# DEVELOPMENT OF PRESSURE ACTUATED LEAF SEALS FOR IMPROVED TURBINE SHAFT SEALING
# FEASIBILITY STUDY

Final Report

Prepared for

**THE NEW YORK STATE**
**ENERGY RESEARCH AND DEVELOPMENT AUTHORITY**

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## NOTICE

## ABSTRACT AND KEY WORDS

### ABSTRACT

The innovative Pressure Actuated Leaf Seal concept incorporates seal elements that elastically deflect with the application of system 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 the large initial seal clearance while preserving small operating clearance and optimum performance at base load conditions (US patent 6,644,667 and continuation-in-part, US publication 2004/0150165). This report presents results from New York State Energy Research and Development Authority (NYSERDA) funded project, Agreement 9887, to evaluate the feasibility of Pressure Actuated Leaf Seals. Discussion includes the design of test seals and test rigs built to demonstrate Pressure Actuated Leaf Seal clearance change function and reduced seal leakage. Test results affirm concept feasibility over a wide range of pressure at ambient temperature. Power system application benefits include: a.) Rub avoidance with pressure actuated seal engagement / disengagement during startup and shutdown; b.) Performance improvement with small running clearance at base load; c.) Extended seal life with non-contacting seal operation; d.) Bi-directional shaft rotation; e.) High differential seal pressure capability and f.) Reduced cost.

### KEYWORDS

Shaft packing, shaft packing rubs, shaft packing seals, rub avoidance; packing life, non-contacting shaft packing; shaft seal leakage, reduced shaft packing leakage; extended seal life; turbine performance, improved turbine efficiency, leaf seal, pressure activated seals, Pressure Actuated Leaf Seal (PALS), retractable packing, metallic seals.

## ACKNOWLEDGMENTS

The management of Turbotechnology Services Corporation (TSC) of Scotia, New York, recently acquired by Piller-TSC Blower Corporation, is acknowledged for its 50% cost share participation in this project. The management of TSC provided engineering support, design and procurement of test seals and test rigs as well as the facilities and personnel to conduct seal tests. The contributions of Jim Flynn, project manager and engineering support, Chris Mulford, design support, Messrs. Art Zdanko, Tim Stadler, and Ted DiMio, providing machine shop services and test support, is noted with appreciation.

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## SUMMARY

Seals used in turbines between high-pressure and intermediate- or low- pressure sections to reduce leakage are critical components to machine efficiency. During start-up and shut-down, rotor dynamic transients experienced in the turbine lead to varying seal clearances, posing a challenge to seal 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 turbine performance. CMG Tech, LLC has developed a pressure-activated seal where seal leaves elastically deflect to reduce operating clearance according to the differential pressure to which they are subjected. The concept has been presented at respected industry conferences where seal industry leaders have affirmed the potential for improved shaft sealing with the technology. It is the subject of US patent 6,644,667 and continuation-in-part, US publication 2004/0150165. The development steps of application-relevant seal design, prototype manufacture, and rig testing were undertaken by the New York State Energy Research and Development Authority (NYSERDA) which funded the project to demonstrate concept feasibility. Pressure Actuated Leaf Seal (PALS) components and assembly are shown in Figure 1. Pressure Actuated Leaf Seal (PALS) functional characteristics are illustrated in Figure 2.

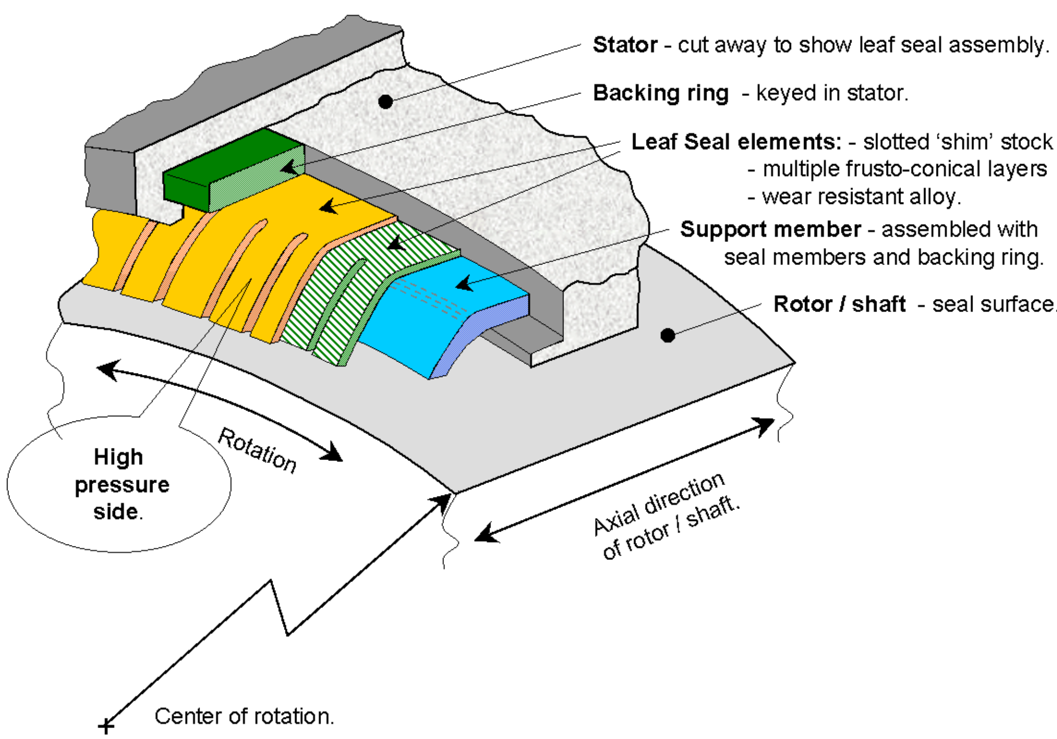



**Figure 1. Pressure Actuated Leaf Seal Assembly.**

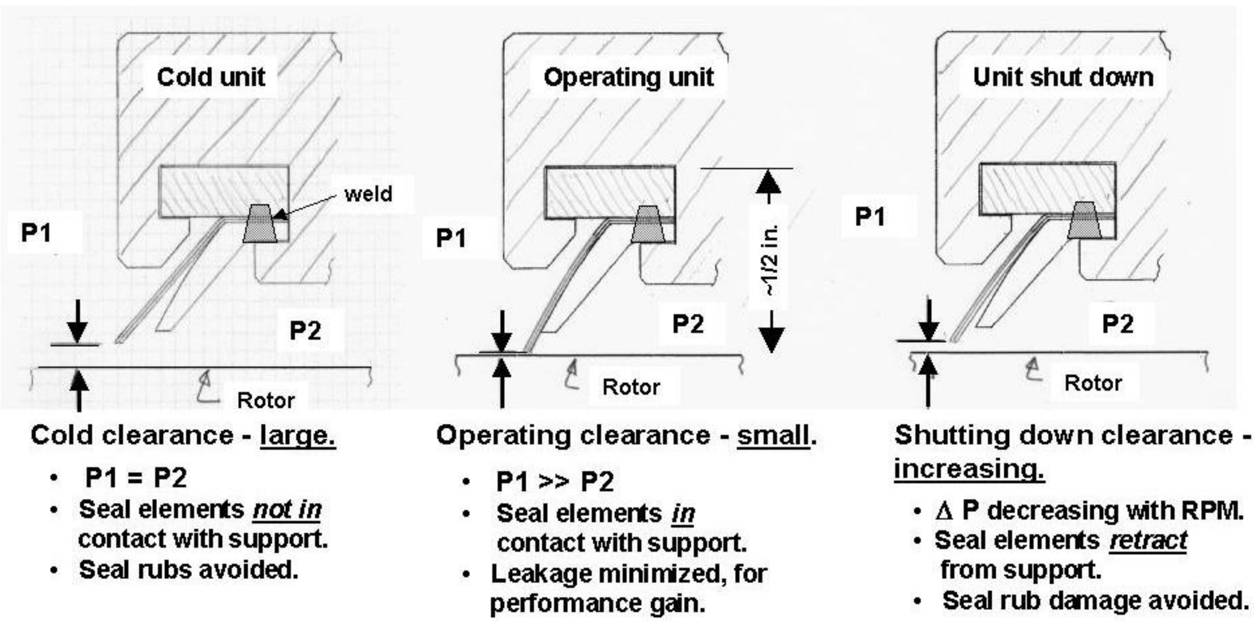



**Figure 2. Pressure Actuated Leaf Seal Functional Characteristics.**

The primary objective of this project 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 was conducted with air at ambient temperature. Test conditions were sufficient to demonstrate concept viability for eventual use in power generation equipment.

Transition from a large cold clearance to a small operating clearance with a rotating shaft was demonstrated with PALS designed, fabricated and rig tested. Initially two-dimensional leaf seal segments of various leaf thicknesses were tested to observe and correlate seal closure with applied pressure. This testing identified means to control leaf oscillations that can occur during seal clearance change as pressure is applied. Other important lessons learned from 2-D testing were incorporated in the design and fabrication of three-dimensional seals subsequently flow tested in a rotating test rig. Measured PALS leakage flow test results are plotted in Figure 3 along with calculated 4-tooth labyrinth seal leakage at clearances found in steam turbines, 0.015 inches and gas turbine packing, 0.025 inches, for comparison. PALS seal leakage exceeds the project goal of 50% reduction relative to the larger labyrinth seal clearance at design seal pressures and approaches a 50% reduction relative to the 0.015 inch labyrinth seal clearance.

