Compliant Turbomachine Sealing

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Abstract

Sealing interface materials and coatings are sacrificial, giving up their integrity for the benefit of the component. Seals that are compliant while still controlling leakage, dynamics, and coolant flows are sought to enhance turbomachine performance. Herein we investigate the leaf-seal configuration. While the leaf seal is classified as contacting, a ready modification using the leaf-housing arrangement in conjunction with an interface film rider (a bore seal, for example) provides for a film-riding noncontact seal. The leaf housing and leaf elements can be made from a variety of materials from plastic to ceramic. Four simplistic models are used to identify the physics essential to controlling leakage. Corroborated by CFD, these results provide design parameters for applications to within reasonable engineering certainty. Some potential improvements are proposed.

Introduction

Brush, finger, and foil seals are representative of compliant seals. Their modeling and applications and other compliant seal configurations are described elsewhere (e.g., see [1] to [4]). Herein we consider some design parameters for the leaf seal described by Nakane et al. [5], Flower [6], and Steinetz and Sirocky [7] with some potential for modifications such as adding an elastic retainer at the outer perimeter, a film-riding ring interface, and side-wall sealing (Figs. 1 and 2).

Several types of retainers and film-riding rings can be envisioned, including the bump foil, garter springs, and sealing elastomers, with shaft or wave-riding interfaces to enable the linear and rotary motions depending on the stiffness requirements, leaf geometry, and rotor interface.

Static and dynamic interface clearance and wear characteristics can be controlled by adjusting elastomer tension. With active control such adjustments can be implemented upon demand through use of piezoelectric actuators for example and a tension band (ring) acting on the elastomer. By controlling the retainer ring tension, the effective leaf seal stiffness can be higher than the brush seal and float within the housing confinement. This implies, for example, that Si_3N_4 wafers or composites or coated plates can be used. It also implies that functionally graded materials can be used as plates. Segmented seals could be made similar to those for the brush seals and like brush seals can be waxed to facilitate installation, and the wear interface can be coated.

The importance of the positioning and functioning of O-ring or piston-ring-type side-wall seals both on the low- and high-pressure side walls cannot be overemphasized. These rings must be close to the interface to mitigate leaf leakage and facilitate leaf dynamics, otherwise excessive leakage can be expected (Fig. 2).

Herein, we (1) present and discuss experimental leakage data for four simplistic models that capture the essential leaf and side-wall physics, (2) corroborate these findings by computational fluid dynamics (CFD) modeling, and (3) suggest some potential improvements.

Modeling

Usual practice is to make and test models that are scalable with high-order accuracy. For exploratory purposes however, the concept herein is that it is only necessary for the model to capture the fundamental physical parameters of a component and validate experimental data via CFD. Once validated, one can then predict with reasonable engineering certainty actual physical components for engines.
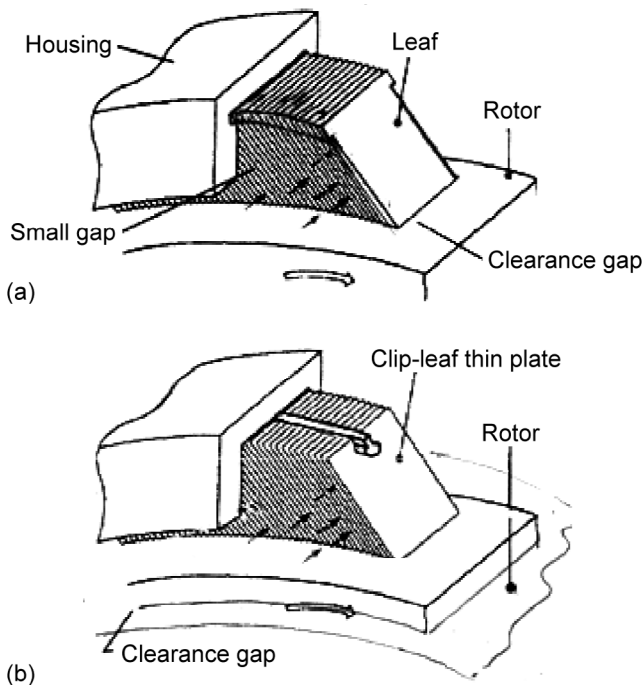
Figure 1.—Compliant interface seal configuration.
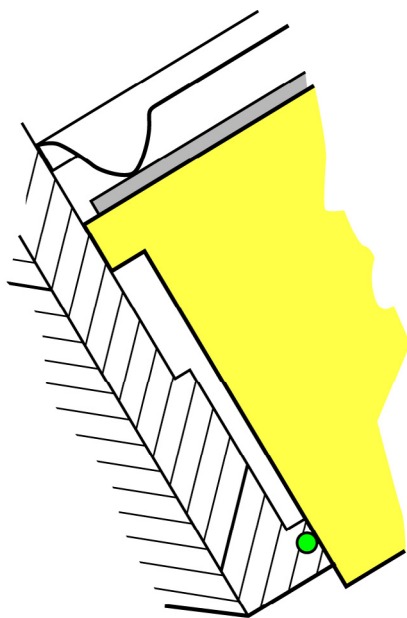(a) Contact. (b) Floating ring.

