Computational Fluid Dynamic Simulation of a Straight Labyrinth Seal for Leakage and Thermal Characterization

by
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Approved:

__________________________
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List of Symbols

\( t \) – Distance between adjacent knife edges (inches)
\( h \) – Knife edge radial height relative to rotor base (inches)
\( w \) – Axial width of knife edge base (inches)
\( c_r \) – Radial clearance between rotor and stator land (mils)
\( b \) – Axial width of knife edge tip (mils)
\( \theta \) – Knife edge angle relative to rotor base (degrees)
\( f_1 \) – Fillet radius between knife edge base and rotor base (mils)
\( f_2 \) – Fillet radius between knife edge tip and knife edge base (mils)
\( r \) – Outer radius of rotor base (inches)
\( r_{L,1} \) – Inner radius of stator at domain inlet (inches)
\( r_{L,2} \) – Inner radius of stator at domain exit (inches)
\( t_L \) – Thickness of seal land (inches)
\( L_1 \) – Axial distance to start of stator land (inches)
\( L_2 \) – Axial distance to end of stator land (inches)
\( L_3 \) – Axial length of stator and rotor (inches)
\( P_{up} \) – Upstream static pressure (psia)
\( T_{up} \) – Upstream total temperature (psia)
\( RPMF_{up} \) – Upstream non-dimensional swirl velocity
\( P_{down} \) – Downstream static pressure (psia)
\( \Omega \) – Angular speed (rad/s)
\( PR \) – Pressure Ratio (-)
\( U \) – Linear wheel speed in tangential direction (ft/s)
\( RPMF \) – Non-dimensional swirl velocity; Normalized by wheel speed (-)
\( V_{rel} \) – Tangential velocity relative to the rotor (ft/s)
\( M_{rel} \) – Mach number relative to the rotor (-)
\( \gamma \) – Ratio of specific heats (-)
\( R \) – Gas constant for air (-)
\( T_s \) – Static temperature (F)
\( T_i \) – Total temperature (F)
\( V_r \) – Radial velocity (ft/s)
\( V_z \) – Tangential velocity (ft/s)
\( \Psi \) – Stream function (ft\(^2\)/s)

\( Q_{visc} \) – Heating due to viscous forces (BTU/s)

\( Q_{convec} \) – Heat transfer due to convection (BTU/s)

\( C_p \) – Specific heat at constant pressure (BTU/lbm-R)

\( \tau \) – Shear stress (psi)

\( H \) – Heat transfer coefficient (BTU/hr-ft\(^2\)-F)

\( T_f \) – Fluid temperature used in evaluating heat transfer coefficient (F)

\( T_{surf} \) – Surface temperature used in evaluating heat transfer coefficient (F)

\( A \) – Surface Area (sqin)

\( t_{foil} \) – Thickness of honeycomb wall (mils)

Cell Size – Size of honeycomb cell (inches)
Acknowledgements

I would like to thank and acknowledge RPI Hartford and in particular my advisor, Ernesto Gutierrez-Miravete. Ernesto provided valuable guidance and encouragement during the course of this project and went out of his way to ensure the project was successful. Also, I would like to acknowledge my wife and family who provided needed encouragement positive feedback.
Abstract

Labyrinth seals are the most commonly used rotating seal in hot regions of gas turbine engines. Accurately characterizing leakage flow rates, windage heating, and heat transfer coefficients is critical for successful engine design. Numerical simulation of a straight type labyrinth seal is performed using commercial Fluent CFD code. 2D axisymmetric CFD is used to study leakage flow and windage heating for various seal clearances ranging from 5-15 mils. Solid (smooth) stator land geometry is considered. The 2D model is subsequently used to determine average heat transfer coefficients on the rotor and stator for heating and cooling situations. A 3D periodic CFD model is used to study leakage and windage for two honeycomb cell sizes.

Results show that for a fixed seal pressure ratio, leakage flow and windage heat increases with increasing seal clearance. Adiabatic temperature rise across the seal decreases with increasing clearance. For smooth land geometry heat transfer coefficients are significantly higher on the stator land than the rotor surfaces. 3D results indicate relatively large honeycomb cells increase leakage flow, while small honeycomb lands can reduce leakage flow.
1. Introduction

Labyrinth (lab) seals are used to seal between high and low pressure regions of rotor/stator systems. Lab seal are prevalent in industries where durability and robustness are paramount including industrial gas turbines, steam turbines, and gas turbines for aircraft. Figure 1 shows a cross section of an aircraft engine highlighting where lab seals are typically used.

Lab seals are used in the low and high pressure compressors on the stator shrouds. A stator shroud isolates the upstream, lower pressure stage, from the downstream, higher pressure stage. Without effective sealing, a significant portion of the compressor flow would leak backwards through the compressor. These leakages are undesirable because they reduce compressor efficiencies, and can lead to compressor operability issues like stall and surge.

Similarly, in the high and low pressure turbines, lab seals are used to prevent flow from leaking between adjacent stages. Lab seals are sometimes used in low pressure turbines at the blade tips. Turbine leakages are also undesirable because erode turbine efficiency, and can contribute to hot gas ingestion.

Figure 1 Locations where lab seals are typically used gas turbine engines
Figure 2 shows the typical makeup of a lab seal. As mentioned, lab seals are situated between a rotating (rotor) and static (stator) component. One or more seal teeth, or “knife edges”, are positioned on the rotor. The designs of the teeth affect seal leakage and thermal characteristics. Seal teeth can be perpendicular to the rotor, or posited at an angle, as shown in Figure 2.

Seal teeth are designed to be in close proximity with a stator land, which is a special region of the stator where designers anticipate interaction between the rotor and stator. These interactions are commonly referred to as “rubs” because the rotor teeth cut into, or rub into the stator land.

![Figure 2 Key components of a labyrinth type seal; Non-Stepped](image)

Rubs occur when the rotor component grows radially outward more quickly than the stator component. In gas turbine operation, these growths are caused by centrifugal and thermal strains.

