IMPROVED STEAM TURBINE LEAKAGE CONTROL WITH A BRUSH SEAL DESIGN

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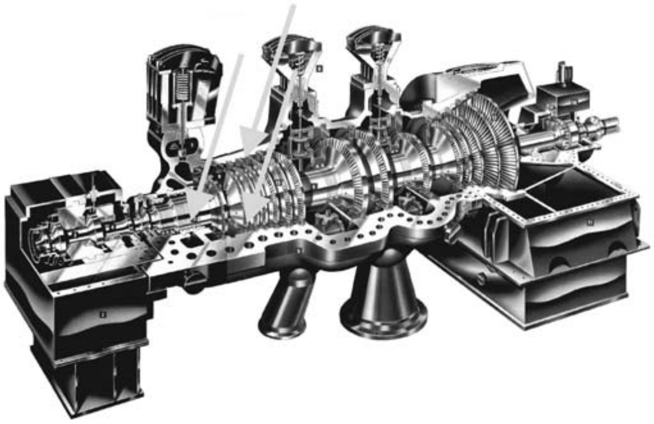
ABSTRACT

Brush seal designs offer improved leakage control over conventional labyrinth seals by applying a compliant bristle with very tight clearance over the rotating shaft. Originally developed for the aircraft engine industry, brush seals have also been applied in land-based gas turbines. More recently, brush seals have been applied in steam turbines. The steam turbine environment adds some unique design considerations that must be addressed to assure a robust and effective brush seal design, and to minimize the impact on the steam turbine as a system. This paper discusses the performance benefits of the brush seal and the design considerations important to a robust design in a steam turbine. The paper also addresses the system characteristics important to

reliable operation, and discusses the current experience basis from one original equipment manufacturer.

INTRODUCTION

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



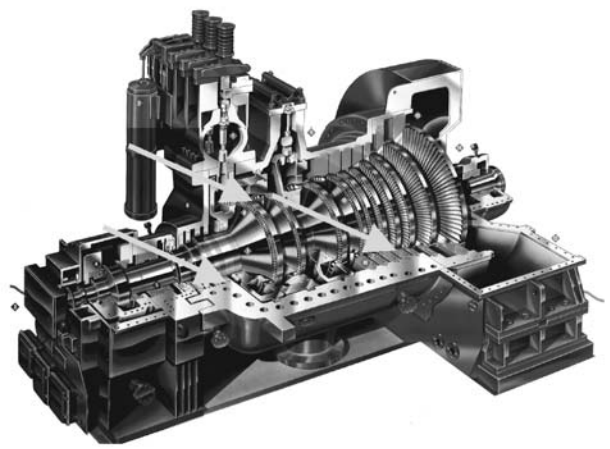


Figure 1. Typical Brush Seal Locations in Industrial Steam Turbines.

Brush seals have been used in aircraft engines since the early 1980s (Flower, 1990; Mahler and Boyes, 1995) and were introduced first into gas and then steam turbines in the mid 1990s (Wolfe, et al., 1997). The continuing development work on steam turbine brush seals leverages effort 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, rotordynamics, and steam chemistry. Typical steam turbine brush seal configurations are shown in Figure 2. These include a brush seal in a utility steam turbine packing ring, and brush seal insert in an industrial steam turbine packing ring.

PERFORMANCE BENEFITS

Steam leakage at the gaps between stationary parts and the rotor can account for as much as 29 percent of the total stage efficiency

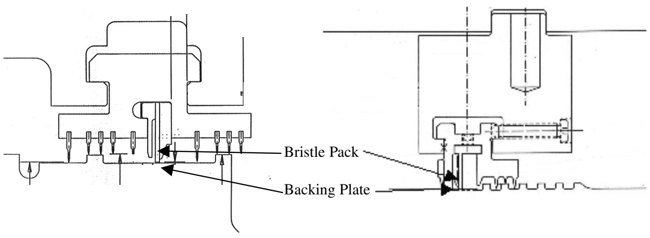


Figure 2. Typical Utility and Industrial Shaft Packing Brush Seal Configurations.

loss (Figure 3), and leakage at the endpackings further reduces overall turbine efficiency (Cofer, et al., 1996). Brush seals fill these gaps, reducing steam leakage to what can flow between bristles and underneath the bristle pack. When assembled with a gap, the flow through the bristle pack tends to blow the bristles down toward the rotor. Brush seals reduce leakage compared to conventional labyrinth seals by some 70 percent, as shown in Figure 4. In the figure, the effects of assembling the seal with interference and with clearance are shown; the leakage performance is compared to a typical labyrinth seal. Typical performance benefits for the reduced leakage of brush seals are shown in Table 1.