Figure 2.—Peripheral loading and side-wall contact
seal (green dot) to permit sliding motion with O-ring,
piston-ring-type, or sliding-contact side-wall seals
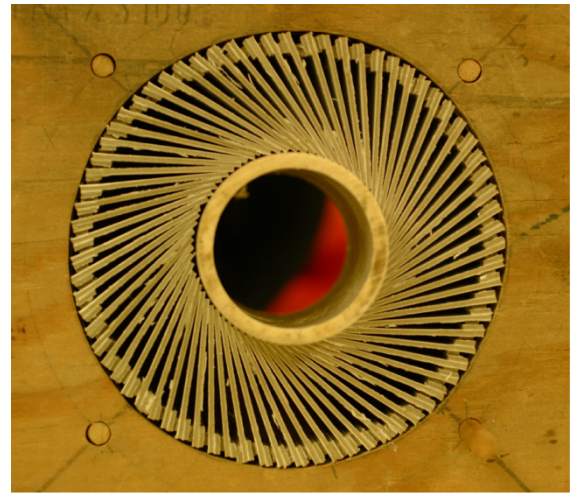as close to interface as prudent design will permit.
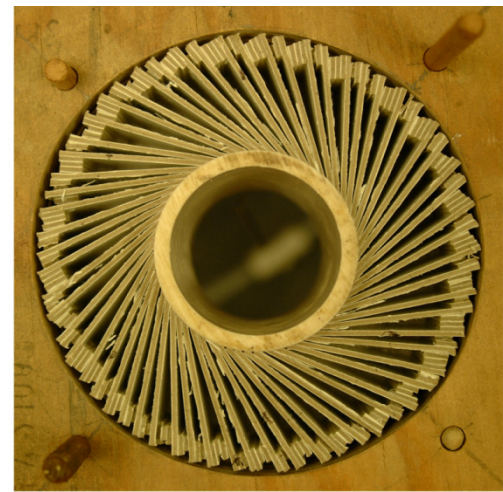

Figure 3.—74-leaf seal configuration. Leaves are
tight packed.

Figure 4.—63-leaf seal configuration. Leaves
nearly touch one another at inside diameter
and do touch simulated shaft.

With that in mind we undertook to fabricate several models that captured the essence of leaf sealing, acquired flow data over a small range in pressure differentials, and determined how well these data are represented by CFD modeling. These models are all static configurations, as rotation accounts for a small decrease in the leakage and dynamics will, if hysteresis is involved, enhance leakage. We first want to establish some concepts for the leakages.

Four such models were fabricated based on the principles of leaf geometry relative to a bore seal configuration. The first model is that of a 74-leaf, tight packed configuration (Fig. 3). These leaves in theory overlap, yet in practice the surfaces deform and are packed about the simulated shaft. In each case an elastomer is used to simulate the outer wave spring that sets the leaves into contact with the shaft and restores sealing during seal dynamics. Here the elastomer also held the leaves into place while facilitating assembly and disassembly. In a real configuration, waxing of the leaves would serve this function and would burn out with operation; the wave spring then becomes effective.

Decreasing the number of leaves to 63 (Fig. 4) allows a configuration without overlapping geometries and similar to

Fig. 3, except that at the outside perimeter, the leaf spacing is alternately increased to account for the change in circumferential spacing.

Details of the leaf configurations are given in Table 1.

TABLE 1. LEAF GEOMETRIES IN SEALS OF FIGS. 3 TO 6
[See schematic in Fig. 7.]

Number of leaves, N	533	74	63	63
Spacing	uniform	uniform	alternating	alternating spring
Figure	5	3	4	6
Thickness, t, in.	0.008	0.052	0.052	0.052
Leaf angle, θ, deg. [a]	40 to 60	33	25/32	22/25
Flow area, in.^{2}	0.2274	0.19	0.228	0.228
Outside diameter, OD	4.38	4.38	4.38	4.38
Inside diameter, ID	1.9	1.9	1.9	1.9
Leaf length, Lo, in.[b]	1.5	1.5	1.5	1.5
Leaf width, W, in.	1.125	1.125	1.125	1.125
Fence O-ring diam., in.	2.25	2.25	2.25	2.25

[a] Angle measured from tangent plane to leaf.

