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Turbomachine Sealing and Secondary Flows

Part 1—Review of Sealing Performance, Customer, Engine Designer, and Research Issues

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July 2004

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Abstract

Although forces outside our control shape our industry, turbomachine sealing research, design, and customer agendas established in 1978 by Ludwig, Campbell, and Smith in terms of specific fuel consumption and performance remain as objectives today. Advances have been made because failures of the space shuttle main engine turbomachinery ushered in a new understanding of sealing in high-power-density systems. Further, it has been shown that changes in sealing, especially for high-pressure rotors, dramatically change the performance of the entire engine or turbomachine. Maintaining seal leakages and secondary flows within engine design specifications remains the most efficient and cost-effective way to enhance performance and minimize maintenance costs. This three-part review summarizes experiences, ideas, successes, and failures by NASA and the U.S. aerospace industry in secondary flow management in advanced turbomachinery. Part 1 presents system sealing, part 2 system rotordynamics, and part 3 modeling, with some overlap of each part.

Keywords: Turbomachine, seals, engine performance, maintenance, design

Introduction

Over the years through research and new design and fabrication methods, engine performance has increased but at the expense of complexity and a rapid decline in margins available for improving conventional turbomachines (see also the section Unsteady Flow Systems in part 3). In the past one could look to percentage improvements in performance, while today researchers, designers, and customers (e.g., airlines) work diligently for fractions of a point.

Turbomachine sealing issues such as controlled leakage and engine dynamics, minimization of secondary coolant and parasitic flows, maintenance, and enhanced component life are fundamental to engine performance. In the past component performance gains could reliably translate into increased profit margins. Today, without fully integrating sealing issues with global engine market issues, component gains could translate into decreased profit margins.

The driver is today's traveling public, who demand increases in security, safety, reliability, comfort, and ease all at low fares. These demands have offset engine and aircraft performance gains to the point where airline profit margins have become razor thin. Nevertheless, the foundations for customer profits, engine design, and research objectives established in the 1970's remain as aggressive goals and objectives in today's market. However, forces outside our control, in reality, shape the goals of our industry.

Herein we illustrate these 1970's turbomachine sealing objectives from the airline-customer, design, and research points of view with comparisons to some of today's methodology and control issues. In part 2 we explore turbomachine sealing dynamics methods, and in part 3 we discuss secondary flows, engine externals, and component life.

Industry Control Issues

In a series of articles investigating the major problems facing the airline industry, Anderson (2003) discusses three general categories: (1) the role of government and public policy, (2) the effect of the economic and geopolitical environment, and (3) the responsibility of airlines to meet customer needs and expectations. Anderson (2002) also cites public policy positions adopted by the Air Transport Association that are vital to the survival of the industry. These practices all center on who ultimately pays for aviation security, taxation, airport screening, extent of regulation, and unfunded mandates. Today's traveler expects and demands safe, reliable, consistent, and clean service, and of course the industry's objective and responsibility are to provide this. However, forces over which the industry has little control determine whether people fly and, if they do, where they fly. These forces include international relations, currency valuation, fuel costs, and war-risk insurance.

Another issue is regulation. The Airline Deregulation Act of 1978 spawned many new carriers, lower ticket prices, and lower profit margins. In some cases current regulations (1) dictate air service to nonprofitable locations and preclude service to potentially profitable cities, (2) restrict foreign equity capital, and (3) complicate computer reservation rules. Nevertheless, hub-and-spoke carriers are critical if smaller communities are to grow (Bethune, 2003).

Anderson (2003(b)) asks what is fair taxation for the airlines. Airlines pay nearly $9.3 billion in annual taxes, double the tax burden placed on the cigarette industry and about 26 percent of the average ticket price (i.e., ticket tax, security fees, and passenger facility charges). A sky marshal seat is lost revenue as is a regulation that restricts the airlines from carrying mail weighing more than 0.45 kg (16 oz). For one U.S. airline alone that mail restriction amounts to a $30 million loss in annual revenue.

These are the realities of some forces that control the aircraft and turbine engine industry and, in a similar manner, the industrial-turbine industry as well. Turbomachine sealing and secondary flows only marginally affect these controlling forces but must provide for safe, reliable, consistent, fuel-efficient, reduced-emission (e.g., noise, oxides of nitrogen (NO_x), and other pollutants) powerplants. With this in mind we now focus our attention on turbomachine sealing and secondary flow issues.

The Customer View

Many airline and aerospace turbomachine sealing issues emerged from the 1978 Advisory Group for Aerospace Research and Development (AGARD) symposium and continue to be of concern today. Smith (1978), Soditus (1999), and Uhl (2001) point out (1) the importance of reliability, durability, and repairability and (2) the necessity of feedback communications between user and designer with specific fuel consumption (SFC) as the cost driver. Although Soditus, Uhl, and Smith differ on specifics, the maintenance trends and costs remain similar. Smith indicates that a fuel flow deterioration of 4 percent after refurbishment is normal and for a 12-aircraft, high-bypass-ratio fleet is equivalent to an expenditure of $1.2 million per year (in 1977 U.S.$) at a fuel cost of

$0.0924 per liter ($0.035 per gallon). Further, if a fleet of four-engine aircraft lost 0.254 mm (0.010 in.) in turbine tip clearance, the fuel usage would increase some 0.6 percent at an additional cost of $0.2 million per year (in 1977 U.S.$) as shown in figure 1. These percentage losses in fuel flow are not directly comparable, but the trends are similar to current estimates of performance degradation with engine refurbishments (fig. 2).

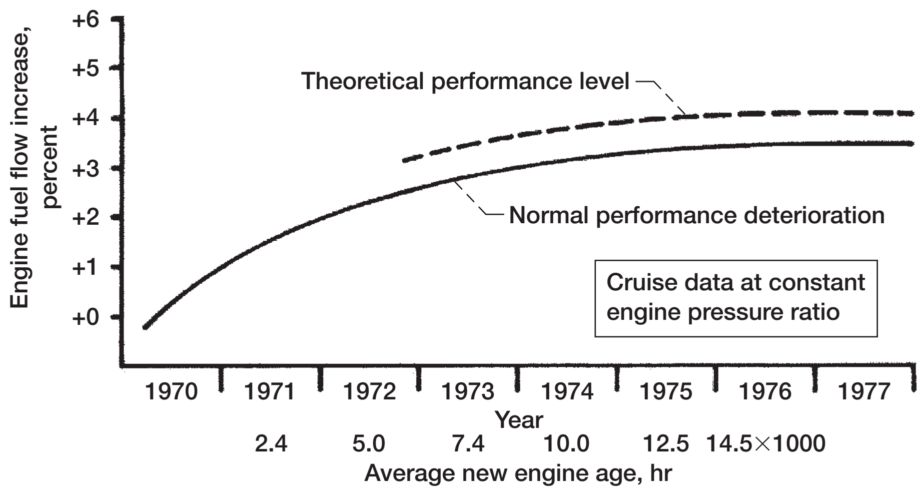

Figure 1.—Example of performance deterioration after normal refurbishment efforts for fleet of aircraft using large commercial high-bypass-flow engines. The dashed line shows the theoretical degradation if the turbine radial tip seal clearance were increased by 0.254 mm (0.010 in.) on all engines. (Smith, 1978.)

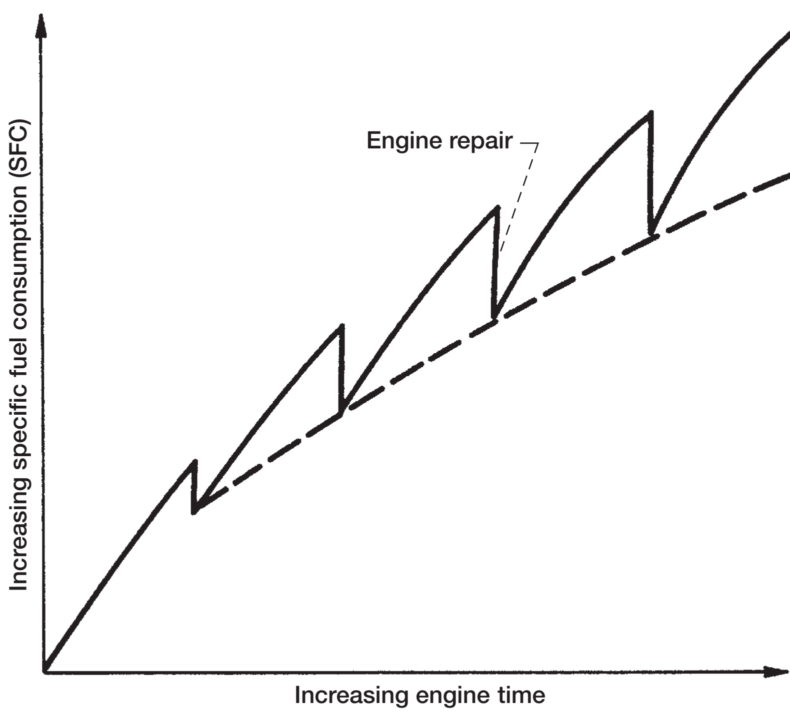

Figure 2.—Performance deterioration. (Ludwig, 1978.)

Some performance degradation arises from pressure problems across oil seals that cause higher oil consumption, even in test rigs, and require very close balance to operate without throwing out all the oil. Smith (1978) goes on to cite seal cracking in the transient wiping of abradable seals as a durability issue; transient wiping is exacerbated by high maneuver loading and the rotordynamics associated with high vibration levels and excessive rotor runouts. Case out-of-round is also a major issue. Seal failures are driven by thermal gradient fatigue or axial and radial thermal expansions during maximum power excursions. In bearing compartments 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.).

Smith (1978) summarizes his overview of the sealing cost effect (in 1977 U.S.$) on the U.S. fleet (fig. 3) as follows: seal deterioration costs 1.5 percent of the fuel bill ($5.85 million), engine entry and repair cost 15 percent of the total engine labor ($2.1 million), and seal material and parts cost $6 million. Smith estimated the total annual cost attributed to engine sealing problems at $14 million, or $14 million/0.33 = $42.42 million in 2003 U.S.$.[1] Smith also suggests that next-generation engines have two clearance settings—one for takeoff and one for cruise. In most aeronautical and aeronautical-derivative gas turbine engines, case cooling is now used to control tip clearances. In some aerospace turbomachines prechilling or preconditioning of the case or turbine is used.

Although in-flight shutdowns have diminished considerably over the years, a major sealing issue of concern is their effect on the compressor. Some older engines and some new ones have compressor surge or stall margin problems caused by "tired old seals." A major benefit can be achieved by reworking both older and new engines with new or refurbished seals. Refurbishing today's engines to maintain engine performance, reliability, durability, and repairability is, and has been, a major industry objective. As an example, in 2001 United Airlines had over 1500 powerplants and 650 auxiliary power units in their fleet service (Uhl, 2001), and the issues are virtually the same as those cited by Smith (1978).

The decreasing trend in SFC over the years (fig. 4) has been and will continue to be modulated by environmental constraints enforced by many regulatory agencies, especially as relates to emissions (e.g., noise and NO_x). However, the demand for lower emissions represents an added opportunity for seal designers (Hendricks, 1994).

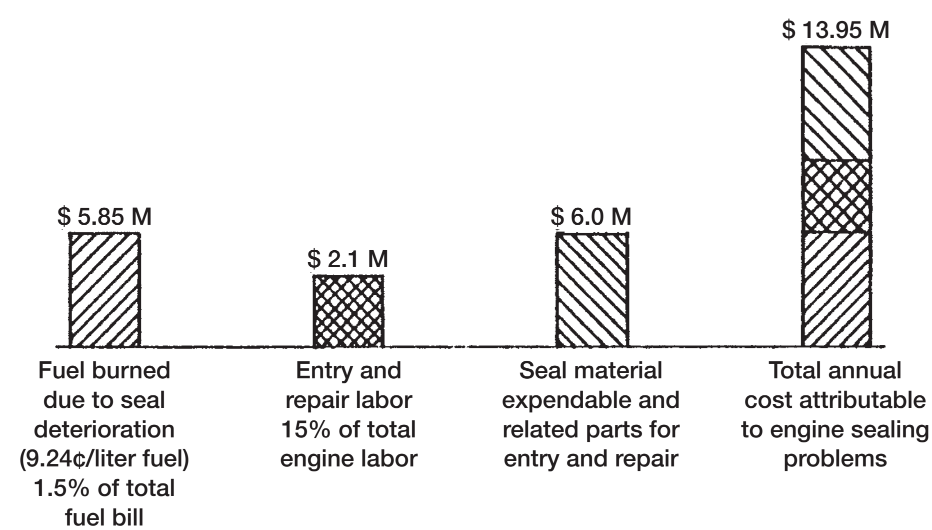

Figure 3.—Annual total cost effect from seal and seal-related problems for large U.S. airline. (Smith, 1978.)

[1]http://oregonstate.edu/Dept/pol_sci/fac/sahr/cv2003.pdf

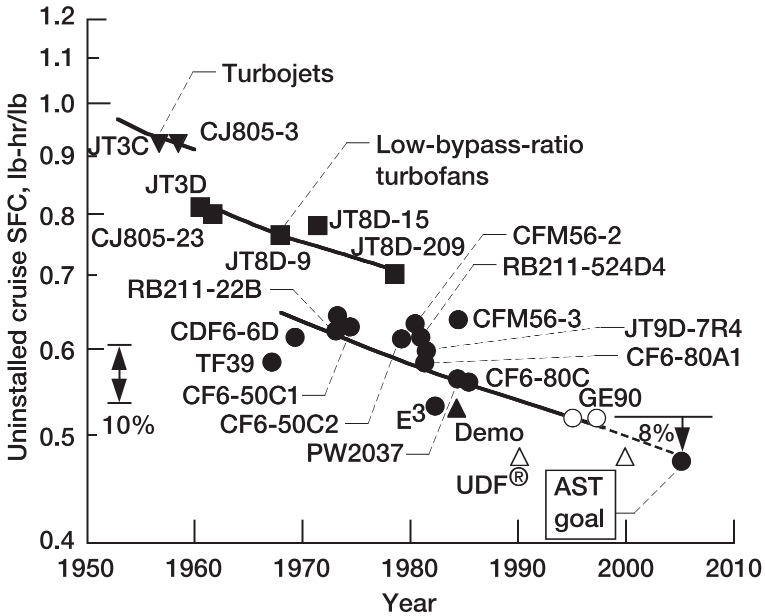

Figure 4.—Subsonic engine historical trend and program goal for specific fuel consumption. (Steinetz et al., 1999.)

Engine performance takes a major hit, up to 5 percent, with degrading seals, especially on the high-spool gas generator (high-pressure compressor (HPC), high-pressure turbine (HPT), and combustor), and operations near the exhaust gas temperature (EGT) margin. The refurbishment of engines costs several million dollars (in 2000 U.S.$) per engine with turbine blade and major component replacements most costly and seals least costly but most effective in restoring performance. As cited in part 3, engine externals are the leading cause of in-flight shutdowns and require constant attention.

The Engine Designer View

Turbomachines are complex systems with secondary (internal) fluid flows (fig. 5) that enable the turbomachine to operate with cool and collective calm but at a cost to the cycle in terms of performance. Even though other seals are being employed (e.g., brush seals), turbomachine systems still depend heavily on labyrinth seals and require good design, performance, and reliability.

Secondary Flow System

According to Campbell (1978), and current practice, the secondary flow system encompasses

1. The flow for the turbine aerofoil and working annulus wall cooling (blades, nozzle guide vanes, rotating and static blade shrouds, blade and vane platforms, and case)
2. The disk sealing and cooling system that protects the disks and blade roots from excessive temperatures and controls end loading of the rotor thrust bearing
3. The bearing sealing system that interacts with disk sealing systems to maintain satisfactory environments in the bearing compartments

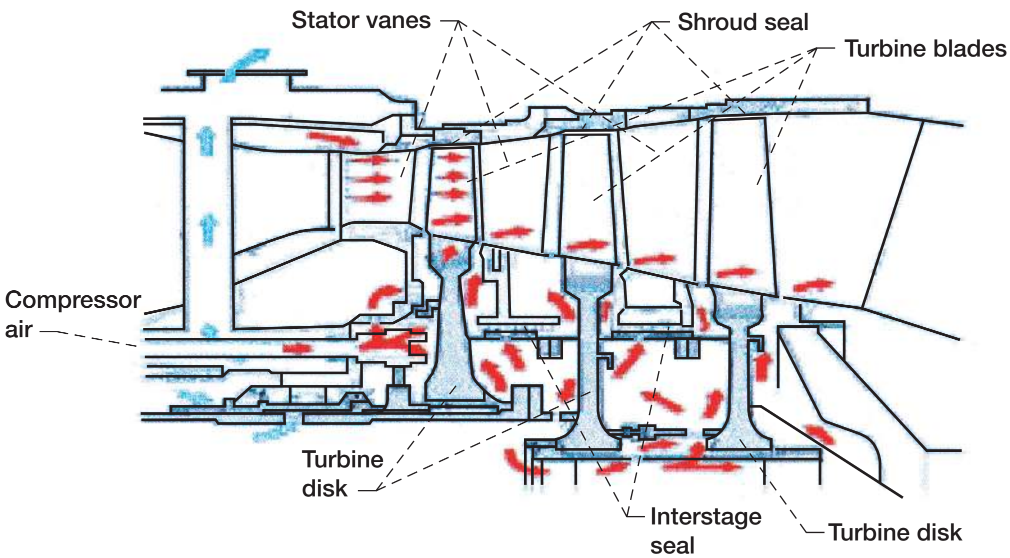

Figure 5.—Hypothetical turbine secondary-air cooling and sealing. (Allcock et al., 2002.) Courtesy AIAA.

