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Advanced Seals for Industrial Turbine Applications: Design Approach and Static Seal Development

Mahmut F. Aksit*
Gebze Institute of Technology, 41400 Gebze, Turkey
Raymond E. Chupp† and O. Saim Dinc‡
General Electric Global Research Center, Niskayuna, New York 12309
and
Mehmet Demiroglu§
Rensselaer Polytechnic Institute, Troy, New York 12180

Changes in the market place are imposing increasing demands to improve efficiency (decreasing heat rate) and power output for both existing and new industrial turbines. The improvement is to be done while maintaining or decreasing emission levels. This demand has led to extensive efforts to improve the performance of the various components in industrial gas turbines, steam turbines, compressors, and generators. One of the critical areas being addressed is reducing the parasitic leakage flows through the various static and dynamic seals. Implementing advanced seals into industrial turbines has progressed well over the last several years, with significant operating performance gains achieved. Advanced static seals have been placed in gas-turbine hot gas-path junctions and steam-turbine packing ring segment end gaps. The status of efforts to develop and implement advanced static seals in industrial turbines is summarized. The design approach following design-for-six-sigma methodology is summarized, and the development efforts for each static seal type are presented.

Introduction

IMPROVED sealing in industrial gas and steam turbines reduces parasitic leakages and gives better control of the secondary flow system. This results in significantly improved performance, both in efficiency (heat rate) and power output.[1] The performance gains being achieved in gas turbines for each sealing location range from a 0.2 to 0.6% reduction in unit heat rate and from a 0.3 to 1.0% increase in power output. Similarly, for steam turbines, overall unit heat rate decreases of 0.1–0.8% are achieved depending on the location, such as end or interstage packing, and turbine section.

Improved sealing in turbomachinery has been under development for several years. Applications have been in gas turbines, steam turbines, aeroengines, industrial compressors, and generators. Static seals have been introduced in gas-turbine combustor and hot gas-path junctions, as well as steam-turbine packing ring segment end gaps. Advanced dynamic seals are being applied at compressor blade tips, high-pressure packing, bearings, turbine interstages, and bucket tip locations. Steam-turbine advanced dynamic seal applications also include end packings.

Various aspects of developing and applying advanced sealing in industrial gas turbines have been described previously.[2–13] The purpose of this paper is to summarize such efforts at General Electric. The paper will include discussion of the development approach, analysis methods, test facilities, and cloth seals for static applications. Figure 1 shows representative static sealing areas that have been addressed in an industrial gas turbine. Dynamic seals are the subject of a companion paper.[14]

Approach

For each turbine model, various sealing locations are examined to determine which areas to address. For candidate locations, possible improved sealing concepts are devised based on previous experience and new approaches. A benefit vs cost analysis is performed to identify the best candidate sealing areas to improve. A system-level analysis is performed with the sealing improvements incorporated to assess impact on turbine operation and hardware life, as well as potential performance gains. Figure 2 gives an overview of a system-level assessment for a gas turbine. Often, there is more leakage through the seals than is necessary for cooling components and purging of hot gas downstream of the seals. These areas provide opportunities for improved sealing. However, the leakage can not simply be decreased to the minimum levels possible with advanced seals without regard to system interactions. The seals must be designed and the flow system modified to maintain required flow rates. Failure to do so could result in decreased part lives or premature failures.

The design/development approach to improve sealing for each application follows design-for-six-sigma (DFSS) methodology[15,16] as depicted in Fig. 3. This ensures that the critical-to-quality (CTQ) parameters for the application are addressed. The CTQs start at the customer level and flow down to CTQs at the various analysis, design, testing, and validation levels. Scorecards are maintained to track the process, and tollgate reviews are held periodically to ensure that all application issues and CTQs are met.