Rubs are undesirable, as they can damage or destroy the tips of the seal teeth. However, rubs are also typically unavoidable during aircraft engine transient operation.

In order to mitigate potential damage from rubs, stator lands are often made of a porous material. This allows the seal teeth to cut into the stator land without significant
wear or damage. A porous material of choice in the gas turbine industry is metal honeycomb. Honeycomb comes in various cell sizes and can be made of various materials. Figure 3 shows several varieties of honeycomb used in lab seal stator lands. The honeycomb shown range in cell size between 1/32” - 3/8”.

![Figure 3 Typical honeycomb used for seal lands [Pratt and Whitney]](image)

The ability to accurately model lab seals leakage flows is extremely important. As previously described, leakages can significantly affect compressor and turbine efficiencies, negatively influence engine operability, and cause undesirable hot gas ingestion.

Seal thermal characteristics are also very important to engine designers. The heat generated by viscous dissipation, or windage, in lab seals can result in temperatures that challenge component material capabilities. It is not uncommon for lab seal locations to represent the hottest locations on a given engine component. Lab seals are also exposed to high flow velocities which enhance heat transfer coefficients, further contributing to elevated temperatures.
Literature Review

Lab seals have been the subject of extensive research starting the 1970s. Stocker et al. [1] performed experiments in 1977 measuring the power losses of straight through, and stepped type lab seals. Stocker also reported data regarding windage characteristics of honeycomb and smooth stator lands, concluding that the windage with honeycomb lands was larger than with smooth lands.

Denecke et al. [2] more recently, in 2005, performed experimental work on stepped labyrinth seals. Axial and swirl velocity measurement were taken using laser doppler velocimetry (LDV) techniques. Windage heating and total temperature rise characteristics were determined. Denecke also proposed non-dimensional methods for lab seal characterization [3].

In the past five years a significant lab seal research has been performed using numerical simulation. Yan et al. [4] investigated leakage, total temperature, and windage characteristics of lab seals using 3D CFD. Yan successfully calibrated a 3D periodic CFD model of a stepped lab seal to the experimental results from Denecke. A good match was described with respect to windage heating between experimental results and the numerical simulation. Yan also compared CFD swirl velocity distribution results to Denecke’s LDV data, reporting an improved match relative to Denecke’s 2D axisymmetric CFD simulation.

After calibrated the CFD model, Yan studied the effect of land geometry, smooth versus honeycomb, and seal clearance on leakage flow rates and windage heating. Various seal pressure ratios were considered. Yan concluded the following [4]:

- Leakage flow rates increase proportionally with seal pressure ratio and seal clearance
- Honeycomb (1/16”) increases seal leakage by roughly 10%
- Windage heating increases with increasing seal leakage and increasing seal clearance
In subsequent research Yan et al. [5] studied how swirl velocities upstream of the lab seal affect seal characteristics. Yan also studied the effects of honeycomb cell size on seal characteristics. The following conclusions were made [5]:

- Upstream swirl velocities significantly affect seal windage characteristics
- Lower upstream swirl velocities increase windage, while higher upstream swirl velocities decrease windage
- Upstream inlet swirl velocities do not significantly affect leakage flow rates
- Seals with honeycomb are less sensitive to upstream swirl velocities with respect to windage
- Larger honeycomb cell sizes result in larger leakage flows. 1/8”, 1/16”, and 1/32” cell sizes considered
- 1/32” cell size and smooth land result in similar leakage flow rates
2. Methodology / Approach

2.1 Geometry

This section describes the geometry to be used for the CFD analysis. Figure 4 and Table 1 describe the dimensions of the rotor, stator and the position of stator land. Figure 5 and Table 2 provide a detailed parameterization of the lab seal teeth and stator land.

As shown, the study geometry consists of a rotor and stator section 5 inches in axial length with the seal section positioned in the middle. The flow relative to the geometry is from left to right. The upstream annular gap is 0.4 inches and downstream annular gap is 0.1 inches.

It is worth briefly discussing the tapering of the downstream annular gap. In rotor stator systems with small axial through flows, strong recirculation vortices can be generated. If these vortices exist near a flow outlet it is possible for the outlet to have regions that reverse flow, enter the domain rather than exit. Tapering the downstream annular gap prevents these large vortices and reduces the possibility of having significant reverse flow.

![Figure 4 CFD geometry: Rotor, stator and land description](image-url)
Table 1 CFD geometry: Rotor, stator and land dimensions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$r_{L,1}$</td>
<td>8.4</td>
<td>inches</td>
</tr>
<tr>
<td>$r_{L,2}$</td>
<td>8.1</td>
<td>inches</td>
</tr>
<tr>
<td>$t_l$</td>
<td>0.15</td>
<td>inches</td>
</tr>
<tr>
<td>$L_1$</td>
<td>2.2</td>
<td>inches</td>
</tr>
<tr>
<td>$L_2$</td>
<td>3.25</td>
<td>inches</td>
</tr>
<tr>
<td>$L_3$</td>
<td>5.0</td>
<td>inches</td>
</tr>
</tbody>
</table>

The lab seal geometry consists of three seal “teeth” and is a “straight through” type as opposed to a “stepped” type. The seal teeth are canted at an angle with the tips of the teeth pointed toward the direction of the flow. The dimensions presented in Table 2 are representative of seals typically used in large high bypass-ratio aircraft gas turbine engines. However, the seal described does not represent an actual engine design.

