Brush Seals for Improved Steam Turbine Performance

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SYNOPSIS

Brush seals, originally developed for aircraft engine and land-based gas turbine applications, have been adapted to steam turbines and have been retrofitted into more than 19 operating machines by GE Energy. Brush seals offer superior leakage control compared to labyrinth seals, owing to their compliant nature and ability to maintain very tight clearances to the rotating shaft. Brush-seal designs have been established for steam turbines ranging in size from 12 MW to over 1200 MW, including fossil, nuclear, combined-cycle and industrial applications. Designs for both high- and low-pressure turbine sections have been developed. Field inspections of seals after several years of service have demonstrated the robustness of brush seals in steam-turbine applications.

Steam turbines present unique design challenges that must be addressed to insure that the potential performance benefits of brush seals are realized. Brush seals can have important effects on the overall turbine system that must be taken into account to assure that the turbine will operate reliably. Subscale rig tests are instrumental to understanding seal behavior under simulated steam-turbine operating conditions, prior to installing brush seals in the field. This paper will discuss the technical challenges of designing brush seals for steam turbines, subscale testing, performance benefits of brush seals, overall system effects, and field applications.

1. INTRODUCTION

Improved seal performance offers substantial opportunities for turbine performance improvement as reduced leakages lead to greater efficiency and power output, and tighter control of turbine secondary flows. There are a number of seal locations in a steam turbine that offer significant performance gain. These include the interstage shaft packings, the end packings and the bucket-tip seals, as shown in Figure 1. This makes them ideal for the application of brush seals.

Brush seals have been used in aircraft engines since the early 1980's (Ref 1 and 2) and were introduced first into gas and then steam turbines in the mid 1990's (Ref 3). The continuing development work on steam-turbine brush seals leverages efforts on aircraft engine, industrial gas turbine and compressor brush seals.

The challenges associated with developing brush seals for steam-turbine applications include very high operating pressures and rotor-dynamic effects. The typical utility steam-turbine brush-seal configuration is shown in Figure 2.

2. PERFORMANCE BENEFITS

Steam leakage through the clearances between stationary parts and the rotor can account for as much as 29% of the total stage-efficiency loss, Figure 3, where 7 percentage points of loss are shown to be due to leakage between the rotating shaft and the diaphragm/fixed blading, and 22 percent is due to leakage between the moving buckets/blades and the stationary casing. Leakage at the shaft end packings further reduces overall turbine efficiency (Ref 4). Brush seals fill the clearances, reducing steam leakage to what can flow between bristles and underneath the bristle pack. When assembled with a clearance between the bristle tips and the rotor, the flow through the bristle pack tends to blow the bristles down towards the rotor. Brush seals can reduce leakage compared to conventional labyrinth seals by two thirds. Some

laboratory test results are shown in Figure 4. The effects of assembling the seal with interference and with clearance are shown. The leakage performance is compared to a representative labyrinth seal. Typical performance benefits for the reduced leakage of brush seals are shown in Table 1, where brush seals are assumed to be applied at most of the packing locations.

Brush seals are equally effective in reducing shaft-end packing leakage. Their application requires consideration of the entire shaft steam-sealing system as discussed in Section 5.4.

When the rotor dynamics of the system are considered, the number of brush seals that can be placed in the turbine without exciting a critical speed can be determined as is discussed in section 5.1. When selecting the locations to place the seals, the end packings and stages with the highest-pressure drops provide the greatest benefit from improved leakage control. Therefore, an offering of brush seals will typically include interstage shaft seals in the early stages of the high-pressure (HP) section of the machine and at the HP end packings. Depending on the rotor dynamics and other system parameters, the application of brush seals at the low-pressure (LP) end of the turbine is then considered.

3. BRUSH SEAL DESIGN

3.1 Pressure-Drop Capability

The brush seal design includes setting parameters such as the backing plate clearance to rotor, bristle clearance, bristle free length, bristle-pack density and bristle diameter. Decreasing the backing-plate clearance increases the pressure capability of the seal. However this clearance must be large enough to ensure that the rotor does not contact the backing plate during a radial excursion. A brush seal with inadequate pressure capability will fail via bristle deformation at the inner diameter of the backing plate; severe overpressurization will cause the bristle material to yield, permanently increasing the leakage flow. GE steam-turbine seals have been tested successfully at pressure drops in excess of 2.76 MPa (400 psid)* differential. Installed in series, these seals can be used for pressure drops in excess of 4.14 MPa (600 psid).

The remaining factors affect both pressure capability and bristle pack stiffness and contact pressure, both of which are discussed below.

3.2 Bristle Stability

A key consideration for brush-seal durability is the stability of the bristles of the brush seals. The geometry of the upstream cavity is key to ensuring the required flow field. Bristle instability can lead to the bristles fluttering and failing rapidly due to high cycle fatigue. A typical computational fluid dynamics (CFD) model, including computed velocity vectors, is shown in Figure 5.

3.3 Wear

Brush seals are contacting seals. Consequently bristle and rotor wear affect the seal's longevity. For most GE steam-turbine applications, the preferred bristle material is Haynes® 25, a cobalt superalloy. For the temperature conditions that are typical in HP and intermediate-pressure (IP) steam-turbine sections, this material has demonstrated minimal bristle and rotor wear against typical uncoated steam-turbine rotor materials in laboratory testing and in the field. However, at temperatures below 260°C (500°F), typical in LP turbine sections, Hastelloy® C-276 has demonstrated superior wear characteristics

to Haynes® 25. It is fortuitous that C-276 is practically devoid of cobalt, making it applicable to steam turbines in nuclear power plants employing boiling-water reactors (BWR). (If transported back to the reactor, cobalt-containing bristle fragments would be exposed to the neutron flux. The cobalt could be converted to Cobalt-60, a long-life high-energy gamma emitter.)

* lbf/in^2

4. SYSTEM CONSIDERATIONS

4.1 Rotor Dynamics

Rotor-dynamic considerations are important when introducing brush seals into steam turbines, owing to the integral-rotor construction of most modern machines. Contact between brush seals and the rotor leads to frictional heating. Any initial bow in the rotor will lead to a high spot on the circumference of the rotor; that spot will undergo the most frictional heating, and the differential heating around the circumference can lead to rotor bowing. Seals at the interstage locations would tend to excite the rotor's first bending mode, while seals at the end packing locations would tend to excite the rotor's second mode. As the typical steam turbine rotor operates between the first and second bending critical frequencies, the interstage seals would tend to affect the turbine's start-up, while the end packing seals would tend to affect stability at running speed.

