Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air
GT2010
June 14-18, 2010, Glasgow, UK

GT2010-23629

FILM RIDING LEAF SEALS FOR IMPROVED SHAFT SEALING

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ABSTRACT

Turbine shaft seals are vulnerable to rubs caused by thermal distortion, mis-alignment and rotor dynamic vibration that are often not well understood. When seals rub as a machine is brought up to operating conditions performance is compromised due to increased seal leakage. Much effort has been extended in recent years to develop seals that mitigate those losses. This paper presents a seal design with segmented film riding runners capable of non-contacting seal operation during rotor transients. Operating differential seal pressure displaces seal leaves and attached runners toward the rotor surface until balanced by hydrostatic and hydrodynamic lift. Sufficient radial range of operation is provided to follow the rotor seal surface during transients while maintaining a small seal clearance. Seal design features and function will be described and illustrated along with analysis of forces and motions for a sample application. Planned modeling and testing will also be presented. This concept promises enhanced shaft sealing by combining a leaf seal structure that provides a large range of motion to avoid rubs during startup and shutdown with runner elements capable of generating hydrostatic and hydrodynamic lift forces to maintain shaft - seal separation during all rotor displacement transients. Improved turbine performance from small operating seal clearance and extended seal life without rubs are expected benefits of the Film Riding Pressure Actuated Leaf Seal (FRPALS).

INTRODUCTION

Seals that reduce leakage between rotating and stationary components are critical to turbine performance. Manufacturing tolerances and installation alignment contribute to seal clearance and leakage. In addition, start-up and shut-down transients, caused by different rates of thermal response to hot working fluid and rotor dynamic shaft movement, often lead to varying seal clearances that are not well understood and cannot be controlled. This technical challenge to prevent seal rubs, lost performance, and costly corrective maintenance has motivated significant seal innovation in recent years as documented by Chupp [1]. Included in that effort is the Pressure Actuated Leaf Seal (PALS) [2], in which seal leaves elastically deflect to reduce operating clearance responding to the differential pressure to which they are subjected. This sealing concept provides the designer with means to avoid startup and shutdown rubs while preserving low seal leakage at operating conditions.

Building on the PALS concept a superior shaft seal capable of maintaining a small, non-contacting clearance under all transient and operating conditions is possible. The FRPALS illustrated in Fig. 1 is such a seal and the subject of this paper. In it, differential seal pressure displaces seal leaves and attached segmented runners toward the rotor surface until balanced by hydrostatic and hydrodynamic lift at a small clearance. Seal design provides sufficient radial range of operation to follow the rotor during transients while maintaining small clearance and low leakage. The concept uses two sets of axially displaced leaf seal elements that engage underlying runners and provide nearly-parallel translation of the seal runners in the radial direction.

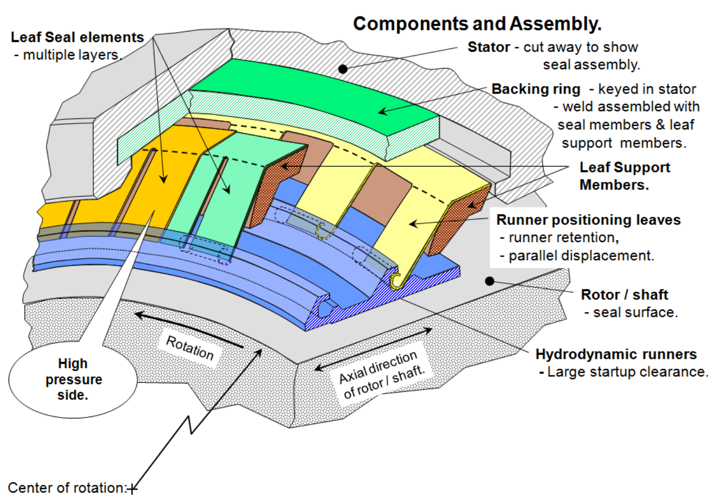

Figure 1. Film Riding Leaf Seal Concept Isometric View

BACKGROUND

Turbine shaft seal design is challenging. Typically a labyrinth seal is pictured suspended from a fixed stator part in close proximity to a large diameter rotor seal surface, as illustrated in Fig. 2A. However, the axial view in Fig. 2B more accurately notes numerous factors that must be considered in seal design and how difficult it can be to predict seal operating clearance with any degree of certainty. The rotor seal diameter increases in response to shaft speed and rising working fluid temperature. On startup, the rotor seal surface center of rotation shifts as hydrodynamic lubricating oil film thickness develops with increasing speed, displacing the shaft in journal bearings. Dimensional tolerances of manufactured rotor and stator seal surfaces and their alignment contribute to non-uniform seal clearance. The turbine stator assembly mounting the seal also changes in response to working fluid temperature but typically at a different rate than the rotor since the thermal mass and exposed heat transfer surface areas are different, causing conventional seal clearance to change with time. Asymmetric thermal growth of a stator assembly with thick flanges, or discrete cooling such as lube oil feed and drain lines also contribute to non-uniform seal clearance around the rotor. In addition rotor dynamic excitation while traversing critical speeds accelerating to operating conditions or decelerating from them, amplifies shaft displacement. Sudden load changes in ground equipment or maneuvers of air or sea vehicles can also displace the rotor. Operating seal clearance is therefore non-uniform to some degree around the circumference of the rotor and subject to transient rotor dynamic response that is difficult to predict. These uncertainties motivate seal innovation capable of maintaining a small clearance through all transient conditions.