Development is pursued for selected improved sealing areas using design criteria established to satisfy system requirements. The design environment and operating conditions (e.g., temperature, pressure, pressure drop, speed, closure rate) are defined for the range of operating conditions to be encountered. The transient relative movement cycle is especially important for both static and dynamic seals. This defines the geometric operating envelope and the range between maximum movement and the steady-state position where leakage performance is most important. The transient movement information also defines the potential for heat generation and surface wear between mating parts in dynamic sealing applications.

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

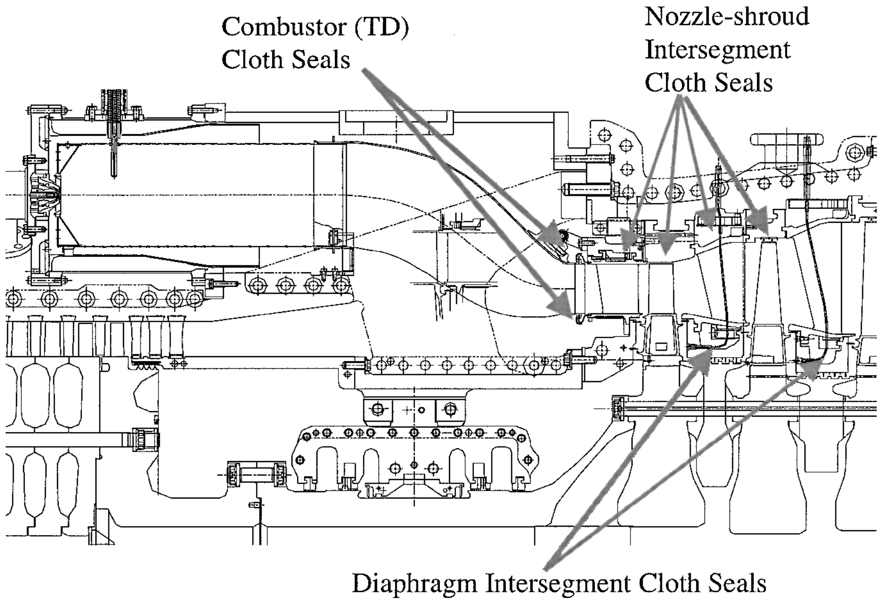

Fig. 1 Advanced static seal locations in a Frame 7EA gas turbine.

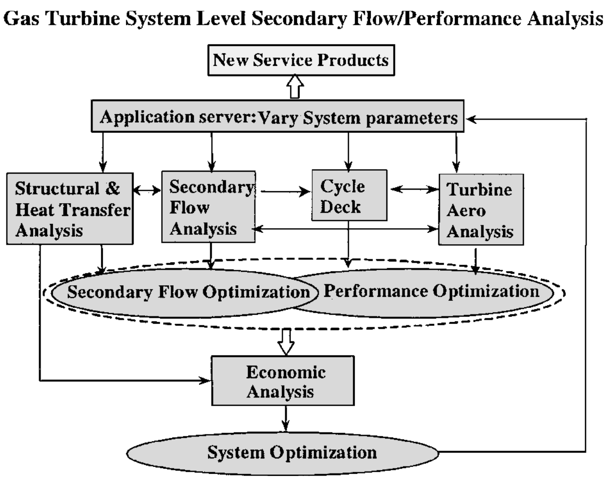

Fig. 2 Gas-turbine system-level assessment of design mods.

Detailed analyses of the seal and surrounding turbine region are performed to properly select seal design parameters. Candidate seals are then designed to meet the requirements. During this process, close coordination is maintained with potential seal vendors to ensure that the resultant seals can be cost effectively produced. Seal designs are then manufactured by the vendors and rig tested. Finally, prototypes of successful seal designs are manufactured and validation tested in operating turbine units.