The successful installation of shaft brush seals requires understanding the relationship between the rotor's stiffness and system dynamics, as well as the number of brush seals installed and their location and contact stiffness. A methodology has been developed to characterize the relationship between rotor stiffness and critical speeds, and brush seal contact force and bristle clearance. The effect of bristle clearance and seal design on a test rotor is shown in Figure 6. Where high-stiffness brush-seals are installed with interference or a line-on-line clearance, the rotor's response at the first and second critical speeds are significant. When the low-stiffness seals are installed with interference, the rotor's response at the second critical is acceptably low. Experience in the field supports these findings; seals with a higher than optimal stiffness or seals that are assembled with an initial interference can have a noticeable effect on rotor response. The effect is significantly reduced or eliminated either by the application of low contact stiffness seals or by assembling the seals with an initial clearance that is then allowed to "blow down" to very light contact when pressurized. Ideally, both methods are employed simultaneously. When brush-seals are installed with the proper design clearance, the rotor response is similar to that with conventional labyrinth packings.

4.2 Turbine Startup

By carefully considering rotor dynamics in the design and application of brush seals, the effect of the seals on turbine startup is negligible. The effects of brush seal-to-rotor contact are different at different rotor speeds; they are dependent on the rotor mode shapes, and the relationship between rotor critical speeds and hold speeds. These are important factors that must be carefully considered when selecting the optimum number and location of brush seals in a steam turbine. Contacting seals located near the rotor midspan will tend to influence rotor behavior at speeds below the first bending critical; seals near the rotor ends will tend to influence rotor vibration at speeds just below the second critical. Depending on the rotor design, this may include operating speed. When brush seals are designed and installed properly, the turbine can be started and operated normally with no special considerations.

4.3 Bucket-Tip Seal Considerations

Brush seals have usually been developed for smooth-rotor applications. On aircraft engines in particular, great care is taken with the surface treatment and finish of the rotor. Bucket-tip seals ride over a discontinuous surface. With integral-cover buckets, circumferential gaps range from 0 to 0.8 mm (0.030") and radial steps range from 0.08 mm (0.003") to 0.2 mm (0.008"). With conventional peened-cover buckets, gaps range from 0.8 mm (0.030") to 1.5 mm (0.060") and the steps can be as high as 0.4 mm (0.015"). Discontinuities have a significant effect on the bristle wear rate and the expected effective clearance. To cope with the surface discontinuities and the particularly turbulent-flow environment, a more robust seal design is used for tip applications. One effect in particular that must be considered is that of radial steam flow between adjacent bucket covers. The initial GE integral-cover buckets feature a contact area near the bucket throat where the gap is zero; however, portions near the cover leading edge and trailing edge had circumferential gaps of about ½ mm (0.02"). Application of brush seals on these bucket covers sets up a radial pressure gradient across the gaps that can induce slight radial inflow or outflow, tending to negate some of the performance advantage that the brush seal provides. Newer integral-cover bucket designs have extended the contact area over the entire axial length of the cover, preventing radial inflow or outflow and maximizing the efficiency benefit of bucket-tip brush-seals.

4.4 Secondary Leakage Flow

It is important to consider the impact of brush seals on the turbine as a system. Clearly the primary motivation for their use comes from the reduction in secondary flows and the corresponding improvement in performance. Reduced leakage affects the flow through the balance holes in the turbine wheels or bucket dovetails. Resizing the balance holes for the reduced leakage flow of the brush seals will minimize the possible intrusion losses from mismatched balance holes. Changing the secondary flows will have an effect on machine thrust. While this is typically not a large effect, it must be evaluated when brush seals are introduced on a retrofit basis.

When end-packing seals are installed, care must be taken to insure that the steam-seal system performance is considered. Reducing leakage at the HP and/or IP rotor ends should lead to a performance benefit. However a minimum leakage flow is required to seal the LP ends to insure that the unit is self-sealing, without using high-pressure steam to make up the difference. This is accomplished either by limiting the number and locations where HP end seals are installed, or by installing brush-seals at the LP section ends as well, to reduce the demand for leakage flow for sealing.

5. LABORATORY TESTING

In developing brush seals for steam turbine applications, laboratory testing plays an important role. GE has a dedicated seals test facility with rigs for testing under conditions that approximate those found in a steam-turbine environment. Component testing is done both at GE's Global Research Center (GRC) in Niskayuna, NY, as well as at Cross Manufacturing's facility in Devizes, England. For the development of steam-turbine seals in particular, initial prototype testing to evaluate leakage and seal stability was performed at Cross. To evaluate seal pressure capability and leakage in a simulated high-pressure environment of a steam turbine, and to investigate the effects of brush seals on rotor dynamics, validation tests were conducted in the high-pressure rig at GE. Further tests to examine the pressure distribution through the bristle pack and to simulate running over bucket tips has been carried out at Cross.

5.1 Seal Performance Testing

Brush seals are flexible by nature, and their leakage behavior is a strong function of the pressures to which they are exposed. In evaluating brush seal leakage as compared to conventional labyrinth-type seals, many subscale test rigs have been employed. Cross sections of three of these are shown in Figures 7 through 9. The static rig shown in Figure 7 was developed by Cross to evaluate the pressure distribution through the bristle pack over a range of pressure drops and pressure ratios. The back-pressure capability of this rig enables high pressure drops combined with low pressure ratios to be investigated. The pressure distribution is measured via a 0.05 mm (0.002") wide slot in the disk as it is traversed axially through the pack. Interesting to note is that much like a labyrinth seal, the distribution of pressure through a brush seal is strongly influenced by the pressure ratio across it, with larger pressure ratios resulting in a greater percentage of the total pressure drop being taken by the last few bristle rows. The implication of this is that the pressure-drop capability of a brush seal is much greater for interstage seals (where the pressure ratio is generally much less than 2), than for end packings where the pressure ratio can be quite high (greater than 2, and often exceeding 4).

The rig shown in Figure 8 is the ambient dynamic rig at Cross. This extremely flexible facility has been used for many different tests since it was commissioned in 1987. For steam-turbine applications in particular, it has been used for many test programs including the following:

a) Evaluating tribo-pairs at representative surface speeds and pressure drops.

b) Pressure capability testing

c) Swirl stability evaluation and visualization using a video bore scope with stroboscopic light source at swirl ratios ranging from 0.3 to 1.2.

d) Investigation into the effects of running over simulated bucket tips with discreet gaps and radially inward or outward flow depending on the axial position of the seal on the bucket tip.

The rig shown in Figure 9 is the GRC sub-scale rig. It is capable of testing seal leakage and durability at pressures up to 8.3 MPa (1200 psi) and at temperatures up to 540°C (1000°F). The rotor is a solid Chromium-Molybdenum-Vanadium (CrMoV) shaft that can be run at any speed above or below its second critical speed, with a maximum rotor surface speed of 245 m/s (800 ft/s). By controlling the seal upstream and downstream pressures independently, mass flow can be quantified over the entire expected range of pressure differentials. These data are then plotted as seal "effective clearance" as a function of pressure differential, and allows different seal types to be compared as shown in Figure 4.

