Measurements of Leakage and Power Loss in a Hybrid Brush Seal
Mesures d'étanchéité et de puissance pour un joint hybride à brosse

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Simplicity, low cost and easy replacement make of labyrinth seals the primary seal type in gas turbines. However, excessive leakage and potential for rotordynamic instability are well known issues. Brush seals effectively control leakage in air breathing engines, albeit only applied for relatively low-pressure differentials. Hybrid brush seals **(**HBS) are an alternative to resolve poor reliability resulting from bristle tip wear while also allowing for reverse rotation operation. The novel configuration incorporates pads contacting the shaft; and which under rotor spinning; lift off due to the generation of a hydrodynamic pressure. The ensuing gas film prevents intermittent contact; thus lowering the operating temperature and thermal distortions, and even eliminating bristles' wear. The hybrid brush seal does improve sealing, and is more durable and reliable than conventional brush seals, while even allowing for reverse shaft rotation. The paper presents measurements of power loss and leakage in a hybrid brush seal (HBS) for increasing pressure differentials and over a range of rotor speeds. The test HBS, Haynes-25 bristle pack [~850 bristles/cm] and 45^o lay angle, is 166.4 mm in diameter and integrates 20-arcuate pads connected with thin EDM-webs to the seal casing. The webs are designed with low radial stiffness to allow for rotor excursions and high axial stiffness to avoid pad pitching at high levels of external pressurization. Measured drag power at low rotor speeds (< 11 m/s at 1,300 rpm) decreases as the pressure differential across the seal increases. At a fixed rotor speed, a significant drop in drag torque (and drag power) ensues as the supply pressure increases, thus demonstrating a gas film separates the rotor from the seal pads. A constant operating temperature (~24^{o}C) at the rotor/seal interface during tests with shaft rotation also indicates the absence of intermittent contact between the seal pads and rotor. Flow rate measurements at room temperature (25^{o}C) show an improved sealing ability with a leakage reduction of about 36%, when compared to a first generation shoed-brush seal. The HBS predicted effective clearance (~50 μm) is a small fraction of that in an equivalent one-tooth labyrinth seal. Improved brush seal technology will increase the efficiency of gas turbines while also aiding to improve the engine stability and to reduce vibrations.

La simplicité, le bas coût et un remplacement facile font des joints labyrinthes le principal type de joints utilisé dans les turbines à gaz. Cependant, des fuites importantes et une possible instabilité dynamique du rotor sont des conséquences inhérentes à leur utilisation. Les joints à brosses contrôlent efficacement les fuites dans les (turbo) moteurs, bien qu'uniquement utilisés dans des cas où les différentiels de pression sont faibles. Les joints à brosses hybrides **(**HBS) sont une alternative permettant de palier à la mauvaise fiabilité résultant de l'abrasion des brins tout en permettant les rotations inverses. Cette nouvelle configuration inclue des lames mettant en contact le joint et l'axe, et qui, lors des rotations du rotor, se décollent grâce à la génération de pression hydrodynamique. Le film de gaz ainsi créé empêche les contacts intermittent, diminuant ainsi la température d'utilisation et les distorsions thermales, éliminant même l'abrasion des brins. Les joints à brosses hybrides améliorent l'étanchéité et ont une durée de vie et une fiabilité plus importante que les joints à brosses conventionnels, tout en permettant les rotations inverses de l'axe. Cet article traite des mesures de pertes de puissance et d'étanchéité d'un joint à brosses hybride (HBS) dans des conditions de différentiels de pression augmentant et pour différentes vitesses de rotation du rotor. Le modèle HBS testé, Haynes-25 brins [~850 brins/cm] et 45° d'inclinaison de brins, a un diamètre de 166.4 mm et comporte vingt lames arquées connectées à la structure du joint par de fines toiles EDM. Ces toiles ont une rigidité radiale faible permettant les excursions du rotor et une rigidité axiale importante évitant ainsi une flexion des lames pour des pressurisations externes importantes. La perte de puissance mesurée pour des vitesses rotor faibles (< 11 m/s à 1300 tr/min) diminue lorsque le différentiel de pression à travers le joint augmente. Pour une vitesse rotor constante, une baisse significative du couple résistant (et de la perte de puissance) est observée lorsque la pression d'entrée augmente, démontrant ainsi qu'un film de gaz sépare le rotor des plaques. Une température de fonctionnement constante (~24^{o}C) à l'interface entre le rotor et le joint lors des tests avec rotation de l'axe indique également l'absence de contact intermittent entre le rotor et les plaques du joint. Les mesures d'étanchéité à température ambiante (25^{o}C) montrent une meilleure étanchéité avec une réduction des fuites d'environ 36%, en comparaison à des joints à brosses avec plaques de première génération. La distance dent/axe équivalente calculée pour le HBS (~50 μm) est faible en proportion de celles obtenues avec des joints labyrinthes à une dent. Les développements de la technologie des joints à brosses permettrant d'augmenter l'efficacité des turbines à gaz tout en améliorant la stabilité du moteur et de réduire les vibrations.