Analysis Methods

A number of tools have been developed to aid in the design of advanced seals. These are based on analytical equations, finite element analysis, computational fluid dynamics, and flow network models, among others. Where possible, tests are performed to support the analytical results and to provide semi-empirical, physics-based transfer functions to aid in design. For example, an extensive brush seal design procedure has been established that includes several design tools, that is, pressure capability of bristle pack, leakage characteristics, radial stiffness, bristle blowdown, bristle tip heat generation, bristle natural frequency, etc. These tools streamline the design process and ensure that design constraints and CTQs are met.

Experimental Facilities

Several experimental rigs are used to quantify the performance and characteristics of advanced static and dynamic seals:

1) A leakage performance test rig is used for testing both static and dynamic seals. The rig is shaped like a shoebox. It is a high-pressure, high-temperature rig that gives comparative leakage performance data for various seal designs, such as cloth, labyrinth, honeycomb, brush, C, and E type seals. As illustrated in Fig. 4, the rig is basically a pressure vessel with high- and low-pressure chambers and a jaw assembly that holds a linear seal sample. Removing the top cover allows easy access to mount the text fixture on the base plate. A purge flow line to the rig provides flow for preheating to accelerate testing and to maintain a constant temperature for low leakage rates. Seal slots are formed by an assembly of jaws. By changing jaw units, various seal designs can be easily tested. Motion of the jaws allows various offsets, mismatches, skews, and gaps. The depth and the height of the slots can be changed by replacing the spacers between the jaws. Air or steam is the working fluid. Capabilities are 3.1-MPa (450-psia) inlet and exit pressure, 540°C (1000°F) air temperature, 400°C (750°F) steam temperature, 0.9-kg/s (2-lbm/s) flow rate, and 30.5-cm (12-in.) length seal samples.

2) A sliding wear test fixture is shown in Fig. 5. This test rig is used to characterize wear behavior of various candidate seal materials. The rig consists of a stationary flat surface and a moving pin loaded against this surface. Various seal and slot material pairs can be tested under different operating conditions. The setup allows heating of the contact area up to 800°C (1470°F). Constant contact load is applied by hanging weights through a lever system. Sliding motion is generated by a pneumatic linear motor that can provide a speed range from cycles per second to cycles per hour. The test conditions are established according to the application, such as a 0.7-MPa unit load on the samples and 1.8-cm stroke length at 1 cycle per second at various temperatures. Typically, wear tests are run for a minimum of 6 h. The friction coefficient is monitored continuously and sample weight loss is measured periodically.

3) A low cycle fatigue (LCF) test rig is used to simulate cyclic strain loads, which are usually experienced by seals during startup and shutdown. The LCF rig is designed to generate seal deflections in one direction. The motion is generated by a pneumatic linear motor that can provide a speed range from cycles per second to cycles per day. By the use of an advanced controller, push and pull speeds and dwell times at either ends of the stroke can be controlled independent of one another. To capture thermal effects, the rig is equipped with an insulated enclosure that has dual heaters capable of maintaining test temperatures up to 800°C (1470°F).

4) A smaller rotary rig, shown in Fig. 6, is employed for dynamic seal testing in air or steam up to 3.1 MPa (air) and 8.3 MPa (steam) (450 and 1200 psia). It is used to test subscale seals at full turbine