5.2 Rotor Dynamics Testing

In addition to the performance data collection described above, the GRC test rig is outfitted with a vibration monitoring system that allows rotor vibration to be measured throughout the duration of a test, at speeds simulating turbine startup, steady-state operation, and coastdown. By studying both the seal's leakage performance and effects on rotor dynamics, the best seal configuration for steam-turbine shaft applications can be established. In addition to these tests, analytical tools have been developed to predict the frictional heat-generation of a brush seal running in contact with a turbine rotor. These are based on heat generation tests in which a thermal-imaging camera was used to measure the rotor temperature at the seal/rotor interface for a range of rotor speeds and seal interferences.

5.3 Wear Testing

In many aircraft-engine applications, the rotor surface immediately under the brush seal is coated with a wear-resistant material such as chrome carbide. Brush-seal wear tests conducted on uncoated CrMoV rotors have demonstrated that for the bristle materials, rotor surface velocities, expected radial interferences, and temperatures that are typical of steam turbines, rotor coating is unnecessary. Furthermore, laboratory wear tests conducted using various bristle materials rubbing against an uncoated CrMoV surface have been conducted to find the bristle material that demonstrates minimal wear, while also providing weldability to the required backplate material and minimal susceptibility to stress-corrosion cracking and hydrogen embrittlement. As discussed in Section 4.3, the bristle material selected for steam-turbine brush seals is temperature-dependent, with Haynes® 25 used for temperatures above 260°C (500°F), and Hastelloy® C-276 used for temperatures below 260°C (500°F) as well as in all nuclear steam turbines employing BWR's.

Field experience to date has correlated very well with the laboratory testing. After three years of operation in one unit, rotor wear was on the order of 25 μm (0.001") and bristle wear was minimal. The bristles were still contacting the rotor surface when pressurized and the seals were performing as expected.

5.4 Steam-Turbine Test Vehicle

An important tool for validating new steam turbine component performance is the Steam-Turbine Test Vehicle (STTV) (Figure 10) developed by GE and located in Lynn, Massachusetts. This is a 3.5 MW boiler feed-pump turbine that has been modified to accurately model the thermodynamic characteristics of a four-admission large steam turbine (Ref 5). Back-to-back tests have been conducted in this turbine in which brush seals at the six interstage locations were shown to improve unit heat rate by 1.0%. This result is consistent with the estimated performance gain based on expected brush-seal effective operating clearances. These tests demonstrate that brush seals provide an appreciable improvement in turbine performance, and that the method used to predict the performance improvement is sound.

6. FIELD EXPERIENCE

GE has been applying brush seals to commercial turbomachinery for over a decade. Early development work focused on aircraft engines and gas-turbines. Later applications were expanded to include industrial and power-generation steam turbines. From this diverse experience, GE has developed design tools and features to improve the unit efficiency and brush-seal life.

The GE gas-turbine fleet has several hundred brush seals in service, the first of which passed their 24000-hour hot-gas-path inspection and were returned to service. The locations include the high-pressure packing, the middle bearing (of three-bearing machines) and the turbine interstage.

There are currently nineteen GE steam turbines, both nuclear and fossil, operating with brush-seals. These units range in output from 12 MW mechanical-drive turbines to power-generation nuclear steam turbines rated 1200 MW. Brush seals are employed at interstage, end packing, and bucket-tip seal locations in HP, IP and LP turbine sections in this population. Steam conditions vary from over 5.5 MPa (800 psi) and 480°C (900°F), to sub-atmospheric pressures and less than 40°C (100°F). An additional 23 units employing brush seals are in various stages of design, manufacture and installation. There are over 130 steam-turbine brush seals installed and approximately the same number on order.

Seven of these steam turbines have been inspected over the last few years. The contained seals are of the first two generations of brush-seal design. These generations varied in bristle pack stiffness. The discoveries made during the inspections were generally well predicted by the design tools. The minor issues found with these designs led to improvements, and to the third generation of brush-seal design. The experience gained during start-ups contributed to achieving a better understanding of the effect of brush seals on rotor dynamics.

The first steam-turbine brush seal was installed in 1996 in the outer shaft packing seal location of a small industrial turbine. The main objective of the test was to prove that brush seals could survive in steam-turbine conditions. The unit was inspected after three years of service and the brush-seals were found to be in excellent condition.

Another unit was inspected during a boiler outage in November 2000. This is a 250 MW utility unit in which the seals were installed during a maintenance outage in June 1999. Six brush seals were installed in various locations of this opposed-flow HP-IP section, as shown in Figure 11. The locations include the diaphragm packing of stage 3, the end packing at the HP exhaust, and the bucket tips on the first two stages of the IP section.

The brush seal installed at the eighth-stage bucket-tip location was intentionally placed in an exceptionally severe operating environment, situated immediately downstream of the tenons of peened-on bucket covers. In addition, this stage employs a notch block taking the place of one bucket in the row. A severe once-per-revolution transient is created in the pressure field immediately beneath the bristle pack, causing the bristles to fail.

The shaft brush seals were in excellent condition after a year and a half of service. Brush seal and rotor wear were at the low levels expected. There was no excessive wear on rotor or brush, and no evidence of bristle failure or damage to the side-plates or packing rings. Some packing teeth showed evidence of light to moderate packing rubs unrelated to the brush seals. The stage-nine tip brush seal, which rides over a row of integral cover buckets, is shown in Figure 12. The segment-end design has since been improved to eliminate catching bristles between segments. The brush seals in this turbine were designed for low contact pressure to mitigate the effect of the seals on rotor dynamics. Start up in 1999 was very smooth, and there have been no rotor-dynamics issues with the unit.

A 700 MW unit was inspected after almost three years of brush-seal service. Ten brush seals were installed in several interstage shaft locations in the HP section of the turbine. The locations include the diaphragm packings of stages 3 through 7. The seals were installed during a major rebuild in April 1998, and inspected in February 2001. All of the brush seals were found to be in excellent condition. Rotor polishing showed that bristles remained in contact with the rotor forming an effective seal. The brush seals of the stage 5 and 6 shaft packings are shown in Figures 13 and 14.

In 2002, four more units of similar characteristics were inspected. These are super-critical units rated at 900 MW. In each machine, eight brush seals were installed at different locations in the HP section. The locations include the diaphragm packings of stages 3 through 6, and the bucket-tip of stage 6. These seals were installed in November and December 1999, and inspected in September 2002. The shaft brush seals were in good condition after almost 3 years of service. Figure 15 shows a close-up view of the stage 5 shaft brush seal.

The application of brush seals has expanded from replacement seals to standard offering in new steam turbines. The positive results from various tests and from field experience confirm the performance benefits of the use of brush seals in steam turbines.

Brush seals have recently been retrofitted to 4 GE nuclear steam turbines. They started up in 2004 and 2005 with no vibration problems encountered. The suitability of the brush-seal designs and application rules has been validated by successful field experience.

Brush seals are employed in the new HEAT™ 107/109 combined-cycle steam-turbines as a standard new-unit offering (Ref 6 & 7), as shown in Figure 16. The first generation of HEAT™ 107 has four brush seals in the end packings. The HEAT™ 109 has six brush seals in the end packings and two brush seals at the bucket-tips. The first of this line of turbines is scheduled to startup in April 2005.

