AN EXPERIMENTAL INVESTIGATION INTO THE OPTIMIZATION
OF PADDED FINGER SEALS

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AN EXPERIMENTAL INVESTIGATION INTO THE OPTIMIZATION
OF PADDED FINGER SEALS

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Thesis

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ABSTRACT

The non-contacting nature of finger seal technology provides a compliant seal which allows both axial and radial adjustments to rotor excursions, without loss of physical integrity or sealing capability. The same cannot be said of labyrinth and brush seals. This fact, along with their relatively low cost, could make finger seals a superior choice in many situations. The performance of twelve different finger seal assemblies has been evaluated. The assemblies are created by combinations of four different sets of finger seal laminates and three front plate designs. Three laminate sets feature a high pressure padless layer, and a padded low pressure layer which generates lift and gives the seal its compliant nature. The fourth laminate set also features an additional padded layer which interlocks with the original low pressure layer, and eliminates leakage between adjacent lifting pads. The original front plate serves only to clamp the laminates to the test rig, while the other two are designed to help restrict flow through the assembly, and force more air under the pads, improving lifting capability. Seal performance is evaluated through a set of parametric studies in which the pressure differential across the seal is brought to 20 psid, and the rotor speed is brought from rest to 15 krpm in increments of 5 krpm. A series of static tests were conducted to ascertain the performance of seal combinations at pressures greater than 20 psi. The study shows that all of the finger seal combinations lift effectively, and experience only minor wear during startup. The double pad design outperforms the original three seals, by providing lower leakage rates and
operating temperatures. The modified front plate designs have shown to reduce leakage and improve lifting capability, making them an excellent supplement to an assembly of finger seal laminates.
# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>LIST OF FIGURES</th>
<th>viii</th>
</tr>
</thead>
<tbody>
<tr>
<td>I. INTRODUCTION</td>
<td>1</td>
</tr>
<tr>
<td>II. FINGER SEAL LITERATURE REVIEW</td>
<td>8</td>
</tr>
<tr>
<td>III. DESCRIPTION OF EXPERIMENTAL INSTALLATION</td>
<td>13</td>
</tr>
<tr>
<td>3.1 Overview</td>
<td>13</td>
</tr>
<tr>
<td>3.2 Basic Setup</td>
<td>13</td>
</tr>
<tr>
<td>3.3 Drive Train</td>
<td>14</td>
</tr>
<tr>
<td>3.4 Overview of the Safety System</td>
<td>15</td>
</tr>
<tr>
<td>3.4.1 Gearbox Lubrication System</td>
<td>15</td>
</tr>
<tr>
<td>3.4.2 Spindle Lubrication System</td>
<td>16</td>
</tr>
<tr>
<td>3.5 Test Section</td>
<td>16</td>
</tr>
<tr>
<td>3.5.1 Assembly Mounting Plates</td>
<td>17</td>
</tr>
<tr>
<td>3.5.2 Exhaust Plate</td>
<td>17</td>
</tr>
<tr>
<td>3.5.3 Seal Mounting Plate</td>
<td>18</td>
</tr>
<tr>
<td>3.5.4 Lucite Enclosure</td>
<td>18</td>
</tr>
<tr>
<td>3.5.5 Air Inlet Plate</td>
<td>19</td>
</tr>
<tr>
<td>3.5.5 End Plate</td>
<td>19</td>
</tr>
<tr>
<td>3.6 Air Supply System</td>
<td>19</td>
</tr>
<tr>
<td>3.7 Description of the Seal</td>
<td>20</td>
</tr>
</tbody>
</table>
# LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.1</td>
<td>5</td>
</tr>
<tr>
<td>1.2</td>
<td>6</td>
</tr>
<tr>
<td>1.3</td>
<td>7</td>
</tr>
<tr>
<td>3.1</td>
<td>26</td>
</tr>
<tr>
<td>3.2</td>
<td>27</td>
</tr>
<tr>
<td>3.3</td>
<td>27</td>
</tr>
<tr>
<td>3.4</td>
<td>28</td>
</tr>
<tr>
<td>3.5</td>
<td>28</td>
</tr>
<tr>
<td>3.6</td>
<td>29</td>
</tr>
<tr>
<td>3.7</td>
<td>30</td>
</tr>
<tr>
<td>3.8</td>
<td>29</td>
</tr>
<tr>
<td>3.9</td>
<td>31</td>
</tr>
<tr>
<td>3.10</td>
<td>32</td>
</tr>
<tr>
<td>3.11</td>
<td>33</td>
</tr>
<tr>
<td>3.12</td>
<td>34</td>
</tr>
<tr>
<td>3.13</td>
<td>37</td>
</tr>
<tr>
<td>3.14</td>
<td>37</td>
</tr>
<tr>
<td>3.15</td>
<td>38</td>
</tr>
<tr>
<td>3.16</td>
<td>38</td>
</tr>
<tr>
<td>Section</td>
<td>Title</td>
</tr>
<tr>
<td>---------</td>
<td>----------------------------------------------------------------------</td>
</tr>
<tr>
<td>6.5</td>
<td>Seal 2-2 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.6</td>
<td>Seal 2-3 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.7</td>
<td>Seal 3-1 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.8</td>
<td>Seal 3-2 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.9</td>
<td>Seal 3-3 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.10</td>
<td>Seal 4-1 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.11</td>
<td>Seal 4-2 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.12</td>
<td>Seal 4-3 Temperature, Leakage, and Wear</td>
</tr>
<tr>
<td>6.13</td>
<td>Interaction of Aligned and Misaligned Rotor with Finger Pads</td>
</tr>
<tr>
<td>6.14</td>
<td>Flow Factor Comparison for Various Seal/ Plate Combinations at 10 psi</td>
</tr>
<tr>
<td>6.15</td>
<td>Typical Flow vs Speed Plot</td>
</tr>
<tr>
<td>6.16</td>
<td>Flow Response to Changes in Rotor Speed</td>
</tr>
<tr>
<td>6.17</td>
<td>Flow Drops from Throttle Adjustments</td>
</tr>
<tr>
<td>6.18</td>
<td>Apparent Flow Loss During Testing</td>
</tr>
<tr>
<td>6.19</td>
<td>Actual Flow Loss During Testing</td>
</tr>
<tr>
<td>6.20</td>
<td>Static Pressure Test for All Seals with 3rd Plate</td>
</tr>
<tr>
<td>6.21</td>
<td>Caulk Added to Fill Gaps Between Layers</td>
</tr>
<tr>
<td>6.22</td>
<td>Back Plate Leakage Path</td>
</tr>
<tr>
<td>6.23</td>
<td>Paper Ring Added to Back Plate</td>
</tr>
<tr>
<td>6.24</td>
<td>Effect of Added Ring on Maximum Pressure</td>
</tr>
</tbody>
</table>
In the modern gas turbine engine, the ability to adequately seal off the path of gas through the system is critical. Without this, proper flow direction and separation would not be possible, and thus cooling and power production efficiency would decrease dramatically. It has been said that the average increase in specific fuel consumption per year of 1% for a large turbofan engine can be attributed directly to the wear and erosion of its seals. (Chupp and Nelson, 1990) This problem is not limited to large engines. The effect of seals on performance of small engines can be as high as a 7% increase in specific fuel consumption. (Ludwig and Bill, 1980) While this may not sound like a serious problem, consider the effect a small increase in fuel consumption on the airline industry. For planes that are running nonstop, this seemingly minute change can amount to millions of dollars in fuel.

For a long time, the only solution for sealing between moving and stationary parts in an engine was by use of rigid seals. Some examples of these are honeycomb, cylindrical, and labyrinth seals. By a wide margin, the most common non-compliant seal is the labyrinth seal. The seal has a number of teeth on the stationary and rotating surface which fit together to form a labyrinth that any escaping air must first pass through. An example of a simple labyrinth seal can be seen in Figure 1.1. Obviously, the smaller the clearances are inside the labyrinth, the higher the resistance to flow will be. That said,
trying to work with small clearances can create problems. In engines, the rotating components have tremendous energy, and transient shaft excursions cause rubbing between teeth can be dangerous and costly, both in terms of hardware replacement and engine efficiency. While trying to balance sealing capability and safe clearance in design, more problems arise. Predictions must be made on the exact motion of the shaft, accounting for motion in the axial and radial directions. Another consideration with shaft and blade seal design are the changes in the geometry of the shaft due to thermal effects. No matter how precisely this analysis is performed, these variables would surely change over the life of the engine, ensuring that the performance of the seal would decline. This fact was realized, and the search began for a seal that would not suffer the same fate.