4. The parasitic leakage or any unwanted leakage between parts with low relative motion, such as in blade and nozzle guide vane platform gaps, shroud segment gaps, etc., that degrades engine performance

Campbell goes on to say that the secondary flow cooling and purging objectives can readily be met by large quantities of bleed air (fluid), which, however, increases turbine entry temperature (TET) and SFC—so the bleed requirements must be optimized. Multistage bleeding of compressor air is much more efficient than using HPC delivery air (fluid). For example, a 2.1 percent SFC penalty for HPC delivery air can be reduced to 0.9 percent by using multistage bleeds (Campbell, 1978). A further advantage of multistage bleeding is that the lower the stage the cooler the air but at a lower available pressure and fluid density. Turbine efficiencies are reduced by sealing flows entering immediately in front of the rotor, underscoring the importance of rim sealing (see also part 3). Campbell stresses that with careful blade and nozzle platform design, efficiency losses can be under 0.3 percent for a sealing flow of 1 percent of turbine flow (many times not included in turbine performance calculations). This effect increases as the core performance continues to deteriorate over the life of the engine. For a fixed core the SFC effects are small, but TET increases 10 deg C for 1 percent of engine bleed. These numbers reflect the importance of making realistic, rather than hopeful, sealing air requirements to ensure the proper flows over the life of the production engine.

Power-Stream Interfacing

Gas ingestion into turbine cavities is determined by circumferential variations in static pressure, including local variations of blade passing and radial flows by disk pumping caused by friction that engenders large changes in disk heating (e.g., a 700 deg F increase due to disk rotation). This heat must be carried away by air supplied for the relevant seals and cavities. Overheated purge air can destabilize the platform airflows (see also part 3 and appendix B of part 2). A major problem centers on engine dynamics (part 2) whereby the rotor and case deflections follow a complex pattern, as illustrated in the x-ray work of Stewart and Brasnett (1978) as shown in figure 6.

Variations in flows upstream of the combustor, which ensue from the compressor and nonuniform distribution within the combustor diffuser, are redistributed by the nozzle guide vanes and

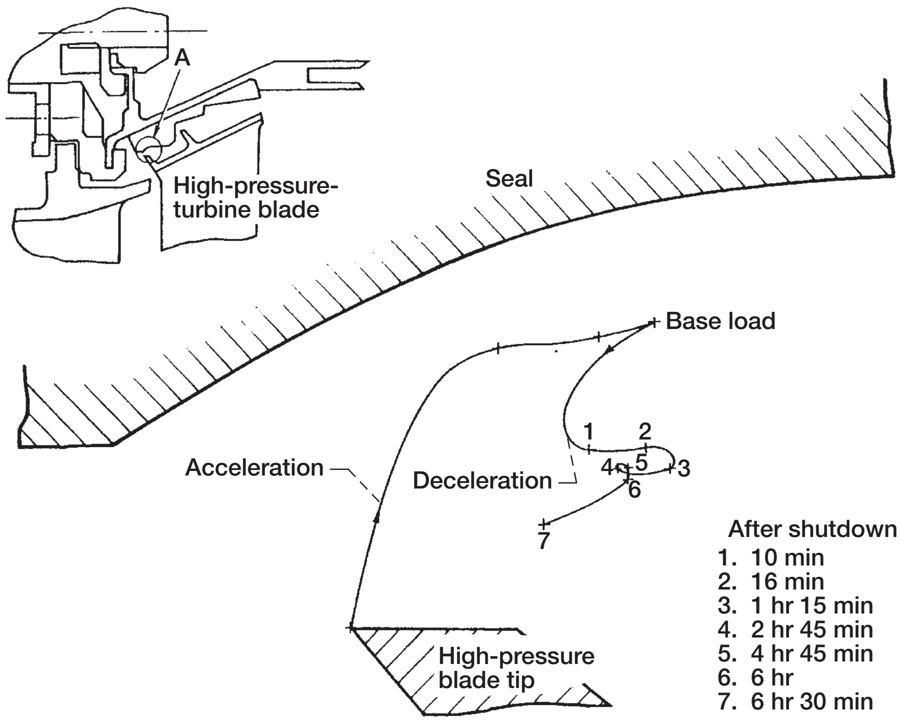

Figure 6.—Enlarged view (scale 50:1) of area A. Locus of high-pressure-turbine (HPT) blade tip seal relative to static seal during acceleration and full-power trip conditions. (Stewart and Brasnett, 1978.)

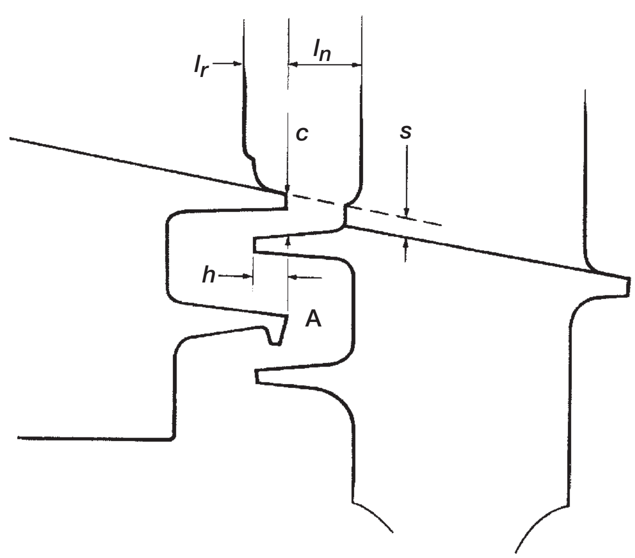

Figure 7.—Generic turbine nozzle rotor gap configuration. (Campbell, 1978.)

vibrate the blades. Variations in nozzle and blade areas and curvature of trailing edges all lead to pressure fluctuations and seal flow perturbations. As a result some gas ingestion into the disk cavities is anticipated. A goal is to minimize this ingestion and reduce these pressure fluctuations. Figure 7 shows that decreases in clearance *c* would minimize leakage ingestion but would make it difficult to hold tolerances in engine manufacturing. Operations with pressure and thermal expansion or contraction of the components lead to difficulties in balancing rotor thrust. Overlap *h* and shingle height *s* tend to minimize local variations in pressure and reduce the effects of cavity flows (see also Teramachi et al., 2002). *It is better to alter the seal geometry to permit some ingestion within the local sealing chamber (*A *in fig. 7); such changes balance the pressure perturbations and minimize the turbine losses.* Disk flow Reynolds numbers are high, of the order of 10^7, with high radial and axial flows. Rotation generates the outward flow to be balanced at the power-stream interface. Many obstacles in the flow path (e.g., bolts, holes, and curved surfaces) make the flow field difficult and time consuming to capture even with today's codes and standards (cited in part 3). Friction heating windage effects must be balanced by the cooling air, and preswirling the cooling air is useful in achieving this balance.

Each disk cavity has one or more flows metered by seals that supply or remove fluid from the cavity. Consider figure 8 where a fixed metering slot denoted by X supplies a fixed amount of air (seal deterioration will not change this flow). Although seal A removes air from cavity 1 and supplies cavity 2, pressures in cavities 1 and 2 are primarily determined by the nozzle guide vane inlet and outlet pressures. If the clearance at seal A increases, the flow increases and starves the annulus; gas ingestion ensues, causing overheating; and hotter gases spread to other parts of the system. The effect is exacerbated when seals have been optimized for single-point design.

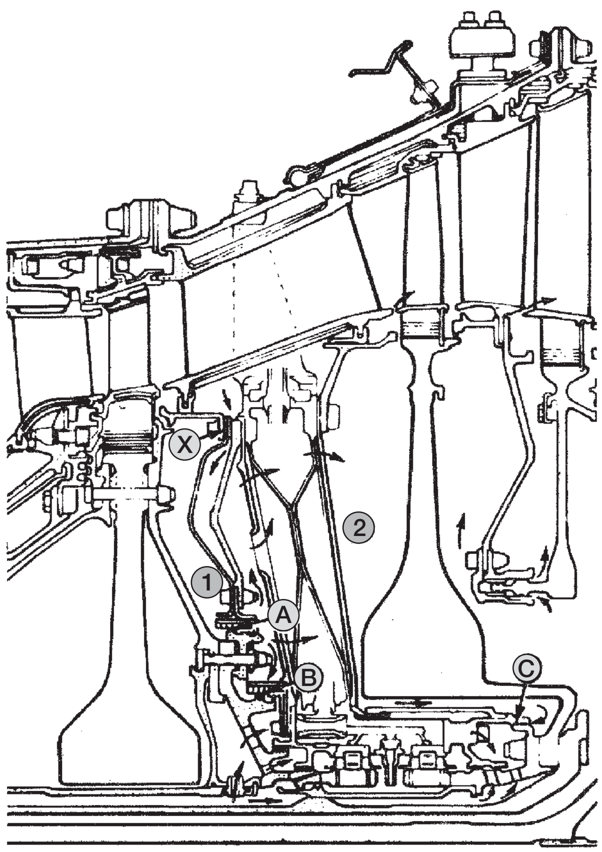

Figure 8.—Cooling airflows and system balancing. (Campbell, 1978.)

Predicting these disk labyrinth seal clearances is difficult, especially in estimating transient behavior and such features as slippage and wear of bolted joint connections. The engine is required to operate with a starved-annulus seal failure. A failed seal usually has a nominal running clearance that is twice the fixed metering slot, and the supplied flow balances the zero sealing flow at the starved annulus. This criterion establishes the basic flow requirements, and additional flows are added to deal with windage and ingestion. The additional flows increase disk temperatures and hence decrease disk reliability. Reliable flow control is preferred over minimizing leakage and is only achieved if the seal, rotor, and stator design is considered as a synergistic system particularly considering the transients.

Blades, Vanes, and Drums

Blade and vane cooling requires passing preswirled cooling air with a minimum addition of leakage air through the blade or vane. Leakage air mixing with preswirl adds to windage heating and reduces whirl velocity. One method of minimizing additional leakage air is to isolate the bleed leakage to cool vane platforms and trailing edges.

Why would one want the HPC and HPT on the same drum? For one thing the net thrust loads require balancing. However, the differential can still be large enough to cause thrust bearing problems, particularly for multishaft, high-bypass-ratio systems where the high loading on bearing balls reduces thrust capacity. Adjusting the loading by altering seal radii usually leads to increased leakages. Larger capacity thrust bearings can improve engine performance, an early factor to be considered in selecting design parameters for the high-pressure system.

Leakage Reductions and Engine Cycle

Seal research and development is typically directed toward reducing leakage, although, as cited previously, that is not always the important objective. Static pressure variation and disk windage in the high-pressure-system annulus determine sealing and cooling flows. At the initial engine design stage bleeds have a relatively small effect on performance. If disk sealing air is reduced from say 3 percent to $1\frac{1}{2}$ percent (50 percent decrease) a $\frac{1}{2}$ percent SFC savings would result, with TET changes eliminated by the choice of bypass ratio. However, once cycle parameters are fixed, the effect of bleed air changes on TET is large. Increasing that same 3 percent airflow to $4\frac{1}{2}$ percent (50 percent increase) would increase TET by 15 deg C, reducing HPT creep life by about 30 percent—underscoring the necessity of realistically estimating disk sealing and cooling airflow and monitoring these flows during development and in field operations.

Of course, fatigue and vibration problems are associated with blade excitation due to variation in upstream nozzle pressures and operational loading (see also parts 2 and 3). Too often overlooked are static sealing issues of nonrotating components and within a rotating component; they remain a big problem. Wire and strip (feather) seals are used, but sealing heavily loaded flanges and guide vanes remains a problem. With concern over O-ring seals sticking in the compressed state, many prefer to use clamped surfaces. These secondary seal concerns should be remembered early in the design phase.

The Research View

The government-university-industry complex has been and remains to some extent heavily involved in seals and secondary flows research. Most innovative techniques are held proprietarily, making communication difficult.

Engine Sealing Demands

An early pioneer of the seals industry was Larry Ludwig. His landmark paper (Ludwig, 1978) set the stage for many gas turbine sealing developments. Ludwig surveys gas path sealing, relating the effects of clearance on efficiency, the effects of gas path sealing on compressor stall margin, and the effects of nonuniform clearance on rotor stability. He sets engine SFC deterioration at 1 to $1^1/_2$ percent per year and estimates that periodic overhauls (fig. 2) do not fully recover SFC, resulting in 3 to 10 percent increases in SFC over that of a new engine, with the deterioration even greater for military engines. In addition to engine efficiency seal clearances also directly affect compressor stall margin and are responsible for "thrust-droop"[2] to 12 percent.

Because for labyrinth seals the gain from more than six teeth (throttles) is small, most labyrinths have six or fewer teeth, with tooth edge sharpness and shroud rubbing contributing to flow losses. Gas turbine engines have over 50 dynamic sealing locations as well as static seals at vane pivots, flanges, duct joints, and blade roots. When one considers the airframe, the number of seals is very large (Miller et al., 1998). Blade tip wear in compressor and turbines becomes a major contributor to engine deterioration. Small engines are a challenge from both sealing and performance viewpoints. What is needed are improved and active clearance control, erosion-resistant abradable materials, and increased energy dissipation in labyrinth seals. Clearance control holds the most promise but requires careful case and rotordynamics control (see part 2). Operation at very tight clearances leads to high rates of deterioration. Abradables should not stick to the blade, should produce little blade wear, and should be erosion resistant with an impermeable, aerodynamically smooth surface. Tip sealing, as cited previously, is a major problem (see also part 3).

Dynamic sealing, especially of the high spool, remains a critical issue and in some of today's high-performance turbomachines makes the difference between achieving operability or not.

Engine Performance

Hendricks et al. (1994) reported engine testing results *conclusively demonstrating that changes in the compressor discharge seal performance changed flows throughout the entire engine.* In separate series of YT–700 engine tests direct comparisons were made between the forward-facing labyrinth and dual-brush compressor discharge seals. The labyrinth bill-of-materiel (BOM) seal tested is shown in figure 9, and the dual-brush replacement seal is shown in figure 10. Compressor speeds to 43 000 rpm, surface speeds to 160 m/s (530 ft/s), pressures to 1 MPa (145 psi), and temperatures to 680 K (765 °F) characterized these tests. Brush seal wear for 46 hr of engine operations was estimated at less than 0.025 mm (0.001 in.) of the Haynes 25 alloy bristles running against a chromium-carbide-coated rub runner. The brush seal demonstrated low hysteresis as evidenced in figure 11, which compares the labyrinth and brush seals' leakage performance. The pressure drops were higher for the dual-brush seal than for the forward-facing labyrinth seal and leakage was lower. The labyrinth seal leakage was $2^1/_2$ times greater, implying that the dual-brush had better seal characteristics, better secondary airflow distribution, and better engine performance. With the dual-brush seal engine experimental SFC was 3 percent better than with the BOM labyrinth seal at high pressure and 5 percent better at lower pressure (fig. 12). However, as brush seals wear down (after 500 to 1000 hr of engine operation), their leakage rates will increase. In some cases new sealing techniques offer potential advantages as with the aspirating seal (see appendix B of part 2). In either case, modification of the secondary flow path requires that changes in cooling air and engine dynamics be accounted for, as cited previously in the Leakage Reductions and Engine Cycle section.

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[2]Rotor lags thrust demand or throttle position setting.

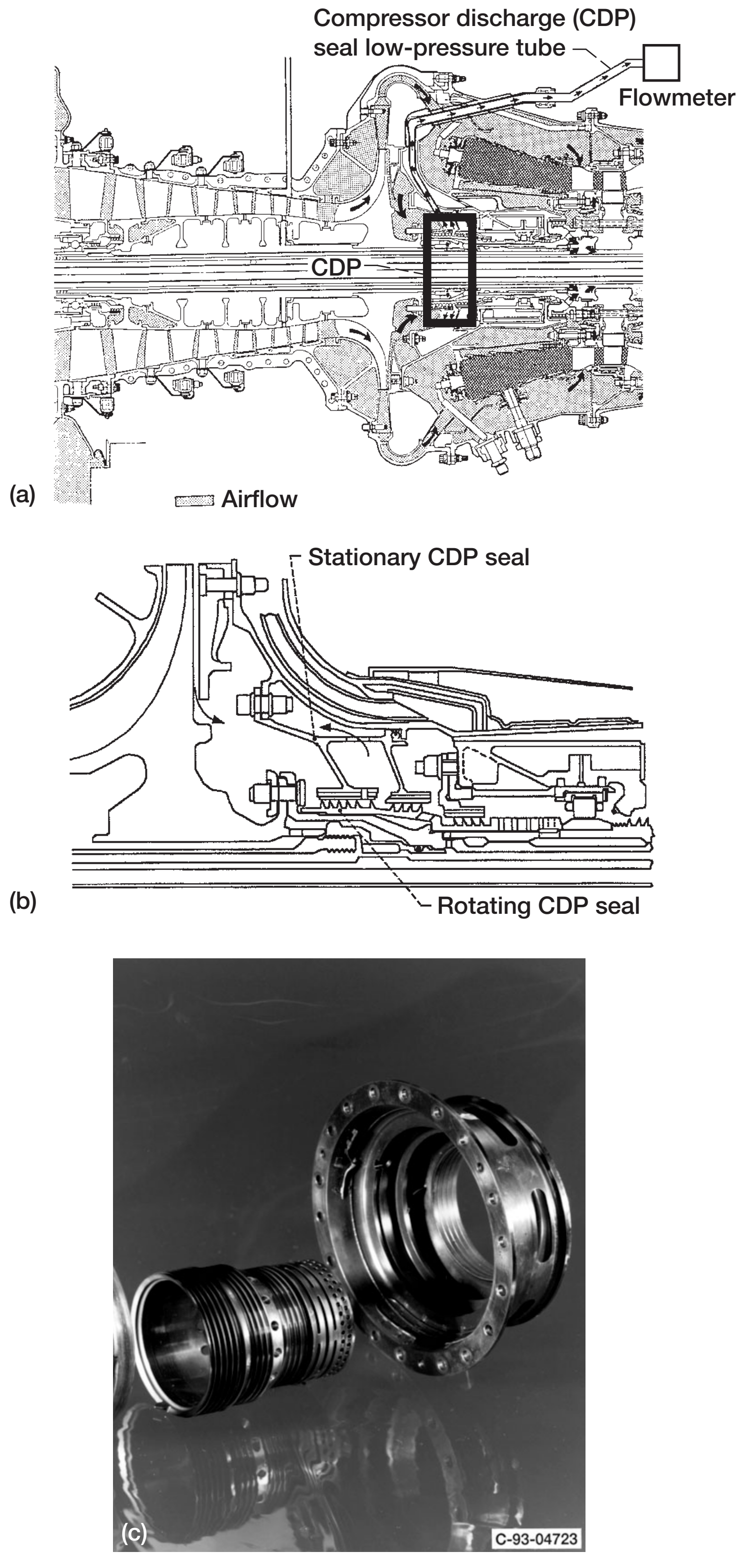


Figure 9.—Labyrinth compressor discharge pressure (CDP) seal. (a) Engine airflow and seal location. (b) Labyrinth seal package and airflow. (c) Simulated exploded view of labyrinth compressor discharge seal system. (Hendricks et al., 1994.)

