Pressure Actuated Leaf Seal Feasibility Study and Demonstration

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An innovative shaft sealing concept was introduced in AIAA-2005-3985, Pressure Actuated Leaf Seals for Improved Turbine Shaft Sealing. Seal elements in that design elastically deflect with differential pressure to intentionally close a large startup clearance to a small, non-contacting, running clearance. Risk of seal rubs during startup and shutdown transients is greatly reduced with large seal clearance preserving unit operating performance. This paper presents developments funded by the New York State Energy Research and Development Authority to evaluate the feasibility of Pressure Actuated Leaf Seals for power generation applications. Discussion includes the design of test seals and test rigs built to demonstrate Pressure Actuated Leaf Seal clearance change function and seal leakage effectiveness. Test results affirm concept feasibility over a wide range of pressure at ambient temperature. Continued product development is proposed to realize the promising benefits effected by Pressure Actuated Leaf Seals of: a) Rub avoidance with pressure actuated seal engagement / disengagement during startup and shutdown, b) Performance and extended seal life with non-contacting seal operation at small running clearance, c) Bi-directional shaft rotation, d) High differential seal pressure capability and e) Reduced cost.

I. Introduction

Seals between high-pressure and intermediate- or low- pressure turbine sections that reduce leakage are critical components to machine efficiency. However, start-up and shut-down rotor dynamic transients often experienced in large rotating equipment leads to varying seal clearances, posing a challenge to designers to prevent seal rubs, lost performance, and costly corrective maintenance. The goal for turbo-machinery efficiency improvement is to minimize seal leakage and rubs that are detrimental to performance. A Pressure Actuated Leaf Seal, provides seal leaves that elastically deflect in an axial direction to reduce operating clearance according to the differential pressure to which they are subjected. This sealing concept, the subject of an issued patent[1] and published patent application[2], provides the designer with a means to avoid startup and shutdown rubs while preserving low seal leakage at operating conditions. The development steps of application-relevant seal design, prototype manufacture, and rig testing to demonstrate concept feasibility were undertaken in a project funded by the New York State Energy Research and Development Authority (NYSERDA).[3] Results of that effort are discussed in this paper.

Pressure Actuated Leaf Seal (PALS) components and assembly shown in Fig. 1 were previously described in AIAA-2005-3985[4] in some detail along with Pressure Actuated Leaf Seal (PALS) functional characteristics illustrated in Fig. 2. In brief review the seal assembly is comprised of three components:

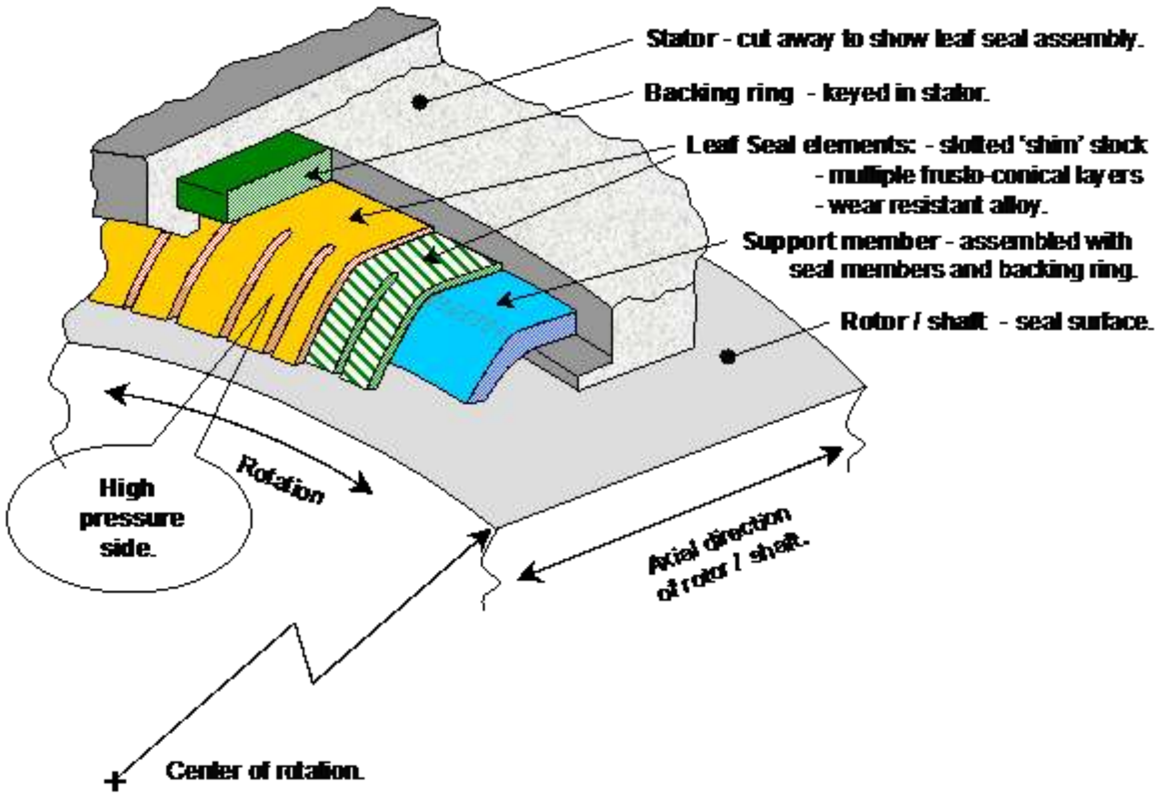

Figure 1. Pressure Actuated Leaf Seal Assembly.