The FRPALS incorporates design features discussed in technical papers AIAA-2005-3985 [2], Pressure Actuated Leaf Seals for Improved Turbine Shaft Sealing, and AIAA-2009-5167 [3], Pressure Actuated Leaf Seal Feasibility Study and Demonstration. They are also the subject of United States

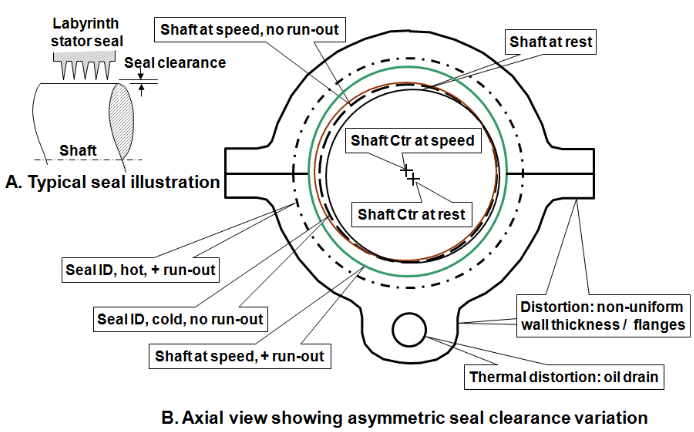

Figure 2. Shaft Seal Clearance Design Considerations.

Patents [4] and [5]. Seal elements in that design elastically deflect with differential pressure to a small shaft clearance near full speed, preferably above shaft criticals, and a selected minimum operating pressure for rub avoidance during startup and shutdown rotor dynamic transients. Once actuated a leaf support restricts further seal closure such that a small clearance is maintained throughout the operating pressure range.

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) [6]. Results discussed by Grondahl [3] document PALS clearance change with pressure and the potential for reduced seal leakage flow. Measured seal leakage was substantially lower than typical power generation labyrinth seals and leakage between adjacent leaf elements adds less than 0.002 in. (0.05 mm) to the effective clearance of the seal. Short duration rub tests also confirmed a measure of 'rub tolerance'. Leaf tip wear at speed and pressure was significantly less than the imposed seal interference with the rotor and evidence that leaves opened during the transient rub. In contrast, labyrinth seal tooth deformation would be expected to be comparable to the amount of rotor-seal interference.

Seal alignment and in-service thermal growth in large equipment may limit PALS application in certain situations. For low leakage PALS clearance must be significantly less than the multi-tooth labyrinth seal it is intended to replace because the pressure drop occurs across only one restriction. For example, a single seal tooth needs a clearance ~50% that of a 4- tooth labyrinth seal for the same leakage. Therefore, reduced PALS leakage will require considerably smaller operating clearances than multi-tooth labyrinth seals. PALS development has been proposed [7] to eliminate seal clearance non-uniformities cited in Fig.2 such as stator misalignment and run-out by developing seals that initially 'wear-in' to close proximity with the rotating seal surface. Subsequent base load seal operation would be non-contacting, by function of the leaf support member, preserving long seal life and performance. Resilient PALS functionality for rub avoidance and rub tolerance would be retained. While that approach is expected to be effective in some applications, this paper presents an alternative approach that avoids all contact between stationary seal elements and the rotating shaft.

FILM RIDING PRESSURE ACTUATED LEAF SEAL

Components of the FRPALS illustrated in Fig. 1 are shown in cross section in Fig. 3. They include:

SEAL LEAVES

Seal members are fabricated from shim stock with slots cut into an edge, forming 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 layers are used to bear the differential pressure and, similar to the tabs of a roofing shingle, leaves are displaced from each other to block airflow through the slots. Thickness and number

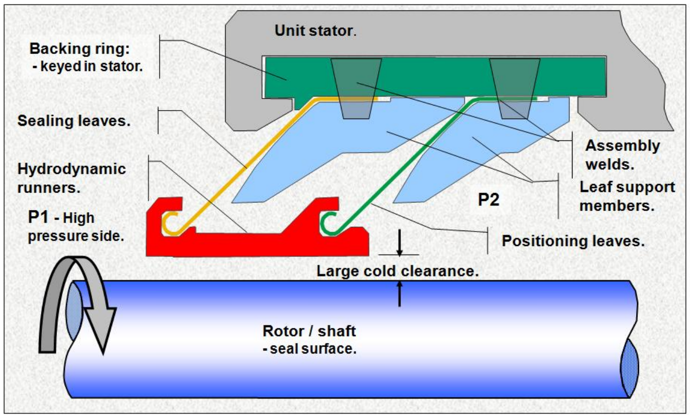

Figure 3. Film Riding Leaf Seal Cross Section View.

of layers are selected to meet application requirements of differential pressure and seal clearance closure objectives. Leaf width is much greater than thickness, so leaves are essentially flat cantilever beams for analysis of stress. Stress is kept well within the elastic limit and high cycle fatigue endurance limit over the entire operating temperature range. In cooperation with the support member, seal leaves are designed to elastically deflect in response to system pressure. At one end the leaves are joined to a support member facing the high pressure side of the seal. This location is referred to as the **knee**, because it is where the bend is located. At the other end, the leaves engage the hydrodynamic runners. One set of leaves blocks flow outside the runners, and the second set keeps the runner nearly parallel to the rotor as illustrated in Fig. 1.

