Brush Seals and Common Issues In Brush Seal Applications

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ABSTRACT

Brush seals are receiving wider acceptance with growing number of successful turbomachinery sealing applications over the past couple of decades. Due to their complaint nature these seals can recover from large transient rotor interference occurrences without any appreciable sustained damage or permanent performance loss. They are formed by a multitude of flexible fine bristles tightly clamped between two metal plates. When subjected to operating pressures and rotor interference, the frictional interactions between thousands of bristles result in very complex seal behavior. Brush seals are known to exhibit load path dependent hysteresis behavior. They also experience increased stiffness under differential pressure loads. While pressure distribution induces some radial forces to close the bristles towards rotor, mechanical interlock and interbristle frictional effects may lead to seal hung up after large rotor interference. In this paper, typical aspects of brush seals are presented. Complex seal behavior and common issues with brush seal applications are also discussed. Recommendations and suggestions for successful seal design are provided.

1.0 BRUSH SEAL

1.1 Seal Design

A brush seal consists of fine diameter fibers densely packed between two plates. A support plate that is called as ¨backing ring¨ or ¨backing plate¨ is positioned downstream of the bristles to provide mechanical support for the differential pressure loads (see Figure 1). In most metallic brush seal applications fibers range 0.07 to 0.15 mm (0.0028 to 0.006 in.) in diameter. Fiber density is defined as number of bristles per length of rotor (seal bore) circumference. Although typical fiber density is around 2000 fibers per inch, depending on the fiber diameter this value ranges between 800-2600.

Some of the main brush seal design parameters can be listed as:

- bristle diameter (d)
- cant angle (θ)
- free bristle height (BH)
- fence height (FH)
- material properties (elastic modulus, yield & ultimate strength, friction & wear coefficients against rotor material etc.)

Parameter	**Units**	**Typical Range**
Bristle diameter	d *[mm]*	0.07 - 0.15
Free bristle height	BH *[mm]*	10 - 20
Fence height	FH *[mm]*	1 - 2.5
Cant angle	θ *[deg]*	35 - 55

Table 1: Typical range of brush seal main design parameters

Table 1 provides typical ranges for the major brush seal design parameters. In addition to these parameters, special features may be needed in backing plate and front plate design. Backing plate pressure pockets are common. Apart from very small seal diameter applications, rotor curvature effects are typically neglected. The following operating conditions are also critical in brush seal design.

- differential pressure (ΔP)
- radial seal interference (ΔR)
- inlet flow temperature (T)
- Type of fluid: air, steam, oil, oil mist

In addition to the above listed operating conditions, in some applications axial rotor displacement also plays an important role in seal design. It may be needed to make a decision whether a coating is required on rotor or not.

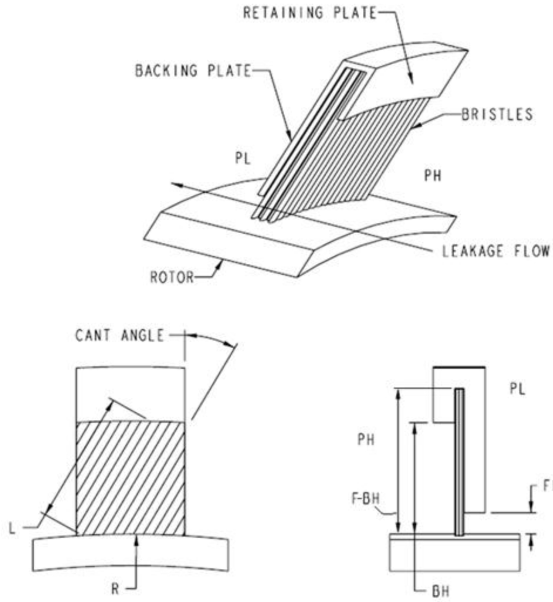

Figure 1: Brush seal schematic and main design parameters.

As illustrated in Figure 1, L denotes free bristle length, and BH shows free bristle height. Fence height (backing plate clearance) is denoted by FH, and R is used for rotor radius. A retaining plate, which is also referred as front plate, tightly clamps and holds the bristles in place. The circular seal is installed in a static member with bristles touching the rotor at an angle in the direction of the rotor rotation. This bristle angle is called as ‘cant angle’ or ‘lay angle’. Typically, the cant angle θ is around 45^o. In the case of rotor excursions, cant angle helps reduce the contact loads, allowing bristles to bend rather than buckle. This inherent flexibility enables the seal to survive large rotor excursions without sustaining any appreciable permanent damage. Because the bristles slide against the rotor surface, wear at the contact becomes a major concern as it determines the life and efficiency of the seal. Brush seals go through an initial wear-in period, and eventually wear line-to-line at the maximum radial excursion point. Ideally, wear in brush seals would be characterized by minimal material removal from both the bristles and the rotor. Practice, however, dictates that one must usually trade a higher degree of bristle wear for minimal rotor wear. The choice is complicated by the fact that the relative rates of wear change with temperature and surface speed [1].

