

**PRESSURE ACTIVATED LEAF SEAL
TECHNOLOGY READINESS TESTING**

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ABSTRACT

This paper continues the evaluation of Pressure Actuated Leaf Seals (PALS) technology readiness for shaft and shroud sealing in power generation and aerospace applications. Seal designs tested are prototypical and constructed using processes appropriate for volume production. Results include both static and dynamic seal leakage measurements running against a 5.1in (129.54mm) diameter smooth surface test rotor and another that simulates sealing against turbine blade shrouds. A further test was undertaken using a 2D static rig that determined acoustic noise experienced during testing was attributed to leaves vibrating at their natural frequency as a result of inter-leaf gaps. The dynamic simulated shroud test includes steps, duplicating small discontinuities of adjacent shroud sealing surfaces and slots to inject air radially under the seal leaves as may occur between shrouds on blades with a high degree of reaction. Consistent seal performance over 15 hours confirms suitability for turbine blade tip applications. Controlled deflection of PALS leaves with operating differential pressure is effective for startup rub avoidance in service as well as conformal wear-in sizing of leaf tips with the rotor. Tested leaf tip wear-in of approximately 0.010in (0.25mm) against rotor discs without hard-face coating, shows potential to eliminate seal misalignment and run-out contributions to operating seal

clearance. PALS design features prevent further rubbing contact with the operating rotor after initial wear-in sizing thereby sustaining a small effective seal clearance and prospects for long seal life. Measurements of rotor surface wear tracks from the wear-in process and endurance runs are included as well as rotor and leaf tip photos. Test results support the technology readiness of the PALS concept as a viable, robust, low leakage dynamic seal for select commercial application.

INTRODUCTION

Shaft sealing is critical to power generation and aerospace turbine performance. Chupp et al ^[1] reviews conventional dynamic seals to control clearances and advanced seal designs under development at the time. Since then, the PALS design ^[2, 3, 4 & 5] has also advanced and is now approaching commercial application for dynamic sealing in turbomachinery. The PALS concept was introduced in AIAA-2005-3985 and initial test results were presented in AIAA-2009-5167. The PALS design, illustrated in Figure 1, thin seal leaves are elastically deflected in an axial direction by system differential pressure to close with the shaft at near full speed, preferably above critical speeds, and minimum operating pressure. Once actuated the support member restricts further seal closure to maintain a small non-contacting clearance throughout the operating

pressure range. This sealing concept provides the designer with means to avoid rubs by providing sufficient seal clearance during startup and shutdown to keep away from normal rotor dynamic transients but preserve low seal leakage at operating conditions. Seal function is independent of direction of rotation and rotor speed.

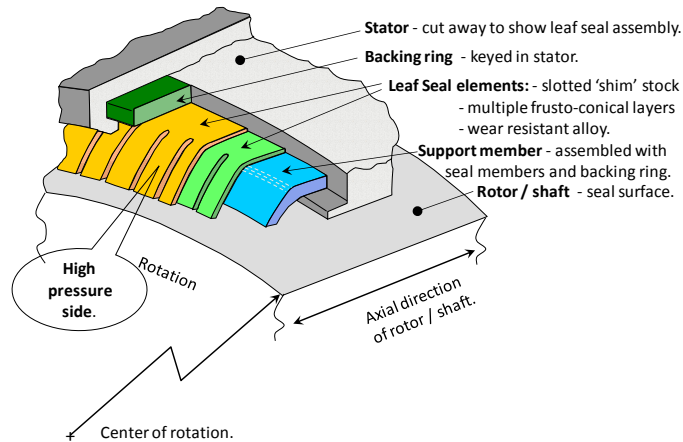


Figure 1: Pressure Activated Leaf Assembly

Labyrinth seals are the most widely used shaft packing in turbomachinery because of relatively low cost and high speed, high pressure and high temperature capability. However, operating experience shows they are often rubbed in service and cause significant loss of performance. For example, Cofer et al [6], reports the normal design clearance for internal mid-span packing of a 500 MW reheat steam turbine, with opposed-flow HP and IP sections in a single casing, is 0.015in (0.4mm). But upon inspection, after five years of operation, packing clearance is "typically opened up to 0.060in (1.5mm)". Regarding gas turbine high pressure packing (HPP) labyrinth seals Johnston [7] notes "In practice, most operating units have HPP clearances significantly higher (0.020 to 0.060in) (0.5 to 1.5mm) than nominal. This increased labyrinth seal clearance results in considerable unit performance loss. For an MS7001E unit, a rub of 0.020in (0.5mm) on labyrinth seal teeth equates to at least 1.0% loss in unit performance." A significant objective of PALS development is rub avoidance. Test results reported in this paper show the potential of the PALS design to avoid damaging startup rubs.

Contributing to labyrinth seal design clearance and magnitude of rubs experienced is the manufacturing tolerance of seal components and their mounting surfaces within a turbine, i.e. the run-out of each seal interface mounting surface as well as alignment of the seal ID with respect to the rotor OD seal surface. All of these eccentricities and thermal growth must be addressed to avoid fixed tooth seal rubs. This is a daunting task and typically additional seal clearance margin is added to calculated values to accommodate undefined part distortion and relative motion between shaft and stator. The wear-in of as built and assembled PALS leaf tips into close conformity with the rotor of an operating unit is another important objective of

PALS development to minimize seal leakage and maximize performance. An aim of this paper is to confirm that benign PALS leaf tip wear-in is feasible when engaging either a smooth rotor seal surface or a discontinuous seal surface that simulates a shrouded turbine blade.

NOMENCLATURE

2D – Two dimensional
 clr – Clearance
 EDM – Electrical discharge machining
 FEA – Finite element analysis
 HCF – High cycle fatigue
 hgt – Height
 HP – High pressure
 HPP – High pressure packing
 ID – Inside diameter
 IP – Intermediate pressure
 LP – Low Pressure
 Ni-Cr-Mo – Nickel – Chromium – Molybdenum
 OD – Outside diameter
 PALS – Pressure activated leaf seal
 p-p – Peak to peak
 PLC – Programmable logic controller

TEST SEAL DESIGN

This seal design is prototypical in cross section, functional characteristics and assembly to demonstrate PALS technology readiness for commercial power generation or aerospace applications. A seal design envelope of 0.55in (14mm) radial height and 0.8in (20mm) axial length was specified to have application potential. The test seal, shown in Figure 2, includes illustration of possible assembly in an application stator.

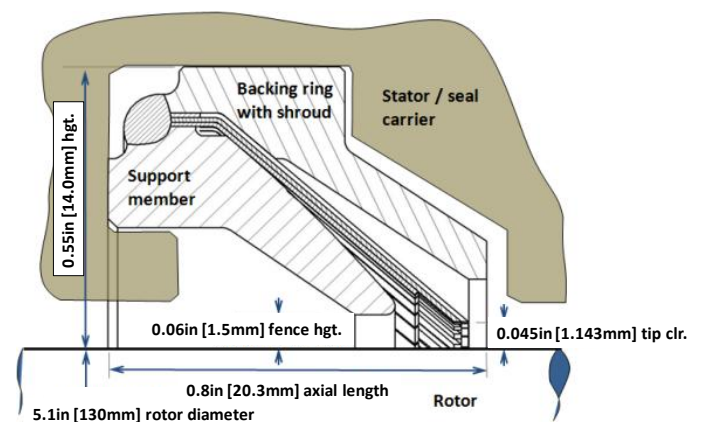


Figure 2: PALS prototype Seal Section

An installed leaf tip clearance of 0.045in (1.14mm) was chosen to illustrate PALS large cold clearance assembly to avoid seal rubs during startup and subsequent closure with the rotor as pressure across the seal increases to 80psid (550kPa), the test seal design pressure. Seal leaves are designed for desired deflection by selection of their length, thickness, number of leaf layers, support member radius, angle with respect to rotor axis

and fence height along with mechanical properties of the seal leaf material. Haynes 25 material in the annealed condition was chosen for the seal leaves of this design because of its wear qualities, suitability for use in other high temperature seals and availability in 0.010in (0.025mm) thickness. There are 2 layers of sealing leaves and a shorter, damping leaf, beneath them that is in register with bottom seal leaf as shown in Figure 3. Space between adjacent bottom seal leaf tips assures that seal pressure drop occurs across top seal leaves. As leaves are pressure loaded they elastically deflect into compliance with the support member contour. Bending stress is held well below material yield stress for long cyclic life without high cycle fatigue. Leaf deflection and stress are typically analyzed using beams in bending elastic analysis. FEA was also performed for this design. The difference in natural frequency of the shorter damper leaves mitigates occurrence of leaf oscillation. Fence height is 0.06in (1.5mm) in this design.