[b] No attempt was made to contour the leaf tip at the leaf-shaft interface.
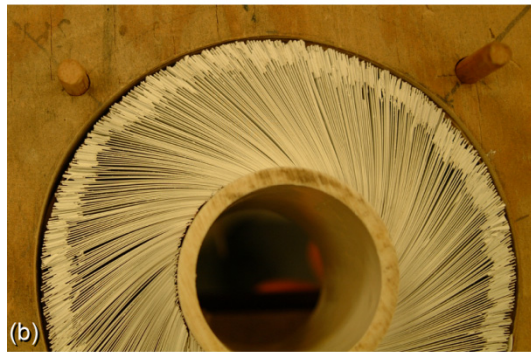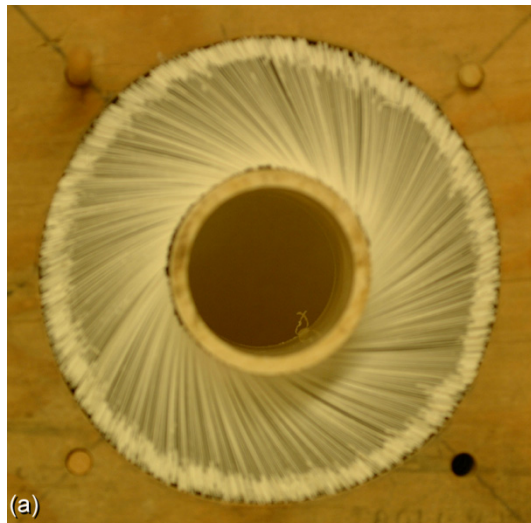

Figure 5.—533-leaf seal configuration. (a) Overview. (b) More detail.

The third and fourth configurations are based on how leaf geometry effects seal leakage. When the leaves are made thin and flexible, which is characteristic of a low-stiffness seal, the number of leaves becomes large (Fig. 5). For leaves of 0.008-in. thickness, compared to the 0.052-in. thickness of Figs. 3 and 4, the number of leaves packed into the seal is $N = 533$.

The fourth configuration considered a closing of the intraleaf triangular spacing generated by the differences between the outer diameter (OD) and inner diameter (ID) providing a leaf spring whereby the inner leaf can be more compatible with the shaft interface and the other leaf spacers of different materials. In Fig. 6(a) the leaves are aligned such as to provide point contact when the principle leaf is flexed while in Fig. 6(b) a line of contact provides additional support to the principle leaf. Leakages for the model of Fig. 6 are similar to the model of Fig. 4, but CFD was not done on this configuration.
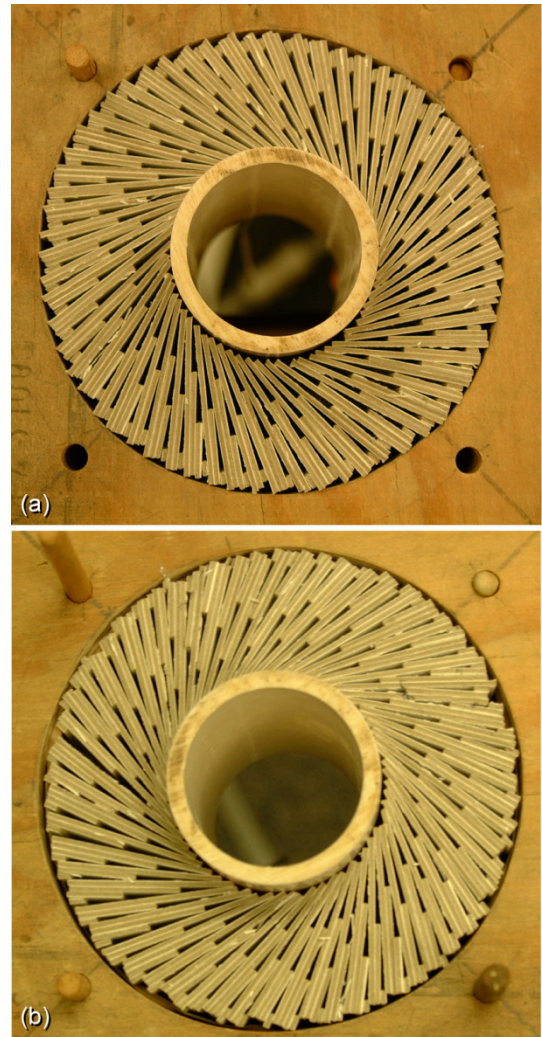

Figure 6.—63-spring-leaf configuration. (a) Spring-leaf point contact (leaves aligned reversed to rotation). (b) Spring-leaf normal contact (leaves aligned to rotation).

These arrangements permit a wider use of materials at the rotor interface and would be beneficial for the case where the leaves are used to spring load a wave-rider rotor, for example. In the wave-rider design, an annular ring (or ring segment) with an inscribed wave [8] would provide a film shielding the shaft from contact while the leaves provide sealing and compliance (Fig. 1(b)).