The test seal cited had an initial rotor rub indicating a very small, worn-in, running seal clearance. Static seal pressurization before and after testing confirmed that the seal was not in contact with the rotor. Constant seal leakage throughout a one-hour test run at 165 psid and 305 ft/sec. affirmed non-contacting seal operation.

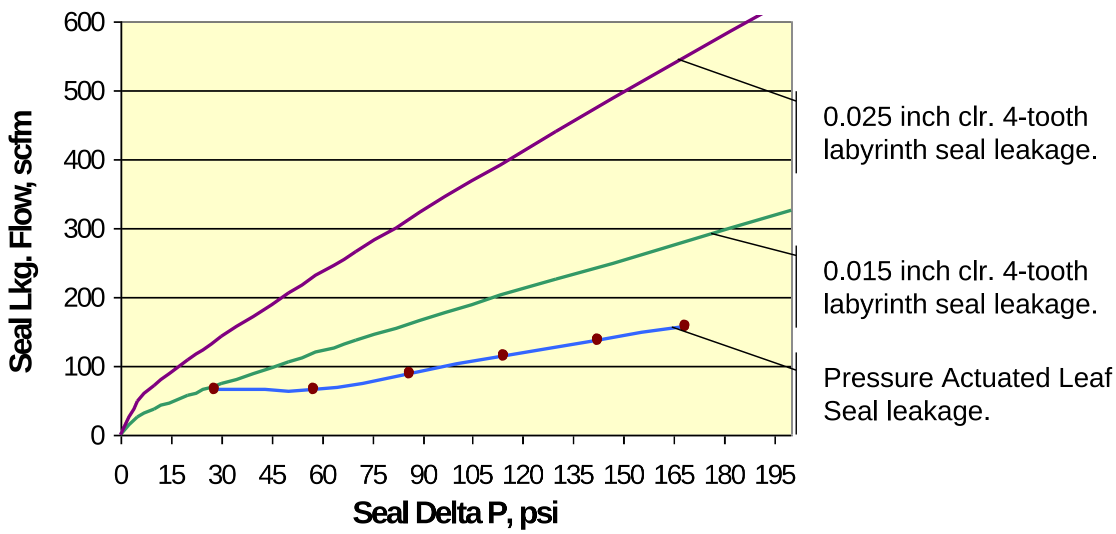



**Figure 3. Pressure Actuated Leaf Seal Leakage Compared.**

PALS rub-tolerance was demonstrated in 2 intentional rub tests. Measured seal wear was considerably less than the amount of seal displacement. Test results show that PALS seal leaves can accommodate transient deflection without permanent deformation which occurs whenever traditional fixed labyrinth seal teeth rub. Rubbing wear of candidate seal leaf materials was evaluated in a study conducted in the RPI Tribology Lab.

The performance benefit from improved shaft sealing in power generation steam turbine applications was evaluated in a study conducted by Encotech, Inc., of Schenectady, NY, using its predictive models and steam turbine audit data. A discussion of the study is included in Section 4.

# TWO-DIMENSIONAL STATIC SEAL TESTING

## SECTION 1

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 Figure 1 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

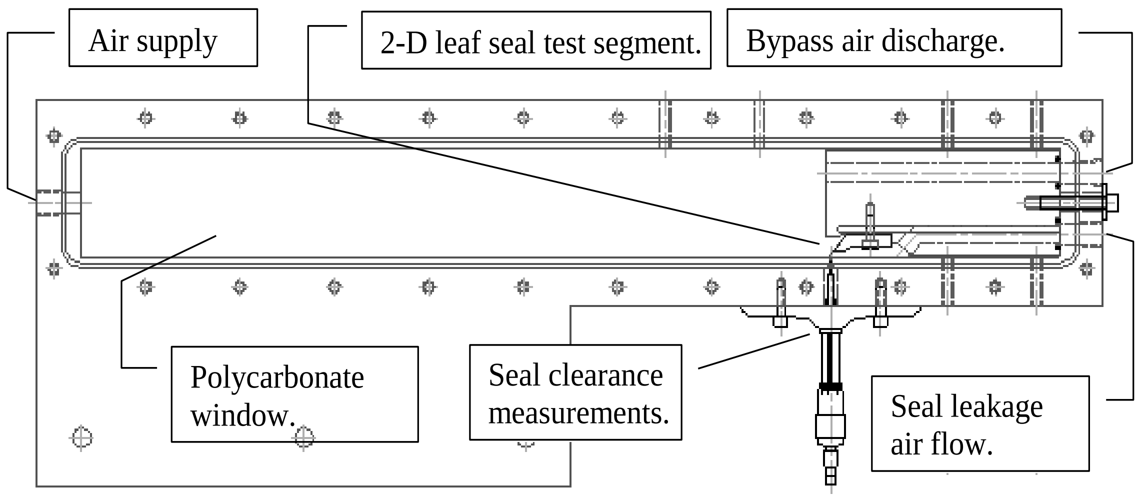



**Figure 4. Two-Dimensional Pressure Actuated Leaf Seal Test Rig.**

and tested. A depth micrometer, show in Figure 4, was used to measure change of leaf tip position, i.e. seal clearance. A micrometer was electrically isolated from the aluminum frame. When the micrometer slide made contact with leaf tips it completed a continuity test circuit and accurately measured operating seal clearance. Seal leakage was measured but was of secondary interest compared to measuring leaf deflection and movement. Audible oscillations, when present, were recorded.

| Alloy: | Nitronic 60 | Hanes 25 | Hastelloy C-276 | Inconel X-750 | Inconel 718 |
|---|---|---|---|---|---|
| Thickness, in. | | | | | |
| 0.004 | X | | | | |
| 0.010 | | X | X | X | |
| 0.012 | | | | | X |
| 0.014 | | | | | X |
| 0.016 | | | X | | |
| 0.018 | | | | X | |

**Table 1. Tested Leaf Seal Alloys and Thickness.**

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 and thicknesses tabulated in Table 1 with the required tensile properties were obtained from commercial vendors thus expensive re-rolling charges were avoided. The Nitronic 60 material was purchased from High Performance Alloys of Tipton, Indiana. Elgiloy Specialty Metals of Elgin, Illinois, supplied all of the other test materials free of charge. Hanes 25 material is a typical brush seal bristle material with good wear resistance and high temperature properties. The HA 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. The 0.010in thick C-276 shim was supplied with spring temper hardness of RC 31and 142 ksi yield strength. The 0.016 in thick C-276 shim was supplied in the annealed condition with 58 ksi transverse yield strength. It was not used because shim stress of that thickness exceeds material yield strength when bent into compliance with the 2.5-inch support radius. Inconel 718 and Inconel X-750 were selected because of their high temperature capabilities. Shim of both Inconel alloys was supplied in the annealed condition and precipitation hardened by heat treatment per AMS 5596 and AMS 5598 for 150 ksi and 115 ksi RT tensile properties, respectively. Nitronic 60 material was selected because it is a non-galling high-temperature stainless steel. It was supplied in a work hardened condition with 126 ksi yield strength.

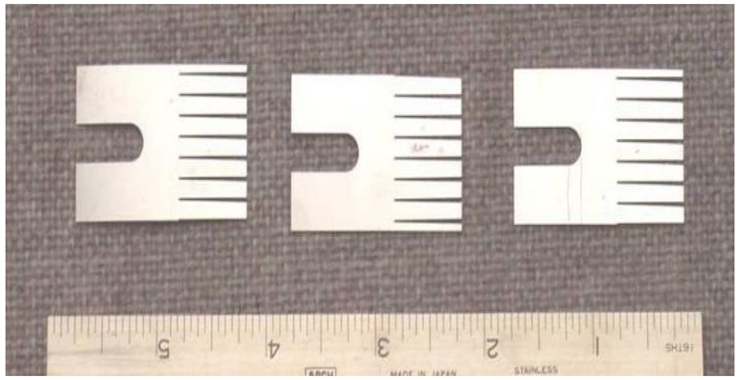


**Figure 5. Two-Dimensional Leaf Seal Segments.**

Two-dimensional leaf seal segments, shown in Figure 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 Figure 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

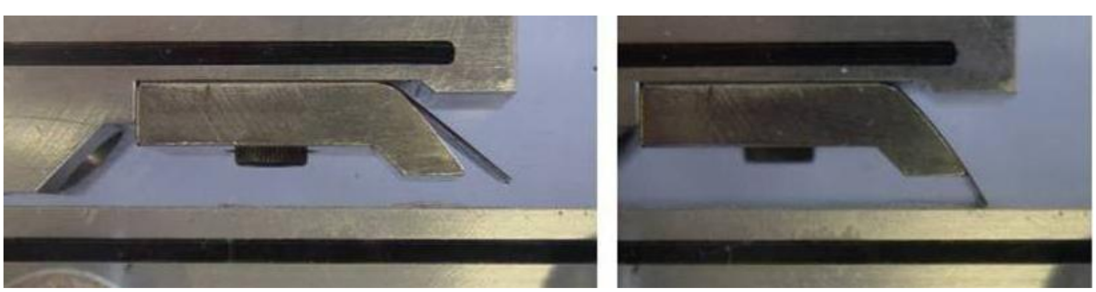


**Figure 6. 2-D Pressure Actuated Leaf Seal Clearance Change.**

test frame.

Stable clearance change of more than 0.040 inches from an inoperative condition to a small operating clearance was evaluated by testing numerous combinations of seal leaves at various pressurization levels and flow rates. Unstable seal actuation that occurred with the application of increased pressure and flowing air was evident with audible seal leaf oscillations or by the failure of the 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 oscillations during pressurization. Seal fabrication, assembly, and testing provided insight to other features that also contribute to stable actuation and low leakage.

Two–dimensional seal testing confirmed that seal clearance change with pressure is a strong function of leaf thickness as illustrated in Figure 7. The curve on the left is that of 2 seal leaves 0.004in thick. Their response to small differential pressure was intended by design 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 0.010in 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 Figure 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, contributes to a smooth transition in proportion to applied pressure. In contrast, when by-pass flow is not present, test seal actuation is abrupt because closure of seal leaves increases upstream pressure accelerating seal closure.

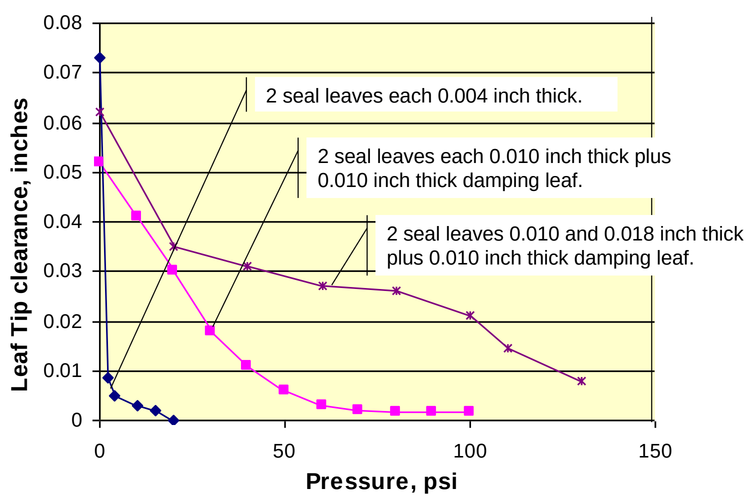



**Figure 7. Clearance Change; Different Leaf Thickness.**

The influence of approaching air flow direction on stable seal actuation, i.e. change in seal clearance as pressure is applied, was evaluated and shown to have little effect. The open air flow approach to 2-D leaf

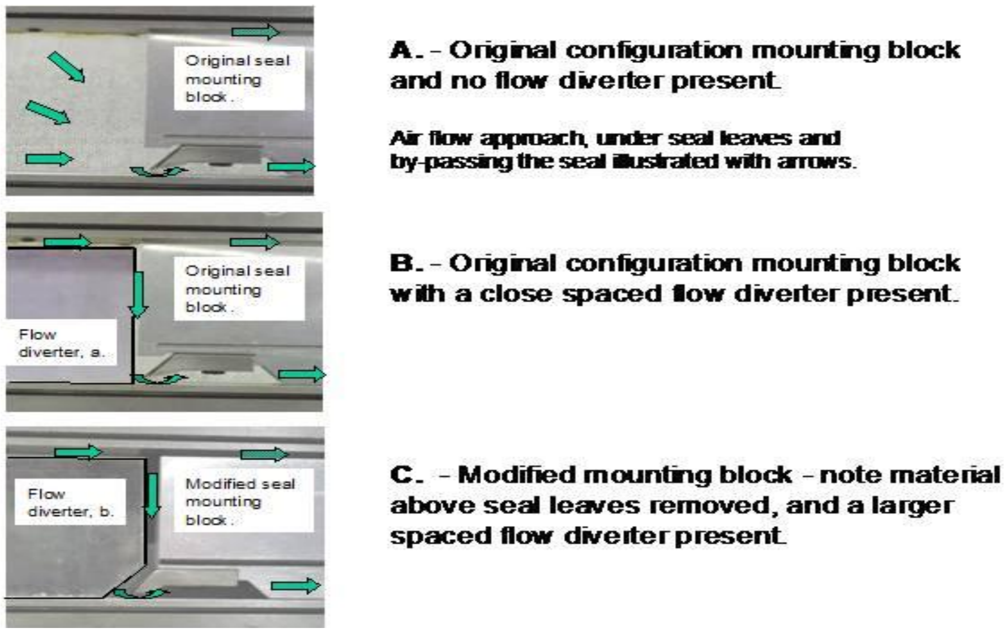



**Figure 8. 2-D Air-Flow Approach Configurations.**

seal segments initially tested is illustrated in Figure 8, panel A. Flow approaching a shaft seal adjacent to a wheel would, however, be in a radial passage. That condition was modeled using a flow diverter as shown in Figures 8, panels B and C. The close spaced flow diverter in Figure 8-B, (0.095 inch), vs. wider spacing in Figure 8-C, (0.190 inch), provided an indication of sensitivity to radial velocity on seal leaves. Figure 8, panel C, includes a modification made to the mounting block where 0.5 inch was removed from the leading edge exposing the entire seal leaf upper surface to the approaching air flow. These tests showed minimal change in seal closure as pressure is applied. The most prominent differences occur at less than ~ ½ the fully actuated seal pressure as shown in Figure 9.

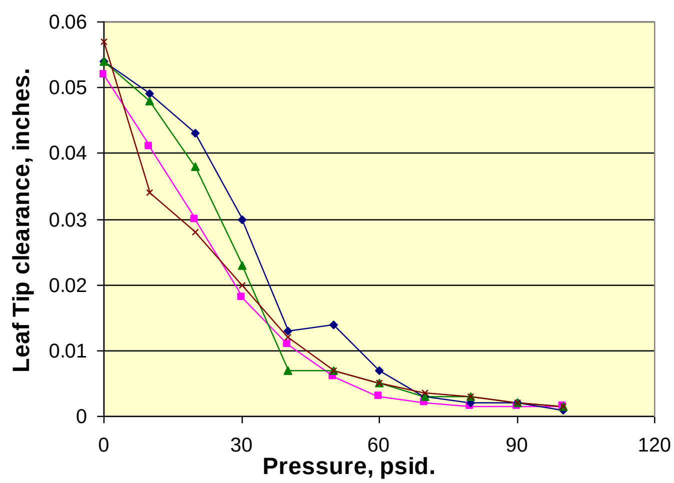



**Figure 9. 2-D Air Flow Effect On Seal Actuation.**

## THREE-DIMENSIONAL PROTOTYPICAL STATIC AND DYNAMIC TESTING

## SECTION 2

### ROTATING TEST RIG DESIGN

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 Figure 10. 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 an axial direction 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 and relative to the tapered rotor OD. Rotating rig components are shown in Figure 11 as prepared for assembly.

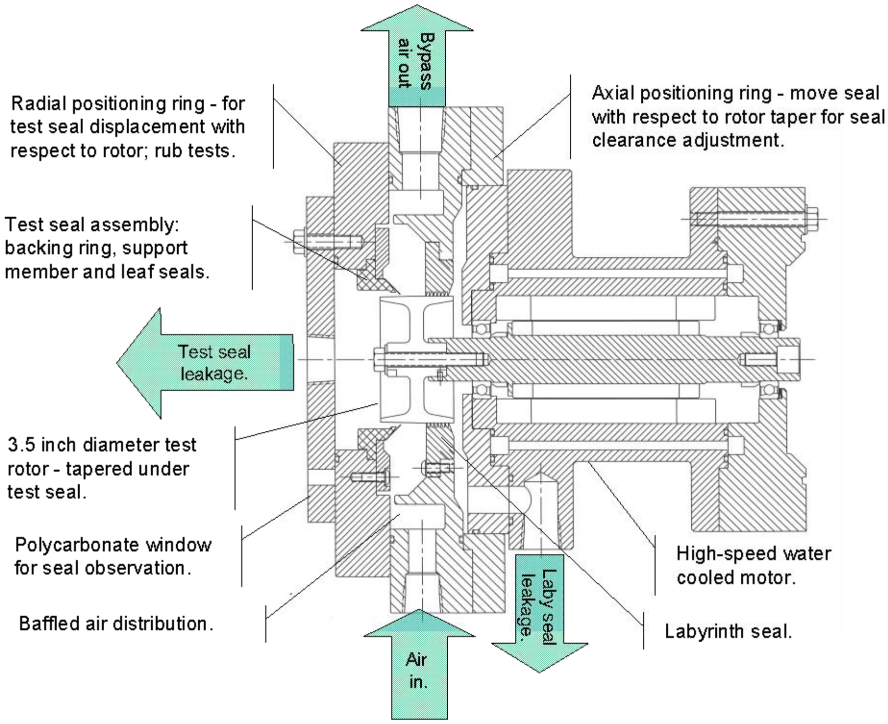



**Figure 10. Rotating Test Rig Cross-section.**

**Figure 11. Rotating Rig Components.**

Test rig instrumentation included a Fox Thermal Instruments flow meter, 2 pressure transducers, and RTD’s for use with a USB data acquisition interface with a laptop computer. Air supply and exhaust pressure gauges were also used for 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 up stream piping.

