often incur permanent clearance increases under such conditions, degrading seal and machine performance; 3) Require significantly less axial space than labyrinth seal; and 4) More stable leakage characteristics over long operating periods.

Brush seals have matured significantly over the past 20 years. Typical operating conditions of state-of-the-art brush seals are shown in table 5.[†]

TABLE 5.—TYPICAL OPERATING LIMITS FOR STATE-OF-THE-ART BRUSH SEALS

Differential pressure	up to 300 psid per stage	2.1 MPa
Surface speed	up to 1200 ft/sec	400 m/s
Operating temperature	up to 1200 °F	600 °C
Size (diameter range)	up to 120-in.	3.1 m

Brush seal construction is deceptively simple, requiring the well ordered layering or tufting of fine-diameter bristles into a dense pack that compensates for circumferential differences between inside and outside diameters, (figs. 36 and 37). This pack is sandwiched and welded between a backing ring (downstream side) and sideplate (upstream side), then stress relieved to insure stability and flatness. The weld on the seal outer diameter is machined to form a close-tolerance outer diameter-sealing surface to fit into a suitable housing. The wire bristles protrude radially inward (shaft-rotor) or outward (drum-rotor) and are machined to fit the mating rotor, with slight interference. Brush seal interferences (preload) must be properly selected to prevent catastrophic overheating of the rotor and excessive rotor thermal growths.

To accommodate anticipated radial shaft movements, the bristles must bend. To allow the bristles to bend without buckling, the wires are oriented at an angle (typically 45° to 55°) to a radial line through the rotor. The bristles are canted in the direction of rotor rotation. The bristle lay angle also facilitates seal installation, due to the slight interference between the bristle pack and the rotor. The backing ring provides structural support to the otherwise flexible bristles and assists the seal in limiting leakage. To minimize brush seal hysteresis caused by brush bristle binding on the back plate, new features have been added to the backing ring. These include reliefs of various forms. An example design is shown in figure 36 and includes the recessed pocket and seal dam. The recessed pocket assists with pressure balancing of the seal and the relatively small contact area at the seal dam minimizes friction allowing the bristles to follow the speed-dependent shaft growths. The bristle free-radial-length and packing pattern are selected to accommodate radial shaft movements while operating within the wire's elastic range at temperature. A number of brush seal manufacturers[‡] include some form of flow deflector (e.g., see flexi-front plate in figs. 36 and 37) on the high pressure side of the wire bristles. This element aids in mitigating the radial pressure closing loads (e.g., sometimes known as "pressure closing") caused by air-forces urging the bristles against the shaft. This element can also aid in reducing installation damage, bristle flutter in highly turbulent flow fields, and FOD.

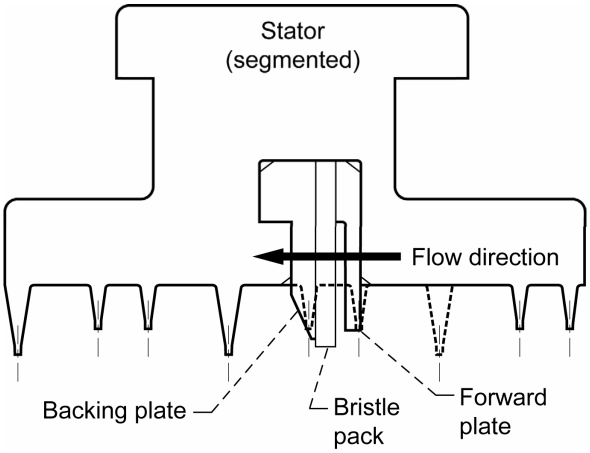

Figure 36.—Typical brush seal configuration and geometric features.[37,80]

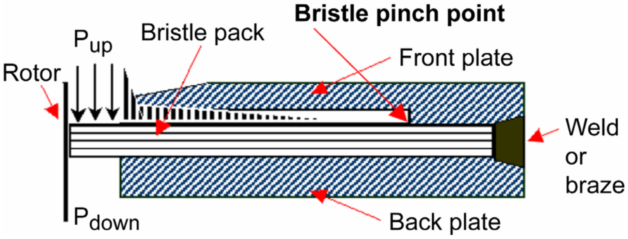

Figure 37.—Brush seal design for steam turbine applications.[37, 80]

Brush seals, initially developed for aero-gas turbines, have also been used in industrial gas and steam turbines since the 1990s. Design similitude, analysis and modeling of brush and woven seals were established earlier in the works of Flower[73] and Hendricks et al.[22] Within in the confines of this paper we are only able to address a few sealing types, their locations and material constraints. For further details, see Hendricks and coworkers[25,74,75] and NASA Conference Publications.[76,77] An extensive summary of brush seal research and development work through 1995 has been published[78,79] and updated in a more recent summary.[37]

1. Brush Seal Design Considerations

To properly design and specify brush seals for an application, many design factors must be considered and traded-off. Comprehensive brush seal design algorithms

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[†] Data available on at http://www.fluidsciences.perkinelemer.com/turbomachinery.

[‡] Data available online at http://www.crossmanufacturing.com.

have been proposed by Chupp et al.,[37] Dinc et al.,[80] Hendricks et al.,[22] and Holle and Krishan.[81] An iterative process must be followed to satisfy seal basic geometry, stress, thermal (especially during transient rub conditions), leakage, and life constraints to arrive at an acceptable design. Many of the characteristics that must be considered and understood for a successful brush seal design are given here:[80] pressure capability, seal upstream protection, frequency, seal high- and low-cycle fatigue (HCF, LCF) analysis, seal leakage, seal oxidation, seal stiffness, seal creep, seal blow-down (e.g., pressure closing effect), seal wear, bristle-tip forces and pressure stiffening effect , solid particle erosion, seal heat generation, reverse rotation, bristle-tip temperature, seal life/long term considerations, rotor dynamics, performance predictions, rotor thermal stability, oil sealing, secondary flow and cavity flow (including swirl flow), and shaft considerations: (e.g., coating, etc.). Design criteria are required for each of the different potential failure modes including stress, fatigue life, creep life, wear life, oxidation life, amongst others. Several important designs parameters are discussed next.

a) Material selection.—Materials in rubbing contact in brush seal installations must have sufficient wear resistance to satisfy engine durability requirements. A proper material selection requires knowledge of the rotor and seal materials and their interactions. In addition to good wear characteristics, the seal material must have acceptable creep and oxidation properties.

Metallic bristles: Brush seal wire bristles range in diameter from 0.071-mm (0.0028-in.) (for low pressures) to 0.15-mm (0.006-in.) (for high pressures). The most commonly used material for brush seals is the cobalt-based alloy Haynes 25 based on its good wear and oxidation characteristics. Brush seals are generally run against a smooth, hard-face coating to minimize shaft wear and the chances of wear-induced cracks from affecting the structural integrity of the rotor. The usual coatings selected for aircraft applications are ceramic, including chromium carbide and aluminum oxide. Selecting the correct mating wire and shaft surface finish for a given application can reduce frictional heating and extend seal life through reduced oxidation and wear. There is no general requirement for coating industrial gas and steam turbine rotor surfaces where the rotor thicknesses are much greater than aircraft applications.

