JOURNAL OF PROPULSION AND POWER
Vol. 18, No. 6, November–December 2002

Advanced Seals for Industrial Turbine Applications: Dynamic Seal Development

Raymond E. Chupp,* Farshad Ghasripoor,† and Norman A. Turnquist‡
General Electric Global Research Center, Niskayuna, New York 12309
Mehmet Demiroglu§
Rensselaer Polytechnic Institute, Troy, New York 12180
and
Mahmut F. Aksit¶
Gebze Institute of Technology, 41400 Gebze, Turkey

The ongoing need for higher performance industrial turbines has lead to extensive efforts to improve various components of gas turbines, steam turbines, compressors, and generators. One area being addressed is improved seals to reduce parasitic leakage flows. Major progress has been made to implement advanced dynamic seals into industrial turbines with resulting performance gains. Brush seals have significantly decreased labyrinth seal leakages in gas-turbine compressors and turbine interstages, steam-turbine interstage and end packings, industrial compressor shaft seals, and generator seals. Abradable seals are being developed for blade-tip locations in various turbine locations. The development and implementation of advanced seals in industrial turbines is summarized and with a focus on dynamic seals.

Introduction

IMPROVEMENTS in dynamic seals between rotating and stationary parts in industrial gas and steam turbines can significantly reduce parasitic leakages and give better control of secondary flow systems. These improvements result in performance improvements in both reduced heat rate (increased efficiency) and increased power output.[1] Brush seals have been developed to improve significantly leakage performance over labyrinth seals in gas-turbine compressor and turbine interstage locations, steam-turbine interstage and end packings, aircraft engine turbine locations, industrial compressor shaft seals, and generator seals. Abradable seals are being developed for several blade-tip locations. Also, aspirating face seals have undergone extensive analysis and testing for potential applications in aircraft engines.[2]

This paper addresses the development of dynamic seals for industrial turbines. A companion paper[3] focuses on the overall design approach, experimental facilities, and static seal development. Two dynamic seal areas are addressed in this paper: brush seals for gas and steam turbines and abradable seals for blade-tip sealing. Figure 1 shows the various areas where these seal types are being incorporated into industrial gas turbines.

Gas-Turbine Brush Seals

Brush seals can significantly reduce leakage flows in turbomachinery between the stationary and rotating components. Brush seals have been under development by General Electric (GE), other turbomachinery manufacturers, seal vendors, universities, and NASA. An extensive summary of brush seal research and development work through 1995 has been published.[4] More recent papers have been published covering brush seal development by GE[5–8] and others.[9–33]

Brush seal development at GE for industrial turbines began in the early 1990s. Initial work focused on gas turbines and built on sealing improvement efforts for the GE90 engine. The performance gains achievable in industrial gas turbines is significant. They range from a 0.2 to 0.6% reduction in heat rate and a 0.3 to 1% increase in power output for each location.[1] Technological development continues to leverage aircraft engine applications where possible and focuses on several industrial turbomachinery applications, including industrial compressors.

A brush seal consists of a bristle pack welded at the outer diameter to the side plates. The downstream plate serves as a backing plate and provides structural support for the bristles. The forward plate protects the bristle pack from incoming flow disturbances and handling damage. In turbine retrofit installations, typically a rotor land is removed to maintain an axial clearance between the brush seal and neighboring lands. Figure 2 show a typical brush seal configuration, and Fig. 3 compares leakages for brush seals compared to typical labyrinth seals.

Figure 4 is a photograph of a representative brush seal taken during a routine inspection. The seal is in good condition after nearly three years of operation. To date, 205 brush seals have been installed in 70 GE industrial gas turbines, nine of which are instrumented for performance monitoring. These turbines have been E and F technology classes in Frames 3, 5, 6, 7, and 9. Brush seals have been installed in the compressor discharge high-pressure packing (HPP), middle bearing, and turbine interstage locations. Of the turbines, 16 with brush seals have been operating since 1996 and 52 since 1997. The current fleet leader is a Frame 7EA HPP brush seal with 40,000 h of operation. Field data and experience from these installations have validated the brush seal design technology.

Brush seals must be carefully designed to achieve the design requirements. Several brush seal design considerations are described hereafter.

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

Presented as Paper 2001-3626 at the AIAA/ASME/SAE/ASEE 37th Joint Propulsion Conference and Exhibit, Salt Lake City, UT, 8–11 July 2001; received 24 August 2001; revision received 15 June 2002; accepted for publication 10 July 2002.

*Mechanical Engineer, Energy and Propulsion Technology; raymond.chupp@crd.ge.com. Member AIAA.
†Engineer, Energy and Propulsion Technology; ghasripo@crd.ge.com.
‡Engineer, Energy and Propulsion Technology; turnquist@crd.ge.com.
§Graduate Student, MANE; demirm@rpi.edu.
¶Chair, Energy Systems; aksit@gyte.edu.tr. Member AIAA.

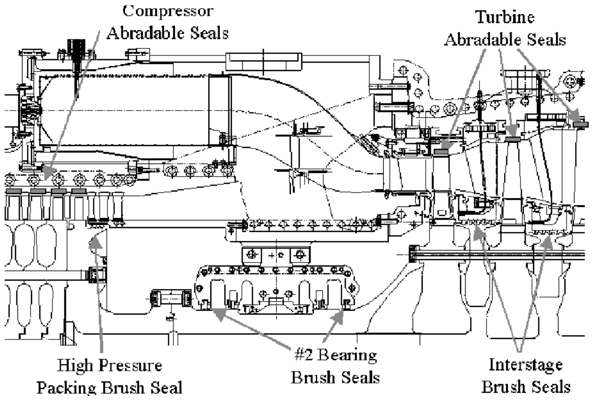

Fig. 1 Advanced dynamic seals locations in a Frame 7EA gas turbine.