1. Introduction

Brush seals improve power and efficiency in high performance turbomachinery by overcoming the limitations associated to labyrinth seals [1,2,3]. Experiments and field tests demonstrate that brush seals can considerably reduce parasitic leakage [4,5,6]; while improving rotordynamic stability and reducing excessive engine vibrations [3,7]. Currently, the primary limitations of conventional brush seals is their inability to withstand high pressure differentials, excessive bristle wear [5], and localized heat generation during shaft rotation [8]. All of these factors can damage the brush seal permanently and increase its leakage. In addition, conventional brush seals cannot accommodate shaft rotation in both directions, an issue of importance in some aircraft gas turbines.

Justak [9] introduces the 1st generation shoed-brush seal (SBS), see Figure 1. This improved brush seal design accommodates shaft rotation in both directions at the same time it eliminates bristle tip wear, pad/rotor contact and thermal distortions by means of a hydrodynamic gas film lifting the pads as the rotor spins. Delgado and San Andrés [10] present a sound identification method to extract structural stiffness and damping characteristics from a 153 mm diameter shoed-brush seal using single frequency shaker loads on a controlled motion test rig and without shaft rotation. A structural loss factor and a dry friction coefficient represent the energy dissipated by the bristles and the dry friction interaction of the brush seal bristles rubbing against each other, respectively.

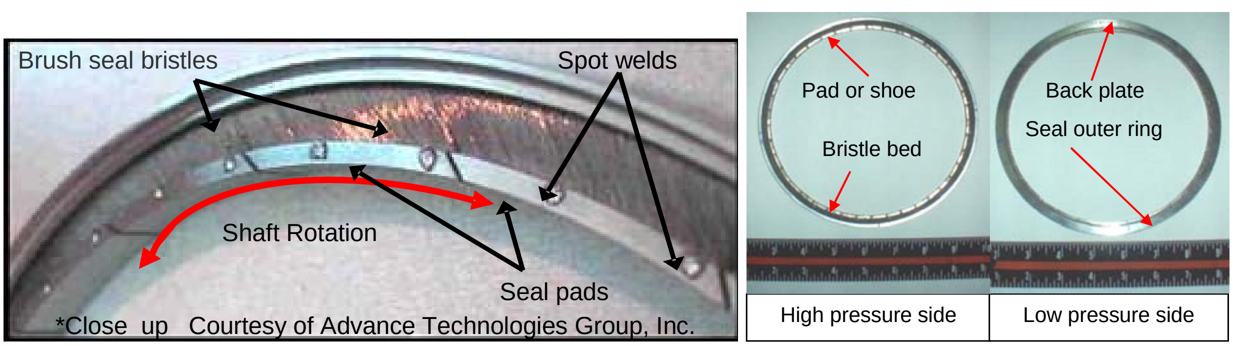


Figure 1. Close up view of shoed-brush seal (SBS) and back and front sides

Delgado *et al.* [11] present a comprehensive analysis for predicting the rotordynamic force coefficients in a SBS. The physical model couples the gas film hydrodynamic forces generated in the thin gap between the rotor and a shoe and the structural characteristics (stiffness and damping) from the bristle pack underneath. Predictions show that the operating gas film clearance and pressure differential do not affect significantly the shoed-brush seal force coefficients. The analysis does not account for the variation of the frictional forces acting within the bristle pack due to increasing pressure differentials.

