PRESSURE ACTUATED LEAF SEALS FOR IMPROVED TURBINE SHAFT SEALING

Clayton Grondahl
CMG TECH, LLC

29 Stony Brook Drive, Rexford, New York, 12148

Abstract

Brush and finger seals running at small effective clearances accommodate relative motion between stator and shaft by resilient deflection of seal elements. Shaft contact, however, causes wear, limiting the life of these seals. This paper presents a leaf seal design where shingle-layered, resilient seal elements do not contact the shaft during steady state operation. Seal elements are designed to elastically deflect with increasing system pressure to close large startup seal clearance to a small, non-contacting steady state running clearance. At shutdown seal elements resiliently retract as differential seal pressure diminishes. Large seal clearance during startup and shutdown substantially reduces the opportunity for contact wear, preserving seal performance at operating conditions. Design features of the Pressure Actuated Leaf Seal (United States Patent 6,644,667 and continuation) will be described and illustrated. Seal stress and deflection calculations for intermediate (~100psid) and high pressure (>1000psid) applications will be presented. Photos of a functional model demonstrating the pressure actuation of leaf seal elements to close clearance with increasing pressure will be shown and progress toward introductory applications and prototype testing discussed. Improvements effected by the Pressure Actuated Leaf Seal include: (1) Non-contacting seal operation with pressure actuated seal engagement / disengagement during startup and shutdown; (2) Performance improvement from extended seal life at small operating clearance; (3) Bi-directional shaft rotation; (4) High differential seal pressure applications and (5) Cost reduction using slotted shim stock seal elements Vs brush seal bristle pack assembly.

I. Introduction

The labyrinth seal is the most widely used shaft packing in today's turbo machinery applications. Among its virtues are relatively low cost and long life in non-contacting seal operation. Labyrinth seals are used in high speed, high pressure and high temperature applications. However, high seal leakage is a significant deficiency. The designer sets labyrinth operating clearance, and seal leakage, to avoid seal tooth rubs during all known transient conditions and allows an additional margin to accommodate unknown extremes in relative motion between shaft and stator. Unfortunately the reality is that well-designed labyrinth seal teeth are often found to have rubbed in service and degraded unit performance for extended periods of time. In Ref. 1, for example, 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 0.015 inch. But upon inspection, after five years of operation, packing clearance is typically rubbed to 0.060 inch.

The unforgiving consequence of rubbing fixed labyrinth teeth, a 4X clearance increase in the example cited, has motivated seal development in recent years, i.e. seal tooth geometry enhancements, honeycomb seals, retractable labyrinth packing, brush seals, finger seals, combinations of these and film riding seals. Of these Ref. 2 cites the brush seal, as the first simple, practical alternative to the labyrinth seal that offers extensive performance improvements. Reduced seal leakage, on initial brush seal run can be as little as 10-20% of comparable labyrinth seals at a pressure ratio of 3. Brush seals are, however, contacting seals and subject to wear as they accommodate transient relative motion between rotor and stator by rubbing displacement of numerous resilient seal elements. Brush seal leakage increases with time from such wear. Leakage of aircraft-engine tested seals have been reported[3, 4] to be less than double in periods approaching one engine overhaul cycle of 3000 hours compared to newly installed brush seals. Other applications in power generation, however, require seal lives to be an order of magnitude greater.

Ongoing developments to mitigate brush seal limitations and extend service life are cited in Ref 2. These efforts address seal “hysteresis”, bristle “stiffening” and “pressure-closing”. Of these, seal “hysteresis” and bristle “stiffening” increase leakage when rotor excursions occur. “Pressure-closing”, or “blow-down” causes wear, adversely affecting seal life. Seal innovation avoiding these phenomena would improve turbine shaft sealing and lead to higher differential pressure and temperature capability. The leaf seal presented in this paper is such a seal. It is the subject of an issued patent [5] and published patent application [6].

II. Pressure Actuated Leaf Seal Assembly Overview

The non-contacting, Pressure Actuated Leaf Seal, is illustrated in Fig. 1 and Fig. 2 is a photo of a full size model.