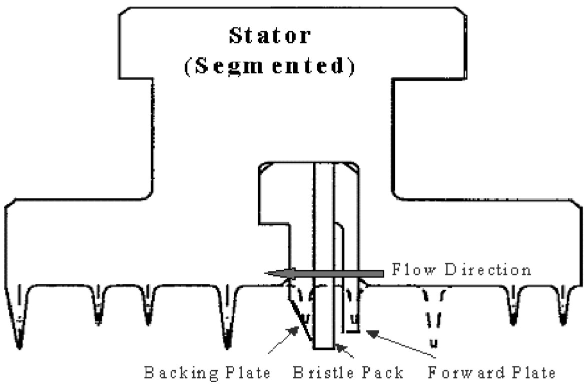

Fig. 2 Typical brush seal designs for steam turbine.

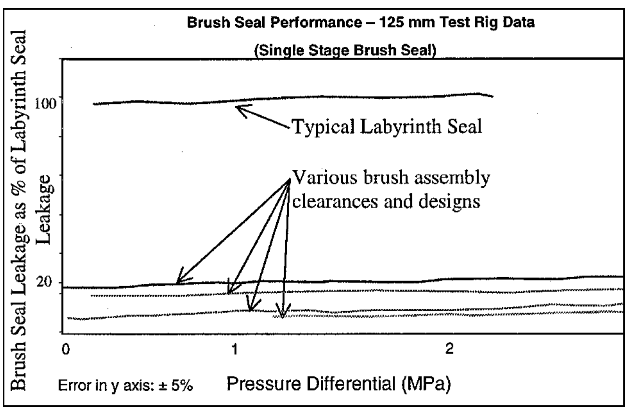

Fig. 3 Representative brush seal leakage data compared to a typical, 15-tooth, 0.5-mm (20-mil) clearance labyrinth seal.

characteristics, the seal material must have acceptable creep and oxidation properties. Commonly, cobalt-based alloys are used as bristle material due to their good wear and oxidation characteristics. There is no general requirement for coating rotor surfaces for industrial gas- and steam-turbine brush seal applications.

Seal Fence Height

A key design issue is the required radial gap (fence height) between the backing plate 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 plate gap is designed to avoid any contact between the rotor surface and the backing plate during any operating condition or under a set of dimensional variations.

Consequently, the design of an effective brush seal hinges on a thorough knowledge of the turbine behavior, operating conditions, and design of surrounding parts.

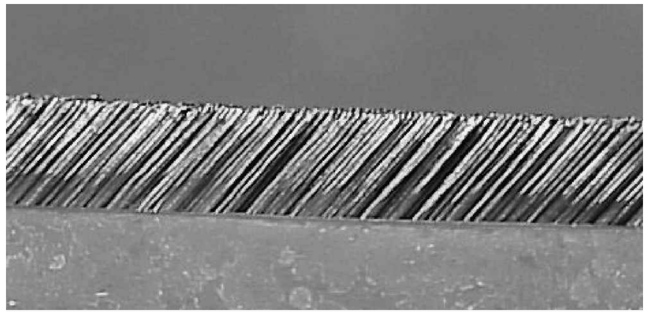

Fig. 4 HPP brush seal for 7EA gas turbine after 22,000 h of operation.

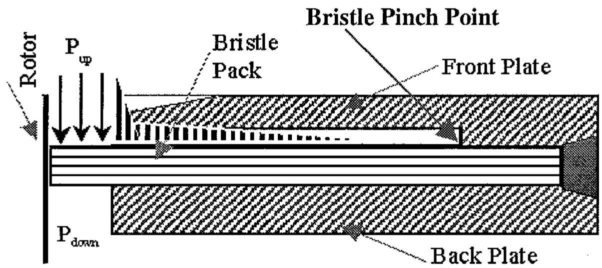

Fig. 5 Pressure forces acting on seal bristle pack.

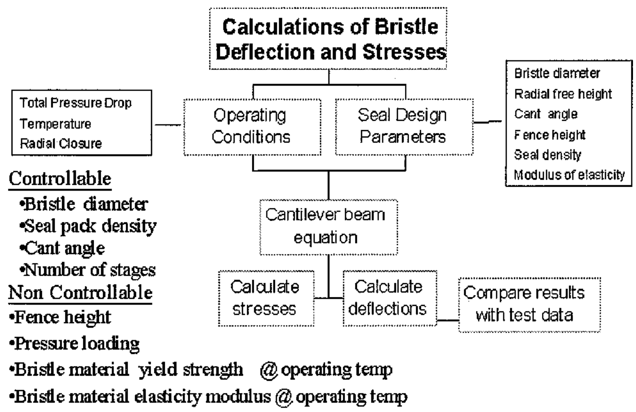

Fig. 6 Bristle stress/deflection analysis.

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 backplate. From a loading point of view, the bristles can be separated into two regions (see Fig. 5), the lower part, of fence region, between the rotor surface and the backplate i.d. and the upper part from the backplate i.d. to the bristle pinch point. The radially innermost portion carries the main pressure load and is the main source of the seal stress.[16] In addition to the mean bending stress, contact stress at the bristle/backplate 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 6 shows the seal stress and deflection analysis and includes a list the controllable and noncontrollable design parameters.