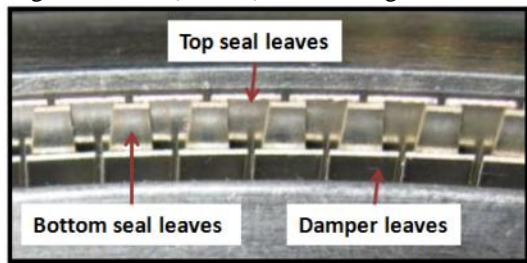


Figure 3: Leaf Tip Detail, LP Side

There are 120 seal leaves in each leaf layer that are cut by wire EDM from sheet. Each leaf layer strip is then bent to the design leaf angle plus some interference of upper leaf tips with lower leaf layers to assure intimate leaf contact as strips are laid up about the cylindrical portion of the support member. When assembled with the backing ring, parts are joined by welding as shown in Figure 2. The final fabrication step is a wire EDM trim of the seal leaves while mechanically constrained in the fully deflected position. This provides a precision seal ID for desired clearance or 'wear-in' with respect to the rotor. The finished test seal is shown in Figure 4. Prominent in the picture is the shroud portion of the backing ring that protects seal leaves from handling damage.



Figure 4: PALS 5.1in (130mm) Diameter Test Seal

An alternate PALS configuration utilizes seal leaves that are cut from a sheet and bent out of a radial plane to assemble tangent with the support ring radius. A cross-section of this PALS configuration in Figure 5 shows mechanical assembly rather than welding. It accommodates those applications where there is limited axial seal envelope length but space for a taller seal. Seals of this configuration have been fabricated and tested with seal leakage comparable to the results presented below.

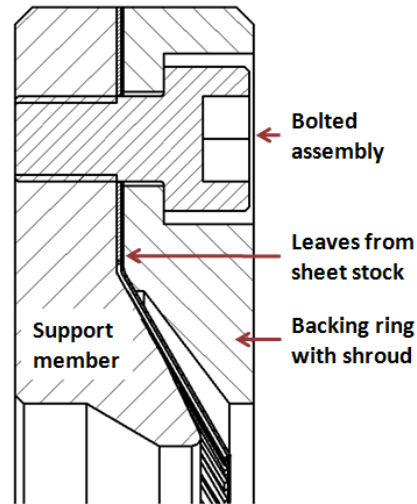


Figure 5: PALS from Sheet Stock

SEAL TEST FACILITY AND OPERATION

The dynamic test facility and air feed system utilized to test the PALS design were first commissioned in 1987 and were described in a paper by Flower in 1990^[8]. The test facilities have evolved greatly over the years and now incorporate an extensive hot test facility as described by Crudginton^[9].

The facility has modern instrumentation, data acquisition equipment, high speed photography and is all controlled, from the separate control room, via PLC's to enable cyclic running to easily be performed.

The rigs have been used in 3 different ways for this paper. The smooth rotor testing is described fully by Flower^[8]. This rig tests a single 5.1in (129.54mm) diameter seal at speeds up to 21,000rpm and pressure drops up to 120psi (827.6kPa). The rig operates at ambient temperature with ambient downstream conditions. Rotors and seals are very easy to change and the rig has the capability to radially offset the seal relative to the disc a predetermined amount during testing. Tests are very quick and easy to run with the seal and rotor typically being exposed for inspection within 5 minutes from the end of a test.

The noise investigation testing utilized the air feed system only of the rig together with the high and low speed data acquisition systems. A unique test fixture was designed and made for this test work.

The set up for the Simulated Shrouded Turbine Blade Testing was a little more complex as this requires two completely independent pressure regulated air feed systems into the rig as previously undertaken for Turnquist et al^[10]. The first controls the seals pressure and the second is fed into the centre of the disk to create a radially outwards flow through the slots in the disk. Both feed systems enable highly accurate control of the pressure and each system has its own flow-metering. This setup still operates at speeds of up to 21,000rpm and all testing is still performed at ambient temperature with ambient conditions downstream of the seal. Even with this added complexity the accessibility to the rig is still excellent with both seal and rotor typically being exposed for viewing and inspection within 5 minutes from the end of a test.

SMOOTH ROTOR SEAL TESTING

The PALS test design installation in the 5.1in (130mm) test rig is illustrated in Figure 6 and Figure 7 photos. Static and dynamic tests were conducted to evaluate sealing performance against a conventional smooth rotor surface using rotor disks of various sizes. Rotor disk material is uncoated Aubert & Duval 819B steel, a Ni-Cr-Mo alloy. The PALS LP side exhausts to atmosphere from the test rig.

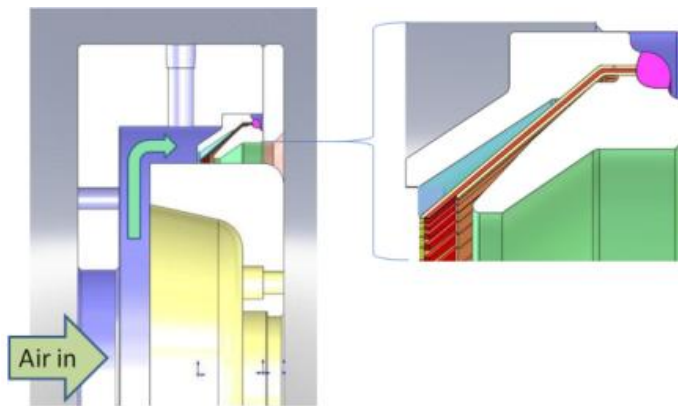


Figure 6: Illustration of PALS in 5.1in (130mm) test rig

Initial tests were static, without rotor rotation, to evaluate PALS closure with pressure and seal leakage with rotor disks of different diameter ranging from interference with deflected PALS leaves to a clearance of 0.012in (0.3mm). The test seal bore diameter with seal leaves mechanically compressed was 5.105in (130mm) for this test. The un-deflected leaf bore was 5.185in (132mm). Static seal leakage verses rotor disk size is plotted in Figure 8 up to 80psi (552kPa). Flow measurements, used for calculation of effective clearance, were not stable below ~30psi (207kPa) and there was a loud tone. The one data point in Figure 8 that is above the line formed by the remaining 30 psi points is attributed to increased effective clearance due to leaf vibration.