## TEST SEAL DESIGN

The 3.5 inch test seal diameter selection was based on the geometry of a possible OEM demonstration application; desire for seals to be of moderate size and expense; capable of operating at up to 200 psid seal

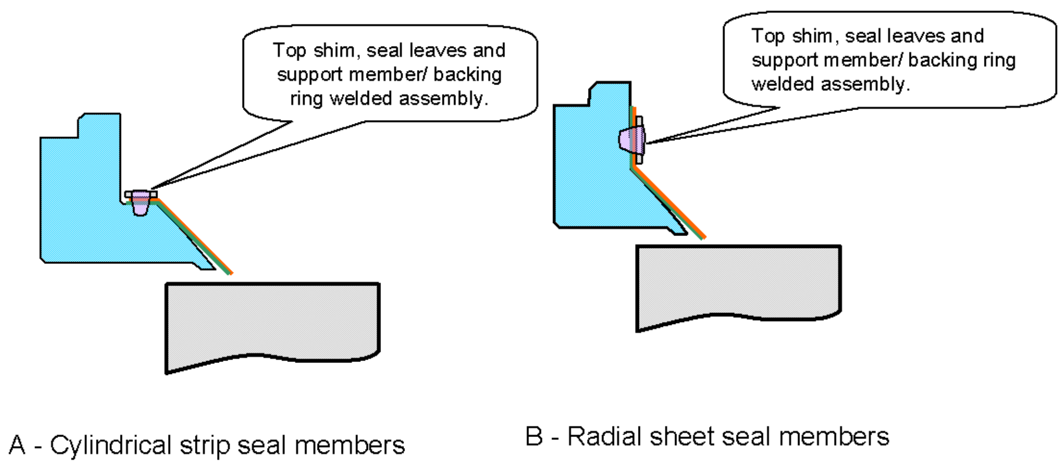



**Figure 12. Pressure Actuated Leaf Seal Design Configurations.**

pressure and 500 ft/sec seal speed. Two seal configurations, illustrated in Figure 12, were established. Seal leaves cut and bent from a strip are laid in tangent contact with the curved surface of the support ring in the first configuration. The unslotted portion of the seal strip is wrapped around a cylindrical shoulder on the seal support ring and welded to it. In the second configuration seal leaves are cut in a sheet and bent out of plane to assemble tangent with 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 2. 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 in Hastelloy C-276 material for Seal # 3 but leaves were not bent or assembled into a test seal. As shown in Figure 12, 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

| | 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 2. Rotating Rig Test Seal Configuration Matrix.**

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. Those changes facilitated test seal assembly builds discussed below to address leaf bending and ‘nesting’ issues that were required for good sealing. This project phase focused on assembly and testing of Seal #2 in Table 2 utilizing 0.010 inch thick Haynes 25 material. The support ring was made from 321 SS with 9.3 x$10^{-6}$ in/in-°F coefficient of thermal expansion, intentionally chosen to be larger than stator parts made of 17-4 PH SS with coefficient of thermal expansion ~ 6x$10^{-6}$ in/in-°F. Stress and twist deflection analysis of the support ring was conducted. The section geometry specified limited seal closure from twist to less than 0.002 inch at 200 psid seal pressure.

## TEST SEAL ASSEMBLY AND INITIAL TESTING

Strip cut seal leaves for an initial 3-D test seal are shown in Figure 13. Seal leaves are bent 45 degrees with respect to the band portion of the strips and assembled to the seal support ring illustrated in Figure 12-A. An assembly fixture was used to hold seal strip leaves in contact with the support ring during seal

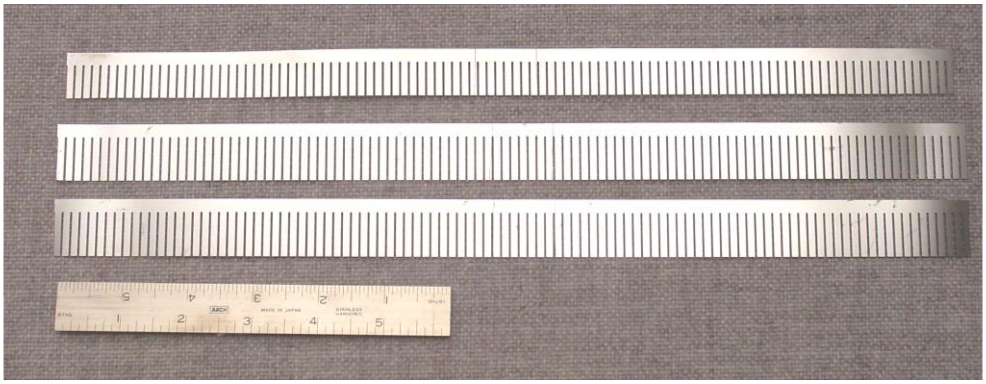


**Figure 13. 3-D Strip Cut Seal Leaves.**

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 are 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 fit over 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 as shown in Figure 17. A final step in preparing test seals was machining of seal tip ID. A fixture was used to deflect seal leaves into their 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. Typical support ring run-out with respect to the rotor was less than +/- 0.001 inch.

Intended seal clearance change with application of pressure was demonstrated in tests and is qualitatively observed in the series of photos in Figure 14. A high intensity light illuminates leaf seal clearance that is seen to decrease as seal pressure is applied. The test seal shown in Figure 14 is composed of 0.004 inch thick Nitronic-60 top leaves, 0.010 inch thick HA25 bottom seal leaves, and damping leaves.

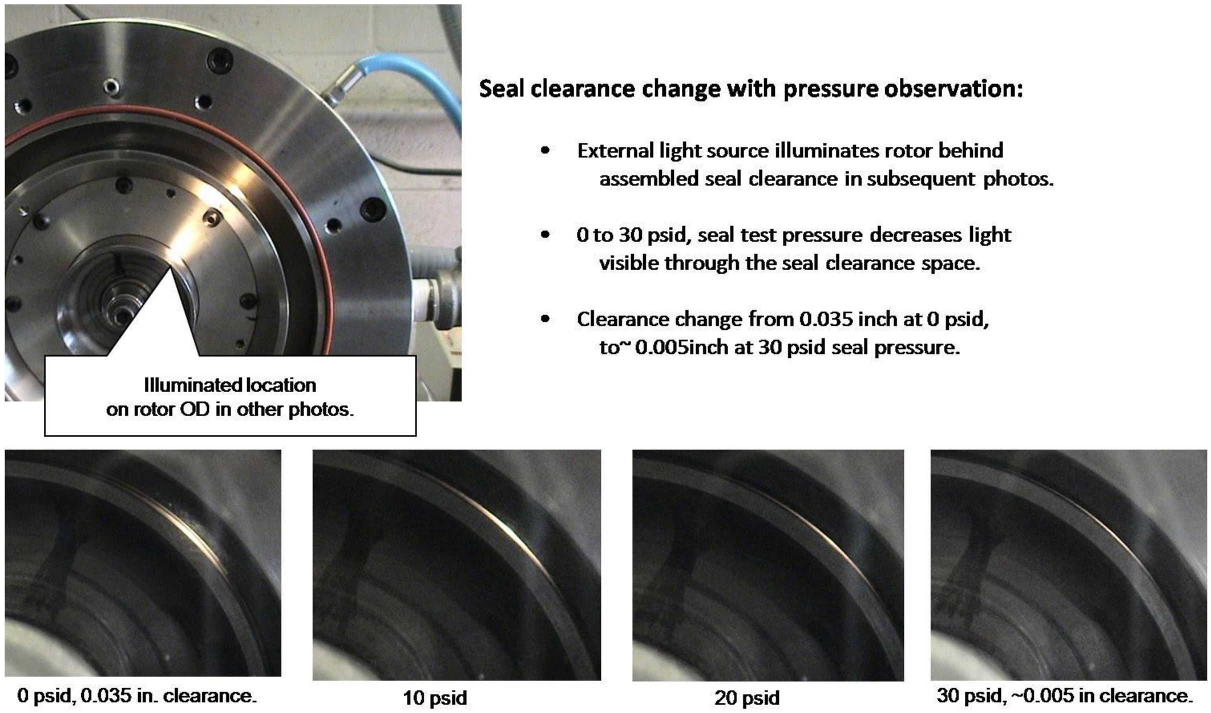



**Figure 14. Seal Clearance Change with Pressure.**

## SEAL LEAKAGE FLOW TEST RESULTS

Test seal #2-5 incorporated the fabrication and assembly lessons cited above and demonstrated reduced seal leakage that met all project goals. This seal, shown in Figure 15, is composed of 2 seal leaf layers that are 0.010 inch thickness Haynes 25 alloy with a shorter damping leaf of the same material beneath them. Good leaf nesting and interleaf contact is visible in Figure 16. Alignment of top leaves with bottom leaf slot centerlines can be improved in future product development test seals and is expected to decrease interleaf leakage somewhat.

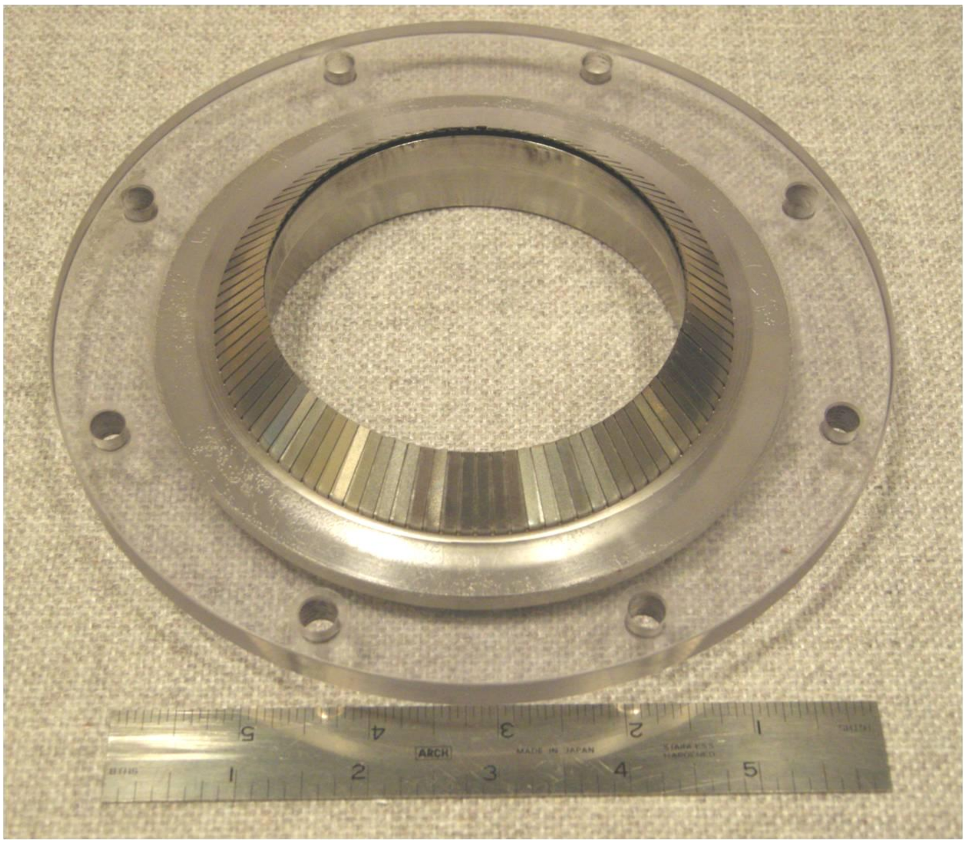


**Figure 15. PALS Test Seal #2-5, After Testing.**

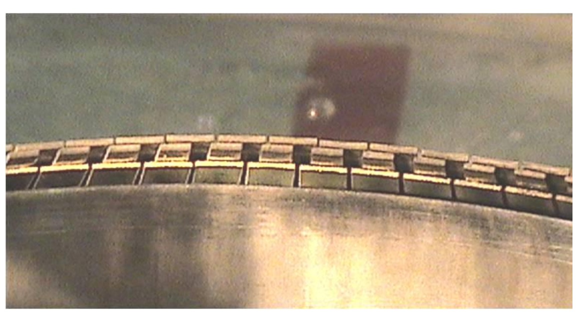


**Figure 16. Good Inter-Leaf Contact.**

Test seal #2-5 seal leakage is presented in Figure 17 as a function of seal pressure along with calculated leakage of 4-tooth labyrinth seals with 0.015 inch and 0.025 inch clearances commonly found in steam and gas turbines, respectively. Also plotted is the 7-tooth labyrinth test rig seal leakage, as measured and calculated. PALS leakage is less than the 7-tooth test rig labyrinth seal leakage with 0.0095 inch clearance

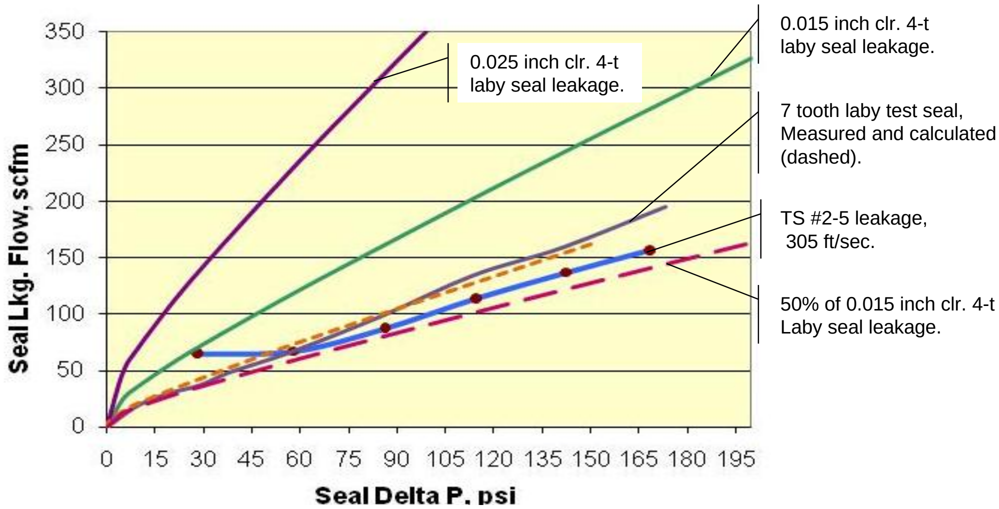



**Figure 17. Seal Leakage Flow Reduction vs. Labyrinth Seals.**

and considerably less than the 4-tooth labyrinth leakage flows with 0.015 inch and 0.025 inch clearance. Good agreement of calculated 7-tooth labyrinth leakage with measured 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.

PALS actuation with seal pressure is seen in the plot of seal leakage vs. pressure in Figure 18. Seal leakage

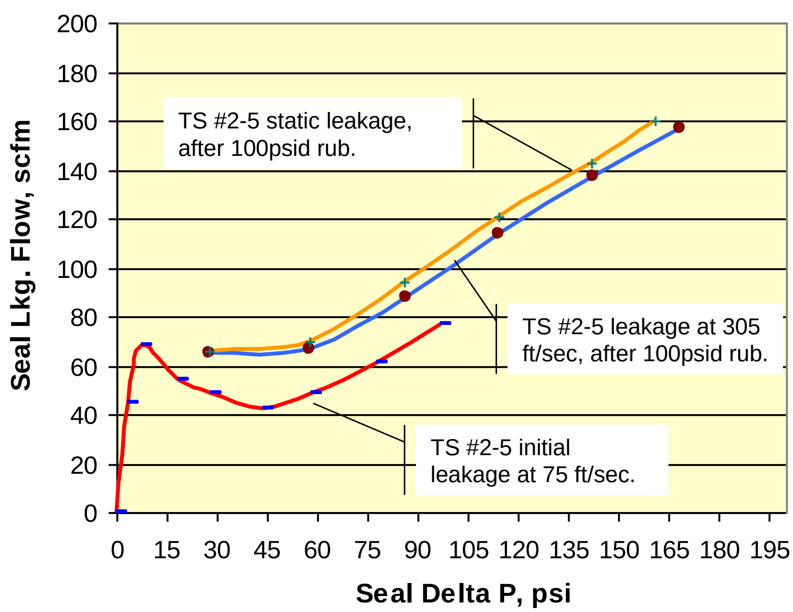



**Figure 18. Test Seal #2-5 Seal Leakage.**

flow increases rapidly at low seal differential pressure when seal clearance is large, ~0.035 inches. 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 compares favorably with a test seal that had inter-leaf leakage blocked with adhesive tape, indicating Test Seal #2-5 has little inter-leaf leakage.

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 seal inspection under microscope showed a small burr on the bottom leaf tips displacing top leaf tips a little, opening more gaps for inter-leaf leakage. Seal leakage shown in Figure 18 increased in both static and rotating tests after the rub.

Plotted seal leakage at 305 ft/sec, in Figure 18, is slightly less than measured seal leakage with the rotor at rest. The flow reduction at speed is caused by rotor rim elastic growth with speed. Radial rim growth at test speed is calculated to be less than 0.0004 inch but sufficient to account for the flow reduction. Figure 19 plots of seal leakage vs. speed at 2 seal differential pressures also show that speed has little effect on PALS seal leakage. Static flow testing may be adequate for most aspects of ongoing PALS development.

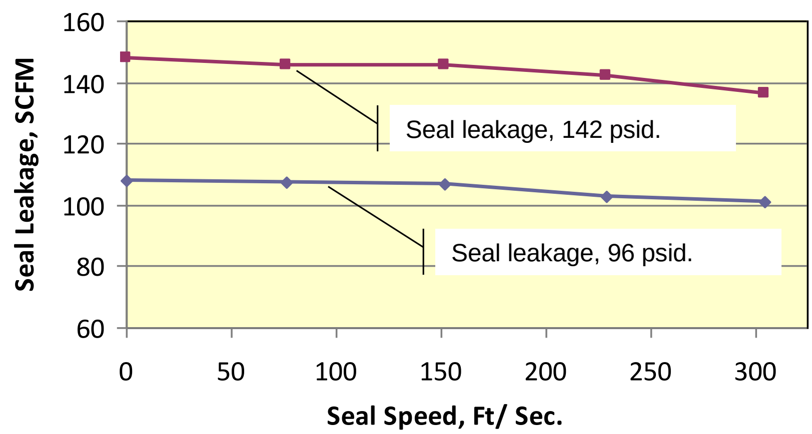



**Figure 19. Test Seal #2-5 Seal Leakage Vs. Speed.**

Test Seal #2-5 leakage plotted in Figures 17 and 18 includes a one-hour run at 305ft/sec rotor speed and 165 psid. Seal leakage did not change throughout the hour at constant test conditions. That relatively short endurance run provided assurance that damaging high-cycle fatigue was not present and that the seal was not in rubbing contact with the rotor.

## SEAL RUB TEST RESULTS

Test Seal #2-5 was subjected to two intentional seal rubs, i.e. rotor interference events, to evaluate PALS rub tolerance and characterize rotor and leaf wear resulting from rubbing interference. Radial misalignment of the seal with respect to rotor simulates potential field installation misalignment. This was accomplished by displacing the seal housing relative to the rotor axis a specified amount: 0.005 inch for the first test and 0.008 inch 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 Figure 20: prior to the 0.005 inch seal displacement, 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 seal rub test, the seal was axially displaced 0.062 inches to reduce 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 0.005 inch seal rub was ~0.0005 inches as shown in Figure 23. That wear track location includes wear from the initial rub previously discussed.

