friction, inertia, and film pressure forces. If the axial force due to fluid pressure on the rubbing face of the free member is not balanced by that on the opposite faces of the free member, the fluid pressure loading will preclude maintenance of constant, low rubbing load. Analytically this can be demonstrated by integrating equation 2 for radial incompressible flow across the nose of a face seal to give the pressure at any radius within the nose, as shown by equation 7.

\[
P_i - P = \pm \frac{12\mu Q}{2\pi h^3} \ln \frac{r}{R_i} \tag{7}
\]

wherein:
- \( P \) = pressure at radius \( r \)
- \( r \) = any radius within limits of nose
- \( P_i \) = inlet pressure
- \( \mu \) = absolute viscosity
- \( h \) = separation of the interface
- \( Q \) = total flow rate

Graphically the total axial pressure force of the fluid in the nose film is represented by area \( a\ b\ c\ d\ e \), as shown by Figure 20. To make a balanced pressure seal, the total pressure force represented by this area is balanced by the pressure force on the opposite side of the seal represented in the diagram by \( f\ g\ h\ i\ j\ k\ l\ f \). Actually, this requires making \( b\ c\ d\ b \) equal to \( g\ h\ i\ j\ g \). When centrifugal effects and curvature of the nose are considered, the total force difference \( \Delta F \) which is represented by \( b\ c\ d\ b \) is given by equation 8.

\[
\Delta F = \pi(R_1^2 - R_0^2) \left[ \frac{\Delta P}{2} \left( \frac{2R_1^2}{R_1^2 - R_0^2} - \frac{1}{\ln R_i/R_o} \right) - \frac{3}{40} \omega^2 \rho \left( \frac{R_1^2 + R_0^2}{\ln R_i/R_o} \right) \right] \tag{8}
\]

wherein:
- \( \Delta F \) = total force difference in the interface
- \( \Delta P \) = pressure differential \( (P_i - P_o) \)
- \( P_i \) = inlet pressure
- \( P_o \) = outlet pressure
- \( R_i \) = inlet radius
- \( R_o \) = outlet radius
- \( \rho \) = density
- \( \omega \) = angular velocity of the rotating boundary

It might be of interest to note that the first parenthesis in the bracket is derived from the curvature component, while the remainder of the bracket is the modification for rotation. More precise analysis must consider deviations from geometry and, in some cases, phase change in the fluid. The force to overcome friction, inertia and other restraints and to maintain contact is supplied by mechanical springing. Since some of the friction and other forces can increase with pressure, the seal face is often designed with a pressure bias to augment the spring loading to a small degree as pressure is increased. This allows use of lighter springs and gains the advantage of less contact loading during operation at lower pressures.

With high rotating speeds, extra consideration should be given to centrifugal effects in the seal face and in the barrier space. It should be clear that a seal which is designed to operate with oil in the interface, and does so successfully, may not be able to operate in the absence of oil. With this in mind, it becomes necessary that operators recognize that the oil pressure which exists at the interface of a seal differs from the oil pressure measured at some point in the seal chamber or in the entering or leaving oil piping. A typical seal has had this centrifugal effect calculated, tested and plotted, as shown by Figure 21. It is imperative that the oil pressure be maintained above that defined by the parabola.

**DRI G AS FACE SEALS**

The grooved face design for dry gas seal rings, as shown by Figure 22, was devised more than thirty years ago to minimize the change in interface pressure caused by thermal and structural deflections of the sealing members. The grooved face, when of proper design, permits pressure balance to be very
nearly maintained despite these deflections, which cannot be entirely eliminated.

By curtailing the pressure changes at the rubbing interface and designing the secondary seal and rotation lock structures to have low and determinable friction, only gentle axial force need be imposed by suitable springs to maintain intimate contact of rubbing surfaces despite reasonable out-of-squareness, vibration and axial movement of the shaft. Low face loadings are absolutely essential when sealing gas which has low cooling capacity and, unlike liquids, no reserve of latent heat to quench transient hot spots.

With the low, positive rubbing load which this design allows, these seals operate dry in the regime of gliding wear at high speeds and at significant pressure differentials. Thus, they are elegant solutions to many gas and vapor sealing problems where low rates of leakage can be tolerated. Depending on conditions, leakage rates of 25 to 100 cc. per minute of gas (SPT) per inch of diameter per psi above atmospheric are obtained for outleakage and less, of course, for inleakage to sub-atmospheric applications. Considerably lower leakages are possible where conditions permit use of special secondary seal designs.