Justak [12] introduces the hybrid brush seal (HBS), a 2nd generation shoed-brush seal, shown in Figure 2. In a HBS, the arcuate pads connect to the seal casing through EDM-webs. The novel construction eliminates reliability issues associated to the originally used spot-welded connections. More importantly, the thin beam connections (webs) provide a high axial stiffness while maintaining a low radial stiffness; thus reducing pad and rotor wear and secondary flow (leakage), by eliminating pad pitching motions caused by a high pressure differential imposed across the seal. Justak and Crudgington [13] evaluate the performance and design of a hybrid brush seal under static (no-rotation) and with shaft rotation conditions (maximum operating speed 15,000 rpm). The seal is tested in both ambient temperature and a high temperature test rigs, at pressure differentials ranging from 0 to 3 bar, to simulate engine conditions. Performance of the test seal is characterized in terms of a semi-empirical effective clearance parameter (C_E) derived from the measured flow rate (leakage) across the seal, inlet pressure and temperature. A minute increase in effective clearance as a function of increasing pressure represents a minimal increase in leakage across the seal. Additionally, a gradual temperature decrease at the rotor/seal interface and a constant power input during tests with shaft rotation and increasing pressure differential evidence the presence of a gas film separating the seal pads from the rotor surface.

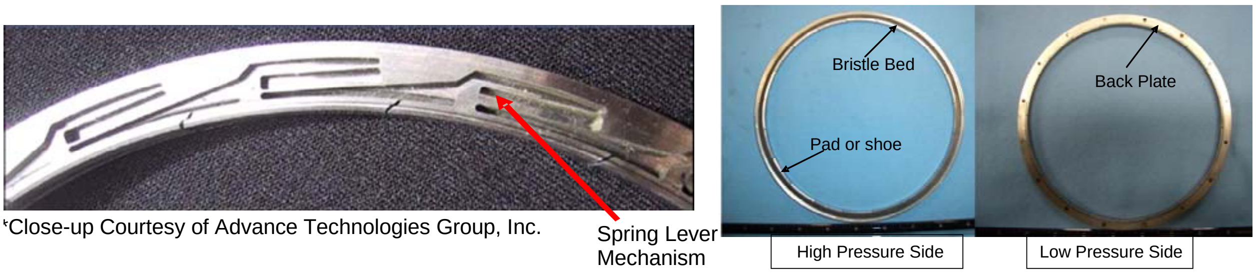


Figure 2. Close up view of hybrid brush seal (HBS) and back and front sides

This paper presents measurements of HBS leakage, drag torque and power loss obtained at low rotor speeds (<1300 rpm) for increasing pressure differentials and at room temperature. The tests aim to characterize the performance of the test seal. In terms of the leakage measurements, the HBS shows a superior leakage performance over the 1st generation SBS, with a nearly 36% leakage reduction over the test pressure range.

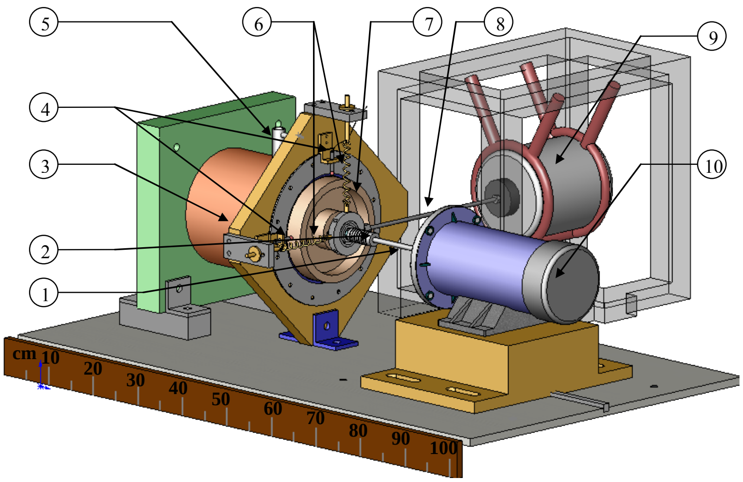
2. Nomenclature

B_w	Brush seal width [mm]
D_j	Rotor diameter [mm]
Ds_i	Brush seal inner diameter [mm]
D_o	Brush seal outer diameter [mm]
g	Gravitational acceleration (9.81 m/s^2)
L	Shaft length (0.284 m)
P_{choke}	Ratio of static pressure to stagnation pressure at Mach 1
P_d	Absolute discharge pressure [Pa]
P_s	Absolute supply pressure [Pa]
P_r	Pressure ratio (P_s/P_d)
R	Specific gas constant [J/ kg.K]
R_i	Radial interference between rotor and seal [mm]
T_u	Upstream (supply) temperature [ºC]
c_E	Effective clearance [mm]
$\dot{m}$	Mass flow rate [g/s]
α	Bristle lay angle [degrees]
φ	Equivalent flow function
γ	Specific heat ratio
ω_s	Rotor speed [rad/s]