Figure 5 CFD Geometry: Lab seal detail description
Table 2: CFD Geometry: Lab seal detail dimensions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>t</td>
<td>0.275</td>
<td>inches</td>
</tr>
<tr>
<td>h</td>
<td>0.25</td>
<td>inches</td>
</tr>
<tr>
<td>w</td>
<td>0.11</td>
<td>inches</td>
</tr>
<tr>
<td>cr</td>
<td>5-15</td>
<td>mils</td>
</tr>
<tr>
<td>b</td>
<td>0.010</td>
<td>inches</td>
</tr>
<tr>
<td>(\Theta)</td>
<td>70</td>
<td>degrees</td>
</tr>
<tr>
<td>(f_1)</td>
<td>0.030</td>
<td>inches</td>
</tr>
<tr>
<td>(f_2)</td>
<td>0.020</td>
<td>inches</td>
</tr>
<tr>
<td>r</td>
<td>8.0</td>
<td>inches</td>
</tr>
</tbody>
</table>

For the 3D numerical analysis, the honeycomb stator land will be explicitly model. Figure 6 shows a schematic of the honeycomb cells along with pertinent dimensions. The cell size is measured from the sides of the hexagon and the foil size represents the thickness of honeycomb walls.

Figure 6: Honeycomb schematic

Figure 7 shows the 3D geometry with 1/8" honeycomb, 5 mil foil size, and a seal clearance of 5 mils. As pictured, 2 honeycomb cells are included in the circumferential direction and 6 are included in the axial direction. The geometry is a revolved through a small angle so the cut faces are rotationally periodic. Figure 8 shows the geometry for
the 1/32” honeycomb land which has 2 cells circumferentially, 32 axially and has a foil size of 2 mils.

Figure 7 3D Geometry with 1/8” honeycomb land

Figure 8 3D geometry with 1/32” honeycomb land
2.2 Numerical Model

Figure 9 and Table 3 describe the initial boundary conditions for the numerical model. The boundary conditions described represent a large pressure ratio across the seal and high rotational speed. Additionally the clearance, $c_r$, is very small (5 mils) necessitating a fine mesh in the clearance gap. Note the clearance gap will be varied between 10-15 mils for subsequent studies.

![Figure 9 CFD boundary condition description](image)

<table>
<thead>
<tr>
<th>Boundary Condition</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$P_{up}$</td>
<td>500</td>
<td>psia</td>
</tr>
<tr>
<td>$T_{up}$</td>
<td>1000</td>
<td>F</td>
</tr>
<tr>
<td>$RPMF_{up}$</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>Turbulent Intensity Up</td>
<td>5</td>
<td>%</td>
</tr>
<tr>
<td>Length Scale Up</td>
<td>0.01</td>
<td>inches</td>
</tr>
<tr>
<td>$P_{down}$</td>
<td>250</td>
<td>psia</td>
</tr>
<tr>
<td>Turbulent Intensity Down</td>
<td>5</td>
<td>%</td>
</tr>
<tr>
<td>Length Scale Down</td>
<td>0.01</td>
<td>inches</td>
</tr>
<tr>
<td>$\omega$</td>
<td>18000</td>
<td>rpm</td>
</tr>
<tr>
<td>$c_r$</td>
<td>5</td>
<td>mils</td>
</tr>
</tbody>
</table>

Reviewing the boundary condition set in more detail, several key observations can be made. First, according to Equation 1, the seal is operating at a 2 to 1 pressure ratio.
This is typically the highest pressure ratio for which this seal type is used, and the highest for which there is significant research.

Next, using Equations 2-5, the relative Mach number can be calculated. For internal air systems in aircraft engines, supersonic and transonic flows are avoided. If we assume negligible swirl velocity in the flow (RPMF=0), the Mach number relative to the rotating structure is 0.78, approaching the transonic regime.

\[
PR = \frac{P_{up}}{P_{down}} \tag{1}
\]

\[
U = r \omega \tag{2}
\]

\[
RPMF = \frac{V_{tan}}{U} \tag{3}
\]

\[
V_{tan,rel} = V_{tan} - U \tag{4}
\]

\[
M_{rel} = \frac{V_{rel}}{\gamma R T_s} \tag{5}
\]

Finally, take note that the radial clearance for this configuration, \( c_r \), is 5 mils. 5 mils is the tightest clearance that is practically achievable in current gas turbine design for large aircraft engines. Because of the large pressure ratio, small radial clearance, and high rotating speeds, this is a numerically challenging configuration.

The CFD mesh was generated using ANSYS Workbench version 14.0, and the CFD solutions were generated using ANSYS Fluent 14.0. Table 4 provides details regarding the Fluent solver settings and modeling options employed.
CFD convergence is monitored internal to the Fluent solver by the residuals of various conservation equations, however it is known that residuals do not always provide accurate evidence of convergence. CFD analysts generally prefer to monitor physical quantities of interest in addition to residuals for determination solution convergence.

Because the focus of this paper is on seal leakage and thermal characterization, flow rates and temperatures will be monitored to determine CFD convergence. Figure 10 shows the planes over which physical quantities will be averaged. Monitoring will include: the flow rate across plane YY, and the total temperatures at planes XX and ZZ. Total temperature is calculated according to Equation 6 and will be reported on a mass weighted average basis.

\[ T_t = T_s + \frac{V_{rel}^2}{2C_p} \]  

(6)
As iterations are performed, monitor quantities will be tracked for convergence. Figures 11-13 show the convergence history of the aforementioned monitors for a 2D CFD solution. The figures show the solution is well converged after 7000 iterations. It is worth noting that the mass flow monitor converges after several hundred iterations, while the total temperature monitors require significantly more iterations to converge.
2.3 Meshing

The quality of any finite element solution is dependent on the quality and length scale of the constituent elements. CFD solutions are particularly prone to numerical
discretization error due to the higher order equations being considered in the finite volume formulations.

While these formulations are beyond the scope of this paper, it is important that grid independence be considered before critical review of CFD results. Grid independence is performed by running several solutions with grids of increasing quality. The solutions are compared, and grid independence is achieved when the solution does not change appreciably as grid quality is changed. Mass flow rate and total temperature will be queried at the locations described in Figure 10, and compared for each grid density.

Figure 14 and Table 5 provide details regarding four 2D meshes of varying quality. Note that “clearance gap” refers to the 5 mil radial clearance.