Nonmetallic bristles: High-speed turbine designers have long wondered if brush seals could replace labyrinth seals in bearing sump locations. Brush seals would mitigate traditional labyrinth seal clearance opening and corresponding increased leakage. Issues slowing early application of brush seals in these locations included: coking (carburization of oil particles at excessively high temperatures), metallic particle damage of precision rolling element bearings, and potential for fires. Development efforts have found success in applying aramid bristles for certain bearing sump locations.[82,83] Advantages of the aramid bristles include: stable properties up to 300 °F (150 °C) operating temperatures, negligible amount of shrinkage and moisture absorption, lower wear than Haynes 25 up to 300 °F, lower leakage (due to smaller 12 μm diameter fibers), and resistance to coking.[82] Based on laboratory demonstration, the aramid fiber seals were installed in a GE 7EA frame (#1) inlet bearing sealing location. Preliminary field data showed that the nonmetallic brush seal maintained a higher pressure difference between the air and bearing drain cavities and enhanced the effectiveness of the sealing system allowing less oil particles to migrate out of the bearing.

b) Seal fence height.—A key design issue is the required radial gap (fence height) between the backing ring and the rotor surface. Following detailed secondary flow, heat transfer, and mechanical analyses, fence height is determined by the relative transient growth characteristics of the rotor vs. the stator and rotordynamic considerations. This backing ring gap is designed to avoid contact with the rotor surface during any operating condition with an assumed set of dimensional variations. Consequently, the successful design of an effective brush seal hinges on a thorough knowledge of the turbine behavior, operating conditions, and design of surrounding parts.

c) Brush pack considerations.—Depending on required sealing pressure differentials and life, wire bristle diameters are chosen in the range of 0.0028 to 0.006-in.[84] Better load and wear properties are found with larger bristle diameters. Bristle pack widths also vary depending on application: the higher the pressure differential, the greater the pack width. Higher-pressure applications require bristle packs with higher axial stiffness to prevent the bristles from blowing under the backing ring. Dinc et al.[80] have developed brush seals that have operated at air pressures up to 2.76 MPa (400 psid) in a single stage. Brush seals have been made in very large diameters. Large brush seals, especially for ground power applications are often made segmented to allow easy assembly and disassembly, especially on machines where the shaft stays in place during refurbishment.

d) Seal stress/pressure capability.—Pressure capacity is another important brush seal design parameter. The overall pressure drop establishes the seal bristle diameter, bristle density, and the number of brush seals in series. In a bristle pack, all bristles are essentially cantilever beams held at the pinch point by a front plate and supported by the back plate. From a loading point of view, the bristles can be separated into two regions (see fig. 36). The lower part, fence region, between the rotor surface and the back plate inner diameter (ID), and the upper part from the back plate ID to the bristle pinch point. The innermost radial portion carries the main pressure load and is the main source of the seal stress.[85] In addition to the mean bending stress, contact stress at the bristle-back plate interface must be considered. Furthermore, bristle stress is a very strong function of the fence height set by the expected relative radial movement of the rotor and seal. Figure 38 shows a diagram illustrating design considerations for seal stress and deflection analysis, and includes a list the controllable and noncontrollable design parameters. As a word of caution, care must be taken in using multiple brush configurations as pressure drop capability becomes more non-linear with fluid compressibility and most of the pressure drop or bristle pressure loading is carried by the downstream brush.

e) Heat generation/bristle tip temperature.—As the brush seal bristles rub against the rotor surface, frictional heat is created that must be dissipated through convection and conduction and is quite similar to the classic Blok problem,[86] where extensive heating occurs at the sliding interface. Brush seal frictional heating was addressed by Hendricks et al.[22,87] and modeled as fin in crossflow with a heat source at the tip by Dogu and Aksit.[88] If the seal is not properly designed, this heating can lead to premature bristle loss, or worst, the rotor/seal operation could become thermally unstable. The latter condition occurs when the rotor grows radially into the stator increasing the frictional heating, leading to additional rotor growth, until the rotor rubs the seal backing plate resulting in component failure. In some turbine designs, brush seals are often assembled with a clearance to preclude excessive interference and heating during thermal and speed transients. These mechanical design issues significantly affect the range of feasible applications for brush seals. Many of these issues have been addressed by Dinc et al.[80] and Soditus.[89]

f) Seal leakage.—Leakage characterization of brush seals typically consists of a series of tests at varying levels of bristle-to-rotor interference or clearance, as shown in figures 39 and 40. Static (nonrotating) tests are run to get an approximate level of seal leakage and pressure capability. They are followed by dynamic (rotating) tests to provide a more accurate simulation of seal behavior. Rotating tests also reveal rotor dynamics effects, an important consideration for steam turbine rotors and turbomachines in general, that can be sensitive to radial rubs due to nonuniform heat generation.

Proctor and Delgado studied the effects of speed [up to 365 m/s (1200 ft/s)], temperature [up to 650 °C (1200 °F)] and pressure [up to 0.52 MPa (75 psid)] on brush seal and finger seal leakage and power loss.[90] They determined that leakage generally decreased with increasing speed. Leakage decreases somewhat with increasing surface speed since circumferential flow is enhanced and the rotor diameter increases; changes in diameter causes both a decrease in the effective seal clearance and an increase in contact stresses (important in wear and surface heating).

g) Other Considerations.—If not properly considered, brush seals can exhibit three other phenomena deserving some discussion. These include seal “hysteresis,” “bristle stiffening,” and “pressure closing.” As described by Short et al.[84] and Basu et al.,[91] after the rotor moves into the bristle pack (due to radial excursions or thermal growths), the displaced bristles do not immediately recover against the frictional forces between them and the backing ring. As a

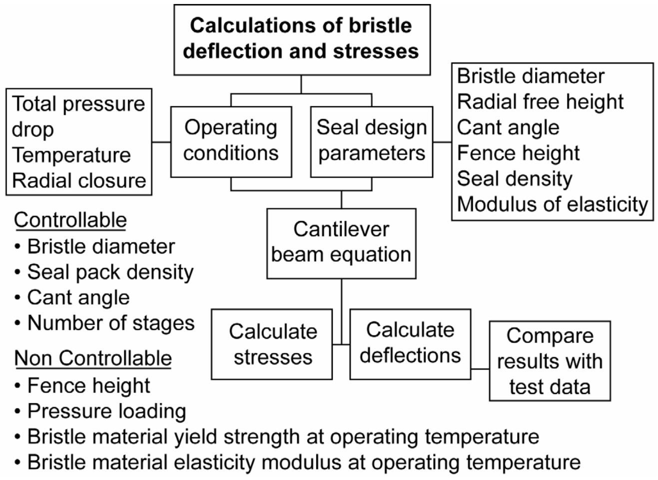


Figure 38.—Bristle stress/deflection analysis.[37, 80]