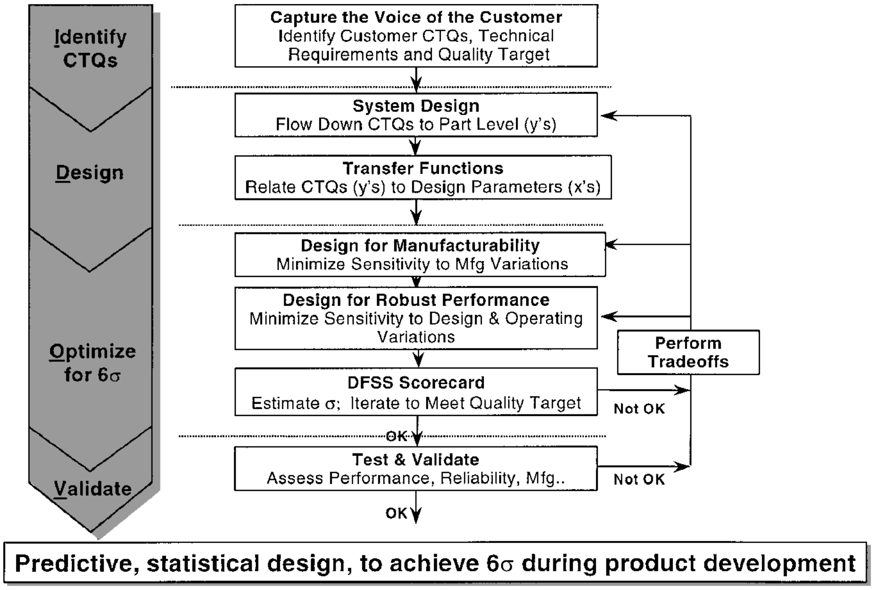

Fig. 3 Overview of DFSS methodology.

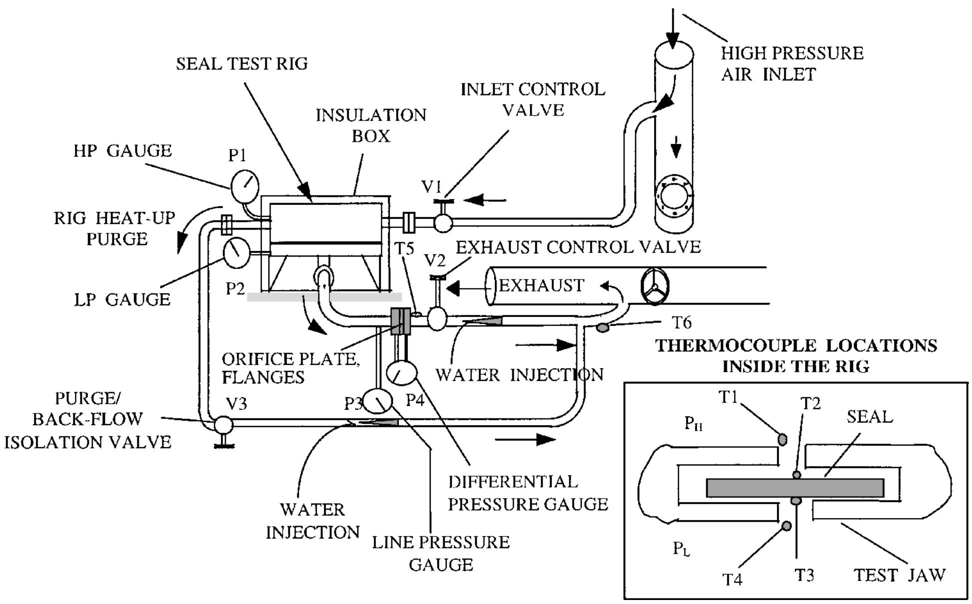

Fig. 4 System layout of shoebox test rig.

conditions (speed, pressure, temperature, and movement cycle). Capabilities are 13-cm (5.1-in.) diameter, 2.1-MPa (300 psia) exit pressure, 550°C (air) and 400°C (steam) (1000 and 750°F) temperature, 244-m/s (800-ft/s) surface speed, 0.9-kg/s (air) and 0.7-kg/s (steam) (2.0- and 1.5-lbm/s) flow rate, and ±1.9-cm (±.0.75-in.) axial motion.

5) A larger rotary rig is used for dynamic testing in air up to 0.86 MPa (125 psia). It is used to test full-scale seals at subscale conditions (see Fig. 7). Capabilities are 0.91-m (36-in.) diameter, which can be configured to 1.25-m (50-in.) diameter, 0.86-MPa (125-psia) exit pressure, 38°C (100°F) temperature, 150-m/s (500 ft/s) surface speed, and 5.4-kg/s (12-lbm/s) flow rate. Aspirating face seals and brush seals have been tested in this rig.