![Figure 14 CFD grid sizing regions](image)

<table>
<thead>
<tr>
<th>Grid Density</th>
<th># Cells</th>
<th>Cell Size Region A (mils)</th>
<th>Cell Size Region B (mils)</th>
<th># Cells in clearance gap</th>
</tr>
</thead>
<tbody>
<tr>
<td>Very Coarse</td>
<td>6,500</td>
<td>10</td>
<td>20</td>
<td>6</td>
</tr>
<tr>
<td>Coarse</td>
<td>2,300</td>
<td>5</td>
<td>10</td>
<td>6</td>
</tr>
<tr>
<td>Medium</td>
<td>91,000</td>
<td>2.5</td>
<td>5</td>
<td>7</td>
</tr>
<tr>
<td>Fine</td>
<td>260,000</td>
<td>1.25</td>
<td>2.5</td>
<td>7</td>
</tr>
</tbody>
</table>

Figure 15 shows total temperature contour plots for the various mesh densities. All the solutions show similar temperature distribution and trends. Table 6 summarizes the
computed flows and total temperatures. Between the “Medium” and “Fine” grid densities there is less than 2% change in mass flow rate, and less than 1% change in mass weighted average total temperature at planes XX and ZZ. Because “Medium” and “Fine” densities produce similar results, and computational resources are limited, the “Medium” grid density is chosen for subsequent studies.

Figure 15 Total temperature contour plot for various grid densities

<table>
<thead>
<tr>
<th>Grid Density</th>
<th># Cells</th>
<th>Flow Rate YY (lbm/s)</th>
<th>Total Temperature at XX (F)</th>
<th>Total Temperature at ZZ (F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Very Coarse</td>
<td>6,500</td>
<td>0.898</td>
<td>1116.8</td>
<td>1249.0</td>
</tr>
<tr>
<td>Coarse</td>
<td>2,300</td>
<td>0.954</td>
<td>1111.9</td>
<td>1230.4</td>
</tr>
<tr>
<td>Medium</td>
<td>91,000</td>
<td>0.917</td>
<td>1106.0</td>
<td>1217.9</td>
</tr>
<tr>
<td>Fine</td>
<td>260,000</td>
<td>0.902</td>
<td>1103.0</td>
<td>1210.0</td>
</tr>
</tbody>
</table>

Figure 16 shows the mesh for the chosen “Medium” grid density. Note the refinements in the seal section and in the clearance gap.
The 3D geometry was meshed using the same procedure as the 2D geometry, using ANSYS workbench 14.0. A fine mesh was applied in the seal section of the domain, and coarser mesh in the upstream and downstream regions. Figure 17 shows several views of the CFD mesh which considers 1/8” honeycomb and a 5 mil seal clearance. Figure 18 shows the mesh for 1/32” honeycomb and a 5 mil clearance. Note the bottom image on both figures shows the honeycomb cells and their proximity to the seal teeth. Table 7 provides details regarding the size of the 3D meshes.

<table>
<thead>
<tr>
<th>3D Model</th>
<th># Cells</th>
<th># Nodes</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/8” Honeycomb</td>
<td>5.3 million</td>
<td>1.1 million</td>
</tr>
<tr>
<td>1/32” Honeycomb</td>
<td>8.4 million</td>
<td>1.8 million</td>
</tr>
</tbody>
</table>

Table 7 3D mesh size summary
**Figure 17** 3D mesh with 1/8" honeycomb

**Figure 18** 3D mesh with 1/32" honeycomb
3. Results and Discussion

3.1 2D Analysis: Leakage and Viscous Heating

It has been established that adequate convergence monitors are in place, and a quality grid is being used. Now, meaningful studies can be conducted with confidence in the results. The study presented in this section will investigate how various clearances affect the leakage and heat generation characteristics of the lab seal. Three clearance values will be considered: 5 mils, 10 mils, and 15mils.

First, a general discussion of the results will be presented where observations be will enable improved interpretation of subsequent material. Next, the discussion of results will detail the behavior of leakage rates and viscous heat generation for the various clearances conditions.

Overview of Results

This section will provide an overview of the results for the 5 mil clearance case, and provide a foundation for more detailed results post-processing.

The pressure, stream function and total temperature are three quantities of interest for rotor/stator systems. Figure 19 shows the contour plots of these quantities for the 5 mil clearance case. The pressure contours provide confirmation in the expected result of large pressure drops across each labyrinth seal tooth. There is a 90 psi drop across the first seal tooth, a 70 psi drop across the second and a 90 psi drop across the third.

The total temperature contour shows a significant total temperature rise across the seal from plane XX to ZZ, about 100F. There is also a significant total temperature increase from the domain inlet to plane XX (upstream of the seal), about 100F. However there is negligible total temperature rise from plane ZZ (downstream of the seal) to the domain outlet. The cause of this behavior is due to velocity changes which will be discussed in a following section.

The middle plot in Figure 19 shows contours of stream function. Stream function is a measure of recirculation for two dimensional flows is defined by Equation 7, where \( \psi \) is the stream function.
\[ \mathbf{v} = \nabla \times \psi \]  

\[ v_r = -\frac{1}{r} \frac{\delta \psi}{\delta z} \]

\[ V_z = \frac{1}{r} \frac{\delta \psi}{\delta r} \]

The stream function plot show several areas of recirculating flow. Large recirculation zones exist upstream and downstream of the seal section and smaller recirculation zones exist within the seal between adjacent seal teeth. Recirculation vortices are expected in rotor stator systems, and the observed recirculation patterns near the seal teeth are consistent with research [4].

**Figure 19** Total temperature, stream function and pressure contours; 5 mil clearance
Tangential velocity, or swirl velocity plays a significant role in the physics of rotor stator systems. Figure 20 shows contour plots of swirl velocity, RPMF (Equation 3), and percent swirl velocity.

The flow enters the domain with no tangential velocity (RPMF=0). As it moves toward the seal section the swirl velocity increases because the rotor is doing work on the flow through viscous forces. At the plane XX, upstream of the seal section, the swirl velocity is about 400 ft/s. Given the linear velocity of the rotor as 1250 ft/s, the non-dimensional swirl velocity (RPMF) here is about 0.3.