Unlike single nose face designs, in which pressure loading is highly sensitive to deflections, the grooved face performs most satisfactorily when completely un lubricated and, consequently, oil jets or other cooling means and oil supply and scavenge systems are eliminated. Where the moderately low gas leakage can be tolerated, buffering and leak-off systems can be avoided. Therefore, in many applications, dry gas seals can reduce auxiliary equipment, conserve shaft length and result in simplification and cost reduction of the machine. The controlled, low face loading allows dry operation with very extended life, ranging from 25,000 to 50,000 hours of actual running at rubbing speeds of 12,900 to 18,000 feet per minute and ambient temperatures of 400 to 500°F. At these conditions, the seal ring and the mating ring will have gross temperatures only 60 to 100°F above the ambient.

When a more accurate isothermal flow of gas between parallel plates must be calculated, equation 2 can be integrated by substituting mass flow times specific volume for volumetric rate of flow, and introducing the proper relation between specific volume and pressure to agree with the actual flow process. The flow rate in weight units per unit width is given by equation 9.

\[ m = \frac{h^3}{24\mu L} \cdot \frac{P_1^2 - P_o^2}{RT} \]  

(9)

wherein

- \( m \) = flow rate in weight units per unit width
- \( h \) = radial clearance
- \( \mu \) = absolute viscosity
- \( L \) = length
- \( P_1 \) = inlet pressure (absolute)
- \( P_o \) = outlet pressure (absolute)
- \( R \) = gas constant
- \( T \) = temperature (absolute)

For a gas seal nose dam, the total interface force difference per unit width may be found by integrating the pressure along the parallel walls and eventually obtaining equation 10.

\[ \Delta F = (P_1 - P_o) \cdot \frac{L}{3} \cdot (1 + \frac{1}{P_o}) \cdot \frac{1}{1 + \frac{P_o}{P_1}} \]  

(10)

wherein

- \( \Delta F \) = total interface force difference per unit width
- \( P_1 \) = inlet pressure (absolute)
- \( P_o \) = outlet pressure (absolute)
- \( L \) = length

When the boundaries of the interface converge or diverge for gas flow, equation 10 is not valid but can be given the form of equation 11.

\[ \Delta F = (P_1 - P_o) \cdot L \cdot \lambda \]  

(11)

wherein the new term is

- \( \lambda \) = balance modulus which can be obtained from Figure 23.
inertia which requires loading to make it follow motions of the collar. The seal ring structure is the optimum compromise between these requirements.

CIRCUMFERENTIAL CONTACT SEALS

The circumferential seal, which is shown by Figure 24 and the exploded view of Figure 25, is a bore rubbing device, highly effective as a gas seal and adaptable to many services. The sealing element is a segmented carbon ring held in an annulus of small cross section and pin-locked against rotation. Sealing is affected by rubbing contact with the shaft runner and by mating of the transverse face of the seal ring with the downstream wall of the annulus. To close off joint leakage, the seal ring is encircled by a cover ring and is backed up by an additional ring, both segmented with joints staggered with respect to those of the seal ring. Each of the rings has three or more segments. Alternate treatment is use of tongue-and-socket or equivalent detail at the seal ring joints, as shown by Figure 26. Garter springs of Inconel X or material of similar quality furnish light radial loading and there are axial springs to hold the assembly against the downstream side of the annulus.

The bore of the seal ring is pressure relieved by vent grooving to minimize pressure load on the rubbing surface and relief grooves on the transverse face reduce load and friction which would restrain radial freedom. Although not pressure balanced, the design has performed in aircraft engines at simultaneous conditions of 90 psi differential, 700°F and rubbing speed of 17,000 feet per minute. Usually, maximum pressure differentials and rubbing speeds are 25% to 30% below these values.

To utilize the advantage of the grooved face, the foregoing analysis and quality control verification must be made. In addition, the design of the entire seal must be consistent with the low loading on the seal ring. Therefore, suspension of the seal ring is as restraint-free as possible and all friction forces are reduced to the lowest practical limit. The best general-purpose secondary seal is a precision piston ring designed and processed to have low, uniform loading in the seal ring counterbore. Its transverse land is lapped flat to mate tightly against the side wall of the piston ring groove. The sealing surfaces of the piston ring are pressure relieved to minimize the pressure loading and the resulting friction.