The introduction of the brush seal occurred in 1980, and while it took nearly a decade to be incorporated in an engine, it seemed to solve many of the problems associated with rigid seals. The basic design concept of the brush seal consisted of a number of rows of wire bristles mounted with preload on the inside of a ring, which is encased between a front and back plate. Thus the circle formed by the tips of the bristles is smaller than the diameter of the shaft that it will be mounted on, so that sealing is ensured. The bristles are mounted at an angle on the ring, so that motion of the shaft does not cause them to buckle. Instead they will flex and adapt easily with any movement of the shaft surface. Schematics of a typical brush seal can be seen in Figure 1.2. (Lattime, 2001) The brush seal provides better sealing capabilities than the labyrinth seal, and has a much longer lifespan. Despite its apparent superiority to the labyrinth seal, the brush seal is not perfect. According to Lattime (2001), the bristles can suffer from increases in stiffness, which will have a negative impact on the performance of the seal. This increase
in stiffness causes the bristles to respond to stimulation more slowly than desired. When
the rotor surface moves towards the bristles, excessive rubbing occurs, and when it
moves away, it takes longer for the newly formed gap to be covered, greatly increasing
leakage. This phenomenon, coupled with the high cost inherent with the production of
these seals is pushing researchers to make advances with a new seal concept.

The development of a new design, aptly named the “finger seal” began in the
early 90s. The concept of the finger seal is similar to that of a brush seal. It is made of a
number of flexible members that are fixed on one end at the inner radius of a ring, and
make contact with the rotor surface on the other end. This is where the similarities end
however. The fingers are produced by making cuts of identical curvature into the inside
of a ring which is relatively thin in the axial direction. This is commonly referred to as a
laminate, and is accompanied by an exact copy of itself, which is then rotated such that
its fingers cover the gaps between fingers of the first laminate. A depiction of an early
finger seal setup can be seen in Figure 1.3. Pioneers of the design Heydrich (1991),
Mackey (1991), and Johnson (1992) showed that when mounted with a preload, the
finger seal showed wear and leakage characteristics that were similar to that of the brush
seal. The difference is that manufacturing of the finger seal costs significantly less than
what was needed to build an equal performance brush seal. The design was later
the end of each finger. These pads experience a hydrodynamic lifting force generated by
air that is pulled in the circumferential direction by the moving surface of the rotor. This
addition creates the possibility for a non contacting seal, which would lengthen the life of
both the seal and the rotor.
The focus of this research will be to gain an understanding of finger seal performance through testing, and to develop design criteria for the next generation of finger seals.
Figure 1.1 - Three Stage Labyrinth Seal (Hendricks, 1987)
Figure 1.2 – Typical Brush Seal from Three Angles
Figure 1.3 – Early Finger Seal (Pierson, 2004)
CHAPTER II
FINGER SEAL LITERATURE REVIEW

The first numerical analysis of finger seals was performed by Hendricks et al (1994). The model presented was similar to that of the brush seal, which had been developed by Braun et al (1990) and Hendricks et al (1991). The porosity of the finger seal was expressed in terms of the finger sticks and the spaces between them. These were modeled as solid and open volumes respectively. The authors calculated the slope of the leakage versus pressure drop curve for two different cases. The model was used first for a zero clearance condition between the rotor and seal, and then for a clearance of .001 inches. Results showed that leakage was highly dependent on clearance, and that performance was similar to that of brush seals and greater than that of labyrinth seals. While this looked promising, this model could not provide hysteresis data.

The first experimentation with finger seals was performed by Arora et al (1999). Thirteen seal configurations were tested to monitor the effect of hysteresis on leakage, and provide a physical understanding that could be used to generate a new design. Testing began with a basic seal package, which consisted of three laminates that were pushed together by two spacers. Leakage of air through the seal was quantified by the difference between flow rates in the air supply and exit lines.

A method for comparing performance of seals with different geometries and operating conditions was developed using the flow factor:
\[ \Phi = \frac{m \sqrt{T_{ave} + 459.6}}{P_u D_{seal}} \]

where

- \( m \) = leakage rate (lbm/s)
- \( T_{ave} \) = average air temperature (°F)
- \( P_u \) = upstream air pressure (psia)
- \( D_{seal} \) = rotor diameter (in)

Leakage due to hysteresis due to changes in rotor speed and laminate position was examined by running the rotor up to 45,000 RPM and back down to zero three times. It was found that the leakage rate was lower during the acceleration phase than the deceleration phase. Furthermore, after a few cycles, the leakage rate remained at a higher value during both phases. It was evident that the fingers were not returning to their original positions after they moved away from the rotor. The authors surmised that as the rotor increased in speed, it expanded radially, pushing the fingers outward.

Unfortunately, the pressure on the seal created a large friction force between the fingers and the low pressure plate, which worked against the spring force in the fingers, preventing them from recovering when the rotor surface receded during deceleration. If the finger seal was going to be successful, the next step was obviously to somehow reduce this frictional force. The authors’ solution to this problem was to create a third plate, which would sit directly behind the low pressure laminate. This plate had a groove in it, which lined up with newly added holes in the laminates. This groove created a pocket behind the fingers, where air that passed through the holes could gather. Now the pressure difference across the fingers was greatly reduced, therefore hopefully reducing...
their friction with the plate. The previous test was repeated with the new back plate, and the results were as expected. Leakage rates were lower, and remained more constant over the course of the test cycles. Two cyclic tests were then run for five hours each. In the first test, the system was held at 500 °F and 30psi, during two minutes at 35,000 RPM, and one minute at 10,000 RPM. The second was held at 800 °F and 60psi, with the same operating speeds. Finally, the authors put the seal through an endurance test with an initial interference of 0.045 inches between the rotor and seal. Air leakage was low and stable for all these tests, compared to the results of the initial design. The authors reported that wear on the fingers and rotor surface was approaching an acceptable “line-to-line” interface contact at the end of testing.

Concerns about finger seal heat generation were first examined in 2002. Proctor et al (2002) tested the seal previously developed by Arora, under the extreme conditions of 1200 °F and 75 psi. The seal was installed on a rotor with a diameter of 8.5 inches, and had an initial clearance of 3 mils. Three tests were designed to evaluate the performance of the seal. The first was a static leakage test. The system was held at 1200° while the pressure was varied from zero to 75 psi. In the second test, the rotor was accelerated to approximately 20,000 RPM and then brought back to rest, while monitoring system temperature and pressure. Lastly, the rotor was run at 20,000 RPM for four hours, with operating conditions of 1200 °F and 75 psi. While the seal exhibited some hysteresis, leakage remained relatively low. The authors believed that movement between the seal holder and the test rotor, may have been what produced the change. Power loss of the seal was comparable to that for a brush seal operating under the same conditions.
In 2002, a new finger seal design concept was introduced. Braun et al began to numerically investigate the addition of lift pads to each finger. A model was developed of two successive fingers, one high pressure padless finger, and one padded low pressure finger. The analysis was split into two extreme cases. The first model assumed that while air could move freely in the axial and radial directions, leakage was allowed only underneath the pads. This was obviously the best case scenario for the seal. The second model was the worst case scenario of unrestricted leakage between all fingers. A 2-D model was used to examine air pressure and flow, as well as pad deformation in the R-Z plane. The air in this model was driven by the pressure difference across the seal. Results showed that a lifting force was created by the air passing under the pads, and the pads rotated about the heel in the axial plane. Flow patterns around the fingers were examined using a 3-D model. The authors used three cases to analyze the effect of upstream flow rate on local velocity fields near the seal. In the high flow case, the velocities remained mostly axial with little to no circumferential component. However, the medium velocity showed significant flow in the circumferential direction, and the lowest flow rate was nearly completely circumferential. This illustrates the authors’ belief that velocity fields around the fingers are heavily reliant on upstream flow rate and geometry.