7. BRUSH-SEAL QUALITY

In order to achieve optimum performance, and repeatable performance from one batch of brush seals to the next, it is important to have certain features tightly controlled. The seals are designed to provide reliable turbine operation with no impact on operability, and a significant reduction in leakage over a 10-year life; to achieve this, the following parameters must be controlled: Seal mating part fit, Bristle Stiffness, Number of bristles, Bristle inner diameter (i/d), and Backing Plate i/d. Bristle stiffness is controlled by the diameter at which the bristles are restrained, the angle of lay, and the bristle wire diameter. It is important that the bristle tips are free to move, and that they are not hooked or fused together.

It is also important to have an integral structure between the bristles and the back plate interface. At Cross Manufacturing these features are all measured and values are recorded. The solid metal features such as the mating fit dimensions and backing plate bores are easily assessed with traditional measuring techniques such as micrometers and co-ordinate measuring machines. However any measurements relating to the bristles must be performed with a non-contacting technique, as the very nature of the bristle is a flexible member. The bristle i/d and bristle angle are assessed with optical techniques including the use of an optical co-ordinate measuring machine. The bristle angle needs to be assessed as a finished seal. One cannot rely on the process capability of the initial manufacturing lay angle, as the angle changes during the manufacture of the seal. To insure the bristle tips are free, a wire electro-discharge machining technique is used, where the settings are tightly controlled. The number of bristles within the seal is numerically controlled. Structural integrity is assured by matching suitable backing plate materials to the bristle material, and using a tightly controlled semi-automatic welding process. Once the optimum weld settings have been established, with the aid of microscopic examination and X-rays, these weld conditions are frozen. As these seals are usually supplied as segments it also easy to assess the weld depth and quality on each end of every segment as a routine quality-control procedure. All the seals are given a post-weld heat treatment to ensure no embrittlement in the weld/ bristle interface area, with the added benefit that the seal bristle pack is also supplied uniform and not “bushy or open”, thus improving seal performance. Figures 17 & 18 show a typical production-quality brush-seal bristle pack.

8. CONCLUSIONS

In this paper, the authors have described the application of brush seals to General Electric steam turbines, considering the benefits in performance and the design issues that must be considered. At the present writing there are nineteen steam turbines in service with various combinations of interstage-packing, end-packing and bucket-tip brush seals. These include both industrial and large-utility steam turbines rated 12 MW to 1200 MW. An additional 23 units employing brush seals are in various stages of design, manufacture and installation. Ongoing programs to develop buckets with tight-fitting, gapless integral covers will permit increased application of brush seals to bucket-tip seals. The brush seals employed have

been a joint development of GE and Cross Manufacturing, evolving from Cross designs for combustion turbines. The quality-assurance procedures followed in their manufacture have been described. While bucket-tip seals continue under active development, steam-turbine shaft brush seals are now a robust product offering, with validated leakage reduction and demonstrated reliability. Development efforts continue both to refine the current designs and to expand the range of possible applications.

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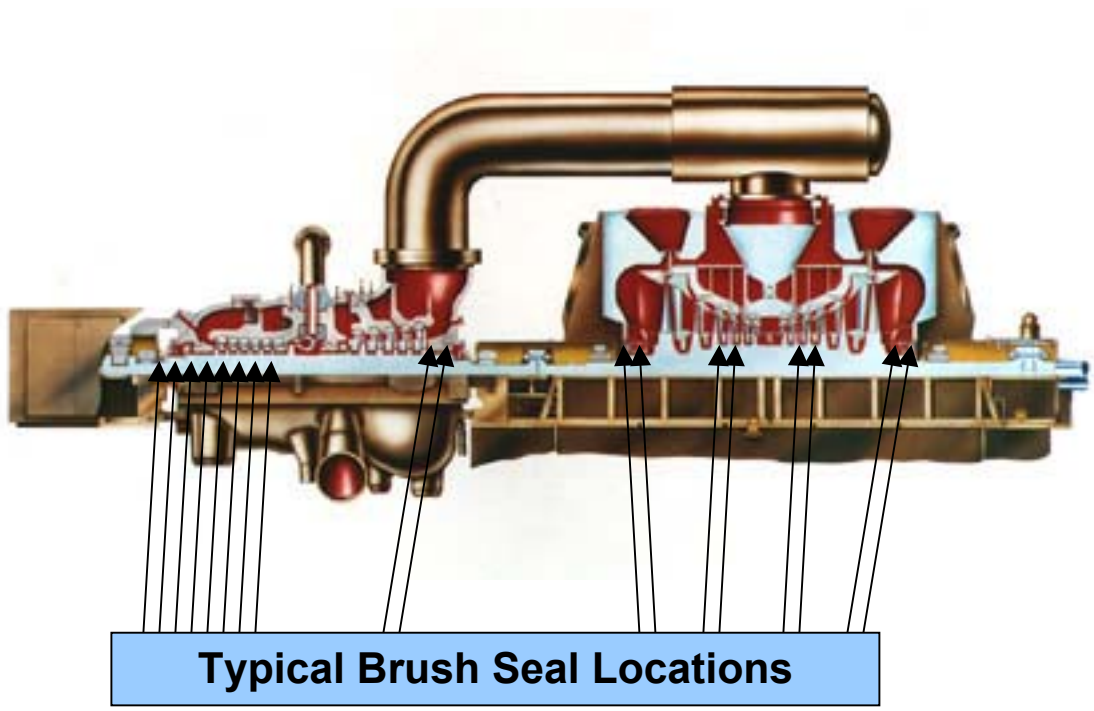


Figure 1. Typical Brush-Seal Locations in a Utility Steam Turbine (© General Electric Company, 2005)

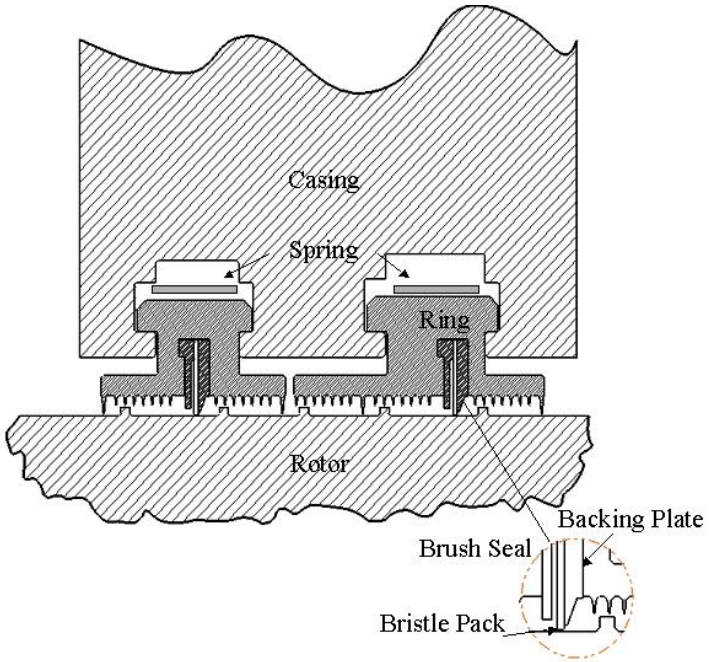


Figure 2. Typical Utility Shaft-Packing Brush-Seal Configuration (© General Electric Company, 2005)