- **Seal members** that are fabricated from sheet metal. Slots cut into the edge of strip shim stock, for example, form leaves that are bent at an acute angle from the uncut edge portion of the strip and wrapped into a frusto-conical shape about the seal longitudinal axis. Multiple seal member layers are used to bear the differential pressure of a particular application and, similar to the tabs of a roofing shingle, leaves are displaced from each other to block airflow. Seal member thickness and number of seal layers are selected to meet application requirements of differential pressure and seal closure. Leaf width is much greater than seal thickness, so leaves are essentially flat cantilever beams for analysis of stress that is kept well within the elastic limit at maximum operating temperature. In cooperation with the support member, seal leaves are designed to elastically deflect in response to system pressure into a close running seal clearance under steady state operating conditions without rubbing contact with the rotor as illustrated in Fig. 2.

- A **support member**, located under the seal leaves on the low-pressure side, bears the upstream pressure applied to leaf seal members. The minor diameter of the support member is set close to the rotor consistent with avoiding rubbing contact under all circumstances similar to the 'fence height' in brush seal designs. Structurally the support member is designed to carry the full differential pressure seal load. The upstream surface of the support member includes an arc that seal leaves conform to as differential pressure is applied. Leaf contact with the arc displaces seal member ends to a smaller radial height and smaller seal clearance as shown in Fig. 2. Arc length and radius determine the amount of change in seal clearance.

- A **backing ring** typically surrounds seal leaves and support member and all are consolidated into a seal assembly by a weld as illustrated in Fig. 2 or other means. Functionally the backing ring provides insertion of a segmented seal assembly into a stator machine component surrounding the rotor or shaft to be sealed. In design of the backing ring consideration is given to the torque applied by differential pressure forces acting on seal and support members and leakage paths around the seal assembly. Space requirement for installation of a Pressure Actuated Leaf Seal assembly will vary with design requirement but 0.5 inches in both axial length and radial height, referenced in Fig. 2 is expected to be typical.

Pressure Actuated Leaf Seal components function together to avoid start-up and shutdown seal rubs as illustrated in Fig. 2. The left ‘cold unit’ panel shows frusto-conical leaves of the seal assembly formed with a large clearance. The ‘operating unit’ view shows differential seal pressure acting on seal leaves to elastically deflect them into contact with the support member’s front face arc, reducing the operating leaf seal clearance. The differential

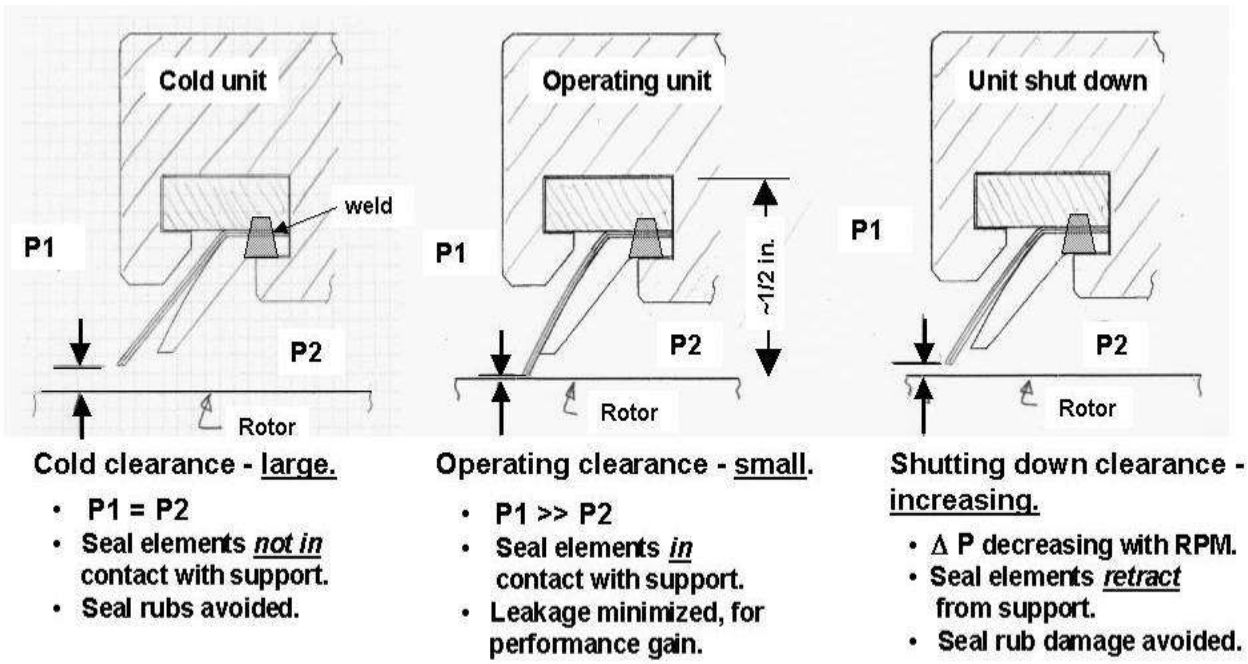

Figure 2. Pressure Actuated Leaf Seal Functional Characteristics.

pressure to bring seal leaves into contact with the support member is selected to meet application objectives. If unit warm-up and acceleration to operating speed is a small fraction of operating differential pressure, leaf engagement pressure may be low. When other criteria are important, such as avoiding rubs on shutdown near full operating differential pressure, a higher leaf engagement - retraction pressure may be selected. Seal deflection is a function of leaf thickness and length, support member forward face arc radius, the number of seal leaf layers and differential pressure. As differential pressure diminishes below the engagement – retraction pressure on shutdown, as illustrated in the ‘unit shutdown’ panel on the right, leaves elastically disengage from the support member arc surface. Increasing seal clearance functions to avoid seal rubs as shaft speed slows. Significant rub avoidance is accomplished by the Pressure Actuated Leaf Seal design in this manner.