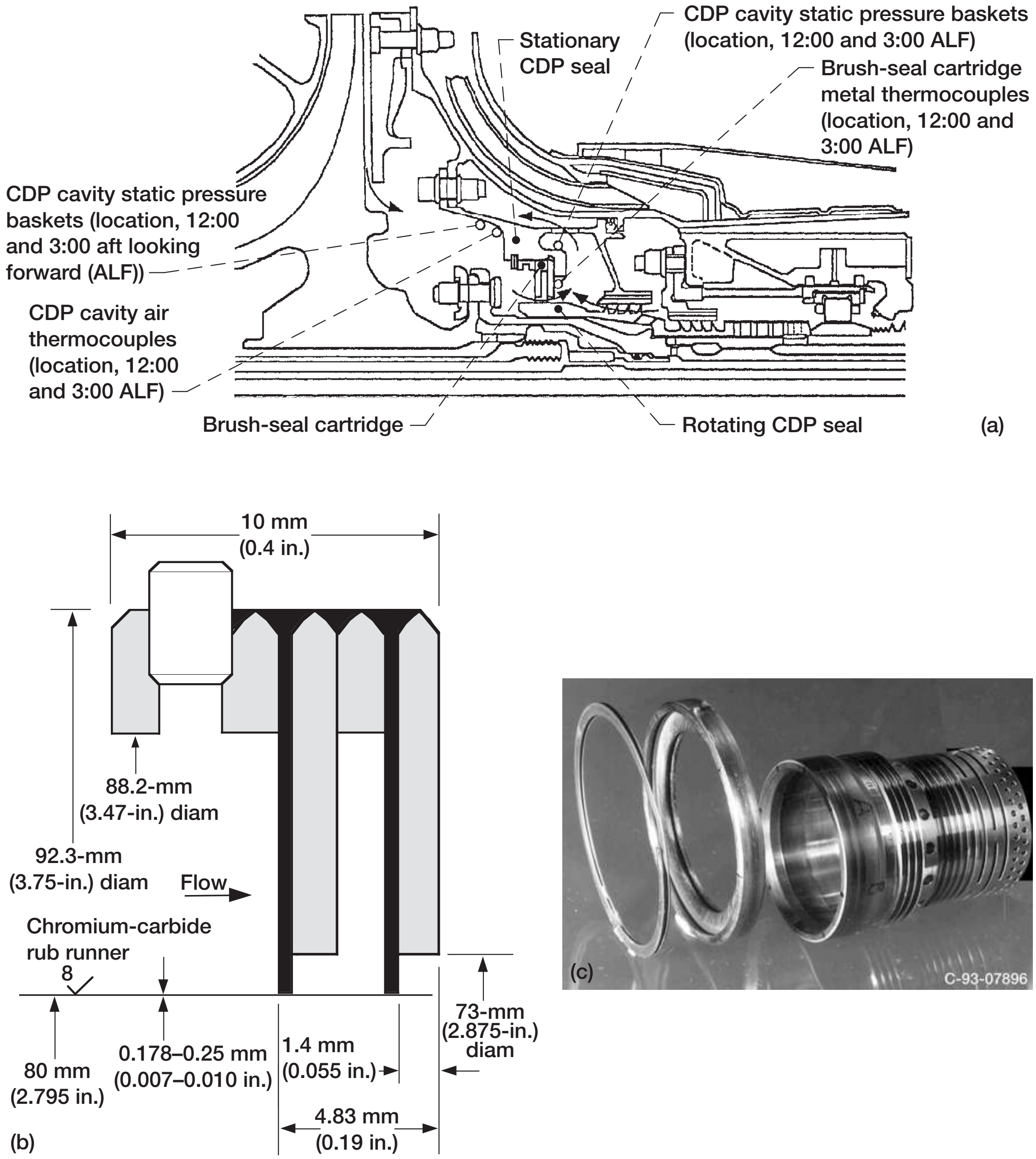


Figure 10.—Brush compressor discharge pressure (CDP) seal. (a) Brush seal package and airflow. (b) Illustration of dual-brush compressor discharge seal system. (c) Simulated exploded view of labyrinth compressor discharge seal system. (Hendricks et al., 1994.)

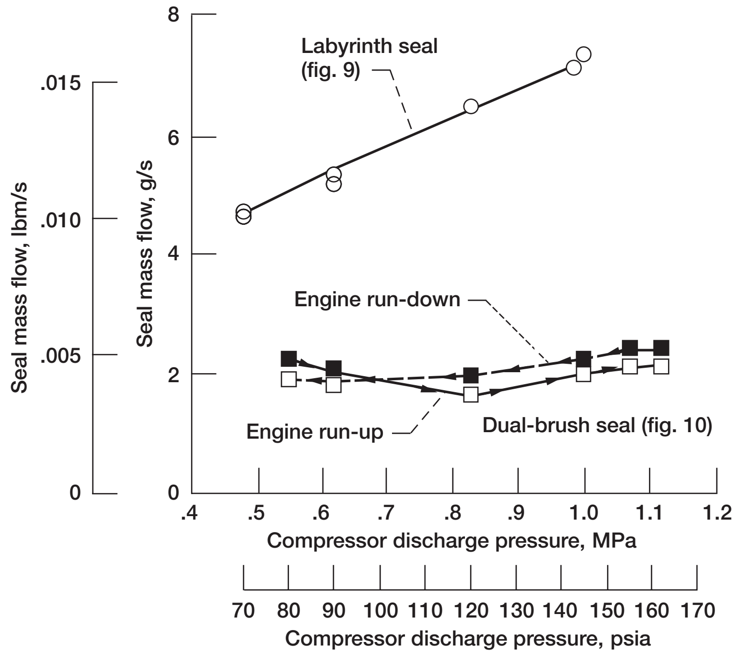


Figure 11.—Seal mass flow as function of compressor discharge pressure for labyrinth and dual-brush seals. (Hendricks et al., 1994.)

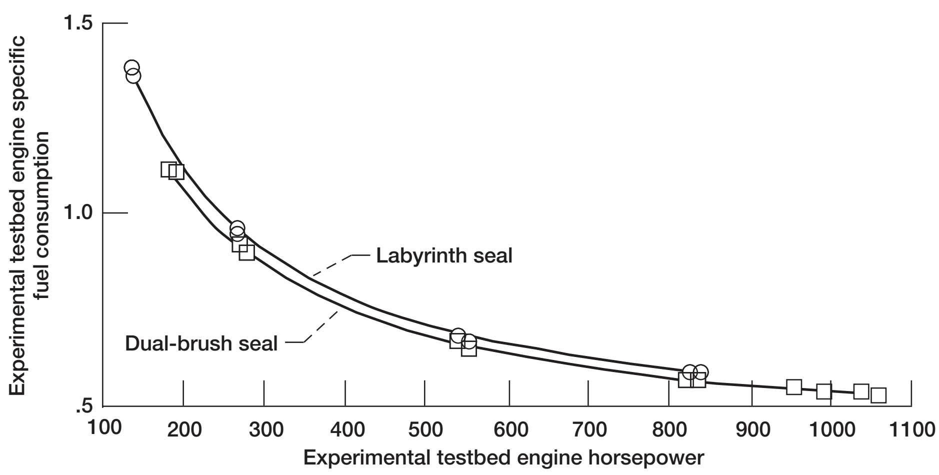


Figure 12.—Experimental testbed engine specific fuel consumption as function of horsepower. (Hendricks et al., 1994.)

Current Sealing Practice and Objectives

Seal Benefits

Performance issues are closely tied to engine clearances. For example, turbine radial growth over the flight cycle includes engine start and takeoff with and without active clearance control (ACC); and ACC is not always functional. Cruise and hot reburst (following a throttle chop) at altitude also occur with and without ACC. These casing and rotor thermomechanical growth and decay issues along with case ovalization and rotor offset are detailed by Davis and Stearns (1985) and Cherry et al. (1982).

Lattime and Steinetz (2002) review some current sealing practices in gas turbine engines. Although the sealing clearances vary over a wide range during the flight cycle, they cite a 0.0254-mm (0.001-in.) change in HPT tip clearance (fig. 13) causing 0.1 percent decreases in SFC and 1 deg C decreases in EGT. Even a 0.1 percent SFC decrease at an annual usage of 16 billion U.S. gallons of aviation fuel results in huge monetary savings as also noted by Smith (1978), Stocker (1978), and Ludwig (1978) (fig. 14). Lattime and Steinetz also cite controlled oxide growth or swelling of shroud material with operating time as potential future seal refurbishment methods. They also cite a need for rub-avoidance methods, presumably by limiting incursion rates through active control. As of this time both schemes are conceptual.

Industrial Gas Turbine Sealing Challenge

As stated previously, one purpose of the seal is to control leakage; another is to maintain a supply of cooling and purge flows into the hot section (fig. 15). Typical labyrinth seals are illustrated in figure 16 with sealing locations shown in figure 17. Typical aeronautical engine compressor sealing locations and applications of abradables are shown in figure 18. Schematics of three types of compressor rub materials with associated incursion types are illustrated in figure 19 for an outer air/blade tip sealing interface in a compressor. These types of materials usually differ from the inner shroud/drum rotor sealing interface in a compressor as illustrated in figure 20.

Chupp et al. (2002) address the tip sealing problems for both the low-pressure and high-pressure spools of industrial gas turbines, with sealing locations illustrated in figure 21, a typical shroud seal element in figure 22, and a typical turbine installation in figure 23. The authors discuss general classes of abradables: (1) a low-pressure compressor (LPC) to 400 °C (750 °F), (2) LPC and HPC ambient to 760 °C (1400 °F), (3) high temperature (to 1150 °C (2100 °F)) for HPT, and (4) application methods (castings for polymer bases, brazing or diffusion bonding for honeycomb or fiber metals, and thermal spray for powdered composites).

The blade tip rub mechanism and material release below surface speeds of 100 m/s is primarily forward-expelled chips; above 100 m/s the release is rearward-expelled particles. Therefore, the cutting tip needs to be thin (say 1 to 3 mm) because thicker tips trap materials. Some abradables are compacted when wear of the rubbed blade tip increases and abradable porosity decreases. Other abradables, such as honeycomb, deform at high speeds and the cell walls will rupture. Ideally, one would want some rubbing over the entire circumference to optimize the running clearances. Because each system differs, each abradable interface has to be tribologically designed for the specific application. Chupp et al. (2002) provide some rub-rig test results but with insufficient detail to be more than a placeholder in one's mind. Nevertheless, the results are impressive, with a fleet range of 0.2 to 0.6 percent reduction in heat rate and 0.3 to 1 percent increase in power output. For these large land-based gas turbines these percentages represent huge monetary returns with the greatest return cited for aging power systems.

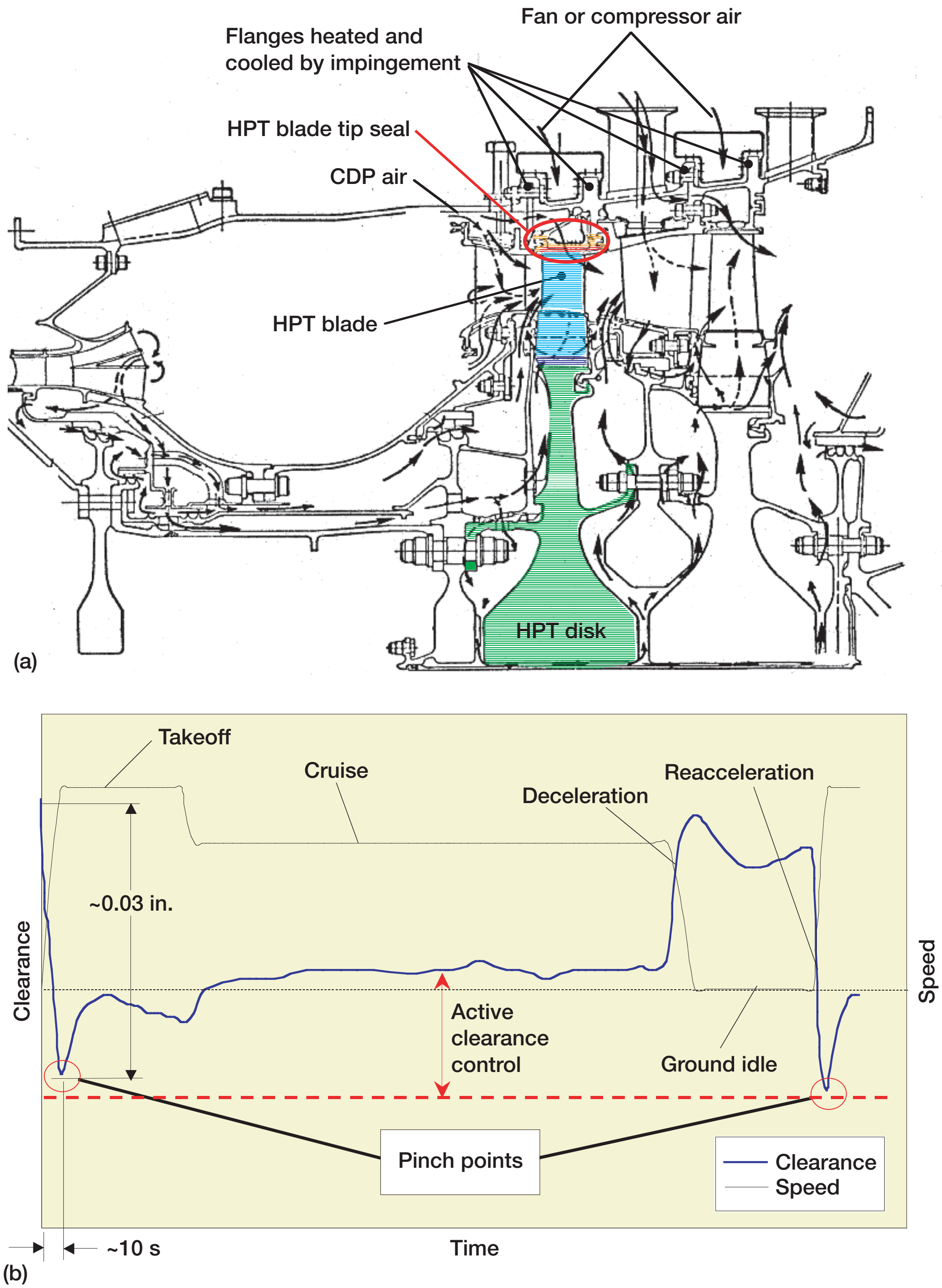


Figure 13.—High-pressure turbine (HPT) and cooling air. (a) HPT blade tip seal location in modern gas turbine engine. (b) HPT tip clearance as function of time over given mission profile. (Lattime and Steinetz, 2002.)

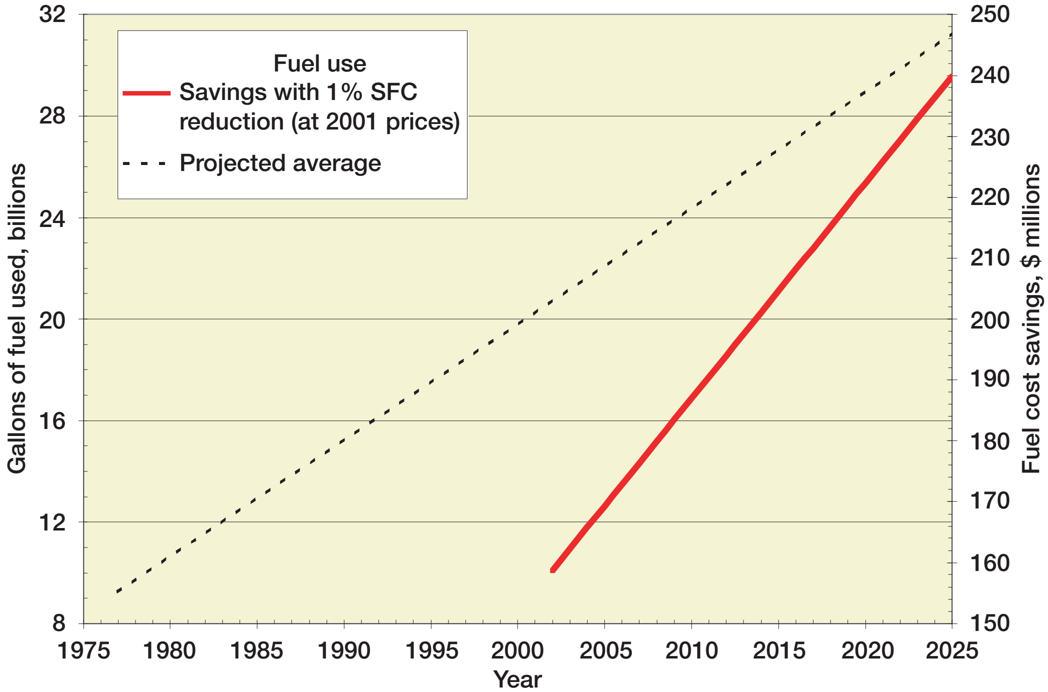


Figure 14.—Past and projected fuel use and projected cost savings for 1 percent reduction in SFC (U.S. fleet). (Lattime and Steinetz, 2002.)