6) An abradable rub rig is shown in Fig. 8. It is a versatile rig for testing candidate abradable shroud materials rubbing against rotating blades or knife edges. The abradable coating samples are put on a stationary shroud and pushed toward the rotating blades or the disk. The rig can simulate turbine blade-tip rotation up to 300 m/s (~1000 ft/s) and incursion speeds from a minimum of 0.0005 mm/s (0.02 mils/s) up to 38 mm/s (1500 mils/s). It has a heating capability for the shroud to operate at turbine environment temperatures up to 927°C (1700°F). Wear characteristics of shroud and blade tribopair materials are determined from the shroud and blade or knife-edge tip wear. The rub rig can simulate radial as well as axial incursions.

7) The Steam Turbine Test Vehicle is a 3.5-MW boiler feed pump turbine that has been modified to model the thermodynamic characteristics of a four-admission large steam turbine. Data for brush seals inserted into six interstage locations have validated predicted performance gains using brush seals.

Static Seals

Advanced static sealing applications in gas turbines include the junctions between the stationary components (combustors, nozzles, shrouds, etc.) throughout the internal cooling flowpath. Typically, adjacent members have to sustain relative vibratory motion with

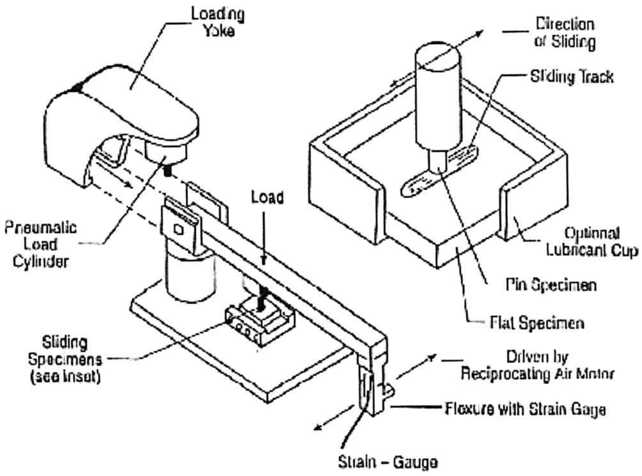

Fig. 5 Sliding wear test rig.

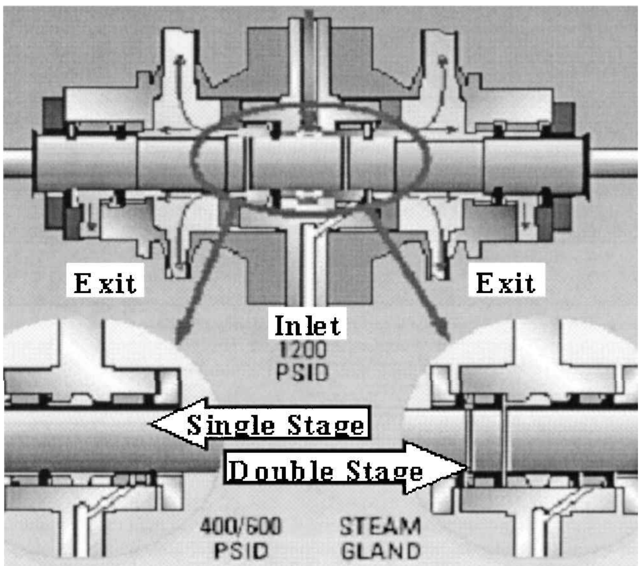

Fig. 6 Schematic view of 13-cm (5.1-in.) rotary seal test rig.