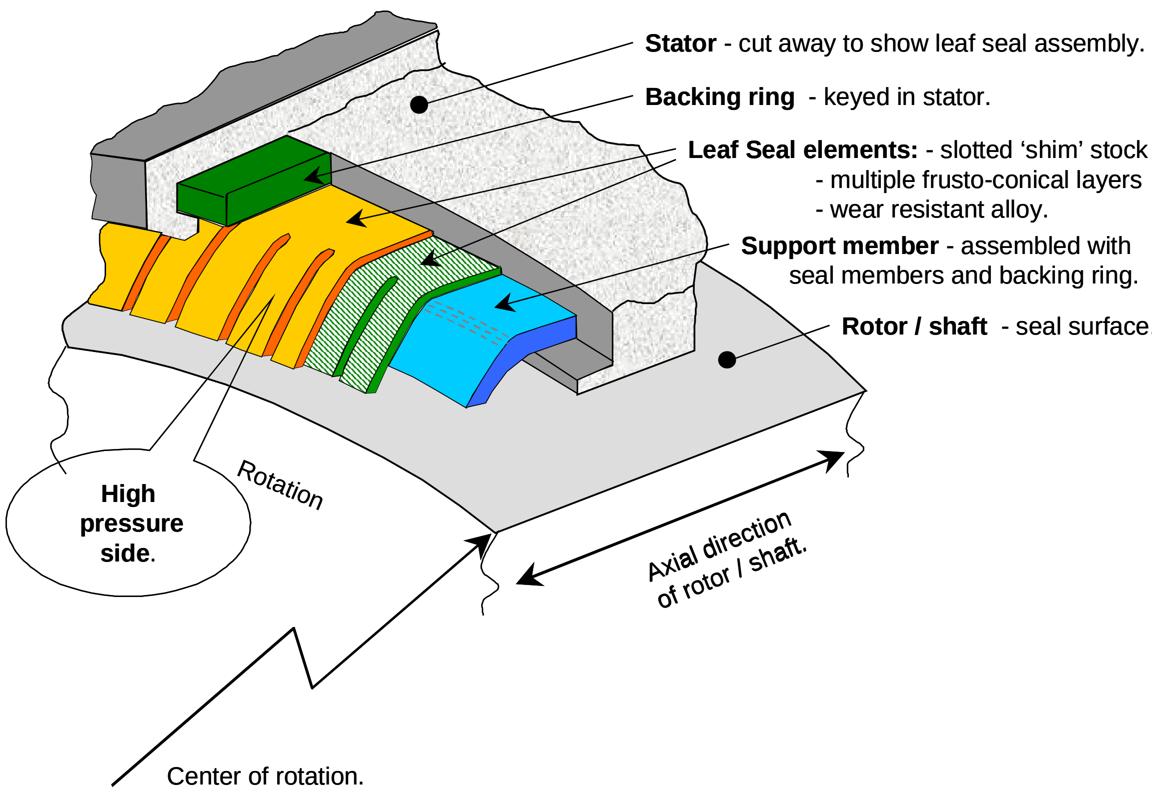

Figure 1. Pressure Actuated Leaf Seal Assembly

Seal members in this concept are fabricated from shim stock. Slots cut into the edge of a strip of shim stock form leaves similar to the tabs of a roofing shingle. Leaves are bent at an acute angle from the uncut edge portion of the strip and formed into a frusto-conical shape relative to the shaft axis. Additional seal member layers are added to bear the differential pressure of a particular application. Slots between leaves of seal member layers are displaced from each other, as in shingles, to block airflow. The support member, facing the high-pressure side of the seal, supports seal leaves against differential pressure. A weld joins support and seal member layers to a backing ring. The backing ring accommodates insertion of seal segments into a slot in the machine stator component surrounding the rotor. Typical space requirement for installation of a Pressure Actuated Leaf Seal is approximately 0.5 inches in both axial length and radial height as illustrated in Fig. 2.

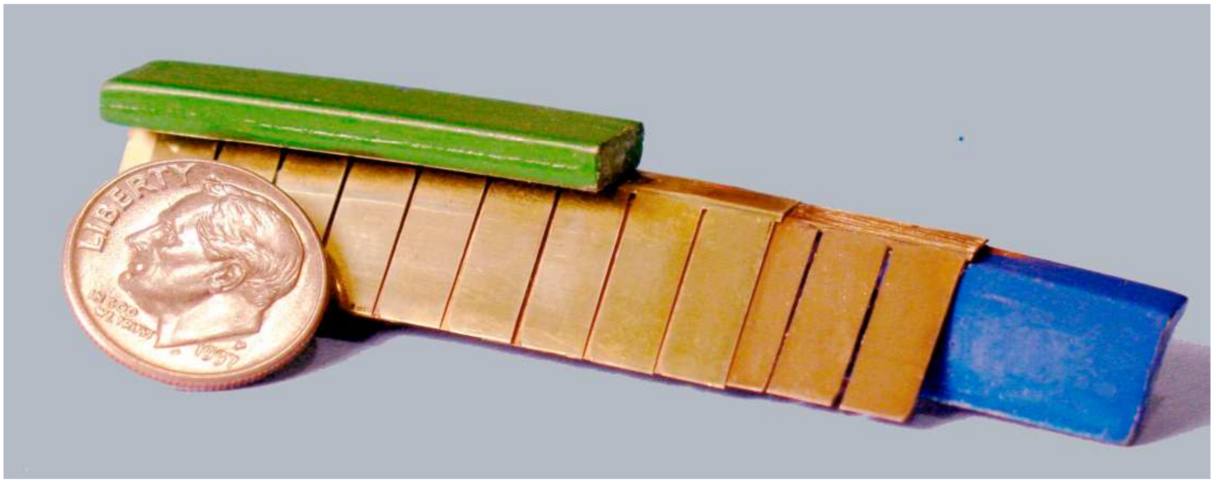

Figure 2. Pressure Actuated Leaf Seal Full Size Model Photo.

III. Pressure Actuated Leaf Seal Components And Their Design

A. Seal members

Leaf seal members are designed to elastically withstand differential seal pressure at system temperature without rubbing contact in normal operation.