Analysis

Flow Through a Slit

Consider a leaf. In general it will be tapered or made in some manner to compensate for the inner and outer circumference. In a brush seal riding on a shaft at its inner circumference, the bristle pack itself is tapered (thicker at shaft and thinner at outer diameter). As the average taper is at this point unknown, we shall consider the geometry at the midplane of the leaf to be representative. This is conservative in that the housing will cover the leaf pack at this point, yet it may compensate for neglecting leakage at the shaft interface. For modeling convenience, we use the leaf midplane and simulate the leakage path as that of a slit (Fig. 7).

In the 74-leaf configuration (Fig. 3), the tight-packed leaves are deformed near the rotor interface. In the 63-leaf configuration (Fig. 4), which is essentially the 74-leaf with 11 leaves removed, tight packing is provided without deformation.

The 533-leaf configuration (Fig. 5) is tight packed with some deformation principally between the seal dam and the interface. Flexibility and sizing of the leaf materials near the interface becomes important as an error of 0.00005 in. in average thickness can result in a 0.043-in. overall error in packing. For some real seal configurations with some 2000 leaves, such an error would be an intolerable 0.1 in.

The 63-spring-leaf configuration is slightly underpacked (Fig. 6).

The leakage flow is simplistically given for an aperture flow model as

$$m = C_D A \sqrt{2\rho\Delta P} = C_D A \left(M_e P_e \sqrt{\frac{\gamma}{RT_e}} \right) \qquad (1)$$

and the exit Mach number is given as

$$M_e = \sqrt{\left[\left(\frac{P_t}{P_e} \right)^{\gamma-1/\gamma} - 1 \right] \left(\frac{2}{\gamma-1} \right)} \qquad (2)$$

where

C_D is the flow coefficient
A is flow area
ρ is density
P is pressure
P_e is exit pressure
γ is the ratio of specific heats
R is the gas constant
T_e is the exit temperature
P_t is total pressure

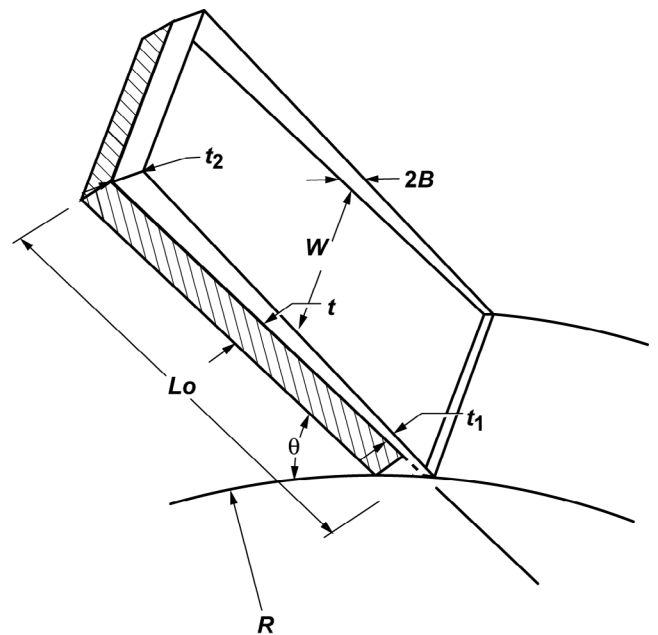**Figure 7.—Tapered-leaf-equivalent rectangular plate and slit-equivalent leakage space 2B×W×Lo. The leaf acute angle θ is measured from the rotor tangent plane to the leaf face. t is leaf thickness; t₁, t₂, and 2B are leaf thickness at interface and seal dam and the average, respectively.**

The parameter $C_D A$ accounts for many unknowns including actual flow area, leaf geometry, friction, and tolerances. Determining the actual values for these parameters is difficult with only an estimate made for the flow area A. C_D is then determined empirically.

CFD Model

The commercial numerical code CFD-ACE+ (CFD Corp., Huntsville, AL) was used to model the 533-, 74-, and 63-leaf configurations. Using the CAD modeling program, geometries for the leaf seal can be constructed. These models serve as the basis for the flow calculations and determination of both flow areas and leaf angle assuming that the leaf contact points with simulated rotor are fixed and leaf interference is allowable. The wire frame model and pressure profiles are illustrated in Fig. 8. The boundary conditions of the triangular volumes are solid wall from the OD to the sealing dam, as are the leaf interfaces and rotor. At the inlet and exit leakage gap the pressure is constant with atmospheric at the exit. The pressure drop is significant at the inlet and exit with little to moderate changes over the width of the leaf, somewhat reminiscent of a labyrinth cavity. Thus, for these seal configurations, the radial position and sealing capability of the inlet and outlet O-ring, piston-ring-type, or rubbing contact side seal become very important.