In the event of an unusual upset rotor transient occurring at full load conditions, radial slots in the frusto-conical seal elements of the Pressure Actuated Leaf Seal allow leaves to deflect in a non-catastrophic manner. The differential seal pressure force on seal leaves is opposed in part by their elastic deflection from their unloaded frusto-conical shape. Friction during a rub at full operating pressure could be significantly higher than during unit startup or shutdown. However, if the rub duration is short, a few revolutions, leaves may deflect with little seal tip wear. The Pressure Actuated Leaf Seal design provides a measure of ‘rub tolerance’ in contrast to labyrinth seal teeth that typically deform or ‘mushroom’ when rubbed, opening seal clearance for all subsequent unit operation until replaced.

The primary objective of this NYSERDA funded development phase was demonstration of Pressure Actuated Leaf Seal (PALS) sealing function and operation in static and rotating rig air tests to confirm viability of the basic concept. Specific project goals included:

- Seal clearance change capability > 0.040 inches.
- Operating differential seal pressure from 2 psid to 200 psid.
- 50% seal leakage flow vs. labyrinth seals.
- Peripheral rotor seal speed to 500 ft/sec.

Testing conducted at ambient temperature in air was considered relevant to demonstrate concept viability for eventual use in power generation equipment at elevated seal temperature ~1000°F.

Figure 3 shows the successful demonstration of Pressure Actuated Leaf Seal leakage in relation to 4-tooth labyrinth seal leakage at clearances typical of power generation steam turbines (0.015 inches) and gas turbines (0.025 inches). The following paragraphs discuss project development activities contributing this demonstration.

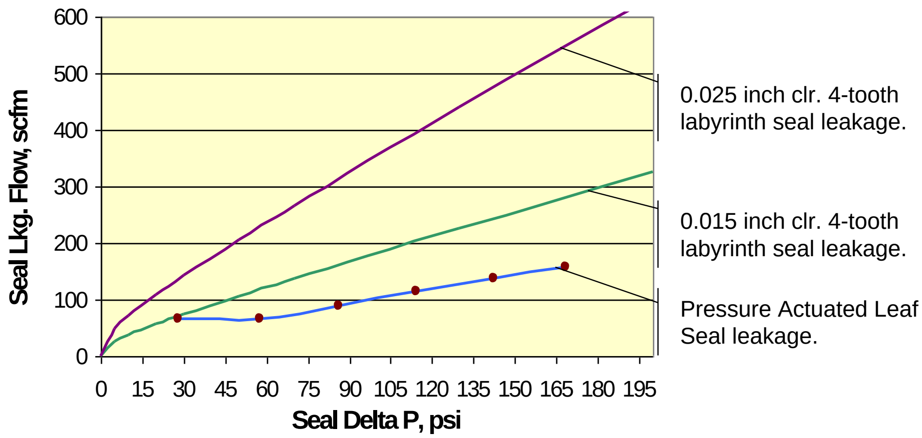

Figure3. Pressure Actuated Leaf Seal Leakage Compared.

II. TWO-DIMENSIONAL STATIC SEAL TESTING

Significant insight to PALS operability was gained in 2-dimensional static seal tests. In these tests, prototypical seal segment leaf deflection was observed and measured. The test rig shown in Fig. 4 was designed for 1 inch length seal segments that could be tested at up to 200 psid seal pressure at ambient temperature. Leaf seal segments of different materials, thickness, length and tip geometry were fabricated and tested. A depth micrometer, show in Fig. 4, was used to measure change of leaf tip position, i.e. seal clearance. The micrometer, electrically isolated from the aluminum frame, completed a continuity test circuit when contact was made with the leaf tip to accurately measured seal clearance. Seal leakage was measured but was of secondary interest compared to measuring leaf deflection and movement. Audible oscillations, when present, were recorded.

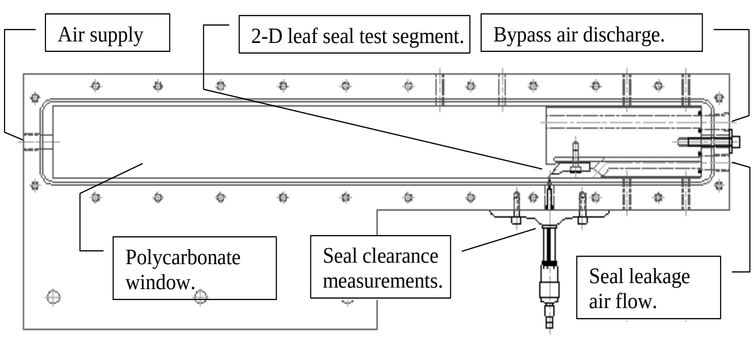

Figure 4. Two-Dimensional Pressure Actuated Leaf Seal Test Rig.

Several test seals of various leaf thicknesses and number of seal leaves were specified to span the range of differential test pressure using CMG Tech seal design tools. A seal support radius of 2.5 inches was held constant for both 2-D and 3-D test seals to minimize design variables in use. Shim stock alloys included Hanes 25, Hastelloy C-276, Nitronic 60, Inconel 718 and Inconel X-750. Hanes 25 material is a typical brush seal bristle material with good wear resistance and high temperature properties suitable for use in power generation turbines. The Hanes 25 shim used in this project was ~15% hard worked with RC 40 hardness and 126 ksi yield strength. Hastelloy C-276 was selected because of its use in some brush seal applications. It was supplied with spring temper hardness of RC 31 and

Figure 5. Two-Dimensional Leaf Seal Segments.

142 ksi yield strength. Inconel 718 and Inconel X-750 were selected because of their high temperature capabilities. Shim of both Inconel alloys, supplied in the annealed condition, was precipitation hardened per AMS 5596 and AMS 5598 for 150 ksi and 115 ksi RT tensile properties, respectively. Nitronic 60 material, selected because it is a non-galling high-temperature stainless steel, was supplied in a work hardened condition with 126 ksi yield strength.