SUPPORT MEMBER

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. 3. Arc length and radius determine the amount of change in seal clearance.

BACKING RING

A backing ring or similar structure, to which the support member and seal leaves are welded, provides means of inserting seal segments into a machined stator slot surrounding the rotor or shaft to be sealed.

HYDRODYNAMIC RUNNERS

Hydrodynamic runners are joined to both the forward and aft sets of leaves. Features on their outer diameter (OD) surface referred to as **ankles**, engage the leaves and allow them to pivot as runners are displaced radially. Seal runners are short circumferential segments with axial length, entrance geometry and inner diameter (ID) surface details to provide needed film force in operation. The runners are displaced nearly parallel to the rotor through curvilinear translation under the influence of differential pressure acting on the forward seal leaves.

RADIAL DEFLECTION

The radial deflection function is illustrated in Fig. 4. During initial start up differential seal pressure is small at low shaft speed. In that condition, the seal leaves are not appreciably deflected and there is a large cold clearance between seal and shaft. Stiffness of the leaves is established by seal design to maintain adequate shaft clearance as the rotor goes through critical speeds. During startup increasing system pressures generate hydrostatic lifting force and increasing speed generates hydrodynamic lift under the seal runners before they come into close clearance with the shaft. As illustrated in the upper panel of Fig. 4, the start up large cold clearance provides means of rub avoidance. At running conditions, illustrated in the Fig. 4 center panel, differential pressure is at design conditions and film load forces are balanced to 'float' the seal runners in small running clearance with the shaft. The low leakage of a film-riding seal is expected to be much less than that of labyrinth seals and will contribute substantial engine performance gain as a non-contacting seal. The lower panel of Fig. 4 illustrates the runner position with large radial shaft eccentricity.

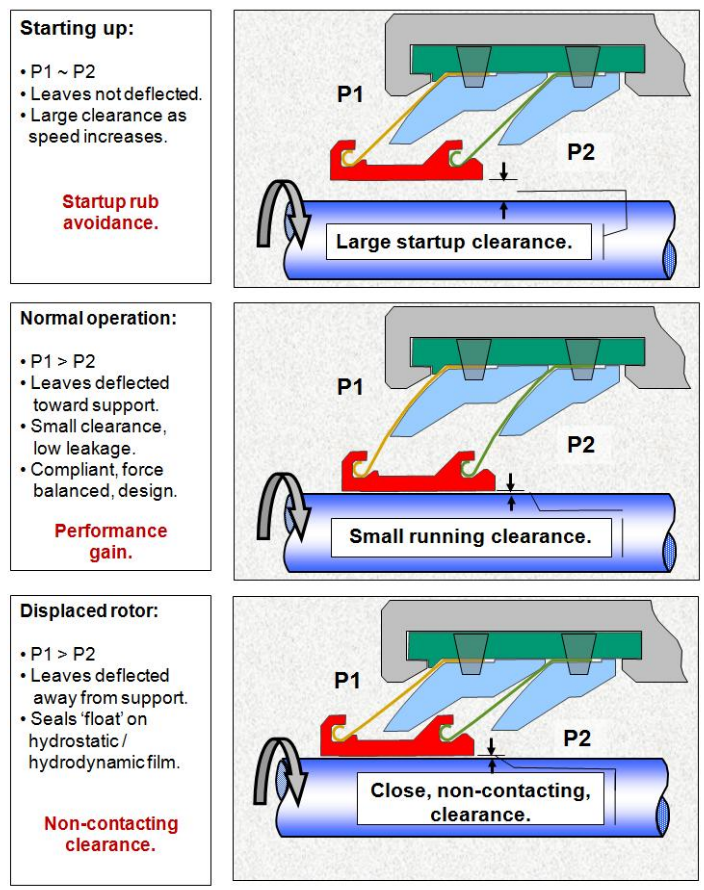

Figure 4. Film Riding Leaf Seal Function.

HYDRODYNAMIC – HYDROSTATIC FILM ANALYSIS

Seal specifications for hydrodynamic-hydrostatic analysis were selected for a generic large diameter power generation gas turbine application as tabulated in Table 1.

Rotor seal diameter	60.0 in.	1.52 m
Runner angle	18.0 deg.	0.314 rad
Temperature	1000 F	538 K
Speed	3600 RPM	3600 RPM
Ambient pressure	50.0 psig	345 kPa
Pressure differential	40.0 psi	276 kPa

Table 1. Seal Specifications.

Conceptual engineering studies were conducted to identify viable seal features at these conditions and the additional requirement that seal leakage rate be less than 2 lbs/sec (0.91 kg/s) with a temperature rise less than 50 F (28 C). Both a plane cylindrical runner configuration and one containing an inlet Rayleigh Step were evaluated. Analytical procedures developed for NASA Industrial Codes by Shapiro [8, 9] were used for analyzing a wide variety of fluid film seals.