As the flow traverses the seal section, the swirl velocity again increases. At the downstream plane the swirl velocity is about 600 ft/s, and the RPMF is about 0.5.

The third contour plot in Figure 20 shows that the swirl velocity is the dominate velocity component in a majority of the domain. The exception is at the radial clearance gap. This is expected, because the axial flow contributes significantly to the resultant velocity when the flow is accelerated through the small gap.

![Figure 20 Contours of tangential velocity quantities; 5 mil clearance](image)
Now that the velocities have been discussed, a better discussion of the total temperature contours in Figure 19 can be provided. The equation for total temperature is provided as Equation 6. Note that the second term includes relative velocity. At the domain inlet, the swirl velocity is zero, so the relative velocity is low. As the flow approaches the seal, the swirl velocity increases, thus relative velocity, and total temperature increase. As the flow passes through the seal section, swirl velocity and total temperature continue to increase. At the exit of the seal section, the swirl velocity is a 600 ft/s, non-dimensionally 0.5. From the exit of the seal section to the exit of the domain, swirl velocity cannot significantly increase. This is because the swirl velocity of the flow is the average of the rotor and stator linear velocities. More concisely, in this region the RPMF is 0.5 and the rotor/stator viscous forces are balanced.

**Viscous Heating and Seal Leakage Results and Discussion**

This section discusses the leakage flow rates, and the viscous heat generation results for the three clearances values: 5, 10, and 15 mils.

The leakage flow rate is easily output by Fluent. The leakage flow rate is quoted as the mass flow across section YY in Figure 10. The calculation of windage heating is performed two ways. The first method employs Equation 9. Equation 9 describes that the amount of heat or work transfer to a control volume is a function of the inlet and outlet total temperatures, the mass flow rate, and the constant pressure specific heat. Figure 21 shows the control volume being employed.

The quantities required for Equation 9 are readily available from the Fluent solution. The mass weighted average total temperatures are queried at the upstream and downstream seal planes, XX and ZZ respectively. The mass flow rate is queried across plane YY, and the specific heat is assumed constant at the average static temperature in the seal section.

\[
Q = Q_{visc} = W = mC_p\Delta T_l
\]  

(9)
There is no mechanical work crossing control volume boundary, thus the total temperature rise in the systems must be due purely to viscous work. To confirm this, the CFD solution was run with the viscous heating option in Fluent turned off. Figure 22 shows the total temperature contour with the viscous heating option turn off. As expected there is no total temperature rise in the CFD domain.

The second method of calculating viscous work involves integrating the shear stress at the rotor and stator wall. Equation 10 is the full differential form of the energy conservation equation, and equation 11 is the reduced form for the problem at hand.

\[
\frac{\delta}{\delta t} \frac{1}{2} \rho v^2 + \rho U = - \nabla \cdot \frac{1}{2} \rho v^2 + \rho U \ v - \nabla \cdot \mathbf{q} - \nabla \cdot \mathbf{p} \mathbf{v} - \nabla \cdot \tau \cdot \mathbf{v} + \rho \ \mathbf{v} \cdot \mathbf{g} \tag{10}
\]

\[
\nabla \cdot \frac{1}{2} \rho v^2 + \rho U \ v = - \nabla \cdot \tau \cdot \mathbf{v} \tag{11}
\]
Integrating Equation 11 yields an equation that relates the change in total temperature to the total shear stress and velocity.

\[ \Delta T_t = \frac{\tau \cdot v}{m C_p} \]  

(12)

It was previously shown in Figure 20 that the tangential velocity is the dominate velocity component, so it is appropriate to replace the vectors in Equation 12 with the scalars \( \tau_{x\phi} \), and \( V_{tan} \). The shear stress in the axial-circumferential direction (\( \tau_{x\phi} \)) is easily queried from Fluent for both the rotor and stator walls, and the swirl velocity (\( V_{tan} \)) can be approximated as the average tangential velocity between upstream and downstream planes.

The results for the 5 mil clearance case are provided in Table 8. The leakage flow through the seal is 0.9173 lbm/s. The observed increase in total temperature through the seal region is 111°F. The control volume approach for calculating heat generation yields 27.27 Btu/s, and the shear stress method yields 25.63 Btu/s. These numbers are in good agreement. Note, using method 2 would result in a 105°F seal adiabatic temperature increase.

**Table 8** Results summary for 5mil clearance case

<table>
<thead>
<tr>
<th>Result</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leakage</td>
<td>0.9173</td>
<td>lbm/s</td>
</tr>
<tr>
<td>( T_{abs,xx} )</td>
<td>1107</td>
<td>F</td>
</tr>
<tr>
<td>( T_{abs,zz} )</td>
<td>1218</td>
<td>F</td>
</tr>
<tr>
<td>( \Delta T_{t,method_1} )</td>
<td>111</td>
<td>F</td>
</tr>
<tr>
<td>( W_{method_1} )</td>
<td>27.27</td>
<td>BTU/s</td>
</tr>
<tr>
<td>( W_{method_2} )</td>
<td>25.63</td>
<td>BTU/s</td>
</tr>
<tr>
<td>( \Delta T_{t,method_2} )</td>
<td>105</td>
<td>F</td>
</tr>
</tbody>
</table>
Two subsequent CFD analyses were run for 10 and 15 mil radial clearances. The same methodology described for the 5 mil case was used. Tables 9 and 10 provide result summaries for the 10 and 15 mil clearances cases respectively. Table 11 summarizes the quantities of interest for the three clearance levels.