Although, when compared with single nose seals, the grooved face is relatively insensitive to small deflections of the sealing members, the primary sealing members must not deform unduly from pressure and temperature and the mating collar should not be distorted by its clamping. To provide strength and to amend its low coefficient of thermal expansion, the carbon seal ring is contained by interference fit in a steel retainer so that it expands equally with the carrier and preserves the clearance. The carbon and the retainer are proportioned to avoid thermal twist with change in temperature.

The collar should be as rigid as space and weight will allow. The seal ring rigidity is governed by two considerations: maximum rigidity to reduce deflections and low mass to reduce inertia which requires loading to make it follow motions of the collar. The seal ring structure is the optimum compromise between these requirements.

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The circumferential seal, which is shown by Figure 24 and the exploded view of Figure 25, is a bore rubbing device, highly effective as a gas seal and adaptable to many services. The sealing element is a segmented carbon ring held in an annulus of small cross section and pin-locked against rotation. Sealing is affected by rubbing contact with the shaft runner and by mating of the transverse face of the seal ring with the downstream wall of the annulus. To close off joint leakage, the seal ring is encircled by a cover ring and is backed up by an additional ring, both segmented with joints staggered with respect to those of the seal ring. Each of the rings has three or more segments. Alternate treatment is use of tongue-and-socket or equivalent detail at the seal ring joints, as shown by Figure 26. Garter springs of Inconel X or material of similar quality furnish light radial loading and there are axial springs to hold the assembly against the downstream side of the annulus.

The bore of the seal ring is pressure relieved by vent grooving to minimize pressure load on the rubbing surface and relief grooves on the transverse face reduce load and friction which would restrain radial freedom. Although not pressure balanced, the design has performed in aircraft engines at simultaneous conditions of 90 psi differential, 700°F and rubbing speed of 17,000 feet per minute. Usually, maximum pressure differentials and rubbing speeds are 25% to 30% below these values.
The seal and cover rings have precisely the same axial thickness so that the back-up ring will engage both at each joint. Seal ring segments, where they adjoin, have equal radial height so that both engage the cover ring. These requirements are satisfied in process, and all segments of seal and cover ring assemblies are scribed for orientation as matched sets. The back-up ring segments, having only flatness and bore diameter requirements, need no indexing. With the three-ring carbon seal, two garter springs are required, one on the seal and cover ring combination, the other on the back-up ring. The latter adds about 20% to the rubbing load at 60 psi differential. For this reason, as well as conservation of weight and space, the single ring design is advantageous.

The single ring as shown by Figure 26 has a triangular tongue on each segment and a corresponding socket at the opposite end. The areuate surfaces of tongue and socket are evolved as true cones, carefully fitted in process. Substantial cross sections are used for the tongue-and-socket detail and, to provide flexibility of the segment, the outer diameter of the carbon is usually scalloped to reduce the section, leaving pedestals for supporting the garter spring. A circumferential groove behind the carbon engages a lip on a thin plate washer to limit the intrusion of the segments into the bore during assembly of the turbomachine.

Runners for circumferential seals are normally made as replaceable shaft sleeves with carefully finished wearing surfaces. The wear surface is usually flame sprayed with ceramics or carbides or hard chrome plated. The runner is a component part of the seal and its truth of geometry is as important as that of the seal itself. Therefore, it must be sealed to the shaft so that there is no leakage under the runner; its faces must be
truly flat; it must be clamped to a flat shaft shoulder with no distortion; the bore and the wearing surfaces must be closely concentric and square with the end surfaces, and the outer surface must be free of waviness, so that chucking and multi-jawed mandrels must not be used in its production. The wearing surface when in operation must remain a true cylinder, without taper. After finish grinding, the wear surface should be polished sufficiently to remove fine grinding burrs and leave small flats at the ridges. These requirements are obvious and comprehensible and should not be ignored.

The length of the wearing surfaces must accommodate the shaft travel without riding off of the seal carbons. There should be means for removing the runner and there must be an entry ramp so that the runner can be inserted into the bore of the seal at assembly without danger of fracturing the carbons. This ramp should have a radial height of approximately one-tenth inch and an angle with the shaft axis of 15° to 30°.

Since circumferential seals are not pressure balanced, the prime consideration for the runner is heat rejection. With a pressure differential of 20 to 25 psi for single ring seals and 10 to 15 psi for three ring seals, heat rejection will be about the same as that for lightly loaded face seals, but at moderate and high pressures, the ability of the runner to carry off frictional heat becomes paramount, not only to protect the rubbing surfaces from excessive wear and damage but also to prevent excessive thermal deflection and resultant leakage.