Material selection criteria to meet design objectives include high temperature strength, wear resistance, availability in sheet or strip stock and acceptable weld properties in joining with other seal components. More optimal material may be selected for specific applications, but Haynes® 25[7], widely used for brush seal bristles, is a good candidate. It is available in sheet form, has desirable wear resistance and is strong at temperature. At 1000'F, 20% cold worked and aged Haynes® 25 sheet has 0.2% yield strength of 129Ksi.

Edge slots in the strip of material forming the seal members may be cut by a variety of productive methods such as wire electro-discharge machining (EDM), water-jet or laser cutting. Since these are typically numerically controlled operations, slot geometry can be as precise as required and as narrow as 0.002 inches. Slots observed in Figures 1 and 2 are perpendicular to the seal member edge, but they may be cut at an angle relative to strip edge or convergent to produce tapered seal element leaves. Narrow slots aid in blocking airflow with a single leaf seal layer. Slot leakage is blocked in applications with multiple displaced layers of seal members. Tapered leaves provide clearance between adjacent seal elements when the prepared strip is formed into frusto-conical shape.

Seal member thickness and number of seal layers are selected to meet application requirements of differential pressure and seal closure needs. Leaf width is much greater than seal thickness. In preliminary design, leaves are considered flat cantilever beams and stress is kept well within the elastic limit of the material at maximum operating temperature. Leaf stress under the support member is a function of the unsupported leaf length, thickness and seal differential pressure. The radius of the arc and leaf thickness determines leaf stresses in that portion where pressure bends them into compliance with the support member arc.

B. Support member

The support member sustains seal leaves against differential seal pressure. Located under the seal leaves, on the low-pressure side, it bears the upstream pressure load on leaf seal members. The minor diameter of the support member, set as close to the rotor as possible consistent with avoiding rubbing contact under all circumstances, is similar to the 'fence height' in brush seal designs. Structurally the support member is designed to carry the full differential pressure load with minimal elastic deformation. The upstream surface of the support member is machined to include an arc. Elastic deformation of seal members into contact with the arc as seal pressure is applied results in a change in the radial height of the seal member ends; i.e. the pressure actuated change in seal clearance. The length of the arc and arc radius determine the amount of change in seal clearance. Support member material needs to be weld compatible for the joining leaf seal member layers and backing ring.

C. Backing ring and assembly

The backing ring surrounds seal member layers and the support member. They are joined together into a seal assembly by a consolidating weld illustrated in Fig. 4. Functionally the backing ring provides insertion of the seal assembly into a stator machine component surrounding the rotor or shaft to be sealed. Geometry of the backing ring and corresponding stator slot may vary as needed. 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.

IV. Seal Requirements And Pressure Actuated Leaf Seal Function

The turbine seal-operating environment presents significant challenges to successful seal design. The static appearance of the labyrinth seal in Fig. 3a, suspended in close proximity to a large diameter rotor seal surface from a fixed stator part, is deceptive. There is little that is stable in the actual turbine seal environment as illustrated in Fig. 3b. The rotor seal diameter increases in response to centrifugal spinning force and rising working fluid temperature. In addition, the rotors seal surface center of rotation shifts with shaft displacement in journal bearings as hydrodynamic lubricating oil film thickness develops with increasing speed. Run-out of the manufactured rotor seal surface may also be several thousandths of an inch. Rotor dynamic excitation, traversing critical speeds, amplifies shaft seal surface displacement. Inertial rotor forces from maneuvers of mobile platforms, as in aircraft jet engines, or sudden load changes in power generation, may transiently displace the rotor seal surface too. Stability of the stationary turbine component mounting the shaft seal is also subject to a variety of forces and displacements. Radial stator position changes in response to working fluid temperature as well as the rotor, but typically at a different rate since the thermal mass and exposed heat transfer surfaces areas are different. Non-uniform stator wall thickness',