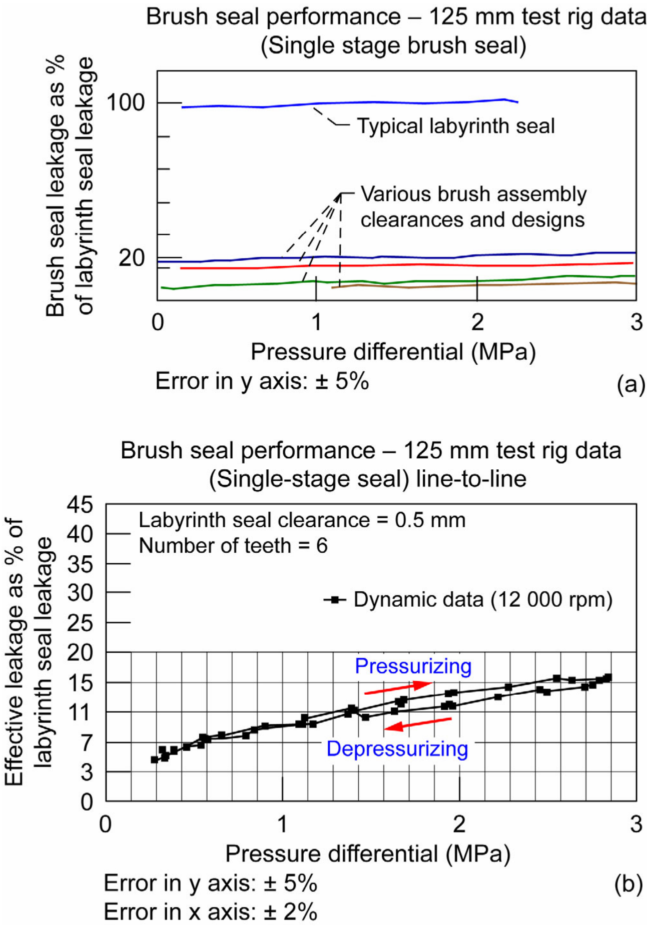


Figure 39.—Brush seal performance as compared to labyrinth seal. Representative brush seal leakage data compared to a typical, 15-tooth, 0.5 mm (20 mil) clearance labyrinth seal. Measured brush seal leakage characteristic with increasing and decreasing pressure drop compared to a typical, 6-tooth, 0.5 mm (20 mil) clearance labyrinth seal.[37]

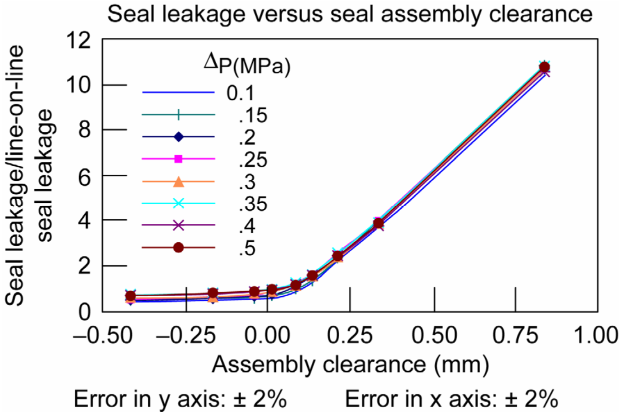


Figure 40.—Measured brush seal leakage for interference and clearance conditions.[37]

result, a significant leakage increase (more than double) was observed following rotor movement.[91] This leakage hysteresis exists until after the pressure load is removed (e.g., after the engine is shut down). Furthermore if the bristle pack is not properly designed, the seal can exhibit a considerable stiffening effect with application of pressure. This phenomenon results from interbristle friction loads making it more difficult for the brush bristles to flex during shaft excursions. Air leaking through the seal also exerts a radially inward force on the bristles, resulting in what has been termed "pressure closing" or bristle "blow-down." This extra contact load, especially on the upstream side of the brush, affects the life of the seal (upstream bristles are worn in either a scalloped or coned configuration) and higher interface contact pressure. By measuring baseline seal leakage in a line-to-line (zero clearance) assembly configuration, bristle blowdown for varying loads of assembly clearance can be inferred from leakage data (see fig. 40).

2. Brush Seal Flow Modeling

Brush seal flow modeling is complicated by several factors unique to porous structures, in that the leakage depends on the seal porosity, which depends on the pressure drop across the seal. Flow through the seal travels perpendicular to the brush pack, through the annulus formed between the backing ring bore and the shaft diameter. The flow is directed radially inward towards the shaft as it flows around individual bristles and collides with the bristles downstream in adjacent rows of the pack and finally between the bristle tips and the shaft.

A flow model proposed by Holle et al.,[92] uses a single parameter, effective brush thickness, to correlate the flows through the seal. Variation in seal porosity with pressure difference is accounted for by normalizing the varying brush thicknesses by a minimum or ideal brush thickness. Maximum seal flow rates are computed by using an iterative procedure that has converged when the difference in successive iterations for the flow rate is less than a preset tolerance.

Flow models proposed by Hendricks et al.,[22,87,93] are based on a bulk average flow through the porous media. These models account for brush porosity, bristle loading and deformation, brush geometry parameters and multiple flow paths. Flow through a brush configuration is simulated using an electrical analog with driving potential (pressure drop), current (mass flow), and resistance (flow losses, friction and momentum) as the key variables. All of the above mentioned brush flow models require some empirical data to establish correlation constants. Once the constants are established, the models can predict brush seal flow reasonably well.

A number of researchers have applied numerical techniques to model brush seal flows and bristle pressure loadings.[94–97] Though these models are more complex, they permit a more detailed investigation of the subtleties of flow and stresses within the brush pack.