Runners should be made of materials with high thermal conductivity, that is, low alloy steels are superior to high alloys. Compatibility of thermal expansion between runner and shaft materials should also be considered. Thin runner sections should be avoided and heat sinks, such as oil jets, should be located as near the wear track as feasible.

The circumferential seal is essentially a unidirectional gas seal. The carbons are lightly sprung to minimize wear and significant pressure reversal will lift them from their coacting surfaces. Oil in the rubbing interface of conventional circumferential seals will induce surf-boarding and result in oil leakage. Therefore, as with other types of shaft seals, pressurizing gas can be brought to a barrier space behind the seal. There it is contained in a labyrinth-restricted chamber at a positive differential above the adjacent pressure for all operating conditions.

The tandem circumferential seal, as shown in Figure 27, is an alternate to this arrangement which saves gas wastage, conserves space on the shaft and saves cost. Furthermore, since it requires little gas flow, pressure control is more determinate than with labyrinth systems having poor clearance control.

The tandem seal is a double seal in a single housing with provisions for pressurization between the sealing elements. The casing is made in two parts, each with a transverse mating face. Gas is brought from the seal flange to the housing chamber by drill holes or welded blisters or by bored openings with casings evolved from investment castings.

Both the seal casing and the carbons are finished with precision. The mating face in the casing and the bolting flange are flat and parallel and the seal ring in the casing will mate simultaneously at its transverse and bore surfaces when the seal is mounted square with the runner axis. Leakage rates less than 0.065 SCFM per diametral inch will result in the static tests made on every seal and spare carbon set. Dynamic testing proves that running leakage equals or better this rate, provided that geometry is maintained.

The turbinomachine's mounting flange must preserve the precision built into the seal parts. Taper, waviness, and structural and thermal deflections will make wasted effort of careful quality control of the seal components. High flange joint and bolt hole leakage can be prevented by control of flange geometry, suitable gasketing and care in fastener design. Metallic O-rings and other highly loaded static seals must not deflect the seal carrier.

FILM RIDING TYPE FACE SEALS

Seals operating with continuous micro-separation of the sealing surfaces can overstep the limitations of rubbing seals with respect to pressure, temperature, velocity and wearing compatibility of materials and, at the same time, restrict leakage rates to small fractions of those attainable with clearance seals. The merit of any seal using separated faces depends upon the response and the stability of the control mechanism, since the separation must remain constant despite shaft motion and wobble of the mating ring and there must not be vibrations or chaotic motions which would cause the relatively moving parts to come into rubbing contact.

Essentially, seals of this nature utilize various hydrostatic and hydrodynamic bearing principles to govern and stabilize the separation. The face design creates a servo-device of such nature that minute changes in separation result in large changes in the pressure profile in the interface and generate high forces to restore the desired separation distance. Thus, in effect, the seal ring rides on a thin film of fluid, acting as a very stiff hydraulic or aerodynamic spring.

This fluid spring is counterbalanced by the pressure load on a constant area on the opposite side of the seal ring and this pressure load is in equilibrium with that at the sealing face when the separation has a predetermined value. The face design generates a large increase in interface pressure with decrease in separation and reduction in the interface pressure as the separation becomes greater. Therefore, any small excursion in either direction gives rise to immediate restoration, as shown by Figure 28.

Separation distances used in design range from 100 to 500 micro-inches, depending on space limitations and magnitudes and variations of pressure, temperature, speed and other parameters. Cleanliness and nature of contamination are obvious considerations.

Two general types of design principles are employed: externally pressurized hydrostatic systems similar to conven-
tional hydrostatic thrust bearings; and autogenous systems wherein the fluid to be sealed is the activating medium and external pressurizing is not necessary. The range of pressure of the fluid to be sealed as well as its nature with respect to hazard, contamination or temperature, will govern choice between external or autogenous pressurization.

The general construction of seals operating with micro-separation does not depart greatly from that of rubbing seals; however, the sealing faces are made large to obtain low leakage and high restoring forces and the seal ring section can be made very rigid because the inertia forces required for following wobble are of much lower magnitude than the servo forces. For the same reason, the friction of the secondary seal is less critical than with rubbing seals.