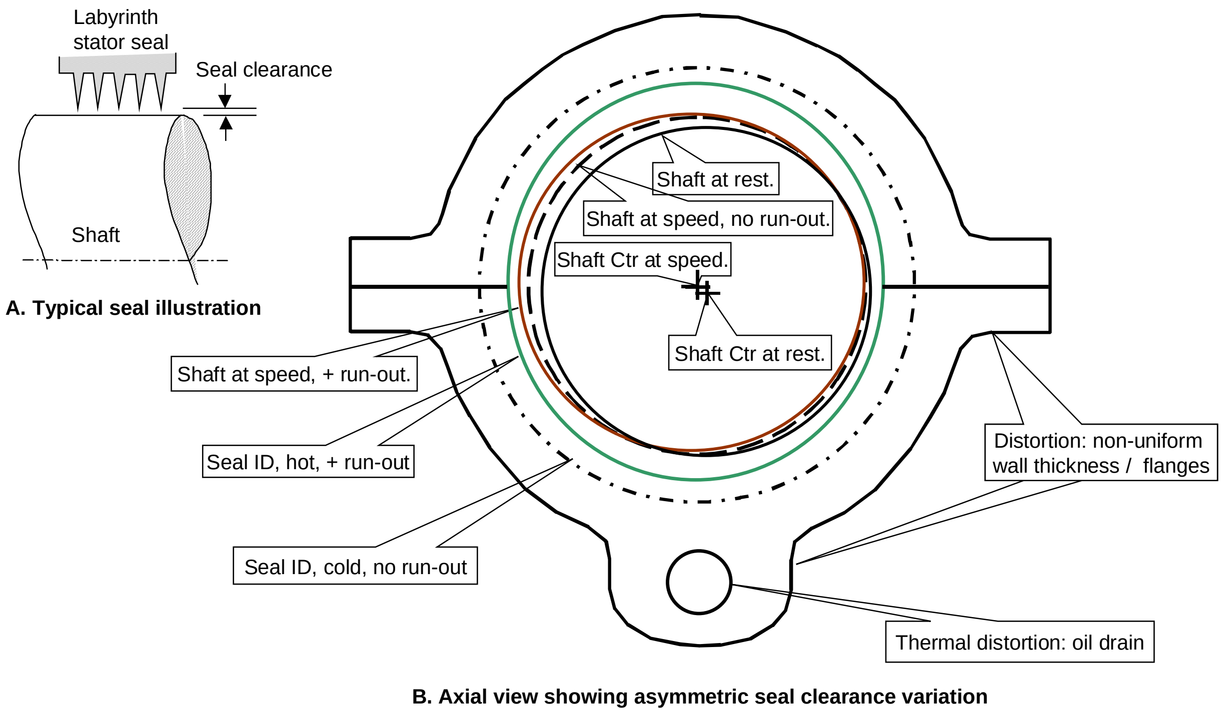

Figure 3. Turbine Seal Operating Clearance Considerations.

such as flanges, and discrete cooling, by lube oil feed and drain for example, also introduce asymmetric thermal growth distorting seal height around the rotor. These effects, along with manufactured seal tolerance, seal mounting slot tolerance and run-out with respect to shaft journal centers, cause seal clearance around the circumference of the rotor to be non-uniform to some degree and certainly different in operation than when cold and static. Axial thermal response of rotor and stator also occur, but at different rates causing displacement of the seal relative to the rotor, establishing minimum axial seal gland length requirements.

While physical turbine seal clearance sums all of these phenomena through operating time, the seal designer works to gain an understanding of them by calculation, clearance measurements and examination of rubbed seals. The example cited in which a seal designed for 0.015 inch operating clearance is rubbed to 0.060 inch illustrates both the need and the challenge of seal design. Start-up and shutdown thermal transients and rotor dynamic displacements are the most challenging seal clearance factors. Since transient rotor displacements are not subject to precise analytic prediction and thermal transients are unavoidable, selection of a seal concept to avoid or minimize their affect is desired. Retractable or variable clearance labyrinth packing is used in some current applications to avoid start-up transients and brush seals, honeycombed labyrinth seals and "Guardian Seal" labyrinth packing by Turbo Parts, Inc are examples of rub tolerant seals. Design features of the Pressure Actuated Leaf Seal will enable both start-up and shutdown rub avoidance and a degree of rub tolerance for upset events.

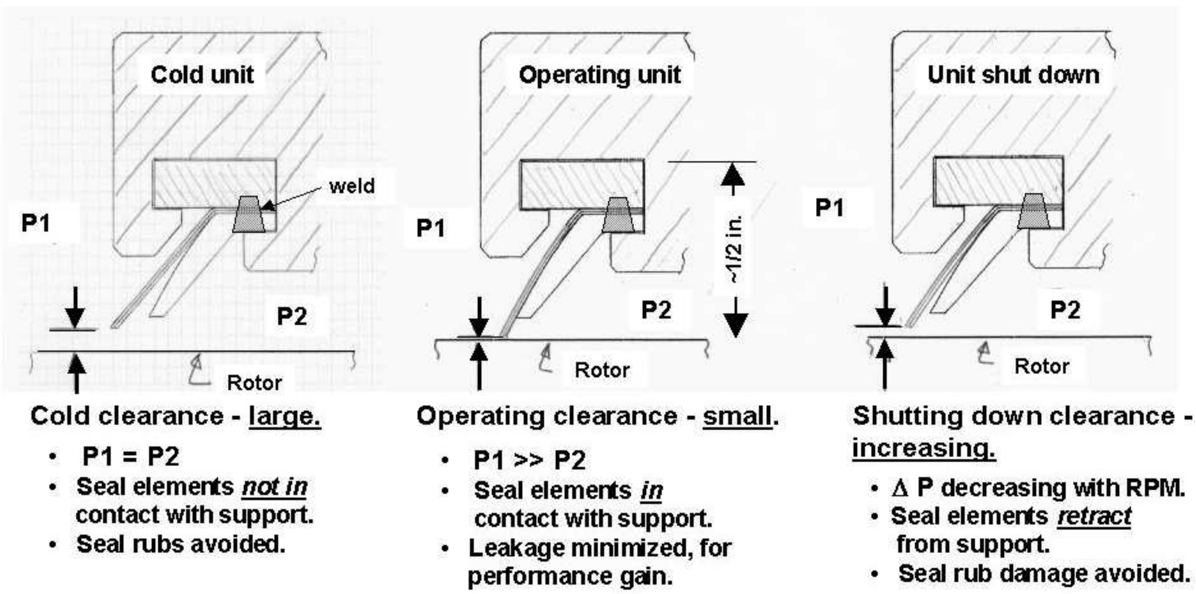

Figure 4. Pressure Actuated Leaf Seal Functional Characteristics.