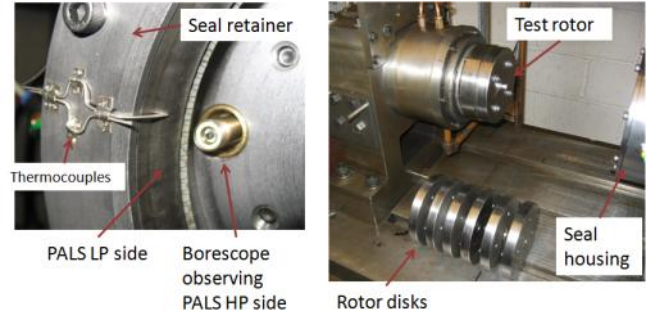


Figure 7: Photos of PALS installation in the 5.1in (130mm) test rig

Audible seal leaf oscillation had occurred in prior PALS development^[3], but was alleviated then with good inter-leaf contact and use of an underlying damping leaf. This test seal includes a damper leaf and was assembled with care so leaf oscillation noise was not anticipated. Inspection of test seal leaf tips, however, found areas where top seal leaves were not in

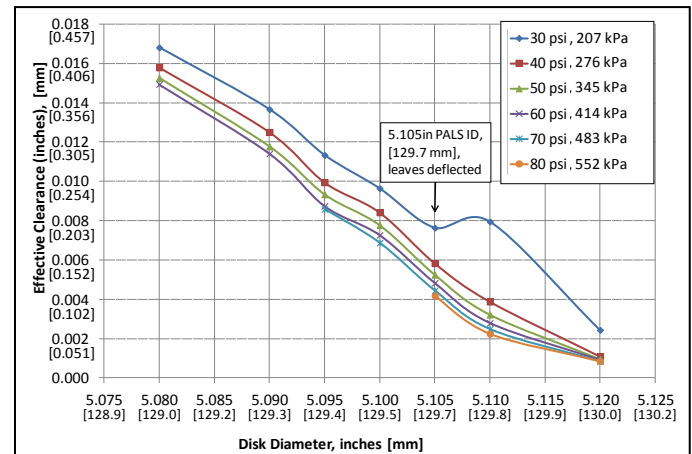


Figure 8: PALS Clearance with Various Disk Sizes

contact with underlying leaves, Figure 9. This finding was recognized as the probable cause of the noise. However, a decision was taken to build a small 2D test rig and thoroughly rule out other possibilities that may be contributing to it. That 2D rig and test program are discussed in the following section. It was undertaken out of concern for prolonged off-design operation at low pressure that could result in high cycle fatigue seal failure.



Figure 9: Test Seal Leaf Tips

Images viewed through a borescope saw the onset of vibration at low pressure with relatively large top leaf tip oscillations at 1160 Hz, the natural frequency of full length

leaves. As pressure increased the magnitude of oscillation decreased and the tone increased to 2000 Hz indicative of shorter free leaf length. Nearing 30 psi (207kPa) the leaves stop vibrating altogether and are quiet. At conclusion of testing this seal was dissected and inspected by dye-penetrant and metallographic section for any evidence of high cycle fatigue. None was found.

Static and dynamic testing in the 5.1in (130mm) test rig continued to evaluate seal performance at higher pressures. Figure 8 shows that PALS can achieve a very tight clearance over a wide range of pressure when disk OD approaches seal ID. Since PALS leaf members are thin, 0.010in (0.25mm) in this case, a primary test objective was to assess the prospect of sizing the seal ID, by ‘wear-in’ engagement with the operating rotor, without hazard to the rotating seal surface or PALS. Doing such a ‘wear-in’ would eliminate seal leakage from the run-out and mis-alignment of seal mounting hardware when seals are designed to avoid shaft contact. PALS pressure actuation of leaf tips from a large inoperative clearance to a small operating clearance provides means of avoiding startup seal rubs caused by rotor dynamic shaft whirl traversing critical speeds coming up to operating conditions. Avoiding startup rubs, a PALS can return to small effective seal clearance established by the initial ‘wear in’.



Figure 10: Leaf Tip Burr - Post Wear-in (View from Bottom)



Figure 11: Leaf Tip Burr - Post Wear-in (View from Top)

The first rub test was conducted at a seal pressure of 50psid (345kPa) while running at 13,500 RPM, (300 Ft/sec) (91.4m/s) rotor speed with an effective seal clearance of 0.007in (0.18mm) and estimated physical leaf clearance of about 0.005in (0.13mm). The seal was abruptly offset 0.010in (0.25mm) for 30 seconds. Borescope observation and video at the rub location show the seal engagement with a brief flash of light and then the stable leaf tip in close proximity to the rotor thereafter. Effective seal clearance increased 0.0007in (0.018mm). Local leaf tip heating was evident and burrs were raised (Figure 10) but the seal was not otherwise affected. The rotor wear track showed only light burnishing. A full 360

degree ‘wear-in’ test was then prepared for by trimming the deflected seal leaves to a diameter of 5.114in (129.9mm) and installation of a 5.133in (130.4 mm) diameter rotor disk in the test rig for a nominal leaf tip ‘wear-in’ of ~0.009inches (0.35mm). At a rotor speed of 15,500RPM, a seal pressure of 30psid (207kPa) was supplied across the seal to displace the leaves toward the rotor. Borescope observation of the resulting ‘wear-in’ rub was a momentary flash of heat and light as before and then the leaf tip remaining in very close running with the rotor. Pressure was raised by steps to 80 psid (552kPa). As pressure was increased there was an occasional flicker of light at the leaf tip – rotor interface that is attributed to burning of loose leaf tip burrs evident in Figure 10 and Figure 11. The rotor wear track was visible but not of any significant depth.

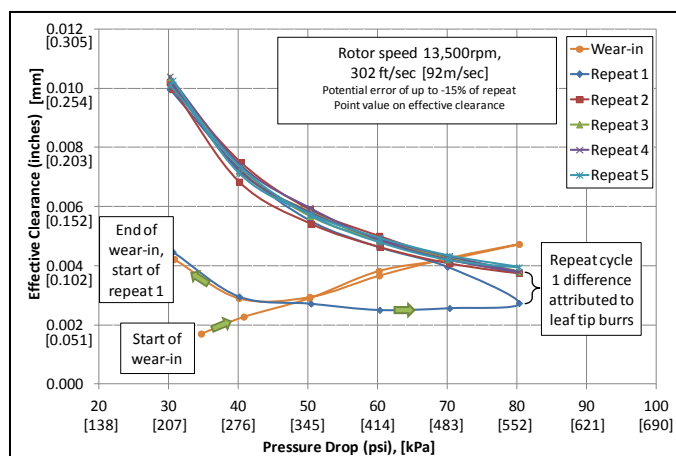


Figure 12: PALS Dynamic Test Clearance During and After Wear-in

Figure 12 is a plot of effective clearance measured during the ‘wear-in’ and 5 repeated dynamic cycles of pressure to 80 psid (552kPa) while maintaining 13,500RPM rotor speed. Irregularities during ‘wear-in’ and the first repeated cycle are attributed to shedding of burrs from seal leaf tips. Effective clearance during the five repeat dynamic pressure cycles track very closely indicating stable and consistent PALS closure with pressure.

Repetitive static tests to 80 psid (552kPa) were performed after the dynamic cycles. That testing extended to lower pressure levels as shown in Figure 13. At seal pressure above 20 psid (138kPa) the effective clearance is essentially unchanged during the 5 pressure cycles. Below that pressure there are differences among the static cycles but airflow measurements were stable and noise was unexpectedly absent. It is believed that leaf tip burrs (Figure 10), may be bridging the gaps to mitigate leaf vibration. Five additional dynamic cycles to a higher pressure of 120psid (827kPa) were performed. Results in Figure 14 show only minor variation in pressure verses effective clearance throughout the test series. The repeatability of PALS closure with pressure was again found to be stable with no evidence of hysteresis.