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. If the seal is not properly designed, this heating can lead to premature bristle loss, or, in the worst case, the rotor/seal operation could become thermally unstable. The latter condition occurs when the rotor grows radially into the stator, increasing the frictional

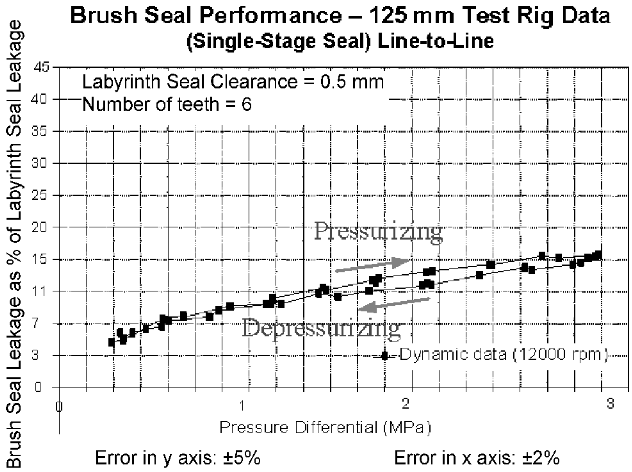

Fig. 7 Measured brush seal leakage characteristic.

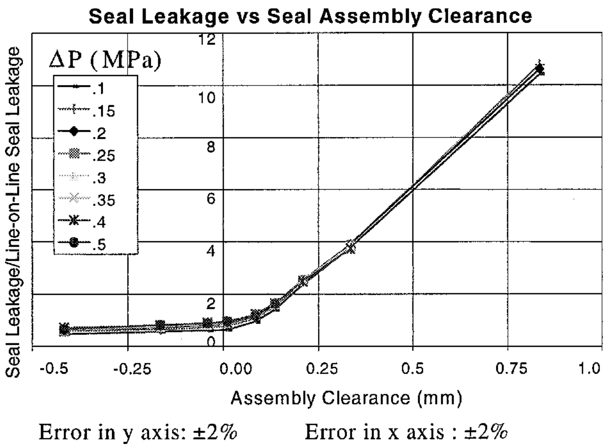

Fig. 8 Measured brush seal leakages for interference and clearance conditions.

heating, leading to additional rotor growth until the rotor rubs the seal backing plate, resulting in component failure. These mechanical design issues significantly affect the range of feasible applications for brush seals. Many of these issues have been addressed in more detail in a recent paper.[33]

Seal Leakage and Blowdown

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 Figs. 7 and 8. 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 that can be sensitive to radial rubs due to nonuniform heat generation. By measuring baseline seal leakage in a line-to-line (zero) assembly clearance configuration, bristle blowdown for varying levels of assembly clearance can be inferred from the leakage data (see Fig. 8).

Finally, leakage, blowdown, and heat generation data from rig tests are integrated into analytical tools used to design brush seals for new applications. Semi-empirical transfer functions are derived from test data or analytical equations and validated through testing. Figure 9 gives a summary of the current status of brush seal technology.

Steam-Turbine Brush Seals

There are several locations in steam turbines where applying brush seals can significantly improve performance. These include the interstage shaft packing, end packing, and bucket tip seals. To date, there are 10 GE steam turbines running with combinations of interstage packing, end packing, and bucket tip brush seals. These units range from a 20-MW industrial turbine to a 900-MW utility turbine. The first steam-turbine brush seals installed have recently been inspected and were in excellent condition after three years of service. Brush seals, which were installed more recently in a utility steam turbine, have also been inspected and were generally in excellent condition after 17 months of service. Figures 10 and 11 show photographs from this latter inspection. Figure 10 depicts an end packing brush seal. Figure 11 shows the stage 9 tip brush seal that rides over a row of integral cover buckets. Whereas bucket tip seals are still under development, steam-turbine shaft brush seals are now a robust product with validated leakage reduction and reliability performance. Development efforts continue to refine the current design and to expand the range of applications.

Transient Capability/ Stable Brush Seals	To the engines3-5 times dynamically more aggressive than current engines
High Surface Speed	From 120 m/s (AE) and 244 m/s of (GT) to 500 m/s
High Pressure Loading	From 0.69 MPa two stage seals to 2.76 MPa single stage seal design. Now going for higher ΔP multi-stage designs.
High Swirl Flow Field	0.3 swirl ratio of current applications to 0.6 – 1.2 region.
Temperature	From 370 °C to 650 – 1000 °C
Seal Life and Durability	Gas Turbine fleet leader w/40,000 field hrs with acceptable degradation.
	Steam Turbine fleet leader 28,000 hrs with minimal degradation
Rotor Surface	Ceramic coated Aircraft engine rotors to uncoated Gas and Steam turbine rotors
	Interrupted surface at bucket tip seals

Fig. 9 Brush seal technology status.

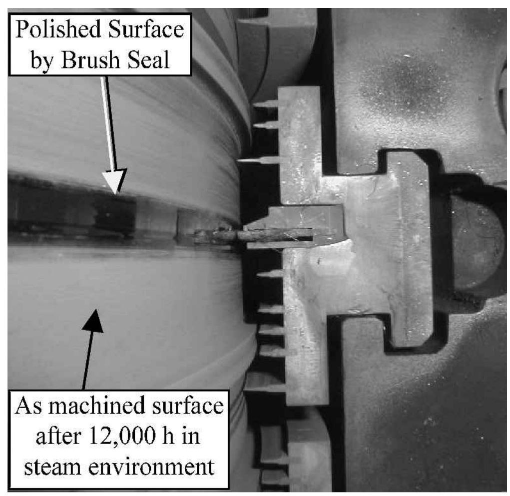

Fig. 10 Typical rotor surface finish under a steam-turbine end packing brush seal.