2.0 MATERIAL SELECTION

Fine diameter bristles under heavy pressure loads may experience excessive stress levels. Therefore, high fiber material strength is needed. As bristles may come into contact with rotor at high surface speeds, careful material selection is required to have adequate 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 characteristics, the seal material must have acceptable creep and oxidation properties [2].

Most brush seals have metallic fibers. Typical wire diameters range from 0.07 mm (0.0028 in.) for low pressure applications to 0.15 mm (0.006 in.) for high pressure applications. Due to its good oxidation resistance and excellent high temperature friction and wear properties cobalt based super alloys are preferred as bristle material. Haynes 25 is the most commonly used material for brush seal bristles [2].

After many successful air applications, brush seals are also considered as oil-mist seals in bearing sump locations. Bearing oil sump applications involve more stringent design requirements than typical air applications. Some of these issues include the risks of coking (carburization of oil at excessively high temperatures), foreign particle damage of precision rolling element bearings, and potential for fires [2]. Among nonmetallic fibers nylon and polyester fibers have been tried first due to availability in similar fiber diameters to those of common metallic versions. Although it has very high strength, nylon has been abandoned due to its low temperature limit and high moisture absorption. After some early trials polyester has been also abandoned, since bristle tip melting and fusing have been observed in subscale prototype testing [3].

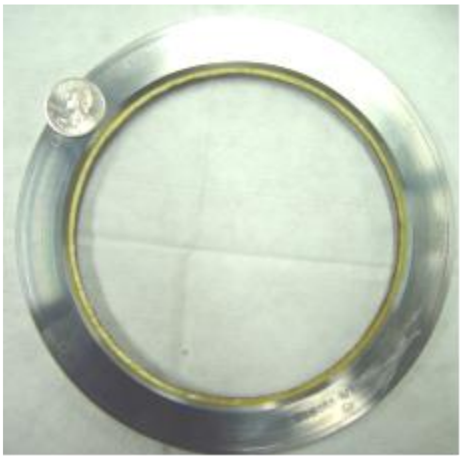

Figure 2: Aramid brush seal prototype [4].

Although they have very small (12 μm) fiber diameters, development efforts in applying aramid bristles for certain bearing sump locations have found success. Figure 2 illustrates an aramid brush seal prototype. Aramid bristles have stable properties up to 150 °C operating temperatures, and have negligible amount of shrinkage and moisture absorption. The tests indicate that brush seals made of aramid fibers have lower wear rate than those with Haynes 25 up to 150 °C (Figure 3). Aramid brush seals have much lower leakage rates than that of metallic brush seals due to higher brush density (Figure 4). Further details on aramid brush seals performance can be found in reference [4].

Brush seals are generally run against a smooth, hard-face coating to minimize shaft wear and the chances of wear-induced cracks from affecting the structural integrity of the rotor. The usual coatings selected for aircraft applications are ceramic, including chromium carbide and aluminum oxide. Selecting the correct mating wire and shaft surface finish for a given application can reduce frictional heating and extend seal life through reduced oxidation and wear. There is no general requirement for coating industrial gas and steam turbine rotor surfaces where the rotor thicknesses are much greater than aircraft applications [2].

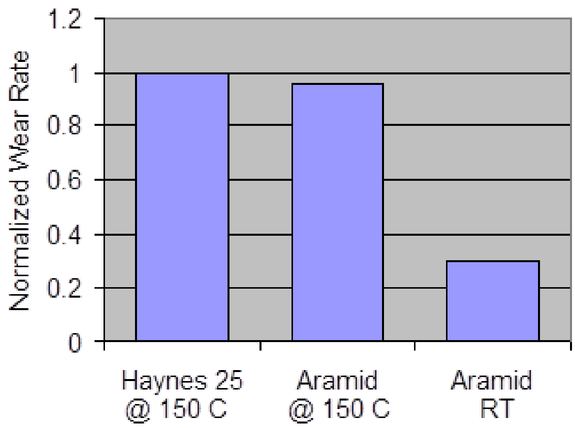

Figure 3: Wear test results for aramid and Haynes 25 tufts against Ni-Cr-Mo-V. Data are normalized with wear rate of Haynes 25 bristles at 150 °C [4].

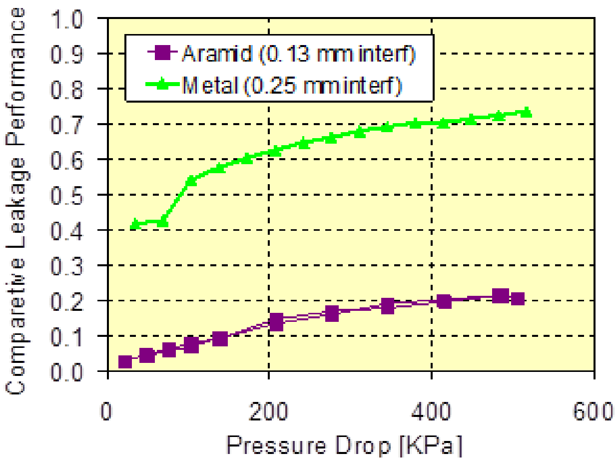

Figure 4: Comparison of aramid and metallic brush seal leakage performances [4].