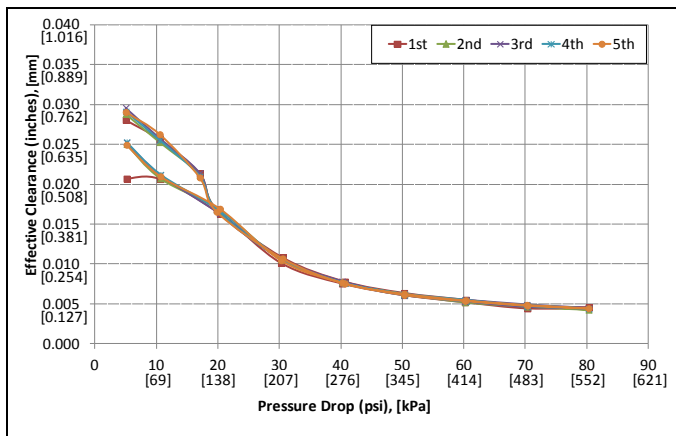


Figure 13: PALS Static Cycling After Wear-in

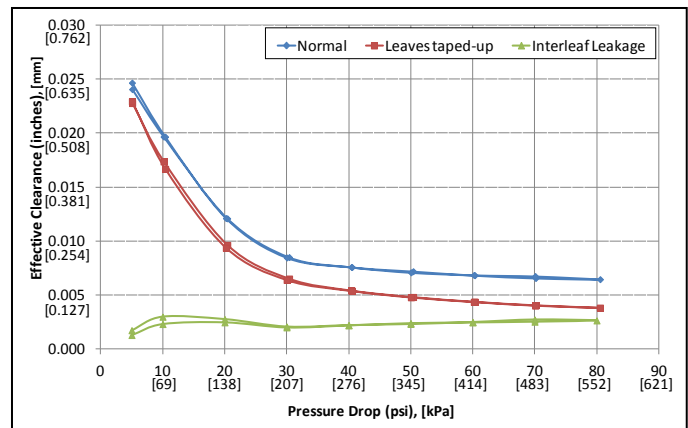


Figure 15: PALS Inter-leaf Leakage Assessment

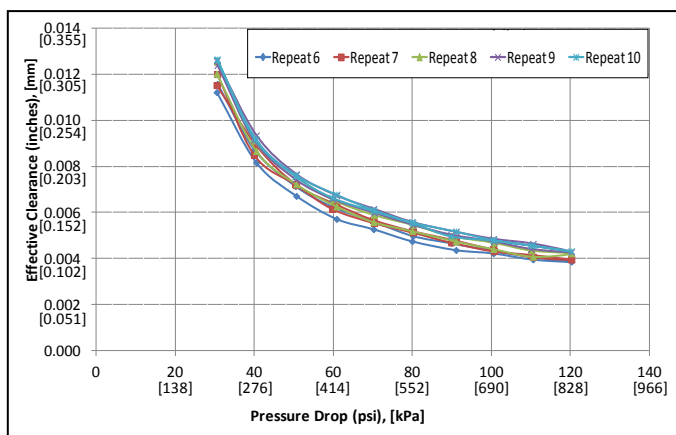


Figure 14: PALS Hi-Pressure Dynamic Cycling Clearance

The minimum effective clearance of 0.004 inches (0.1mm) is the same at 120psid (827kPa) as it was at 80 psid (552kPa) seal pressure. The incremental leaf tip deflection calculated for this seal is 0.001in (0.025mm) when seal pressure is increased to 120psid (827kPa) from 80psid (552kPa). It is worn away by leaf tip engagement with the rotor and results in a larger effective clearance at lower pressure as seen at 80psid (552kPa) in Figure 15. To assess the contribution PALS interleaf leakage to the minimum effective clearance, tests were run after the protective shroud was machined away to expose the seal leaves. The first test established a base line with seal leaves exposed normally. In the second, insulation tape covered the up-stream face of leaves to prevent passage of air between them. The difference in measured effective clearance is attributed of interleaf leakage. Results in Figure 15 show the effective clearance of interleaf leakage at 80 psid (552kPa) to be approximately 0.0025in (0.064mm). The rest of minimum measured effective seal clearance of 0.004in (0.1mm) is attributed to leakage at the seal backing ring and other test rig leaks, run-out of the rotor seal surface and instrument error.

PALS closure observed after ‘wear-in’ with the 5.133in. (130.38mm) diameter disk was compared to that predicted by the FEA model. Using the measured un-deflected seal leaf ID of 5.202in (132.13mm) the calculated FEA seal closure, tracks very closely to measured effective clearance when an adjustment is included for measured interleaf leakage. The favorable comparison (Figure 16) confirms accuracy of design analysis. Acoustic noise was present from 30psi to 0psi when reducing the pressure back to ambient. As a result there is a reduction in effective clearance compared to when applying the pressure. This reduction in effective clearance is due to the oscillation of the leaves causing disturbed flow and irregular pressure transducer readings.

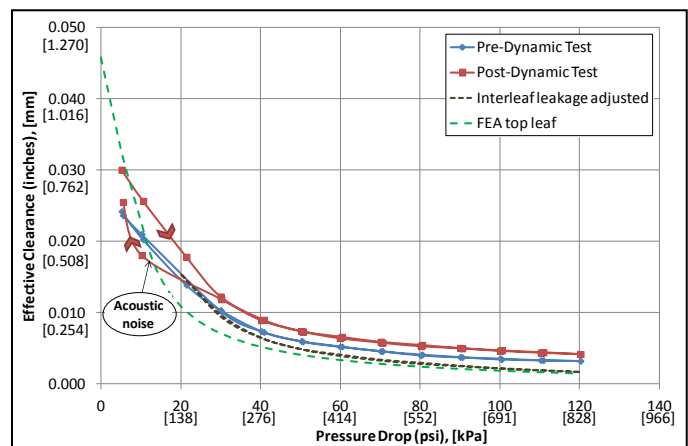


Figure 16: PALS Post Test Closure Analysis

NOISE INVESTIGATION AND 2D LEAF TEST RIG

Leaf vibration resulted in an audible noise at low differential pressure across the seal leaves in early testing. The vibration and noise ceased when the pressure increased above 30 psid (207kPa). Testing of an early generation seal ^[3] also experienced a similar noise that was traced to interleaf gaps.

An investigation was undertaken to identify the cause of the leaf vibration and noise, find a solution and then

demonstrate a design that is free from leaf vibration and noise. It started with the whole team conducting a brain storming activity to list all potential causes. In all, eleven possible causes were identified, Table 1. A test plan was then formulated to positively prove or disprove each of the possible causes of noise. This test plan involved a series of tests in both the 5.1in (130mm) test rig (Figure 7), and the 2D Leaf Test Rig (Figure 17).

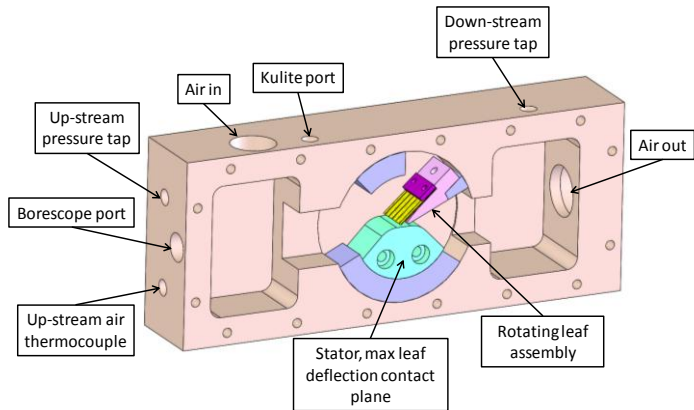


Figure 17: 2D Leaf Test Rig Design

A special test rig was designed and constructed which facilitated experimentally varying seal parameters in a well-controlled manner while providing access for good visual observation and accurate measurement of seal behavior. Shown in Figure 17 & Figure 18, the 2D Leaf Test Rig consists of a pressure vessel body with chambers upstream and downstream of a rotating leaf assembly in the round chamber. The leaves can easily be positioned to adjust the distance from the contact