Braun et al (2004) continued analysis and optimization of the padded finger seal design. A method of seal construction using ten geometric parameters was developed. These parameters could be varied to create seals with characteristics that were needed for specific conditions. A finite element analysis was performed to see the effect of loading on the seal. While the results showed the need for improvement, the main purpose was to get a good approximation of finger stiffness. In order to minimize the amount of finger
twisting due to lift on the pads, the top of the pad was tapered to produce a thicker cross section where it meets the finger, and a fillet was introduced. Numerical thermofluid investigation was conducted to determine lift characteristics for different pad geometries. The first case was a pad with a slanted bottom that created a converging region in the circumferential direction. The second case had a bottom surface that was converging in both the circumferential and axial direction. Using this model, variables such as under pad pressure, under pad force, fluid leakage, and finger lift could be seen as functions of rotor speed and system pressure. Lastly, an experiment was conducted to determine the accuracy of the numerical models and their predictions. A seal using the single pad with a circumferential wedge design for a low pressure laminate was used. A new compact pressure balance ring was designed to minimize friction with the fingers. Data for pressure, pad temperature, and flow leakage was collected, and were close to the values predicted in the numerical analysis. As expected, fluid and pad temperature was more dependent on rotor speed than the pressure differential across the seal. Unlike the numerical model, the experiment revealed that leakage was dominated by the high pressure difference across the seal, and much less by the rotor speed. However, the authors felt that rotor speed may have a bigger role past their limitation of 12,000 RPM. Finally, the test proved that there was very little drop in pressure between the high pressure side of the seal and the pressure equalization chamber, as compared to that across the seal. This meant that the design objective of the new back plate was achieved, and finger friction was reduced, thus allowing the fingers to float with possible shaft excursions.
CHAPTER III

DESCRIPTION OF EXPERIMENTAL INSTALLATION

3.1 Overview

In order to evaluate the performance of different finger seal designs, a custom testing installation had to be developed. The purpose of the test rig was two-fold. First, it would be used to collect performance data on the seal, in order to determine the viability of the design. This was a necessary preliminary step to any possible testing of the seal in a high temperature test setup at NASA Glenn Research Center. Second, the results of the tests would provide evidence to the validity of the numerical tests that were previously conducted, which included several thermofluid and dynamic simulations.

3.2 Basic Setup

The main design requirements for the test rig were to drive the rotor that the seal would operate on, and to provide a high pressure air supply to the test region that would encase the seal. A picture showing the overall setup can be seen in Figure 3.1. Specifically, the test would require rotor speeds upwards of 15,000 revolutions per minute, and up to 120 pounds per square inch of pressure. The complete system that was created will be described in the following sections. In order to best delineate its intricacies, the description of the test installation will be split into two parts; the drive train and its supporting lubrication systems, and the high pressure air delivery circuit.
3.3 Drive Train

We can see in Figure 3.2 the drive train for the experiment, which is made up of an electric motor, a gearbox multiplier, and the high speed spindle that the rotor is mounted to. In order to compensate for any small amounts of misalignment between the gearbox and either the motor or the spindle, elastic couplings were used for the two interconnections in the drive train. Auxiliary power, lubrication circuits, and a blast shield were created to insure safety for both the instrumentation and personnel. Details of all parts of the drive train and its supplementary circuits will be examined further.

The test rotor was connected to the gearbox output of the spindle from Whitnon Manufacturing, seen in Figure 3.3. It was designed to have a maximum shaft run-out less than .0001 inch for speeds up to 30,000 rotations per minute. This precision was paramount to creating a stable testing platform for the seal. The output shaft of the spindle was tapered to provide a connection with the test rotor that would not slip. The shaft was supported on both sides of the spindle by high precision ball bearings that were lubricated by an oil mist. Lubrication by liquid oil or packed grease would cause large frictional heating in this high speed test, and thus, were not desirable.

Built by R.F. Cook of Akron Ohio, the test rotor supplied the moving surface for the seal to interact with. A solid model of the rotor can be seen in Figure 3.4. The inside of the rotor was tapered to match the output shaft of the spindle, and also had a lip for the locking nut to tighten against. After the rotor was properly installed, a dial indicator was used to check the runout of the rotor surface. The best value achieved was less than 0.0002 inches.
3.4 Overview of the Safety System

A system had to be built in order to provide a safe test setup. The first objective was to guarantee that the system would not be able to run without proper lubrication. This was achieved by routing the power to the motor through two mercoid pressure switches. These switches were connected to the lubrication lines for both the gearbox and the spindle. The electrical circuit would close once a preset pressure value was reached for each of the mercoid switches. These values were set by the user as seen in Figure 3.5. Not only would the system not run if it was not lubricated, but if either of the lubrication systems failed during testing, the circuit would open, and power to the motor would be cut off. The second objective was to lower the chance of personnel being struck by flying debris from any possible failures in the drive train. The best solution was to mount one inch thick plexiglass to the already existing steel blast shield that surrounded the area of the test rotor, as seen in the picture of the test area. This provided protection on all sides of the test area from the front of the rotor, back to the front end of the motor. A detailed look at the lubrication system follows.

3.4.1 Gearbox Lubrication System

Lubricating oil was supplied to the gearbox by means of the two pumps above. These pumps could provide oil at a rate up to 4 gallons per minute, and were powered by two Leeson ¼ horsepower motors. The supply pump was used to pull oil from a reservoir into the system. The reservoir was pressurized by argon gas to reduce frothing of the returning oil. The oil was pushed through a filter that would remove contaminates or wear particles. It then moved into the distributing manifold, where it was directed to each of the five lubrication regions within the gearbox. An overfill valve was installed at the
top of the gearbox to send any extra oil immediately to the reservoir. While oil was needed to lubricate the gears and bearings, excessive amounts in the box would create resistance to movement. For this reason, a scavenge pump was used to draw oil out of the gearbox, and keep a regulated oil level within it. With this, the teeth of the gears could dip into the oil, but any frictional losses could be minimized. The oil then moved back to the reservoir, which contained an immersed cooling coil. The cooling coil circulated tap water that was discharged to a heat exchanger.

3.4.2 Spindle Lubrication System

In Figure 3.6 we see the oil mist generating system, which would supply misted lubrication air to the spindle. Air is supplied at 100 psi, and is controlled by a pressure regulator to approximately 40 pounds per square inch before entering the mist generator. Air then passes through a venturi nozzle inside the generator, and oil, carried from the reservoir by the airflow, was atomized into a fine mist. Larger particles were filtered out by baffles under the venturi, leaving a mixture containing only oil particles of an average size of 60 microns in diameter. The mixture then traveled to the spindle, where the oil mist cools on the surface of the ball bearings. Excess oil dripped from exit locations on each end of the spindle, into a collection tub below, and was disposed of. This process provided a clean supply of lubricant, and prevented contaminants from entering the system by way of the positive pressure created with the spindle.

3.5 Test Section

An assembly was built to house the rotor and test seal. It was designed to provide a safe and stable test region for the seal, through which test data could be easily gathered. The assembly of the test head enclosure can be seen in Figure 3.7. Constructed from
aluminum and one inch thick Lucite, the housing was a 1-ft tall, 1-ft wide, and 10-in long box. This created a pressurized cavity for the seal that could be easily viewed by operators at any time. Description of the assembly will be broken down into its six main components.

3.5.1 Assembly Mounting Plates

Alignment of the housing with respect to the test rotor was critical to our setup. This was achieved through use of the vertical and horizontal mounting plate (Figure 3.7-1). The horizontal plate had four thru holes for mounting to the test rig. The vertical plate was mounted on the horizontal plate, perpendicular to the rig table, using dowel pins and four socket head cap screws. The mounting plates were aligned using a base plate and shims, so that the cut-out hole in the vertical plate was concentric with the output shaft of the spindle, and consequently, the rotor. Access holes were drilled for the four thermocouple wires and five pressure transducers that would be used to monitor the seal and test section. A sight port was cut out of the side of the vertical plate so that movement of the low pressure seal could be observed during testing.

3.5.2 Exhaust Plate

The exhaust plate (Figure 3.7-2) was mounted onto the front of the vertical mounting plate, and provided an exit route for air which had passed through the seal. Two additional pressure tap holes were drilled on the sides of the plate, so that measurements inside the manifold could be taken as air moved to the exhaust port. The plate also had matching holes for those on the vertical plate used for the pressure taps and thermocouples mentioned previously, as well as cut-outs for the sight port and rotor. The
plate also featured a machined groove for the seating of an O-ring that would seal the contact with the Lucite enclosure.

3.5.3 Seal Mounting Plate

The seal plate was bolted to the exhaust plate, and provided a mounting place for the seal. On this plate (Figure 3.7-3), the seal could be positioned correctly on the rotor both axially and radially. This plate had four sight ports to provide viewing of the low pressure side of the seal. Three were located on the left, right, and top faces of the plate, and the fourth lined up with the port in the vertical and exhaust plates. Just as with the exhaust plate, matching holes for pressure taps and thermocouples were drilled. Two of the five pressure taps ended in this plate, so that the pressure near the seal pads could be measured. The remaining taps and guide holes ended in parts of the finger seal assembly that were mounted to the seal plate, and will be discussed at greater length.

3.5.4 Lucite Enclosure

The Lucite enclosure (Figure 3.7-4) provided a safe way for operators to view the high pressure side of the test seal. It was aligned with the plates by two dowel pins, and its contact with the exhaust plate was sealed with an O-ring, preventing any air leakage from the high pressure cavity. Previously, finite element analysis was performed on the enclosure to ensure that it would be safe under the high pressures involved. A distributed load of 60 psig was applied to the surfaces of the enclosure, which was much higher than any possible testing pressure, and the Lucite showed only very small amounts of deflection.
3.5.5 Air Inlet Plate

The air inlet plate (Figure 3.7-5) provided a stable pressure load on the seal that would be measurable. Air entered the cavity via the two tangential ports that directed the flow in a circumferential manner. The feature was designed in order to try and eliminate additional load on the seal due to high air velocities whose values could never be determined. An eighth pressure transducer was installed in this plate to monitor the cavity.