3. Applications

a) Aero gas turbine engines.—Brush seals are seeing extensive service in both commercial and military turbine engines. Lower leakage brush seals permit better management of cavity flows and significant reductions in specific fuel consumption when compared to competing labyrinth seals. Allison Engines has implemented brush seals for the Saab 2000, Cesna Citation-X, and V-22 Osprey. General Electric has implemented a number of brush seals in the balance piston region of the GE90 engine for the Boeing 777 aircraft. Pratt & Whitney has entered revenue service with brush seals in three locations on the PW1468 for Airbus aircraft and on the PW4084 for the Boeing 777 aircraft.[98]

b) Ground-based turbine engines.—Brush seals are being retrofitted into ground-based turbines both individually and combined with labyrinth seals to greatly improve turbine power output and heat rate.[37,80,99–103] Dinc et al., report that incorporating brush seals in a GE Frame 7EA turbine in the high pressure packing location increased output by 1.0 percent and decreased heat rate by 0.5 percent.[80] Figure 41 is a photo of a representative brush seal taken during a routine inspection. The seal is in good condition after nearly three years of operation (~22,000 hr). To date, more than 200 brush seals have been installed in GE industrial gas turbines in the compressor discharge high-pressure packing (HPP), middle bearing, and turbine interstage locations. Field data and experience from these installations have validated the brush seal design technology. Using brush seals in the interstage location resulted in similar improvements. Brush seals have proven effective for service lives of up to 40,000 hr.[80]

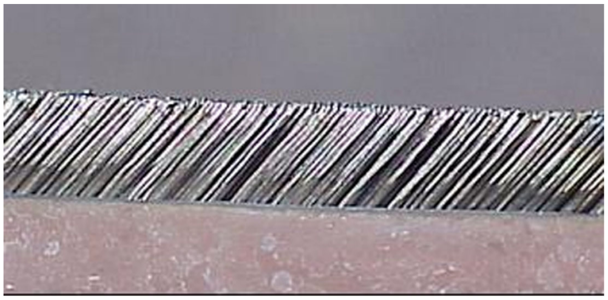


Figure 41.—7EA Gas turbine high-pressure packing brush seal in good condition after 22,000 hr of operation.[37, 80]

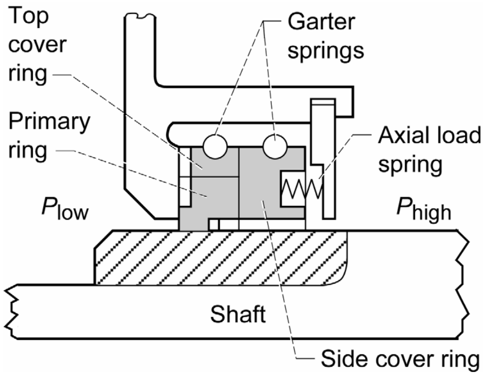


Figure 42.—Shaft riding or circumferential contact seal.[104]

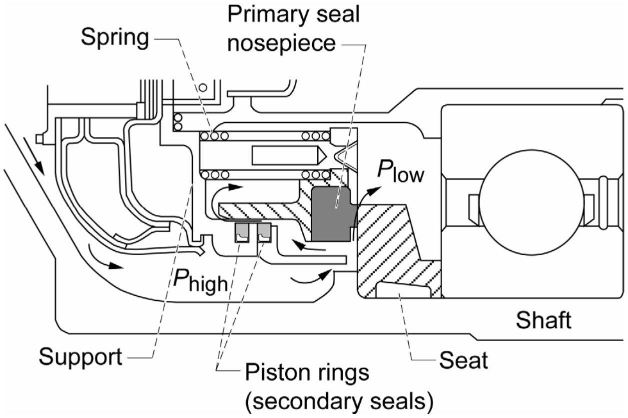


Figure 43.—Positive contact face seal.[106]

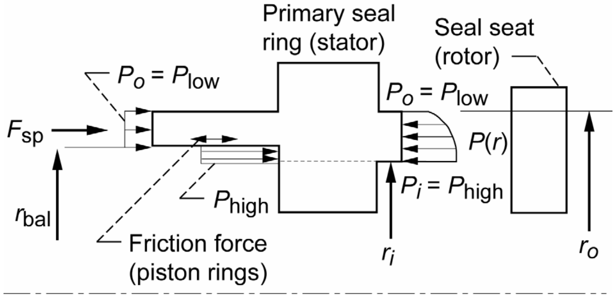


Figure 44.—Pressure balancing forces in face sealing.[108]

E. Face Seals

Labyrinth seals are less impacted by FOD-debris than other types of seals, yet also pass that debris to other components such as bearing cavities. One of the major functions of face and buffer sealing is to preclude debris from entering the bearing or gear-box oil yet an equally important function is to prevent oil vapors from leaking into the wheel-space and from entering the cabin air stream. Debris in the bearing or gear-box oil can radically shorten life and oil-vapor in the wheel space can cause fire or explosions. Oil vapors in the cabin are unacceptable to the consumer-traveler.

Face seals are classified as mechanical seals. They are pressure balanced contact or self-acting seals. The key components are the primary ring (stator) or nosepiece, seat or runner (rotor), spring or bellows preloader assembly, garter or retainer springs, secondary seal and housing (figs. 42 and 43).[105,106] There is a wealth of information on experimental, design and application of mechanical seals in the literature, including Ludwig[4] to books by, for example., Lebeck.[107]

For the face seal, the geometry of the ring or nosepiece becomes critical. For successful face sealing, the forces due to system pressure, sealing dam pressure and the spring or bellows must be properly balanced and stable over a range in operating parameters (pressure, temperature, surface speed) (fig. 44).[108]

Contact seals wear and are generally limited to surface speeds less than 76 m/s (250 ft/s). To mitigate the wear, prolonging life and decreased leakage Ludwig[109] and Dini[110] promoted the self-acting Rayleigh step and spiral groove seal, (figs. 45 to 47). A labyrinth seal or a simple projection representing a single throttle is used for presealing to control excessive leakage should the dam of the face seal "pop" open; for example, the labyrinth preseal as is illustrated in figure 45 (and aspirating seal of section V.E). Spiral groove (fig. 47), slot and T-grooving (bidirectional) are more commonly used than Rayleigh steps to provide more lift at less cost to manufacture.