The study identified satisfactory seal performance with the following geometry:

- Runner axial length: 1.500 inch (38.1 mm)
- Inlet Rayleigh step height: 0.010 inch (0.254 mm)
- Inlet Rayleigh step length: 0.300 inch (7.62 mm)
- Runner inside radius: 30.100 inch (0.76454 m)

The runner lifting force from the film at the specified conditions and selected runner geometry is plotted as a function of minimum clearance in Fig. 5. The plotted values are for one 18-degree (0.314 rad) runner segment. The target runner lifting force of 330 lbs (150 kg) per segment was based on a preliminary estimate of runner OD pressure and seal leaf loading with the rotor displaced 0.100 inch (2.54 mm) radially toward the particular segment. Actual lifting force needed to avoid runner contact

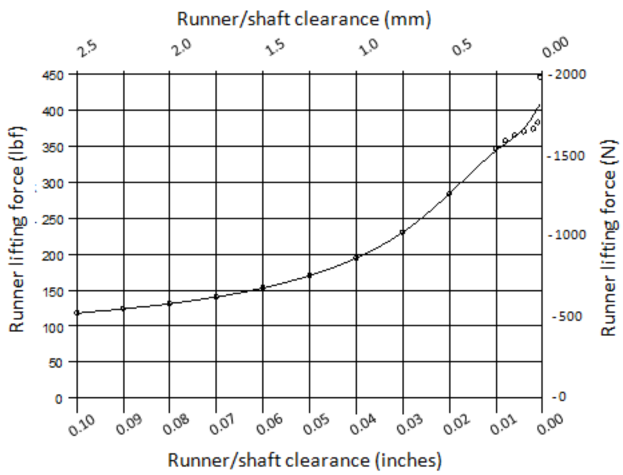

Figure 5. Runner Segment Lifting Force.

with the rotor is a function of the seal leaf forces and pressure acting on the runner OD and is addressed below.

Seal leakage under all seal segments is plotted in Fig. 6 for the selected runner geometry. This plot assumes all 20 seal segments comprising the 360-degree (6.28 rad) seal assembly are at the minimum seal clearance on the abscissa. Segment end gap leakage and leakage through and around the supporting leaf seal structure are neglected in this plot. The target leakage rate of 2 lb/sec (0.9 kg/s) was selected to be ~ 75% seal leakage reduction relative to a 4-tooth labyrinth seal with nominal clearance of 0.06 inch (1.52 mm). The leakage goal is met at an average minimum seal clearance of approximately 0.013 inch (0.33 mm) (still neglecting leaks between leaves and between runners). As noted before, equilibrium runner operating clearance is established by the balance of film pressure under the seal runner and forces acting on the OD of the seal runners, addressed in the next section.

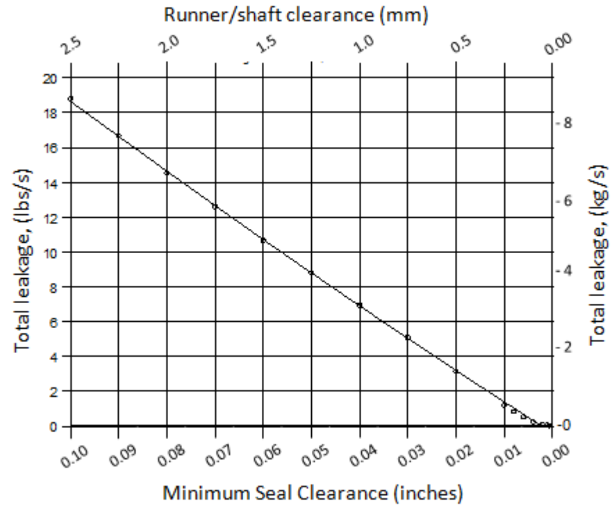

Figure 6. Seal Leakage Under All Runners.

Air temperature rise as a function of minimum film clearance is plotted in Fig. 7 at the specified conditions and selected seal runner geometry. Temperature rise is predicted to be less than 50 F (10 K) down to minimum seal clearance of approximately 0.006 inch (0.152 mm). This modest temperature rise is not expected to adversely impact anticipated applications.

The analysis also provided calculation of center of pressure, cross coupled stiffness coefficients and cross coupled damping coefficients which will be useful for subsequent dynamic analyses.

Power loss for all segments was less than 10 hp (7.5 kW) at seal clearances larger than 0.005 inch (0.127 mm).

An assessment of the contribution of hydrodynamic action vs. hydrostatic was made by comparing results at rest with full speed. The difference in runner lifting force between 0 and 3600 rpm is negligible. The leakage at full speed is only slightly greater ~ 0.2 lb/s (0.09 kg/s). It is concluded that there is practically no hydrodynamic effect in the operating range, and that the predominant effect is hydrostatic. This is attributed to the relatively short axial seal length.