3.5.5 End Plate

The end plate (Figure 3.7-6) closed off the test chamber, and included a 10 inch diameter Lucite sight glass. As with the other plates, it was aligned by two dowel pins. The Lucite was mounted to the main plate by 12 socket head cap screws and a retaining ring. Safety was also checked using finite element analysis identical to that for the enclosure. Just as with the enclosure, it was obvious that deflections would be miniscule.

3.6 Air Supply System

Air is delivered to the system via the 30 horsepower compressor seen in Figure 3.8. It is pumped into a tank continuously until the pressure reaches 250 psi. At this point, the compressor begins to idle. Once the test has begun, and pressure inside the tank drops, the compressor starts up again. This enables us to run tests for longer periods of time without losing too much pressure. The air travels from the tank to our testing room, where it then passes through the digital flow meters. Next in the line is the throttling point, where we control the amount of air flow entering the test section. From here, air moves through the seal, and out the top of the mounting plate.
3.7 Description of the Seal

The basis of this experiment is to test the performance of four different finger seal designs. The four seals of interest are the Single Wedge, Double Wedge, Rayleigh Step, and Double Pad. These seals are composed of front and back plates, and the finger laminates, as seen in Figure 3.9 Our experiment is intended to provide an understanding of the effect that each of these parts has on the seal performance, and to help us derive a design which provides good sealing characteristics without suffering loss of integrity over time. We will begin by looking at the high pressure laminate, which is the same for all the seals.

3.7.1 High Pressure Laminate

Both the high and low pressure laminates can be broken down into three simple shapes. These are the root, the finger, and the foot. An exploded view of this can be seen in Figure 3.10.

True to its name, the root provides the structure from which all the fingers originate. The basic shape of the root is the ring of material which remains untouched during the cutting of the fingers and feet. The inside diameter of the ring is formed by the base of each finger, and the outer diameter matches that of the back plate. In our case, these values are $D_I$ and $D_O$ respectively. The root also contains the holes necessary for pressure equalization, which will be discussed in the back plate section, as well as the three holes for mounting and centering of the seal assembly.

The fingers are what give the seal the ability to comply with external forces, and therefore hopefully eliminate wear. They are simply cantilever beams that respond to the forces experienced by their respective “feet”. The shape of the fingers is determined by
the geometry of the cuts made into the original ring. Figure 3.11 depicts the geometric parameters of the fingers and feet. The fingers are created by making cuts of thickness $I_s$ along an arc $R_s$ from the inner radius of the root, to the foot upper diameter $D_f$. The center of this arc lies on a circle of diameter $D_{cc}$, which is concentric about the shaft center $G_c$. The circle which contains the centers of these arcs has been dubbed the “circle of centers”. On this circle, the center of each arc is offset from the next by the repeat angle $\alpha$, which is the fraction of 360 degrees over the number of fingers. Although our finger geometry was fixed, we can see in Figures 3.12 a-c the effect that the circle of centers would have on the finger shape, and therefore the response of the seal. Increasing the diameter of the circle creates fingers which are short and thick, while shrinking the circle creates fingers that are long and thin. No doubt the mechanical and dynamic properties of the finger seal are dependent on these parameters, but all the seals that we will be dealing with have the same finger geometry.

The feet of the seal are formed by the cutting tool when the arc is completed at $D_f$. At this point, the tool makes a straight cut towards the shaft center $G_c$. This cut ends at the inside diameter of the seal $D_i$, which matches the outer diameter of the shaft. What remains is the characteristic foot shape of Figure 3.13.

3.7.2 Low Pressure Laminate

The root, finger and foot geometry of the low pressure laminate is the same as the high pressure laminate, but its differences provide two very important functions. The first is to block any flow that would pass through the interstices of the high pressure laminate. This is achieved by rotating the laminate so that the fingers cover the gaps between the fingers of its high pressure counterpart. The position of both the mounting and pressure
holes are shifted to facilitate this. The second role of this laminate is to provide lift to the fingers, eliminating rotor contact with the seal. This is made possible by the addition of pads to each foot on the seal. The thickness of the pad can be described in terms of $L_c$, $L_r$, and $L_a$, which are the circumferential, radial, and axial lengths respectively. This can be seen in Figure 3.14. While the pads for each of the test seals were different, they all feature geometry on the bottom surface of the pad that creates a small hydrodynamic lifting force. The pad profile in the $r-\theta$ plane is the same as that of the foot it lies on, because they are made with the same cut. We will now take a closer look at the differences between the four seals that will be tested.

The basic shape of the pads can be described as a parallelepiped. The sides of this box are perpendicular to the rotor surface, and the top of the pad is an arc which is concentric about the center of the rotor. The corner where the connection to the finger is made has a fillet to alleviate stresses caused by lift on the pad. The bottom of the pad is an arc of equal radius to the rotor surface, but is rotated counter-clockwise to create a wedge shape between the rotor and pad surface, as seen in Figure 3.15. The converging region accelerates the flow, generating a net force upward. The second seal is nearly identical to the first. The difference is that it also features a converging region in the axial direction. This creates lift from the air leaking through the seal under the pads, which is driven by the high pressure region. This is illustrated in Figure 3.16. As you can see from Figure 3.17, the third seal has a step in its pad, created by arcs of two different radii, rather than the gradual converging region. This is also a self lifting geometry.
3.7.3 The Double Pad

While the fourth seal follows the same concept as the first three, there are a few major differences in the geometry of the feet and pads. The design of this seal was based around solving the problems associated with the other seals. We wanted to give both laminates the capability to lift, and to reduce any possible leakage between the pads. This was achieved by first decreasing the circumferential foot and pad length of the low pressure laminate, and then adding similar pads to the high pressure side. A picture of this intermediate design can be seen in Figure 3.18. Now both laminates can lift, but we have not solved the leakage problem. To do this, we will rotate the sides of the pad so that they will no longer be normal to the rotor surface, but 45 degrees clockwise from it. A look at Figure 3.19 may make this clearer. The theory here is that when the pads lift, the toe of one pad will lift into the heel of the next. This will cut off any air trying to move radially through the pads. You may have noticed that the high and low pressure pads have different circumferential lengths. The different sizes will generate different amounts of lift, and therefore motion, guaranteeing contact between adjacent pads. Unfortunately the extended high pressure feet intended to block axial flow could not be machined, because the cutting tool used to make the sides of the pads had to pass through the feet as well. While the gap between each pad is only around 0.002 inches, it is now a path which air can pass straight through. Our solution to this was to use the high pressure laminate from the other seals as a third layer, to block these gaps. A look at Figure 3.20 shows that one problem still remains. Air can still escape into the pocket between the first and third laminate. From there it can easily pass between the pads. In order to fill that gap, we extruded the pad on the low pressure laminate through the finger and out the
other side. The pad extends only far enough to reach the same plane as the front of the high pressure pads, or in our case thirty mils. The end result can be seen in Figure 3.21. Axial flow through the seal is now limited to only what passes under the pads.

3.7.4 The Back Plate

If the laminates that make up the seal were simply mounted alone, the seal would suffer a diminished ability to lift. This is due to the friction between the fingers and mounting plate that is created by the upstream air pressure. In order for the seal to be compliant, this friction had to be reduced or eliminated. The solution was to mount the laminates to a specially designed back plate. As we can see in Figure 3.22, this plate has a recess that has been cut out in order to collect air that passes through the aforementioned “pressure equalization” holes in the seal. Upstream air flows through these holes, and gathers in this pocket, creating a pressurized region behind the fingers. While air can pass out of the pocket between the low pressure fingers, there is enough resistance to flow to generate a pressure close to that of the air on the high pressure side of the fingers. This effect greatly diminishes the net normal force and the associated friction experienced by the fingers.