Two-dimensional leaf seal segments, shown in Fig. 5, were wire EDM'ed from shim stock. After bending to a 45° angle, leaves of alternate slot spacing were aligned on a support member and bolted to the 2-D rig mounting block. An assembly of two 0.004 inch thick Nitronic 60 seal leaves is shown in Fig. 6. On the left, without differential pressure, there is a large clearance between leaf tips and frame. On the right, with pressure applied, leaves are displaced to within a few thousandths of an inch clearance with the test frame.

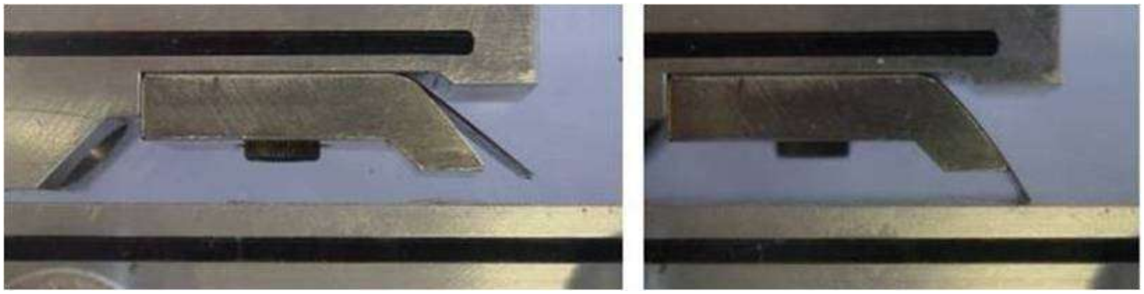

Figure 6. 2-D Pressure Actuated Leaf Seal Clearance Change.

Stable clearance change of more than 0.040 inches from an inoperative condition to a small operating clearance was evaluated by testing combinations of seal leaves at various pressurization levels and flow rates. Initial unstable seal actuation occurred with the application of pressure that was evident with audible seal leaf oscillations or by the failure of top seal leaves to close. A layer of shorter leaves, under the primary seal leaf layers, was identified as an effective means of damping seal leaf oscillations during pressurization. Seal fabrication, assembly, and testing provided insight to other features that also contribute to stable actuation and low seal leakage.

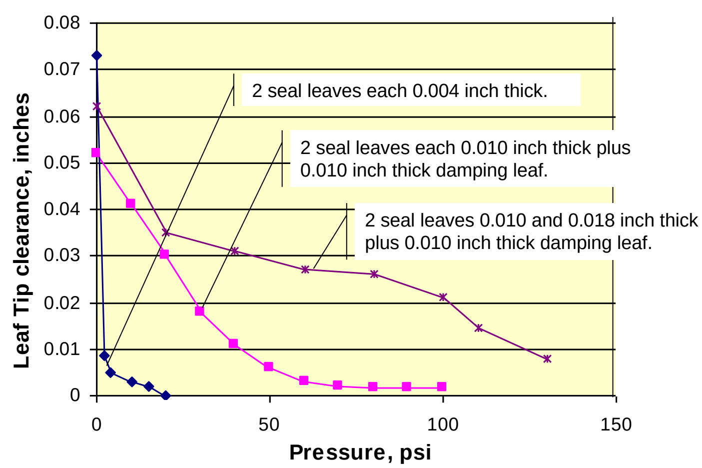

Figure 7. Clearance Change; Different Leaf Thickness.

Two–dimensional seal testing confirmed that seal clearance change with pressure is a strong function of leaf thickness as shown in Fig. 7. The curve on the left is that of 2 seal leaves 0.004 inch thick. Their response to small differential pressure was intended to show that clearance changes at very low seal pressures are possible if desired. The center curve is that of 2 seal leaves that are 0.010 inch thick with an additional leaf of the same thickness beneath them that is shorter. As noted above, the shorter bottom leaf dampens leaf oscillations. Seal leaf deflection is initially proportional to applied pressure but less so at higher pressure as leaves come into compliance with the support member. The seal leaf closure curve on the right in Fig. 7 occurs at substantially higher seal pressure because seal leaves are 0.010 and 0.018 inch thick. These curves show clearance change of more than 0.04 inch that can be widely adjusted by design to occur at different seal pressures as may be required for different applications. The transition or actuation pressure is in reasonable agreement with analytically predicted values; however, thicker leaf seals closed to smaller clearance at somewhat higher pressure than anticipated. A significant volume of air by-passing the seal, typical of operating equipment gas path flow, contributed to stable seal transition in proportion to applied pressure. In contrast, when by-pass flow is not present, seal actuation can be abrupt because leaf closure increases upstream pressure accelerating the transition.

The influence of approaching air flow direction on stable seal clearance change as pressure is applied was evaluated and shown to have little effect. Noted differences occurred at less than ~ ½ the fully actuated seal pressure.

III. THREE-DIMENSIONAL PROTOTYPICAL STATIC AND DYNAMIC TESTING

A. ROTATING TEST RIG

A test rig with rotating components was designed and built to test 3.5-inch diameter Pressure Actuated Leaf Seals. The test rig is shown in cross-section in Fig. 8. Air pressure is supplied to the test rotor rim outer diameter between the test seal and a 7-tooth labyrinth seal. The motor and rotor assembly is capable of 32,740 RPM to produce planned seal speed of 500 ft/sec. Stator parts supporting the test seal and labyrinth seal were made of 17-4 PH stainless steel. The test rotor was also made of 17-4 PH. Seal displacement in both radial and axial directions was provided to facilitate seal rub experiments and adjust seal clearance with the rotor. The latter is accomplished by moving the seal assembly parallel to the rotor axis relative to the tapered rotor OD. Rotating rig components are shown in Fig. 9 as prepared for assembly.