3. Description of Test Rig and Hybrid Brush Seal (HBS)

Figure 3 depicts the rotordynamic test rig and instrumentation for hybrid brush seal evaluation. A slender steel shaft is affixed to the base of a cylindrical steel vessel via two taper roller bearings and the free end of the shaft holds a steel disk. A DC motor (746 W) drives the overhang disk/shaft assembly at the shaft free end through a flexible coupling. The test brush seal (166.4 mm in diameter at the pads circumference) is secured atop of the vessel with a retainer ring. The seal assembly nominal interference fit with the disk is 0.381 mm (0.015 in). Two eddy current sensors, 90^{o} apart, are secured on the front plate of the cylindrical vessel and face the outer diameter of the steel disk. The sensors record the disk displacements along two orthogonal directions in the vertical plane.

A slender rod (stinger) connects an electromagnetic shaker to the free end of the shaft. The magnitude of the periodic load is controlled with a piezoelectric load cell fastened at one end of the stinger. Two soft springs located at the drive end of the shaft, in the vertical and horizontal directions, allow centering the rotor free end respect to the test seal. The springs connect to the shaft through a ball bearing enclosed within an aluminum housing. A compressed air line connected to the cylindrical housing is instrumented with multiple pressure taps, a turbine flow meter, a static pressure transducer and thermocouples. Table 1 details the dimensions and material properties for the test hybrid brush seal and Table 2 shows the air flow conditions for the leakage measurements obtained at ambient temperature T_u. A turbine flow meter (± 0.2 SCFM) and a strain gauge pressure sensor (± 0.3 kPa) measure flow rate across the seal and upstream (supply) pressure P_s, respectively. The discharge pressure P_d is ambient (1 bar).



1	Quill shaft
2	Flexible coupling
3	Pressurization vessel
4	Eddy current sensors
5	Supply pressure inlet
6	Soft centering springs
7	Rotor
8	Stinger
9	Electromagnetic shaker
10	DC motor

Figure 3. Rotordynamic test rig for a hybrid brush seal (HBS)

Physical Properties	**Magnitude**
Rotor diameter, D_j	167.1 mm
Brush seal (pads) inner diameter, Ds_i	166.4 mm
Brush seal (retainer) outer diameter, D_o	183.1 mm
Brush seal width, B_w	8.53 mm
Radial Interference between rotor and seal, R_i	0.381 mm
Number of pads	20
Width of pads	7.23 mm
Bristle lay angle, α	45 degrees
Bristle modulus of elasticity, E	22.48x10^{5} bar
Bristle density (circumference)	850 bristle/cm

Table 1: Dimensions and material properties of test hybrid brush seal

Fluid: Air	**Magnitude**
Absolute supply pressure (upstream), P_s	1.01 to 3.04 bar
Absolute discharge pressure (downstream), P_d	1.01 bar
Temperature upstream, T_u	22 to 25.5 ^{o}C
Specific gas constant, R	0.287 kJ/kg. K

Table 2: Flow conditions for leakage measurements in a hybrid brush seal

4. Power loss and drag torque measurements for a hybrid brush seal (HBS) at low rotor speeds

Rotating tests to characterize the hybrid brush seal performance in terms of drag power and torque are conducted for rotor speeds (ω_s) ranging from 400 to 1,300 rpm (surface speed ~11 m/s), and increasing supply pressure (P_s) to discharge pressure (P_d) ratios, $P_r = P_s / P_d$. In the case of no external pressurization, i.e. $P_r = 1.0$, no air flows across the seal as the rotor spins. The seal power loss is derived from the drive motor electric power, i.e. voltage *x* current, at each test rotor speed. Drag torque is estimated by dividing the electrical power by the rotational speed (ω_s). Prior to installing the HBS, a baseline torque is estimated to account for frictional drag arising from the roller bearings located at the base of the cylindrical vessel (see Fig. 4). In addition, with rotor spinning, a baseline drive power without the hybrid brush seal in place is recorded. This power accounts for windage drag on the large steel disk.