3.7.5 The Front Plate

The purpose of the front plate is to secure the stack of finger laminates and back plate to the mounting plate. An illustration of this concept can be seen in Figure 3.23. This plate simply slides over the 3 seal alignment pins, and is secured to the test section by 12 bolts. It was later determined that too much air was passing through the seal holes and through the back plate pocket. In an attempt to reduce this leakage contribution, a
new front plate design was developed. This can more easily be understood by viewing a cross sectional cut of the seal assembly, as seen Figure 3.24. We can see in Figure 3.25, that the original front plate design allows the test section air to pass unchecked through the pressure balance holes in the laminates. This air then escapes from the pressure balance ring by way of the gaps between adjacent fingers in the low pressure laminate. Figure 3.26 shows the second front plate design. Notice that the inner diameter of the plate has been decreased to cover the majority of the finger, and force the air down near the feet of the seal. In the low pressure side, a .01 inch deep ring shaped cutout is added to provide a path through which air can reach the pressure equalization holes. The purpose here was to provide enough resistance to flow to reduce the amount of air escaping through the back plate pocket, but still retain enough pressure to reduce friction on the fingers. A third model was later created with a .005 deep recess.
Figure 3.1 – Seal Testing Rig
Figure 3.2 – Drive Train

Figure 3.3 – High Precision Spindle
Figure 3.4 – 3D Model of Test Rotor

Figure 3.5 – Mercoids for Lubrication System
Figure 3.6 – Oil Mist Generator

Figure 3.8 – Air Compressor
Figure 3.7 – Test Section Assembly

1. Assembly Mounting Plates
2. Exhaust Plate
3. Seal Mounting Plate
4. Lucite Enclosure
5. Air Inlet Plate
6. End Plate
Figure 3.10 – Exploded Laminate View (Pierson 2004)
Figure 3.11 – Laminate Geometry
Figure 3.12a – Geometry of Cuts with Standard Circle of Centers
Figure 3.12b - Geometry of Cuts with Enlarged Circle of Centers

Larger Circle of Centers Creates Steeper Cuts and Stiffer Finger Geometry
Figure 3.12c - Geometry of Cuts with Reduced Circle of Centers

Smaller Circle of Centers Creates Shallow Cuts and More Flexible Finger Geometry
Figure 3.13 – Characteristic Foot Shape (Pierson 2004)

Figure 3.14 – Basic Pad Geometry (Pierson 2004)
Figure 3.15 – Circumferential wedge profile

Figure 3.16 – Converging axial wedge
Figure 3.17 – Circumferential step profile

Figure 3.18 – Double Pad Concept Model

Dual Pads Enable Independent Lifting Capability to Both Laminates
Figure 3.19 – 45 Degree Double Pad Concept Model

Figure 3.20 – Lift Reducing Air Path in Initial Design
Figure 3.21 – Air Path Closed by Extension of Low Pressure Pad

Extended Low Pressure Pad Completes the Lifting Surface
Pressure Balance Holes Align With Pockets in the Back Plate

Pressure Balancing Manifold

Figure 3.22 – Back Plate Illustration
Figure 3.23 – First Front Plate Serves Only to Clamp the Laminates to the Test Section

Figure 3.24 – Cross Section of Seal Assembly
Figure 3.25 – Leakage Path Through Original Front Plate
Figure 3.26 – Leakage Path Through Second Front Plate
CHAPTER IV

INSTRUMENTATION

4.1 Data Acquisition System

Many characteristics of the system had to be monitored in order to determine the performance of the seal. Values of temperature and pressure on both sides of the seal had to be measured, as well as the leakage rate through the seal, and the rotor speed. What follows is a detailed description of the sensors and how they were utilized in conjunction with the NEFF 470 system seen in Figure 4.1 to maintain the accurate measurements required.

4.2 Measurement of Temperature

In addition to the overall system air itself, it was very important to monitor the temperature of the seal, specifically the pads. This would give us any easy way to know if the fingers had lifted. A quick spike in temperature on the pad would indicate that it was rubbing on the rotor and heating up due to friction. Type K thermocouples were used for our measurements. These were made by Omega Engineering. As mentioned before, four of these thermocouples were chemically bonded to top of the seal pads. They were mounted 90 degrees apart from each other, on the upper surface of the pads as seen in Figure 4.2. Two more thermocouples were used to monitor both the temperature of the room, and that of the air entering the chamber. The necessary reference temperature for
the Neff was provided by an RTD sensor on the connecting board. In this experiment, it was of the utmost importance that the response time of our temperature sensor be very fast. For this reason, the sensing bid on the thermocouple was chosen to be only one hundredth of an inch in diameter, which would obviously be more sensitive to temperature changes than a larger size bid. The thermocouples output voltage range of zero to 4.5 millivolts corresponded to a temperature range of 32 to 212 degrees Fahrenheit. This signal was interpreted by the NEFF system with an approximate error of .75%.

4.3 Measurement of Pressure

The eight pressure taps previously mentioned were used to provide pressure data throughout the system. Each tap was connected to a K8 type pressure transducer from Ashcroft Industries. The transducer contained a stainless steel diaphragm and a polysilicon strain gage. When subjected to pressure, the film would compress, and stimulate the attached Wheatstone Bridge configuration, creating an analog signal that was proportional to the deformation. Due to the small nature of these particular transducers, potentiometers for both gain and offset were not available. Therefore, a direct calibration of the transducers was needed.

For proper calibration, the transducers needed to be powered by the same source, so that they receive the same excitation. In this case, all wiring was channeled through a common bus before going to our 5 volt HP power source. The high precision dial indicator used to read pressure values was connected to the same source. Pressures and their corresponding voltages were recorded simultaneously and imported into an Excel spreadsheet in order to establish a relationship between the two. Values were taken at
increments of 5 psi for a range of zero to 80 psi. This process was repeated ten times in order to ascertain standard deviations and errors. The location of each individual transducer can be seen in Figure 4.3, and all calibration curves can be seen in Table 1.

4.4 Measurement of Rotor Speed

The basic tool used to determine the speed of the rotor was a magnetic pickup sensor. This sensor was positioned next to a gear on the output shaft of the gearbox as seen in Figure 4.4. The movement of the gear teeth through the magnetic field disturbed the flux density of the probe’s field. The frequency of this disturbance was then used to determine the rotor speed, given the number of teeth on the gear. For calibration purposes, the speed was cross checked with a Strobotac. This data was filtered to the NEFF system, where it could be displayed to the operator.

4.5 Measurement of Air Flow

The most important function of the seal was to slow the flow of air through the system. If it didn’t actually seal, then it was of no value. Therefore it should go without saying that measurement of leakage through the seal was paramount. Any leakage through the seal will result in an equivalent flow rate upstream from the test section. Three flow meters were installed parallel upstream from the seal, in order to evaluate this flow rate.

The HFM-200 series flowmeters operate on a thermo-electric principle. A heated capillary tube is mounted in the flow path, with a thermocouple welded to its surface. When fluid passes over the tube, energy is convected away from the surface, and a temperature difference is created between the center and outside of the tube. A voltage output signal is created which is proportional to this temperature differential. The change
in temperature of the tube is a function only of the mass flow rate and heat capacity of the fluid. The output signals were routed into a display box which was calibrated to determine volumetric flow rates for each of the 3 instruments.

Each flow meter in Figure 4.5 is designed for a different range of flowrates. They are installed in a small, medium, and large PVC tubes in parallel. The sum of their readings provides the total airflow though the seal. These signals were sent to both the NEFF and a nearby display. Data was collected in Excel as with the pressure transducers, in increments of 40 SLPM. The resulting curves can be seen in Table 2.
Figure 4.1 – NEFF Data Acquisition System

Figure 4.2 – Thermocouple Locations Seen from Front of the Rotor, and Mounting with Epoxy to Pad Surface
Figure 4.3 – Pressure Transducer Locations (Pierson 2004)
Figure 4.4 – Rotor Speed Sensor
Figure 4.5 – Flow Meters
CHAPTER V

TEST PROCEDURE

5.1 Overview

For consistency in the results, ease of comparison, and repeatability, we developed a test procedure. The procedure provided us with a detailed plan that was used for every test, ensuring that results would be as accurate and as reproducible as possible for our installation. The objective of the test was to acquire data from 10 pressure transducers, 4 thermocouples, 3 flow meters, and one speed sensor. Data from this instrumentation was collected by means of the NEFF system previously mentioned, along with use of the Mozayik data acquisition software.

5.2 Basic Test Setup

We began by installing the seal into the test rig. The four thermocouple wires attached to the low pressure laminate were pulled through specially designed ports on the back plate, and the laminate was aligned with the back plate. The high pressure laminate was then placed on top of the low pressure laminate, and three alignment pins were inserted into the group. Before the seal could be put into position, the thermocouples would have to be threaded through their respective holes in the mounting plate. In Figure 5.1, we can see a port in the back plate that allowed thermocouples passage to the data acquisition system. Once there was no slack in the wires, the seal was placed around the rotor, pushing up on the fingers to allow the seal to slide on. The slack created in the
wires was once again pulled out while sliding the seal towards the mounting plate. When
the seal was flush with the mounting plate, with the wires carefully tucked away, each
end of the thermocouple wire was installed into a male-type thermocouple connector, and
connected to its respective female-type that ran to the NEFF system. Now we began the
process of attempting to center the seal about the test rotor. To do this, thin strips of
transparency were cut, and inserted under the pads, as shown in Figure 5.2. The front
plate was next placed over the alignment pins and pushed up against the seal.