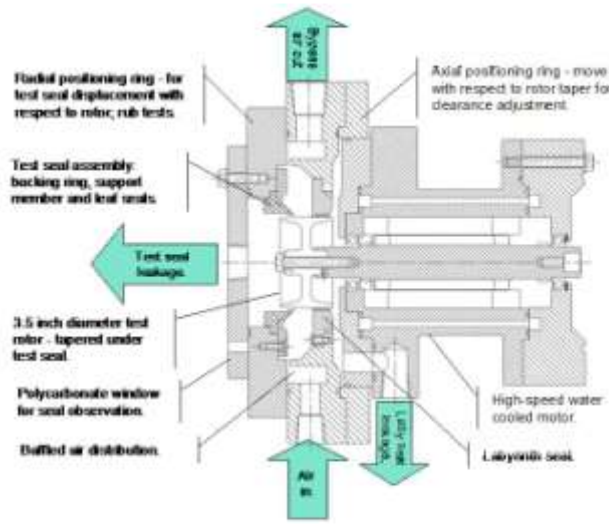

Figure 8. Rotating Test Rig Cross-section.

Figure 9. Rotating Rig Components.

Test rig instrumentation included a thermal flow meter, 2 pressure transducers, and RTD's for use with a USB data acquisition interface and laptop computer data logging. Air supply and exhaust pressure gauges facilitated manual data recording. Two holes through the stator provided radial access for proximity probes to measure rotor orbits. Those measurements were not made in this phase, but the access holes were used to illuminate test seal clearance with a high intensity microscope light source and to measure seal pressure within the test rig rather than in upstream piping.

B. TEST SEALS

Nominal 3.5 inch test seal diameter was selected based on the geometry of a perspective demonstration application; desire for seals to be of moderate size and expense; capable of operating at up to 200 psid seal pressure and 500 ft/sec seal speed. Two seal configurations, illustrated in Fig. 10, were established. Seal leaves cut and bent from strip material are laid in tangent contact with the curved surface of the support ring in the first configuration.

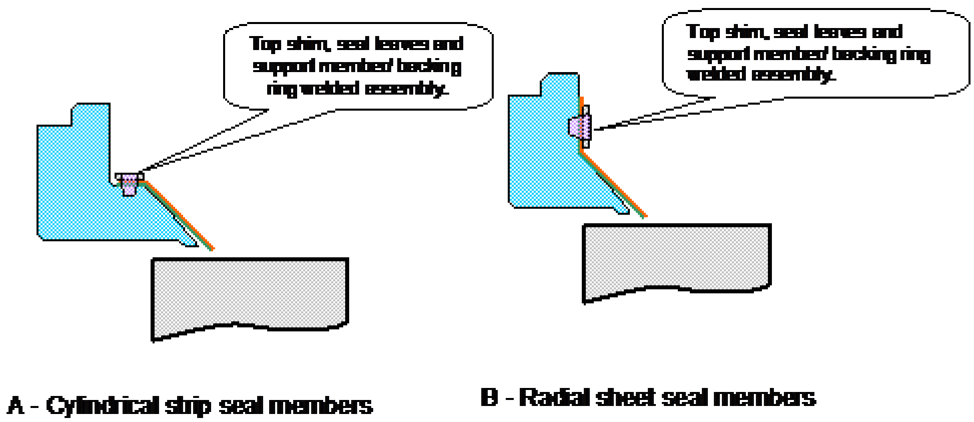

Figure 10. Pressure Actuated Leaf Seal Design Configurations.

The unslotted portion of the seal strip is wrapped around the cylindrical shoulder of the seal support ring and welded to it. In the second configuration seal leaves are cut from sheet material and bent out of plane to assemble tangent to the support ring radius. Seal leaf layers are displaced circumferentially in both configurations to block seal slot leakage. Seal designs for operation at various test pressures are tabulated in Table 1. All 5 seal designs utilize two sets of overlapping seal leaves that range from 0.004 inch to 0.018 inch in nominal thickness. A layer of shorter leaves is provided under the seal leaves for damping as discussed in the 2-D test section. Slots in seal leaves of all thickness were wire EDM cut for both the strip and sheet seal leaf configurations. Sheet seal leaves were cut in 0.010 inch Hastelloy C-276 material for Seal # 3 but leaves were not bent or assembled into a test seal. As shown in Fig. 10, the design intent is to weld seal leaves to the support member / backing ring in both seal configurations. The weld assembly includes a thicker shim on top of the seal leaves to strengthen the weld attachment to the support ring. Laser welding was identified as the prefered method of joining seal leaves to the support ring. However, welding of test seals in this feasibility development phase was omitted along with plans to make test seals in 180 degree halves. Instead, test seals were assembled using a shrink fit lexan ring to constrain cylindrically wrapped strip seal leaves on the support ring. That change facilitated test seal assembly builds discussed below to address leaf bending and 'nesting' issues required for good sealing. This project phase focused on assembly and testing of Seal #2 in Table 1 utilizing 0.010 inch thick Haynes 25 material. The support ring was made from 321 stainless steel.

	Leaf Alloy	Leaf Thickness	# of leaves	Seal Ring Configuration	Calculated Actuation Pressure	Test Pressure, psi Min.	Max.
Seal 1	Nitronic 60	0.0040	2	Split seal ring, strip seal leaves.	1.1	2	75
Seal 2	Hanes 25	0.0098	2*	Split seal ring, strip seal leaves.	26	40	200
Seal 3	Hastelloy C-276	0.0098	2*	360' seal ring, sheet seal leaves.	26	40	200
Seal 4	Inconel 718	0.0120	2*	Split seal ring, strip seal leaves.	39	~75	200
Seal 5	Inconel X-750	0.0175	2*	Split seal ring, strip seal leaves.	109	TBD (>120)	200

* indicates an additional short, non-sealing, leaf 0.010in thick under other leaves.

Table 1. Rotating Rig Test Seal Configuration Matrix.