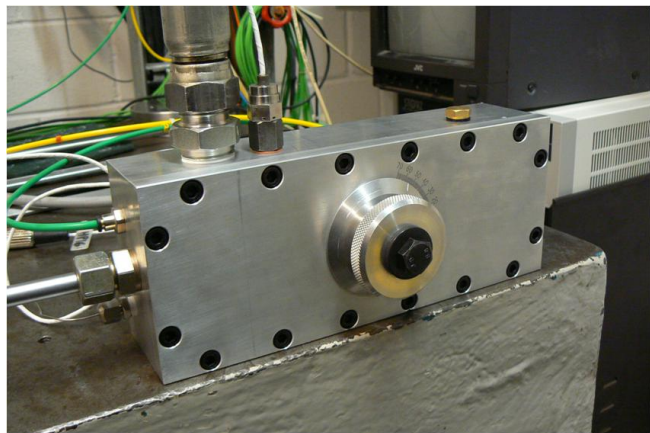


Figure 18: 2D Leaf Test Rig Photo

plate that simulates a rotor surface. The whole assembly can be rotated to vary the approach angle of the flow and the leaves can be rotated relative to the contact plate to obtain a wide range of angles between the leaves and the plate. Ports for measuring pressure and temperature are visible in the figure. This configuration also made it easy to place thin shims

between the leaves at their attachment location to test the influence of a gap between leaves on leaf vibration and noise.

	Hypothesized cause of leaf vibration and noise
A	Upstream pressure in the test rig fluctuates in response to varying open area for flow past PALS.
B	A cavity in the air supply system resonates at the frequency of the leaves.
C	Interleaf gaps allow relative motion without damping.
D	After wear-in burrs at the tip of the middle leaf silence the noise by supporting top leaves.
E	Vortex shedding from leaf tips excites leaves at their natural frequency.
F	Shortening the leaves during wear-in changed their natural frequency enough to avoid resonance
G	The approach angle between the airflow and the leaves is a key factor in vortex shedding.
H	HCF cracking occurred in the top leaf near the knee, allowing better interleaf contact or damping causing the noise to disappear.
I	Vibration is caused by a cushion of air between the support and the leaves.
J	Blunt ends of leaves on the 2D Leaf Test Rig redirect flow up between leaves causing vibration.
K	There is insufficient damping in the current PALS design.
Table 1: Comprehensive List of Possible Causes of Leaf Vibration and Noise	

The same air supply was used for both the 2D Leaf Test Rig and the 5.1in (130mm) test rig so one of the first tests was to determine whether the noise was related to {A}, upstream pressure in the test rig fluctuating in response to variation in the open area for flow past the PALS, or {B}, a cavity in the air supply system that resonates at the frequency of the leaves. It was reasoned that rapid response of upstream pressure could cause a synchronized flutter of all leaves in the set. It was acknowledged that this would not be an independent cause because it would require {C}, interleaf gaps, to activate. Air supply plumbing was revised to eliminate any potential cavities and a bypass was provided around the seal such that the flow through the seal was about 20% of the total flow to dramatically reduce the upstream response to seal closure. In tests at 5 psid a 2000 Hz tone increased from 0.6 to 0.75 psi p-p as flow was adjusted from no bypass to 80% bypass. Thus, testing with and without bypass flow resulted in the essentially the same level of leaf vibration and noise, discrediting

hypotheses {A} and {B}. Later on in the test program after a wear-in test in the 5.1in (130mm) test rig there ceased to be leaf vibration or noise at any test condition, with or without bypass flow, positively disproving hypotheses {A} and {B}.

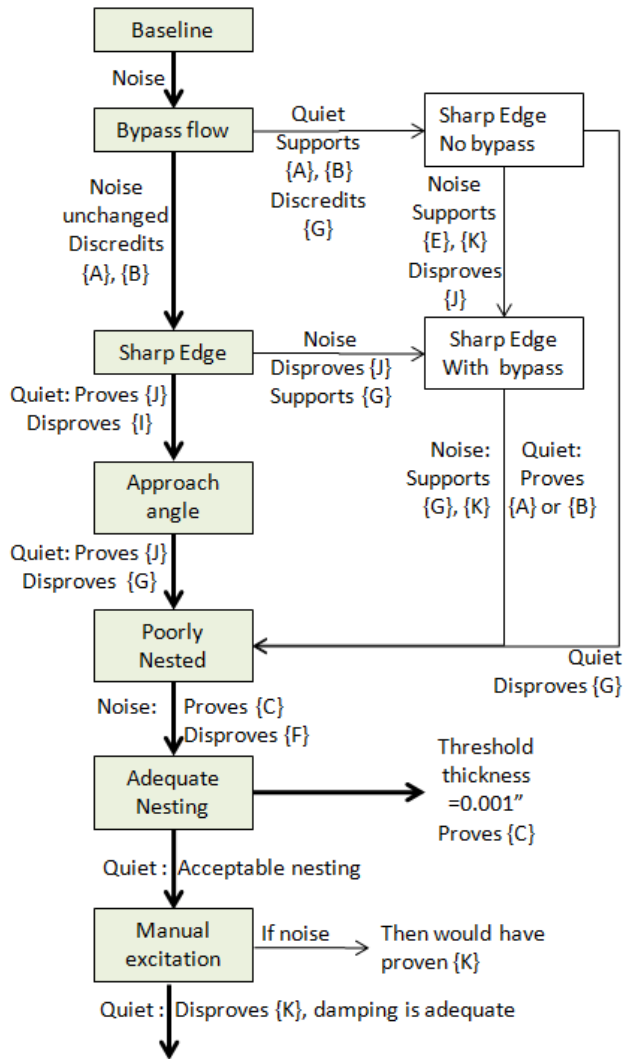


Figure 19: Sequence for Testing in 2D Leaf Test Rig

The next step was to re-cut the ends of the leaves for the 2D Leaf Test Rig to give them a chisel-shaped end instead of a square end. In the assembled seal the leaves were clamped against the support at an angle of about 29 degrees and cut on a horizontal line (parallel with the centerline of the rotor). This also assured that under full pressure the second layer of leaves would not protrude beyond the top layer and that each layer would have a sharp edge for sealing. Test results did not reveal any leaf vibration or noise with or without bypass and at any pressure. This demonstrated that the square ends of the leaves were adversely affecting flow, proving hypothesis {J}.

It did allow the possibility that a burr from the machining was providing support or damping between the two layers of

leaves where there were interleaf gaps previously. This possibility was tested at a later time and is discussed below. It had been speculated that the blunt ends of the leaves and the resulting protrusion of the middle layer beyond the top layer was creating a cushion of air between the support and the leaves that was a key factor. There had been a plan to drill a vent hole through the support to vent that area, but the noise was eliminated without any venting, so there was no need for that test and hypothesis {I} was eliminated.

If vortex shedding from the leaf tips was causing leaf vibration then the approach angle between the airflow and the leaves should be a key factor in controlling the vibration, {G}. A series of tests was performed in the 2D Leaf Test Rig in which the leaf support was rotated to different angular positions. The entire assembly rotated as a unit, so only the flow approach angle was being changed. Figure 20 shows that the effective clearance and rate of closure changed as the assembly was rotated over a range of leaf angles from 29 to 56 degrees. This is accounted to the fixed geometry of the rotating leaf assembly. At large angles the leaf deflection is sufficient for the leaf tip to nearly contact the simulated rotor, however, as the angle is reduced the deflection becomes restrained by the support plate geometry and the leaf is too short to reach the simulated rotor. Consequently, for the rotating leaf assembly, as the leaf assembly angle increases the tip clearance decreases. All angular positions tested remained quiet; therefore hypothesis {G} was eliminated.