Variations in flow area and leaf angle with the number of leaves for fixed-leaf, simulated-shaft, and housing geometries is illustrated in Fig. 9.

Grid independence was checked. One needs to also take care in specifying that the inlet pressures and temperatures are stagnation values; otherwise the code will not converge
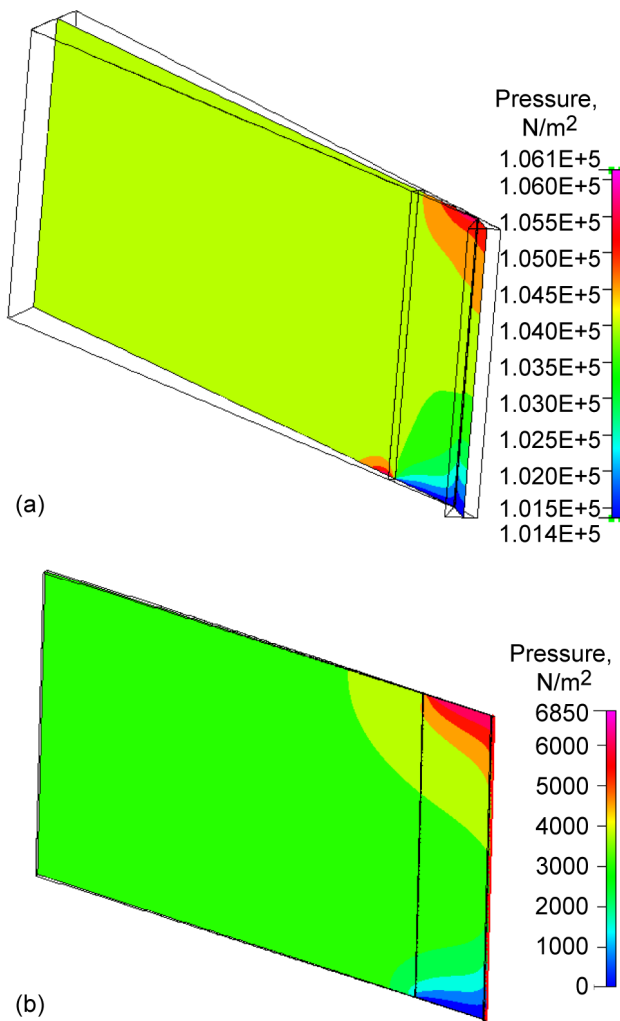
Figure 8.—Wire-frame computational model and corresponding pressure distribution for leaf-seal configurations at 1 psi pressure drop. (a) $N = 63$ leaves at 28°. (b) $N = 533$ leaves at 50°.
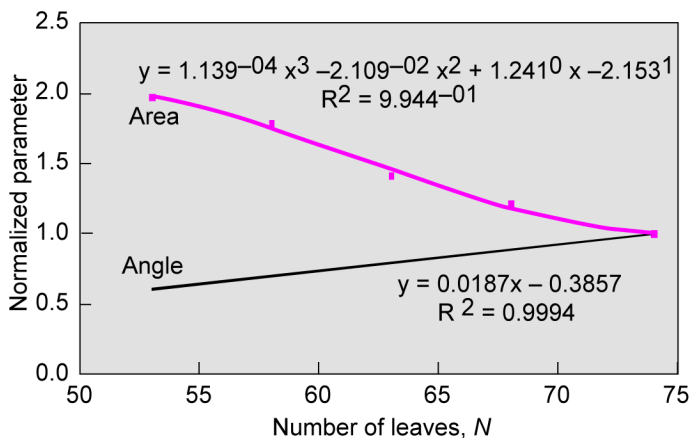

Figure 9.—Flow area and leaf angle for N-leaf seal configuration, normalized with respect to 74-leaf configuration.

properly. Also checked were the effects of turbulence and leaf blunt-edge inlet (by adding a plenum region), which lowered flow rates between 8 and 10 percent in each case for the $N = 63$ leaf configuration. These real effects are to be considered in leaf-seal design and indicate a flow rate lower than predicted.

Results

Flow within the leaf seal, like the labyrinth seal, is relatively unperturbed and dominated by the exit piston-ring or O-ring side-wall sealing. The difference between the outer and inner circumference engenders large unimpeded flow apertures as illustrated by the streak lines and velocities shown in Fig. 10 and the near constant pressure in the cavity region shown in Fig. 8. Figs. 10(a) and (b) illustrate the main flow path with low circulation in the leaf cavity on the centerline between leaves at 1 psi pressure drop for the configurations $N = 63$ leaves at 28° and $N = 533$ leaves at 50°, respectively. In both cases, the flow rapidly expands through the inlet and the exit, but for $N = 533$ the flow is near the sealing dam at the inlet and more uniform at the exit. Readily apparent are the differences in the circulation patterns within the cavity region: in Fig. 10(a) it is more inertia dominant and in Fig. 10(b), more viscous dominated. Figs. 10(c) and 10(d) contrast the differences in streak lines within the leaf cavity and illustrate a more uniform behavior for the $N = 533$ configuration. This also serves to illustrate both the function and importance of locating the side-wall seals (piston-type, O-ring, or sliding contact seal) close to the rotor interface.

As can be seen from Fig. 11, the leakage is marginally decreased as the packing of the leaves is increased from 63 to 74, yet a considerable decrease is seen for $N = 533$. The temperature is ambient and the pressures are inlet-stagnation values.

For the $N = 74$ model (Fig. 3) the simplistic leakage model gives $\langle C_D \rangle = 0.7$ over the range of data which is in reasonable agreement with other known aperture flow experiments. The CFD solution for this same model provides a reasonably good prediction to within 8 percent of the data. The effects of turbulence were checked and the effects of entry losses were found by adding an upstream plenum region. Each were found to decrease the leakage by 8 to 10 percent, which places experiment and analysis in reasonably good agreement. The effects of irregular leaves, interface contact and laminate irregularities of the experimental model can all augment leakage, yet are difficult to assess.

CFD simulations illustrate that the leaf angle and leaf interference can play a role in leakage as illustrated in Figs. 9 and 11 for the $N = 63$ leaf configuration where the angle is changed from 33° to 28° to 23°. The leaf angle is measured from the tangent plane to the simulated rotor. Because of the inserted spacers, the leaf angle alternated from 25° to 32° every other leaf, and as such 28° would be a reasonable average (see Table 1). It is seen that the data follow the CFD solution to a point and then depart to a lower leakage more closely associated with the 25° angle and may be affected by turbulence. Correcting the $N = 63$ leaf at 28° for inlet losses and turbulence underpredicts leakage, in good agreement with experimental results (Fig. 11). There are many flaws in the crude experimental models, yet the principle physical
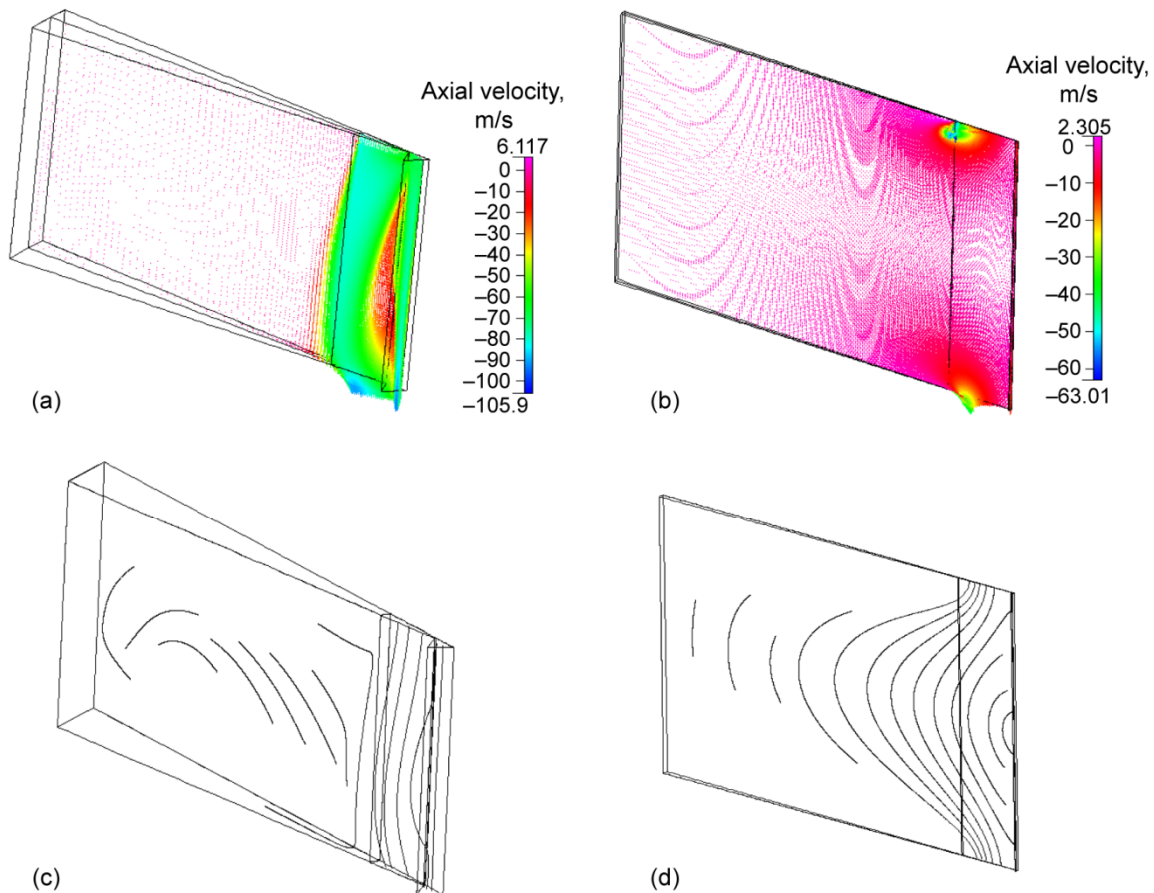
Figure 10.—Interleaf velocity and streak lines for the configurations *N* = 63 leaves at 28° and *N* = 533 leaves at 50° at 1 psi pressure drop. Parameters at midplane between leaves. (a) Velocity vectors for 63-leaf configuration. (b) Velocity vectors for 533-leaf configuration. (c) Streak lines for 63-leaf configuration. (d) Streak lines for 533-leaf configuration.