C. TEST SEAL ASSEMBLY AND INITIAL TESTING

Strip cut seal leaves for an initial 3-D test seal are shown in Fig. 11. Seal leaves are bent 45 degrees with respect to the band portion of the strips and assembled to the seal support ring illustrated in Fig. 10-A. An assembly fixture was used to hold seal strip leaves in contact with the support ring during seal assembly. Seal leaf pitch, the distance between slots, is designed for an integer number of leaves to equal the seal layer circumference. Seal strip ends were trimmed during assembly to provide a small gap between strip ends. A snap ring was used to hold seal layers in place as subsequent seal leaves were added. The assembly fixture applies axial force on the bent portion of all leaf layers holding them in place as the heated lexan ring is shrunk onto the assembled test seal leaf layers. At room temperature, the lexan ring shrink fit holds the assembled leaves in place and provides a flange for assembly in the test rig shown in Fig. 8. A final step in preparing test seals was machining of the seal tip ID. A fixture was used to deflect seal leaves into their deflected, operating position while being machined. Initial grinding raised undesirable burrs, so wire EDM was specified for the seal ID trim. Completed test seals were accurately aligned in the test rig for support ring run-out of less than +/- 0.001 inch with respect to the rotor.

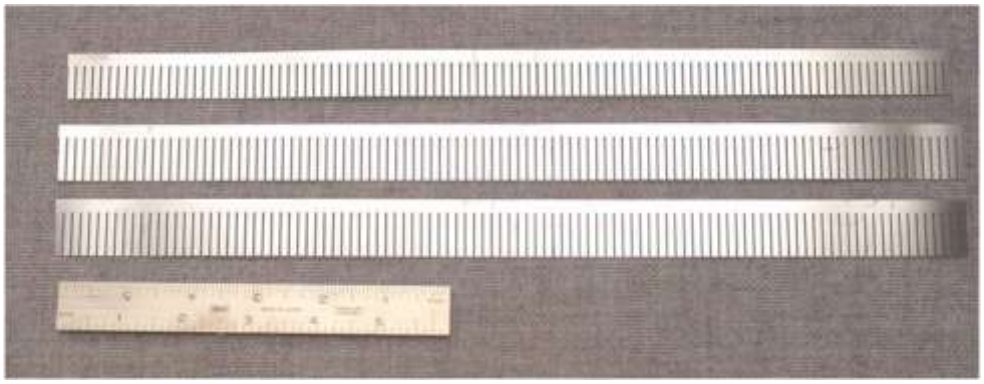

Figure 11. 3-D Strip Cut Seal Leaves.

Intended seal clearance change with application of pressure was demonstrated in tests and is qualitatively observed in the series of photos in Fig. 12. A high intensity light illuminates leaf seal clearance that is seen to decrease as seal pressure is applied.

Figure 12. Seal Clearance Change with Pressure.

D. SEAL LEAKAGE FLOW TEST RESULTS

Test seal #2-5 demonstrated reduced seal leakage that met project goals. Shown in Fig. 13, it has 2 seal leaf layers each of 0.010 inch thickness Haynes 25 alloy with a layer of shorter damping leaves of the same material beneath them. Good interleaf contact or 'nesting' is visible in Fig. 14 and top seal leaf slot flow is blocked by seal leaves below them.

Test seal #2-5 measured seal leakage is plotted in Fig. 15 vs. seal pressure along with calculated leakage of 4-tooth labyrinth seals with 0.015 inch and 0.025 inch clearances that are commonly found in steam and gas turbines, respectively. Also plotted is 7-tooth labyrinth test rig seal leakage, both measured and calculated. PALS leakage is less than the 7-tooth labyrinth seal leakage with 0.0095 inch clearance and considerably less than 4-tooth labyrinth leakage flows with 0.015 inch and 0.025 inch clearance. Good agreement of calculated and measured 7-tooth labyrinth leakage affirms the adequacy of flow calculations being used. In relation to the project goal of 50% seal leakage reduction, Test seal #2-5 leakage is more than 50% of 0.015 inch clearance 4-tooth labyrinth seal leakage but considerably less than 50% of the 0.025 inch clearance 4-tooth labyrinth seal leakage.

Figure 13. PALS Test Seal #2-5, After Testing.

Seal leakage during PALS actuation, change of clearance, with application of seal pressure is seen in Fig. 16. Initially seal leakage flow increases rapidly at low seal differential pressure when seal clearance is large, ~0.035 inches. However, as pressure increases seal leaves start to close and leakage diminishes to a minimum at ~ 45 psid where physical clearance is ~ 0.001 to 0.002 inches. Effective seal clearance, including inter-leaf seal leakage, is estimated to be 0.003 to 0.004 inches. This seal leakage is about the same as a test

Figure 14. Good Inter-Leaf Contact.

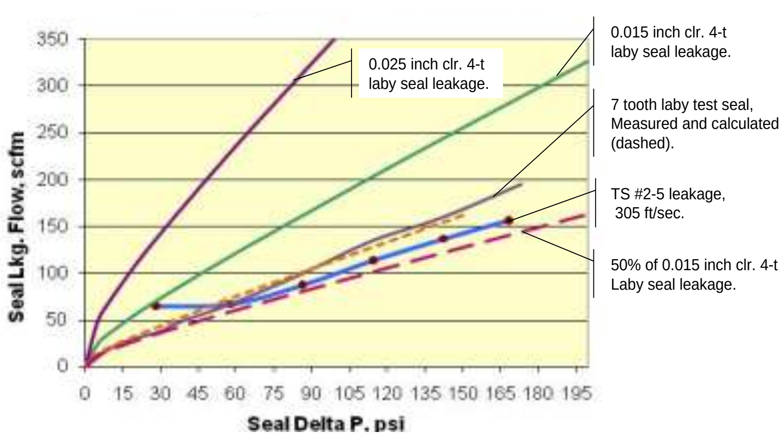

Figure 15. Seal Leakage Flow Reduction vs. Labyrinth Seals.

seal that had inter-leaf leakage blocked with adhesive tape, indicating Test Seal #2-5 has little inter-leaf leakage. However, better alignment of top seal leaves, shown in Fig. 14, with bottom leaf centerlines is expected to decrease interleaf leakage in the future.