Several precautions had to be taken when installing the mounting bolts. Simply
screwing them in would create problems later. The front must be held flush against the
seal, while installing each bolt in a star pattern. This concept is illustrated in Figure 5.3.
The idea here is that the force exerted on the seal by the front plate is distributed as
equally as possible. We also had to take precautions when tightening the bolts. If any of
the bolts were over-tightened, the front plate would be flexed about the seal. This would
create a torque on the seal laminates, twisting the fingers outward from the center of the
seal, towards the high pressure side. This phenomena caused a few problems, the most
serious being the increased difficulty in the closing of the seal. When the fingers of the
high pressure laminate are flexed away from their low pressure counterparts, air can
easily pass through the new space created between adjacent fingers. For this reason, not
enough pressure is created to clamp the two laminates together and close the seal. The
difference in flow through a seal experiencing this phenomenon, and that of a normally
closing seal is enormous. One example showed an increase in leakage in excess of 60%.
The simple solution to this problem was to tighten the bolts only a little more than what
could be achieved by hand. This would insure only that the seal could not move around.

55
The vast majority of the mounting load would come later from the pressure developed inside the test section. The air pressure would provide a much larger and more uniform load on the seal than we could ever hope to create with mounting hardware.

Once the seal was mounted, the test section had to be closed. The Lucite enclosure was placed over the seal, and bolted into the back plate as shown in Figure 5.4. The rest of the eight bolts were installed, and nuts were placed on each end. The sealing bolts were then tightened in a star pattern, just like the seal mounting bolts, in order to insure that a good seal was created between the plate and the enclosure.

After the test section was closed, the rest of the testing procedure was as follows. Power was supplied to the motor controller, speed sensor, and monitoring computer. The air supply to the lubrication systems was opened. Now the spindle was being lubricated, and the lubrication circuit for the gearbox was pressurized. The gearbox lubrication pumps were then powered up. Next DAQ was started, and our custom test layout was loaded into the program. The compressor supplying air to the test section was then started. Typically, the air storage tank was already at maximum capacity, and the compressor was used only to keep the supply full during testing. If the tank was not already full, the compressor was run long enough to fill the tank before the test was started. The data acquisition now had to be started. A name and storage directory for saved files was created by the operator, and the software began acquiring data.

5.3 Dynamic Testing

The basis of our testing centered around the dynamic testing of the four different seals and three different front mounting plates previously discussed. The purpose here was to see how each of these combinations would compare to one another in terms of
leakage rates and rotor contact, for various rotor speeds. We began by opening the valve which regulated air flow into the test section, until a pressure of 10 psi was developed inside. This was determined to be the point at which there was enough pressure to close the seal, but not enough to impose too much axial force on the rotor and damage the spindle bearings. Once this working pressure was reached, the rotor was accelerated to approximately 5,000 RPM. The pad temperature was allowed time to stabilize, and the speed was then increased to 7,500 RPM. This was repeated at speeds of 10,000, 12,500, and 15,000 RPM. The rotor was then brought to rest over a span of about 15 seconds.

While the amount of leakage allowed by the seal was very important, the crux of the finger seal was that it was “non-contacting”. Therefore, the amount of wear on the seal had to be thoroughly examined. The test section and seal were removed, and the seal was pulled out far enough to be placed on a white sheet in front of the rotor. A strong lamp was then placed in front of the seal and pictures were taken using a high resolution camera and a magnifying lens.

This entire procedure was then repeated for each of the other eleven seal and front plate combinations.

5.4 Static Testing

While we realized that the rotor could not be run at high test section pressures, it is still important to see what maximum pressure each seal could reach. It was becoming obvious that the capabilities of each seal could not be evaluated completely at our relatively low operating pressure of 10 psi. A simple test was developed for this purpose.

The test was set up in the fashion described above, but when it was time to let air into the system, a change was made. The valve controlling the flow rate was quickly
opened, and the maximum possible amount of air entered the test section. The Mozayik software acquired the flow rates and corresponding pressures as the test section stabilized. After the system reached equilibrium, the valve was closed, and the test was stopped. This test was performed on each of the four seals using only the third front plate.
Figure 5.1 – Thermocouple Pocket

Figure 5.2 – Transparencies Added Between Rotor Surface and Seal Pads
Figure 5.3 – Front Plate Bolt Tightening Pattern

Figure 5.4 – Mounting of Enclosure to Test Section
CHAPTER VI

TEST RESULTS & DISCUSSION

6.1 Dynamic Test Results

In order to simplify our analysis, we need an easier way to reference each seal and front plate design. The single wedge, double wedge, step pad, and double pad finger seals will be known as seals 1, 2, 3, and 4 respectively. Front plate 1 refers to the original front plate, which has no flow limiting component. Front plates 2 and 3 feature the smaller inner diameter that reduces flow, and they have flow path cutouts of 0.01 and 0.005 inch respectively. Refer to sections 3.7.2 through 3.75 for a detailed description of the various seals and front plates. Table 3 gives a brief summary of every seal and front plate combination.

6.1.1 Temperature Analysis

For a general temperature comparison, one must look at Figures 6.1a-6.12a. These 12 plots depict the relationship between pad temperature and rotor speed for each seal and front plate combination. We can see that, for the most part, the pad temperatures closely follow the profile of the rotor speed curve. Each increase in RPM raises the temperature of the pads in kind. Anomalies occur only when a pad makes physical contact with the rotor. Obviously this creates a spike in the temperature profile. This can be seen in the tests with the first and second front plates and the original three seals, and
seems to occur during the acceleration phase from rest to 5,000 RPM. The third front plate apparently forces enough air under the pads to protect against this startup wear. The fourth seal does not experience pad touchdowns with any of the front plates.

As one would expect, a seal’s pad temperature profile is closely related to its under-pad geometry. Seal number 3 has no axial wedge, much like seal 1, and its step design gives it no clearance whatsoever on the “toe” half of the pad. This combination explains its large temperature spike during startup with the first two front plates. Only the third plate provides enough cushion to eliminate this contact. The first and second seals touchdown only slightly compared to the step seal, due to the larger clearance of the circumferential wedge.

It’s hard to make a direct comparison between the first seal and the other three. This is because the low pressure laminate of the first seal is thinner than the other seals. Consequently, the fingers of the first seal require less force to be generated by their pads in order to lift. As far as temperature is concerned, this allows the first seal to perform better than seal 2, which is superior in terms of design for leakage and lift. That said, the fourth seal experiences much lower temperatures than seals 2 and 3. This is most likely due to the increasing lifting ability provided by the interlocking pads.

One should notice that with every test, the bottom thermocouples, T1 and T2, are always reaching higher temperatures than the top thermocouples, T3 and T4, even if there are no pad touchdowns. There must be something about the relative position of the seal with respect to the rotor that’s creating this effect. The first thought was that this was a result of an unavoidable error in seal alignment during installation. If this was the case, it would be very unlikely that the same thermocouples would be hottest in each case. It
would be very hard to make the same alignment mistake in every test that was conducted. Another possibility is that the test rotor is not perfectly aligned. Under ideal circumstances, the seal would be perpendicular to and perfectly centered about the rotor. Now of course we cannot make either of these conditions perfect, but our pad temperature anomaly may be explained by an alignment problem. If the rotor were tipped forward with respect to the test section mounting plate, as shown in Figure 6.13, then once the seal was clamped down, the top and bottom pads would experience very different operating conditions. On the top of the rotor, the pads would have an increased axial wedge due to the tilting of the rotor surface. However, on the bottom, the orientation of the rotor creates a slight obstruction of airflow under the pads, and diminishes the effect of the axial wedge. Of course these pads can still lift due to air being forced under them, as well as the circumferential wedge effect, but this tilt in the rotor could reduce the lifting enough to generate the increased pad heating and localized touchdowns that we’ve been seeing. Further proof of this theory can be seen in the section on pad wear analysis.

6.1.2 Leakage Analysis

Figures 6.1b-6.12b show the relationship between seal leakage and rotor speed. The profiles depicted here follow the trends that we predicted, for the most part. Just like with the temperature profiles, flow through the seal is highly dependent on the pad geometry. As expected the first seal, with no axial wedge, had the most leakage. The step seal, having no axial wedge, but a smaller under pad clearance, performs noticeably better. Seal number 2, with both the axial and circumferential wedges, shows the best sealing characteristics at 10 psi. The front plate modifications effect the seal leakage rate
as one would expect, by choking off air flow through the assembly. Plate three experiences the least leakage, followed by plate two, and then plate one. All of the seals were evaluated through the use of Arora’s flow factor defined previously. The results at 10 psi can be seen in Figure 6.14. As we can see, the performance of the finger seals falls in between the labyrinth and brush seals, which is to be expected. An adaptive seal such as this will never seal as well as a brush, because of the flow that must be allowed through to create a lifting film, but will outperform a labyrinth by providing smaller clearances.