These seals are useful with both liquids and gases in medium and high pressure regions. Their behavior with liquids and gases is not entirely identical because of the differences in fluid proprieties. Liquids are excellent damping media because of incompressibility and high viscosity. Vibrations and oscillations correspond to rapid increase and decrease of the interface opening. This must be accompanied by corresponding outflow and inflow of fluid in the minute interspace. With liquids, the viscous friction accompanying such motion is of tremendous magnitude and stability problems are practically non-existent.

Gases do not exhibit such restrained behavior. Because their volume changes with pressure, they have less stiffness as fluid springs than do liquids and their lower viscosity further reduces their damping characteristics. Thus, careful attention must be given to capacity effects in separated seals used for gas, and attention must be directed to sensitivity and stability. The low order of heat capacity of gases makes safeguarding against transient hard rubbing much more important with gas seals than with liquid seals.

These more complex design requirements of separated gas seals are not of such nature that they cannot be overcome or circumvented by careful engineering. Gas seals have been constructed and have operated with separated faces in pressure regions from 60 psi to 1290 psi and in ambients of from 400°F to 1050°F. At the 400 psi pressure, seals have been operated in air with surface speeds of 18,000 feet per minute.

The seal shown in Figure 29 is of the externally pressurized thrust bearing type. This device is a seal for hot gas in a turbine engine. Pressurizing air, cooled by passage through a fuel heat exchanger, enters the region behind the seal ring bounded by the two piston ring secondary seals. The cooler barrier air is conducted to the face grooving through three orifice cascades wherein the pressure loss increases with flow rate. Thus, increase in interface opening, which would be accompanied by increase in leakage rate, results in increased loss through the orifices and lower pressure in the face groove. Any closing of the separation reduces the leakage, decreases the orifice loss, increases the groove pressure and restores the equilibrium separation. The seal face is carbon and the mating ring has a flame-coated ceramic surfaced. Carbon is used because rubbing contact occurs during start-up, idling and stopping of the engine.

It will be noted that the seal illustrated by Figure 30 is not externally pressurized but has an autogenous servo, such as

![Figure 29. Externally Pressurized Film Riding Face Seal Assembly.](image)

![Figure 30. Tapered Face Film Riding Face Seal Assembly.](image)
that previously described and defined by Figure 28. The pressure on the seal ring at the interface is opposed by the pressure on the rear surfaces. The face is fed only from the bore, flows past the tapered transverse inlet section to a junction point with the parallel surfaces of the outlet section and then proceeds past the latter to the outlet cavity. As previously described, increase in separation results in higher face leakage which results in reduction of junction pressure so that the separation must close to restore equilibrium.

There are a number of other types of hydrostatic seal designs which use controlled micro-separation. These also depend on a variable pressure gradient at the interface to control separation distance. Closely associated with the hydrostatic designs are the hydrodynamic designs such as shown by Figure 31. These require relative rotation to obtain the required controlled micro-separation. Essentially, the various seals of this nature utilize hydrodynamic bearing principles of operation, which are somewhat subtle and, therefore, require more technical material than can be given herein.

SEAL SYSTEMS

In recent years more sophisticated types of seal systems have been generated to cope with governmental restrictions and modern chemical process requirements. Because of their magnitude, we cannot within the allotted time cover them all. In fact, a previous presentation [7] by the writer that was devoted entirely to the subject was not itself an "all inclusive" listing. Many of the previously discussed seal assemblies are examples of direct, buffered, vented and simple combination systems.

Closely related to these simple seal systems is the buffered educted restrictive-ring seal system, such as shown by Figure 32. This is an important type for consideration, primarily because it sometimes encounters difficulty due to faulty installation or operation. To operate properly, the seal rings must be installed with the directional sense shown, the buffering pressure, \( P_b \), must be greater than both the process pressure, \( P_p \), and eduction pressure, \( P_e \), while the latter must be maintained below atmospheric pressure, \( P_a \). Common mistakes are to install the rings in the wrong direction, not to have the buffering pressure above the process pressure, and not to have sufficient eduction capacity.