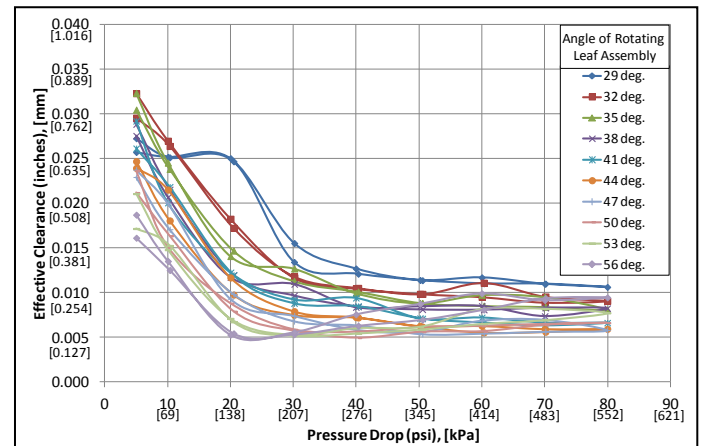


Figure 20: 2D Leaf Clearance, Varying Leaf Angle

Taking the quiet seal with the chisel-end leaves, tests were next performed with shim stock between the top and middle leaves where they were clamped together. Tests were performed with 0.010, 0.005, 0.0025 and 0.001in (0.25, 0.13, 0.06 and 0.025mm) shim stock. All tests with shims showed leaf vibration and noise at 5 and 10 psid (34 and 69kPa). Using bypass did reduce the noise but the effects were inconsistent and never eliminated the noise. With a 0.005in (0.13mm) shim at 5 psid (34kPa) the primary frequency was reduced by 18-24% when the flow was bypassing. At 10 psid (69kPa) the

primary frequency was 2036 Hz and it was reduced 47% by bypass. With a 0.0025in (0.06mm) shim the leaf vibration and noise persisted. A back-to-back test with no shim was quiet with no leaf vibration. With a 0.001in (0.025mm) shim the seal was quiet except for a brief “bleat” as flow was started; then it was quiet for all other test conditions. This series of tests proved conclusively that interleaf gaps (greater than 0.001in (0.025mm)) could initiate leaf vibration {C} and the noise could be eliminated by achieving good interleaf contact (not more than 0.001in (0.025) gap).

This test also disproved hypothesis {F} that shortening the leaves during wear-in changed their natural frequency enough to avoid resonance. There was no change in the length of the leaves between these tests, but adding or removing shims could consistently control whether the seal was noisy or quiet. So, hypothesis {F} was eliminated.

Although it was proven that interleaf gaps could allow leaves to vibrate, it did not prove that leaves were adequately damped to prevent vibration initiated by an external stimulus. So, tests were performed in the 2D Leaf Test Rig using contacting leaf configurations that do not vibrate. Tests were performed over a range of pressures to determine if manual excitation by “poking” the leaves could trigger vibration. None of the poking tests caused any vibration, demonstrating that damping is sufficient and disproving {K}.

It still remained to determine whether a burr on the ends of the leaves resulting from the machining was providing contact between layers of leaves or providing added damping. This was especially interesting because when the first prototype seal was tested in the 5.1in (130mm) test rig it had leaf vibration and noise up to 30 psid (207kPa) on each of the early tests. Once a wear-in test was performed in which a larger diameter rotor was used to assure that the leaves would contact the rotor as pressure was increased, the seal was silent for all subsequent testing at all pressures. Hypothesis {E} also involved burrs, speculating that vortex shedding from the leaf tips excited the leaves and after wear-in the burrs on the leaf tips changed the character of vortex shedding, explaining why noise stops. To test these hypotheses, {D} and {E}, the 5.1in (130mm) test rig was used.



Figure 21: Seal with Intentionally Damaged Leaves

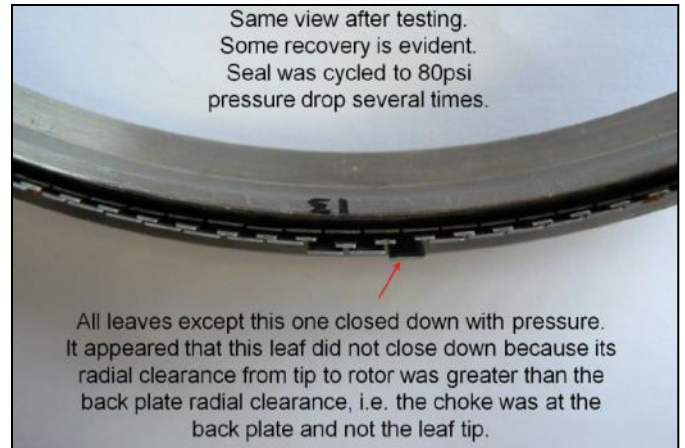


Figure 22: Intentionally Damaged Leaves, Post Testing

After thorough inspection of the seal used for initial testing, the rig was reassembled and static (non-rotating) and dynamic (rotating) tests were repeated. As was the case for all testing after the wear-in test, there was no leaf vibration and no noise at any pressure level, with or without bypass flow. Then the seal was removed and clamped into a fully-deflected condition. A wire EDM was used to remove burrs, cutting a new chisel-edge to 5.128in (130.25mm) diameter fully deflected. Remaining burrs were removed by hand. It was reassembled into the test rig with the same 5.133in (130.25mm) diameter rotor. Tests were performed, taking care to limit the differential pressure to assure that the leaves did not rub the rotor. There was a slight hum detected as the flow was initiated, but effectively the leaf vibration and noise were still absent. This seemed to indicate that burrs were not a factor, disproving {D} and {E}. However, one additional possibility was considered. If HCF cracking had occurred because of the length of time that the leaves had been allowed to vibrate then that could have altered the stiffness and the natural frequency or provided some damping. So, the seal was removed from the test rig and examined by microscope and dye penetrant for indications of HCF. None were found, eliminating hypothesis {H}.

As the 2D test rig has indicated that the interleaf gaps result in leaf vibration and noise at low differential pressures a test of the impact of handling damage was conducted. Starting with the seal for the 5.1in (130mm) test rig which was quiet for all conditions, leaves were manually bent up to simulate handling damage (Figure 21 & Figure 22). Strong vibration and a loud noise resulted at 5 and 10 psid (34 and 69kPa), (Figure 23). One leaf that had been deflected to a tip radius greater than the support inner diameter did not close under pressure. Several cycles to 80 psi (552kPa) yielded repeatable results with no change in the appearance during the test (Figure 23). This test confirmed that interleaf gaps can lead to the initiation of vibration and noise.

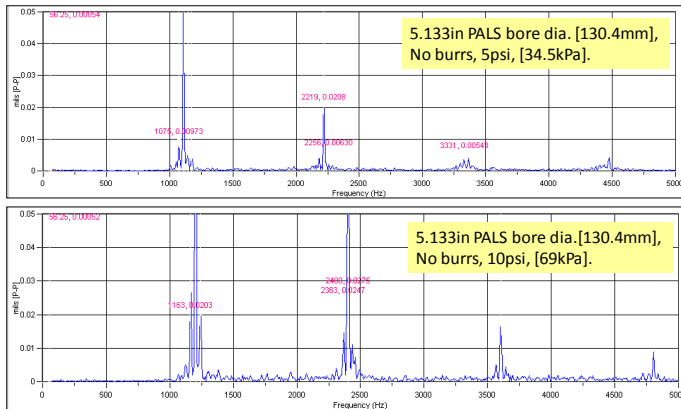


Figure 23: Damaged Seal Noise Signal at 5 and 10psi respectively

SIMULATED SHROUDED TURBINE BLADE TESTING

The PALS simulated turbine blade test installation can be seen in Figure 24 & Figure 25 using the 5.1in (130mm) test rig. The test was conducted on a 5.12in (130.0mm) diameter rotor with 12 equal spaced step and slot combinations round the circumference of the rotor.