To understand how the Pressure Actuated Leaf Seal functions to avoid start-up and shutdown rubs refer to Fig. 4. In the 'cold unit' panel, frusto-conical leaves of the seal assembly have 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 pressure to bring seal leaves into contact with the support member can be selected to meet application objectives. If unit warm-up and acceleration to operating speed is accomplished at 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, seal leaves elastically disengage from the support member arc surface increasing seal clearance to avoid seal rubs as shaft speed slows. Significant rub avoidance is accomplished by the Pressure Actuated Leaf Seal design.

Figure 5a is a photo of a functional model showing a simulated shaft seal gland and segment of a leaf seal connected to an air supply. Without differential pressure in Fig. 5a, leaf tip clearance to the simulated seal surface is ~0.040 inches. In Fig. 5b, the leaf seal is in compliance with the support member, with 20 psid pressure applied, and leaf tip clearance to simulated seal surface is ~ 0.005 inches.

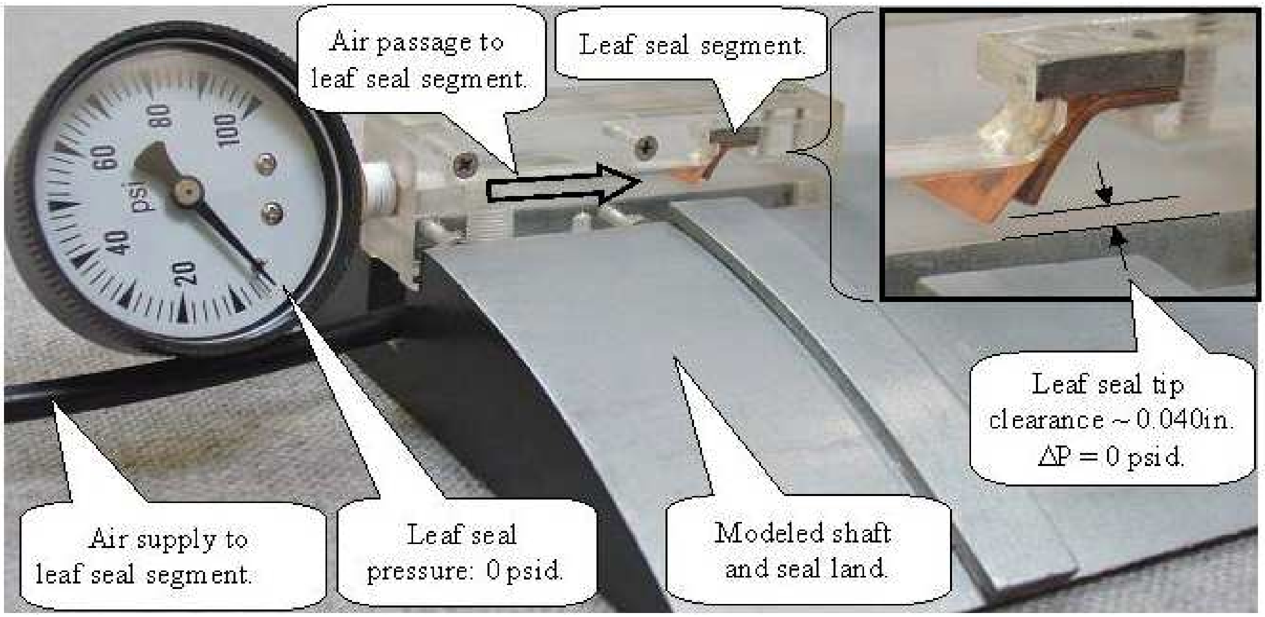

Figure 5a. Pressure Actuated Leaf Seal Functional Model - Static Description.