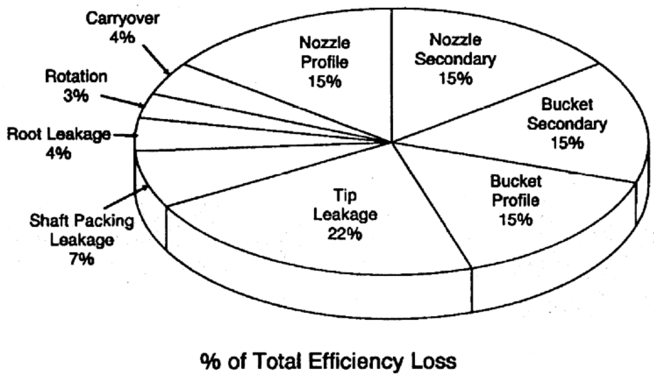


Figure 3. Sources of Steam Turbine Efficiency Loss.

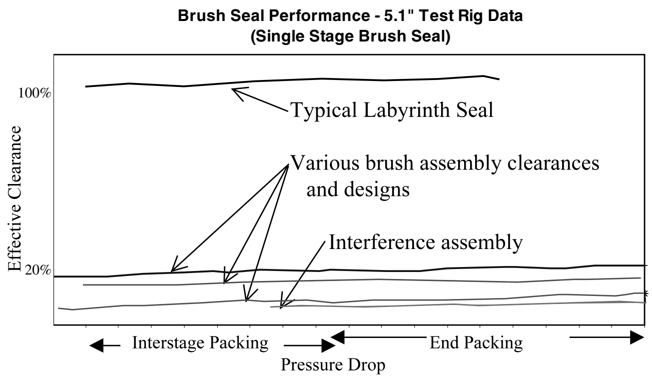


Figure 4. Leakage Rates Versus Pressure for Various Brush Seal Assembly Clearances, Compared to a Typical Labyrinth Seal.

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

When the rotordynamics 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; this will be discussed later in this paper. When selecting the locations to place the seals, the endpackings and stages with the highest pressure drops provide the greatest benefit to leakage control. Therefore an offering for brush seal replacement will typically include interstage shaft seals toward the high-pressure (HP) section of the machine and at the HP endpackings. Depending on the rotordynamics and other system parameters, application of the brush seals at the low-pressure (LP) end of the turbine is then considered.

BRUSH SEAL DESIGN

Pressure Drop Capability

The brush seal design includes setting parameters such as the backing plate clearance to rotor, the bristle clearance, the bristle free length, the bristle-pack density, and bristle diameter. Decreasing the backing plate clearance increases the pressure capability of the seal; however, the clearance must be set to ensure that it does not rub the rotor. A brush seal with inadequate pressure capability will fail by having the bristles deformed at the inner diameter of the backing plate, permanently increasing the leakage flow. One manufacturer's steam turbine seals have been tested successfully at pressure drops in excess of 400 psid. Installed in series, these seals can be used for pressure drops in excess of 600 psid.

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

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 model is shown in Figure 5.

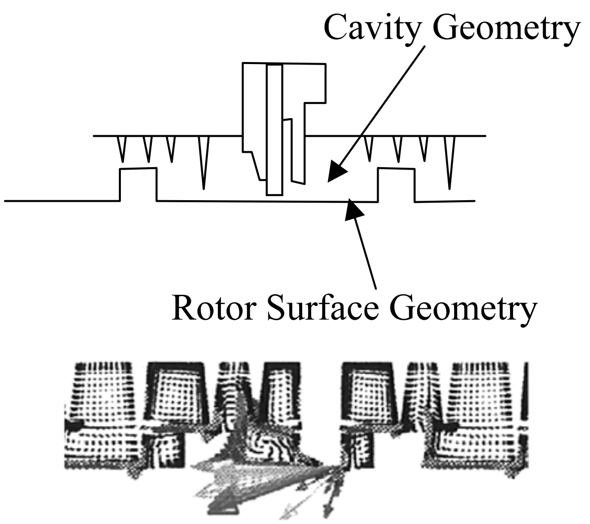


Figure 5. CFD Model of Brush Seal with Velocity Vectors.