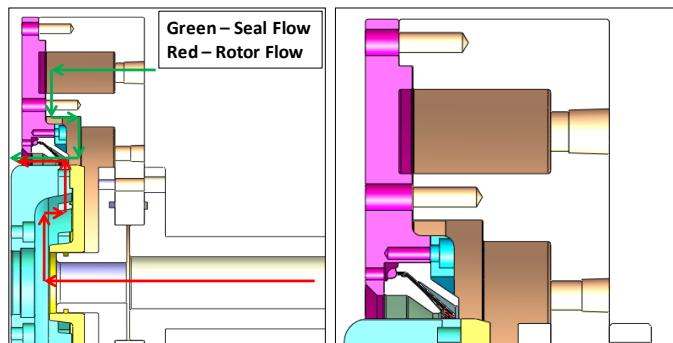


Figure 24: Shrouded Rotor Test Schematic with Air Flow Path

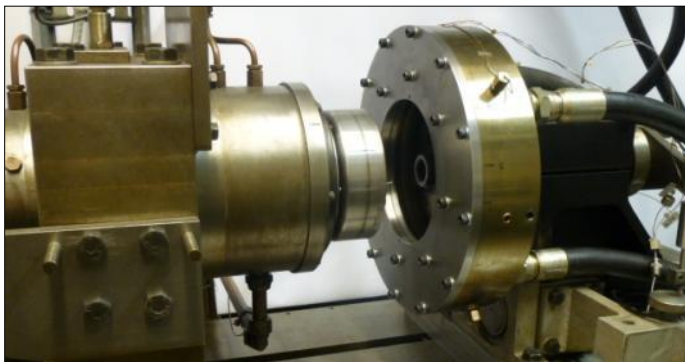


Figure 25: Shrouded Rotor Test Setup

The radial height of each step was 0.003in (0.076mm) with slots allowing for radial air to be injected under the seal leaves (Figure 26) at a higher pressure than that at the front of the leaves. This type of rotor set-up simulates any discontinuation of the shroud sealing surfaces as typically seen in steam turbine locations. The rotor was again uncoated Aubert & Duval 819B as per the smooth rotor test.

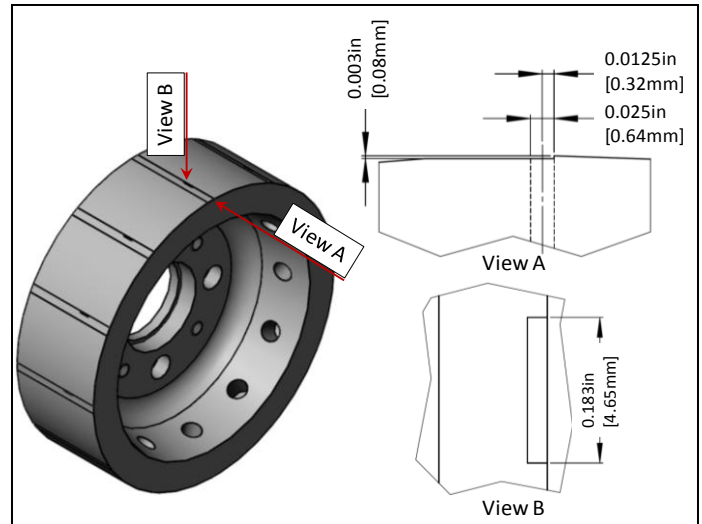


Figure 26: Rotor Slot and Step Detail, 12 Places

A 15 hour steady state dynamic test was conducted with a seal pressure of 55psi (379kPa), rotor pressure of 65psi (448kPa) and venting to atmosphere in both instances. The rotor speed was held at 20,000rpm, equating to a slot frequency of 4000Hz. The seal had a cold clearance of 0.018in (0.5mm) and an initial deflected leaf to rotor interference of 0.003in (0.076mm). The test was broken down into 4 intervals with inspections taken at 1hr, 3hrs, 9hrs and 15hrs. The rotor and seal flow were measured independently of each other to calculate the effective clearance in both instances, with the total effective clearance being reported. The results for total effective clearance in Figure 27 show a period of wear in before stabilization. The seal first experiences an increase in sealing performance as the leaves bed-in to the rotor from ~0.006in (0.152mm) to ~0.004in (0.102mm) total effective clearance during the initial hour long test. The seals performance then decreased gradually over a period of 3 hours by ~0.001in (0.025mm) total effective clearance before stabilizing between 0.0035 (0.089mm) and 0.004in (0.102mm) for the remaining 11 hours.

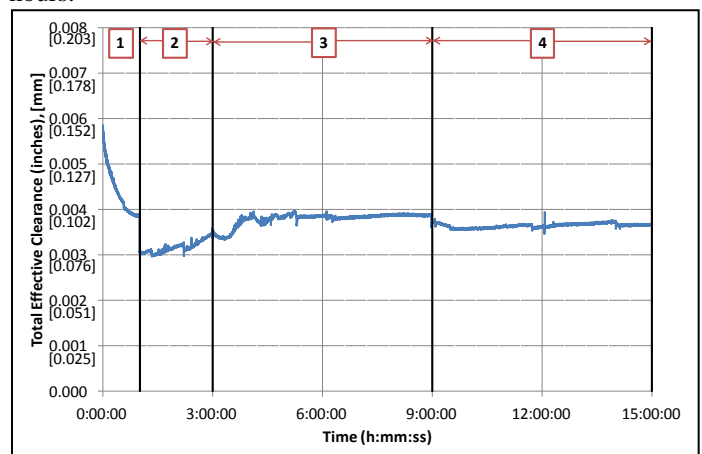


Figure 27: Total Effective Clearance over the 15hr Test with Test Breakdown Intervals Shown

The seal was inspected at intervals through the testing with a comparative on leaf wear shown in Figure 28. The seal was EDM cut with only shakedown wear to the tips at 0 hours of testing. After the 15 hour simulated shroud test, wear was visible to the leaf tips, with small burr formation on both the top and bottom leaves (Figure 29). In total the free diameter of the bore increased by 0.008in (0.2mm) from 5.156in (130.96mm) to 5.164in (131.17mm) and the leaf tips became profiled to an angular deflection relative to the rotor.

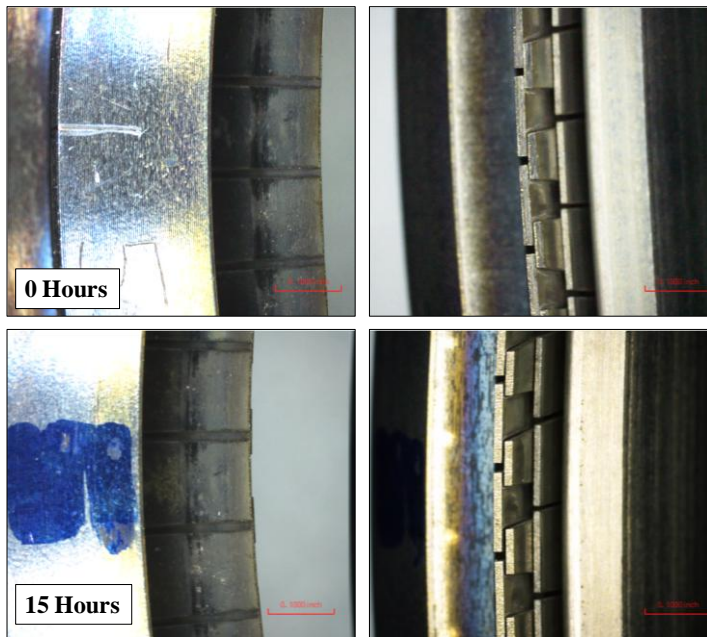


Figure 28: Leaves at 0hr and 15hrs of Testing



Figure 29: Burr Formation in Both Top and Bottom Leaves

The rotor wear track was again visible, with some heat generation marking on the upstream side of the track (Figure 30) generated at the point of initial contact and wear. The majority of the track was created within the first 3 hours, with the wear track at 0.0001in (0.0025mm) depth after 1 hour of running and only 0.0003in (0.0076mm) after the full 15 hour test as seen in Figure 31. The results for the total effective clearance and wear traces indicate that frictional heating has minimal influence on sealing performance after the designed wear-in stage.