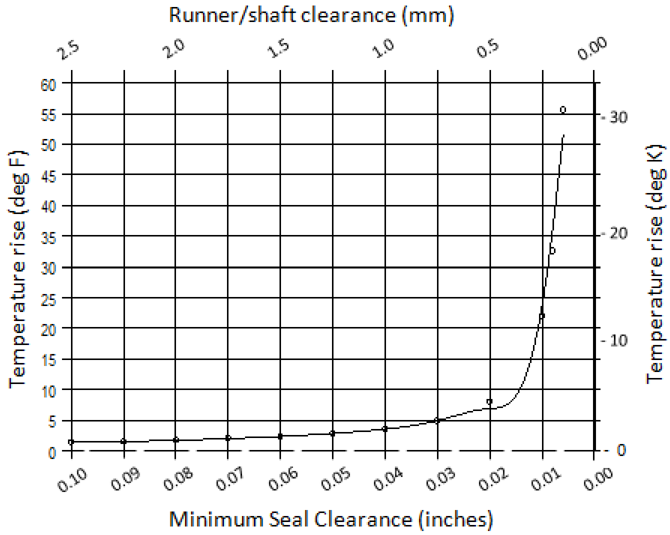

Figure 7. Seal Leakage Temperature Rise.

Comparative analysis was also done to assess the significance of temperature on seal performance. Compared to full temperature operation at 1000 F (538 K), startup operation at 70 F (21 K) reduces the viscosity from 5.307×10^{-9} to 2.629×10^{-9} lb-s/in^2 (3.66×10^{-5} to 1.81×10^{-5} Pa-s), so performance does not deteriorate markedly at the lower temperature. The explanation is that the performance for both temperatures is predominantly hydrostatic and is thus a function of the pressure differentials. Runner lifting force at the higher temperature is only slightly greater, than at ambient temperature; power loss is higher by 5 hp (3.7 kW) at the lower temperature, even though the viscosity is lower; leakage is higher (nearly double) at the lower temperature because of lower viscosity and temperature rise is greater at the higher temperature because of lower flow.

Sensitivity of seal performance to variation in the selected design parameters of inlet Rayleigh step were evaluated. Height change from 0.010 to 0.030 inch (0.254 to 0.762 mm) and length change from 0.30 to 0.5 inch (7.62 to 12.7 mm) had minimal influence on plots of runner lifting force and leakage as a function of minimum film thickness. Seal performance with an axial runner length of 2.0 inch (50.8 mm) was also compared to the selected 1.5 inch (38.1 mm) axial length. Seal leakage was essentially unchanged, but lifting force increased more than 30 percent. Seal performance change with runner inside radius of 30.010 inch vs. 30.100 inch (0.76225 m vs. 0.76454 m) was also evaluated. At this large seal diameter there is very little change in calculated seal leakage between the 2 cases. Runner lifting force is about 10% lower with the 30.010 inch radius than for a radius of 30.100 inches (Fig.5).

Misalignment studies were also performed to evaluate the effect of seal runner segment rotation in both clockwise and counter clockwise directions. Derivatives with respect to angle of runner lifting force, center of pressure and righting moment from these studies are used in the force/moment analysis of seal leaf interactions with the runners. In general, the seal appears to accommodate rotations without serious effects and performance of the selected configuration is considered good from the hydrodynamic-hydrostatic film analysis point of view.

Figure 8 is conceptual solid model view of a candidate, 1.5 inch (38.1 mm) axial length FRPALS.

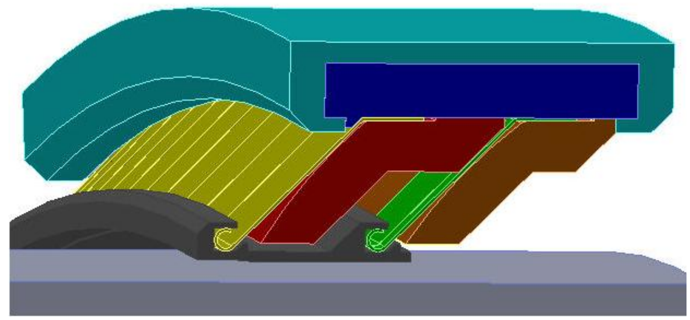

Figure 8. Film Riding Leaf Seal Solid Model.

FORCE AND CLEARANCE ANALYSIS

In the FRPALS design, the runner lifting force developed hydrostatically under the runners is opposed by pressure and seal leaf forces applied on the outer surfaces of the runner. The design process includes detailed analyses to select design parameters that result in tight clearance between the rotor and runners at base load conditions where low seal leakage is essential. The parameter selection process must also assure that the runners avoid contact with the rotor at maximum transient radial rotor displacement relative to the seal. Seal leaf elastic properties are determined by leaf thickness, length, and modulus of elasticity. Upstream pressure on exposed runner surfaces applies loads on the runners as well as on the sealing set of leaves. The leaf tip load on the runner OD is a function of the radial displacement of the seal runner relative to its shutdown position. Similarly, the film forces acting on the runner ID are a function of clearance with the rotor. The following paragraphs outline this design analysis and resulting operating clearances.

Forces acting on the seal runners are shown in Fig. 9. The forces between the leaves and the runners at the pivot point in the ankle can be resolved into axial, Fz, and radial, Fr, components. These components may be different in the two sets of leaves because leaf 1 sustains all of the differential pressure, DP, and also the hydrostatic and hydrodynamic pressures, Fpi, acting radially outward on the runner shift axially a small amount as clearance changes. The challenge for the designer is to balance these forces at the appropriate operating point such that the design is stable and responsive over the full operating range, including high eccentricities encountered during transient operation discussed above.