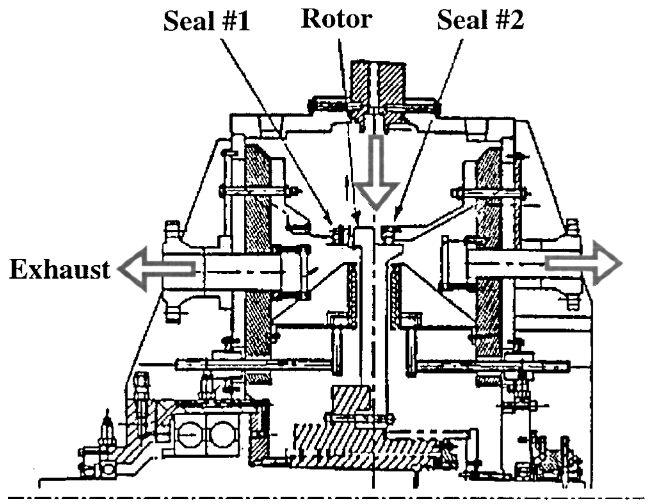

Fig. 7 Cross section view of 0.91-m (36-in.) rotary seal test rig.

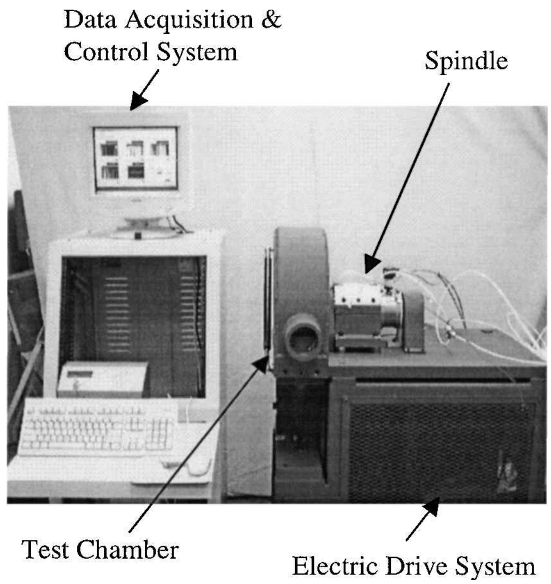

Fig. 8 Abradable rub test rig. Fixed shroud samples can be tested against shrouded and unshrouded rotor blades.

minimal wear or loss of sealing. In addition, they must accommodate thermal growth and misalignment. Modern gas turbines require high-temperature, low-leakage, and compliant seals to control parasitic leakage flows between turbine components. Effective sealing not only increases turbine efficiency and output, but also improves the main gas-path temperature profile.

Combustor Seals

In gas turbines with can-annular combustion systems, the combustor seals are used to seal the gap between the cans [transition duct (TD)] and the first-stage nozzles (FSN). Combustion dynamics and excessive thermal misalignments make combustor sealing more challenging than the nozzle-shroud intersegment sealing. A typical sealing junction involves two TDs (cans) and FSN segments. Large axial offsets and relative skew/misalignments between neighboring cans are quite common. As shown in Fig. 9, these junctions are typically sealed using formed metal strips designed to take relative axial and radial motion by sliding in grooves machined in the TD and FSN. However, the FSN is made of segments that experience relative misalignments, causing the seal to stick in the FSN slot. Jamming the seal on the FSN side results in wearing of the seals by the TD due to the relative dynamic motion. Heavy wear on the seal and in the TD slots is commonplace. Seal failure has caused occasional forced power outages.

The need for flexibility at the TD–FSN junction was addressed first by the use of brush seals[5] and later by the use of cloth seals.[6,7] As illustrated in Fig. 10, TD–FSN cloth seals utilize a radial lip formed by a flexible cloth–shim assembly. The cloth seals also incorporate an interference fit, allowing for a uniform seal–slot contact under any condition, thereby providing reduced leakage. When the seal is jammed in the FSN slot, relative vibratory motion is absorbed by flexing of the cloth assembly rather than the wearing on the rigid seal frame.