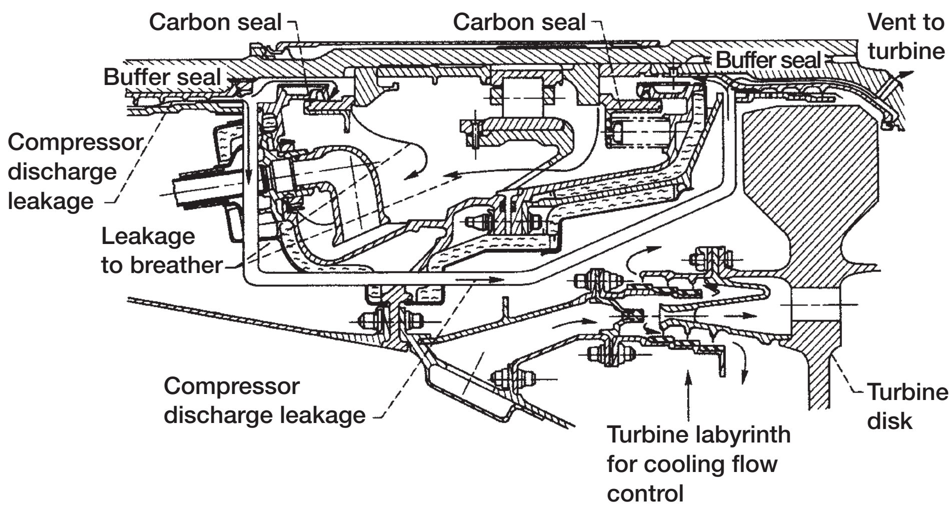


Figure 15.—Main-shaft and high-pressure-turbine labyrinth seal. (Ludwig, 1978.)

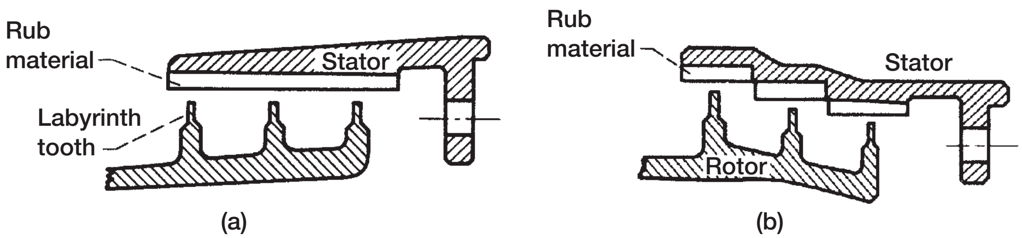


Figure 16.—Labyrinth seal types. (a) Straight. (b) Stepped. (Ludwig, 1978.)

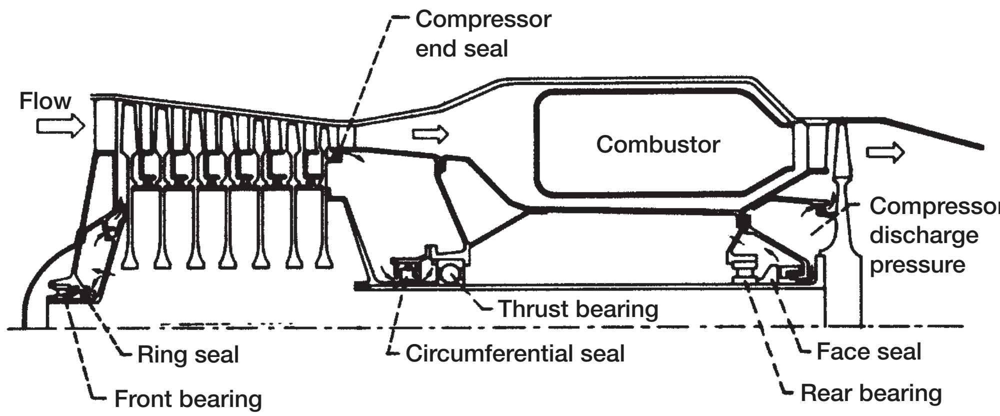


Figure 17.—Engine schematic showing main-shaft seal locations. (Ludwig, 1978.)

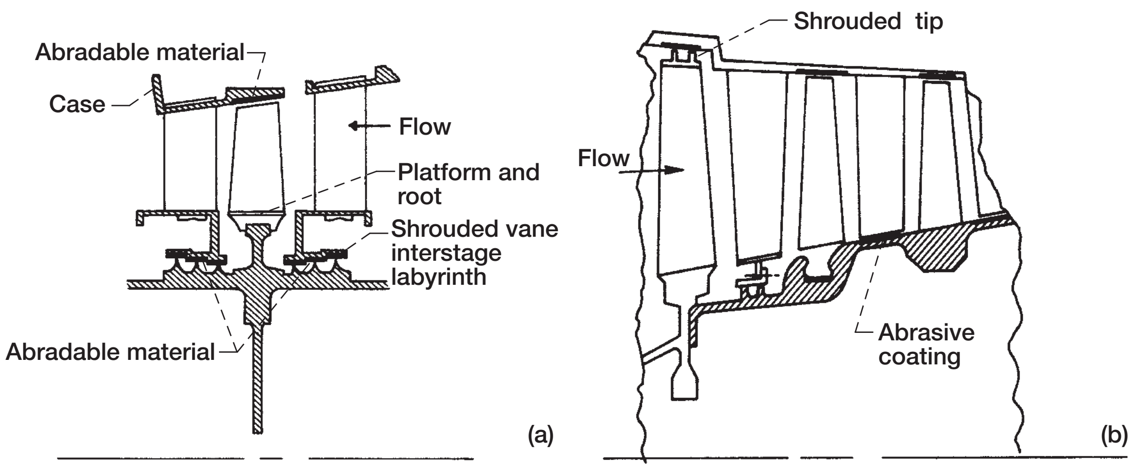


Figure 18.—Compressor sealing locations. (a) Blade tip and interstage. (b) Drum rotor. (Ludwig, 1978.)

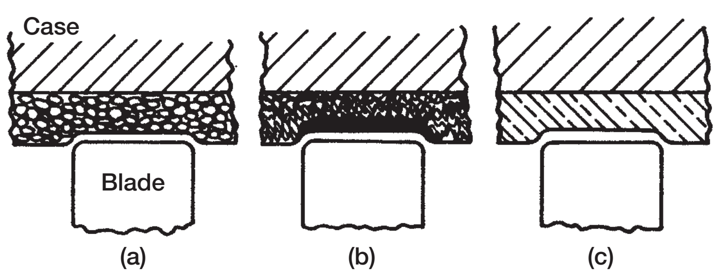


Figure 19.—Illustration of types of compressor rub materials for outer air sealing. (a) Abradable (sintered or sprayed porous materials). (b) Compliant (porous material). (c) Low shear strength (sprayed aluminum). (Ludwig, 1978.)

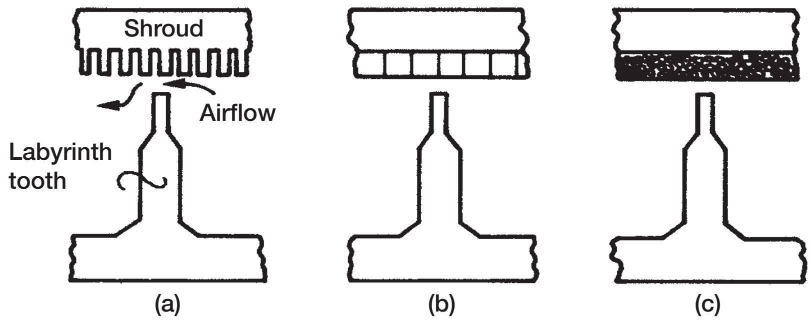


Figure 20.—Inner shrouds for compressor interstage labyrinths. (a) Striated. (b) Honeycomb. (c) Porous material (abradable or compliant). (Ludwig, 1978.)

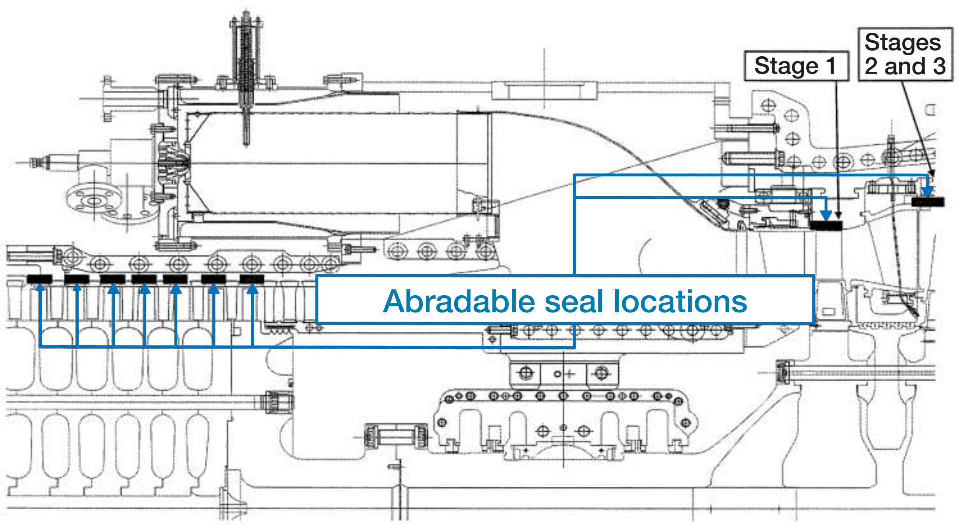


Figure 21.—Schematic of E-class industrial gas turbine showing focus locations of abradable blade tip sealing. (Chupp et al., 2002.) Courtesy GE.

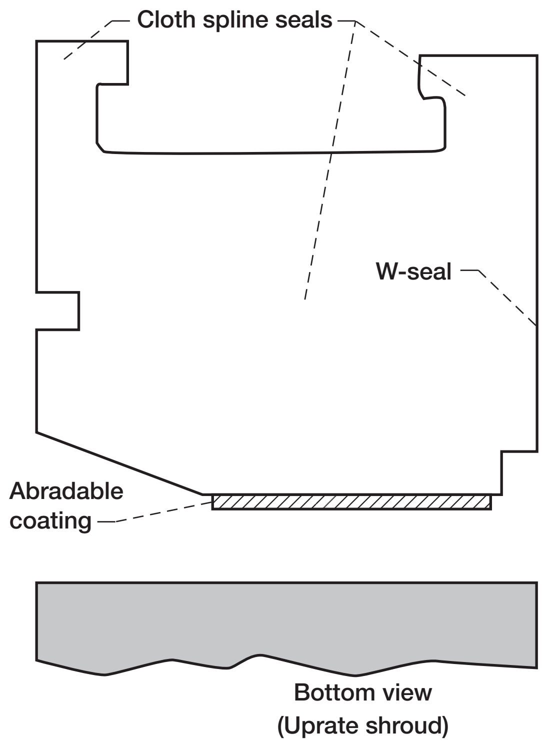


Figure 22.—Schematic of stage 1 shroud block with abradable material placed on surface opposite rotating blade tips. (Chupp et al., 2002.) Courtesy GE.

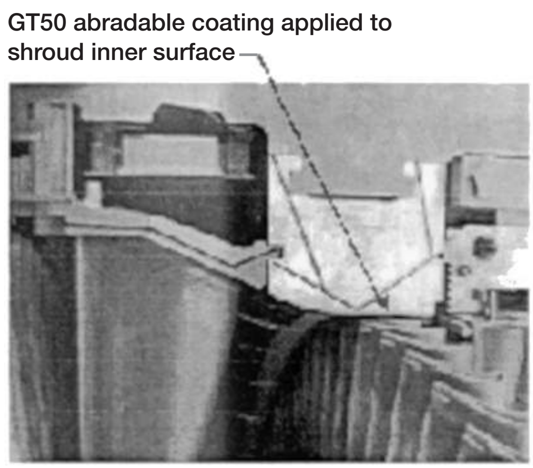


Figure 23.—Stage 1 shroud blocks installed in turbine. (Chupp et al., 2002.) Courtesy GE.

Shroud Sealing and Abradables Standards

Glen McDonald at NASA Lewis was the first to introduce ceramic shroud seals into an engine (fig. 24 from Biesiadny et al., 1985). BOM first-stage-turbine gas path shroud seals were replaced by plasma-sprayed zirconia, ceramic-coated seals (fig. 25). The coatings were 1-mm (0.040-in.) ZrO_2-$8Y_2O_3$ over a 1-mm (0.040-in.) NiCoCrAl-based bond coat onto a BOM Haynes 25 substrate. The seals successfully ran for 1001 cycles between flight idle and high power as well as at steady state for a total of 57.8 hr. Characteristic "mudflat" cracking of the ceramic occurred at the blade interface, but measured back-side seal temperature reductions over BOM seals of 78 deg C (140 deg F) were noted, with gas path temperatures estimated at over 1205 °C (2200 °F).

Chappel et al. (2001a,b) address the need for standardization of abradable seals. Their work directly compares abradable materials in high-speed tests at 275 m/s (900 ft/s) with temperature to 482 °C (900 °F) and in low-speed tests at 60 m/s and room temperature. They compare fiber-metal, honeycomb, and thermal spray materials. The material characteristics are given in table I.

Figure 26 shows a generic fiber-metal material. At 15 to 30 percent porosity the ultimate tensile strengths (UTS) of sintered materials vary from 3.45 to 20.7 MPa (500 to 3000 psi) with UTS of 10.3, 15.5, and 19 MPa (1500, 2250, and 2750 psi) selected for high-speed testing. UTS values of 5.7, 6.9, 10.3, 13.8, and 17.9 MPa (827, 1000, 1500, 2000, and 2600 psi) were selected for low-speed testing. The honeycomb material was Hastelloy-X felt, 0.05-mm (0.002-in.) wall and 1.59-mm (0.063-in.) cell. The two thermal-sprayed coatings (conditions of bond coat and spray not described) were nickel graphite (0.75Ni-0.25C self-lubricating, suitable to 480 °C) and MNiCrAlY/hBN/PE metal (cobalt) alloy with hexagonal boron nitride (hBN) for lubricity and polyester (PE) for porosity.

Abradability and erosion resistance represent conflicting demands on the felt-metal seal, as illustrated in figure 27, and provide the designer with some flexibility. Low-speed (47 m/s; 154 ft/s) abradability tests showed no appreciable blade wear. To 10.3 MPa (1500 psi) UTS rub depths were clean. However, at 13.8 to 17.9 MPa (2000 to 2600 psi) UTS, networks of fine cracks appeared with

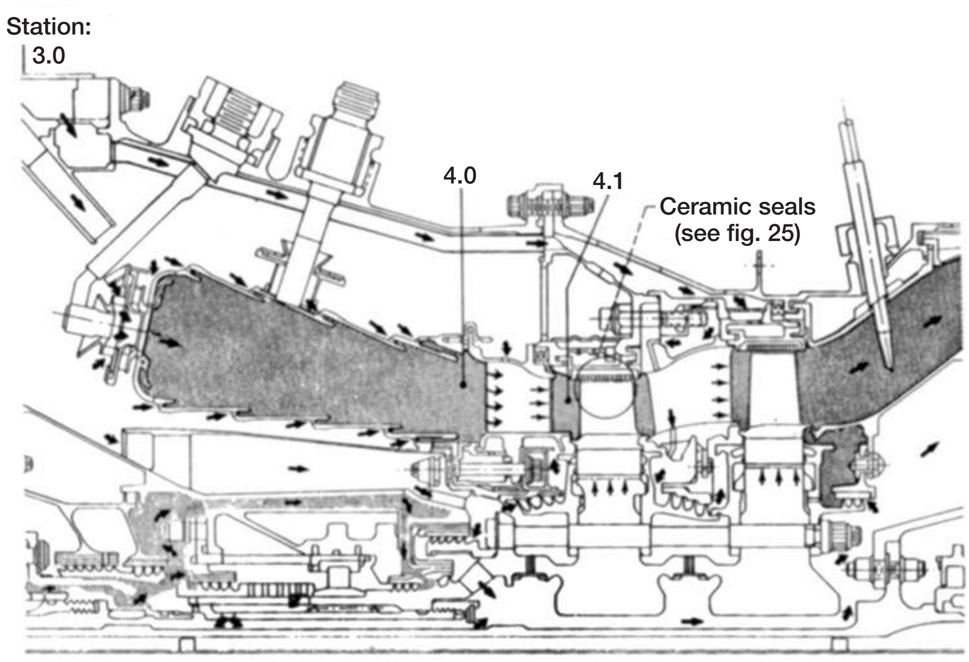


Figure 24.—Schematic of engine hot section. (Biesiadny et al., 1985.)