The leaves are treated as individual cantilever beams, built in where trapped between the backing ring and the leaf support members. As the leaves respond to increasing pressure they

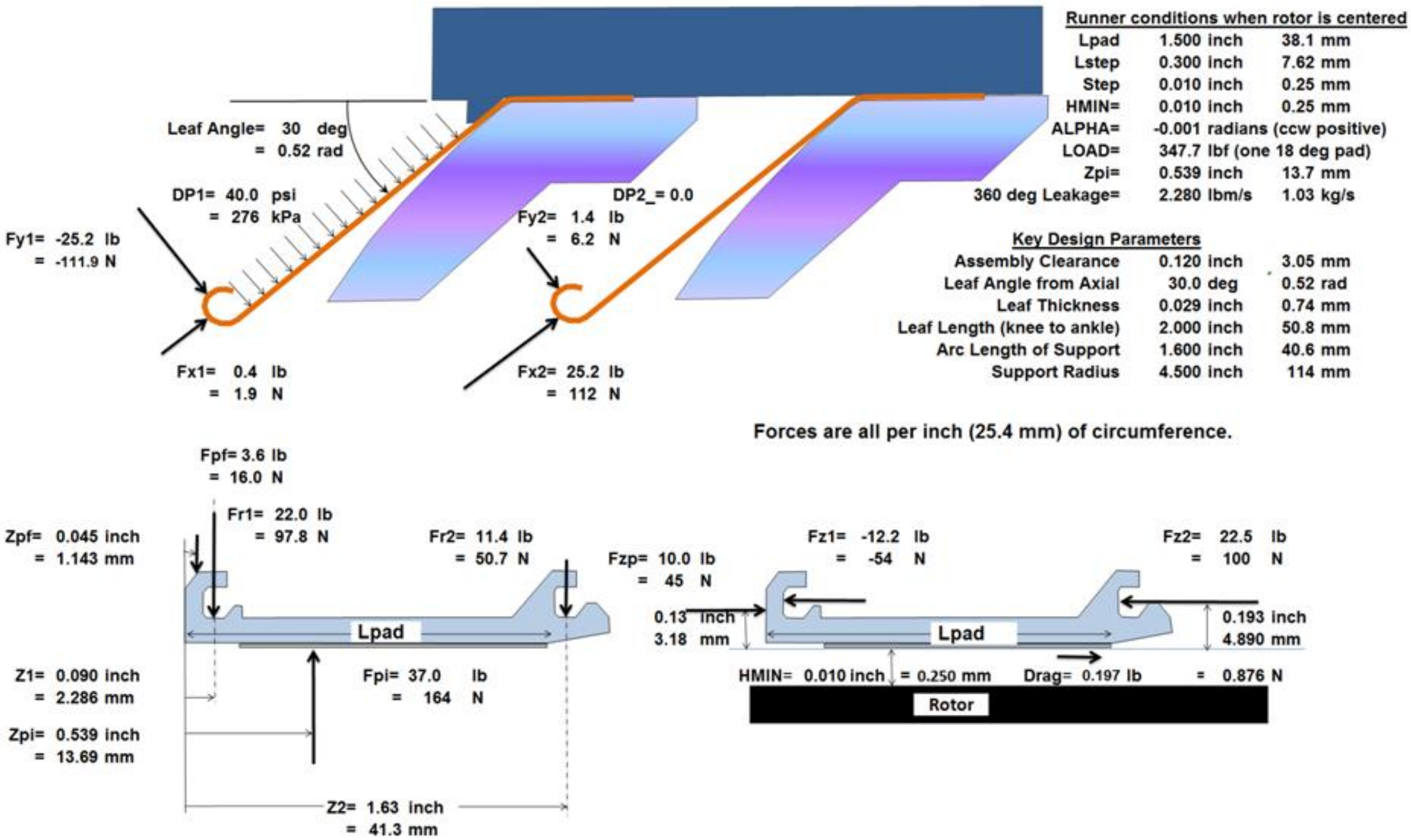

Figure 9. Leaf and Runner Forces, Centered.

may contact the leaf support members all the way to the end of the supports. The last point of contact between the leaves and support is found by calculating the moment required to bend the leaf to the same curvature as the support and then calculating the point along the leaf at which that moment first occurs. That point is taken as the “built-in” point in the beam analysis process with force being exerted on the leaf by the support and moment transmitted through the leaf. Adjustment to the stiffness of the leaves is also made for the taper (reducing width) of the leaves between the knee and the ankle due to the changing radius. For the leaf analysis the force vectors acting between the runners and leaves are resolved into components perpendicular to the leaves (causing bending) and parallel with the leaves (causing tension/compression in the leaves) . As the leaves bend the tension/compression forces also contribute additional moment acting to bend the leaves which is included in the analysis.

Although there is friction between the leaves and the runner at the pivot location, it is not included in the force/moment analysis to determine the stable operating position. The curl at the ends of the leaves is sized to be tight within the slot on the back of the runners, referred to as the ankle, in order to seal against leakage. The frictional forces from rotation of the leaf curl within the ankle slot has been neglected in the leaf deflection analyses because machinery vibration and fluid dynamic fluctuations will be constantly moving the runners a miniscule amount and relieving the torque. That friction is a key damping mechanism to assure dynamic stability of the seal and is included in dynamic analysis. Without rotating motion at the pivots there is no moment transmitted between the leaves and runners. A feature on the center leaf for each runner resists the torque from fluid flowing between the runner and the rotor.