Many applications require seal systems that incorporate different types of components, as for example the multiple combination gas seal system shown by Figure 33. Not only does this have different buffer and vent ports at various pressure levels but it includes three different types of seals together with some redundancy. The primary seals in this combination seal system are the four segmented restrictive-ring seals. This type was selected because the four were subjected to a large pressure differential and had to minimize the natural gas leakage from the buffer inlet toward the low pressure fuel cavity, even though this fuel could be used directly in the process. Two segmented circumferential contact seal rings separate the low pressure fuel vent cavity from the F-4 furnace vent cavity. Controlled low pressure differentials together with low leakage characteristics indicated that circumferential contact seals should be used at this location. If seals at this location were to leak excessively, all of the gas could not be economically burned. Fortunately, these contact seals have performed well within the design specification; i.e., 0.3 SCFM at 35 psi differential. The F-4 furnace vent chamber is separated from the atmosphere by two more circumferential contact seal rings. Low leakage and a very low pressure differential were again the reasons circumferential contact seals were selected for this location. Since this leakage went to the atmosphere, it was economically wasted and raised the pollutant reading. The latter also gave support to the circumferential contact seal selection because their small cross sections permitted the possible future insertion of an inert gas buffering system, which could reduce the pollutant level to zero. Fortunately, during the four years of operation, the inert gas buffering system has never had to be installed because of the low leakage from the direct contact seal system on the atmospheric end. The last two of the six circumferential contact seal rings restrict the flow of buffer gas into the compressor. Ring selection was again based on the fact that the pressure could be regulated to maintain a small differential and that these rings reduced leakage to a minimum, thereby avoiding diluting the compressor gas stream with natural gas. Downstream of these process side seal rings is the labyrinth. Because of the low mass flows, this labyrinth is essentially a back-diffusion restriction which prevents the polymers contained in the process gas from clogging the seal rings.

Some compressor applications are so dangerous that redundant systems are required, such as that shown by the seal system example illustrated by Figure 34. The primary source of buffering nitrogen gas enters at a pressure \( P_i \), which is greater than the machine pressure \( P_m \), between the dry gas...
face seal and the two restrictive ring seals that are separated from the process gas by the single labyrinth knife. The knife functions as a gas pressure integration cavity separator as well as an emergency seal. The secondary source of buffering nitrogen enters at a pressure $P_4$, which is normally only a few pounds above $P_4$ and $P_5$. Should the primary system fail, the line leading from the $P_5$ cavity is automatically blocked off and buffering pressure $P_5$ is raised. It should be noted that even without nitrogen to serve as a buffer, the assembly has an inherent characteristic of acting as a redundant direct seal system and therefore always inhibits process gas flow to the surroundings. A variation of this system is to replace the dry gas face seal with restrictive ring seals but this sacrifices efficiency because of the increased leakage and also restrictive rings do not give the direct seal redundancy previously mentioned. If the pressure levels had been low, the dry gas face seal could have been replaced with circumferential contact seals. More basic alterations have also been applied.

Refrigeration and similar applications sometimes require that a gas buffer be used in conjunction with the previously
mentioned liquid buffered face seal and bushing seal combination to avoid liquid contamination of the process. A popular method is to use the buffered labyrinth seal together with the liquid seal as depicted in Figure 35. The example used is an overhung single stage machine wherein the buffer gas is filtered process gas obtained from the discharge of the compressor. After being injected between the labyrinth knives, some of the buffering gas flows to the back of the impeller and then into the discharge while the remainder flows toward the atmosphere into the contaminated oil chamber. From this chamber the oil buffering gas mixture is piped to a trap where the separated oil is collected and periodically dumped to be reused or reclaimed. The process gas escaping from the top of the trap is normally orifice controlled and then sent to the compressor inlet, to flare or to a recycle section of the process. Notice that in this application both oil and gas are used as the fluids while the active components are a labyrinth, a contact face seal, and redundant fluid film seals together with a low efficiency slinger. Many multi-stage process compressors have the labyrinth as separate items.

Buffered seal systems generally use the conventional components previously discussed. However, even those applications requiring no process outleakage and a minimum of inleakage can be satisfied with a film riding hydrostatic face seal, such as shown by Figure 36. The illustration shows such a device for operation with process gas at 400 psi. An autogenously actuated device would have resulted in outleakage and an externally pressurized device would have prevented outleakage only if the barrier fluid were at a substantial pressure differential above internal machine pressure. The device shown overcomes this difficulty by incorporating an additional internal seal, maintained at a very low differential above machine pressure.

Barrier fluid is fed into the innermost groove through a port of low resistance. The servo-groove pressure adjusts itself to conditions of flow across the faces and through the orifice cascade and controls the separation. Barrier fluid then leaks inward at low differential across the innermost pad and outward at higher differential across the other two pads.

Figure 36. Buffered Triple Dam Hydrostatic Film Riding Face Seal Assembly.

REFERENCES:


