Performance Characteristics of Brush Seals for Limited-Life Engines

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ABSTRACT

Brush seals are potential replacements for air-to-air labyrinth seals in gas turbine engines. An investigation has been conducted to determine the performance characteristics of brush seals for application in limited-life gas turbine engines. An elevated temperature, rotating test rig was designed and built to test labyrinth and brush seals in simulated subsonic and supersonic engine conditions. Results from initial tests for subsonic applications demonstrated that brush seals exhibit appreciably lower leakage compared to labyrinth seals, and thus offer significant engine performance improvements. Performance results have been obtained showing the effect of various brush seal parameters including: initial interference, backplate gap and multiple brush seals in series.

NOMENCLATURE

\[ \begin{align*}
CL &= \text{labyrinth seal radial clearance, in} \\
N &= \text{disk rotational speed, rpm} \\
Pd,PD &= \text{exit air pressure, psia} \\
Pu,Pin &= \text{inlet air pressure, psia} \\
T,Ta &= \text{inlet air temperature, } ^\circ\text{F or } ^\circ\text{R} \\
Tm &= \text{seal material temperature, } ^\circ\text{F} \\
V &= \text{disk surface speed, ft/sec} \\
W &= \text{air flow rate, lb/sec} \\
\Delta P &= \text{pressure drop across seal, psid} \\
\Phi &= \text{flow factor} = \frac{W/T}{Pu, \sqrt{R}/(\text{psia*sec})}
\end{align*} \]

INTRODUCTION

Aggressive pursuit of increased performance in gas turbine engines is driving the thermodynamic cycle to higher pressure ratios, bypass ratios, and turbine inlet temperatures. As these three basic parameters increase, airflows in the internal air system and resultant thermodynamic cycle losses increase [Moore (1975)]. This conflict of reducing internal airflows while increasing thermodynamic efficiency and performance is putting more emphasis on improvements to the internal flow system. Studies have shown that improvements to the internal flow system to reduce leakage can yield an increase in thrust by as much as 17%, or a decrease in the low pressure turbine inlet temperature of 175 \( ^\circ\text{F} \). One such improvement that is receiving renewed interest and research is the brush seal. The brush seal offers as much as an order of magnitude reduction in leakage flow over a conventional labyrinth seal.

A brush seal is shown schematically in Figure 1. It has a set of densely packed, fine diameter (typically 0.003 inches) metallic fiber bristles sandwiched between inner and outer packing plates. The bristles are angled in the direction of rotation of the shaft to reduce friction and wear. The angle...
causes the bristles to bend rather than buckle when shaft excursions occur. The bristles run against a ceramic shaft coating to prevent/reduce wear on the shaft. These features allow the brush seal to retain its performance even after large excursions, whereas the labyrinth seal would incur a permanent increase in clearance under such conditions.

The purpose of this paper is to present results from a brush seal experimental investigation. The objectives of the investigation were to evaluate the performance improvements, wear characteristics, and cost impacts of replacing labyrinth seals with brush seals in a limited life engine. Initial results were published by Chupp and Nelson (1990).

BACKGROUND

For over 50 years now, labyrinth seals have been the primary seal used for air-to-air sealing in gas turbine engines. However, labyrinth seals have an intrinsic clearance associated with them due to the nature of their design. This clearance can initially be tight, but over time increases appreciably due to shaft excursions and thermal growth. This increase in clearance increases parasitic leakage and engine performance losses. Ludwig and Bill (1980) point out that the losses associated with the increase in clearance of the labyrinth seal can result in as much as 17% loss in power and a 7.1/2% increase in specific fuel consumption (SFC) over time. With the demands of greater performance and reduced fuel consumption being placed on advanced engines, the need for improved sealing techniques is imperative.

The idea of replacing labyrinth seals with brush seals was first investigated in 1955 under the General Electric (GE) J-47 engine test, that proved unsuccessful. Rolls Royce successfully tested brush seals in some of their demonstrator engines in the 1960’s, followed by a test on a production standard RB-199 in 1987. At present, the only production engine with brush seals is the IAE V2500, that was certified in 1987. This engine has brush seals in three locations, the high pressure compressor delivery and the front bearing chamber as pressure balance seals. Allison has successfully tested brush seals in two of their engines to date [Holle and Krishnan (1990)]. The first was the T800, that was tested with a brush seal at the power turbine discharge location. The second was the T406 Plus, that had 13 brush seals at compressor interstage locations and six in the engine hot section. These Allison tests not only demonstrated leakage flow reductions of up to an order of magnitude over labyrinth seals, but also the tolerance of brush seals to transient clearance changes.