Surprisingly, even though the fourth seal should be superior in design, compared to the other three, it experiences more leakage than seals two and three at this pressure. This is startling at first, but the reason becomes clear if we remember how the fourth seal functions. Unlike the other seals, the new seal is made up of three laminates instead of two. For this reason, more pressure is required to “close” the seal. While 10 psi is enough to close the two laminate seals, this seal’s laminates are not clamping together, and there is some air passing between the fingers of each successive laminate. To determine if this theory is correct, another test was developed to see what maximum pressure could be sustained by each seal. The results of the static test will be discussed in a later section.

Figure 6.15 shows the typical variation of air flow with rotor speed during a dynamic test. This graph was chosen from one of the dynamic tests to illustrate key points of discussion. The most important thing to see here is the inverse proportionality of the relationship between changes in the rotor speed and the air leakage through the seal. As we can see in Figure 6.16, each time the rotor accelerates to the next speed, there is a corresponding drop in air flow, and when the rotor is stopped, a large increase in flow
occurs. This effect is a product of the radial growth of the rotor. As the rotor accelerates, it experiences an increasing amount of centrifugal force, and increases in diameter. This effect is may be only a fraction of a mil, but in this case, that is the same order of magnitude as the clearance under our pads. Consequently, this results in a measurable change in leakage.

It is important to note that the larger drops in flow highlighted in Figure 6.17 are a result of adjustments in air flow by the operator. In order to make accurate comparisons between flow rates for different tests, it was important to keep the test section pressure as close to 10 psi as possible at all times. Unfortunately, when the rotor increases in speed, and decreases leakage, it also causes a rise in test section pressure. If the pressure got too far from 10 psi, the throttling valve for the air had to be closed enough to bring it back to within an acceptable tolerance.

At first glance, it may seem that the seal experiences a rather large permanent decrease in leakage over the time of the test. In this case, Figure 6.18 seems to show that the flow rate has dropped more than 200 SLPM. The first thought was that the seal somehow centered itself during the test. This seems unlikely, given that the root of the seal is bolted down, and unable to move without considerable force. Another possibility is that the position of the rotor with respect to the seal has somehow changed. Considering that this drop is seen in every test, that theory also does not make sense. A look at Figure 6.19 sheds some light on the subject. We see here, that if we add back in the flow loss created by operator adjustment, the actual loss is really only about 60 SLPM. This smaller drop can be explained by taking a look at our air delivery system. Once testing begins, air is being continually pulled from the storage tank, and must rely
on the compressor to keep the pressure up. During the course of a dynamic test, where air is being pulled from the tank for an extended period of time, the compressor does not have the ability to keep the pressure steady. Tank pressure drops considerably, and thus our flow rate suffers. This phenomenon was easily reproduced with a static test, to eliminate any other possibilities.

6.2 Wear Analysis

If we want to gain a complete understanding of the operation of these seals, examination of the pad wear is vitally important. Close up pictures of the typical wear for each seal can be seen in Figures 6.1c-6.12c. Note that in each case, wear occurs in a different circumferential position on the pad, and never on the “toe” of the pad. This is important, because if the pads were floating due to the circumferential wedge, then the toe portion of the foot would usually be closest to the rotor, and we would expect to see wear in that region. Any wear seen on the first three seals occurs near the center of the pad, which implies that this region was closest point to the rotor. If this is the case, then our circumferential wedge was already gone at this point, and lift was completely dependent on the axial wedge effect. Of course the significant wear patterns were a result of the touchdowns discussed earlier, and those always occurred during startup. This seems to suggest that the seal lifted past the operational range of the circumferential wedge instantly. In fact, it’s possible that the act of pressurizing the test section provided enough lift from the axial wedge to eliminate the circumferential wedge before the test even began. Obviously this calls the value of the circumferential wedge into question. The minimal wear on the fourth seal occurs only on the heel portion, showing that given
the same test section pressure, each foot rotated even farther, and thus generated even more lift than the other three.

As we mentioned previously, all wear is occurring on the bottom of the seal, in a pattern centered about the “6 o’clock” position. Seals 1 and 2 show wear from 5 to 7 o’clock, while seal three shows a wider pattern of 4 to 8 o’clock. Seal four shows a small pattern spanning only about 5 adjacent pads. All of these wear patterns have a common feature however. No matter the amount of wear on the pad, there is always a triangle shape developed, with its “base” on the high pressure side of the pad. This is especially obvious in Figures 6.1c and 6.4c. If the rotor surface were parallel with the bottom surface of the pad when they touched, this would not happen. In that case, we would see a straight wear pattern across the width of the pad. The only way that this triangle wear shape can be created is if the rotor is tilted downward, as seen in Figure 6.13, so that its surface gets closer to the pad as we approach the high pressure side of the pad. This proves our tilted rotor theory that was developed in the temperature analysis section.

6.3 Static Test Results

As stated earlier, the static test was developed to determine the max pressure that each seal can reach in our setup. This would give us an idea of the effectiveness of each seal at the higher pressures. The results can be seen in Figure 6.21, and there are no surprises. The fourth seal is indeed more effective than seal two, proving our theory that it requires more upstream pressure to properly close.
6.4 Leakage Investigation

While our seal design has proven to be more effective than its predecessors, it’s still not perfect. We had assumed that leakage between the pads had a large impact on our performance. While closing that gap decreased our leakage and improved our lifting capability, there is still more progress to be made. Further testing has been developed in an attempt to determine additional sources of leakage.

6.4.1 Loss Between Plates

One possible source of performance loss could be leakage between each successive part in the seal assembly. This includes front plate and laminates, laminates and back plate, and back plate and test section. The mating between these parts does not include a gasket. This metal on metal contact may introduce paths through which air can escape. Small as they may be, it was worth looking into.

This phenomenon was examined by a two stage test. In the first stage, a static test was conducted at 10 psi. Only the combination of seal 4 and front plate 3 was tested, because this was the most successful combination. The purpose here was to obtain a base leakage value for this configuration, which could then be compared to using data from the next stage.

In preparation for the second test, a caulking compound was used to fill in the space between the front plate and test section. The compound was compressed into the gap, insuring that this space was completely filled, and then the excess was scraped off the sides. The end result can be seen in Figure 6.22. Now that every possible leakage path between the plates has been eliminated, another 10 psi static test was run, and the leakage rates of the two tests were compared. While the comparison showed a drop in leakage
after the addition of the calking compound, it was almost negligible. The scant 1% decrease in flow rate is not really enough to even rule out the unavoidable variation between successive tests.

6.4.2 Loss Through the Back Plate

As discussed in the section covering the description of the back plate, the air that gathers in the back plate manifold constantly escapes through the gap created between consecutive low pressure fingers, as shown in Figure 6.23. This was certainly a rather large source of leakage. The solution to this problem was rather simple. We placed the back plate on a sheet of poster board, and fixed it with rubber cement. We then cut out the extra paper, leaving a ring with geometry matching that of the back plate. Next, we poked out the alignment holes, and one third of the pressure equalization holes in the paper ring, creating what we see in Figure 6.24. The remaining equalization holes were to be cut out in future experiments.

The seal was then mounted to the test section, and another 10 psi static test was performed. This time the results were more impressive. Compared to the base experiment conducted earlier, this new configuration experienced approximately 20% less leakage. With such a substantial improvement, we wanted to see the effect that the added ring had on maximum seal pressure. The seal was subjected to the maximum available flow rate, just like with the maximum pressure static tests conducted on the other seals. The results were plotted alongside those of the other 4 static tests in Figure 6.25.