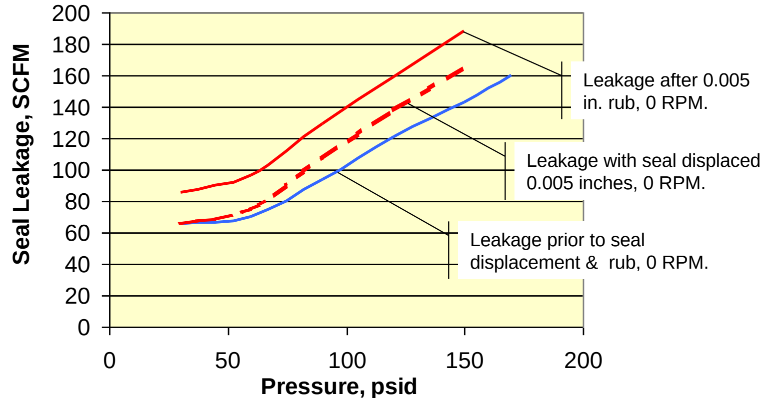



**Figure 20. 0.005 inch Seal Displacement Rub Test Leakage.**

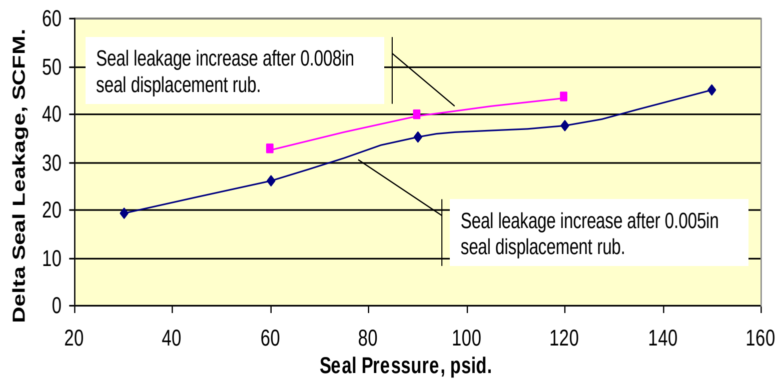



**Figure 21. Rub Test Seal Leakage Increase.**

The second seal rub test, with a 0.008 inch seal displacement, and the same test sequence and higher peak seal pressure of ~150 psid, produced very similar change in seal leakage as shown in Figure 21. Test seal #2-5 ID with seal leaves depressed was measured by coordinate measurement system (CMS) after the

0.005 inch displacement rub and again after the 0.008 inch displacement rubs. These measurements are plotted in Figure 22. The change in seal ID is 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 with the seal leakage change observed. The second rub test was conducted at a clean, axially displaced location on the rotor. That wear track is ~0.0003 inch deep as shown in Figure 23.

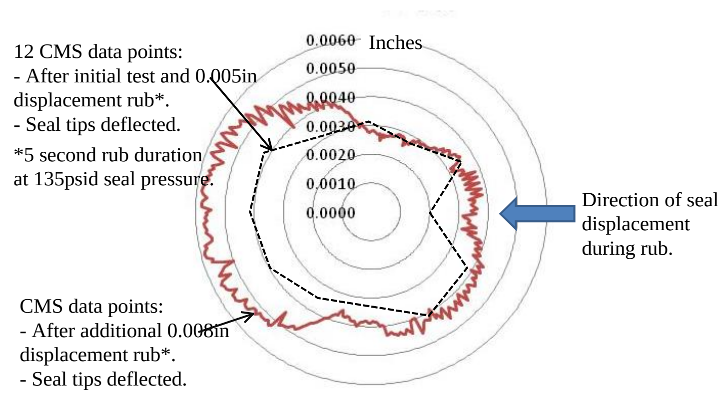



**Figure 22. Rub Test Seal Leaf Wear.**

In both rub tests, wear of the HA25 leaf tips was ~0.001 inch, much less than the 0.005 inch and 0.008 inch seal displacement. Fixed labyrinth seal tooth wear would likely have been comparable 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. Since seal wear and leakage change is approximately the same for this 2X change in seal displacement rub depth, seal wear was more dependent on applied load, from seal delta P, and speed than the amount of seal interference with the rotor. The maximum rub test seal pressure was 135 psid in the first test and 150 psid in the second rub test. Both rubs were imposed for 5-second duration at 305 ft/sec seal speed (~1500 linear feet). These conditions are not correlated with known power generation equipment startup / shutdown transient data, but the modest 0.001 inch, seal wear suggests a reasonable degree of rub tolerance for the Pressure Actuated Leaf Seal concept. A leaf and rotor material wear pair is needed; however, 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 may eliminate rotor wear vs. Haynes 25 leaves. Alternatively, leaf tips fitted with abradable material may provide a wear pair that would not score un-coated rotor materials during an inadvertent seal rub. It is also possible that this rub test load, speed, or duration is excessive with respect to application specific requirements.

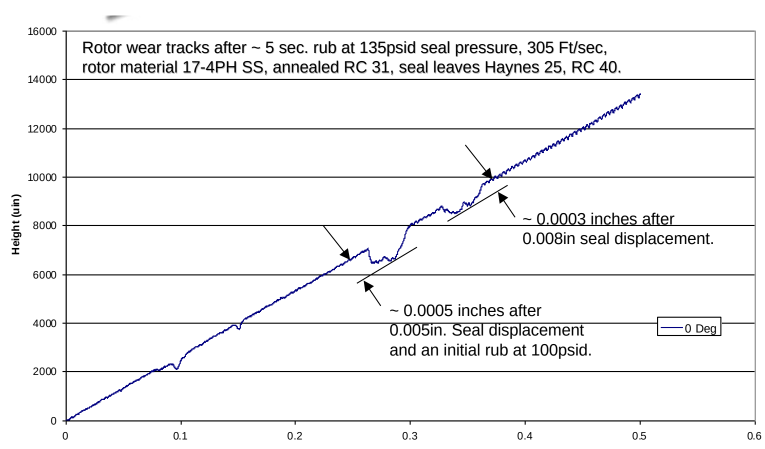



**Figure 23. Rub Test Rotor Wear.**

## LEAF MATERIALS WEAR CHARACTERIZATION
## SECTION 3

An evaluation of the rub tolerance and wear characteristic of candidate leaf seal materials against a relevant, 17-4 PH stainless steel, rotor material was conducted in the Rensselaer Polytechnic Institute Tribology Laboratory, Troy, New York, under the direction of Thierry Blanchet, Associate Professor of Mechanical Engineering. Five candidate seal materials (Nitronic 60, Haynes 25, Hastelloy C-276, Inconel 718, and Inconel X-750) were evaluated. The wear specimen design replicated the geometry of a single seal leaf. Test rig featuring oriented the leaf with respect to the rotor rim at the same angle a Pressure Actuated Leaf Seal leaf has with a shaft seal surface. These wear tests were conducted on a modified WAM-1 tribo-tester. A two-by-two test matrix of high and low sliding speed (100 or 40 $^{ft}/_{sec}$) and high and low contact pressure (430 or 43 psi) was conducted for each of the 5 materials. Plots and tables compare the different materials against each other while holding each of the four sliding condition combinations constant to provide a rough ranking of candidate leaf materials. Since test seal materials were of different thicknesses, the estimated equivalent seal pressure at constant contact pressure varied as shown in Table 3, assuming there are 2 seal leaves.

| Wear test contact pressure: | | 43 psi | 430 psi |
|---|---|---|---|
| **Material** | **Thickness, inches.** | **Seal DP, psid.** | **Seal DP, psid.** |
| **Nitronic 60** | 0.004 | 6 | 63 |
| **Haynes 25** | 0.01 | 16 | 159 |
| **Hastelloy C-276** | 0.01 | 16 | 159 |
| **Inconel 718** | 0.014 | 22 | 222 |
| **Inconel x-750** | 0.0175 | 28 | 278 |

**Table 3. Wear Test Materials and Estimated Equivalent Seal Pressure.**

Inconel 718 performed the best for all test conditions except the high pressure and low rotational speed combination where Nitronic 60 test results were better. That Nitronic 60 test was the first conducted, and re-testing was suggested to confirm the result. The observed rub distances and loss of material, even at the higher load, are small relative to the limited duration of transient rubs that a Pressure Actuated Leaf Seal is expected to experience when equipment traverse rotor critical speeds during startup and shutdown. The seal leaves in rub tests, discussed in Section 2 above, were of Haynes 25 material 0.010 inch thick. Those intentional rubs were at 305 ft/sec, substantially higher than the wear testing done at RPI, but the seal delta pressure of 135 to 150 psid corresponds reasonably well with the higher contact pressure of the wear testing program. Seal leaf rub distance in ~5 seconds at 305 ft/sec is ~1500 ft (~460 meters) resulted in max seal wear of ~0.002 inches. Extrapolating the reduced loss of Haynes 25 material that occurs at 100 Ft/sec vs.

40 Ft/sec and high contact pressure, the 0.002 inch seal wear (~0.05 mm) in ~500 m rub distance, appears plausible.

The temperature sensitivity of material wear was not addressed in this study but is of concern for high-temperature applications. Preferably, seal rubbing wear should be verified at application temperature and the most relevant conditions of shaft speed and seal pressure where a rub is anticipated. This study, showing a preference for Inconel 718, did not reveal a big distinction among leaf materials in rubbing wear against the 17-4PH stainless steel rotor.