3.0 SEAL BEHAVIOR

The complicated sealing mechanism of brush seals dictates a strong coupling between differential pressure and seal stiffness. When coupled with frictional effects, loads created by the leakage flow and differential pressure cause three major phenomena commonly observed in most brush seal applications; pressure stiffening, hysteresis, and bristle blow-down. Below, common seal behavior is discussed.

3.1 Pressure-Stiffness Coupling

Differential pressure load across the brush pack pushes the bristles against the backing plate. As shown in Figure 5, under the effect of frictional mechanisms, bristles stick to one another, and the bristle pack sticks to the backing plate. The frictional resistance at the backing plate causes a large increase in the contact load at the rotor surface. When pushed radially during a rotor excursion, the seal feels much stiffer than it does without any pressure load. This leads to increased wear rates which reduce the seal life.

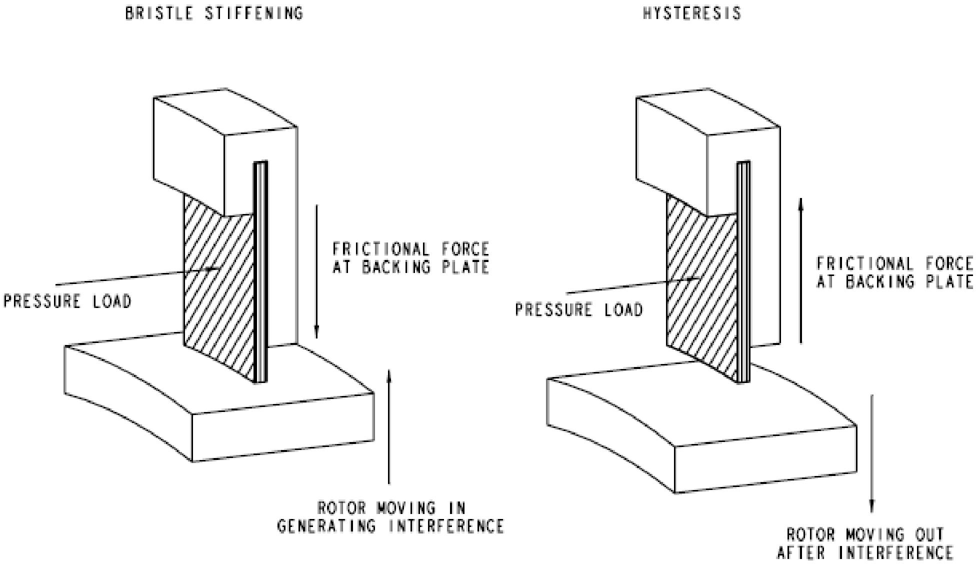

Figure 5: Role of friction in seal stiffening [5].

3.2 Hysteresis

Hysteresis is an immediate consequence of the pressure stiffening effect. During a rotor excursion, while the seal is stiff under the differential pressure load, bristles are pushed radially out in order to accommodate an eccentric or thermally-expanded rotor. After the transient conditions are over, the rotor returns to its original or steady state position or dimension. At this point, radially displaced bristles do not follow the rotor, and remain 'hung up' as they are locked in position by the pressure load tightly pressing them against the backing plate (see Figure 5). Once again, friction plays a major role here. Hysteresis is more a leakage performance problem rather than a durability issue. With the bristles 'hung up', leakage rate is increased until the seal recovers when pressure load drops below a certain level. Hysteresis can also be observed without any rotor interference. If a sample pressurization-depressurization cycle is applied on a brush seal, leakage is different for pressurization leg than that for the depressurization leg. Once the bristles are pushed and locked in a certain position, they leak differently during depressurization. The leakage results of a pressurization-depressurization cycle is called the 'hysteresis curve.'

3.3 Pressure-Closure

When a differential pressure load is applied on a brush seal, bristles are known to move radially inwards towards the rotor. This is referred as 'pressure closure' or 'blow-down' in various literature. Although the exact nature of the factors causing pressure closure is unknown, it is believed that a limited amount of radial flow within the seal might cause the high bristles to close towards the rotor. Contrary to pressure stiffening and hysteresis, pressure closure is caused by a lack of restraints. Across the bristle pack from the high pressure (front) side to the low pressure (rear) side, cumulative pressure load increases due to load transfer from front bristle rows to rear ones. This results in a case where the rearmost bristle row next to the backing plate experiences the full pressure load, while the very front row at the high pressure side suffers only a small drag due to the leakage flow. With little restriction, high pressure-side bristles are prone to move towards the rotor under the influence of secondary radial leakage flow. On the other hand, as illustrated in Figure 6, pressure gradient across the pack reduces as moved radially out. Therefore, a large pressure difference also occur radially along the backing plate. Therefore, towards downstream side, aerodynamic blow-down forces are also increasing. Although the main leakage is axially oriented, there is some radial flow occurring above the fence height as shown in Figure 6. High pressure air penetrates the bristle pack and flows radially towards the rotor before turning axial under the backing plate. This radial flow (or the radial pressure difference driving this flow) pulls the bristles towards the rotor closing the tip clearance to some extent.