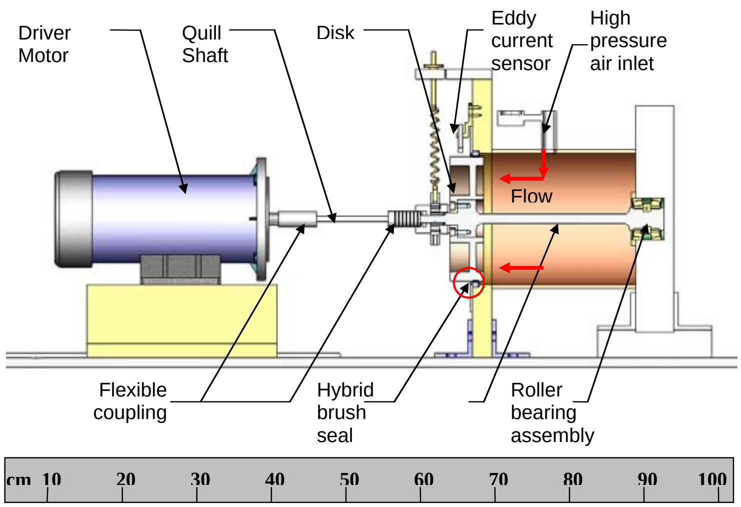


Figure 4. Cut view of hybrid brush seal rotordynamic test rig

Figures 5 and 6 show the HBS power loss and drag torque versus rotational speed, respectively, for increasing supply to discharge pressure ratios, $P_r = P_s / P_d$. The baseline motor drag and disk windage power are subtracted from the power measurements. Maximum power loss (~350 W) occurs at $P_r = 1.0$ due to the high contact forces at the rotor/seal interface and remains approximately constant over the test speed range. Power losses and drag torque for a HBS decrease about 90% and 50%, respectively, from $P_r = 1.0$ to 1.7, over the test speed range. Static break away torque measured at 0 rpm, i.e. torque required to initiate shaft rotation, drops significantly from $P_r = 1.0$ to 1.7 evidencing hydrostatic liftoff that causes the seal pads to separate from the rotor surface prior to actual shaft rotation. The hydrostatic liftoff effect is further enhanced by the hydrodynamic film action created once the rotor spins. The results denote a significant reduction in the *contact* forces between seal pads and rotor surface that should be apparent during rotor startup and actual steady operation. Hence, rotor and pads' seal wear are virtually eliminated. It is thought that the hydrodynamic liftoff effect will become more noticeable at higher rotational speeds. Additionally, a low constant operating temperature (~24^{o}C) at the rotor/seal interface during the rotating tests with external pressurization evidence the absence of intermittent contact between the seal pads and rotor. Recall that power loss and drag torque estimations, depicted in Figures 5 and 6, correspond to a HBS with a radial interference of 0.381 mm (0.015 in) with its rotor.

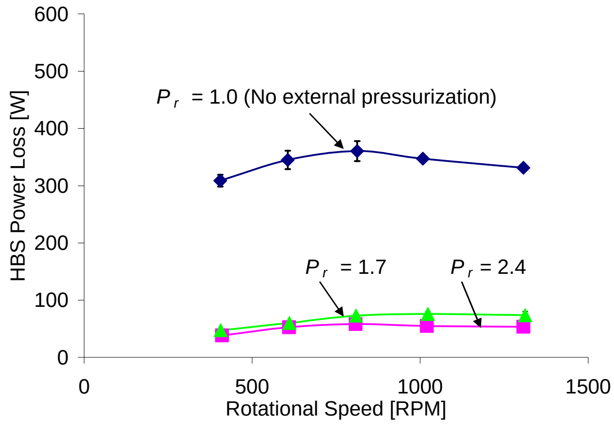


Figure 5. Test power loss for hybrid brush seal (HBS) versus rotational speed for increasing supply to discharge pressure ratios.