The steam-turbine environment adds some unique design considerations to be addressed to assure a robust and effective seal design and to minimize the impact of the seal on the steam turbine. The design challenges include very high operating pressures, rotor dynamics, turbine startup, and steam chemistry.

Steam leakage at the gaps between stationary parts and the rotor can account for as much as 29% of the total stage efficiency loss, and leakage at the end packings further reduces overall turbine efficiency.[8] The end packings and stages with the highest pressure

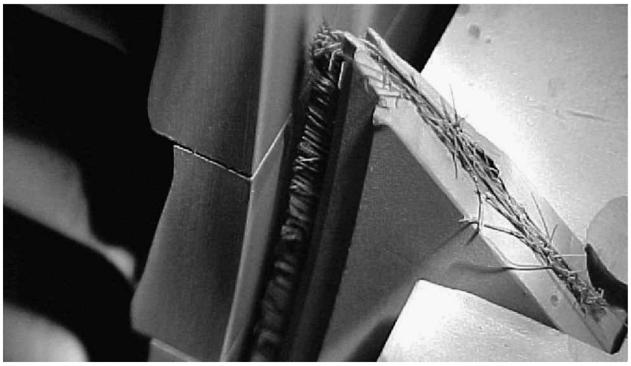

Fig. 11 Horizontal-joint view of brush seals above the buckets tips; design has been improved to eliminate bristles being caught between segments.

drop provide the greatest benefits for reducing leakage. Thus, incorporating brush seals will typically include interstage shaft seals toward the high pressure (HP) section of the turbine and at HP end packings. The number of brush seals in a turbine is influenced by critical speed considerations derived from a rotor dynamics system analysis. Such analyses and other system parameters indicate if it is feasible to use a brush seal in the low-pressure (LP) end of the turbine. These steam turbine design issues are discussed in more detail elsewhere.[8]

Abradable Seals

As the name suggests, abradable seal materials are worn in by the rotating blade during service. They are applied to the casing of gas and steam turbines to decrease clearances to levels difficult to achieve by mechanical means. Abradable seals are gaining appeal in gas turbines as a relatively simple means to reduce gas-path clearances in both the compressor and turbine. They offer clearance reductions at relatively low costs and minor engineering implications for the service fleet. Abradable seals have been in use in aviation gas turbines since late 1960s/early 1970s.[34] However, they have been used less in land-based gas turbines for power generation, primarily because of the long cycle times that the materials are in service. With increasing fuel prices and advances in materials to allow extended service periods, abradable seals are gaining popularity within the power generation industry.

Without abradable seals, the cold clearances between blade or bucket tips and shrouds must be large enough to prevent significant contact during operation. Use of abradable seals allows the cold clearances to be reduced with the assurance that if contact occurs, the sacrificial part will be the abradable material and not the blade or bucket tips. Also, abradable seals allow tighter clearances with common shroud or casing out of roundness.

Abradable seals are generally classified according to their temperature capability[35] 1) low temperature, usually for LP compressors, ambient to 400°C (750°F); 2) Midrange for LP and HP compressors, ambient to 760°C (1400°F); and 3) high temperature for HP turbines, from 760°C (1400°F) to 1150°C (2100°F).

Abradable seals can be also characterized by the method of application[36]: 1) castings for polymer-based abradable materials, 2) brazing or diffusion bonding for honeycomb and/or fiber metals (porous fiber metal structures), and 3) thermal spray coatings for a large range of powdered composite materials.

Many attempts have been made to study the wear mechanisms of abradable structures using conventional tribometers[37] or specially designed test rigs.[38,39] However, due to the high relative speeds, >100 m/s (>330 ft/s), between the abradable seal and the rotating blade tip surface, the mechanisms of wear/cutting differ considerably from low-speed tribology normally associated with machining operations. At high speeds, the removal/cutting of thermal spray abradable coating is done by release of small particle debris, that is, <0.1 mm (<0.004 in.). In contrast to conventional (low-speed) cutting in machine tools, the particle debris released in abradable materials is ejected at the rear of the moving blade.[40] This, therefore, partly sets the criteria for the design of such materials. It also sets a limiting design criterion for blade-tip thickness. Generally, a cutting element (blade-tip) thickness less than 1.3 mm (0.05 in.) allows release of the particles from the coating. Thicker tips tend to entrap the loose particles between the blade and the abradable material. As a result, special considerations have to be given to the design of the materials to allow for the cutting mechanisms (for example, altering the base material particle morphology and size).

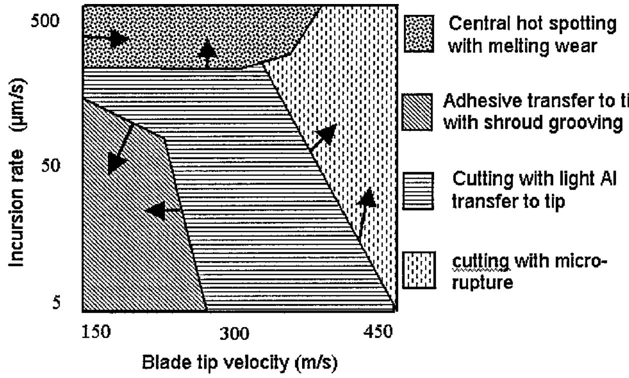

Fig. 12 Aluminium–silicon–polyester coating wear map.[47]