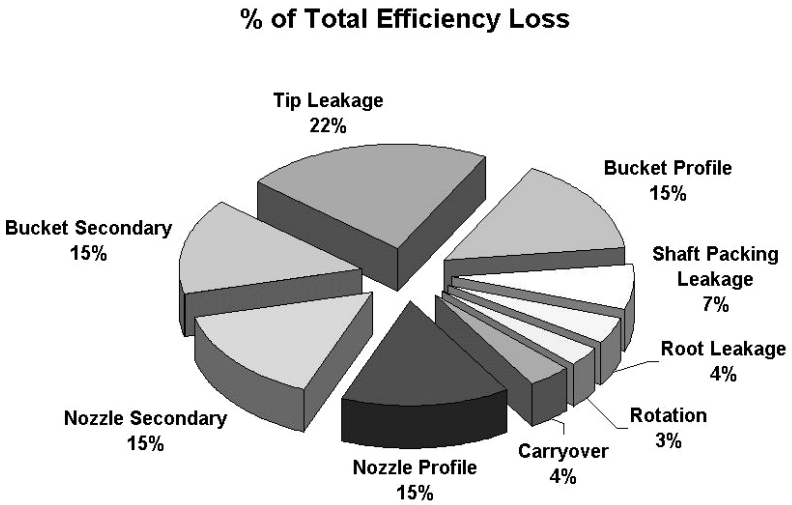
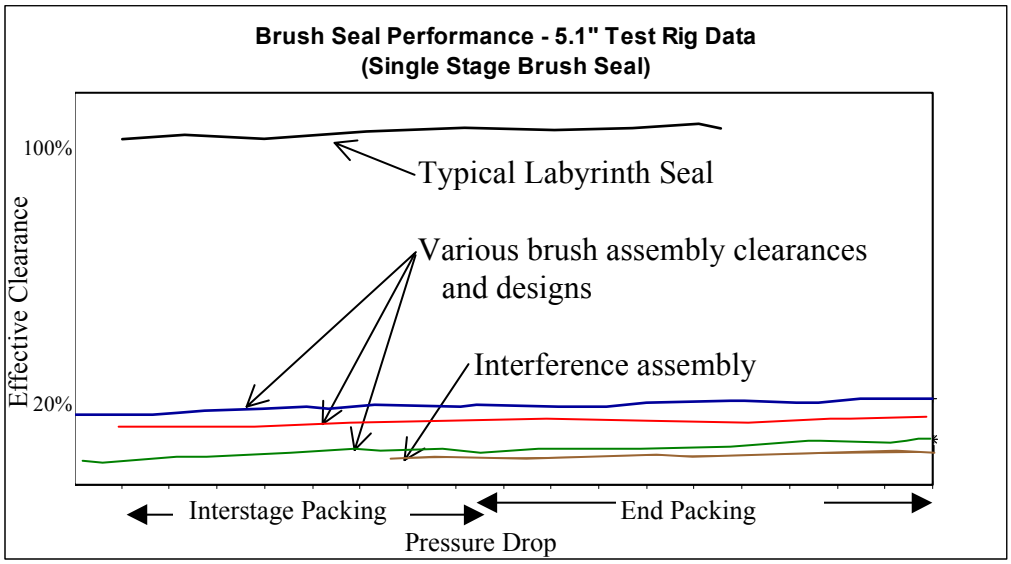


Figure 3. Typical Turbine-Stage Efficiency Losses (© General Electric Company, 2005)



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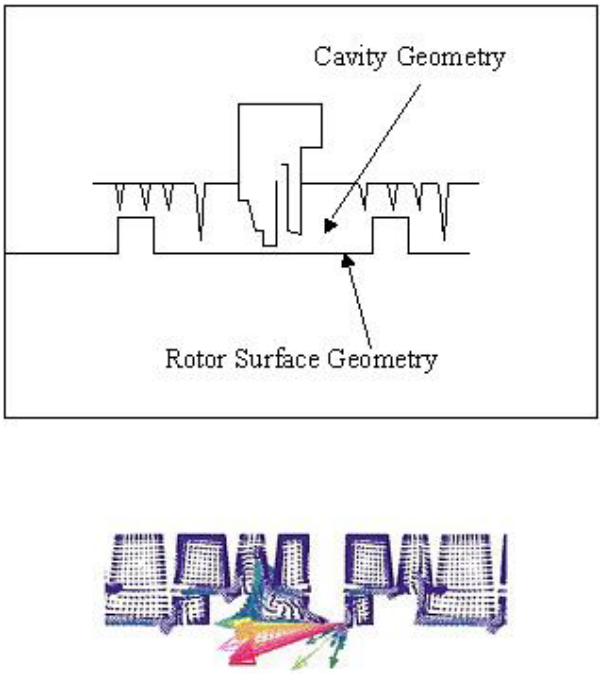

Figure 5. CFD Model of Brush-Seal Showing Computed Velocity Vectors (© General Electric Company, 2001)

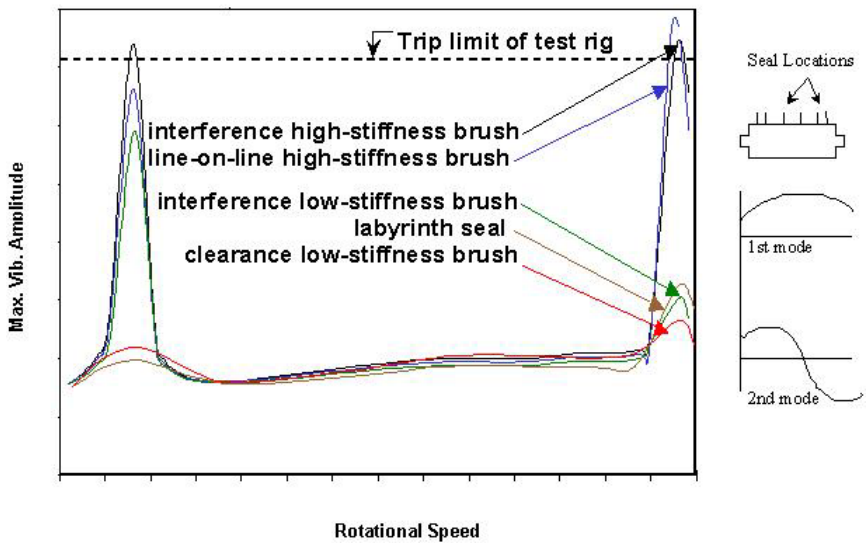

Figure 6. Test-Rig Rotor Vibration Reponse Versus Speed for Various Seal Configurations (© General Electric Company, 2001)