A seal rub occurred during initial testing of Test Seal #2-5 at 75 ft/sec and seal pressure of 100 psid that increased effective seal clearance approximately 0.001 inch. Post-test inspection of rubbed seal leaves under microscope revealed a small burr on bottom leaf tips that displaced top leaves opening a small gap for inter-leaf leakage. Seal leakage shown in Fig. 16 is increased for both static and rotating tests after the rub.

Measured seal leakage at 305 ft/sec, Fig. 16, is slightly less than static seal leakage with the rotor at rest. Calculation of radial rim growth at test speed confirmed the flow reduction to be caused by elastic rotor growth reducing seal clearance. Fig. 17 plots of seal leakage vs. speed at 2 differential pressures that show seal leakage reduction with speed. Since reduction in seal leakage correlates with rotor speed, static flow testing should be adequate for routine flow testing of PALS.

Plotted seal leakage in Figures 15 and 16 were measured after a one-hour, steady state, test run at 305ft/sec rotor speed and 165 psid. Constant seal leakage throughout the hour confirmed that the seal was not in rubbing contact with the rotor and that damaging high-cycle fatigue of seal members was not present.

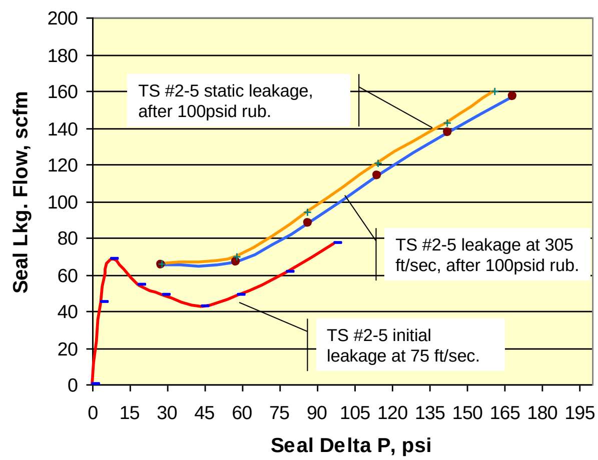

Figure 16. Test Seal #2-5 Seal Leakage.

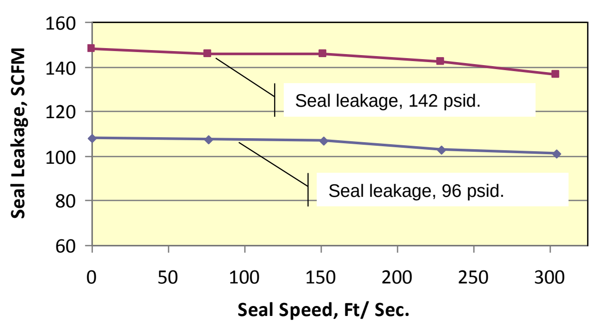

Figure 17. Test Seal #2-5 Seal Leakage Vs. Speed.

E. SEAL RUB TEST RESULTS

Two intentional seal rubs were conducted to evaluate PALS rub tolerance and characterize rotor and leaf wear resulting from the rubs. Radial displacement of the test seal with respect to the rotor simulates a seal damaging rotor upset event at operating conditions, a startup rub resulting from seal misalignment or intentional interference to 'wear-in' the seal. The test seal was displaced in the seal housing relative to the rotor axis 0.005 inches for the first test and 0.008 inches for the second. This displacement was made with the test rig rotating at 305 ft/sec speed but at low seal pressure, below the seal actuation pressure. Seal pressure was then increased to ~135 psid and held for 5 seconds followed by rapid reduction in seal pressure. Static, 0 RPM, seal leakage is plotted in Fig. 18 prior to the 0.005 inch seal displacement along, with the static seal displacement and after the seal rub at 305 ft/sec. Critical seal leakage flow calculations indicate ~0.001 inch clearance increase. After the first rub test, the seal was axially displaced 0.062 inches to a fresh location on the tapered rotor. That displacement reduced physical seal clearance 0.001 inch. Seal leakage at the reduced clearance was reduced to pre-test values, confirming that effective seal clearance changed about 0.001 inch during the seal rub event. Profile measurement of the rotor wear track in the location of the first 0.005 inch seal rub was ~0.0005 inches as shown in Fig. 21. That wear track location includes wear from the initial rub previously discussed.

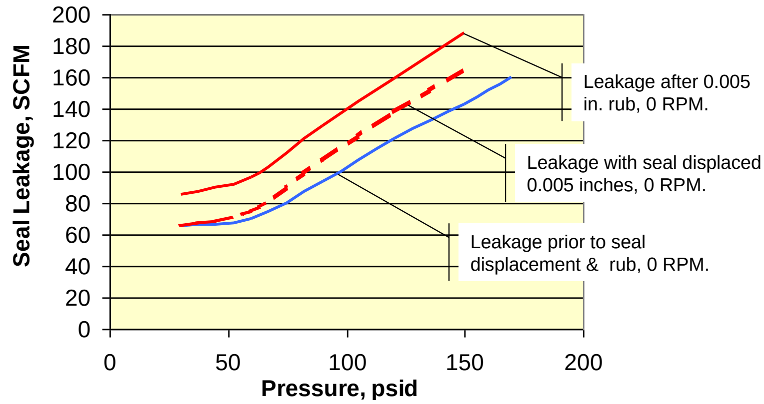

Figure 18. 0.005 inch Seal Displacement Rub Test Leakage.

The second seal rub test was a 0.008 inch seal displacement with the same test sequence and a higher peak seal pressure of ~150 psid. This rub produced very similar change in seal leakage as the first rub test as plotted in Fig. 19. The test seal ID was measured by coordinate measurement system (CMS) after both rub tests with seal leaves depressed. These measurements are plotted in Fig. 20. The change in seal ID was not uniform around the circumference, and maximum wear is not located where the seal was displaced. However, the mean change in seal ID, ~ 0.001 inch, correlates well with the seal leakage change observed. The second rub test, conducted at a clean, axially displaced location on the rotor produced a wear track ~0.0003 inch deep as shown in Fig. 21.