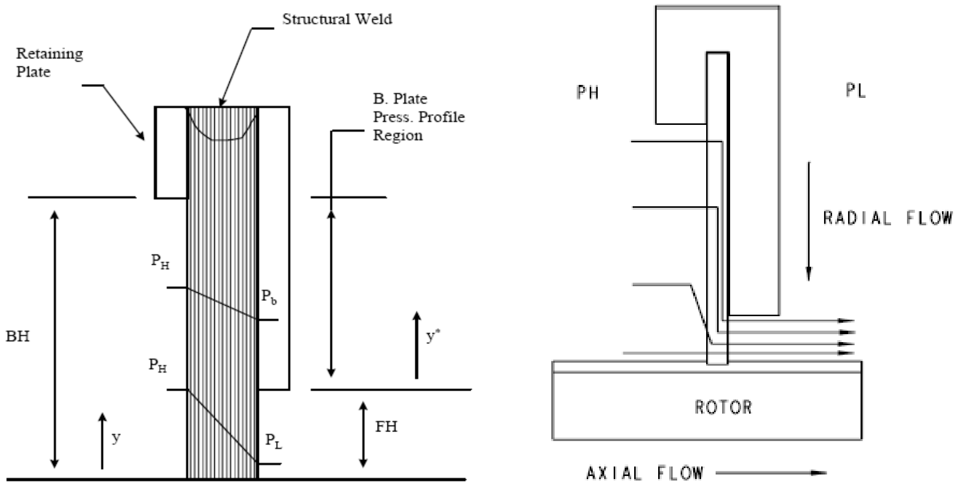

Figure 6: Brush seal internal pressure distribution and leakage flow through the bristle pack [5].

Recently, improved seal designs utilize backing plate pressure balance cavities or extended front/retaining plates with some proper relief to reduce hysteresis behavior. Backing plate pressure balance cavities reduce frictional forces between the backing plate and the brush pack, and reduce stiffening and hysteresis. The extended retaining plates also help reduce pressure closure by restricting the radial flow through the seal. However, experience shows that some pressure closure may still be present in certain instances. This indicates the presence of some other causes for this phenomenon. Under the axial pressure load, the overhanging portion of the bristles bend below the backing plate in the flow direction. While bristles bend under the backing plate, they also tend to slide down a little towards the rotor due to lay angle. Basically, the geometry of the structure with some cant angle is also contributing to the pressure closure under axial bending loads. Level of closure is dependent on pressure as well as the cant angle. It is more severe with increased pressure and cant angle. This may also explain the reason that uneven wear is only visible in some cases.

Overall, pressure closure helps improve leakage performance, reducing leakage even when there is a clearance. On the other hand, it also leads to continued wear of front side bristles beyond the maximum rotor interference.

3.4 Bristle Stability

Being relatively free to move, the high pressure side bristles tend to flutter under the influence of aerodynamic disturbances at the inlet region. The disturbances are usually turbulence or jet flow coming from the upstream in a cascade setup. Jet flow coming beneath the upstream seal or labyrinth tooth creates a vortex which tends to pull upstream front row bristles away from the brush pack, and causing them to flutter. The upstream vortex is clearly illustrated by O'Neill et al. [6]. Extended front plates help protect the upstream bristle from aggressive incoming flow turbulence and play an important role in minimizing aero instability damage to front row bristles. Similarly, in some improper designs, downstream bristles may be lifted off the rotor surface due to very high pressure loads, and fail under high bending stress and aerodynamic flutter combination.

Coupled together, flutter and pressure closure cause uneven wear of bristle tips at high pressure side. As illustrated in Figure 7 after some time in service, seal profile changes, and bristles at the front side wear more than the bristles near the backing plate. However, although irregular front row wear may be observed, seal may continue to perform blocking of leakage flow.