### Table 9 Results summary for 10mil clearance case

<table>
<thead>
<tr>
<th>Result</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leakage</td>
<td>2.07</td>
<td>lbm/s</td>
</tr>
<tr>
<td>$T_{cabs,xx}$</td>
<td>1066</td>
<td>F</td>
</tr>
<tr>
<td>$T_{cabs,zz}$</td>
<td>1138</td>
<td>F</td>
</tr>
<tr>
<td>$\Delta T_{t,method 1}$</td>
<td>71</td>
<td>F</td>
</tr>
<tr>
<td>$\dot{W}_{method 1}$</td>
<td>39.45</td>
<td>BTU/s</td>
</tr>
<tr>
<td>$\dot{W}_{method 2}$</td>
<td>32.75</td>
<td>BTU/s</td>
</tr>
<tr>
<td>$\Delta T_{t,method 2}$</td>
<td>60</td>
<td>F</td>
</tr>
</tbody>
</table>

### Table 10 Results summary for 15 mil clearance case

<table>
<thead>
<tr>
<th>Result</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Leakage</td>
<td>3.40</td>
<td>lbm/s</td>
</tr>
<tr>
<td>$T_{cabs,xx}$</td>
<td>1044</td>
<td>F</td>
</tr>
<tr>
<td>$T_{cabs,zz}$</td>
<td>1100</td>
<td>F</td>
</tr>
<tr>
<td>$\Delta T_{t,method 1}$</td>
<td>56</td>
<td>F</td>
</tr>
<tr>
<td>$\dot{W}_{method 1}$</td>
<td>50.98</td>
<td>BTU/s</td>
</tr>
<tr>
<td>$\dot{W}_{method 2}$</td>
<td>38.76</td>
<td>BTU/s</td>
</tr>
<tr>
<td>$\Delta T_{t,method 2}$</td>
<td>42.7</td>
<td>F</td>
</tr>
</tbody>
</table>
Table 11 Leakage and windage summary for clearances (5-15mil)

<table>
<thead>
<tr>
<th>Clearance (mil)</th>
<th>Leakage (lbm/s)</th>
<th>Windage Heating Rate (Btu/s)</th>
<th>Adiabatic Total Temperature Rise (°F)</th>
<th>Mass Weighted Average Upstream RPMF</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.92</td>
<td>25.6</td>
<td>111</td>
<td>0.322</td>
</tr>
<tr>
<td>10</td>
<td>2.07</td>
<td>32.7</td>
<td>71</td>
<td>0.255</td>
</tr>
<tr>
<td>15</td>
<td>3.4</td>
<td>38.8</td>
<td>56</td>
<td>0.173</td>
</tr>
</tbody>
</table>

Figure 22 shows a plot of leakage rate versus clearance. As expected the leakage flow rate increases fairly linearly with clearance. The larger gap allows increased flow at a given pressure ratio.

![Figure 23](image.png)

**Figure 23** Leakage versus clearance for 2:1 seal pressure ratio

Figure 24 shows that heat generation due to windage increases with increasing seal clearance. This is an expected result as similar results were reported by Yan et. Al [4]. However, the physics behind this trend requires further explanation.
The increased viscous heating is caused by lower swirl velocities (RPMFs) upstream of the labyrinth seal section [5]. The flow enters the domain with zero swirl velocity and the swirl velocity increases as it approaches the seal section. It is expected that as the flow rate increases, the swirl velocity (or RPMF) upstream of the seal section will drop because larger flows will preserve their angular momentum better than smaller flows. Figure 25 shows contours of RPMF for the various seal clearances. The RPMFs upstream of the section drop as seal clearance is increased.

Figure 25 also shows the downstream RPMF is roughly the same for each clearance case, about 0.5. So for a larger clearances, more work/heat needs to be added to achieve the same downstream condition.
Figure 26 shows that the adiabatic temperature rise across the seal decreases with increasing clearance. This is an expected result because the increase in windage heat is more than offset by the increase in heat capacitance due to increased flow rates. Comparing the 5 mil clearance case to the 10 mil clearance case, the flow rate increases roughly 100%, but the windage heat increases only 30%. According to Equation 9, this requires the temperature difference to decrease.
Viscous Heating Results and Discussion – Constant Leakage Flow

As explained in the previous section, the varying flow rate has a significant effect on seal windage characteristics, primarily due to the changing swirl velocity upstream of the seal section. These results are realistic since typically the pressure ratio across the seal is generally fixed. This section, however, considers a constant flow rate and varying clearances.

Since the flow rate is constant it is expected that swirl velocities upstream of the seal will be constant. The windage is expected to decrease with larger clearances due to lower axial velocity gradients in the vicinity of the seal teeth. It is also expected that the temperature rise across the seal will decrease with larger clearances because the heat capacitance of the flow is constant and the heat load is lower.

Table 12 summarizes the windage and temperature rise results for varying clearances at constant flow. For each clearance condition, a leakage rate of ~0.9 lbm/s is imposed. As expected, the windage and temperature rise decreases with increasing clearances. Additionally, the upstream swirl velocities, represented by RPMF, remain constant. Figure 27 shows the RPMF contours for the various clearance conditions verifying that the upstream RPMFs are indeed constant.

Figure 26  Total temperature rise versus clearance for 2:1 seal pressure ratio
Table 12 Leakage and windage summary for clearances (5-15mil); Constant flow

<table>
<thead>
<tr>
<th>Clearance (mil)</th>
<th>Leakage (Ibm/s)</th>
<th>Windage Heating Rate (Btu/s)</th>
<th>Adiabatic Total Temperature Rise (F)</th>
<th>Mass Weighted Average Upstream RPMF</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>0.91</td>
<td>27.27</td>
<td>111</td>
<td>0.32</td>
</tr>
<tr>
<td>10</td>
<td>0.9</td>
<td>21.46</td>
<td>89</td>
<td>0.33</td>
</tr>
<tr>
<td>15</td>
<td>0.9</td>
<td>19.33</td>
<td>81</td>
<td>0.31</td>
</tr>
</tbody>
</table>

Figure 27 RPMF versus seal clearance comparison; Constant flow

Figure 28 is a plot of windage versus seal clearance which shows that for a constant flow rate, the windage heating decreases non-linearly with increasing seal clearance. Figure 29 shows a corresponding trend in adiabatic temperature rise.
3.2 2D Analysis: Seal Convection Heat Transfer Characterization

Up to this point heat transfer has been neglected, that is the walls in the CFD domain were assumed to be adiabatic. In this section, wall temperatures will be assigned.
in the seal region, and heat transfer coefficients will be calculated. First, heat transfer coefficients will be determined using results that include viscous heating, then viscous heating will be turned off and coefficients will be recalculated. Comparisons between these two approaches are presented. For this study only the 5 mil clearance is considered.