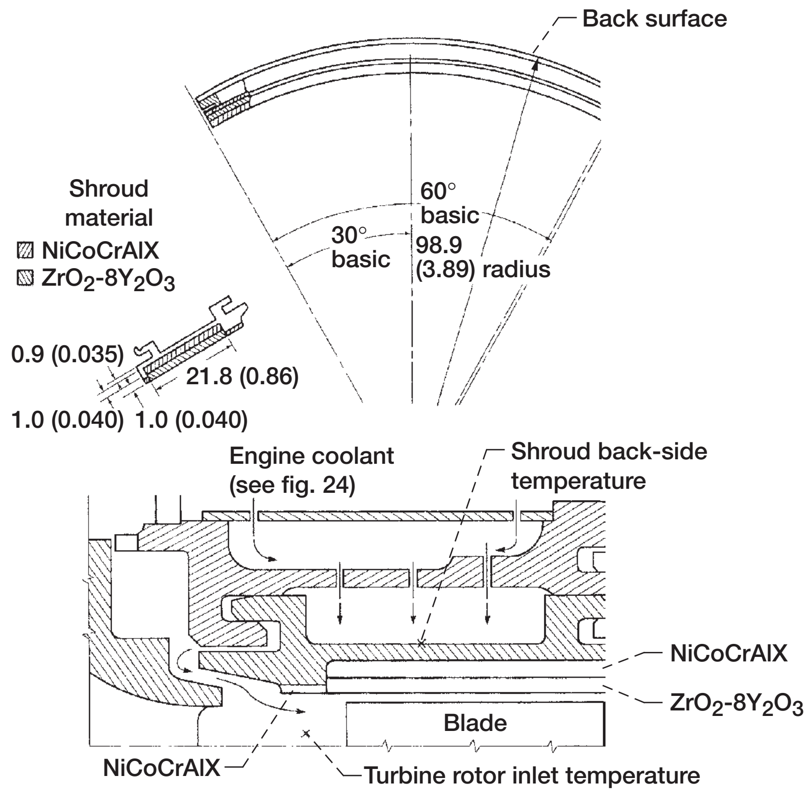


Figure 25.—Schematic of ceramic-coated shroud seal (dimensions in millimeters and inches). (Biesiadny et al., 1985.)

TABLE I.—ABRADABLE MATERIALS USED BY CHAPPEL ET AL. (2001)

Sample	Density, percent	Ultimate tensile strength, psi
Fiber metal 1	22	1050
Fiber metal 2	23	2150
Honeycomb	Hastelloy-X, 0.05-mm foil, 1.59-mm cell	
Nickel graphite	Sulzer Metco 307NS (spray)	
CoNiCrAlY/hBN/PE[a]	Sulzer Metco 2043 (spray)	

[a]Hexagonal boron nitride (hBN) acts as a release agent; polyester (PE) controls porosity.

the surface temperature rise shown in figure 28. The lower the fiber-metal UTS the lower the temperature rise, with the other tested materials in between (apparently the ceramic was not tested). For the high-speed (244 m/s; 800 ft/s) thick-blade utility turbine application, the fiber metal showed essentially zero wear and the ceramic and NiC produced the highest wear. However, for the thin-blade aerospace turbine, wear increased rapidly with UTS (fig. 27). For erosion the high-UTS fiber metal and ceramic did well with others less satisfactory (fig. 28).

For industrial power systems under the conditions tested and the materials of table I, fiber metal had the best abradability-erosion characteristics; honeycomb materials collected on the blade tips; and sprayed materials were less satisfactory at the prescribed porosity. These relative ratings are illustrated as table II. Therefore, inroads toward industrial standards have begun, but more comparison tests are required before standardization becomes a reality.

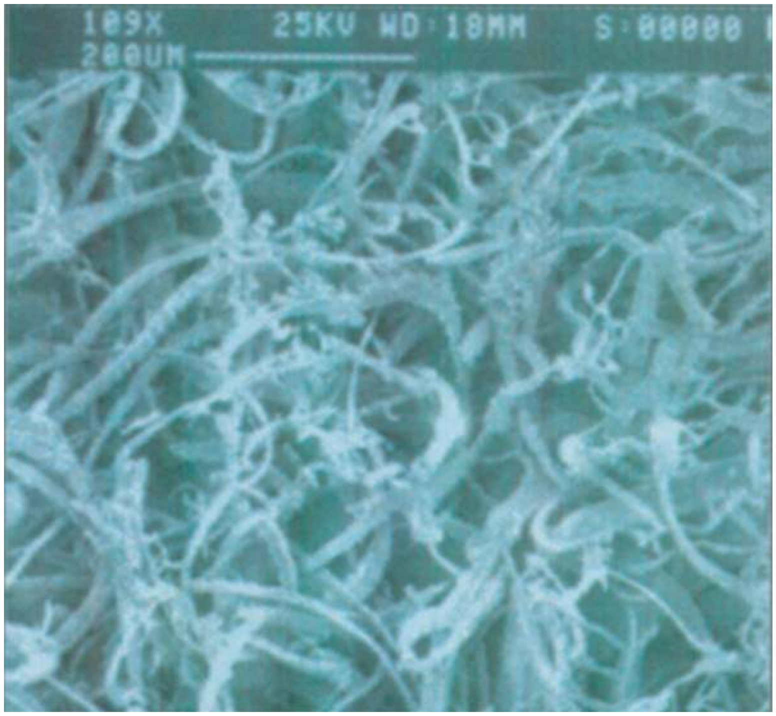


Figure 26.—Scanning electron microscope view of fiber metal. (Chappel et al., 2001a.) Courtesy Technetics Corp.

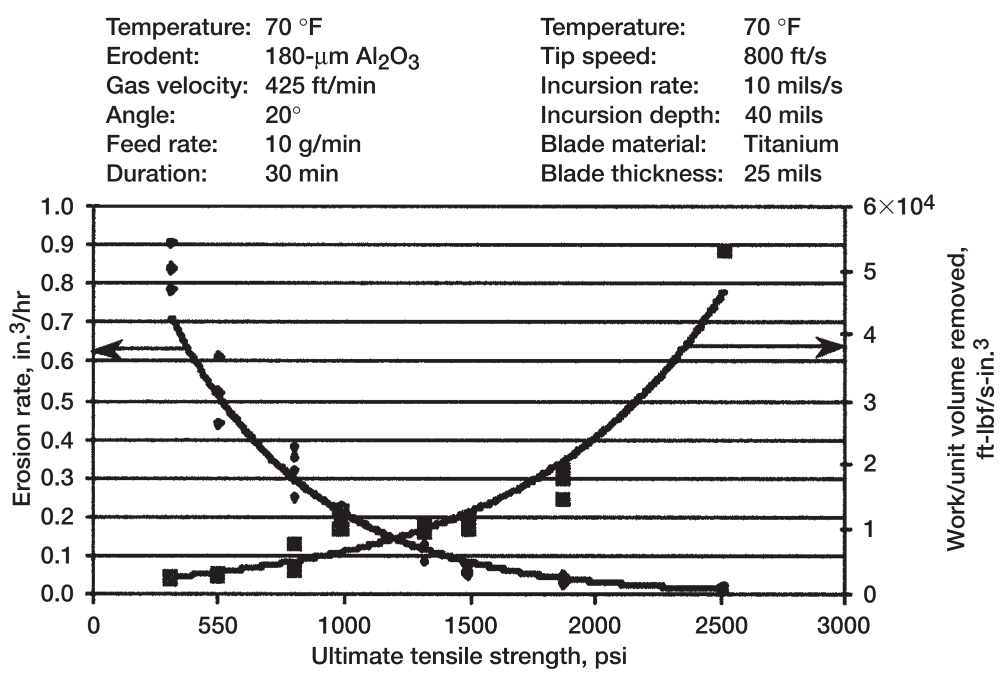


Figure 27.—Erosion and abradability as function of ultimate tensile strength. (Chappel et al., 2001a.) Courtesy Technetics Corp.

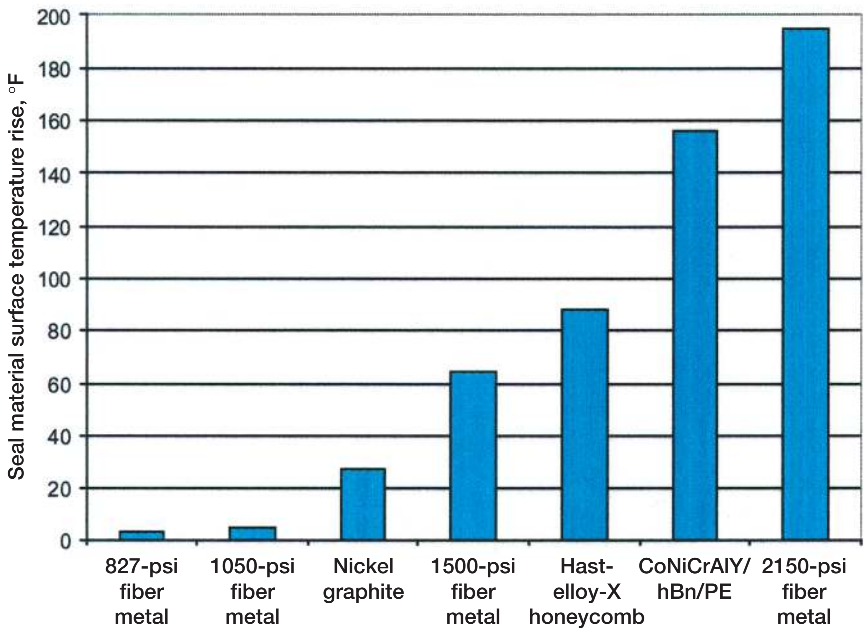


Figure 28.—Surface temperature rise with low-speed abradability test. (Chappel et al., 2001b.) Courtesy Technetics Corp.

TABLE II.—OVERALL PERFORMANCE RANKINGS OF ABRADABLE MATERIALS[a]

Abradable material	High-speed abradability	Low-speed abradability	Erosion resistance
1050-psi fiber metal	1	1	3
2150-psi fiber metal	1	1	1
Hastelloy-X honeycomb	2	3	2
Nickel graphite	3	1	2
CoNiCrAlY/ hBN/PE	3	3	1

[a]Where 1 = best and 3 = worst.

Blade Tip Sealing

The subtle differences in sealing requirements and secondary flows for the shrouded or unshrouded compressor, turbine, and fan affect the entire engine. It is important to evaluate sealing and secondary flows at the blade tip to understand how these disturbances propagate through the engine (see also parts 2 and 3).

Compressor.—In an unshrouded compressor axial leakage opposes the flow path, and circumferential leakage across the blade tip opposes rotation. These factors lead to inversions in the tip flow velocity profiles and engender unstable vortex flow fields with more intense fluctuations at the pressure side of the blade tip, which can cause stall and engine unstart. These effects are discussed further in parts 2 and 3.

Although the blade thickness is small, the ratio of thickness to tip clearance is large (e.g., factor of 10). Nominal tip clearances are 1 to 2 percent of tip chord with 4 to 5 percent considered excessive, depending on engine size and operating envelope. In the work of Copenhaver et al. (1996) rotor efficiency dropped 6 points when the tip clearance/chord ratio was increased from 0.27 to 1.87 percent, and flow decreased 30 percent with a stronger interaction between the tip-leakage flow passage shock.

Turbine.—Unshrouded turbine blades are thicker, better cooled, and more highly contoured than compressor blades. Further, the circumferential and axial pressure gradients are in the directions of rotation and flow, respectively, but sealing similarities to the compressor persist. The tip flow-field vortex couples with the pressure-side-to-suction-side passage vortex, causing passage blockage that decreases through mass flow and turbine work. These effects are discussed further in parts 2 and 3. Hot section sealing and clearance control are discussed in the previous section, case clearance control methods and benefits are discussed in Lattime and Steinetz (2002), and engine externals are discussed in part 3.

Fan.—The tip speeds of high-bypass-ratio engine fans can reach 457 m/s (1500 ft/s). Blade-tip-seal rub strips vary with engine and flight profile (e.g., smooth, honeycomb impregnated, and ribbed). It seems that each engine and company has its own philosophy when it comes to fan rub strips. For example, the PW4090 uses a filled-honeycomb configuration (fig. 29). The uneven rub, caused by in-flight maneuvers, can become, relatively speaking, quite deep. Although it is difficult to tell from the figure, the rub-in is deeper than with other engines. The PW4000 and PW2000 have a very similar labyrinth rub strip (fig. 30). On the other hand, the CFM56 engine uses a smooth surface that gets repotted during overhaul but is usually not refurbished unless considerable damage has been incurred (fig. 31).

Honeycomb Friction

Friction factors are used in modeling or predicting leakages and dynamics of the complex flow fields in honeycomb seals. Allcock et al. (2002) expanded the experimental work of Ha and Childs (1992), where they investigated flow through a flat-plate honeycomb test apparatus. Allcock's similar apparatus tested three honeycomb surfaces. The cell depth was constant at 5 mm with cell sizes of 3.18, 1.59, and 0.79 mm (1/8, 1/16, and 1/64 in., respectively). Clearance was simulated with 0.5-, 1-, 2-, and 3-mm fixed constant-clearance spacers. For the honeycomb surface the friction factor decreased with cell size (with the exception of the 0.5-mm spacing). For all three cell sizes the friction factor increased with spacing (increasing clearance) and approached a constant at cell Re_w greater than 100 000 (e.g., values of 0.04, 0.06, 0.85, and 0.95 for spacings of 0.5, 1, 2, and 3 mm, respectively (fig. 32).

Several types of seals (smooth, straight through, and stepped labyrinth) were modeled, tested, and reported, but slant knives or teeth were not, and results are presumably proprietary. In some turbomachines honeycomb seals can be unstable and can require modifications such as for a large-scale compressor (see appendix B of part 2).

Figure 29.—PW4090 fan and rub strip. (a) Fan and shroud. (b) Fan/rub strip interface. (c) Rub strip. Courtesy Sherry Soditus, United Airlines Maintenance, San Francisco, CA.

Figure 29.—Concluded. (c) Rub strip.

Seals and Bearings

Munson et al. (2002) describe and provide operations data for room-temperature testing of a seal/bearing concept. The basic concept (fig. 33) was advanced by von Pragenau (1982, 1992). In Munson's seal configuration the foil thrust bearing was combined with a mating flat interface to make a device called a foil face seal (fig. 34). The interface foils are supported by multiple-wave bump foils. With pressure drop and rotation this interface gives rise to a compliant, hydrodynamic, film-riding face seal. The seal can handle conical distortions of at least 0.52° and circular out of flatness; stiffer bump foils gave better load capacity. Lift-off speed essentially followed as $(\text{load})^{1/2}$ up to 266.9 N (60 lb) with leakage following the pressure drop (ΔP) relation

$$\text{Leakage rate (scfm)} = -0.0213\ \Delta P\ \text{(psid)} + 3.3$$

where 1 scfm (standard cubic foot per minute) = 0.028317 m^3/min and 1 psid = 1.9524×10^{-3} MPa, within ±0.2 scfm over the range 28 000 to 56 000 rpm. As a comparison the leakage flows for several sealing configurations are illustrated in figure 35. For a 89-kN- (20 000-lb-) thrust class engine, with this technology, impressive estimated mission fuel burn reductions of –1.85 percent for a fixed engine and "rubber" airframe and –3.17 percent for both engine and airframe being rubber were reported. (Here rubber refers to allowing for design parameter changes.)

Cost Comparisons

Munson's et al. (2002) SFC reductions would save today's airline customer over 0.5 billion gallons of fuel—a significant effect on both the environment and the customer. Stair and Ludwig (1978) and Stocker (1978) noted similar annual savings and environmental benefits. Stocker stated

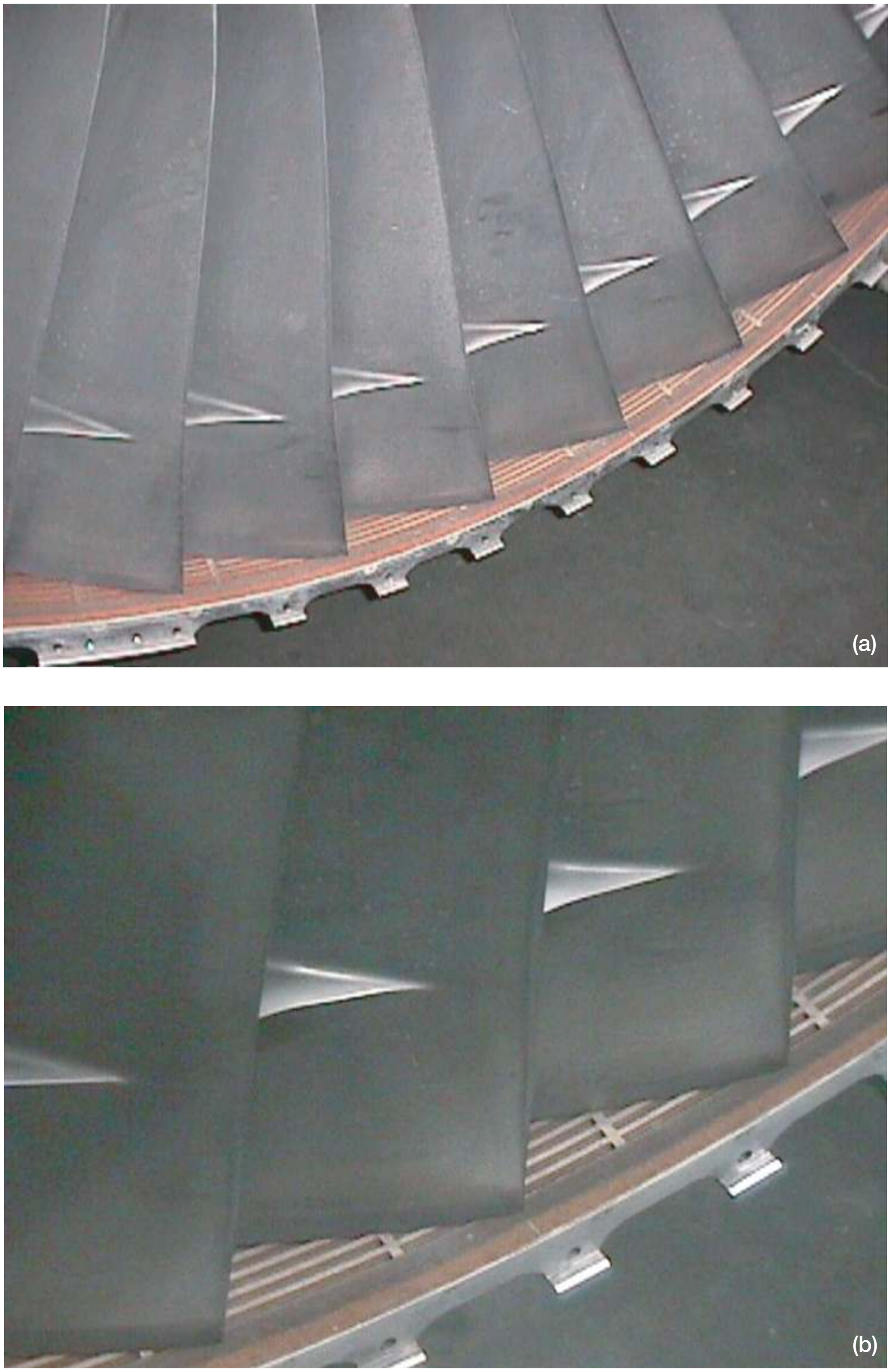


Figure 30.—PW2000/4000 fan rub strip. (a) Fan and shroud. (b) Fan/rub strip interface. (c) Rub strip. Courtesy Sherry Soditus, United Airlines Maintenance, San Francisco, CA.