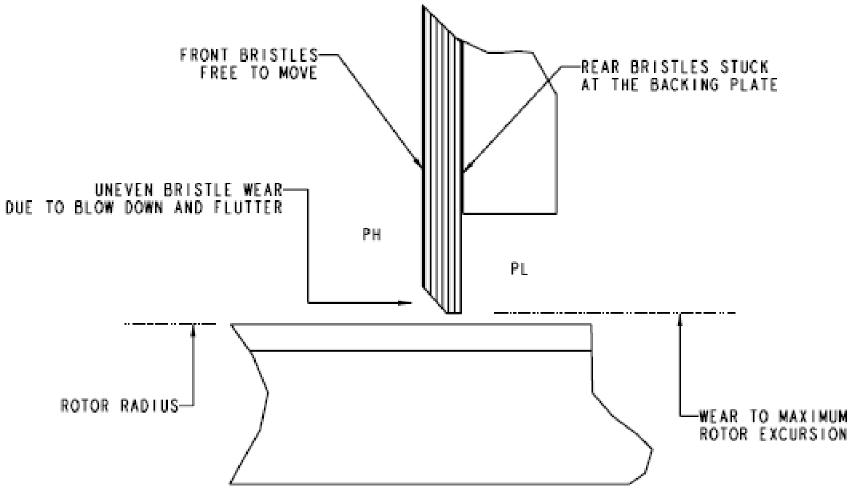

Figure 7: Uneven wear of bristle tips caused by blow-down and flutter [5].

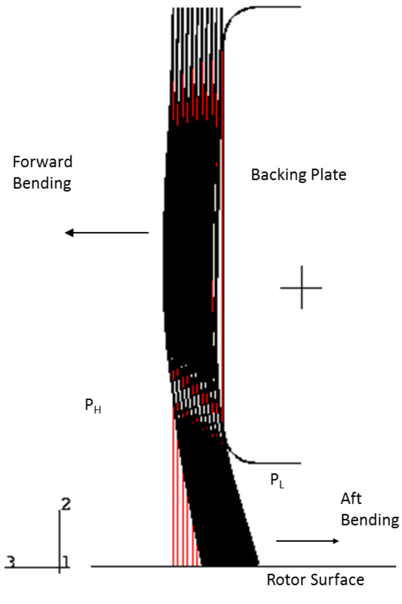

Figure 8: Brush pack behavior under pressure load [5].

3.5 Pressure Capability

Brush seal pressure load capability typically dictated by fence height (backing plate to rotor radial clearance), bristle diameter and bristle lay angle among others. As illustrated in Figure 8, under differential pressure load, brush pack deflects downstream in the unsupported fence height region while it bows forward in the backing plate supported section. When pressure load is increased, brush seal leakage gradually increases with pressure after an initial settling as shown in Figure 9. If pressure load reaches excessive levels, leakage rate increase becomes exponential beyond certain pressure load. This sudden increase in leakage flow can be attributed to lifting the bristles off the rotor surface and reduction in the brush density in the fence height region due to excessive bending of bristles under heavy pressure load. The onset of exponential leakage increase signals the pressure load limit of the brush seal. For a proper brush seal design, one should stay safely below this limit during operation.

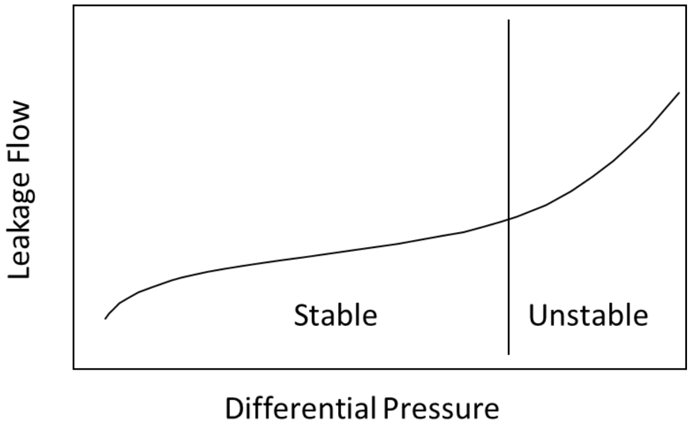

Figure 9: Brush seal leakage under increasing pressure load.

4.0 APPLICATION ISSUES

There are some practical issues with brush seal turbine applications. As higher pressure differences drive larger leakage flows, potential for engine efficiency and power gains are higher at large differential pressure locations. Naturally, engineers want to apply brush seals at increasingly high pressure applications. As stated under pressure capability discussion above, there is a limit to pressure load which a brush seal can safely handle. Brush pack thickness cannot be increased beyond some practical limits due to manufacturing issues, as well as the fact that the pack starts losing its uniformity, and intermixing of stray fibers increases brush pack porosity and affects stiffness. On the other hand, extra caution should be taken while using larger bristle diameters, as seal gets very stiff. If not designed properly, stiff seals may damage rotor. It may be possible to control rotor wear damage by some hard coating applications. However, the excessive heat generated during large rotor excursions may cause micro cracks, or lead to shaft thermal deflections and thermal rotor instability. Use of multiple softer brush seals in cascade applications has other known issues. When multiple brush packs are used in cascade configuration, typically the last brush seal at the downstream side takes major portion of the pressure load across the entire cascade. Therefore, it may fail first. Once the last brush pack fails under heavy pressure load, the majority of the pressure difference will shift to the next brush pack causing its failure, and so on. However, successful single brush stage application has been reported up to 2.76 MPa differential pressure [7].