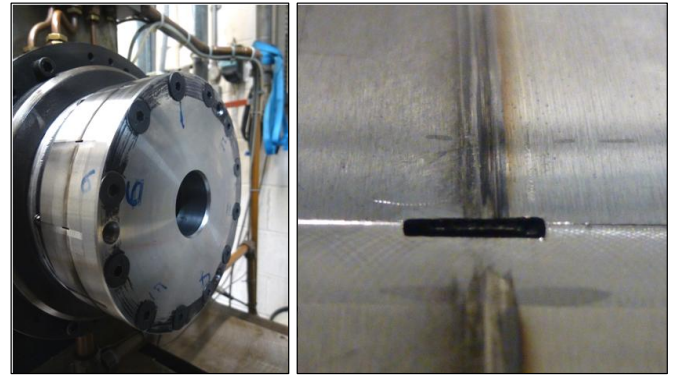


Figure 30: Rotor Wear Track at 15hrs of Testing

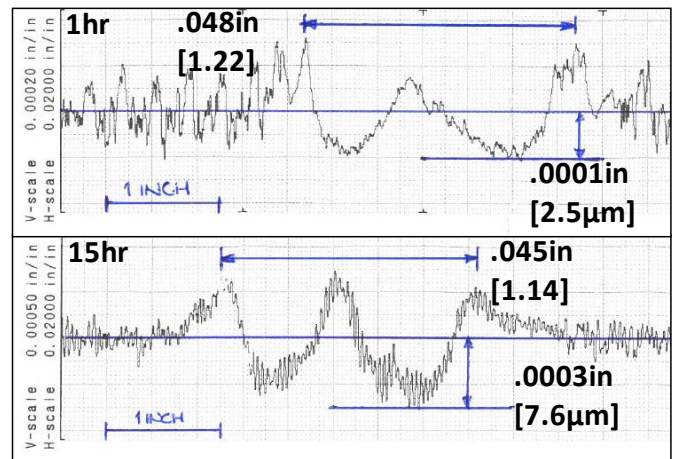


Figure 31: Rotor Wear Traces after 1hr and 15hrs of Testing

A reverse rotation test was carried out with a seal pressure of 55psi (379kPa), rotor pressure of 65psi (448kPa) and venting to atmosphere as per the 15 hour test. The test was run at speeds from 5000rpm to 20,000rpm and held for a few minutes. The results in Figure 32 demonstrate the seals ability to run in the reverse direction, against the steps. Further, the total effective clearance improved with speed and at 20,000rpm the results were comparable to that measured in the 15 hour test.

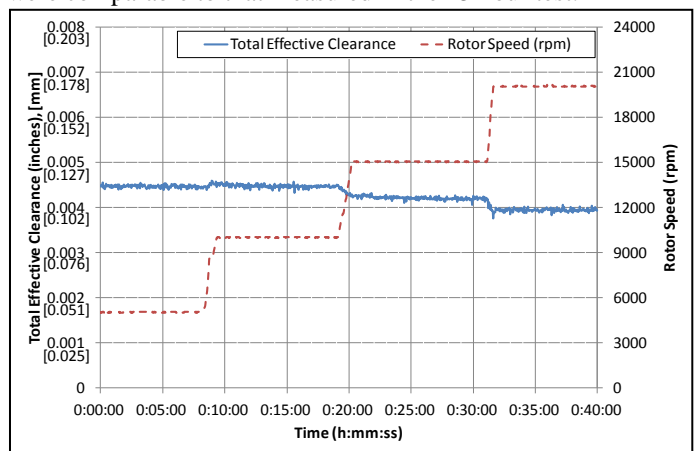


Figure 32: Total Effective Clearance running Reverse Rotation at Varying Speeds

Finally static leakage tests were conducted pre and post testing with the results shown in Figure 33. These were undertaken with the leaves located over the slots, increasing the seal pressure from 0-70psi (0-483kPa). No rotor pressure was applied and the air path to the slots closed. At pressures below 20psi (138kPa) the pre-testing static leakage encountered leaf oscillation as seen by the discontinuity of the total effective clearance when applying the pressure verses removing the pressure. The leaf oscillation was not present on the post-testing static leakage test. Pressures above 20psi (138kPa) show an improvement on total effective clearance from pre to post-testing as a result of the leaf wear in. There is also an increase in effective clearance when comparing static to dynamic results. This can be attributed to the radial growth of the disk due to rotation.

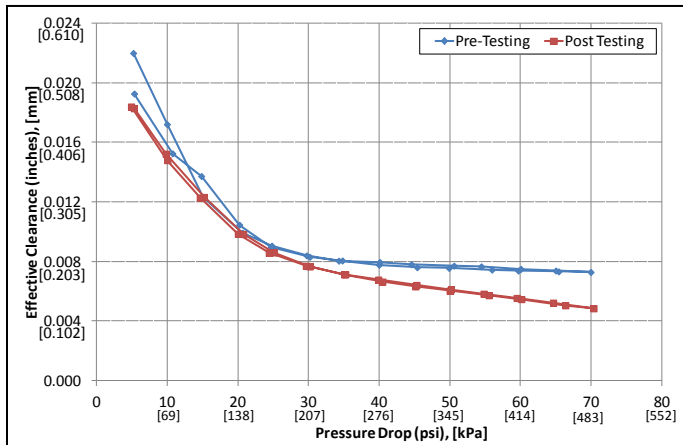


Figure 33: Static Leakage Results Pre and Post Testing

CONCLUSIONS

The testing described in this paper demonstrates the ability of the PALS design. The unique design, that allows a large cold build clearance to reduce by a predetermined amount as pressure drop is applied, is now looking for suitable applications.

Dynamic testing of the PALS with smooth uncoated rotors has demonstrated the performance of the seal at surface speeds of 300ft/sec and pressure drops of up to 120psi (828kPa). The ability to tolerate radial offsets and full 360° rub with no loss of seal integrity and only small performance reductions has been clearly demonstrated. This ability to tolerate rubs enables the seal to be built to a size that would enable the seal to ‘wear-in’ during the first run of an engine and thus relax the manufacturing tolerances.

The acoustic noise experienced during early testing at pressures before the seal is properly activated has been thoroughly investigated in a 2D static rig and 5.1in (130mm) test rig. The source of this noise has been confirmed as leaves vibrating at their natural frequency. The test work confirmed that this excitation can only occur if there is a gap between the layers of leaves that make up the PALS. The noise has only

been observed at low pressures, typically less than 30psid (207kPa), even when mechanical poking was used for excitation. Once the pressure is high enough to cause firm contact between layers of leaves, the damping is adequate to stop all noise. Consequently, stable operation is anticipated at all higher pressure levels.

Dynamic testing of the PALS design on a rig designed to simulate running on shrouded turbine blade tips has confirmed the suitability of the seal for use in turbine blade tip applications. Even with a rotor pressure 10psi (69kPa) higher than the seal pressure and radial steps of 0.003in (0.076mm) all passing the seal at 4000Hz the seal maintained a consistent performance for 15 hours of running.

RECOMMENDATIONS FOR FURTHER WORK

Current work is in progress on the influences of manufacturing tolerances and preload.

Additional reverse rotation testing is recommended with varying degrees of rotor offset, rotor eccentricity and rotor pressure can be implemented to investigate the effects on stability, wear, excitation and high cycle fatigue.

Also, an analytical approach to predicting the wear rate of the leaves based on operating conditions is to be evaluated against further test data.

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