Certain abradable materials rely more on densification (compaction) of the structure than on particle debris removal.[41] Material compaction limits the functional depth of the abradable material because the compacted material will increase the wear of the rotating blade tips as the porosity is reduced. These types of seal materials include some of the thermal spray coatings and porous metal fiber structures (fiber metals). Fiber metals can be designed and constructed with varying fiber sizes and densities to alter their tribological behavior.[42,43]

Low-Temperature Abradable Seals

For thermally sprayed abradable coatings, different classes of coating materials behave tribologically differently. Traditionally, most of the powder metals available for low-temperature applications, that is, <400°C (<750°F), are aluminum–silicon based. To make them abradable, a second phase is added.[38] This phase is usually a polymeric material or a release agent and is often called a solid lubricant. The role of the second phase in aluminum–silicon based abradable material is primarily to promote crack initiation within the structure. The size, morphology, quantity, and material of the second phase determine the wear mechanisms and abradability of the seal coating under various tribological conditions. Figure 12 shows a typical wear map for an aluminum–silicon–polyester coating. This map displays the numerous wear mechanisms that are dominant under different blade-tip velocities and incursion rates when rubbed by a 3-mm (0.12-in.) thick titanium blade at ambient temperature. The arrows indicate the movement of wear mechanism boundaries when a stiffer polymer than polyester is used as the second phase.

Midtemperature Abradable Seals

For temperature applications up to 760°C (1400°F), Ni- or Co-based alloy powders are commonly used as the basis of the abradable seal matrix. Other phases are added to the base metal powder to make the material abradable. These added phases are polymeric materials that are fugitive elements used to generate coating porosity and act as release agents.[39,44]

Figure 13 displays a wear map of a midtemperature coating system abraded at 500°C (930°F) using titanium blades. The map shows the wear mechanism domains vs blade-tip velocity and incursion rate. The arrows indicate the movement of the wear regime boundaries as the polyester level increases. As polyester content and, thus, porosity increases, cutting becomes increasingly predominant mechanism over the entire range of the speeds and incursion rates.

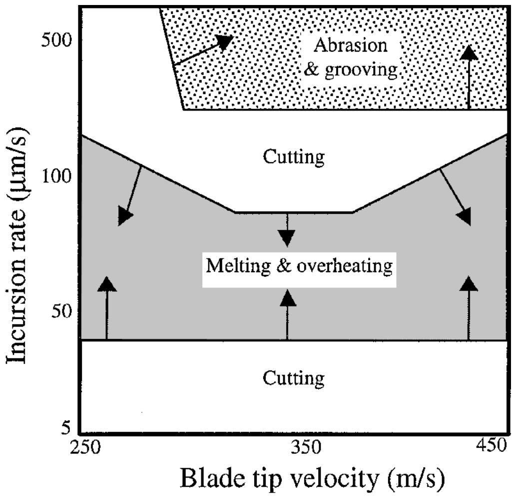

Fig. 13 Midtemperature abradable coating (CoNiCrAlY) wear map at 500°C (930°F) by titanium blades.[43]

However, increasing porosity has a negative effect on coating cohesive strength and erosion properties.

High-Temperature Abradable Seals

For operating temperatures above 760°C (1400°F), common practice is to use porous ceramics as the abradable material. The most widely used material is yttria-stabilized zirconia (YSZ), which is usually mixed with a fugitive polymeric phase. There are a number of important considerations to make regarding porous ceramic abradable materials. To achieve an acceptable abradability, the cutting element/blade generally has to be reinforced with hard abrasive grits. Choosing these grits and processes to apply them has been the subject of numerous research activities. There are a number of patents that deal with this aspect of ceramic abradable materials.[45–49] Abrasive grits considered include cubic boron nitride (cBN), silicon carbide, aluminum oxide, and zirconium oxide. Published data suggest that cBN particles of a given size range tend to be the best abrasive medium against YSZ porous ceramics.[45,48] This is because cBN poses a high hardness (second to diamond) and a high sublimation temperature, >2980°C (>5400°F), which makes it an ideal candidate to abrade ceramic abradable materials. However, cBN's relatively low oxidation temperature, ~850°C (~1560°F), allows it to function for only a limited time. This has prompted the use of other abrasives such SiC.[46,49] Despite successful functionality of SiC against YSZ, SiC has been met with limited enthusiasm. SiC requires a diffusion barrier to prevent its reaction with transition metals at elevated temperatures.[50] This adds to the complexity and the cost of the abrasive system.

The ceramic abradable coating microstructure and its porosity is another essential consideration. Clearly, the higher the porosity, the more abradable the coating is. However, YSZ is strongly susceptible to high-angle erosion because of its brittle nature,[51] and adding porosity makes it prone to low-angle erosion also. Thermally sprayed porous YSZ coatings show different tribological behavior when compared to metallic abradable materials. They tend to show a strong influence of blade-tip velocity on abradability (see Fig. 14).[40] Abradability tends to improve with increasing blade-tip velocity. On the other hand, porous YSZ coatings show less dependency on incursion rate. They tend to have poor abradability at very low incursion rates, <0.005 mm/s (<0.2 mils/s), thus requiring blade-tip treatments.