Wear

Brush seals are contacting seals. Consequently bristle and rotor wear affect the seal's longevity. Extensive testing under a variety of conditions has led to the use of Haynes® 25, a cobalt superalloy, as the bristle material on the uncoated rotor.

SYSTEM CONSIDERATIONS

Rotordynamics

Rotordynamic 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 rotor leads to frictional heating. Any initial bow in the rotor will lead to a high spot in the circumference of the rotor; that spot will see the most frictional heat, and the differential heating around the circumference can lead to rotor bow. 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 frequency, the interstage seals would tend to affect the turbine's startup, 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 the number of brush seals installed and their location and contact stiffness. A transfer function 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 conventional brush seals are installed with interference or a line-on-line clearance, the rotor's response at the first and second critical speeds is significant. When the low contact 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 the brush seal is installed with the proper design clearance, the rotor response is similar to that with a labyrinth packing.

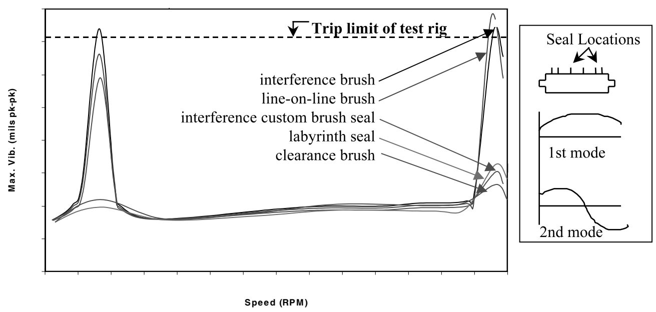


Figure 6. Test Rig Rotor Response Versus Speed for Various Seal Configurations.

Turbine Startup

By carefully considering rotordynamics in the design and application of brush seals, the impact of the seals on turbine startup is negligible. As mentioned above, 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.

Tip Versus Shaft Seals

Brush seals have traditionally been developed for smooth rotor applications. On aircraft engine applications in particular, great care is taken with the surface treatment and surface finish of the rotor. Bucket tip seals ride over a discontinuous surface. With integral cover buckets, circumferential gaps range from 0.020 to 0.030 inch and radial steps from 0.003 to 0.008 inch; with conventional peened cover buckets, gaps range from 0.030 to 0.060 inch and the steps can be as high as 0.015 inch. This has a significant effect on the wear rate and the effective clearance to be expected. A more robust seal design is used for tip applications.

Secondary Leakage Flow

It is important to consider the impact of brush seals on the turbine as a system. Clearly the primary motivation comes from the consideration of the reduction in secondary flows and the corresponding impact on performance. The reduced leakage impacts 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 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 ensure that the steam seal system performance is considered. Reducing leakage at the HP ends will lead to a performance benefit. However a minimum leakage flow is required to seal the LP end to ensure that the unit is self-sealing. This is accomplished either by limiting the number and locations where HP end seals are installed or by installing the brush seals at the LP end as well to reduce the demand for leakage flow.

Steam Deposits

Steam chemistry is a concern with some steam turbine owners who felt that deposits in their steampath would possibly hinder the brush seal effectiveness and/or adversely influence the rotordynamic characteristics of the seals. Boiler pressure is the driving force for determining when steam deposits will occur. When pressures are high enough, particulates vaporize in the boiler and mix with the steam. As the steam and vaporized minerals go through the steampath and pressure is gradually reduced, the minerals precipitate out of the steam, and can deposit on the turbine internals. Good examples of this are sodium chloride (NaCl) or potassium chloride (KCl) deposits in process steam in industrial applications, or silica deposits in large, supercritical utility units (GEK-72281A, 1996). For the cases of sodium chloride or potassium chloride, the deposits can be effectively removed by washing the steam path with wet steam. Silica deposits are much more tenacious, as they are insoluble in water and may have to be removed mechanically.

Tests to evaluate the effects of sodium chloride and potassium chloride deposits on brush seals have been conducted by one manufacturer. Deposits were formed by immersing brush seal samples in water solutions of sodium chloride and potassium chloride, in various concentrations, then boiling off the water. Bristle pack stiffness and leakage characteristics were then measured in static tests. Not surprisingly, with a heavily encrusted bristle pack (from a 5 percent NaCl solution), seal leakage is actually reduced by 28 percent due to a reduction in the porosity of the bristle pack. Seal stiffness is only increased substantially until the deposits are either broken by force (in which case the stiffness recovers to within 2.5× its original value), or the deposits are washed off. In general, seal behavior in the presence of potassium chloride deposits did not vary significantly than for sodium chloride.