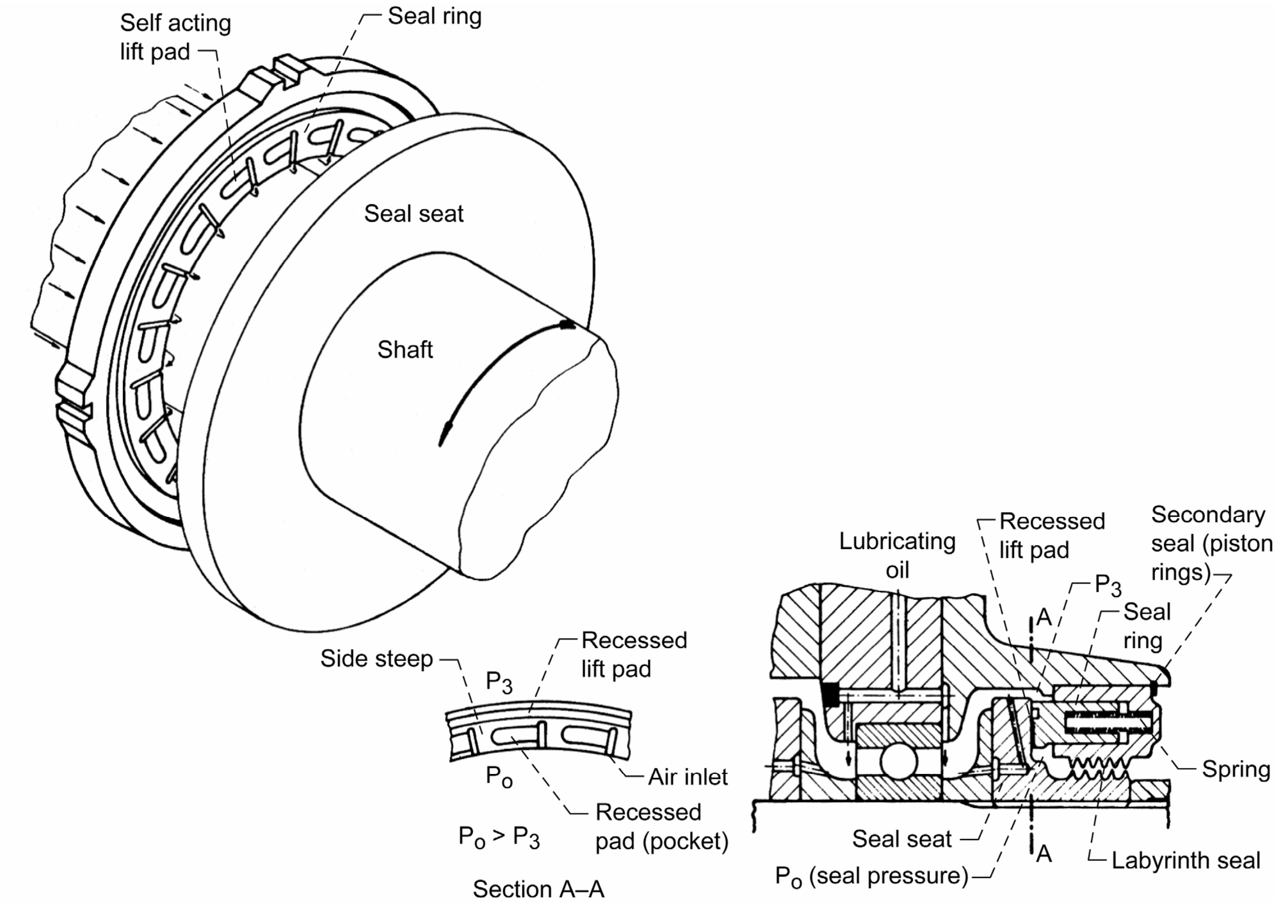
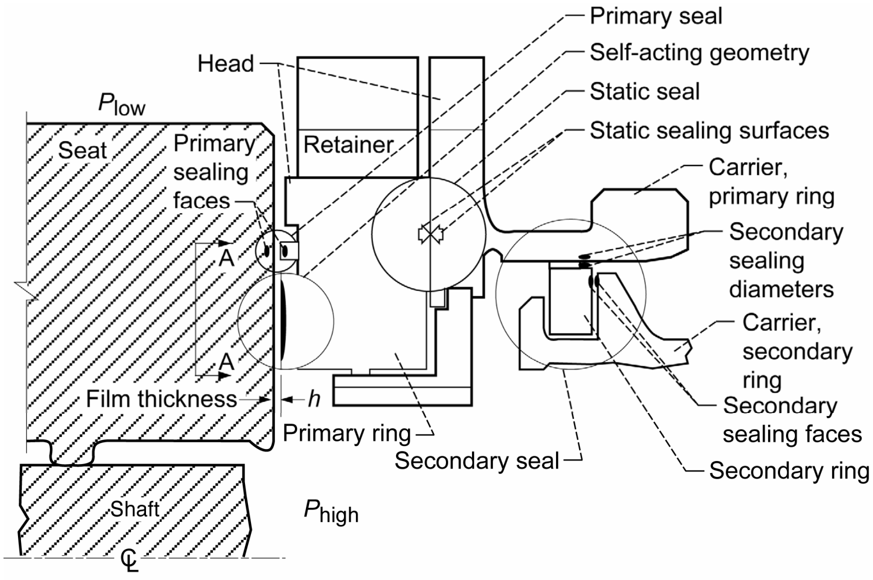


Figure 45.—Self-acting face seal with labyrinth seal presealing.[110]



(a) Nomenclature.

Figure 46.—Component schematic Rayleigh pad self-acting face seal.[109]

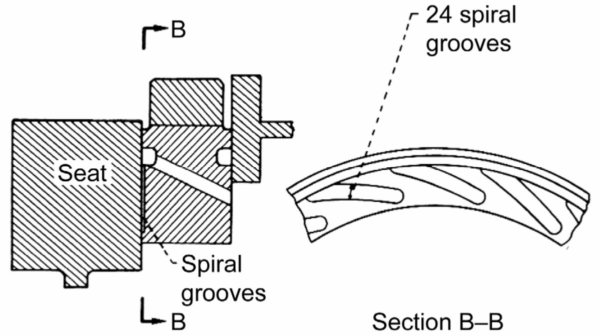


Figure 47.—Spiral groove sealing schematic.[109]

Self-acting seals permit tighter clearances and better control of the sealing dam geometry as sealing pressure drops are increased, providing lower leakage. Figure 48 provides a comparison of the leakage rates between labyrinth, face-contact and self-acting seals. While self-acting face sealing greatly reduces leakage, surface speeds are generally limited to less than 213 m/s (700 ft/s), but nearly triple the limits of contact face sealing 61 to 91 m/s (200 to 300 ft/s).

F. Oil Seals

Gas turbine shaft seals are used to restrict leakage from a region of gas at high pressure to a region of gas at low pressure. A common use of mechanical seals is to restrict gas leakage into bearing sumps. Oil sealing of bearing compartments of turbomachines is difficult. A key is to prevent the oil side of the seal from becoming flooded. Still, oil-fog and oil-vapor leakage can occur by diffusion of oil due to concentration gradients and oil transport due to vortical flows within the rotating labyrinth-cavities (crude distillation columns). Bearing sumps contain an oil-gas mixture at near-ambient pressure, and a minimal amount of gas leakage through the seal helps prevent oil leakage and maintains a minimum sump pressure necessary for proper scavenging. Bearing sumps in the HPT are usually the most difficult to seal because the pressure and temperatures surrounding the sump can be near compressor discharge conditions.