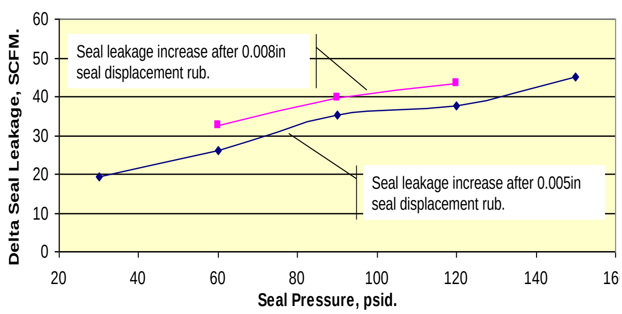

Figure 19. Rub Test Seal Leakage Increase.

In both rub tests, wear of the Haynes 25 leaf tips was ~0.001 inch, much less than the 0.005 inch and 0.008 inch seal displacement. Fixed labyrinth seal tooth damage would likely have been equal to the seal displacement less running clearance estimated to be 0.001 to 0.002 inch, i.e., net rubs of ~0.003 inch and ~0.006 inch respectively. Because measured seal wear and leakage change is approximately the same for this 2X change in seal rub depth, PALS wear appears to be more dependent on seal delta P and speed than the amount of seal interference with the rotor.

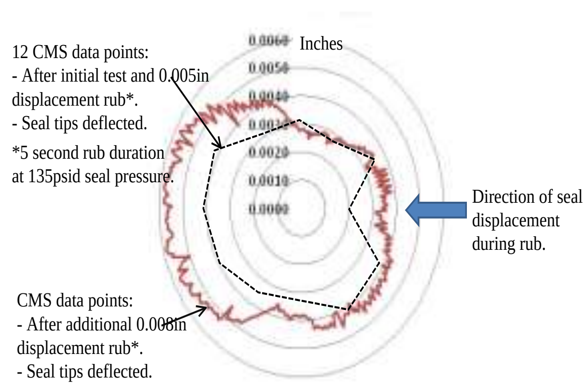

Figure 20. Rub Test Seal Leaf Wear.

Both rubs were imposed for a 5-second duration at 305 ft/sec seal speed, i.e. for ~1500 linear feet of rub distance, at the relatively high seal pressures previously noted. These rub conditions are not correlated with known power generation equipment upset events or startup / shutdown transients, but the modest 0.001 inch, seal wear suggests a reasonable degree of rub tolerance for Pressure Actuated Leaf Seals. However, a leaf and rotor material wear pair is needed that does not remove material from the rotor for commercial applications. The test rotor rub surface was annealed 17-4 PH stainless steel with a Rockwell C hardness of 31 while seal leaves were work hardened Haynes 25 alloy with a Rockwell C hardness of 40. Greater rotor material hardness or hard-face coating would mitigate rotor wear vs. Haynes 25 leaves. Alternatively, leaf tips fitted with abradable material would not readily score un-coated rotor materials during an inadvertent seal rub. Evaluation of these approaches to accommodate PALS 'wear-in' on startup and upset events when operating is planned as part of a follow-on NYSERDA project for product development. [5]

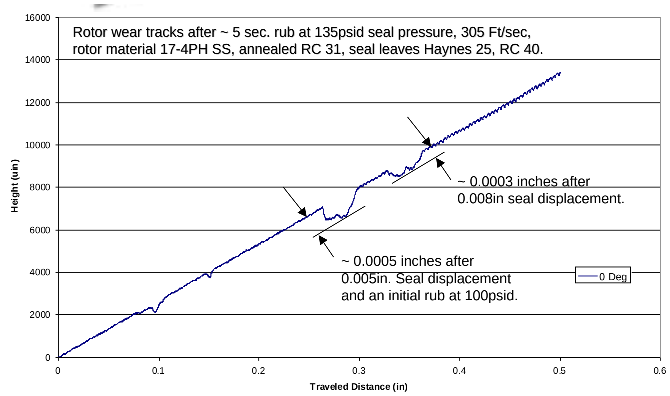

Figure 21. Rub Test Rotor Wear.

An evaluation of the rub tolerance and wear characteristic of candidate leaf seal materials against a 17-4 PH stainless steel rotor material was conducted as part of the NYSERDA project. [6] This room temperature study at 2 friction loads and 2 speeds, 40 ft/sec and 100 ft/sec showed Inconel 718 to be somewhat preferred but did not reveal a big distinction among leaf materials in rubbing wear against a 17-4PH stainless steel rotor.

IV. Conclusion

Pressure Actuated Leaf Seal function has been demonstrated in prototypical seals designed, fabricated and tested in this NYSERDA funded project. Successful seal leaf transition from a large cold clearance to a small operating shaft clearance in response to increasing seal pressure was observed at various pressures in 2-D testing and rotating rig tests. Large unpressurized seal clearance reduces risk of seal rubs during startup and shutdown of power generation equipment. At operating conditions reduced seal clearance and leakage improves unit performance. Measured seal leakage met project goals and further reduction is expected. Testing confirmed non-contacting seal operation when fully pressurized that is independent of shaft speed. Intentional seal rub tests showed that PALS leaf tip wear is independent of the amount of interference with the rotor that is expected to mitigate seal damage in the event of an unanticipated rub at full operating conditions. These encouraging results have secured a follow-on NYSERDA project phase for the product development of Pressure Actuated Leaf Seals.

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Turbotechnology Services Corporation (TSC) of Scotia, New York, recently acquired by Piller-TSC Blower Corporation, is acknowledged for its cost share participation in the NYSERDA project providing engineering support and test facilities.

References

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