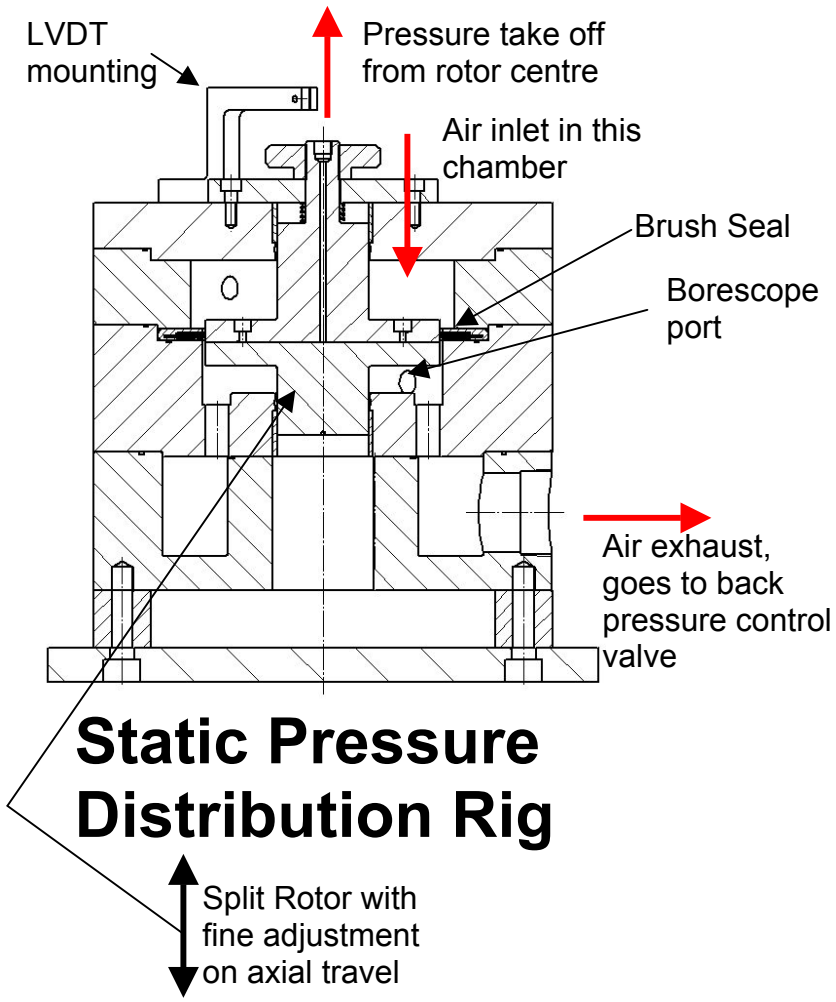

Figure 7. Static Seal Test Rig

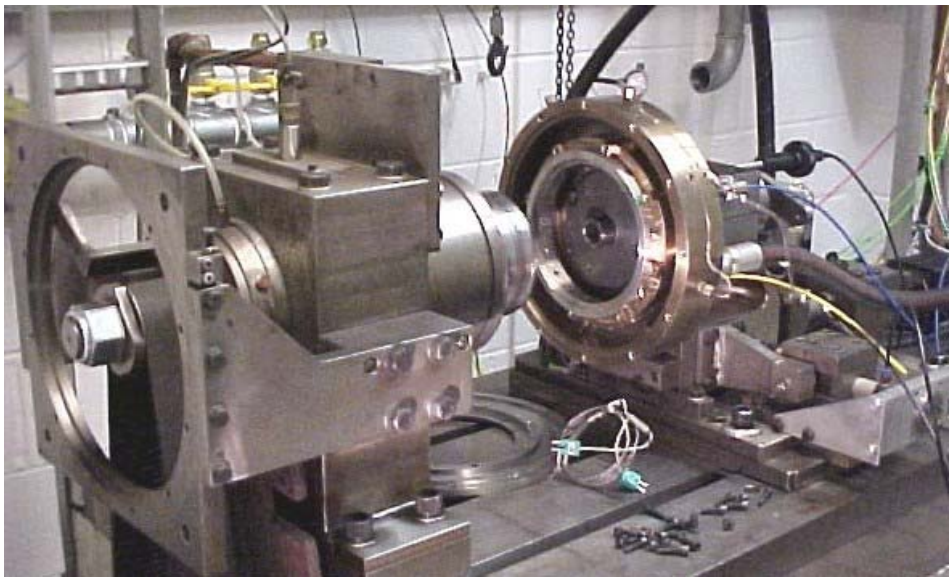

Figure 8. Ambient Dynamic Seal Test Rig

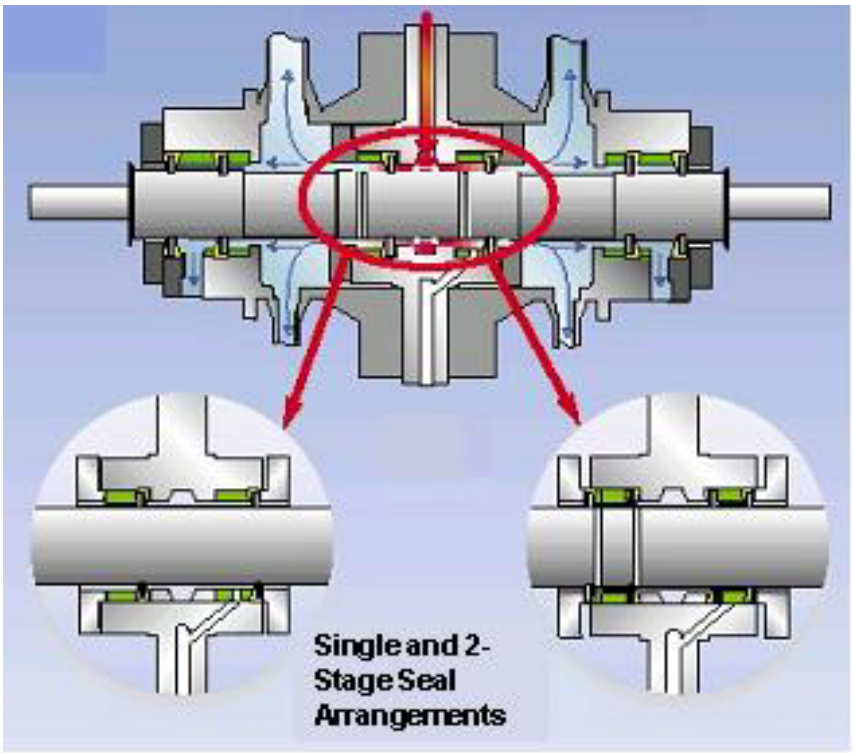

Figure 9. GRC Sub-Scale Seal Test Rig (© General Electric Company, 2005)