# COMMERCIALIZATION PLANS.

# SECTION 4

## PERFORMANCE BENEFIT ASSESSMENT.

To better understand the commercial potential of Pressure Actuated Leaf Seals in power generation steam turbines, a study was conducted by Messrs. Fred Kindl and Paul Roediger, of Encotech, Inc., 207 State Street, Schenectady, NY. The performance benefit assessment used their database of past Steam Path Audit seal clearances and software that assigns power, heat rate, and fuel cost penalties to each type of deterioration on a stage-by-stage basis. The study utilized Encotech's Etracker-PE Computer Program that analyzes the entire power generation cycle and is capable of modeling change of seal type and clearance and then calculating the impact of those changes on the overall efficiency or heat rate and output.

A starting point for the study was a statistical survey of labyrinth seal clearances found during the steam path audit of 27 steam turbines entering overhaul. Figure 24 is a histogram of measured shaft end packing, EP, clearances from their survey. Rubbed packing clearance is frequently 3 to 4 times larger than OEM design clearance of 0.015 inch. Histograms of high-pressure, HP, and intermediate-pressure, IP, turbine shaft packing are similar.


**Figure 24. Shaft End Packing Clearance.**

The study evaluated the combined performance change of rubbed HP, IP, and shaft end packing at 0.010 inch intervals up to 0.080 inch clearance in a 730 MW impulse type steam turbine with 2400 psia inlet pressure, 1000°F inlet temperature ,and 1000°F reheat temperature. The expected performance benefit of reduced shaft leakage using PALS at the same locations, i.e., a single seal tooth at clearances from 0.001 inch to 0.020 inch PALS seal leakage relative to a labyrinth seal, is a function of the number of labyrinth seal teeth it replaces and clearance of both seals. A 50% seal leakage with PALS packing relative to a 4-tooth labyrinth with 0.015 inch clearance would require an effective PALS clearance of ~0.005inches. Report plots show performance change with PALS clearance increases faster than the multi-tooth labyrinth seals they replaced. This is expected since they seal with a single tooth; however, with the rub tolerance shown in tests conducted (~0.001 inch clearance increase), only minimal change in PALS clearance is anticipated with infrequent upset rub events. Startup and shut down rubs that hazard conventional labyrinth seal are avoided with the PALS design. The Etracker-PE computer program calculation of performance change with rubbed labyrinth seals for this turbine was less than expected in light of performance benefits cited by others in the Metrics Report below. A heat rate increase of 25 Btu/Kw-hr was calculated for all study labyrinth seals rubbed to 0.06 inch clearance vs. 0.015 inch design clearance from unit cycle heat rate of 7977 Btu/Kw-hr. That change is a 0.31% loss in overall heat rate. While that value is less than anticipated, it is in agreement with GE Power Generation Reference Library, GER-3713E, “Advances in Steam Path Technology,” by Cofer, Reinker and Sumner, where a 0.35% loss in unit heat rate was calculated for a situation where the internal mid-span, N2 packing, and first 3 HP and IP stages of diaphragm packing of a 500 MW reheat unit were rubbed to 0.060 inch. vs. 0.015 inch design clearance. The Etracker-PE program calculated performance benefit of PALS at 0.005 inch clearance vs. labyrinth seals rubbed to 0.06 inch is a 0.33% improvement in heat rate and 0.15% increase in generator output with 0.32% less main steam flow.

In addition to the Etracker-PE study, several additional cases were run using Encotech’s eSTPE (Encotech Steam Turbine Performance Evaluation) program. That program quantifies performance on a stage-by-stage, component-by-component basis at constant unit steam flow. The performance impact of rubbed N2 packing and three HP stages of inter-stage packing in a 550 MW, 3500 psi, opposed flow steam turbine was evaluated. Assuming those labyrinth seals are rubbed to a clearance of 0.060 inch vs. 0.015 inch design clearance, unit heat rate increase was 0.45% (7443 vs. 7409 Btu/kWh) with 0.98% loss in output. This performance impact, without consideration of IP stage inter-stage packing, is greater than that calculated for the 730 MW, 2400 psi steam turbine evaluated with the Etracker program. The net performance benefit from retrofit of labyrinth shaft packing with PALS in this steam turbine is ~ 0.5% improvement in heat rate and 1% in output as shown in Figure 25. The Etracker-PE assessment of a 730 MW, 2400 psi steam turbine performance is Case 1 and the eSTPE analysis of the 550 MW, 3500 psi unit is Case 2.

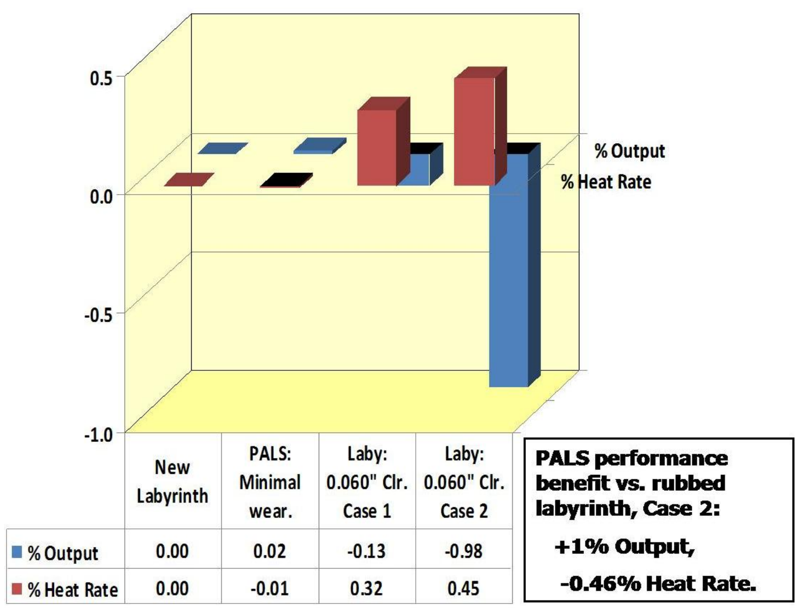



**Figure 25. PALS Steam Turbine Performance Benefits.**

The different steam turbine pressure, size and configuration were cited as contributing to the Case 2 performance difference from Case 1. It is apparent that steam turbine performance benefits are highly dependent on unit specific design conditions. These PALS performance results are consistent with the range of GE steam turbine performance benefits listed in Table 4 for brush seals installed in various locations as reported in "Fundamental Design Issues of Brush Seals for Industrial Applications," ASME 2001-GT-0400 by Dinc, Demiroglu, Turnquist, Mortzheim, Goetze, Maupin, Hopkins, Wolfe, and Florin.

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

**Table 4. Brush Seal Performance Benefits in Steam Turbines. (Source: ASME 2001-GT-0400.)**

HP turbine tip seal leakage in the 550 MW, 3500 psi unit was also evaluated with the eSTPE program. Bucket tip ‘spill strip’ seals are typically designed with 0.040 inch clearance but are commonly found rubbed to ~ 0.08 inch in Encotech opening audits. The performance impact of HP stages 2, 3, and 4 tip seal leakage with 0.080 inch clearance vs. 0.040 inch design clearance is 0.3% increase in unit heat rate and loss of 0.58% in Kw output. This relatively large performance loss is typical of Encotech analysis of opening steam turbine clearances in their experience. That observation is consistent with GE report, GER-3731E, “Advances in Steam Path Technology,” noted above, that states HP turbine efficiency losses from tip leakage is 3 times that of shaft packing leakage. Development of PALS for this location has a significant potential steam turbine performance benefit. The combined PALS shaft packing and tip seal performance improvement would approach 0.8% in heat rate and 1.6% in output.

Annual fuel cost penalty of worn labyrinth seals was calculated for the 730 MW steam turbine unit and Etracker, Case 1, performance results. The calculation assumed a fuel cost of $\$2.00/10^6$ BTU, boiler efficiency of 90%, and a 90% capacity factor. Annual increase in fuel cost is plotted vs. seal clearance in the Ebcotech report. The calculated annual fuel cost penalty of HiLo labyrinth packing worn to 0.060 inch clearance vs. 0.015 inch design clearance, Case1 heat rate loss of 0.32%, is $330,000. At a fuel cost of

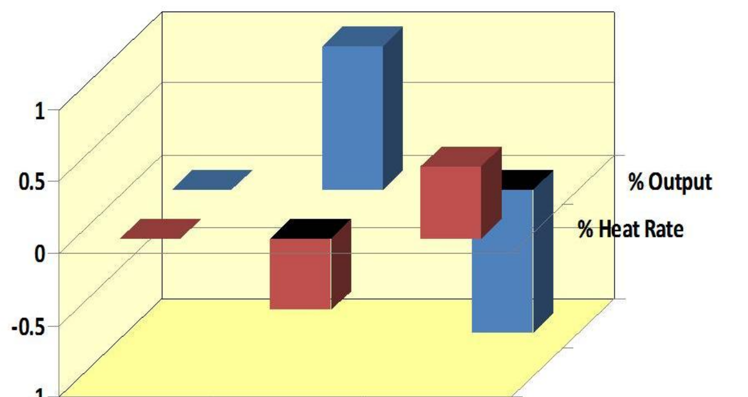
$6.00/10$^6$ BTU, more typical of natural gas, that would be ~$1,000,000/year fuel cost penalty. Installation of PALS packing that avoids damaging seal rubs, would save the $1,000,000/year fuel cost. Retrofit of PALS in the Case2 unit, with 0.46% heat rate improvement, would produce an annual cost saving over $1,400,000 at the higher fuel cost. If PALS were also applied as bucket tip sealing, the combined heat rate improvement would be ~0.8%, with annual fuel saving of $2,500,000 at the higher fuel cost as well as providing 1.6% more KW output.