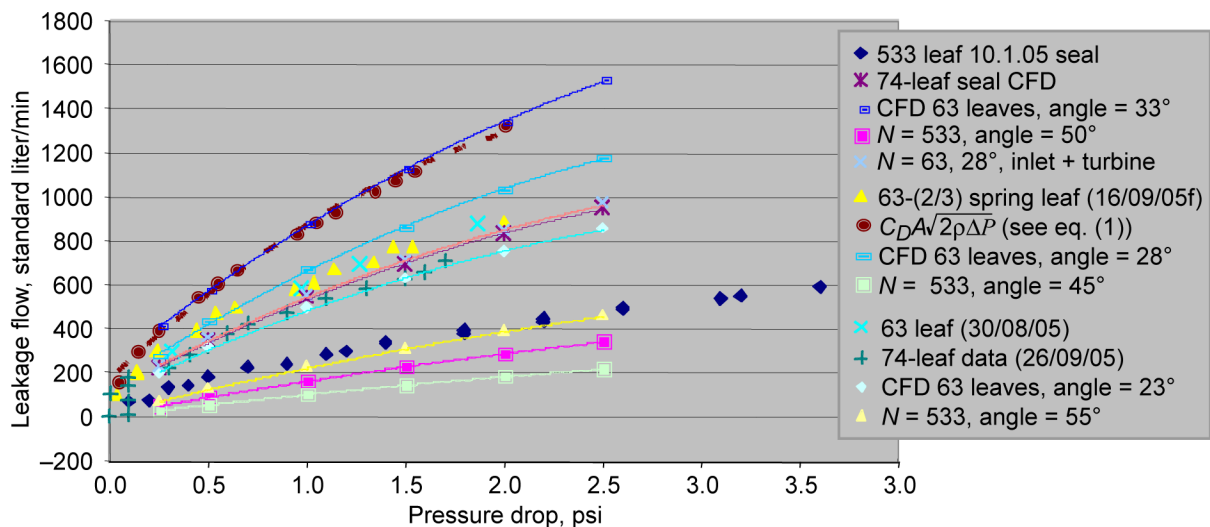
Figure 11.—Leaf seal data and simulation. Data for four configurations: 533 leaves, 74 leaves, 63 leaves with alternating spacing, and 63 spring-leaves (2-leaves/3-leaves) with side plate filter paper seals. CFD solutions for *N* = 533 at leaf angles of 45°, 50°, and 55°; *N* = 74 at leaf angle of 33°; and *N* = 63 at leaf angles 23°, 28°, 33°. CFD solution for simplified aperture flow model is also illustrated. Standard ft³/min = 35.315 standard liter/min.

parameters have been retained and are captured by the CFD model.

For $N = 533$, the leaves are distorted and curved between 40° to 60°, where 50° seems most reasonable. For this case, the predictions are less promising, and model losses are significant. Leakages for leaf angles of 45°, 50°, and 55° are shown in Fig. 11.

Potential Improvements in Design

Suggested design improvements include diamond like coatings (DLC) [8] that could be applied to the leaf interfaces to resist abrasion and provide low friction if the operating temperatures are less than 500 °C (930 °F). For example, in a wave bearing application, Dimofte et al. [8] tested coatings applied to both the rotor and stator: (1) silicon diamondlike carbon (SiDLC), (2) diamondlike carbon (DLC), (3) tungsten carbide/carbon (WC/C), and (4) titanium carbide (TiC). SiDLC performed well over 1000 start-stop cycles followed by an additional 50 oil-off cycles where friction torque lockup occurred in about 10 min. WC/C also did well except at oil-off, because of higher friction. TiC failed at oil-off, while DLC ran 20 min, but seized on the first cycle. For new coated bearings operating at low supply pressures both SiDLC and WC/C performed well, yet at oil-off WC/C degraded rapidly (3 min) versus 2½ hr for SiDLC. Tribologically, SiDLC performed very well for the wave bearing tested and currently would be the coating of choice for the plates and the rotor surface; however, statistically significant data are still required [9].