The introduction of flexible combustor cloth seals demonstrated the potential to extend service life by 50% or more. Combustion laboratory tests indicated a 30–35% reduction in leakage. Combustor cloth seals are now standard for all new Frame 6F and 7F gas turbines. They are also offered as part of an extended life kit for the older units.

Nozzle–Shroud Intersegment Seals

Industrial gas turbines have many shroud and nozzle segments that require high-temperature sealing at the interfaces. Typically, sealing is accomplished by machining deep slots in the mating parts and inserting fairly stiff metal strips as seals (see Fig. 11). However, relative motion between the members can cause these seals to tip and toe, or jam against the slots. Lack of flexibility results in poor sealing and excessive wear. Various seals have been developed to address these compliance issues. For small gap changes, conventional sheet metal seals like c seals or w seals are adequate. Braided rope seals are used for demanding cases.[17–19]

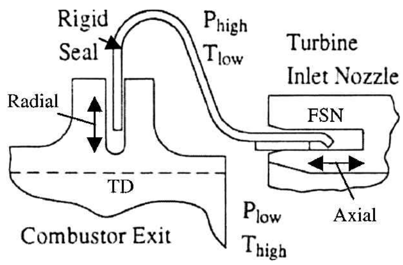

Fig. 9 Typical combustor seal assembly.

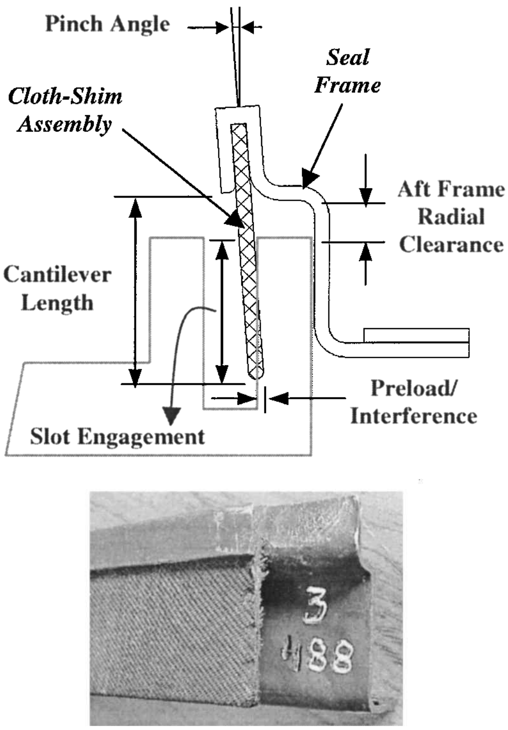

Fig. 10 Combustor cloth seal.

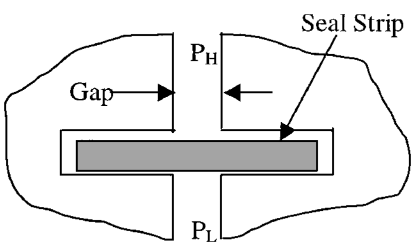

Fig. 11 Typical nozzle–shroud intersegment sealing arrangement.

For large interface gap changes, rigid metal strips were the main sealing method until the development of metal cloth seals. In large gap/misalignment applications, compliance cannot be achieved by reducing the thickness of the seal strips. The use of foil seals, like in aircraft engine applications, results in large stress levels and limited wear life. To solve seal compliance issues at such interfaces, early trials included the use of static brush seals.[5,20] Cloth seals were later developed as a better and less expensive alternative.[8–10]

Cloth seals are formed by combining thin sheet metals (shims) and layers of densely woven metal cloth. Although shims prevent through leakage and provide structural strength with flexibility, external cloth layers add sacrificial wear volume and seal thickness without adding much stiffness. As illustrated in Fig. 12, a typical design requires simply wrapping a layer of cloth around thin flexible shims. The assembly is held together by a number of spot welds along the seal centerline. Further leakage reduction can be achieved by a crimped design where double shims are used. The shims are