Figure 30.—Concluded. (c) Rub strip.

Figure 31.—CFM56 fan and rub strip. (a) Fan overview. (b) Fan/shroud interface. (c) Fan rub strip. (d) Closeup of fan rub strip. Courtesy Sherry Soditus, United Airlines Maintenance, San Francisco, CA.

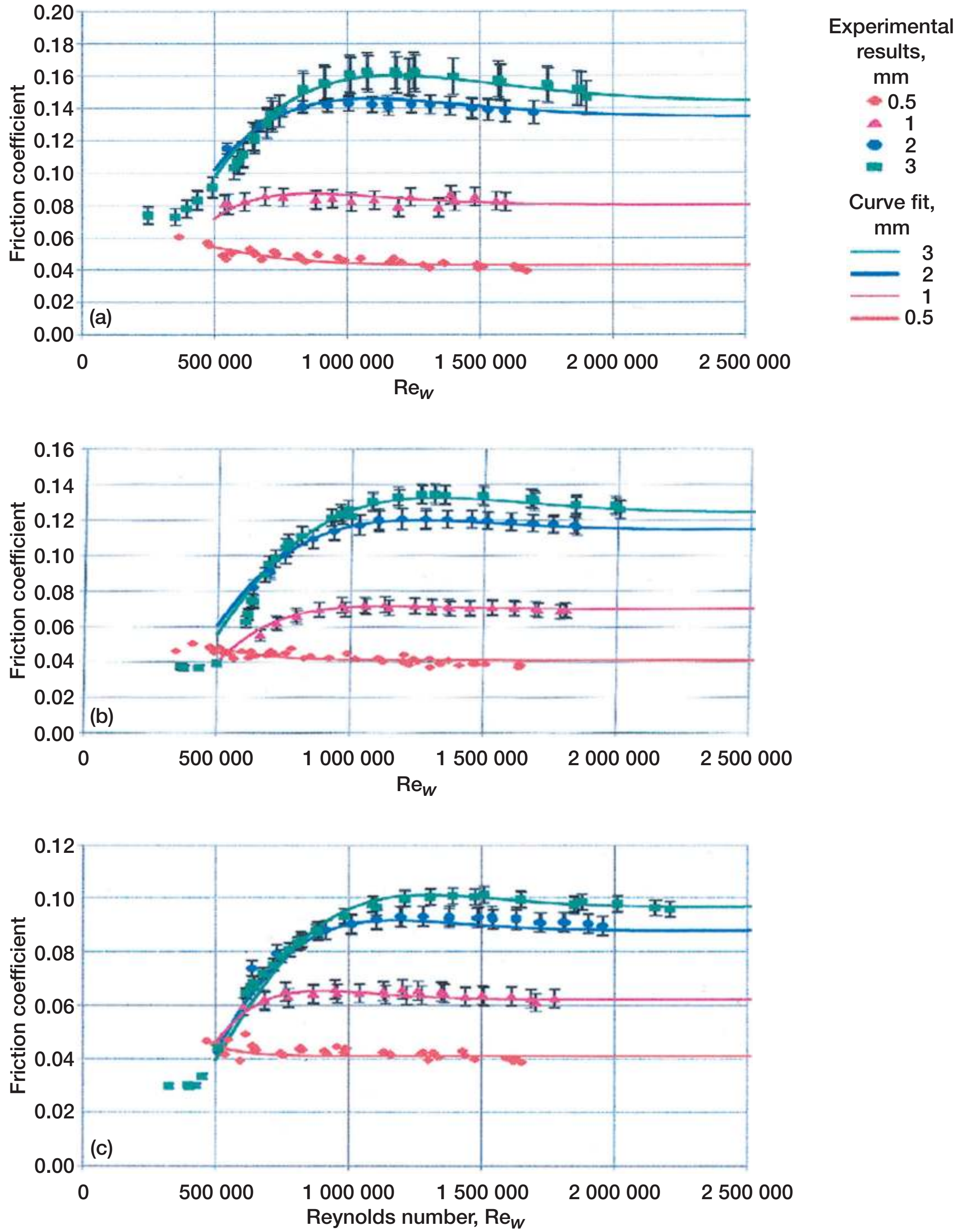


Figure 32.—Friction factor versus Reynolds number, all clearances. (a) 3.18-mm honeycomb. (b) 1.59-mm honeycomb. (c) 0.79-mm honeycomb. (Allcock et al., 2002.) Courtesy AIAA.

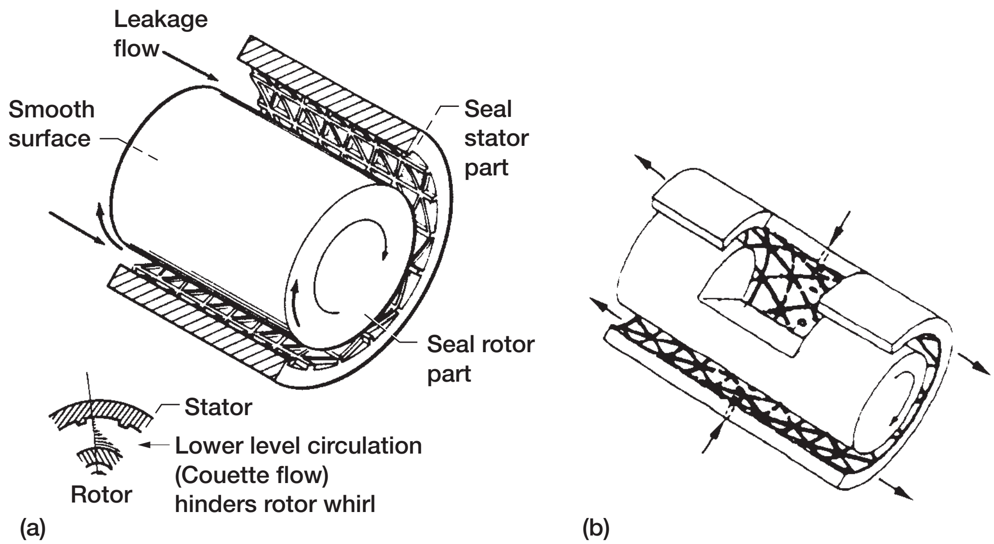


Figure 33.—Damping seals and bearings. (a) Damping seal concept with isogrid stator. (b) Damping bearing concept with stationary orifices (orifices can also be in the shaft). (Inventions of George von Pragenau, Huntsville, Alabama; Hendricks et al., 1995.)

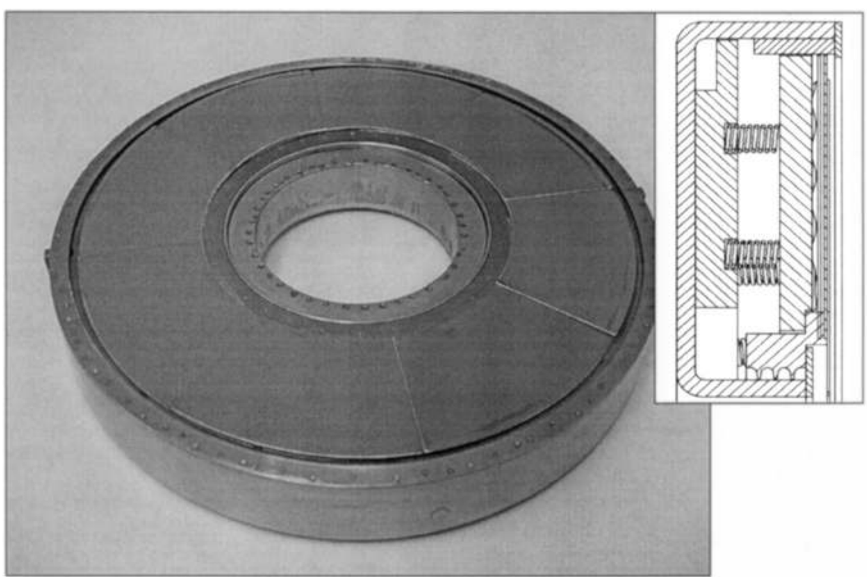


Figure 34.—Proof-of-concept foil face seal. (Munson et al., 2002.) Courtesy Rolls-Royce/Allison.

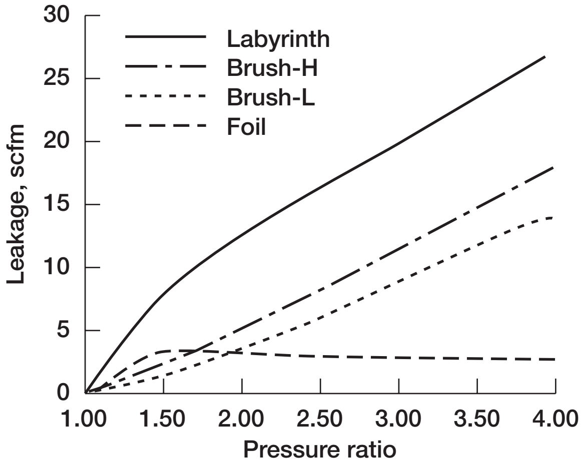


Figure 35.—Seal leakage comparison. (Munson et al., 2002.) Courtesy Rolls-Royce/Allison.

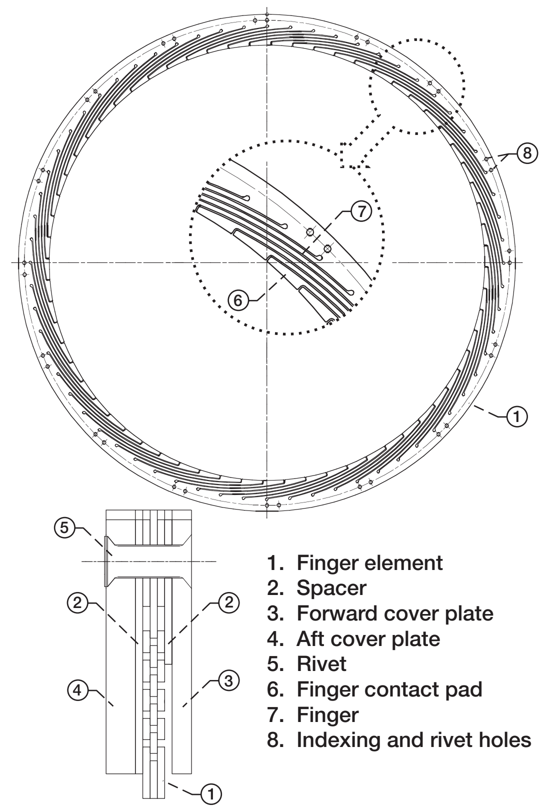


Figure 36.—Finger seal design. (Proctor et al., 2002.)

that a 0.25 percent reduction in engine seals leakage would give a 2.75 percent reduction in SFC, which translates into nearly 0.45 billion gallons of U.S. airlines fuel savings[3] (1977 statistics). Stocker also viewed stepped, forward-facing conical labyrinth seals to be more efficient than see-through labyrinth seals with honeycomb lands. These conical labyrinth (also known as slant-knife) seals over honeycomb lands provided efficient reduction in leakage, whereas other abradable lands leaked significantly compared with a smooth interface.

Stair and Ludwig (1978) determined that improvements in fluid film sealing resulting from a proposed research program could lead to an annual energy saving, on a national basis, equivalent to about 37 million barrels (where 1 barrel = 42 U.S. gallons) of oil or 0.3 percent of total U.S. energy consumption. Further, the application of known sealing technology could result in an annual saving of an additional 10 million barrels of oil. In addition to energy saving, cost effectiveness is further enhanced by reduction in maintenance and by minimization of equipment required for collecting leakage and for meeting environmental pollution standards.

Finger Seals

Proctor et al. (2002) continued to provide data for the testing of a finger seal configuration (fig. 36) over a range of temperatures to 650 °C (1200 °F) and rotor speeds to 366 m/s (1200 ft/s) at pressure drops to 0.52 MPa (75 psid). They found power losses to be similar to those of a brush seal. Radial finger wear in terms of finger thickness was about 0.5 over 11 hr of testing with 70 percent occurring during the first 3.5 hr of testing. Some hysteresis was noted but was cited as within design

[3]Implies over 1 billion gallons world aviation savings.

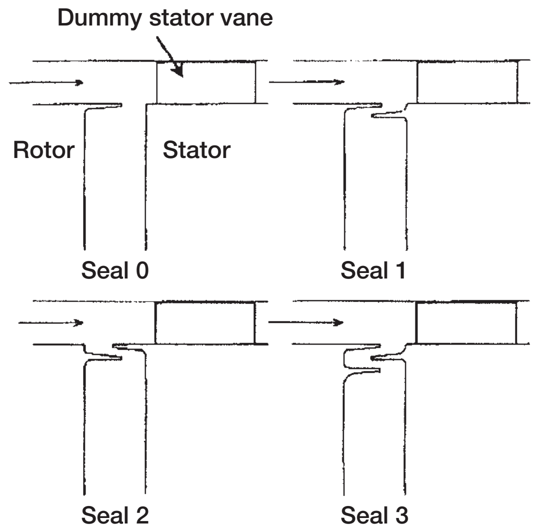


Figure 37.—Experimental rim seal configurations. (Teramachi et al., 2002.) Courtesy AIAA.

expectations. The flow parameter[4] $\phi = \dot{m}\sqrt{T_0}/P_0 D_r$ was generally less than 0.006, with most data falling near 0.004. A second generation of finger seals (illustrated in appendix B of part 2) is also being investigated.

Rim and Platform Interface Sealing

Teramachi et al. (2002) investigated rim interface sealing (discussed in more detail in part 3). They tested four rim seal configurations (fig. 37): (0) T-on rotor, (1) T-on rotor with overlap T-on stator, (2) T-on stator with overlap T-on rotor, and (3) fish mouth on rotor with overlap T-on stator. Configuration 3 was the least affected by changes in overlap and configuration 0 the most; configuration 2 was quite sensitive to overlap (see also fig. 7).

Wellborn et al. (1996) investigated the effect of leakage in a four-stage, low-pressure compressor with blading design based on the NASA E^3 engine. Seal leakages did not affect upstream stages but did progressively degrade downstream stage performance. For each 1 percent change in clearance/span ratio the pressure rise penalty was nearly 3 percent with a 1 percent drop in efficiency (see also part 3).

Heidegger et al. (1996) performed a parametric study on a three-tooth labyrinth seal/cavity configuration (fig. 38) and a sensitivity study on various sealing parameters (fig. 39). A class of generic labyrinth seal tooth geometries is nicely imaged in figure 40. The most significant effect of their study shows that the leakage flow out of the seal cavities can affect the power stream and hence engine performance as found by Hendricks et al. (1994) in their series of YT–700 engine tests. (See the section Engine Performance on page 10 and see also part 3.)

Smalley et al. (1975) introduced a high-pressure, high-temperature seal that responds dynamically through elastic deformation of the J-seal membrane. The J-seal consists of a sealing ring supported on a thin membrane and separated by a fluid film from the runner. It thus behaves as a face seal with a resilient (deformation) interface. The toroidal J-seal can accommodate axial changes to 0.254 mm (0.010 in.) caused by runout, distortion, or axial shift. The larger shifts due to throttle chops are to be carried by a slower moving carrier (not described in their paper).

[4]Where T_0 and P_0 are the plenum temperature (°R) and pressure (psia), D_r is the rotor diameter (in.), and $\dot{m}$ is the mass flow rate (lb/s).