Obviously the addition of the new ring had a large impact on the seal leakage, but it has another effect. The flow that would have passed through the old back plate now
remains inside the pressure equalization pocket. This in turn raises the pressure inside the pocket. While it was first thought that this would further decrease friction on the fingers, we no longer believe this to be the case. Unfortunately, the addition of this ring has likely not only eliminated any possibility for friction reduction, but actually increased it. However, with nowhere else to go, the test section air can escape only under the pads. This will enhance the lifting ability of the seal. It remains to be seen whether this will be enough to overcome the increased friction experienced by the fingers. Without the paper ring, at 10 psi, pressure loss from the high pressure side of the seal to the back plate pocket was between 9%-10%. The addition of the ring brings that value down to less than 1%. At the higher pressures reached in the second test, the pocket actually builds slightly more pressure than the upstream condition. This is probably due to the air being jammed into the recess, and not having anywhere else to go. Further dynamic testing must be performed in order to determine the viability of the 3rd plate. If nothing else, this has shown the severity of leakage through the back plate. Future designs should include modifications that would decrease this effect.
Figure 6.1a – Seal 1-1 Temperature Development (See Table 3 Appendix A)

Figure 6.1b – Seal 1-1 Leakage Development (See Table 3 Appendix A)

Figure 6.1c – Seal 1-1 Wear
Figure 6.2a – Seal 1-2 Temperature Development (See Table 3 Appendix A)

Figure 6.2b – Seal 1-2 Leakage Development (See Table 3 Appendix A)

Figure 6.2c – Seal 1-2 Wear
Figure 6.3a – Seal 1-3 Temperature Development (See Table 3 Appendix A)

Figure 6.3b – Seal 1-3 Leakage Development (See Table 3 Appendix A)

Figure 6.3c – Seal 1-3 Wear
Figure 6.4a – Seal 2-1 Temperature Development (See Table 3 Appendix A)

Figure 6.4b – Seal 2-1 Leakage Development (See Table 3 Appendix A)

Figure 6.4c – Seal 2-1 Wear
Figure 6.5a – Seal 2-2 Temperature Development (See Table 3 Appendix A)

Figure 6.5b – Seal 2-2 Leakage Development (See Table 3 Appendix A)

Figure 6.5c – Seal 2-2 Wear

75
Temperature and Rotor Speed
10 Psi Pressure Differential

Figure 6.6a – Seal 2-3 Temperature Development (See Table 3 Appendix A)

Rotor Speed and Leakage
10 Psi Pressure Differential

Figure 6.6b – Seal 2-3 Leakage Development (See Table 3 Appendix A)

Figure 6.6c – Seal 2-3 Wear
Figure 6.7a – Seal 3-1 Temperature Development (See Table 3 Appendix A)

Figure 6.7b – Seal 3-1 Leakage Development (See Table 3 Appendix A)

Figure 6.7c – Seal 3-1 Wear
Figure 6.8a – Seal 3-2 Temperature Development (See Table 3 Appendix A)

Figure 6.8b – Seal 3-2 Leakage Development (See Table 3 Appendix A)

Figure 6.8c – Seal 3-2 Wear
Figure 6.9a – Seal 3-3 Temperature Development (See Table 3 Appendix A)

Figure 6.9b – Seal 3-3 Leakage Development (See Table 3 Appendix A)

Figure 6.9c – Seal 3-3 Wear
Figure 6.10a – Seal 4-1 Temperature Development (See Table 3 Appendix A)

Figure 6.10b – Seal 4-1 Leakage Development (See Table 3 Appendix A)

Figure 6.10c – Seal 4-1 Wear
Temperature and Rotor Speed
10 Psi Pressure Differential

Figure 6.11a – Seal 4-2 Temperature Development (See Table 3 Appendix A)

Rotor Speed and Leakage
10 Psi Pressure Differential

Figure 6.11b – Seal 4-2 Leakage Development (See Table 3 Appendix A)

Figure 6.11c – Seal 4-2 Wear

Region of Wear
Figure 6.12a – Seal 4-3 Temperature Development (See Table 3 Appendix A)

Figure 6.12b – Seal 4-3 Leakage Development (See Table 3 Appendix A)

Figure 6.12c – Seal 4-3 Wear
Figure 6.13 – Interaction of Aligned and Misaligned Rotor with Finger Pads

Figure 6.14 – Flow Factor Comparison for All Seal Combinations at 10 psi and 15 kRPM
Figure 6.15 – Typical Flow vs Speed Plot (Example taken from Seal 8)

Figure 6.16 – Flow Response to Changes in Rotor Speed (Example taken from Seal 8)
Flow Reductions to Maintain Pressure at 10 psi

Figure 6.17 – Flow Drops from Throttle Adjustments (Example taken from Seal 8)

Figure 6.18 – Apparent Flow Loss During Testing (Example taken from Seal 8)
Figure 6.19 – Actual Flow Loss During Testing (Example taken from Seal 8)

Figure 6.20 – Static Pressure Test for All Seals with 3rd Plate at Room Temperature
Figure 6.21 – Caulk Added to Fill Gaps Between Layers

Caulking compound used to fill possible gaps between assembly components

Figure 6.22 – Back Plate Leakage Path

Inner Lip of Back Plate (Back Plate not shown)

Pathway for escaping air
Figure 6.23 – Paper Ring Added to Back Plate

Figure 6.24 – Effect of Added Ring on Maximum Pressure
CHAPTER VII

CONCLUSION & FUTURE WORK

In this research, we tested several compliant finger seal configurations in order to gain a better understanding of the complex inner workings of this sealing technology. Four different finger seal geometries were tested. These were the original circumferential wedge, the circumferential and axial wedge, the step, and the double pad, or seals 1, 2, 3, and 4 respectively. The first three seals performed as expected. Seal 1 allowed the most leakage, but due to its thinner fingers, lifted more easily than the others, and stayed lower in temperature. This made comparison with the other three seals difficult. Seal 2 experienced the least amount of leakage, and experienced few lifting problems. Seal 3 had some serious lifting problems during startup, producing a great deal of wear and overall higher temperatures, and also allowed more leakage than seal number 2.

The fourth seal was designed to be superior to all previous finger seal designs in every way. It featured the same under-pad geometry as that of seal 2, with the circumferential and axial wedges. In this design, both laminates now featured pads, which had slanted sides which interlocked with adjacent pads. This blocked air escaping up through the pads, thus trapping more air and improving lifting capability. The high pressure laminate from the other seals was used in this system to block axial flow between the pads. Dynamic testing showed seal 4 to experience less wear and lower temperatures than seal 2, but slightly higher leakage. This occurred because the 10 psi
pressure used in the dynamic tests was not enough to completely close this three laminate system, unlike the other configurations, and thus the comparison was flawed. A static test verified this, and showed the true capability of seal 4. All seals were subjected to maximum airflow, and seal 4 established the highest pressure differential.

Each of the four seals was tested using three different front plates. The first plate served only as a ring that was used to fix the laminates to the back plate. The second plate had a much smaller inner diameter, which limited flow into the back plate, and forced it under the pads. The third plate further decreased flow into the back plate by reducing depth of the recess which the air moved through. As expected, front plate 3 provided the lowest leakage, followed by plate 2, and then plate 1.

In order to determine the true potential of finger seal technology, continued testing needs to be done. Unfortunately, our test rig is very limited in what testing conditions it can provide. A system must be developed which could provide higher temperature, pressure, and rotor speed operating conditions. This would paint a much clearer picture of the abilities and limitations of each seal design. Also, further investigation into the effectiveness of a ring added to the back plate in order to decrease flow would prove valuable. Our tests showed this ring to drop leakage through the seal by 20%, but it remains to be seen whether this ring also creates enough undesirable friction to prevent lifting of the fingers.

While we have shown the new double pad seal to be superior to its predecessors, there is still room to improve this technology with future designs. Under-pad geometry will continue to be critical. I believe that we have shown impact the circumferential
wedge to be minimal at best. Future designs should focus on a simpler geometry which utilizes the vastly superior axial wedge effect and minimizes pad to rotor clearance.
REFERENCES


Proctor, Margaret P. and Bruce Steinetz. “Noncontacting finger seal”, United States Patent No. 6,811,154, November 2004
APPENDIX

<table>
<thead>
<tr>
<th>Pressure Transducer</th>
<th>Curve Fit Equation</th>
<th>Error (psi)</th>
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<td>1</td>
<td>$P = 1740V - 4$</td>
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</tr>
<tr>
<td>2</td>
<td>$P = 2370V - 1.91$</td>
<td>0.6548</td>
</tr>
<tr>
<td>3</td>
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<td>4</td>
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Table 1 - Curve Fit Equations for Pressure Transducers

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<th>Curve Fit Equation</th>
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<tr>
<td>400 SLPM</td>
<td>Flow = 39.921V - .2158</td>
</tr>
<tr>
<td>1500 SLPM</td>
<td>Flow = 150.26V - .5767</td>
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Table 2 – Curve Fit Equations for Flow Meters

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<th>Third Layer</th>
<th>Front Plate</th>
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<td>Circumferential Wedge</td>
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<td>5 Mil Recess</td>
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<tr>
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<td>Double Wedge</td>
<td>-</td>
<td>Original</td>
</tr>
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<td>-</td>
<td>10 Mil Recess</td>
</tr>
<tr>
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<td>Double Wedge</td>
<td>-</td>
<td>5 Mil Recess</td>
</tr>
<tr>
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<td>Step</td>
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Table 3 – All Seal Configurations