Figure 10. Steam-Turbine Test Vehicle in Lynn, Massachusetts (© General Electric Company, 2001)

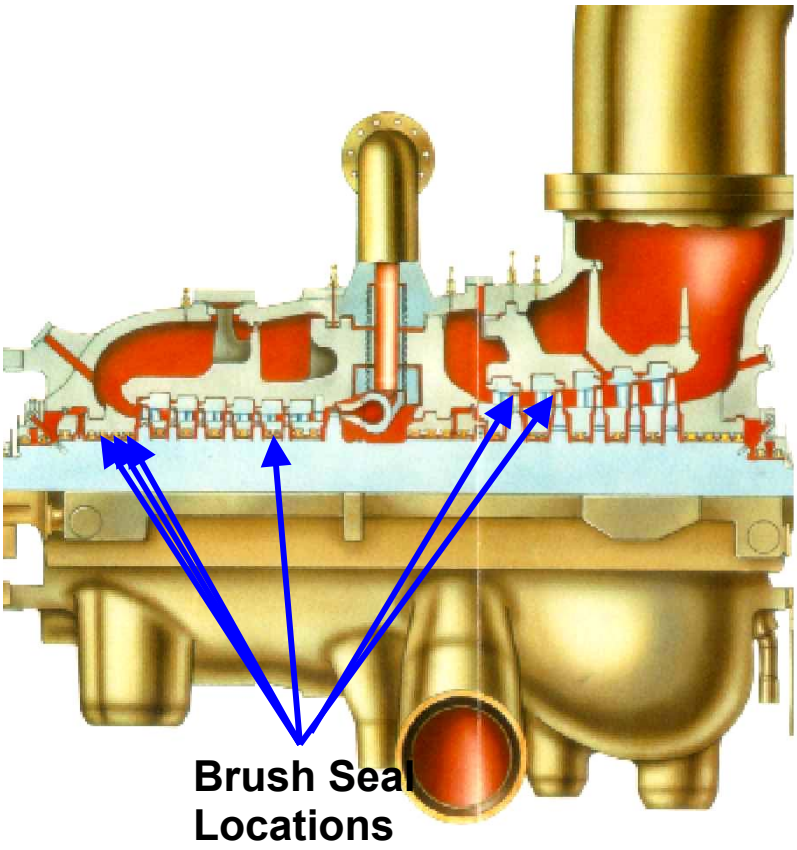

Figure 11. Brush-Seal Locations in Opposed-Flow High-Pressure-Intermediate-Pressure Turbine (© General Electric Company, 2001)

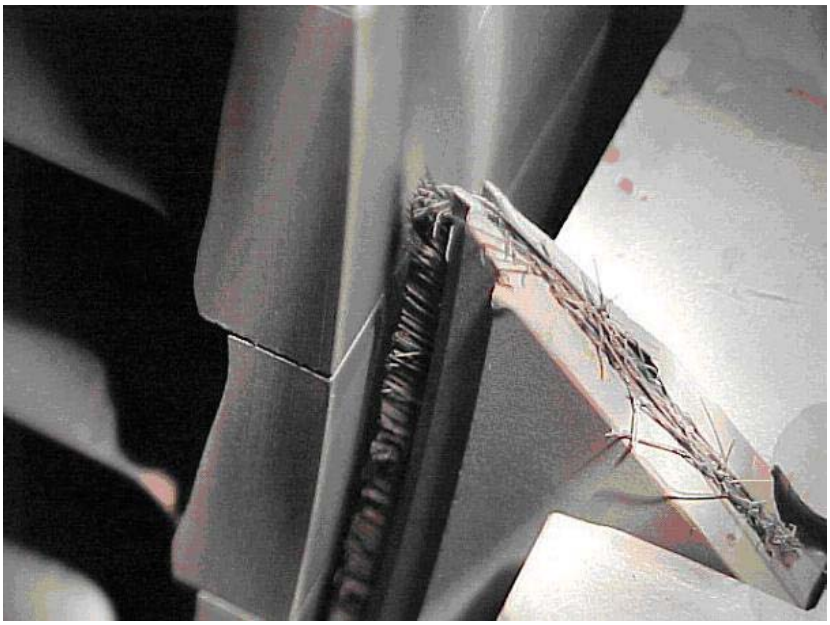

Figure 12. Horizontal-Joint View of Brush Seals at Bucket Tips (© General Electric Company, 2001)

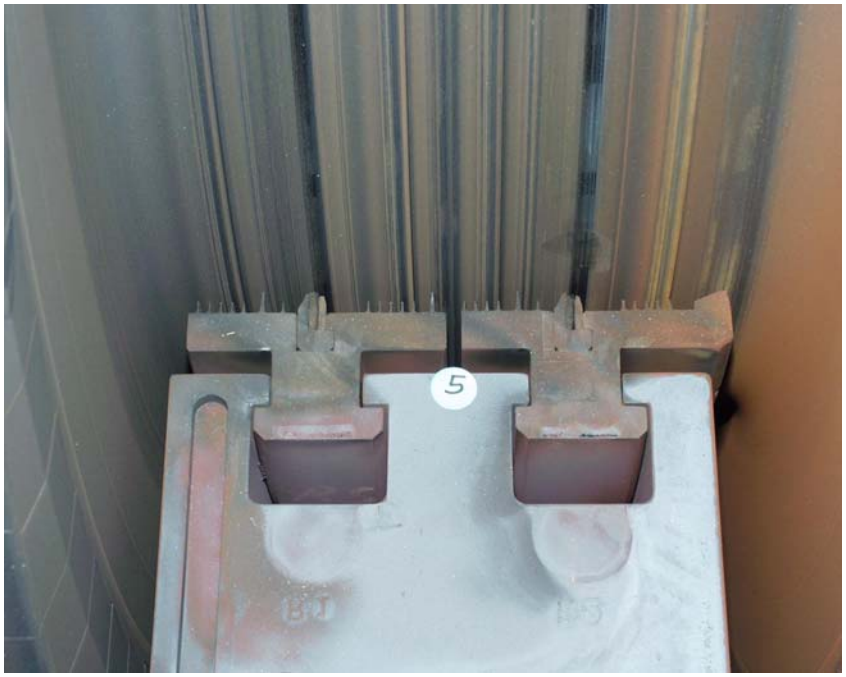

Figure 13. 700 MW – Diaphragm Stage 5 Brush Seal (© General Electric Company, 2001)