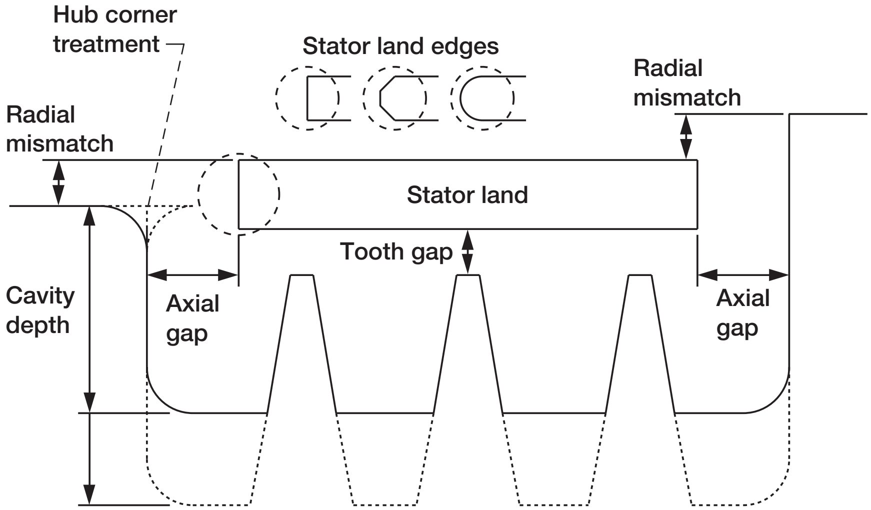


Figure 38.—Geometric parameters defining overall seal/cavity geometry. (Heidegger et al., 1996.)

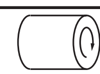
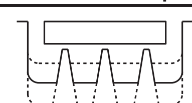
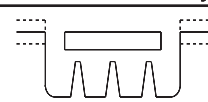
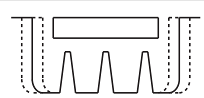
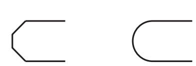
Parameter	Baseline case	Variations
Seal tooth gap	0.010 in.	 No cavity, no gap, 0.020 in., 0.040 in.
Wheel speed (rpm)	100%	 58% baseline speed
Seal cavity depth	0.184 in.	 ± 50% of baseline cavity depth
Radial mismatch of hub flow path:		
Upstream	0.000 in.	± 5% stator span
Downstream	0.000 in.	± 5% stator span
Axial trench gap:		
Upstream	0.081 in.	
Downstream	0.061 in.	± 20% of baseline gap
Hub corner treatment:		
Leading edge	Sharp	Rounded back; Sharp; Rounded forward
Trailing edge	Sharp	
Stator land edge treatment	Faceted	 Faceted; Rounded

Figure 39.—Test matrix of geometric parameters to be tested relative to baseline configuration. (Heidegger et al., 1996.)

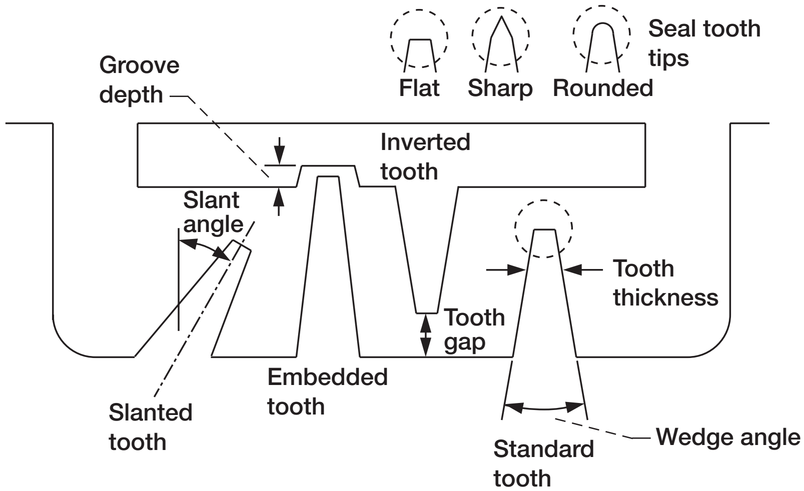


Figure 40.—Geometric parameters defining individual and generic seal tooth geometries. (Heidegger et al., 1996.)

Spiral-Groove Seals

Another type of seal perhaps worth revisiting is the spiral-groove seal advanced by Strom et al. (1967) (figs. 41 and 42), who reported that for similar geometries the spiral-groove face seal showed negligible leakage and wear compared with the contact face seal. Groove feeding produced greater film thickness.

The visco-seal may also have applications. See Fisher et al. (1968) for theoretical and experimental results of studies with visco and buffered shaft seals.

Oil Sealing

Oil sealing of bearing compartments in turbomachines is difficult. In many aerospace and some advanced aeronautical engine applications, the working fluid is used as the lubricant. However, in most aeronautical and aeronautical-derivative engines, oil is used as the lubricant. Ludwig (1978) and Whitlock (1978) provide schematic sketches, albeit perhaps dated, of oil sealing requirements for bearings. Figure 43 represents classes of labyrinth seals; figure 44 represents sealing functions and requirements. These figures provide concepts for typical gas turbine engine oil-sealing functions (e.g., single-seal bearing compartment, secondary venting, cooling air blanket, typical bearing compartment, and pressure balance passage).

A key is to prevent the oil side of the seal from becoming flooded (i.e., to keep the mass fraction of oil low, nearly all vapor, at the seal inlet). Hughes and Chao (1980) studied the limiting cases of all liquid at the inlet or at the exit of a face seal, which gave rise to two film thicknesses that could support the same load. Beehler and Hughes (1984) completed the transient analysis for laminar flows and found that for a near-saturation inlet condition the film suddenly drops its support. It may (1) with sufficient damping be restored, (2) collapse to smaller or contact film thickness and still operate but at higher temperatures, or (3) cycle in sustained oscillation if damping is small. These concepts may explain the dynamics of seal popping. Again, the key is keeping the liquid away from the inlet to enhance stability (see also part 2).

Nevertheless, oil fog and oil vapor produced in the bearing regions may be transported through labyrinth seals opposite to the buffering fluid flows. Leakage can occur by diffusion of oil due to concentration gradients and by oil transport due to vortical flows within the rotating labyrinth cavities

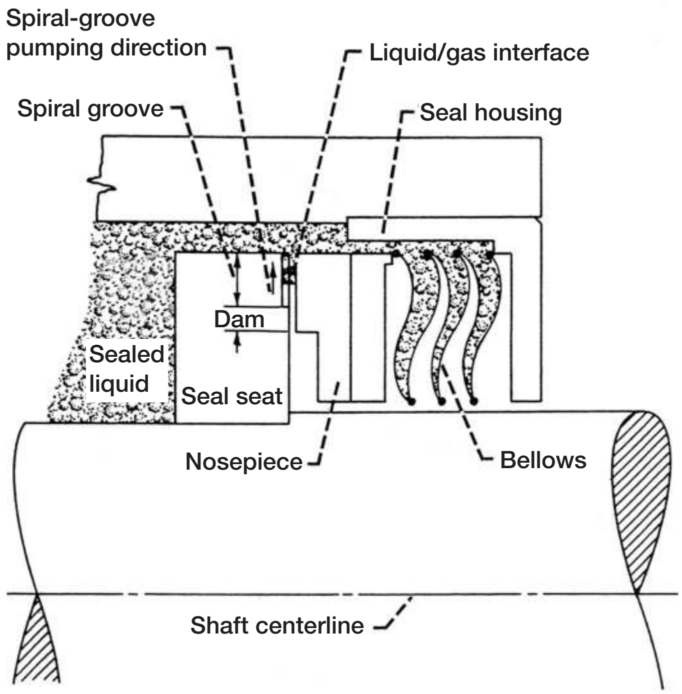


Figure 41.—Spiral-groove seal placement in seal seat for sealing liquid at outside diameter of seal dam. (Strom et al., 1967.)

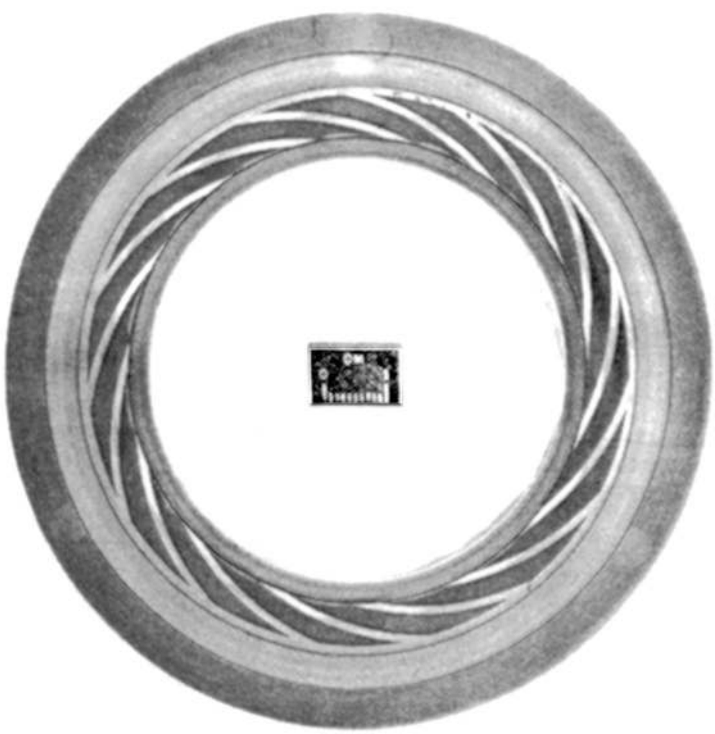
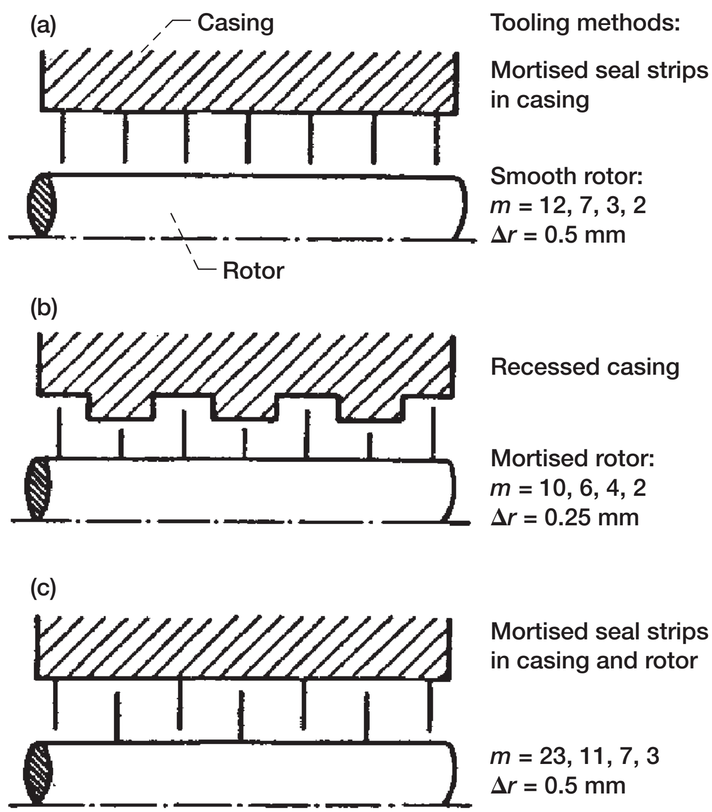


Figure 42.—Seal nosepiece with chemically etched inward-pumping spiral grooves. (Strom et al., 1967.)



Testing parameters of labyrinth:

Entry swirl speed, C_{u0}
Differential pressure, Δp_{st}
Eccentricity, ε
Rotor speed, U_w
Number of whirling chambers, m
Geometry, Δr, h, t

Figure 43.—Close-clearance labyrinth shaft seals, circumferential types. (Benckert and Wachter, 1978.)

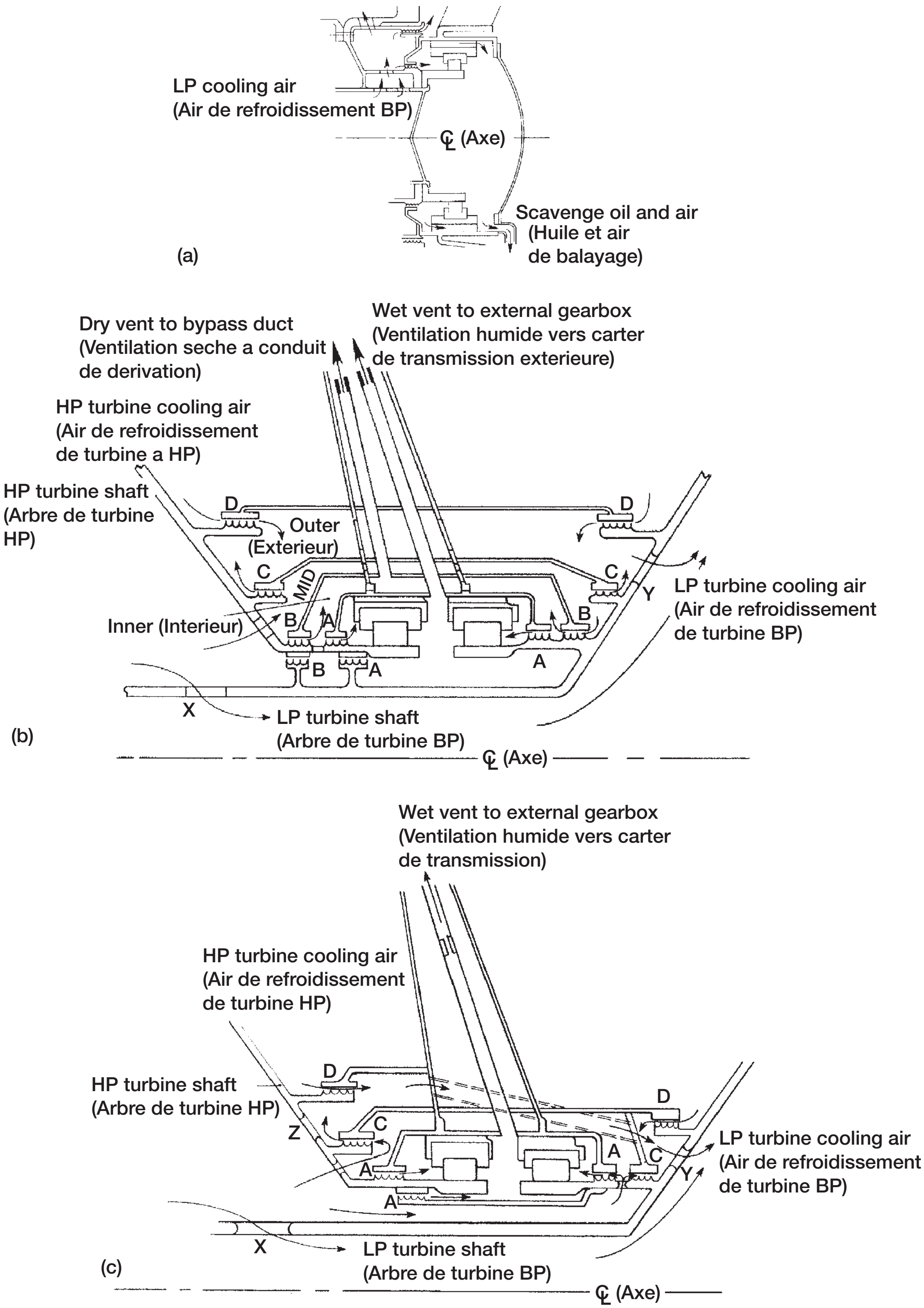
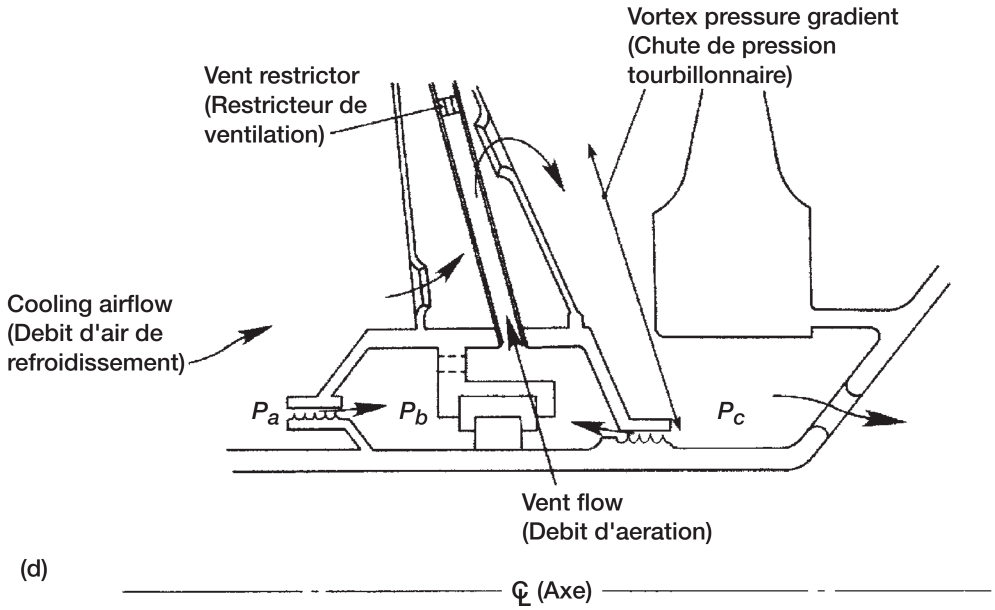


Figure 44.—Typical gas turbine engine oil-sealing functions and requirements. (a) Single-seal bearing compartment. (b) Secondary venting. (c) Cooling air blanket. (d) Typical bearing compartment. (e) Pressure balance passage. (Whitlock, 1978.)