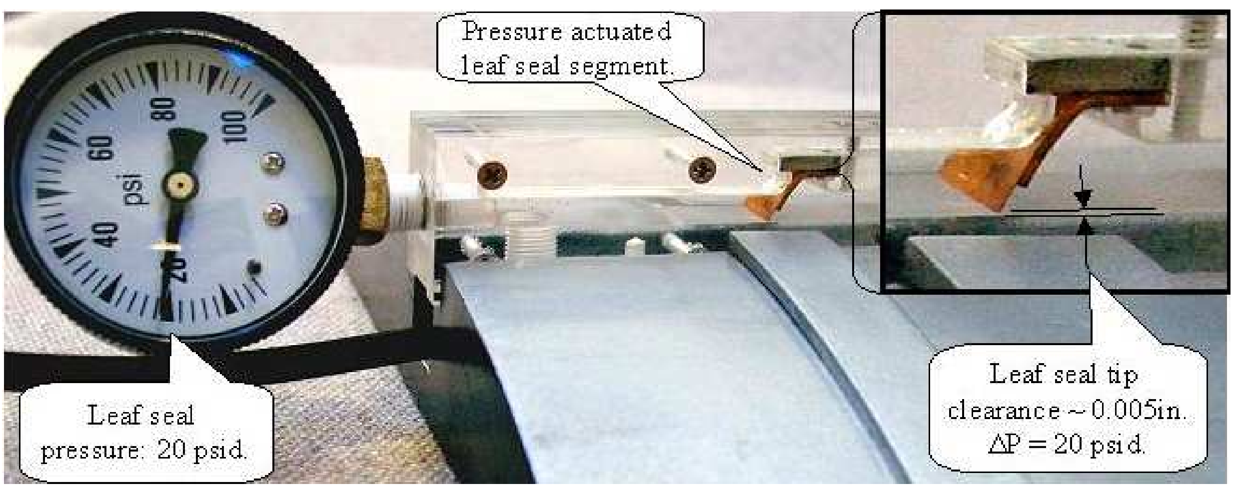

Figure 5b. Pressure Actuated Leaf Seal Functional Model Operation – Pressure Actuated.

Avoiding seal rubs during all turbine operations may not be possible. In such cases, seal survival requires a measure of ‘rub tolerance’. Solid labyrinth seal teeth typically ‘mushroom’ when rubbed, opening seal clearance for all future operation. In contrast, radial slots in the frusto-conical seal elements of the Pressure Actuated Leaf Seal allow leaves to deflect in a non-catastrophic manner when rubbed by the rotor. The differential seal pressure force on seal leaves is opposed in part by their elastic deflection from their unloaded frusto-conical shape. Rub force on a leaf seal tip at time of contacting the rotor may therefore be light during unit startup or shutdown. However a rub at full operating pressure could have significant force. If the duration of such a rub is short, (a few revolutions) leaves may deflect undamaged, but if the rub condition is imposed for longer periods of time, seal tip wear would likely result. Experimental evaluation of candidate material’s rub tolerance is needed to access Pressure Actuated Leaf Seal endurance under anticipated conditions.

Rub tolerance that allows a measure of initial operation 'wear-in' would also contribute to maximum performance benefit with smaller operating seal clearance. At operating conditions the minimum non-contacting seal clearance is determined by run-out, i.e., the deviation in radial height, of the seal around its circumference, and rotor seal surface run-out. Favorable rub tolerance of leaf seal tips would permit a slightly undersized leaf seal to be worn in for optimum fit. Elimination of stator seal run-out, for example, would remove a significant contributor to effective seal clearance. Non-contacting seal clearance could then approach one half of rotor seal surface run-out; i.e. clearance in one revolution would vary from 0 to the total indicated reading, TIR, run-out value on the rotor seal surface. Targeted operating seal clearance must, of course, meet downstream secondary flow requirements. As documented in Ref. 8, changes in seal leakage flow with improved seals can adversely affect component cooling. Seal clearance may, therefore, need to be greater than the minimum possible value to provide such cooling at the expense of unit performance.

V. Pressure Actuated Leaf Seal Preliminary Design Analysis

Major features of Pressure Actuated Leaf Seal design include: seal clearance change, seal height and approximate length, leaf thickness, bending stress, and the number of layers of leaves. Application data required to conduct preliminary design analysis includes: seal differential pressure at operating conditions, the differential pressure for seal closure during start-up to full operating conditions, a target value for the change in seal clearance, and desired support member ID. Operating seal temperature is also required for material selection, allowable bending stress and modulus of elasticity for calculations.

Inoperative seal geometry, illustrated in Fig. 4, is oriented as shown in Fig. 6 for evaluation as a 'built-in' cantilever beam. In the present design beam loading is differential seal pressure. 'Beam' deflection is evaluated at the desired seal actuation pressure for various combinations of seal design variables, i.e., leaf support radius, leaf thickness, length and number of layers of leaves. When leaf deflection equals the support member arc at its extremity, there is approximate compliance with the support member. Bending stress in that leaf portion is a function of leaf thickness and support radius. Cases with acceptable bending stress are then considered for change in seal clearance, other seal geometry and stress in the unsupported leaf portion beneath.

Figure 7 illustrates the variables used in calculation of leaf seal clearance and geometry changes when leaves come into compliance with the support member arc in the pressure actuated configuration. The change in seal clearance, ΔCL, is the actuated change in radial height of the leaf tip. Refinements in seal design variables of leaf length and support radius are made to attain design objectives. Segments of overall leaf length are used in these calculations. L1 is the arc length of support radius, L2, is specified to separate the leaf bending stress over the arc from the cantilever bending stress of the unsupported leaves beneath the support member, and L3, is the unsupported leaf length beneath the support member. These variables also determine support member radial height, H_S, relative to leaf seal tips, i.e. the approximate 'fence height' of the support member.