Dynamic tests to evaluate heat generation in the presence of NaCl deposits were also conducted. In this test, a brush seal was assembled with a nominal 0.010 inch bristle interference on a 5.100 inch diameter test rotor, and the rotor speed held at 4000 rpm for one hour. Temperature measurements taken using thermal imaging equipment showed a maximum steady-state temperature at the seal/rotor junction of 34°C (93.2°F), reached after 10 minutes. The seal was removed, NaCl deposits were created in a 5 percent solution, the seal with deposits was baked in an oven at 900°F for one week, and the test was repeated. After reaching a peak of 108°C (226.4°F) in 3.5 minutes, the maximum seal temperature quickly settled to a steady-state value of approximately 40°C (104°F) within 20 minutes. No adverse effects on rotordynamics were observed throughout the tests.

In addition to the seal performance evaluations described above, metallurgical investigation of the Haynes® 25 bristles revealed no adverse effects from the exposure to NaCl or KCl solutions.

Investigations into the effects of silica deposits have not yet been completed, simply due to difficulties in finding a suitable method for simulating silica deposits in the laboratory.

LABORATORY TESTING

In developing brush seals for steam turbine applications, laboratory testing plays an important role. One original equipment manufacturer (OEM) has a dedicated seals test facility with rigs for testing under conditions that approximate those found in a steam turbine environment.

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, a subscale test rig was employed, as shown in Figure 7. The rig is capable of testing with either air or steam as the working fluid, at pressures up to 1200 psia and temperatures up to 1000°F. The rotor is a solid CrMoV (chromium molybdenum vanadium) shaft that can be run at any speed above or below its second critical speed, with a maximum rotor surface speed of 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 allow different seal types to be compared, as shown in Figure 4.

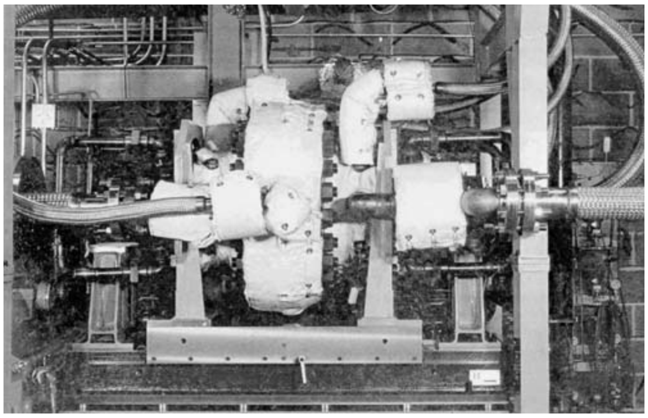


Figure 7. Seals Test Rig.

Rotordynamics Testing

In addition to the performance data described above, the corporate research and development (CRD) 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 rotordynamics, 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.

Wear Testing

In many aircraft engine applications, the rotor surface immediately under the brush seal is coated with a wear-resistant material, typically 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, the 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 (SCC) and hydrogen embrittlement.

Field experience to date has correlated very well with the laboratory testing. After three years of operation in one unit, rotor wear is on the order of one mil and bristle wear is minimal. The bristles are still contacting the rotor surface when pressurized and the seals are performing as expected.

Steam Turbine Test Vehicle

An important tool for validating new steam turbine component performance is the steam turbine test vehicle (STTV, Figure 8) developed by one OEM and located in Lynn, Massachusetts. This is essentially 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 (Willey, et al., 2000). 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 percent; this result is consistent with the estimated performance gain based on expected brush seal effective operating clearances. These tests demonstrate not only that brush seals provide an appreciable improvement in turbine performance, but that the method used to predict the performance improvement is sound.

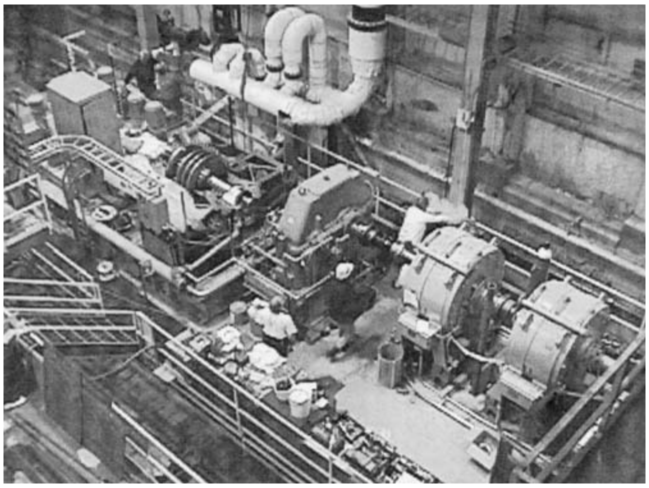


Figure 8. Steam Turbine Test Vehicle.