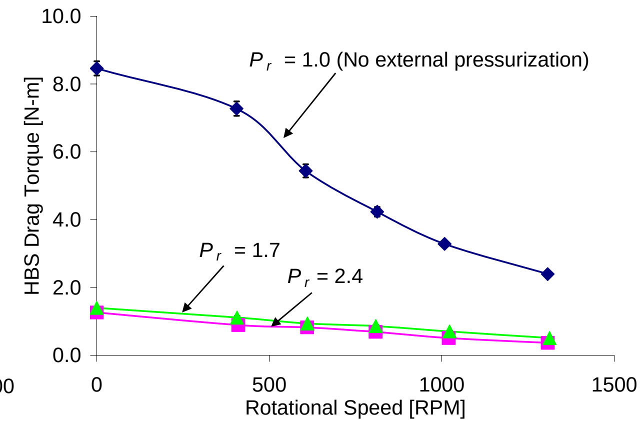


Figure 6. Drag torque for hybrid brush seal (HBS) versus rotational speed for increasing supply to discharge pressure ratios

5. Leakage characteristics of a hybrid brush seal under static (non-rotating) conditions and at low rotor speeds

Leakage (flow rate) in the hybrid brush seal is measured for increasing supply pressures (P_s up to 3 bar) at ambient temperature (~25°C), under static (non-rotating) conditions and at low rotor speeds (<1,300 rpm). Figure 7 depicts the measured mass flow rate versus pressure ratio ($P_r = P_s/P_d$) under static conditions (i.e. without shaft rotation) for the hybrid brush seal (HBS) and a 1st generation shoed-brush seal (SBS) tested at identical feed pressure conditions [10]. Note, that both test seals have an interference fit to the rotor. The HBS shows a better sealing performance over its predecessor, reducing overall leakage by about 36% over the test pressure range. This sealing improvement is due to the pads larger axial stiffness, which improves the pads (shoes) ability to resist axial and twisting motions induced by high pressure differentials. Error bars in the graph denote a rather low uncertainty, less than 5.0% of the leakage measurement.

In terms of HBS leakage versus rotational speed, tests results in Figure 8 indicate the flow rate for each supply pressure (P_s) remains nearly constant, i.e. independent of rotor speed. The test results indicate that HBS leakage depends mainly on the pressure differential across the seal. Most air flows through the minute air gap formed between the seal pads and rotor (i.e. hydrostatic liftoff) induced primarily by the increasing pressure differential across the seal rather than shaft rotation.

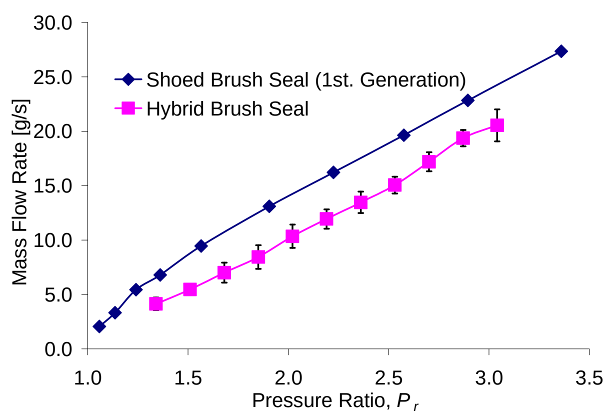


Figure 7. Test flow rates for 1st generation shoed-brush seal (SBS) and hybrid brush seal (HBS) versus supply to discharge pressure ratio under static conditions (no shaft rotation)

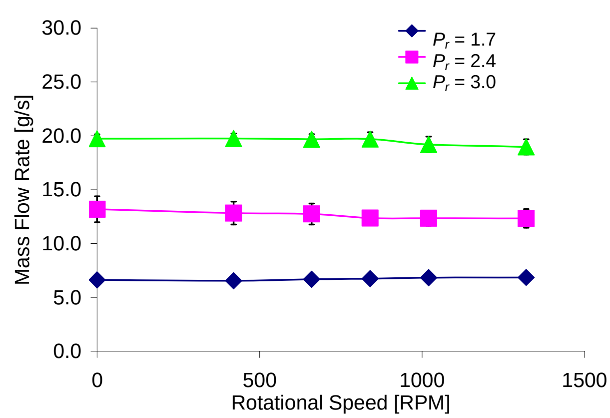


Figure 8. Test flow rate for hybrid brush seal (HBS) versus rotor speed for increasing supply to discharge pressure ratios