Designing Abradable Materials for Turbomachinery

Because abradable seals are low-strength structures that wear without damaging the mating blade tips, they are also susceptible to gas and solid particle erosion. Abradable structures intended for use in harsh temperatures occurring in gas turbines can also be prone to oxidation because of the inherent material porosity. These

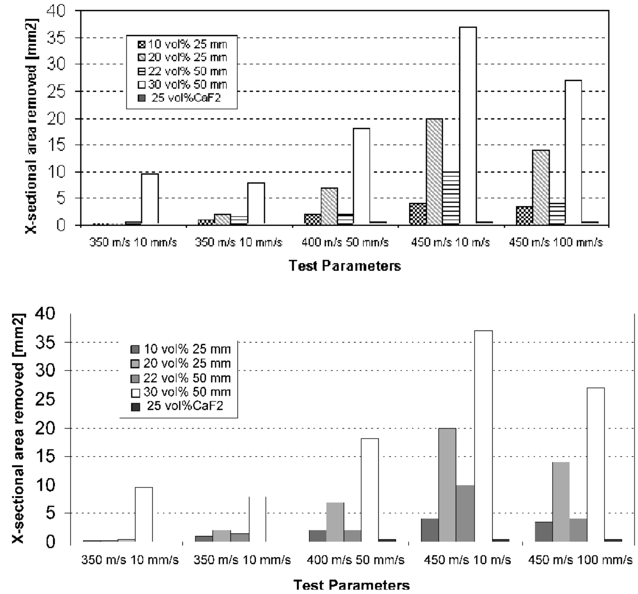

Fig. 14 Abradability of high-temperature materials by SiC tipped blades at 1025°C (1880°F): right set of data is for a CaF abradable, other data are for YSZ with polyamide; the *x* axis lists velocity and incursion rates, and the legend gives porosity levels and average pore sizes after the polyamide is burnt out.[49]

conflicting properties need to be accounted for in designing abradable seals. Abradable seals then have to be considered as a complete tribological system that incorporates 1) relative motions and depth of cut, blade-tip speed and incursion rate; 2) environment temperature, fluid medium, and contaminants; 3) cutting element geometry and material, blade-tip thickness, shrouded or unshrouded blades; and 4) counter element, abradable seal material and structure. Manufacturing processes as well as microstructural consistency of abradable seals can have a profound effect on their properties.[42,52]

Considering the aforementioned elements makes the abradable system quite unique, that is, designed to suit the particular application. Thus, despite the availability of many off-the-shelf materials, abradable seals have to be modified or redesigned in most applications to meet the design constraints. More extensive lists of references on abradable seals and their use have been published elsewhere.[40,42]

Conclusions

Implementation of advanced seals into industrial turbines has progressed well over the last few years. Applications have been seen in gas turbines, industrial and utility steam turbines, aeroengines, industrial compressors, and generators. Brush seals have been incorporated into many labyrinth seal locations, and abradable seals have been employed to reduce blade-tip clearances. These sealing improvements have resulted in significant reductions in parasitic leakage flows, thereby increasing turbine efficiency, power output, and, in some cases, reducing emissions.

Extensive development efforts have been carried out for the various types of advanced seals and application. These include the following:

1) Brush seals have been implemented into many industrial gas turbines at the compressor discharge, middle bearing, and turbine interstage locations. Experience has shown that these seals operate well for their expected life (24,000 h and more) with significant turbine performance benefits (up to 2% combined).

2) Brush seals are also being applied to interstage shaft packings, end packings, and opposite the bucket tips in steam turbines. Inspection of brush seals from initial applications shows that these brush seals are operating well. Unique design issues for steam turbines include very high operating pressures, rotor dynamics, turbine startup, and steam chemistry.

3) Abradable seals offer an effective means of improving turbine performance through reduced clearances. These seals are being applied to the casings and shrouds of gas and steam turbines to decrease the blade-tip clearances beyond what is practical by mechanical means. Abradable materials are selected based on the environment temperature, that is, relatively low temperatures (compressors), midtemperature (HP compressors and LP turbines), and high temperatures (HP turbines). Besides temperature, other issues include the method of application, cutting mechanism, need for blade-tip treatment, etc. As for all advanced seals, many engineering and material issues need to be addressed to ensure an abradable seal is properly designed and applied for the specific turbine application.

Acknowledgments

The advanced dynamic seal development work summarized in this paper was carried out primarily at General Electric (GE) Global Research and at GE Power Systems, who sponsored the work. The authors would like to acknowledge the contributions of the many other GE authors of publications on which this paper was based; Mark Burnett, Frederick Brunner, Paul Chiu, Robert Cromer, Gregory Crum, Saim Dinc, Mark Florin, Gayle Goetze, James Hopkins, Iain Kellock, Onika Kerber, James Lawen, Paul Marks, John Maupin, James Maynard, Jason Mortzheim, Ryan Pastrana, George Reluzco, Albert Stuck, and Christopher Wolfe.

References

[1]"GE Advanced Sealing Technologies," General Electric Brochure, Atlanta, GA, 2000.

[2]Turnquist, N. A., Tseng, T., McNickle, A., and Steinetz, B., "Full Scale Testing of an Aspirating Face Seal with Angular Misalignment," AIAA Paper 99-2682, June 1999.

[3]Askit, M. F., Chupp, R. E., Dinc, O. S., and Demiroghu, M., "Advanced Seals for Industrial Turbine Applications: Design Approach and Static Seal Development," *Journal of Propulsion and Power*, Vol. 18, No. 6, 2002, pp. 1254–1259.

[4]Chupp, R. E., and Holle, G. F., "Generalizing Circular Brush Seal Leakage Through a Randomly Distributed Bristle Bed," *Journal of Turbomachinery*, Vol. 118, No. 1, 1996, pp. 158–160.

[5]Wolfe, C. E., Chiu, R. P., Cromer, R. H., Crum, G. A., Marks, P. T., Stuck, A. E., Turnquist, N. A., Reluzco, G., and Dinc, O. S., "Brush Seals in Industrial Gas Turbines," AIAA Paper 97-2730, July 1997.