1. Radial Face Seals

Conventional rubbing-contact seals (shaft-riding and radial face types) are also used to seal bearing sumps. Because of their high wear rates, shaft-riding and circumferential seals (fig. 42) have been limited to pressures less than 0.69 MPa (100 psi); and successful operation has been reported at a sealed pressure of 0.58 MPa (85 psi), a gas temperature of 370 °C (700 °F), and sliding speed of 73 m/s (240 ft/s).[105]

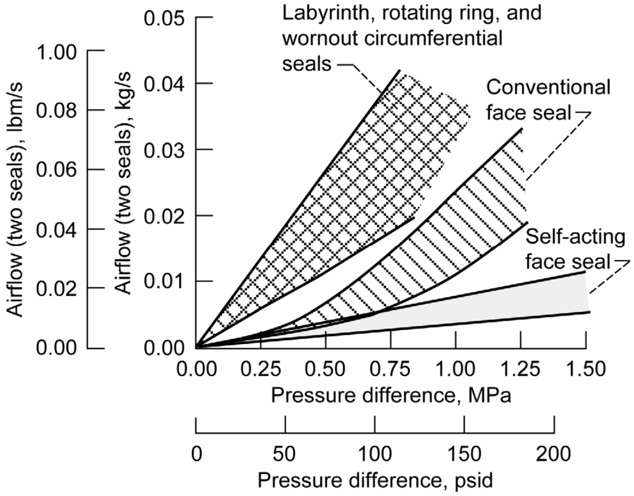


Figure 48.—Comparison of leakage characteristics for labyrinth, conventional (contact) face seal and self-acting face seals.[109]

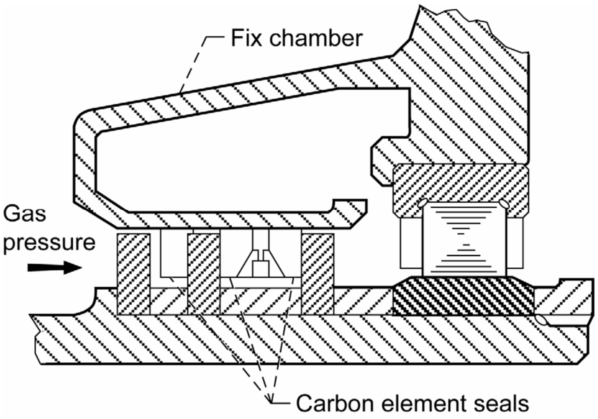


Figure 49.—Expanding ring seal.[106]

2. Ring Seals

The ring seal, as described by Whitlock[111] and Brown,[106] is essentially an expanding or contracting piston ring. The expanding design is simpler and is illustrated in figure 49. Other designs that can be grouped in the ring seal family include the circumferential segmented ring seal and the floating or controlled-clearance ring, as described by Ludwig.[4] The material requirements for these seals are essentially the same as those for the expanding ring seal. The ring seals are carbon and they seal radially against the inside diameter of the stationary cylindrical surface as well as axially against the faces of the adjacent metal seal seats (fig. 49). The metal seal seats are fixed to, and rotate with, the shaft. The sealing closing force is provided by a combination of spring forces and gas pressures. Ring seals are employed where there is a large relative axial movement due to thermal mismatch between the shaft and the stationary structure. Ring seals are limited to operation at air pressure drops and sliding speeds considerably lower than those allowed for face seals. However, they can be used to gas temperature levels in the same range as for positive-contact face seals, approximately to 480 °C (900 °F). Generally, a minimum pressure differential of 14 kPa (2 psid) must be maintained to prevent oil leakage from the bearing compartment.

Carbon ring and face sealing of the sumps described by Ludwig,[4] Whitlock,[111] and Brown[106] are fairly standard. Boyd et al.[112] have investigated a hybrid ceramic shaft seal which is comprised of a segmented carbon ring with lifting features as the outer or housing ring and a silicon-nitride tilt-support arched rub runner mounted on a metal flex beam as the inner ring (fig. 50). The flex beam added sufficient damping for stability and no oil seepage was seen at idle speed down to pressure differentials of 0.7kPa (0.1 psia), air to oil.

3. Materials

Selecting the correct materials for a given seal application is crucial to ensuring desired performance and durability. Seal components for which material selection is important from a tribological standpoint are the stationary nosepiece (or primary seal ring) and the mating ring (or seal seat), which is the rotating element. Brown[106] described the properties considered ideal for the primary seal ring as shown here: 1) mechanical—high modulus of elasticity, high tensile strength, low coefficient of friction, excellent wear characteristics and hardness, self-lubrication; 2) thermal—low coefficient of expansion, high thermal conductivity, thermal shock resistance, and thermal stability; 3) chemical—corrosion resistance, good wetability; and 4) miscellaneous—dimensional stability, good machinability, and low cost and readily available.

Because of its high ranking in terms of satisfying these properties, carbon graphite is used extensively for one of the mating faces in rubbing contact shaft seals. However, in spite of its excellent properties, the carbon material must be treated in order for it to satisfy the operational requirements of sealing applications in the main rotor bearing compartment of jet engines.

Seal failures are driven by thermal gradient fatigue or axial and radial thermal expansions during maximum power excursions. Bearing compartment carbon seals will fail from the heat generated in frictional rub. Excessive face wear occurs during transients and, as mentioned, labyrinth seals can allow oil transport out of the seal and oil contamination by the environment (moisture, sand, etc.) [113]

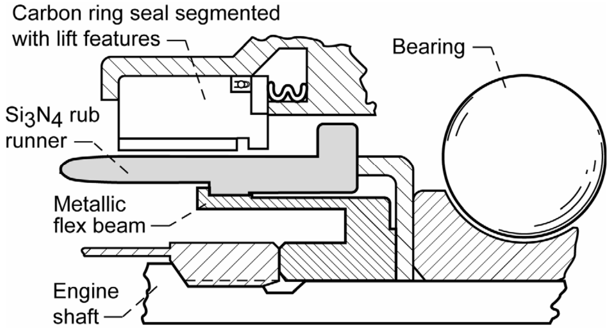


Figure 50.—Hybrid ceramic carbon ring seal.[112]

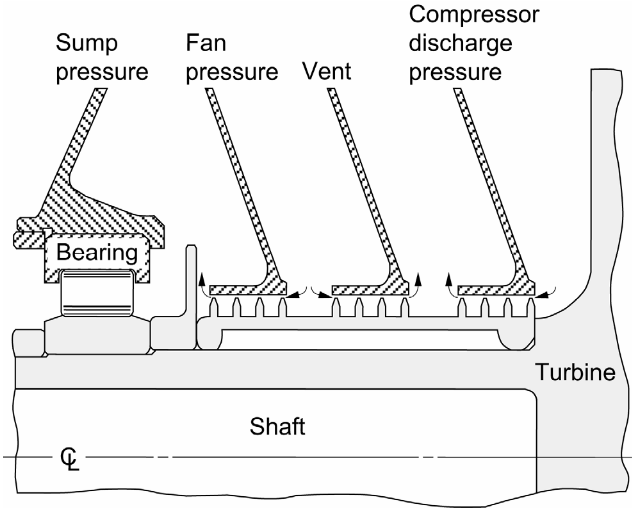


Figure 51.—Schematic of aero-gas-turbine buffer sealing of oil cavity.[109]

G. Buffer Sealing

Public awareness of environmental hazards, well-publicized effect of hazardous leakages (Three Mile Island, Challenger), and a general concern for the environment, have precipitated emissions limits that drive the design requirements for sealing applications. Of paramount concern are the types of seals, barrier fluids, and the necessity of thin lubricating films and stable turbomachine operation to minimize leakages and material losses generated by rubbing contacts.[104]

A zero-leakage seal is an oxymoron. Industrial practice is to introduce a buffer fluid between ambient seals and those seals confining the operational fluid (fig. 51) with proper disposal of the buffered fluid mixture,[109,111] A second example is for shaft sealing as shown in figure 52 where buffer fluids are introduced. In the case of oil sumps, the buffered mixture is vented to the hot gas exhaust stream and is presumed to be consumed. Within the nuclear industry, this becomes a containment problem where waste storage now becomes an issue. In the case of rocket engines, the use of buffering or inerting fluids (e.g., helium) is commonplace to separate fuel and oxidizer-rich environments for example in the Space Shuttle main engine (SSME) turbomachinery.