Labyrinth seals remain as the standard seal element for rotating machinery that requires restricting gas flow from high to low pressure regions. Due to this circumstance, a brush seal manufacturer [9], compares leakage performance of brush seals to that of a labyrinth seal in terms of an *effective clearance*, C_E, corresponding to an equivalent film thickness in a "one-sharp tooth" labyrinth seal. The definition for effective clearance, C_E [9], is

$$c_E = \dot{m}\frac{\sqrt{T+273.15}}{P_s \pi D_j \varphi} \tag{1}$$

where

$$\varphi = \begin{cases} \sqrt{\dfrac{\gamma}{R}}\sqrt{\dfrac{2}{\gamma-1}P_r^{\frac{-(\gamma+1)}{\gamma}}\left(P_r^{\frac{(\gamma-1)}{\gamma}}-1\right)} & P_r \le P_{choke} \\ \sqrt{\dfrac{\gamma}{R}}\sqrt{\dfrac{2}{\gamma+1}\left(\dfrac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}}} & P_r > P_{choke} \end{cases} \tag{2}$$

γ =1.4 denotes the gas (air) ratio of specific heats, and $P_{choke} = \left(\dfrac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$

Figure 9 depicts the calculated effective clearance (C_E) from the leakage data recorded for the shoed-brush seal (SBS) and hybrid brush seal (HBS) at static conditions (i.e. no rotation). Additionally, the HBS effective clearance is shown for two rotor speeds, 600 and 1300 rpm. The derived effective clearances for the SBS and HBS are a fraction of a typical labyrinth seal clearance. For non-rotating conditions, the HBS shows a lower effective clearance, ~ 30% in average, than that of the SBS over the test pressure range. The HBS effective clearances at 600 and 1,300 rpm are similar to the effective clearance predicted for no shaft rotation condition. The largest effective clearance is just over 0.050 mm at the highest test pressure ratio, P_r = 3.0. Typical operating clearances (diametral) for a labyrinth seal range from 1.0 to 2.0 mm [14], about 20 times larger than the calculated effective clearance for a HBS. Also note that in labyrinth seals, issues like rotor thermal expansion and continuous rub against knife edges minimize their sealing performance over time.

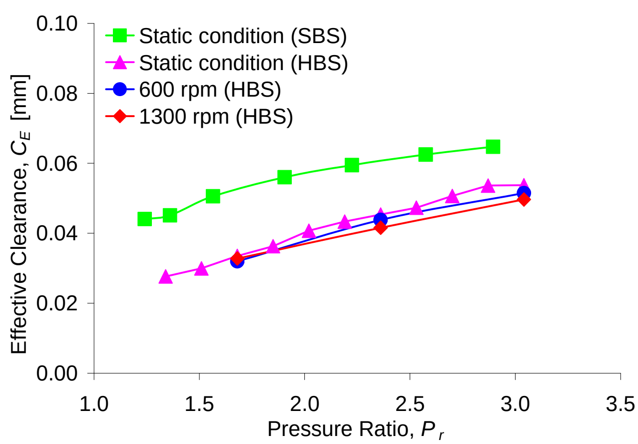


Figure 9. Calculated effective clearance for shoed-brush seal (SBS) and hybrid brush seal (HBS) versus supply pressure to discharge pressure ratio at static condition (no rotation) and selected shaft speeds (600 and 1,300 RPM)

6. Conclusions

Gas turbines typically use labyrinth seals to restrict leakage between stages. Labyrinth seals inevitably wear, thus increasing their leakage and at times inducing harmful rotordynamic instability. Brush seals also restrict effectively secondary flows, but since the seal bristles are in contact with the rotor, they also wear and can induce severe shaft thermal bow. Hybrid brush seals (HBS) resolve poor reliability resulting from bristle tip wear while also allowing for reverse shaft rotation operation. A hybrid brush seal incorporates pads contacting the shaft; and which under rotor spinning; lift off due to the generation of hydrodynamic gas film pressure. The gas film preventing intermittent contact; lowers the shaft operating temperature, reduces thermal distortions, and even eliminates bristles wear. This paper presents laboratory experiments aiming to quantify the leakage and power loss performance of a hybrid brush seal. Measurements show the advantages of a hybrid brush seal over conventional brush seals and a 1st generation shoed-brush seal (SBS), overcoming main deficiencies associated with rotor and seal wear and low pressure differential capability.

Flow rate (leakage) measurements demonstrate a better sealing performance of the HBS with respect to a 1st generation shoed-brush seal (SBS). This sealing improvement is due to an increase in pads' support axial stiffness, provided by the EDM-webs connecting the shoes to the seal casing, and which effectively prevents pads pitching motions caused by pressure differentials across the seal. Furthermore, the bristle bed in the HBS is now a secondary seal to the gas film riding pad element. In addition, the empirically derived effective clearances for the SBS and HBS are a fraction of a typical labyrinth seal clearance, with the HBS effective clearance being lower (~ 30% in average) than that of the SBS effective clearance.