Another issue with brush seal engine applications is the segmented installation. Fully circular brush seal installations are possible in aircraft engine applications. However, in large industrial turbines seal diameters beyond 1000 mm are very common. For large industrial gas and steam turbine sealing applications, brush seal has to be segmented due to practical issues in manufacturing and turbine assembly procedures. When a brush seal needs to be assembled in segments, the seal has to be sliced at bristle lay angle. Otherwise, bristle loss and excess leakage occur at seal segment interfaces. Overlapping the backing plates of neighboring segments will further reduce intersegment leakage. Care should be taken in segmented applications as gravity may push all segments to one side. If properly designed, intersegment leakage is negligible in large seals.

As bristles touch rotor surface with some lay angle, reverse rotation is another issue in brush seal applications. Especially in large industrial gas and steam turbines, reverse shaft rotation is common occurrence during maintenance and blade installations. The field experience shows that very low speed reverse rotation with smooth unsegmented shaft surfaces during turbine maintenance and assembly does not cause any brush seal damage. However, reverse rotation at high speeds or with segmented shaft surfaces will damage the brush pack. On the other hand, brush seals can be designed to run on segmented rotor surfaces, such as shrouded blade tips in steam turbines, if reverse rotation is avoided during turbine installation.

Hydrodynamic lift and oil coking are other issues in oil sealing applications of brush seals. The measurements in continuous oil flow show quick rise in leakage rate with initial rotor speed, a clear indication of hydrodynamic lift of bristles. However, the amount of lift and leakage get stabilized as speed increases due to shear thinning of oil and balancing of the lift force. When the lift/radial clearance increases, the hydrodynamic lift force decreases, while the bristle tip force (due to bristle bending, blow-down and frictional interactions) increases. The seal operating clearance stabilizes when these forces are balanced. Oil coking may be an issue if seal is designed too stiff or makes large interference rubs with rotor. Further discussion on brush seal application issues in oil sealing can be found in references [8-13].

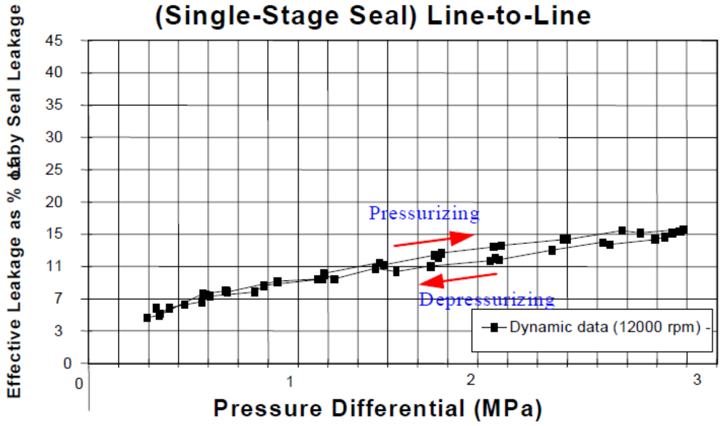

Figure 10: Comparison of labyrinth and brush seal leakage performances [7].

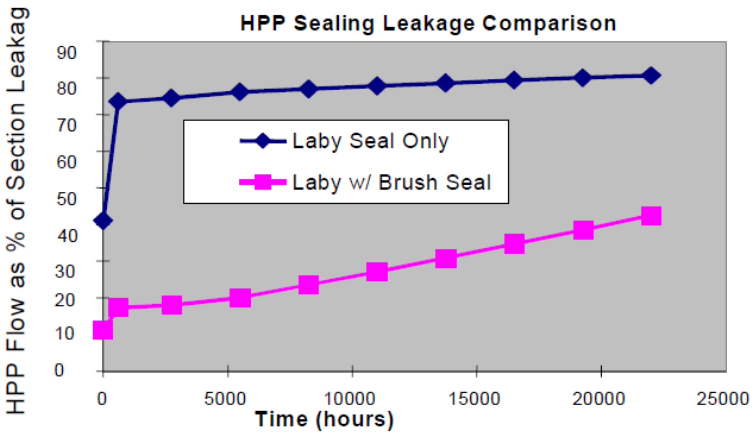

Figure 11: Comparison of labyrinth and brush seal performance degradation during service [7].