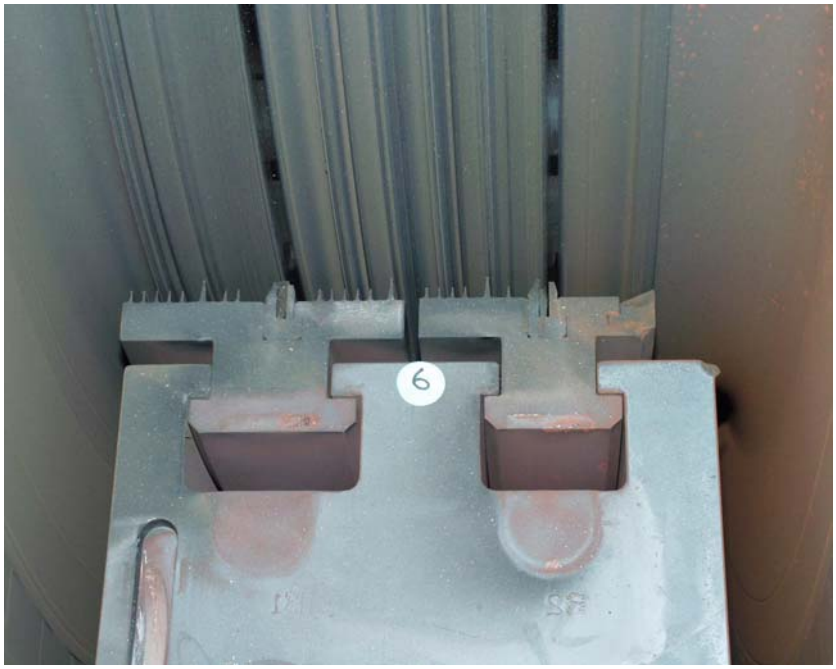

Figure 14. 700 MW – Diaphragm Stage 6 Brush Seal (© General Electric Company, 2001)

Figure 15. 900 MW – Diaphragm Stage 5 Brush Seal (© General Electric Company, 2005)

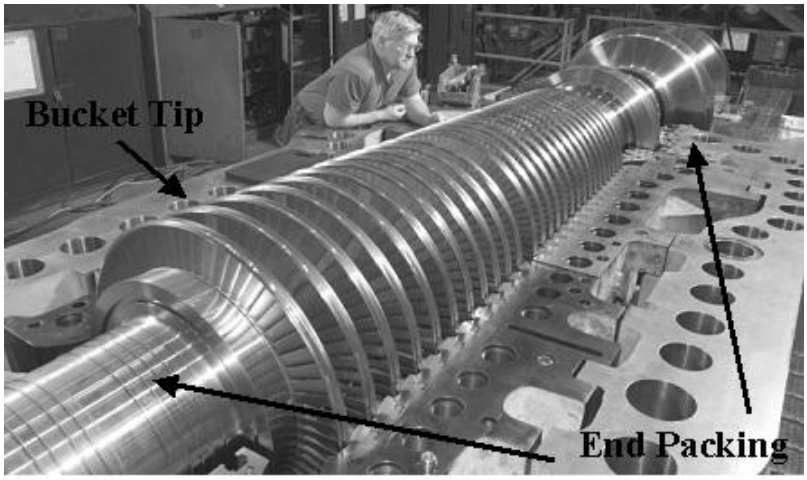

Figure 16. Heat™ HP Section Showing Brush Seal Locations (© General Electric Company, 2005)

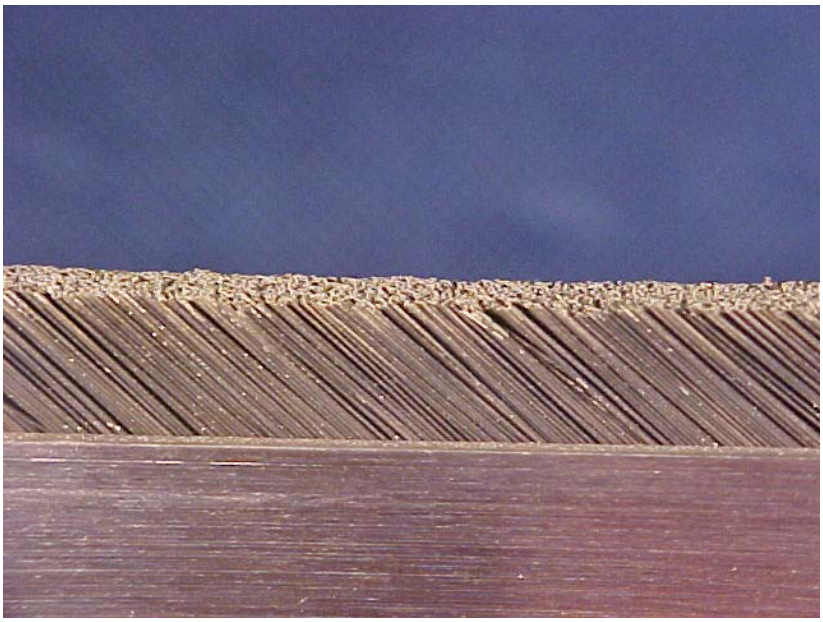

Figure 17. Typical Axial View of Bristle Pack

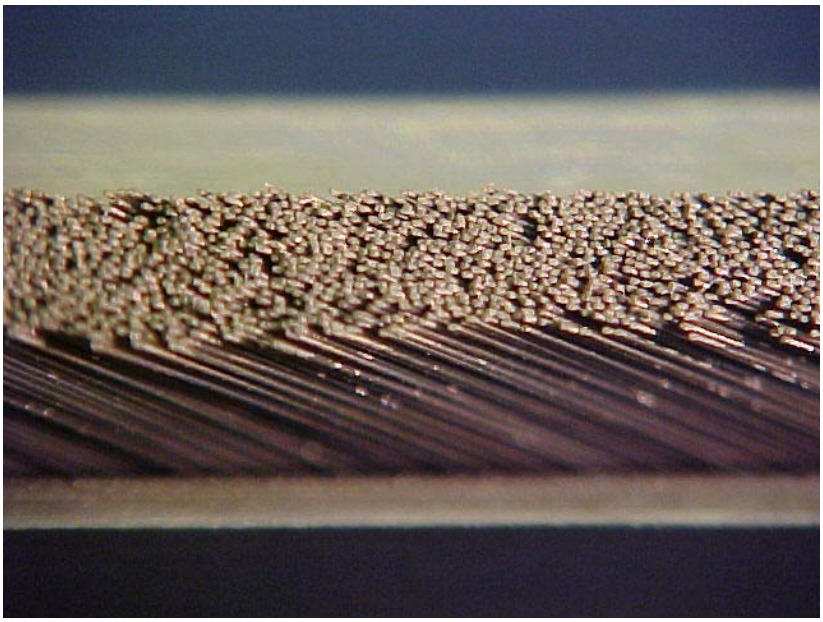

Figure 18. Typical Radial View of Bristle Pack

List of Tables

Table 1 Typical Performance Benefits of Brush Seals

	Utility ST	*Industrial ST*
Interstage	0.5-1.2% HP section efficiency; 0.1-0.2% unit heat rate	0.2-0.4% efficiency
End Packing	0.1-0.2% unit heat rate	0.4-0.8% efficiency
Bucket Tip	0.5-1.0% HP section efficiency; 0.1-0.2% unit heat rate	0.7-1.1% efficiency

Table 1 Typical Performance Benefits of Brush Seals (© General Electric Company, 2001)