Field Experience

The gas turbine fleet of one OEM has several hundred brush seals in service, the first of which passed their 24,000 hour hot gas path inspection and were returned to service; the locations include the high pressure packing, the middle bearing (for three bearing machines), and the turbine interstage. Nine steam turbines, ranging from 20 MW industrial turbines to a 900 MW supercritical utility turbine, are operating with brush seals at interstage, end packing, tip seal locations.

The very first steam turbine brush seal, an outer packing seal in a small industrial turbine, was inspected after three years of service and found to be in excellent condition. More recently, a number of brush seals were inspected after 17 months of services on a 250 MW utility steam turbine.

Six brush seals were installed in various locations of an opposed flow high pressure-intermediate pressure steam turbine, as shown in Figure 9. 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 intermediate-pressure (IP) section. The seals were installed during a maintenance outage in June 1999, and inspected during a boiler outage in November 2000.

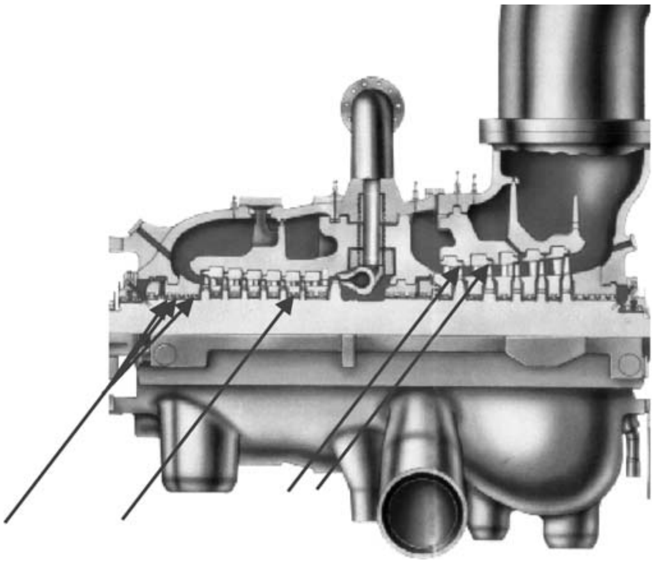


Figure 9. Brush Seal Locations on Opposed Flow High Pressure-Intermediate Pressure Turbine.

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 featured the presence of a notch block taking the place of one bucket in the row, creating a severe once per revolution transient in the pressure field immediately beneath the bristle pack. The bristles of this seal did not survive, and work to extend brush seals' current limits of durability to these types of application is ongoing.

The other 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 sideplates or packing rings. The end packing is shown in Figure 10, and closer views of the packing ring with brush seal are shown in Figures 11 and 12. Some of the packing teeth in Figure 12 show evidence of light to moderate packing rubs. The stage nine tip brush seal, which rides over a row of integral cover buckets, is shown in Figure 13. Note that the segment end design has since been improved to eliminate the presence of bristles caught between segments.

The brush seals in this turbine were designed for low contact pressure to mitigate the impact of the seals on rotordynamics. Startup in 1999 was very smooth, and there have been no rotordynamics issues with the unit since.

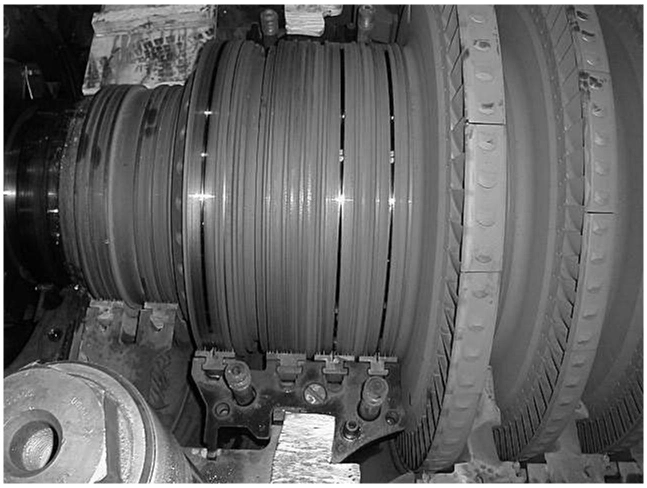


Figure 10. View of Brush Seals at HP Endpacking During Recent Inspection.