Power loss and drag torque measurements performed on a HBS at low rotor speeds (i.e. a maximum surface speed of 11 m/s at 1,300 rpm) reveal a surprising relation with respect to the increasing pressure differentials imposed on the seal. Maximum power losses (~350 W) occur without external air pressurization. Power losses decrease by approximately 90% over the test speed range (400 to 1300 rpm) as the seal is pressurized, evidencing the generation of a hydrodynamic gas film separating the seal pads from the rotor surface. Most importantly, the break away torque drops by more than 50% as the HBS is pressurized, thus indicating that seal pads liftoff prior to actual shaft rotation. The hydrostatic liftoff effect is further enhanced by the hydrodynamic action caused by rotor spin. Thus, when pressurized, the HBS drag torque is lowest at rotor startup and steady operating conditions, a finding in direct opposition to that of typical brush seals. Power loss and drag torque measured under external pressurization, i.e. P_r = 1.7 and 2.4, show no significant variation with increasing rotor speeds. Additionally, a low constant temperature (~24°C) at the rotor/seal interface also indicates the absence of intermittent contact between the rotor and hybrid brush seal pads.

Currently, a model for identifying rotordynamic force coefficients of a hybrid brush seal as functions of rotor speed and excitation frequency is under scrutiny. Model results will validate a rotordynamic predictive code [11]. A high temperature experimental facility under construction will characterize HBS performance at higher shaft speeds and feed pressures, thus simulating actual engine conditions. Additional testing will further determine the reliable performance of the hybrid brush seal and provide predictive tools facilitating the design, application and improvement of this promising sealing technology.

7. References

[1] Xi J. and Rhode D. L. (2006) Rotordynamics of Impeller Eye Seals with Wear-Damaged Teeth in Centrifugal Compressors. *Tribol. Trans,* **49**(3), pp. 328-337.

[2] Benkert H. and Wachter J. (1980) Flow Induced Spring Coefficients of Labyrinth Seals for Applications in Tubomachinery. NASA CP2133.

[3] Childs D.W. and Vance J.M. (1997) Annular Gas Seals and Rotordynamics of Compressors and Turbines. *Proc. 26th Turbomachinery Symposium,* pp.201-219.

[4] Ferguson J.G. (1988) Brushes as High Performance Gas Turbine Seals. ASME Paper No. 88-GT-182.

[5] Proctor M.E. and Delgado I.R. (2004) Leakage and Power Loss Tests Results for Competing Turbine Engine Seals. NASA TM—2004-213049.

[6] Chupp R.E. and Dowler C.A. (1993) Performance Characteristics of Brush Seals for Limited Life Engines. *Trans. of the ASME,* **115**, pp.390-396.

[7] Conner J.K. and Childs D. (1993) Rotordynamic Coefficient Test Results for a Four-Stage Brush Seal. *AIAA J. Prop. Power,* **9**, pp.113-119.

[8] Dogu Y. and Aksit M.F. (2005) Brush Seal Temperature Distribution Analysis. ASME Paper No. GT2005-69120.
[9] Justak J.F. (2002) Hybrid Brush Seal Capable of Reverse Rotation Technical Report. *Navy SBIR Phase II Project*, Advanced Turbomachinery Solutions, Miami, FL., April.
[10] Delgado A. and San Andrés L. (2005) Measurements of Leakage, Structural Stiffness and Energy Dissipation Parameters in a Shoed Brush Seal. *Sealing Technology,* **12**, pp. 7-10.
[11] Delgado A. San Andrés L. and Justak J.F. (2004) Analysis of Performance and Rotordynamic Force Coefficients of Brush Seals with Reverse Rotation Ability. ASME Paper No.GT2004-53614.
[12] Justak J.F. (2002) Robust Hydrodynamic Brush Seal. U.S. Patent No. 6,428,009.
[13] Justak J.F. and Crudgington P.F. (2006) Evaluation of a film riding hybrid seal. AIAA Paper No. 2006-4932.
[14] Stephen D. and Hogg S.I. (2003) Development of Brush Seal Technology for Steam Turbine Retrofit Applications. ASME Paper No.IJPGC2003-40103.

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