The rear set of leaves does not bear pressure because there are open spaces between leaves as shown in Fig. 1, but they do function to provide nearly parallel translation of the seal runner and when displaced exert force on the runner, resolved as Fz2 and Fr2. By judicious selection of design parameters the seal is designed for nearly the same deflection of each set of leaves in order to cause the runner to remain essentially parallel with the rotor surface. The changing hydrodynamic/hydrostatic force and center of pressure is incorporated into the force and moment balance. The analysis allows the runners and leaves to move freely until all forces and moments are in balance. This includes applying the fluid forces appropriate to the angle between the runner and the rotor surface. This angle is a maximum of about -0.3 degrees (-0.005 rad) for the runner closest to a 0.100 inch (0.762 mm) eccentric rotor and it is about +0.01degrees (0.0002 rad) for the runner farthest from the 0.100 inch (0.762 mm) eccentric rotor. At the nominal operating point with the rotor centered within the seal the runner is essentially parallel to the rotor surface.

The forces acting on the runners are the force vectors from the leaves acting at the ankles decomposed into components Fr1, Fz1, Fr2 and Fz2. Additionally, the seal runner has radial hydrostatic-hydrodynamic lifting force, Fpi, and axial film force, Drag, which are functions of the clearance with the rotor seal surface. There also is differential seal pressure loading on exposed upstream and downstream surfaces, Fzp, and a radial force, Fpf, due to the full upstream pressure acting radially inward forward of the ankle.

The same force vectors acting between leaves and runners can be decomposed into components parallel, Fx1 and Fx2, and perpendicular, Fy1 and Fy2, to the leaves for use in analyzing deflection of the leaves. As the leaves move, the angle formed at the ankles changes and so the decomposition of this force vector must be constantly recalculated. Additionally, the leaves have a moment and force acting at the built-in location such that all forces and moments are in balance. An iterative balance of forces and moments was performed to satisfy equilibrium requirements consistent with the elastic properties of the seal leaves and the radial position of the seal runner. Included in the analysis is consideration of the optimal axial placement of the seal runner with respect to the forward set of seal leaves and selection of seal leaf properties. Parameters selected satisfy equilibrium requirements as well as producing small seal clearance at normal operating conditions for reduced seal leakage flow and providing non-contacting clearance at a large transient eccentricity of 0.100 inch (2.54 mm).

Figure 10 is a plot of results that shows both small operating clearance, 0.010 inch (0.254 mm), when the rotor is centered within the seal (eccentricity = 0), typical of steady state turbine operation, and a non-contacting clearance of 0.002 inch (0.051 mm) at an eccentricity of 0.100 inch (0.254 mm). Leaves in this design are 0.029 inch (0.737 mm) thick and 2.0 inch (50.8 mm) long. Unpressurized seal runner clearance with the rotor is 0.120 inch (3.05 mm), and the radius of the support under the seal leaves is 4.5 inch (114 mm). The runner radial position

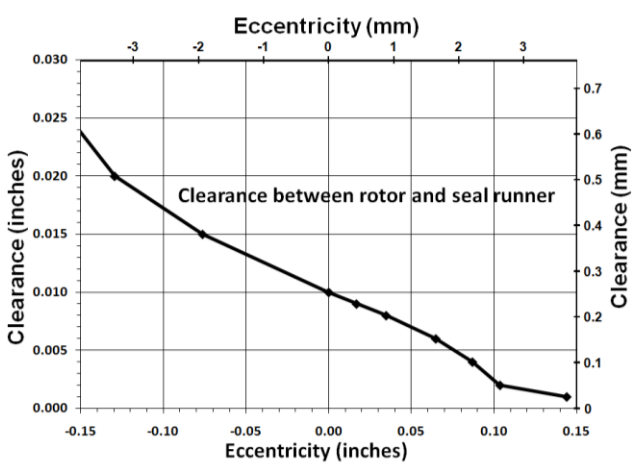

Figure 10. Seal Clearance vs. Eccentricity.

and flexure of the leaves, changes with the eccentricity of the rotor with respect to the seal as illustrated in Fig. 11. The eccentricity can be caused by either rotor displacement within an aligned seal arrangement, or it can be caused by non-uniform radial displacement of the turbine stator component in which the seal is mounted. Eccentricity under normal operating conditions, where low seal leakage is essential, is the sum of component alignments, stator radial fit, machine surface run out, and rotor seal surface run out, and is likely less than 0.02 inch (0.51 mm). The results plotted in Fig. 10 and 11 are for individual, 18-degree (0. 314 rad), seal segments. When the maximum eccentricity is occurring at one seal segment location the other side of the turbine has large clearance, i.e. ~ 0.017 inch (-.431 mm) with -0.100 inch (2.54 mm) eccentricity. The average seal clearance of all segments at steady state operating conditions is expected to be near 0.010 inch (0.254 mm) when the rotor is concentric with the seal. With reference to Fig. 6, leakage of approximately 1.2 lb/sec (0.54 kg/s) is predicted under the runners. Total seal leakage will include leakage through seal leaf slots and passages between seal runners. Those leakages are estimated to be 0.8 lb/sec (0.36 kg/s)and 0.1 lb/sec (0.04 kg/s) respectively, giving a total seal flow of 2.1 lb/sec (0.95 kg/s).