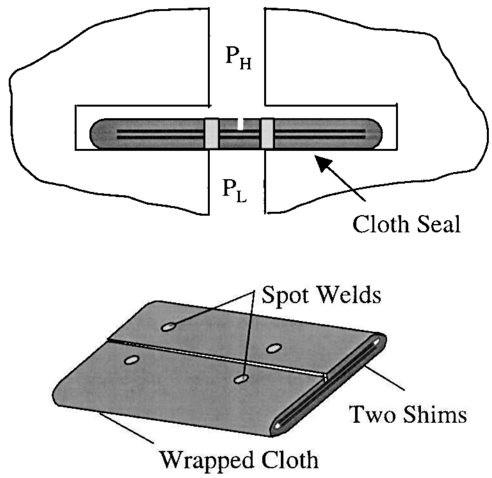

Fig. 12 Wrapped cloth seal design.

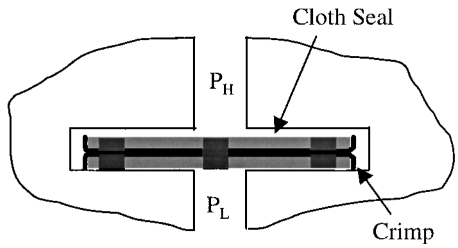

Fig. 13 Double shim crimped cloth seal design.

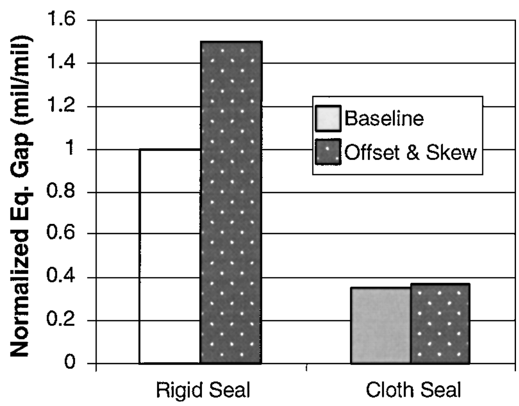

Fig. 14 Leakage performance of different nozzle–shroud intersegment seals (normalized by rigid seal baseline leakage).

crimped over the edge of the cloth layers (Fig. 13). By maintaining contact with the slot surface, crimped shims block the flow through the cloth layer. Therefore, sealing is achieved by restricting the flow at the crimps, rather than through the cloth material.

The flexibility introduced by cloth seals ensures a uniform slot contact over a range of relative excursions and provides reduced leakage rates. Back-to-back rig leakage tests demonstrated leakage reductions of up to 70% under various conditions (see Fig. 14).

The flow savings achieved by nozzle–shroud cloth seal applications translate into a 0.50% output increase and 0.25% heat rate reduction in 7F gas turbines. The flow savings have been verified through field tests of Frame 7E first-stage shrouds. Currently, nozzle and shroud cloth seals are standard for all new Frame 7F gas turbines. They are also offered as upgrades to the older E and F class units, Frames 3–9.

Conclusions

Implementing advanced static seals into industrial turbines has progressed well over the last few years. Applications have been in gas turbines, industrial and utility steam turbines, aeroengines, industrial compressors, and generators. System-level analyses have been performed to ensure that the seal leakage reductions do not cause overheating or purging problems. The analytical models have been employed to establish goals for leakage reduction without interfering with turbine operation and to define any turbine design modifications necessary to incorporate the sealing changes. Various cloth seal designs have been employed at junctions between stationary components (combustors, nozzles, shrouds, diaphragms, etc.) throughout the internal cooling and leakage flowpaths and steam-turbine packing ring segment end gaps. The sealing improvements have resulted in significant reductions in parasitic leakage flows, thereby increasing turbine efficiency and power output and, in some cases, reducing emissions. Advanced sealing has yielded a 0.3–1.0% increase in gas-turbine power output, and a 0.1–0.8% decrease in steam-turbine unit heat rate, depending on location.

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