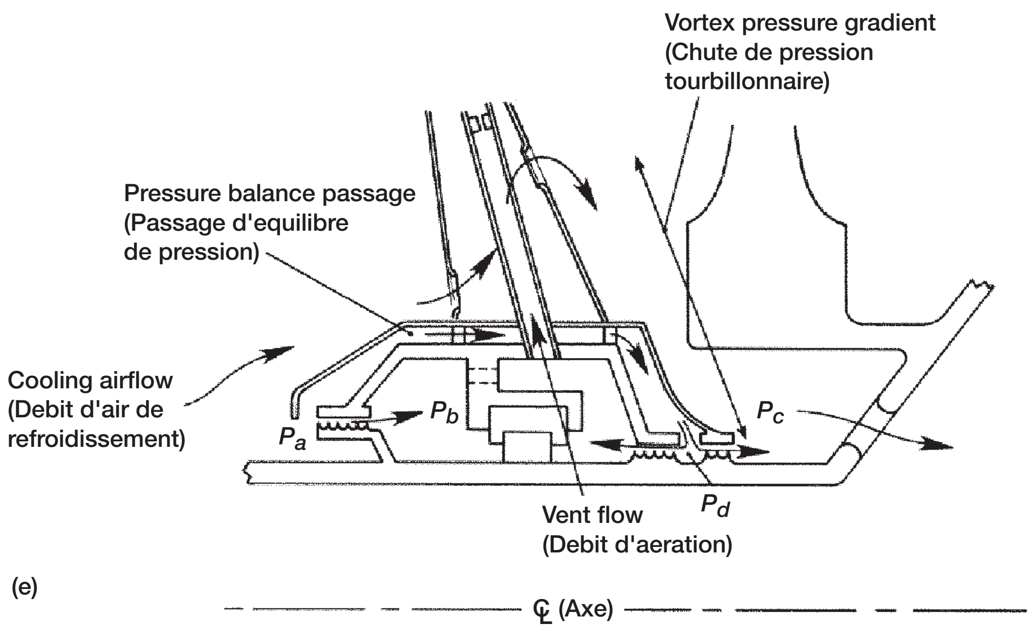


Figure 44.—Concluded. (d) Typical bearing compartment. (e) Pressure balance passage. (Whitlock, 1978.)

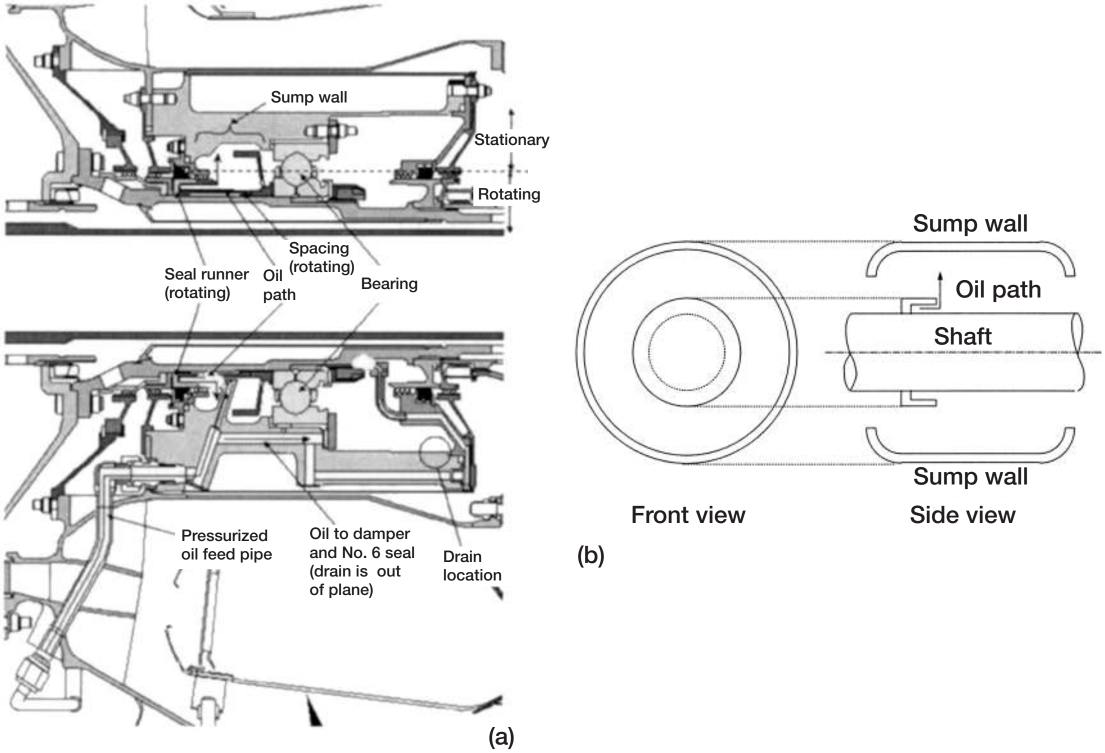


Figure 45.—Oil sump. (a) Cross-sectional view of engine. (b) Model of oil sump. (Shimo and Hesiter, 2002.) Courtesy AIAA.

(Boyman and Suter, 1978). Seal deterioration results in an unbalancing of the bearing cavity flows. Especially prone are see-through labyrinth configurations, but the effect decreases with an increasing number of teeth and labyrinth cavities (crude distillation columns).

Weinstock and Hesiter (2002) recently addressed the issues of oil containment. They provide simplified modeling and numerical results for droplets flung from a spinning cylinder into a concentric cylinder. Collection efficiency went down with decreasing ratio of wall radius to shaft radius and increasing sump pressure and was not much affected by either temperature or shaft speed. Thick films did not affect collection efficiency, but with thin films, on the order of 1/2 to 1 drop diameter, the collection efficiency increased.

Shimo and Hesiter (2002) provide computational fluid dynamics (CFD) analysis of an oil slinger for the 61-mm-shaft-radius AE3007 (Rolls Allison) at 95.8 m/s with a 21- by 8-mm C-shaped annular slinger and a 129.2-mm-radius concentric cylinder sump (fig. 45). Temperature had the largest effect on the film. For the sump a competition between gravity, windage, and viscosity effects gave rise to oscillations and an orientation-dependent sump. As the film thickened, the viscous effects decreased but windage and gravity forces increased (and vice versa). However, the time-averaged film was nearly constant. Still when windage forces or the sump wall was too far from the shaft, dryout could occur over a significant part of the sump.

Carbon ring and face sealing of the sumps described by Ludwig (1978) and Whitlock (1978) is fairly standard. However, hybrid ceramic seals, whereby both the rotor and stator are ceramic, have achieved good sealing and life between overhauls. Boyd et al. (2002) describe one such hybrid ceramic shaft seal comprising 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 flexible beam as the inner ring (fig. 46). The key element is a near match of the thermal expansion coefficients of the materials (Si_3N_4 at 2.9×10^{-6} cm/cm/°C versus carbon nanofibers at 4×10^{-6} cm/cm/°C). In addition, Si_3N_4

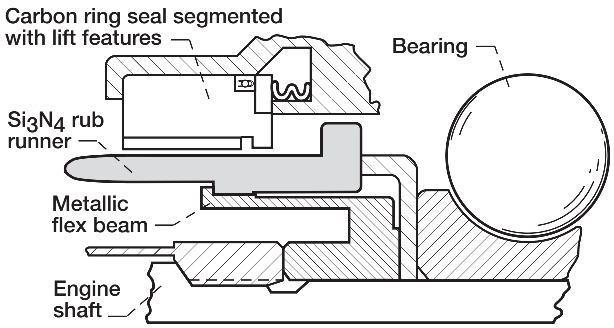


Figure 46.—Hybrid ceramic carbon ring seals. (Boyd et al., 2002.)

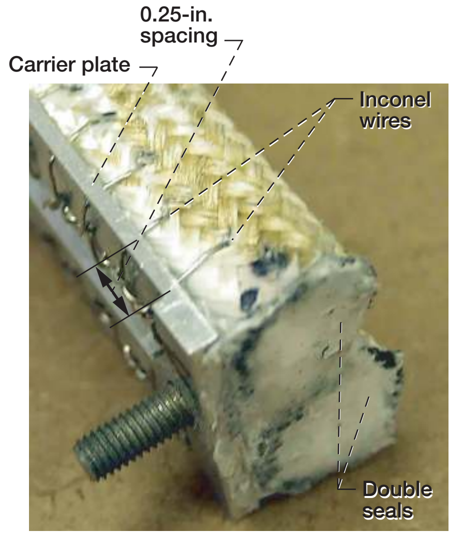


Figure 47.—Double seals mechanically attached to carrier plate with Inconel wires for scrub testings. (Dunlap et al., 2002.)

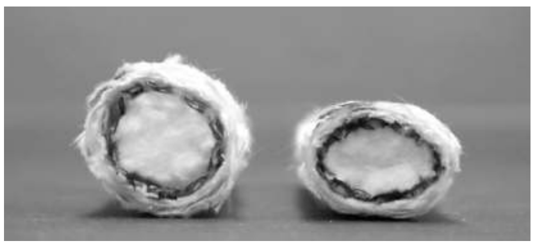


Figure 48.—6-pcf-X-38 seals before (left) and after (right) 1900 °F exposure. (Dunlap et al., 2002.)

has higher Young's modulus, lower density, higher hardness, and lower friction than metals and must be carefully screened to eliminate flaws. The flexible beam added sufficient damping for stability over the range of operation, but Boyd et al. (2002) provide no rotor-fluid dynamic coefficients. They saw no oil seepage at idle speed down to air-to-oil pressure differentials of 0.69 kPa (0.1 psia).

Static- or Slow-Displacement Sealing

Static- or slow-interface-movement rope, string, and wadding seals have been around for quite some time. Various impregnants are used to satisfy sealing fluid and material compatibility requirements. These seals are effective, low cost, and readily available and are often a plumber's best friend.

In addressing high-temperature static- or slow-interface-movement sealing issues, Dunlap et al. (2002) provide experimental data for single and dual braided-rope seals for sealing the control flap gap during reentry conditions (fig. 47). The 15.7-mm- (0.62-in.-) diameter rope seal consisted of an Inconel X–750 spring filled with Saffil insulation covered by two layers of Nextel 312 fabric wrap. Tests were conducted under static and rubbing oscillating conditions when the seal was heated with an arc jet to 1200 °C (2200 °F) and 2.8-kPa (0.4-psid pressure) (ambient to 5.5 kPa (0.8 psid)). Under compression and temperatures of 1038 °C (1900 °F) for 7 min the seal compressed to 0.7 by 1.06 relative units in an elliptical pattern, indicating a degree of spring set (fig. 48). One-thousand-cycle wear tests against shuttle tiles were conducted with primary degradation occurring in the first 50 cycles. In a recent patent Steinetz and Dunlap (2002) describe a new braided, carbon fiber, thermal barrier designed to protect critical nozzle joints and O-ring seals in the space shuttle reusable solid rocket motors. The new thermal barrier reduces the temperature of the 3040 °C (5500 °F) rocket combustion gas (nonoxidizing environment) and permits only relatively cool (<110 °C; <200 °F) gas to reach the O-rings. The seal enhances safety margins and enables solid rocket motor joint assembly in approximately one-sixth the time.

Applications to turbojet engines include a hybrid braided seal, developed by Steinetz and Adams (1997), to seal the last-stage articulated turning vane of the F119 turbine engine. The seal limits flow of fan cooling air past the turning vane flow path (or power stream)/fairing interface and also prevents backflow of potentially damaging high-temperature core air (fig. 49). Key seal design conditions include sealing air pressure differentials across the seal of 0.345 MPa (50 psid) and temperatures in the range 540 to 650 °C (1000 to 1200 °F). The hybrid seal consists of a Nextel ceramic core overbraided with high-temperature superalloy wires. This seal configuration limits leakage and reduces the chances of extrusions because of its tougher, more abrasion-resistant sheath.

Inexpensive, flexible cloth seals provided uniform slot or gap contact and good compliance over a range of excursions and demonstrated leakage reductions up to 30 percent in combustors and 70 percent in nozzle segments (Askit and Demiroglu, 2002). A cloth seal can be fabricated much like a covered feather seal. The backbone or body is made of thin sheets of fabric, metal, or ceramic that are covered with sacrificial, yet flexible, high-temperature woven fabric, metallic, or ceramic material. The flexibility and compliance absorb rather than transmit vibrations thereby also functioning as a damper. Problems can arise if electrical isolation or conduction is required but can be overcome by materials selection. For low-temperature applications (190 °C; 380 °F) the cloth can be impregnated with rubber; for higher temperatures (540 °C; 1000 °F) it can be impregnated with texturized fiberglass yarn; or when coated with abrasion-resistant vermiculite it can be used to 1255 °C (1800 °F). Silica cloth (amorphous silica fibers), used for personnel protection, is also good to 1255 °C (1800 °F). And for even higher temperatures (1530 °C; 2300 °F) ceramic fiber cloth can be used, with Inconel for greater tensile strength. The cloth seal is similar to the rope seal, and the backbone or body could be plasma sprayed, forming a robust, high-temperature seal. However, in most cases impregnated cloth seals lack the high resilience of the cloth or rope configuration.

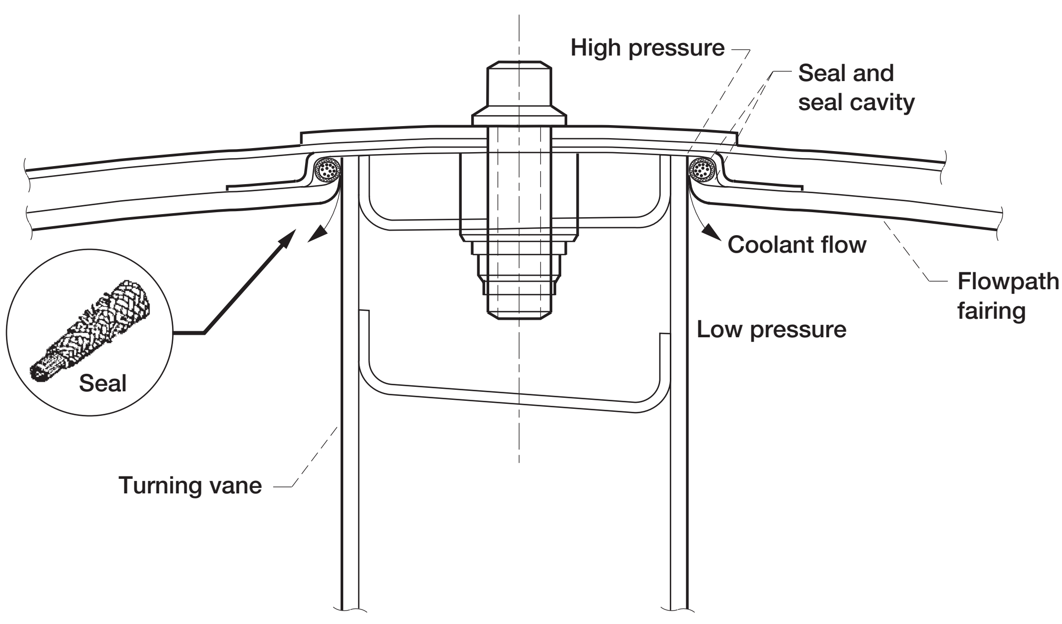


Figure 49.—Cross section of PW F119 engine showing last-stage turning vane with hybrid braided seal sealing perimeter.

Summary

Pay very close attention to sealing and secondary fluid streams. Interaction between seal and engine flows is important to the implementation of new seal technology. This does not mean that gas path or power-stream dynamics are unimportant. Brayton-cycle machines, for example, are fairly mature. However, turbomachine sealing represents your competitive edge.

1. Maintaining seal leakages and secondary flows within engine design specifications is the most efficient and cost-effective way to enhance the on-wing performance and minimize maintenance costs. Even with refurbishment engines become worn and their performance is less than for an out-of-the-box engine. The performance loss is significant, being 10 to 15 percent with each refurbishment depending on engine size and operating history, and over time the engine becomes uneconomical.
2. Changes in sealing, especially on the high-pressure spool, dramatically change the performance of the entire engine. This result has been conclusively demonstrated in a YT–700 engine test where the compressor discharge pressure seal, a labyrinth seal, was replaced by a dual-brush seal. A direct comparison of specific fuel consumption over a range in developed power showed a consistent reduction of over 1 percent.

The sealing research and development agenda of Ludwig (1978), the engine design requirements established by Campbell (1978), and the airline customer needs given by Smith (1978) remain as current objectives. However, to these objectives one must add the market-force control issues of Anderson (2003) and turbomachine seal rotordynamics. It was the failures of the space shuttle main engine turbomachinery that ushered in a new understanding of seal rotordynamics (see part 2) and the terrorist attack of September 11, 2001, that introduced new forces into the industry relative to customer profits and sealing and secondary flow requirements.

Today's advanced sealing applications require time-unsteady computations and control surfaces with attention focused on control of vertical structures and flow separation within the tip, platform, and cavity regions of the turbomachine. These unsteady flows and their temperature and pressure requirements afford many challenges to sealing (see part 3).

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13. ABSTRACT ***(Maximum 200 words)***
Although forces outside our control shape our industry, turbomachine sealing research, design, and customer agendas established in 1978 by Ludwig, Campbell, and Smith in terms of specific fuel consumption and performance remain as objectives today. Advances have been made because failures of the space shuttle main engine turbomachinery ushered in a new understanding of sealing in high-power-density systems. Further, it has been shown that changes in sealing, especially for high-pressure rotors, dramatically change the performance of the entire engine or turbomachine. Maintaining seal leakages and secondary flows within engine design specifications remains the most efficient and cost-effective way to enhance performance and minimize maintenance costs. This three-part review summarizes experiences, ideas, successes, and failures by NASA and the U.S. aerospace industry in secondary flow management in advanced turbomachinery. Part 1 presents system sealing, part 2 system rotordynamics, and part 3 modeling, with some overlap of each part.

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