H. Rim Sealing and Disk Cavity Flows

Turbomachine blade-vane interactions engender unsteady seal and cavity flows in multiply connected cavities with conjugate heat transfer and rotordynamics. A comprehensive review of seals-secondary flow system developments are documented by Hendricks et al.[114,115] and NASA Seals Code and Secondary Flow Systems Development publications.[77]

Unsteady flows perturb both the power and the secondary flow streams.[2] A T1 turbine (first stage of the HPT) can have 76 blades and 46 stators all interacting with unsteady loadings (fig. 53).[116] Cavity ingestion of rapidly pulsating hot gases induce cavity heating, increases disk temperature, which in turn limits disk life and can compromise engine safety. Proper sealing confines these gases to the blade platform regions.

Rotordynamic issues further complicate rim seal and interface seal designs. These issues are addressed in: Thomas,[63] Alford,[64,117,118] Benckert and Wachter,[66] NASA Conference Publications,[76] Abbott,[65] von Pragenau,[119] Vance,[120] Childs,[121] Muszynska,[68] Bently and Hatch,[122] Hendricks,[115] and Temis.[123]

Cavity and sealing interface requirements differ between industrial and aero-turbomachines. Major differences include split casings and through bolted disks, and compressors and turbines with common drive shafts for industrial machines vs. cylindrical casings and drum rotors on multiple spools for aero machines. Figure 53 shows a typical aero multistage turbine cavity section. Several experimental studies have been reported that consider both simplified and complex disk cavity configurations (e.g., Chen;[124] Chew et al.[125,126] Graber et al.,[127] and Johnson et al.[128,129]). Cavity sealing is complex and has a significant effect on component and engine performance and life. However, several analytical and numerical tools are available to help guide the designer, experimenter and field engineer in addressing these challenges (see appendix A).

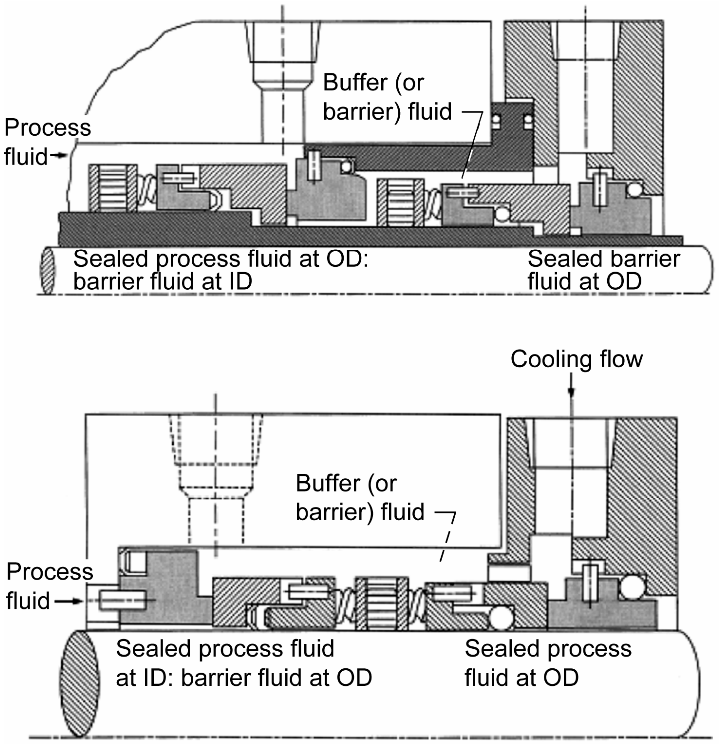


Figure 52.—Schematic of buffer fluid use in system sealing.[104]