[6]Dinc, S., Reluzco, G., Turnquist, N. A., Lawen, J., Kerber, O., Brunner, F., Crum, G., Stuck, A. E., Cromer, R. H., Marks, P., Chiu, P., Wolfe, C. E., and Crudgington, P., "Brush Seals in Industrial Gas Turbines—Turbine Section Interstage Sealing," AIAA Paper 98-3175, July 1998.

[7]Dinc, S., Demiroglu, M., Turnquist, N., Mortzheim, J., Goetze, G., Maupin, J., Hopkins, J., Wolfe, C., and Florin, M., "Fundamental Design Issues of Brush Seals for Industrial Applications," *Journal of Turbomachinery*, Vol. 124, No. 2, 2002, pp. 293–300; also American Society of Mechanical Engineers, ASME Paper 2001-GT-0400, June 2001.

[8]Pastrana, R. M., Wolfe, C. E., Turnquist, N. A., and Burnett, M. E., "Improved Steam Turbine Leakage Control with a Brush Seal Design," Texas A&M 30th Turbomachinery Symposium, Sept. 2001, pp. 33–38.

[9]Chupp, R. E., Johnson, R. P., and Loewenthal, R. G., "Brush Seal Development for Large Industrial Gas Turbines," AIAA Paper 95-3146, July 1995.

[10]Chupp, R. E., Prior, R. J., and Loewenthal, R. G., "Update on Brush Seal Development for Large Industrial Gas Turbines," AIAA Paper 96-3306, July 1996.

[11]Chupp, R. E., Prior, R. J., Loewenthal, R. G., and Menendez, R. P., "Advanced Seal Development for Large Industrial Gas Turbines," AIAA Paper 97-2731, July 1997.

[12]Willey, L. D., Maughan, J. R., Hill, J. M., and Walsh, D. J., "New Steam Turbine Test Vehicle for the Verification of Improved Efficiency Power Generation Steam Turbines," *Proceedings of IMECE-2000 International Mechanical Engineering Conference and Exposition*, Orlando, FL, Nov. 2000, pp. 1–9.

[13]Arora, G. K., and Proctor, M. P., "JTAGG II Brush Seal Test Results," AIAA Paper 97-2632, July 1997.

[14]Berard, G., and Short, J., "Influence of Design Features on Brush Seal Performance," AIAA Paper 99-2685, June 1999.

[15]Chen, L. H., Wood, P. E., Jones, T. V., and Chew, J. W., "An Iterative CFD and Mechanical Brush Seal Model and Comparison with Experimental Results," *Journal of Engineering for Gas Turbines and Power*, Vol. 121, No. 4, 1999, pp. 656–661; American Society of Mechanical Engineers, ASME Paper 98-GT-372, June 1998.

[16]Chen, L. H., Wood, P. E., Jones, T. V., and Chew, J. W., "Detailed Experimental Studies of Flow in Large Scale Brush Seal Model and a Comparison with CFD Predictions," *Journal of Engineering for Gas Turbines and Power*, Vol. 122, No. 4, 2000, pp. 672–679; American Society of Mechanical Engineers, ASME Paper 99-GT-218, June 1999.

[17]Crudgington, P. F., "Brush Seal Performance Evaluation," AIAA Paper 98-3172, July 1998.

[18]Crudgington, P. F., "Recent Brush Seal and Testing Developments at Cross," AIAA Paper 2001-GT-3480, July 2001.

[19]Demiroglu, M., Aksit, M. F., and Tichy, J. A., "A Numerical Study of Brush Seal Leakage Flow," AIAA Paper 98-3173, July 1998.

[20]Fellenstein, J. A., DellaCorte, C., Moore, K. D., and Boyes, E., "High Temperature Brush Seal Tuft Testing of Metallic Bristles vs Chrome Carbide," AIAA Paper 96-2908, July 1996.

[21]Fellenstein, J. A., DellaCorte, C., Moore, K. D., and Boyes, E., "High Temperature Brush Seal Tuft Testing of Selected Nickel–Chrome and Cobalt–Chrome Super Alloys," AIAA Paper 97-2634, July 1997.

[22]Gail, A., and Beichl, S., "MTU Brush Seal—Main Features of an Alternative Design," AIAA Paper 2000-3375, July 2000.

[23]Hendricks, R. C., Wilson J., Wu, T., and Flower, R., "Bidirectional Brush Seals, American Society of Mechanical Engineers, ASME Paper 97-GT-256, June 1997.

[24]Lattime, S. B., Braun, M. J., Choy, F. K., Hendricks, R. C., and Steinetz, B. M., "Advances in Hybrid Floating Brush Seals," AIAA Paper 98-3171, July 1998.

[25]Lattime, S. B., Braun, M. J., Choy, F. K., Hendricks, R. C., and Steinetz, B. M., "Continued Evaluation of the Hybrid Floating Brush Seal (HFBS)," AIAA Paper 99-2688, June 1999.

[26]Menendez, R. P., and Xia, J., "Recent Developments in Brush Seals for Large Industrial Gas Turbines," AIAA Paper 2000-3374, July 2000.

[27]O'Neill, A. T., Hogg, S. I., Withers, P. A., Turner, M. T., and Jones, T. V., "Multiple Brush Seals in Series," American Society of Mechanical Engineers, ASME Paper 97-GT-194, June 1997.

[28]Prior, R., Short, J., and Basu, P., "Brush Seal Wear Model," AIAA Paper 98-3170, June 1998.