Over the past couple of years, research on brush seals has increased with several papers published [Braun et. al. (1990), Conner and Childs (1990), Ferguson (1988), Flower (1990), Gorelov et. al. (1988), Holle and Krishnan (1990), Mullen et. al. (1990)]. Results from these research programs have been very positive. They have shown that brush seals can offer dramatic reductions in parasitic leakage, improved flow metering capability, and better rotordynamic characteristics compared to conventional labyrinth seals.

BRUSH SEAL INVESTIGATION

The brush seal investigation, described in this paper, was initiated in September of 1988 and was completed in July of 1990. The objective was to determine leakage and wear characteristics and cost impacts of brush seals compared to labyrinth seals. Limited-life engines were the primary applications considered. The investigation included: an engine study to identify potential locations to install brush seals; design and fabrication of an elevated temperature, rotating test rig; and testing of candidate brush seal configurations. The rig was designed to simulate subsonic and supersonic applications of limited-life engines, but the initial testing was limited to subsonic simulations.

In the engine study, selected state-of-the-art, limited-life engines were considered in determining potential locations for brush seals to replace labyrinth seals. Figure 2 summarizes the results of the study. This figure illustrates seven different locations to install brush seals and gives the corresponding operating conditions at these locations. Two prime locations, i.e., compressor backface ("B") and turbine front face ("C"), were identified as the best locations to apply brush seals. These locations offer the maximum potential for reducing leakage and thus increasing engine performance.

Also in the study, engine cycle data were analyzed to determine a test cycle for the seal rig testing. The cycle chosen is representative for limited-life engines and has an initial maximum power condition (100% engine speed) for 30 minutes, corresponding to launch at altitude, followed by a cruise power condition (85% engine speed) for 35 minutes, corresponding to cruise operation at sea level.

![Figure 2: Summary of limited-life engine study](image)

EXPERIMENT

The brush seal test rig, designed and built in the investigation, is driven by an air turbine that rotates the rig up to 40,000 rpm. The test disk diameter is 5.10 inches so that the maximum test surface speed is nearly 900 ft/sec. The rig is provided with various cooling and buffering air to protect the static structure. The bearings are lubricated and cooled...
by a circulating lubrication system. Heated air is supplied to the rig at temperatures up to 1500 °F. The maximum allowable pressure of the test air is between 65 and 200 psia depending upon the air temperature level.

Figure 3 shows a cross section of the test rig with the various rig interfaces and features identified. Figure 4 is a photograph of the rig with a brush seal being installed and Figure 5 is a photograph of a brush seal and test disk. The disks are coated to investigate the effect on performance and wear of the surface running against the brush seal. The seal holder as configured at the top of the disk in Figure 3 can hold up to three brush seals so that multiple brush seals in series can be tested. The seal holder axial location is adjustable, as shown below the disk in Figure 3, so that the seal could be located at four different axial locations. This allowed up to four different individual brush seals to be tested separately with a given disk. A labyrinth seal ring that fit into the holder was provided so that flow comparison between labyrinth and brush seals could be made.

Rig instrumentation was provided to measure: shaft rotational speed, inlet and exit pressures, flow rate, air temperatures, and various rig monitoring temperatures, pressures and vibrations. Multiple probes were used for the seal inlet and exit pressure and the seal inlet temperature so that more accurate, representative data could be obtained. The seal flow rate was measured using an upstream orifice.

An initial series of tests were conducted on the brush seal rig. The testing was limited to investigating current brush seal designs as applied to subsonic, limited-life gas turbine engines. The maximum test temperature was approximately 600 °F that simulates subsonic applications. The initial test matrix included fifteen brush seals and one reference labyrinth seal as summarized in Table 1. The seal parameters investigated include: initial seal/disk interference, backplate gap height, repeatability, bristle surface finish operation, disk surface treatment, number of seals in series, operating temperature, and seal vendor. Brush seal Configuration #1 in Table 1 was the baseline brush seal. It had nominal values for the various seal design parameters. The other configurations in Table 1 had variations in one or more of the seal/disk parameters to see their effect. All the seals tested except the last two listed were supplied by Cross Manufacturing Co. (1938) LTD of Devizes, Wilts, England. The last two seals were supplied by EG&G Sealol Engineering Products of Warwick, Rhode Island and Textron Turbo Components of Walled Lake, Michigan. These last two seals replaced Configurations #12 and #15 that were originally planned to be tested.

A four phase approach was followed in testing each brush seal. First, an ambient temperature flow performance check was conducted at two moderate rotational speeds
that most of the wear occurs in the first four hours. The rig was established after it was determined for Configuration #1 tested for a total of four to eight hours. This total test time at a higher speed (35,000 rpm) and making an abbreviated performance check after each hour. Most of the seals were made by testing a seal for a series of one hour time intervals then acquired at successively lower pressure drops until nearly zero is reached again. Fourth, have wear-in period (of about four hours) and then wear slowly and (4) have performance and wear characteristics that vary with initial interference and disk surface finish/bristle finish machining operation. These findings were based on data reported for Configurations #1, #3 and #9 (see Table 1).

In this paper, results for four additional configurations will be presented and compared with the previous data. These four configurations are #4, #5, #8, and #A1, as listed in Table 1, and reflect the effects of backplate gap, repeatability, two seals in series, and seal vendor, respectively. The results presented will primarily involve leakage performance, i.e., seal flow factor versus seal pressure ratio parameter. Typical engine cycle simulation results also will be given. Wear testing data for these configurations are similar to that reported previously and will not be covered here.

**Engine Cycle Simulation Results**

Typical engine cycle simulation results for two brush seal configurations are given in Figure 6. Configuration #1 is the baseline and Configuration #5 has the same seal construction parameter values as the baseline to determine repeatability. This figure is a plot of leakage flow factor and disk rpm versus test time. The points are individual data points and the lines show approximately how the flow and speed vary between points. During the initial period, the speed is stepped up to 35,000 rpm and then maintained at
that speed (launch condition). The leakage flow starts out low for the well compacted brush with a moderate interference. As the disk diameter increases with the increasing speed, the bristles adjust slowly to the diameter change because the pressure drop across the seal holds them in place. After the initial "launch" period, the change in flow is small indicating that a nearly equilibrium condition has been reached. The rig speed is then decreased to 30,000 rpm (cruise condition). The leakage flow initially jumps up as the disk diameter decreases with the sudden decrease in speed, and the bristles respond slowly with the imposed pressure drop. Within about seven minutes steady state is reached. The leakage flow characteristics of Configurations #1 and #5 in Figure 6 are similar with the final flows agreeing within three percent. Thus, the response of the bristle packing was very repeatable. The data in Figure 6 demonstrate the performance repeatability obtainable from brush seals and from the test rig.

**Performance Testing Results**

Testing to acquire performance results was done immediately after the cycle simulation testing for most of the brush seals. Some seal wear would have occurred by then, but the flow characteristics should still be representative for limited-life engine applications. Figure 7 shows performance plots obtained for Configurations #1 and #3 brush seals and the reference labyrinth seal at approximately 30,000 rpm and 500 to 600 °F air temperature. These data were presented previously in separate plots [Chupp and Nelson (1990)] and are given here for comparison purposes. Figure 7 is a plot of flow factor versus a pressure ratio parameter that approximately linearizes the data at lower pressure ratios. The pressure drop parameter was derived from porous-wall flow expressions [Green and Duwez (1951)]. The data points in Figure 7 are stabilized ones taken three to six minutes after setting off a pressure drop condition. For the brush seal data, the first data point was for a very low pressure drop for which the bristles are in a relaxed position and a minimum flow factor is measured. Pressure drop was then successively increased yielding the lower portion of the curve. The pressure drop was then successively decreased and the upper curve was obtained. The resulting brush seal data in Figure 7 are classical hysteresis curves. The difference between the upper and lower curves is due in part to the bristles being displaced radially outward by the disk during the performance testing as the rig encounters small speed changes between test points. Speed increases cause the disk diameter to increase slightly and the bristles to be pushed out. Although the rig speed returns rapidly to the nominal value, the bristles respond slowly for the higher
imposed pressure drops (higher values of pressure ratio parameter).

The data in Figure 7 demonstrate the significantly lower leakage (factor of four to seven) of the brush seals compared to the reference labyrinth seal (with a radial clearance of about 0.006 in.). Further, these data show a significant difference in flow characteristics between the two brush seal configurations. Configuration #3 is identical with #1 except the initial interference values (see Table 1). The difference in leakage was attributed primarily to seal-to-seal variations and amount of bristle wear incurred, and not the initial interference.

Two Seals in Series

Multiple brush seals in series were included in the test matrix because the pressure drop across any one seal is lower. This is important for higher pressure drop applications where higher forces on the bristles would deflect them under the backplate. Figure 8 gives flow performance results for Configuration #8 that had two baseline configuration brush seals (#1's) in series. The resultant hysteresis curve is similar to that for single brush seals except the level is lower and the variation between the upper and lower portions of the curve is less. In all the testing of this configuration, the leakage was not as sensitive to disk speed changes as it was for the single brush seals. Thus, two brush seals in series not only allow a higher pressure drop, but also reduce the flow leakage even for the lower pressure drops tested in this investigation and are less sensitive to disk speed changes than a single brush seal.

Different Seal Vendor

Figure 9 gives the leakage performance results for a brush seal manufactured by EG&G Sealol. It had dimensional parameter values approximately the same as those of the baseline configuration. The resultant hysteresis curve is similar to that of Configurations #1 and #3 in Figure 7. The leakage level is between that for these two configurations, but closer to the lower results of Configuration #3. Thus, this brush seal from the second vendor yields comparable leakage characteristics as those of the baseline configurations.

Backplate Gap

Figure 10 gives the leakage performance results for Configuration #4 that had a backplate gap of 0.020 in. compared to 0.030 in. for the baseline configuration. The resultant flow factor level is within the range of the other seals. The hysteresis curve in Figure 10 has a very small difference between the upper and lower portions, but this seal in other testing reacted to disk speed changes similarly as the baseline one. Thus, lowering the backplate gap to 0.020 in. has little effect on the leakage flow characteristics.

To evaluate further the effect of backplate gap, a special series of tests were run at Cross Manufacturing facilities on brush seal Configuration #10. This configuration is the same as #1 and was to be rig tested later. The purpose of these tests was to quantify the relationship between maximum recommended pressure drop across a single brush seal and the backplate gap as the gap is increased above the nominal 0.030 in. Although Cross's rotating seal test rig operates with ambient temperature air and ambient downstream pressure, the results are representative of engine applications. The test approach was to evaluate the leakage
performance for several backplate gaps of the same seal by successively removing material from the backplate inner surface and measuring performance. The backplate machining was done without affecting the bristles.

Figure 11 is a summary plot of the data obtained by Cross. It is a cross plot of the individual performance data curves and gives flow factor versus backplate gap for six pressure drop values. The smallest gap (0.032 in.) represents the nominal value for the baseline configuration. The data for this gap were obtained before removing any backplate material. The increase in flow factor with increasing pressure drop at the lower gap values is the expected effect of pressure ratio across the seal on leakage. For backplate gap values up to 0.055 in., the flow factor is essentially constant for a given pressure drop. As the gap increases above 0.055 in., the flow factor becomes a function of gap and a stronger function of pressure drop. Thus, at backplate gap of 0.055 in., the bristles begin to deflect under the backplate and the effective flow area increases. The amount of bristle deflection depends upon the pressure drop, but some increase in flow factor is noted even for the 30 psid data. The maximum recommended backplate gap for the bristle thickness and packing density considered in this investigation is then 0.055 in.

Performance Comparisons

Table 2 is a summary of leakage flow results obtained in this investigation. This table gives the available seal flow factor data measured at the end of the engine simulation test and at three pressure ratios in the performance testing. Also included are data for the labyrinth seal and results from the Cross testing with variable backplate gap. In the performance testing portion of the data in Table 2, the flow factor value listed is a mean value from the upper and lower portions of the hysteresis curve at a given pressure ratio.

Observations that can be drawn from the data comparison in Table 2 are:
- The performance testing gave a lower flow factor than that at the end of the simulation testing (i.e., pressure ratio near 2.3). This is due to differences in the pressure/speed history before acquiring the data points. Toward the end of the simulation testing, the bristles have come to an equilibrium point but are still not as close to the disk surface as they would be in the performance testing where the pressure drop has been decreased until the bristles are in a relaxed position. Thus, in engine applications the actual leakage may not be as low as measured in performance testing.