Improvements in the design may be found in the use of spring-loaded side-plate sealing, for example, similar to a piston ring. This will reduce the flow through the open triangular passages between leaves due to the change in radii. In the leaf seal experiments, filter paper side-wall seals were used, and of course this is a static configuration.

The use of the OD wave spring will be useful in many cases to provide fresh interface, as this is a contact seal and will wear like the brushes in an electric motor, a little at a time, until they wear out. It will be necessary to provide leaf configurations that will not disengage from the holder.

It may be that radial triangular-shaped leaves will also work, as they would be bidirectional for rotation; that is, the shaft can be rotated either way without damage. The OD wave spring would keep them in line, yet low-friction interfaces would still be needed, otherwise they would lock up; a SiDLC coating would be of great value on the leaves, yet make the seal more costly. So there will become a tradeoff between closing down the triangular area due to differences in radii and the restoring moments or leaf hysteresis. The leaves could be made triangular with a small bulge (like the head of a pin or needle) that will space them a slight distance apart and keep them from friction locking. These plates can then be quite short and would make a very compact seal with high length-to-diameter ratio (L/D) versus the conventional ones with low L/D.

In our models, the leaf interfaces are not contoured to the shaft. In a real seal, this would become a necessity except in the case of a radial leaf configuration. Radial leaves do not need to be a "solid pack" but could be optimized between hysteresis effects, the weight penalty, and the effectiveness of sealing. This same criteria can be applied to other configurations as well as the use of side plates.

The end plates or side plates need to have ID that are a few percent—about 5 percent—greater than the largest clearance requirement to ensure noncontact. In most cases a single labyrinth tooth upstream and downstream would suffice to provide sure sealing in case of leaf failures; however, in extreme cases up to three at each end can be used; for that matter any number can be used and one essentially has a leaf-augmented labyrinth seal rather than a labyrinth augmented leaf seal. Usually the labyrinth is more effective on the rotor with abradable materials for the stator interface. Here the labyrinth is less effective yet essential to the safety of the turbomachine.

Leaf-seal leakages could be improved If a compliant material could be found to fill the triangular gaps between the leaves while still enabling interface dynamics.

As cited prior, the materials of the leaf-spring seal can differ. The material interfacing with the shaft needs to have good wear characteristics and withstand a lot of heat. The shorter ones may be of different materials. It becomes a manufacturing-cost-weight benefits issue.

Patents by Shinohara et al. (Japan) [10], Flower (Cross Mfg, England) [6], and Steinetz and Sirocky (United States) [7] need be considered prior to manufacturing. Herein we have added to these concepts and then compared the modeling methods along with the results and suggested methods of improvements.

Conclusions

Experimental data from simple models using convenient working fluids, (e.g., air and water) that capture the essential physics, supported and verified by CFD, were used to provide scalable parameters for design of leaf-seal configurations for applications, to within reasonable engineering certainty. Simple experimental tests on four leaf-seal configurations have identified principle physical parameters, and CFD was employed to verify that these parameters can be estimated and applied to sealing designs.

Leaf-seal internal flow and pressure drop characteristics are somewhat similar to a labyrinth seal cavity yet thin leaf flow characteristics tend to be dominated by viscous effects with a skewed flow expansion at the inlet.

Leaf-seal leakage can be similar to brush seal leakages, and indeed the patents are derivatives of those issued for brush seals.

Several potential improvements have been delineated including dynamic or spring-loaded side plates, a compliant interleaf filler, sandwiching the leaves between sets of labyrinth teeth, and a floating-leaf-pad interface seal providing a film-riding noncontact seal.

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13. SUPPLEMENTARY NOTES

14. ABSTRACT
Sealing interface materials and coatings are sacrificial, giving up their integrity for the benefit of the component. Seals that are compliant while still controlling leakage, dynamics, and coolant flows are sought to enhance turbomachine performance. Herein we investigate the leaf-seal configuration. While the leaf seal is classified as contacting, a ready modification using the leaf-housing arrangement in conjunction with an interface film rider (a bore seal, for example) provides for a film-riding noncontact seal. The leaf housing and leaf elements can be made from a variety of materials from plastic to ceramic. Four simplistic models are used to identify the physics essential to controlling leakage. Corroborated by CFD, these results provide design parameters for applications to within reasonable engineering certainty. Some potential improvements are proposed.

15. SUBJECT TERMS
Seals; Coatings; Thin films; Turbomachines; Life

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