Gas turbine performance benefit from Pressure Actuated Leaf Seals was not subject of a separate study. However, performance benefit of brush seal installations in GE large frame gas turbine compressor high pressure packing (HPP), turbine interstage packing (ISTG), and other locations is published information as shown in Table 5 from ASME 2001-GT-0400, previously cited. PALS seal leakage reduction in these locations is expected to be comparable to that of brush seals and produce similar performance benefits.

| Seal location & Turbine | % MW output increase, + & % Heat Rate decrease, -. |
|---|---|
| **High Pressure Packing (HPP):** | |
| MS32G | +0.7% / -0.5% |
| MS32J | +0.7% / -0.5% |
| MS51P | +0.6% / -0.45% |
| MS61B | +1.0% / -0.5% |
| MS71E, EA | +1.0% / -0.5% |
| MS91E | +1.0% / -0.5% |
| **Turbine Interstage Packing (ISTG):** | |
| MS51N | +1.0% / -0.5% |
| MS51P | +1.0% / -0.5% |
| MS61B | +1.0% / -0.5% |
| MS71E, EA | +1.0% / -0.5% |
| MS91E | +1.0% / -0.5% |
| **# 2 BRG forward and aft seals:** | |
| MS71E | +0.3% / -0.2% |
| MS91E | +0.3% / -0.2% |
| **HPP and # 2 BRG combined:** | |
| MS52C | +0.7% / -0.5% |
| MS52D | +0.9% / -0.6% |
| MS71B | +1.0% / -0.5% |
| MS91B | +1.0% / -0.5% |

**Table 5. Brush Seal Performance Benefits in Gas Turbines. (Source: ASME 2001-GT-0400.)**

Rubbed HPP labyrinth packing causes significant performance penalties that can be avoided with PALS packing. It is stated that "In practice, most operating units have HPP clearances significantly higher (20 to 60 mils) than nominal." and lost performance is large: "For a MS7001E unit, a rub of 20 mils on HPP labyrinth seal teeth equates to at least 1.0% loss in unit performance." Source: GE Power Generation Reference Library, GER-3571H, "Performance and Reliability Improvements for Heavy Duty Gas Turbines," by JR Johnston. The combined performance benefit of more than 50% seal leakage reduction, nominally 0.5% heat rate improvement and 1% in output, and avoiding the performance penalty of rubbed HPP labyrinth seal teeth is illustrated in Figure 26. The net gas turbine HPP performance benefit is a 1% decrease in heat rate and a 2% increase in unit output that can be expected from retrofit of PALS in a unit with labyrinth seal packing rubbed only 0.02 inches more than nominal design value.




| | New Labyrinth | PALS | Rubbed Laby, 0.02 inch. |
|---|---|---|---|
| % Output | 0 | 1 | -1 |
| % Heat Rate | 0 | -0.5 | 0.5 |

**Figure 26. PALS Gas Turbine Performance Benefits.**

**COMMERCIALIZATION PLAN.**

The performance benefit assessment confirms significant commercial value of PALS application in steam turbines for both shaft sealing and, potentially, bucket tip sealing. Application of PALS in gas turbines to reduce high pressure packing leakage is expected to produce performance benefits similar to that of newly installed brush seals as discussed above. A strategic plan for the commercial introduction of PALS technology in power generation equipment is outlined here as a means to realizing these performance benefits. Its objective is to identify necessary steps to advance the PALS concept from current seal test results and fabrication experience to product end users who can confidently install and use them. Power generation end users as well as original equipment manufacturers and after-market service businesses are risk averse. Therefore, the following plan is aimed at addressing known concerns to reduce commercial risk:

- Continue PALS development to address product design and durability issues. A CMG Tech proposal submitted under NYSERDA PON 1200 has been selected for funding of a follow-on product development phase. It is aimed at mitigating risk through seal testing at higher temperature, 1000°F, and speed, 1000 ft/sec, to confirm seal operability at conditions more typical of power system turbines. Included in the scope is development of seal elements that initially ‘wear-in’ to close proximity with the rotating seal surface and then operate at minimal clearance without further rotor contact except during an upset event.

- Fabricate larger diameter, segmented test seals that are readily scaled to power generation diameters. Larger, ~8 inch diameter, seals are planned to be produced and tested in NYSERDA Phase 2. These seals are to be purchased from a commercial seal manufacturer with known engineering capability to produce turbine worthy seal products. While the quantity of test seals is small, insight into manufacturing issues is expected to be an important step in developing a viable commercial supply of PALS for power generation customers.

- Conduct seal tests in power generation equipment or engine test environments that closely approximate power generation equipment. The NYSERDA Phase 2 efforts include consultation and preliminary planning for a future ‘Piggy-Back’ seal test in a Rolls-Royce factory test turbine at conditions similar to those in power generation equipment. Many cycles and hours of testing will demonstrate seal durability and build confidence for power generation users.

- Deploy PALS technology in other rotating equipment to gain operating experience in less severe environments that can bolster confidence in more demanding power system applications. For example, successful deployment of Pressure Actuated Leaf Seals in Dresser-Rand centrifugal compressors and steam turbines under 60 MW in size would be very helpful to the introduction of PALS in larger steam turbines and gas turbines. Other such fields of use include small air compressors, pumps and barrier seal applications.

- Seek out niche opportunities to introduce PALS technology in power generation equipment where there is a particular need and where cost of failure is low. Power system users with installed brush seals may be candidates for retrofit replacement with PALS directly into the brush seal stator slot as a means of introducing them. New unit production of large steam turbines and gas turbines in power generation is several hundred units per year in the US compared to thousands of operating units worldwide. All operating power system units are candidates for improved performance from reduced seal leakage at the time of their major overhaul at intervals of 3 to 6 years. After-market overhaul and service providers are prime candidate organizations to introduce advanced sealing as well as power system OEM's.

- CMG Tech, LLC plans to transition from the current R&D stage to commercialization of PALS technology products through license agreements and strategic partners. Licensees of specific fields of use will have seal application design, manufacture, marketing, sales, and customer service responsibilities for their market segment. A strategic partnership or license with a seal manufacturer and / or engineering service organization is planned for product deployment outside specifically licensed fields of use. CMG Tech, LLC plans to provide technology transfer support and engineering consulting services to licensees as well as continue to promote the technology in technical presentations. Examples of such activity in support of commercialization include:

- Technical paper, AIAA-2005-3985, "Pressure Actuated Leaf Seals for Improved Shaft Sealing," and presentation at the July 2005 AIAA Joint Propulsion Conference, Advanced Seal Technology session.

- Presentation at the NASA Seals Workshop, November 2005, Society of Tribologists and Lubrication Engineers (STLE) Hudson-Mohawk Section and

- Presentation at the 2nd Annual Tech Valley Engineering Symposium in 2006.

- License of Pressure Actuated Leaf Seal technology to Dresser-Rand Company in April, 2008, for use in centrifugal compressors and steam turbines under 60 MW

# METRICS REPORT.

# SECTION 5

Successful demonstration of improved turbine shaft sealing with Pressure Actuated Leaf Seals has significant application potential in power generation equipment in New York State. Seals that replace rubbed labyrinth packing with retractable packing are reported to improve performance in both heat rate and unit output of up 1-2% and 2-3%, respectively. (Source: Brandon Engineering and TurbCare Inc web pages.) Details of seals replaced, and their condition, are not stated, but the performance is assumed to be with respect to a unit with rubbed seals. The calculated benefit of Pressure Actuated Leaf Seals, in Section 4, is about half of the heat rate and output improvement reported by these field retrofit seals that restored original labyrinth packing to the design seal clearance. The difference between calculated and reported performance is not understood, but in either case, there are large performance benefits from reduced seal leakage. They reduce energy cost in $/kWh, air pollutant emissions in lbs/MWh of SOx, NOx, CO2, and particulates. Reduced equipment maintenance expense with durable, long life seals is a further economic benefit, as is the development of new seal technology business activity. The energy, environmental, and economic benefit potential for New York State from development of Pressure Actuated Leaf Seals is illustrated by the following estimates for a 1% decrease in heat rate. These estimates were submitted in the CMG Tech proposal for NYSERDA PON 1042 that funded this feasibility study. They have not been updated, and no estimates have been made of the rate of commercial implementation.

- Economic value of the energy cost savings for a 1% improvement in the heat rate of coal, natural gas and petroleum fired power generation equipment in New York State is estimated to be an annual saving of approximately $40,000,000 in primary energy cost. Approximately 85,000 gigawatthours of annual electricity generation in New York State (26,284 from coal, 44,304 from natural gas and 13,985 from petroleum) is produced from 845 Tbtu of fossil fuels (269.8 coal, 428.5 natural gas, and 145.6 petroleum). Reference 2001 New York State Energy Fast Facts by NYSERDA. New York State electric power delivered fuel prices from Energy Information Administration/State Electricity Profiles 2004, in cents/ million Btu, was used in this calculation (175.6 coal, 652.6 natural gas, and 486.2 petroleum).

- Environmental benefits of a 1% improvement in heat rate reduction in pollutant emission are estimated using the Energy Information Administration/State Electricity Profiles 2004 for New York State. Emissions tabulated for that year are as follows: Sulfur Dioxide – 236 thousand metric tons, Nitrogen Oxide – 77 thousand metric tons and Carbon Dioxide – 57,625 thousand metric tons. The potential benefit of applying Pressure Actuated Leaf Seal in all New York power generation equipment, (assumed to be turbine powered with 1% improvement in heat rate) is an annual reduction of 576 thousand metric tons of Carbon Dioxide, 2,360 metric tons of Sulfur Dioxide and 770 metric tons of Nitrogen Oxide.