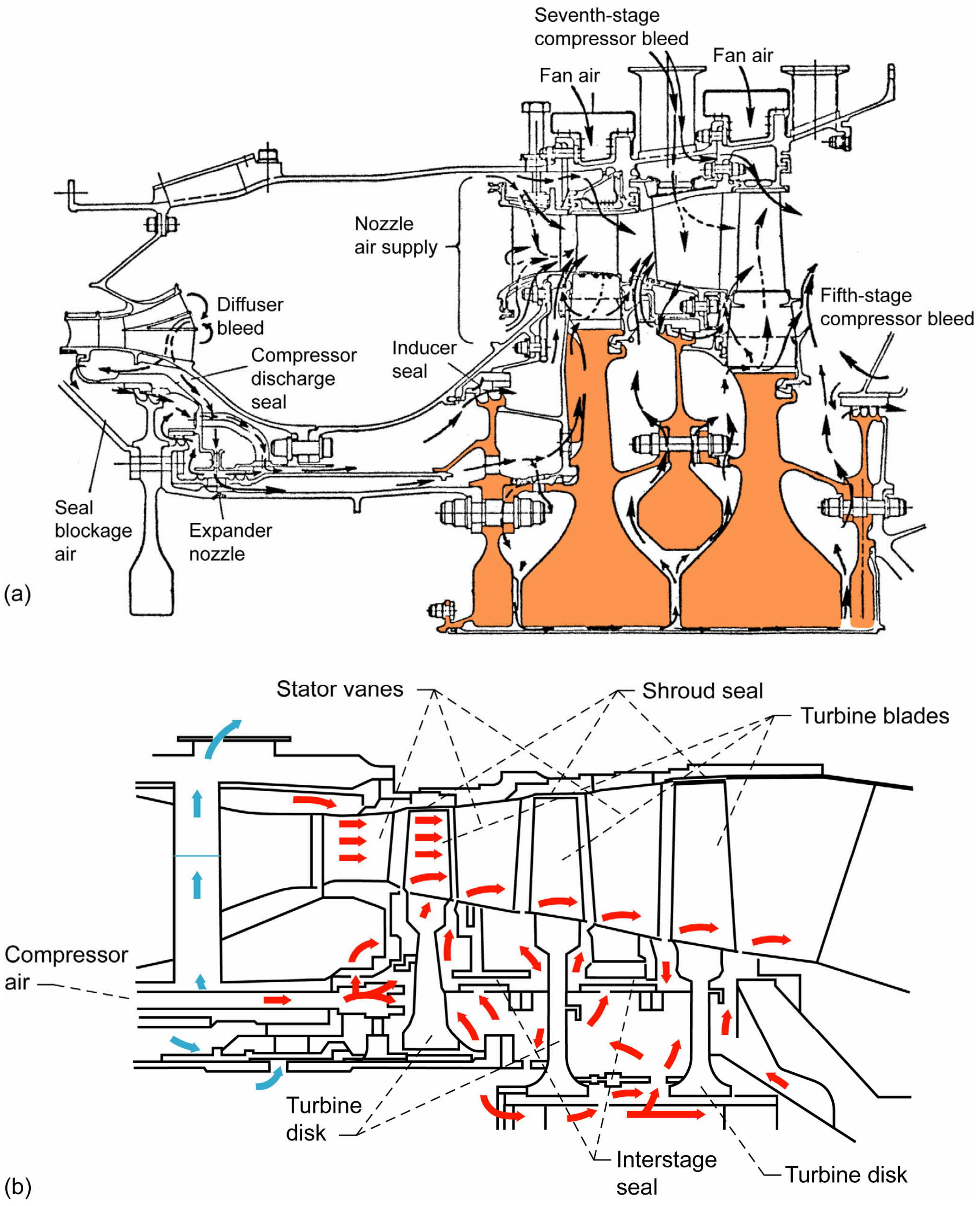


Figure 53.—Typical multistage turbine cavity section. (a) Energy Efficient Engine high-pressure turbine.[2] (b) Hypothetical turbine secondary-air cooling and sealing[116] (courtesy AIAA).

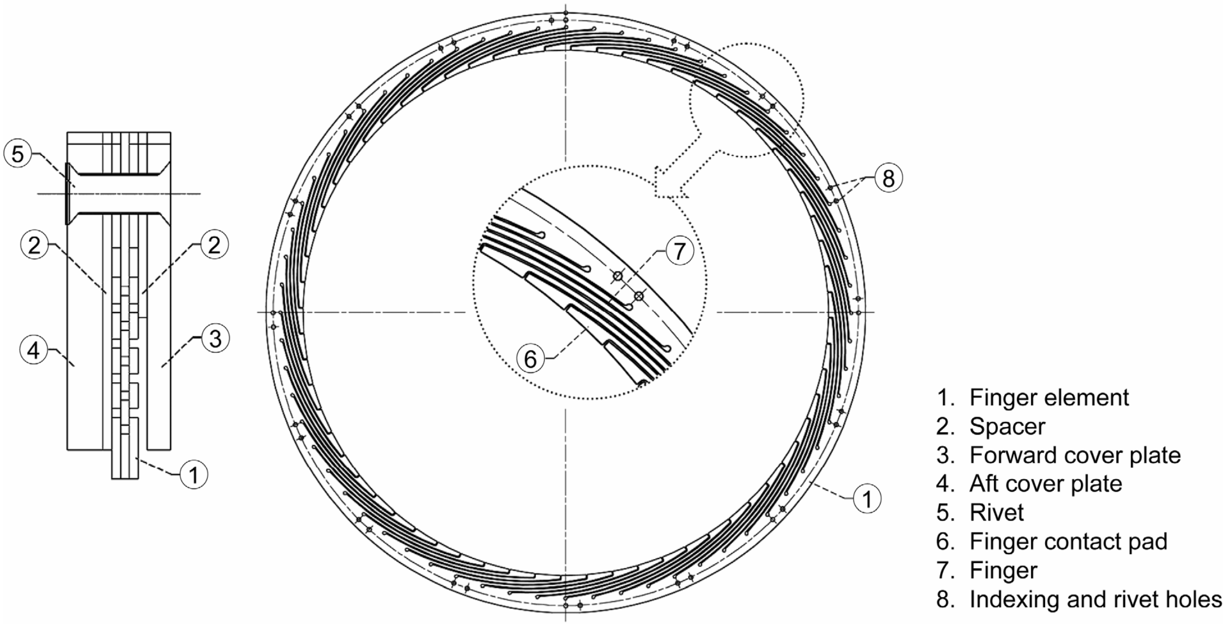


Figure 54.—Finger seal and detailed components.[132]

V. Advanced Seal Designs

A. Finger Seal

The finger seal is a relatively new seal technology developed for air-to-air sealing for secondary flow control and gas path sealing in gas turbine engines.[130–132] It can easily be used in any machinery to minimize airflow along a rotating or nonrotating shaft. Measured finger seal air leakage is 1/3 to 1/2 of conventional labyrinth seals. Finger seals are compliant contact seals. The power loss is similar to that of brush seals.[133] It is reported that the cost of finger seals are estimated to be 40 to 50 percent of the cost to produce brush seals.

The finger seal is comprised of a stack of several precisely machined sheet stock elements that are riveted together near the seal outer diameter as shown in figure 54. The outer elements of the stack, called the forward and aft coverplates, are annular rings. Behind the forward coverplate is a forward spacer, then a stack of finger elements, the aft spacer and then the aft coverplate. The forward spacer is an annular ring with assembly holes and radial slots around the seal inner diameter that align with feed-thru holes for pressure balancing. The finger elements are fundamentally an annular ring with a series of cuts around the seal inner diameter to create slender curved beams or fingers with an elongated contact pad at the tip. Each finger element has a series of holes near the outer diameter that are spaced such that when adjacent finger elements are alternately indexed to the holes, the spaces between the fingers of one element are covered by the fingers of the adjacent element. Some of the holes create a flow path for high pressure upstream of the seal to reach the pressure balance cavity formed between the last finger element, the aft spacer and seal dam, and the aft coverplate. The aft spacer consists of two concentric, annular rings. One is like the forward spacer. The second is smaller with an inner diameter the same as the aft coverplate and forms the seal dam. It is connected to the outer annular ring by a series of radial spokes.

The fingers provide the compliance in this seal and act as cantilever beams, flexing away from the rotor during centrifugal or thermal growth of the rotor or during rotordynamic deflections. The pressure balance cavity reduces the axial load reacted by the seal dam and hence minimizes the frictional forces that would cause the fingers to stick to the seal dam and cause hysteresis in the finger seal leakage performance. In this seal there are two leakage paths. One is thru (around and under) the fingers at the seal/rotor interface. The other is a radial flow across the seal dam. When a pressure differential exists across the seal the fingers tend to move radially inward towards the rotor. Test results confirm this pressure closing effect. The pressure closing effect is largely due to the pressure gradient under the finger contact pads. The bulk of the radial pressure loads on the curved beam of the finger balance out to a zero net load. Ideally, one would design finger seals to have a line-to-line fit during operation. However, most applications involve a range of operating conditions and seal-to-rotor fits and clearances change due to different coefficients of thermal expansion, centrifugal rotor growth, pressure closing effects, and dynamics of the rotor. Depending on the requirements of the application it may be desirable to start with an interference-fit at build and allow the seal to wear in