5.0 TURBINE APPLICATIONS & PERFORMANCE BENEFITS

Brush seal leakage performance is complicated issue that depends on many factors. To name a few

- bristle density: changes porosity
- fence height: Backing plate-rotor clearance determines exposed porous flow area
- compliance: individual bristles move around, inter-bristle gaps change with pressure
- hysteresis: changes seal tip clearance
- blow-down/pressure closure: changes seal clearance
- seal wear: changes seal tip clearance
- hydrodynamic (oil applications): changes seal tip clearance

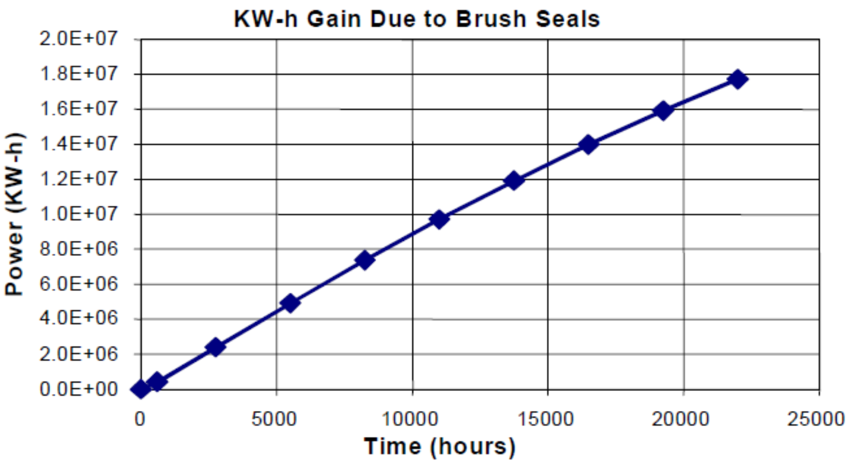

Figure 12: Cumulative power gain due to sustained brush seal performance during service [7].

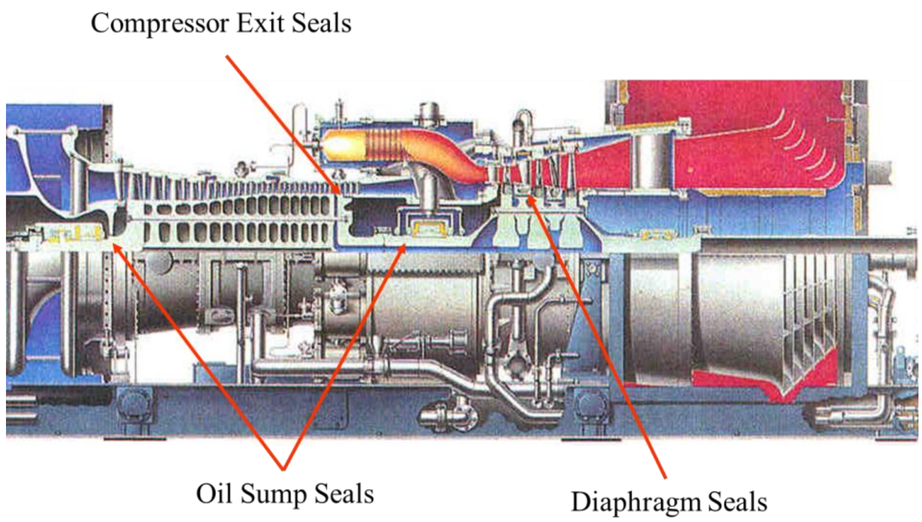

Figure 13: Typical brush seal application areas in a gas turbine.

Brush seals have good leakage performance to begin with. In fact, Ferguson [14] reports that a brush seal replacing the best possible finned labyrinth seal, with a clearance of 0.7 mm (0.027 in.), can reduce the flow to approximately 10% of that of the finned seal. As also illustrated in Figure 10, Dinc et al. [7] reports 85-95% leakage reduction with brush seals over labyrinth seal. Moreover, this performance gain is sustained over much longer operational period as compared to sudden permanent performance loss of labyrinth seals after initial rotor rubs (Figure 11). This sustained leakage performance leads to large cumulative savings during the life of the seal (Figure 11) [7].

HPP	%MW & %Heat Rate
32G	+0.7%/-0.5%
32J	+0.7%/-0.5%
51P	+0.6%/-0.45%
61B	+1.0%/-0.5%
71E,EA	+1.0%/-0.5%
91E	+1.0%/-0.5%
ISTG	
51N	+1.0%/-0.5%
51P	+1.0%/-0.5%
61B	+1.0%/-0.5%
71E,EA	+1.0%/-0.5%
91E	+1.0%/-0.5%
BRG2*	%MW & %Heat Rate
71E	+0.3%/-0.2%
91E	+0.3%/-0.2%
HPP and BRG2**	
52C	+0.7%/-0.5%
52D	+0.9%/-0.6%
71B	+1.0%/-0.5%
91B	+1.0%/-0.5%

Table 2: Performance benefits for gas turbine brush seal applications [7].
(* Combination of forward and aft seals)
(** HPP and #2 bearing seals combination required for these machines)

The substantial cumulative power gains due to sustained brush seal performance has led to multitudes of brush seal applications in different turbomachinery. As illustrated in Figure 13, typical brush seal applications in gas turbines include diaphragm locations in turbine section, compressor exit (high pressure packing) and recently oil sump applications. Steam turbine applications include interstage locations, end packings and recently shrouded bucket/blade tip applications. Dinc et al. [7] report efficiency and power increase benefits for different gas turbine (see Table 2) and steam turbine applications (see Table 3).