[29]Short, J. F., Basu, P., Datta, A., Loewenthal, R. G., and Prior, R. J., "Advanced Brush Seal Development," AIAA Paper 96-2907, July 1996.

[30]Tseng, T. W., Short, J. F., and Steinetz, B. M., "Development of a Low Hysteresis Brush Seal Modern Engine Applications," AIAA Paper 99-2683, June 1999.

[31]Turner, M. T., Chew, J. W., and Long, C. A., "Experimental Investigation and Mathematical Modeling of Clearance Brush Seals," *Journal of Engineering for Gas Turbines and Power*, Vol. 120, No. 3, 1998, pp. 573–578; American Society of Mechanical Engineers, ASME Paper 97-GT-282, June 1997.

[32]Wood, P. E., and Jones, T. V., "A Test Facility for the Measurement of Torques at the Shaft to Seal Interface in Brush Seals," *Journal of Engineering for Gas Turbines and Power*, Vol. 121, No. 1, 1999, pp. 160–166; American Society of Mechanical Engineers, ASME Paper 97-GT-184, June 1997.

[33]Soditus, S. M., "Commercial Aircraft Maintenance Experience Relating to Current Sealing Technology," AIAA Paper 98-3284, July 1998.

[34]Morrell, P., Betridge, D., Greaves, M., Dorfman, M., Russo, L., Britton, C., and Harrison, K., "A New Aluminium–Silicon/Boron Nitride Powder for Clearance Control Application," ITSC 98, ASM International, Materials Park, OH, 1998.

[35]Ghasripoor, F., Schmid, R., and Drofman, M., "Abradables Improve Gas Turbine Efficiency," *Journal of the Institute of Materials*, Vol. 5, No. 6, 1997.

[36]Schmid, R. K., Ghasripoor, F., Dorfman, M., and Wei, X., "An Overview of Compressor Abradables," ITSC 2000, ASM International, Materials Park, OH, 2000.

[37]Guilemany, J. M., Navarro, J., Lorenzana, C., Vizcanio, S., and Miguel, J. M., "Tribological Behaviour of Abradable Coatings Obtained by Atmospheric Plasma Spraying (APS)," Thermal Spray 2001, ASM International, Materials Park, OH, May 2001.

[38]Ghasripoor, F., Schmid, R. K., Dorfman, M., and Russo, L., "A Review of Clearance Control Wear Mechanisms for Low Temperature Aluminum Silicon Alloys," ITSC 98, ASM International, Materials Park, OH, 1998.

[39]Nava, Y., Mutasim, Z., and Coe, M., "Abradable Coatings for Low-Temperature Applications," Thermal Spray 2001, ASM International, Materials Park, OH, May 2001.

[40]Schmid, R., "New High Temperature Abradables for Gas Turbines," Ph.D. Dissertation, Thesis 12223, Dept. of Materials, Swiss Federal Inst. of Technology, Zurich, 1997.

[41]Borel, M. O., Nicoll, A. R., Schlaepfer, H. W., and Schmid, R. K., "Wear Mechanisms Occurring in Abradable Seals of Gas Turbines," *Surface Coating Technology*, Vol. 39, 1989, pp. 117–126.

[42]Chappel, D., Vo, L., and Howe, H., "Gas Path Blade Tip Seals: Abradable Seal Material Testing at Utility Gas and Steam Turbine Operating Conditions," American Society of Mechanical Engineers, ASME Paper 2001-GT-0583, June 2001.

[43]Chappel, D., Howe, H., and Vo, L., "Abradable Seal Testing: Blade Temperatures During Low Speed Rub Event," AIAA Paper 2001-3479, July 2001.

[44]Ghasripoor, F., Schmid, R., Dorfman, D., and Wei, X., "Optimizing the Performance of Plasma Control Coatings up to 850C," Surface Modification Technologies XII, ASM International, Materials Park, OH, 1998.

[45]Shell, J. D., and Farr, H. J., "Abrasive Ceramic Matrix Turbine Blade Tip and Method for Forming," U.S. Patent 5,952,110, Sept. 1999.

[46]Ghasripoor, F., Schmid, R. K., and Dorfman, M., "Silicon Carbide Composition for Turbine Blade Tips," U.S. Patent 5,997,248, Dec. 1999.

[47]Benoit, R., Beverly, E. M., Love, C. M., and Mack, G. J., "Abrasive Blade Tip," U.S. Patent 5,603,603, Feb. 1997.

[48]Draskovich, B. S., Frani, N. E., Joseph, S. S., and Narasimhan, D., "Abrasive Tip/Abradable Shroud System and Method for Gas Turbine Compressor Clearance Control," U.S. Patent 5,704,759, Jan. 1998.

[49]Johnson, G. F., and Schilke, P. W., "Alumina Coated Silicon Carbide Abrasive," U.S. Patent 4,249,913, Feb. 1981.

[50]Pan, Y., and Baptista, J., "Chemical Stability of Silicon Carbide in Presence of Transition Metals," *Journal of the American Ceramic Society*, Vol. 79, No. 8, 1996, pp. 2017–26.

[51]Hutchings, I. M., "Erosion by Solid Particle Impact," *Tribology; Friction and Wear of Engineering Materials*, Edward Arnold, London, 1992, Sec. 6.4, pp. 171–197.

[52]Wei, X., Mallon, J. R., Correa, L. F., M. Dorfman, M., and Ghasripoor, F., "Microstructure and Property Control of CoNiCrAlY Based Abradable Coatings for Optimal Performance," ITSC 2000, ASM International, Materials Park, OH, 2000.