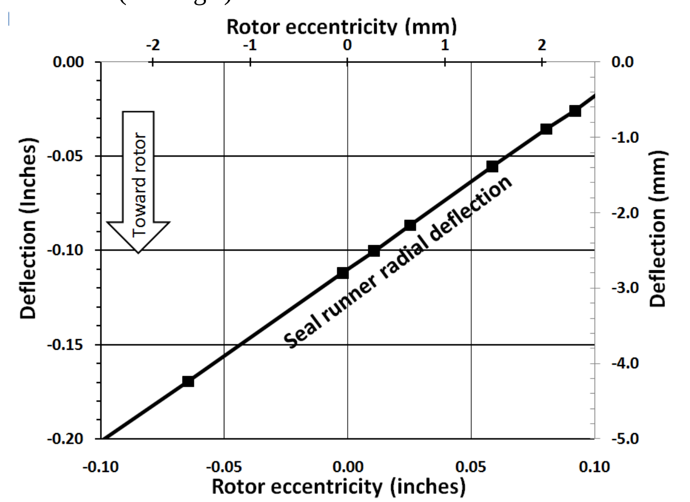

Figure 11. Seal Runner Radial Deflection vs. Eccentricity.

SEAL VALIDATION TEST PLANS

The hydrostatic-hydrodynamic film analysis showed that essentially all of the seal runner lifting force is derived hydrostatically. Further analysis comparing ambient temperature performance with 1000 F (538 K) predicts similar film forces. Therefore, meaningful seal validation can be conducted in static testing at room temperature. Initial tests of two-dimensional seal segments that are full size in cross-section but short in length is planned. This testing will demonstrate seal leaf interaction with the seal runner. Varying the radial height of the 'rotor' seal surface under the seal runner will model a range of seal eccentricity. Applied differential seal pressure will produce the balance of hydrostatic and seal leaf forces contributing to seal clearance. Seal clearance can be measured by depth micrometer and seal leakage by flow meter.

This 2-D static testing is expected to validate predictive seal design tools and provide observation of the dynamic behavior of the seal assembly by ring testing the 2-D model and acoustic monitoring during flow testing.

A second static test is planned of a full-scale film riding seal segment in three-dimensions. This testing will more closely model all seal leakage paths and seal leaf interaction with the seal runner when transitioning from shutdown seal clearance to simulated operating conditions. As in 2-D testing, the radial height of the 'rotor' surface can be shimmed to model a range of assumed eccentricity and seal clearance measured by depth micrometer probes at several locations. Proximity probes may also be used for measurement of dynamic behavior. Full segment 3-D static testing will demonstrate seal operability and leakage flow over the range of anticipated transient eccentricity.

These static model tests are expected to provide significant insight to seal behavior before more expensive, sub-scale, rotating testing is conducted. The dynamic response of the seal will be a primary objective of rotating rig testing. This will include studying the response to shaft run-out and fluctuations in upstream pressure as well as measuring the behavior over the full range of rotational speed to assure that there are no critical frequencies at which there is a risk of contact between the runners and the shaft. If adjustments to mass or damping are shown to be required, those steps will also be taken and tests repeated.

CONCLUSIONS

The feasibility of the Film Riding Pressure Actuated Leaf Seal design concept has been evaluated in design studies and appears to be viable. Film flow analysis identified seal runner geometry with adequate hydrostatic radial lifting force to maintain non-contacting seal clearance at normal operating conditions and over a large range of eccentricity of the rotor with respect to the seal. The seal clearance calculation included analysis of seal runner lifting force, seal differential pressure, and seal leaf forces acting on a seal segment runner. Predicted seal clearance at steady state, base load, operating conditions is 0.010 inch (0.254 mm) in the example seal design. A significant additional finding is that seal clearance change in this design is insensitive to seal eccentricity, varying only 0.002 inch (0.05 mm) with 0.03 inch (0.76 mm) eccentricity. This flat response of clearance with eccentricity will facilitate the design of low-leakage, non-contacting seals without precise knowledge of seal alignment or transient seal eccentricities. Validation of these results in relatively inexpensive static testing outlined will provide a confident basis to continue their development. The Film Riding Pressure Actuated Leaf Seal is a promising concept for improved shaft seal in power generation applications.

ACKNOWLEDGMENTS

Mr. Wilbur Shapiro is acknowledged for the hydrostatic-hydrodynamic seal analysis reported herein.

NYSERDA, New York State Energy Research and Development Authority, is acknowledged for work sponsored under Agreement No. 9887 that is background to the material presented. The opinions expressed in this report do not necessarily reflect those of NYSERDA or the State of New York.

NOMENCLATURE

Ankle - Features on the outer diameter of runners that engage the leaves and provide for pivoting of the leaves relative to the runners.
DP – Differential pressure across seal
FRPALS – Film Riding Pressure Actuated Leaf Seal
Fr – Radial component of ankle force vector
Fx – Ankle force component parallel with leaf
Fy – Ankle force component perpendicular to leaf
Fz – Axial component of ankle force vector
ID – Inner Diameter
Knee – Location at which leaves are held between support and backing ring. Beyond this point leaves are free to bend.
NYSERDA – New York State Energy Research and Development Authority
OD – Outer Diameter
PALS – Pressure Actuated Leaf Seal

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