Both cooling and heating conditions are considered. Wall temperatures of 500F (fluid heats wall), and 1500F (fluid cools wall) are applied independently to the rotor and stator walls. Figure 30 illustrates the regions on the rotor and stator where temperature boundary conditions are applied. Table 13 describes the four cases being considered.

![Figure 30 Application of wall temperature boundary conditions](image)

**Table 13** Summary of heat transfer studies

<table>
<thead>
<tr>
<th>Case #</th>
<th>Inlet Total Temperature (F)</th>
<th>Rotor Temperature (F)</th>
<th>Stator Temperature (F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1000</td>
<td>500</td>
<td>Adiabatic</td>
</tr>
<tr>
<td>2</td>
<td>1000</td>
<td>Adiabatic</td>
<td>500</td>
</tr>
<tr>
<td>3</td>
<td>1000</td>
<td>1500</td>
<td>Adiabatic</td>
</tr>
<tr>
<td>4</td>
<td>1000</td>
<td>Adiabatic</td>
<td>1500</td>
</tr>
</tbody>
</table>

In order to calculate heat transfer coefficients the following approach will be used. Given the same control volume considered in Figure 21 the energy balance yields Equation 13, where \( \Delta T_t \) is the difference between the upstream and downstream total temperatures.
For this study it is assumed that $Q_{\text{visc}}$ is the same as calculated in the section 3.1. There is some error in this assumption since it is known that temperature affects viscosity, and thus viscous heating. However as noted, the heat transfer coefficient calculations will be performed in subsequent sections without the inclusion of viscous heating terms to assess this error.

Heat transfer coefficients are calculated using Equations 14 and 15. When rotor heat transfer coefficients are considered, the fluid temperature in Equation 15 is the total temperature relative to the rotor. When stator heat transfer coefficients are considered the total temperature is relative to the stator (Equation 6). Combining Equations 13-15 yields an expression for the heat transfer coefficient, Equation 16.

$$Q_{\text{convec}} = HA(T_s - T_f)$$  

(14)

$$T_f = \frac{T_{t,\text{up}} + T_{t,\text{down}}}{2}$$  

(15)

$$H = \frac{mC_p}{A T_{\text{surf}}} \left( \frac{\Delta T_{t,\text{down}} - \Delta T_{t,\text{up}} - Q_{\text{visc}}}{T_{t,\text{down}} - T_{t,\text{up}}} \right)$$  

(16)

Figure 31 shows total temperature contours for the heated conditions, cases 1 and 2 from Table 12. The fluid temperature contours are consistent with the applied wall temperatures. Note that wall temperatures applied to the stator affect the downstream fluid temperature to a greater extent than wall temperatures applied to the rotor. That is, the downstream temperature of the fluid for $T_{\text{stator}}=500\,\text{F}$ is closer to 500F than for $T_{\text{rotor}}=500$. This observation leads one to expect that the heat transfer coefficients on the
stator should be higher than on rotor. Figure 32 shows temperature contours for the cooled conditions, and shows the same downstream temperature trend.

Although the stator has more of an effect on the downstream temperature, the rotor temperature is responsible for setting the temperature environment in the pockets between the seal teeth.

![Temperature Contours](image)

**Figure 31** Total temperature contours for heated wall condition

Table 14 shows the calculated average heat transfer coefficients for each of the respective runs. As expected the stator heat transfer coefficient are significantly higher than the rotor. Table 14 also shows the heat transfer coefficients for a heated wall are very similar to those for a cooled wall. The values for the rotor are about 6% different, and the values for the stator are less than 1% different.
It was noted that the viscous heat rate may be slightly different with applied wall temperatures because the fluid temperature is being affected by the convection to the walls. In order to assess this error and gain confidence in the heat transfer coefficients, cases 1 and 2 were re-run without the viscous heat term, $Q_{\text{visc}} = 0$. These cases are presented as 1a and 2a in Table 15.

Figure 33 shows the total temperature contours for 1a and 2a. As expected, the trends from the previous section still hold true. The stator heat transfer is higher, resulting in a lower downstream temperature condition compared to the rotor. Also the
rotor temperature boundary condition continues to be responsible for setting the temperature environment inside the seal pockets.

![Figure 33 Total temperature contours for heated wall condition; No viscous heating](image)

The heat transfer coefficients are presented in Table 15. As expected the stator heat transfer coefficients are significantly higher than the rotor. In comparing heat transfer coefficients between result with and without viscous heating, both rotor and stator heat transfer coefficient are lower by about 10% when viscous heating is turned off.

<table>
<thead>
<tr>
<th>Case #</th>
<th>Rotor Temperature (F)</th>
<th>Stator Temperature (F)</th>
<th>Average Rotor HTC (BTU/hr-ft²-F)</th>
<th>Average Stator HTC (BTU/hr-ft²-F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1a</td>
<td>500 Adiabatic</td>
<td></td>
<td>594</td>
<td>---</td>
</tr>
<tr>
<td>2a</td>
<td>Adiabatic 500</td>
<td></td>
<td>---</td>
<td>1097</td>
</tr>
</tbody>
</table>

The average heat transfer coefficients on the stator are significant higher than on the rotor because a significant portion of the stator land sees a high relative axial velocity. The rotor, on the other hand, only sees this relative velocity contribution at the
seal tips. The seal tips have minimal surface area, and thus do not significantly contribute to the average heat transfer coefficient.

The average rotor heat transfer coefficients are primarily determined by the relative swirl velocity rather than axial velocities. Figure 34 shows the swirl velocities in the seal pockets are significantly lower than the axial velocities at the tips. This helps explains the lower heat transfer coefficients on the rotor.