Turbine Class and Location	Efficiency Benefit
Utility Steam Turbines (HP Section)	
End Packings (multiple locations)	0.1-0.2% unit heat rate
Interstage Packing (multiple stages)	0.5-1.2% HP section efficiency; 0.1-0.2% unit heat rate
Industrial Steam Turbines	
End Packings (multiple locations)	0.4 - 0.8% efficiency
Interstage Packing (multiple stages)	0.2 - 0.4% efficiency

Table 3: Performance benefits for steam turbine brush seal applications [7].

6.0 CONCLUSION

With their flexible fibers, brush seals can maintain much tighter rotor clearance levels than rigid labyrinth seals. By providing physical blockage of clearances via flexible bristles, brush seals provide up to 10 folds improvement in leakage performance over labyrinth seals. Moreover, the inherent flexibility of brush seals ensures prolonged superior leakage rates without any appreciable permanent performance loss after large rotor excursions. The sustained performance advantage results in much bigger cumulative gains in practical applications. With all the above mentioned positive aspects brush seals gain more and more popularity in many turbomachinery applications. However, their complex nature requires in-depth knowledge and experience with brush pack behavior. Flexible bristles lead to unique seal behavior. If not designed properly, pressure-stiffening, blowdown, hysteresis, and flutter may cause performance and durability issues. With their superior leakage performance brush seals applications are expected to grow across industrial turbines and aircraft engine applications.

7.0 REFERENCES

[1] Atkinson, E. and Bristol, B., 'Effects of Material Choices on Brush Seal Performance,' Lubrication Engineering, 48, pp 740-746, 1992.

[2] Chupp R.E., Hendricks R.C., Lattime S.B., Steinetz B.M., Aksit M.F., "Turbomachinery Clearance Control," NASA Technical Report ID: 20070005016, NASA Glenn Research Center, NASA Scientific and Technical Aerospace Reports (STAR), No.3, Vol.45, pp.114, February 2007.

[3] Aksit, M.F., Bhate, N., Demiroglu, M., and Bouchard, C., "Evaluation of Brush Seal Performance for Oil Sealing Applications," 39th AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, AIAA-2003-4695, 2003.

[4] Bhate, N., Thermos, A.C., Aksit, M.F., Demiroglu, M., Kizil, H., "Non-Metallic Brush Seals For Gas Turbine Bearings", Procs. of ASME Turbo Expo 2004, GT2004-54296, 2004.

[5] Aksit M.F., "Evaluation of Brush Seal Performance for Oil Sealing Applications," PhD. Dissertation, Rensselaer Polytechnic Institute, Troy, NY, May 1998.

[6] O'Neill, A.T., Hogg, S.I., Withers, P.A., Turner, M.T. and Jones, T.V., 'Multiple Brush Seals in Series,' ASME Gas Turbine and Aeroengine Congress Paper ASME 97-GT-194, (1997).

[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," ASME Journal of Turbomachinery, No.2, Vol.124, pp.293, 2002.

[8] Aksit, M.F., Dogu, Y., J.A. Tichy and Gursoy M., "Hydrodynamic Lift of Brush Seals in Oil Sealing Applications," Proceedings of 40th AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, Fort Lauderdale, Florida, AIAA–2004–3721, July 2004.

[9] Cetinsoy, E., Aksit, M.F., and Kandemir, I., "A Study Of Brush Seal Oil Lift Through Long Bearing Analysis," Proceedings of IJTC2006 STLE/ASME International Joint Tribology Conference, San Antonio, TX, ASME Paper IJTC2006–12370, 2006.

[10] Duran, E.T., Aksit, M.F., and Dogu, Y., "Effect of Shear Heat on Hydrodynamic Lift of Brush Seals in Oil Sealing," Proceedings of 42nd AIAA/ASME/SAE/ASEE Joint Propulsion Conference & Exhibit, Sacramento, California, AIAA Paper AIAA–2006–4755, 2006.

[11] Duran E.T., Aksit M.F., and Dogu Y., "Oil Temperature Analysis Of Brush Seals," Proceedings of STLE/ASME International Joint Tribology Conference, San Diego, California USA, ASME Paper IJTC2007-44397, 2007.

[12] Duran E.T., and Aksit, M.F., "A study of brush seal oil pressure profile including temperature-viscosity effects," 44th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit, Reston, VA, USA, AIAA Paper AIAA-2008-4622, 2008.

[13] Duran, E. T. and Akşit, M.F., "Effect of Shear Heat on Hydrodynamic Lift of Brush Seals in Oil Sealing," Proceedings of The STLE 65th Annual Meeting & Exhibition, Atlanta, GA, USA, 2010.

[14] Ferguson, J.G., "Brushes as High Performance Gas Turbine Seals," ASME Gas Turbine and Aeroengine Congress Paper ASME 88-GT-182, 1988.