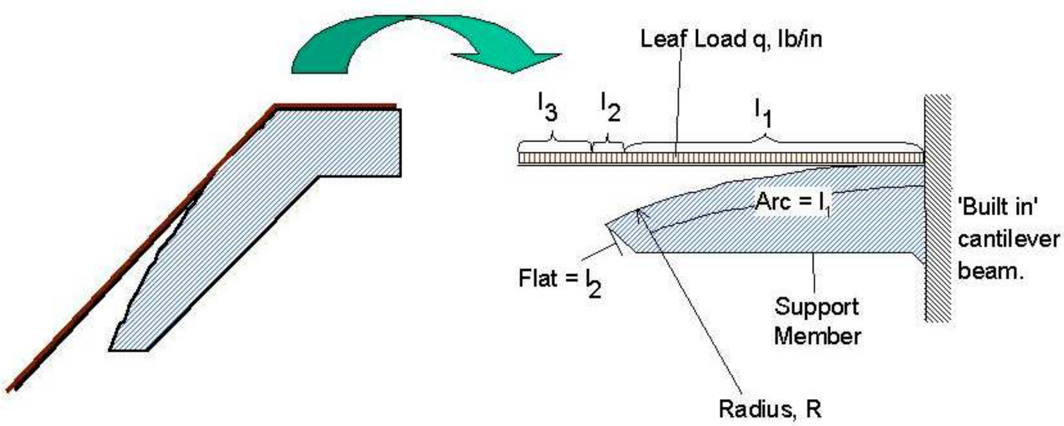

Figure 6. Leaf Seal Analysis Orientation.

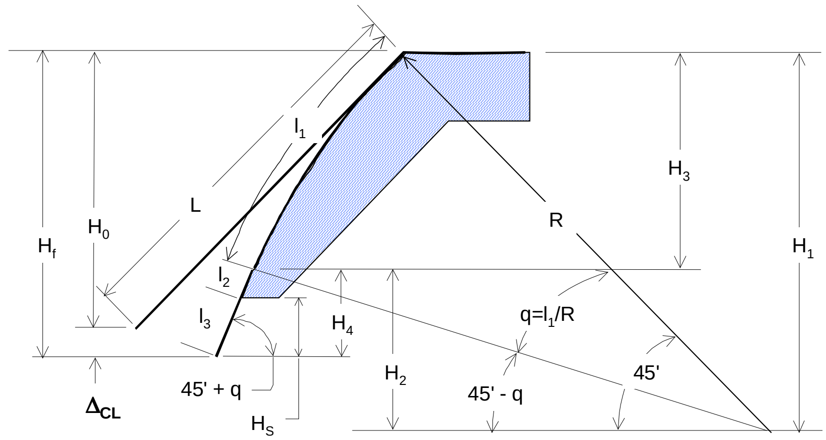

Figure 7. Leaf Seal Geometry Changes

Cantilever bending stress in the unsupported leaves extending below the support member is calculated at the full operating seal differential pressure. The additional deflection at the end of that leaf segment is also calculated.

A. Analysis results

Several seal designs with operating seal differential pressures from 40 psid to 2400 psid, have been analyzed. The results are tabulated in Fig. 8. Different actuating pressures, as a fraction of operating seal differential pressure, were selected for target seal engagement. Changes in seal clearance of 0.045 to 0.056 inches were calculated, support height was maintained at about 0.100 inches and leaf stress was limited to 100ksi, consistent with Haynes ® 25 elastic capability. The number of seal leaves varies from a minimum of two to a maximum of 12 where a high differential operating pressure and engagement differential pressure is used. Seal leaf angle relative to shaft axis is also calculated and included in the geometry tabulation.

The sample tabulations reveal that leaf seal design can accommodate a wide range of application operating conditions. High differential pressure, in excess of current brush seal capability, is feasible. It is recognized that a complete design analysis will also include structural analysis of the support member, the backing ring and assembly weld to assure robust seal life. More sophisticated finite element stress analysis and deflection calculations will provide further confidence in the seal structure.

Seal design pressures.		Seal design variables -						Δ Seal Clearance and geometry				Bending stresses	
Seal ΔP, Unit design point, psid	**Engagement ΔP, psid**	**Support Radius,** inches	**Leaf thickness,** inches	**Length, l_1:** over arc support portion.	**Length, l_2:** length flat support seg.	**Length, l_3:** unsupported leaf.	**# Leaves**	**Seal angle to shaft,** deg.	**ΔC_L** : change in leaf tip clearance, In.	**Support Height** from seal ID.	**Deflection,** unsupported cantilever, l_3, in.	**Stress, psi** Unsupported length.	**Stress, psi** Leaf thicknes to radius R.
40	20	2.00	0.010	0.425	0.075	0.120	2	57.2	0.056	0.101	0.0002	8640	77747
75	35	2.00	0.012	0.425	0.075	0.120	2	57.2	0.056	0.101	0.0002	11250	93297
150	50	2.20	0.014	0.425	0.075	0.120	2	56.1	0.051	0.100	0.0003	16531	98951
300	50	2.20	0.014	0.425	0.075	0.120	2	56.1	0.051	0.100	0.0006	33061	98951
600	150	2.50	0.016	0.425	0.075	0.120	4	54.7	0.0454	0.098	0.0008	25313	99516
1200	200	2.50	0.016	0.425	0.075	0.120	6	54.7	0.0454	0.098	0.0016	33750	99516
2400	200	2.50	0.016	0.425	0.075	0.120	6	54.7	0.0454	0.098	0.0032	67500	99516
2400	400	2.50	0.016	0.425	0.075	0.120	12	54.7	0.0454	0.098	0.0032	33750	99516

Figure 8. Preliminary Design Analysis Sample Results.