![Figure 34 Absolute value of axial to swirl velocity ratio](image)

3.3 3D Analysis: Leakage and Viscous Heating

This section describes the 3D CFD results for two honeycomb land geometries: a 1/8” honeycomb land, and a 1/32” honeycomb land. A 5 mil clearance is used for this study. Note that all contour plots in the section are taken at the mid plane.

Figure 35 shows the pressure contours for the 1/8” honeycomb land. The contours are very similar to the 2D analysis with 90, 70 and 90 psi pressure drops across first, second and third knife edges respectively.
Figure 35 3D CFD pressure contours for 1/8” honeycomb

Figure 36 shows contours of total temperature. Here there is significant difference between the similar 2D analysis (Figure 19). Although the radial clearance between the knife edge and the seal land is 5 mils for both analyses, the large honeycomb effectively increases metering flow area, allowing for significantly more leakage. This increase in seal leakage then mitigates the total temperature increase according to Equation 9.

The flow area increase can be seen in the velocity contours on Figure 37. At the mid plane, the first seal tooth is positioned directly under a solid portion of the seal land. This is similar to the treatment in the 2D analysis, however, for the second and third seal teeth, the seal tip is offset from solid portion of the honeycomb. This offset creates a gap larger than the 5 mil seal clearance, so the flow gets directed into the honeycomb cells. The velocity contour in Figure 37 shows the flow being directed into and out of these gaps.
Figures 36 and 37 show the contours for swirl velocity and RPMF. These contours also differ significantly relative to the 2D analysis. The mass weighted RPMF
at the upstream plane, XX is 0.16 compared to 0.32 for the 2D analysis. This is understandable given the increased leakage with the honeycomb configuration.

Due to the different leakage flow levels it is difficult to accurately compare the 2D and 3D analyses with respect to windage. However other interesting comparisons can be made. The downstream mass weighted RPMF at plane YY is nearly 0. Recall that in the 2D results the RPMF was ~0.5. The RPMF is low in the 3D results because a significant portion of the flow is directed into the honeycomb cells. This mitigates angular momentum increases imposed by the rotor.

![Swirl Velocity Contours](image1)

**Figure 38** 3D CFD swirl velocity contours for 1/8" honeycomb

![RPMF Contours](image2)

**Figure 39** 3D CFD RPMF contours for 1/8" honeycomb
Table 16 provides a comparison between the 2D and 3D 5 mil clearance cases. Interestingly, the 1/32” honeycomb analysis produces a leakage level less the 2D solid land results. One explanation for the leakage reduction is that the 1/32” honeycomb cell size (0.03125”) is similar to the axial width of the tip of the knife edge (0.010”). This limits the flow’s ability to find a less restrictive path than the clearance gap itself.

Choi et al reported similar findings with small honeycomb cell sizes and attributed the leakage decrease to turbulent friction [6]. Note contour plots for the 1/32” honeycomb analysis are provided in the appendix.

<table>
<thead>
<tr>
<th>Clearance (mil)</th>
<th>Honeycomb Size (in)</th>
<th>Leakage (lbm/s)</th>
<th>Windage Heating Rate (Btu/s)</th>
<th>Adiabatic Total Temperature Rise (F)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>None; Solid</td>
<td>0.92</td>
<td>25.6</td>
<td>111</td>
</tr>
<tr>
<td>10</td>
<td>None; Solid</td>
<td>2.07</td>
<td>32.7</td>
<td>71</td>
</tr>
<tr>
<td>15</td>
<td>None; Solid</td>
<td>3.4</td>
<td>38.8</td>
<td>56</td>
</tr>
<tr>
<td>5</td>
<td>1/8”</td>
<td>2.25</td>
<td>28.1</td>
<td>47</td>
</tr>
<tr>
<td>5</td>
<td>1/32”</td>
<td>0.54</td>
<td>21.0</td>
<td>144</td>
</tr>
</tbody>
</table>
4. Conclusions

This paper explored several aspects of gas turbine labyrinth seal behavior. A 2D axisymmetric model was used to investigate the leakage rates and windage heating for various seal clearances with a solid stator land. The 2D model was also used to study the heat transfer characteristics on the rotor and stator for heating and cooling situations. A periodic 3D model was employed to study the effect of honeycomb cell size on leakage flow and windage heating.

The results for the 2D analysis were as expected. For a given pressure ratio, an increase in seal clearance resulted in an increase in leakage. The relationship between seal clearance and leakage level was linear. The windage heat also increased with seal clearance due to lower upstream RPMFs in cases with larger seal clearances. The adiabatic total temperature rise across the seal dropped as seal clearance was increased due to additional heat capacitance provided by more leakage flow.

When the flow rate across the seal was held constant the windage heat decreased non-linearly with increasing seal clearance due to lower axial velocity gradients in between the rotor and seal land. Correspondingly, temperature rise across the seal decreased with increasing seal clearance.

Heat transfer results for the 2D analysis were also as expected. For a solid seal land, heat transfer coefficients were higher than the rotor heat transfer coefficients due to the significant axial velocity component. Heating versus cooling did not have a significant effect on heat transfer coefficients for neither the rotor nor the stator. Removing viscous heating from the solution equations also did not appreciably change heat transfer coefficients.

Honeycomb geometry was studied using a rotationally periodic 3D CFD sector model. Two honeycomb geometries were studied: 1/8” honeycomb, and 1/32” honeycomb. The results were as expected. The larger 1/8” honeycomb showed a significant increase in leakage flow because of the larger effective flow gap. The 1/32” honeycomb showed a slight decrease in leakage flow relative to solid land 2D results.
5. References


6. Appendix

![Figure 40](image1) 3D CFD pressure contours for 1/32" honeycomb

![Figure 41](image2) 3D CFD total temperature contours for 1/32" honeycomb
Figure 42 3D CFD velocity magnitude contours for 1/32” honeycomb

Figure 43 3D CFD swirl velocity contours for 1/32” honeycomb
Figure 44 3D CFD RPMF contours for 1/32" honeycomb