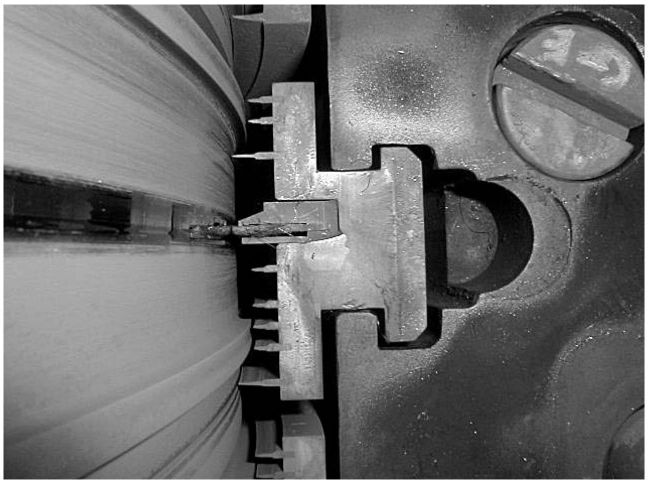


Figure 11. Typical Rotor Surface Finish Under a Brush Seal.

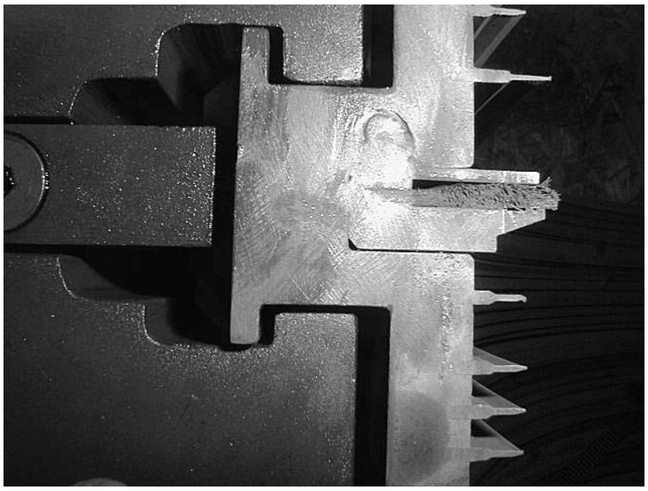


Figure 12. Brush Seal View at Horizontal Joint.

CONCLUSION

In this paper, we have described the application of brush seals to steam turbines, covering the benefits in performance and the design issues that must be considered. To date there are nine steam turbines running with a combination of interstage packing, end packing, and bucket tip seals. These include both industrial steam turbines from 20 MW to large utility turbines of 900 MW.

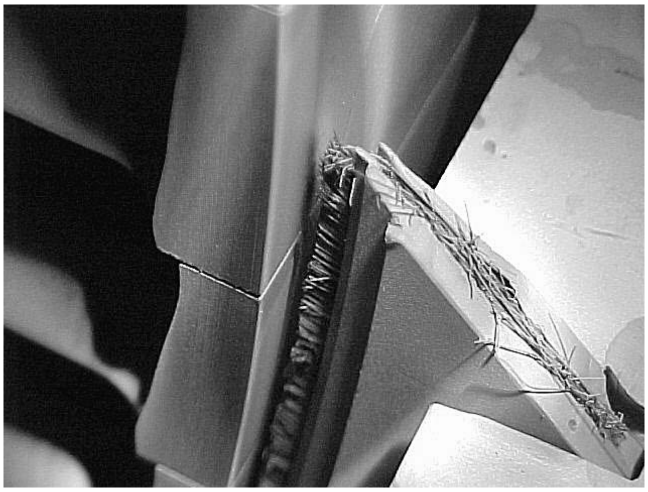


Figure 13. Horizontal Joint View of Brush Seals at Bucket Tips.

The first steam turbine brush seals to be installed have recently been inspected and were found to be in excellent condition after three years of service. Brush seals installed more recently in a utility steam turbine have also been inspected and were found to be in excellent condition after 17 months of service, with the exception of one bucket tip seal subjected to a particularly abusive operating environment. While bucket tip seals are still under development, steam turbine shaft brush seals are now a robust product offering from the OEM with validated leakage reduction and reliability performance. Development efforts continue both to refine the current design and to expand the range of possible applications.

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