VI. Pressure Actuated Leaf Seal Benefit Assessment

The Pressure Actuated Leaf Seal concept is compared to labyrinth seals and brush seals in Fig. 9 and the discussion below. Eight seal attributes are considered and evaluated as a strength or weakness inherent to the design or operating experience for each seal type. An up arrow indicates positive design features and operating experience and a down arrow a negative design feature or unfavorable experience. For the Pressure Actuated Leaf Seal, in its concept feasibility phase, these evaluations are based on design features and expectation of successful reduction to practice.

	Rub Avoidance	Rub Tolerance	Seal Life	Low Leakage	Hi Δ P >300psid	Reverse Rotation	Seal Length	Cost to Manufacture
Labyrinth Seals	↓	↓	↑ w/o rub	↓	↑	↑	↓	↑
Brush Seals	↓	↑	↓ improving	↑	↓	↓	↑	↓
Pressure Actuated Leaf Seal	↑	TBD	↑	↑	↑	↑	↑	↑

Figure 9. Pressure Actuated Leaf Seal Benefit Comparison.

A. Rub avoidance:
- *Labyrinth seals* ↓ - are often damaged by unplanned rubs compromising their effectiveness.
- *Brush seals* ↓ - bristles are in rubbing contact with the rotating seal surface during normal operation and transients, limiting seal life.
- *Actuated leaf seals* ↑ - incorporates design features to avoid start-up and shutdown rubs.

B. Rub tolerance:
- *Labyrinth seals* ↓ - teeth are permanently deformed when rubbed.
- *Brush seals* ↑ - bristles are designed to resiliently deflect on rubbing contact with acceptable wear.
- *Actuated leaf seals* ? - seal elements can resiliently deflect, a positive feature, but wear will be dependent on materials selected, pressure loading and surface speed that can only be determined by testing.

C. Seal life:
- *Labyrinth seals* ↑ - typically have long life when operated without rubs.
- *Brush seals* ↓ - development activity continues to be required to reduce bristle wear and extend brush seal life, but progress is reported.
- *Actuated leaf seals* ↑ - design features for non-contacting seal operation contributes to long life expectancy.

D. Low seal leakage:
- *Labyrinth seals* ↓ - large seal clearances, needed to avoid rubs, contributes to large seal leakage.
- *Brush seals* ↑ - initial leakage is cited at 10 to 20% of a typical 4 tooth labyrinth seal.
- *Actuated leaf seals* ↑ - shingled seal members and small operating clearance to effectively block flow bodes well for small seal leakage. Leakage increase with time is mitigated by its non-contacting design.

E. High seal differential pressure, > 300psi:
- *Labyrinth seals* ↑ - have been designed to dependably operate at high differential pressure.
- *Brush seals* ↓ - bristle stability limits current brush seal differential pressure capability to ~ 300psid.
- *Actuated leaf seals* ↑ - design at high differential seal pressure is feasible with multiple seal layers.

F. Reverse rotation capability:

- *Labyrinth seals* ↑ - are non-contacting seals and capable of reverse rotation without damage.
- *Brush seals* ↓ - are at risk of damage with reverse seal surface rotation, even for maintenance procedures, because bristles contact the rotating surface at ~45 degrees angle.
- *Actuated leaf seals* ↑ - are non-contacting seals and capable of reverse rotation without damage.

G. Seal length:

- *Labyrinth seals* ↓ - are inherently long because sealing is derived from small pressure drops across a succession of seal teeth. High differential pressure requires more seal teeth, increasing shaft-packing length and turbine bearing span.
- *Brush seals* ↑ - seal with one or two stages of seals that require little axial length.
- *Actuated leaf seals* ↑ - require less than 1 inch of axial length in cases considered.

H. Cost to manufacture:

- *Labyrinth seals* ↑ - are typically less expensive than brush seals though features such as honeycomb rub strips and coated teeth labyrinth seals add cost.
- *Brush seals* ↓ - inherent use of a large quantity of small diameter bristles that must be oriented, closely packed and secured by weld or other means contribute to high manufacturing cost.
- *Actuated leaf seals* ↑ - feature seal elements readily fabricated from sheet metal strips and weld assembly of a small number of parts is expected to be cost effective in production.

In summary, this benefit assessment of eight important seal attributes shows the Pressure Actuated Leaf Seal to have significantly more positive features than both labyrinth and brush seals. Prototype flow testing and introductory service of Pressure Actuated Leaf Seals are recognized as essential to verify this assessment, but the comparison is good reason to proceed.

VII. Conclusion

Pressure Actuated Leaf Seal design features have been discussed in some detail to illustrate the potential for improved turbine shaft sealing. The pressure actuation of seal elements provides a mechanism of rub avoidance during start-up and shut down transients. At operating conditions, minimum seal clearance is expected to result in performance gains comparable to newly installed brush seals. Long Pressure Actuated Leaf Seal life expectancy, with non-contacting seal elements, can sustain those performance gains, undiminished, for extended periods of time. In addition, high differential seal pressure applications are feasible and manufacture cost is anticipated to be modest. In all, the concept is ready to progress to the next steps of application specific design, manufacture and prototype testing for introductory service. Promising partnership arrangements for this next phase are underway. Enhancements to the basic concept are also the subject of ongoing patent activity.

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