### Table 2  Summary of leakage flow results for the various seals tested.

<table>
<thead>
<tr>
<th>CONFIG #</th>
<th>PARAMETER VARIED</th>
<th>BACKPLATE GAP (in)</th>
<th>TEST:</th>
<th>FLOW FACTOR - W/TPU</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>SIMULATION</td>
<td>PERFORMANCE</td>
</tr>
<tr>
<td>1</td>
<td>Baseline</td>
<td>0.030</td>
<td>600</td>
<td>0.0075 0.0061 0.0063 0.0062</td>
</tr>
<tr>
<td>2</td>
<td>Baseline at lower Temp.</td>
<td>0.030</td>
<td>400</td>
<td>0.0058 0.0060 0.0055</td>
</tr>
<tr>
<td>3</td>
<td>Initial interfer. = 0.001 in.</td>
<td>0.030</td>
<td>600</td>
<td>0.0066 0.0034 0.0035 0.0035</td>
</tr>
<tr>
<td>4</td>
<td>Backplate gap = 0.020 in</td>
<td>0.020</td>
<td>600</td>
<td>0.0057 0.0040 0.0044 0.0043</td>
</tr>
<tr>
<td>5</td>
<td>Repeat of baseline</td>
<td>0.030</td>
<td>600</td>
<td>0.0073</td>
</tr>
<tr>
<td>6</td>
<td>Two seals in series</td>
<td>0.030</td>
<td>600</td>
<td>0.0036 0.0027 0.0029 0.0030</td>
</tr>
<tr>
<td>7</td>
<td>No I.D. grinding/shaft coating</td>
<td>0.030</td>
<td>600</td>
<td>0.0064 0.0033 0.0039 0.0040</td>
</tr>
<tr>
<td>A1</td>
<td>EG&amp;G Sealol seal</td>
<td>0.030</td>
<td>600</td>
<td>0.0064 0.0036 0.0041 0.0041</td>
</tr>
<tr>
<td>Ref</td>
<td>Four-knife labyrinth seal</td>
<td>N/A</td>
<td>600</td>
<td>0.0234 0.0245</td>
</tr>
<tr>
<td>10</td>
<td>Baseline seal tested at Cross</td>
<td>0.032</td>
<td>70</td>
<td>0.0048 0.0052</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.045</td>
<td>70</td>
<td>0.0049 0.0052</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.055</td>
<td>70</td>
<td>0.0051 0.0055</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.065</td>
<td>70</td>
<td>0.0058 0.0070</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.075</td>
<td>70</td>
<td>0.0062 0.0100</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.085</td>
<td>70</td>
<td>0.0088 0.0168</td>
</tr>
</tbody>
</table>

* Flow factor for the Performance data are mean values of the upper and lower parts of the hysteresis curves at the given pressure ratios.
- The leakage flow factor for brush seals with baseline parameter values is between 0.0035 and 0.0060. The actual baseline configuration (#1) had a flow factor at the higher end of this range, but the other seals indicate that a more representative baseline flow factor is closer to 0.0040.
- Testing the baseline configuration at a test temperature near 400 °F instead of 600 °F (Configuration #2 versus #1) had only a small effect on leakage flow factor.
- Decreasing the initial interference from a baseline value of 0.005 to 0.001 in. (Configuration #3 versus #1) decreased the leakage, but the decrease is small considering the results from the other seals tested with near baseline parameter values.
- The repeat simulation test of the baseline configuration (Configuration #5) compared very closely with the baseline itself (Configuration #1); thus, proving brush seal and rig repeatability.
- Doubling the number of brush seals (Configuration #8) yielded the lowest leakage performance (flow factor less than 0.003).
- Eliminating the final bristle grinding operation and running on an uncoated shaft (Configuration #9) yielded flow performance close to the low end of the baseline range.

CONCLUSIONS

The results from the brush seal experimental investigation further demonstrate the potential of replacing labyrinth seal with brush seals, especially for subsonic, limited-life engine applications. The decreased leakage using brush seals would allow a better control of the cooling and leakage air flow rates that would decrease parasitic leakages and result in improved engine performance.

Specifically, the results of this investigation have shown that:
- brush seals have a factor of four to seven less leakage than labyrinth seals
- performance characteristics, i.e., flow factor variations with pressure ratio across that seal, are a representative indication of the flow performance of brush seals, but actual flow leakage achieved in applications can be higher because of the pressure drop/shaft speed history
- brush seal leakage performance follows a hysteresis curve with the difference between upper and lower curve portions depending in part upon the disk speed excursions during the test
- changes in the initial bristle/disk interference for modest interference levels have marginal effect on the leakage characteristics
- multiple brush seal in series not only permit a higher pressure drop across the seals but also reduce the leakage level and sensitivity to disk speed changes
- the maximum recommended backplate gap for the nominal bristle packing thickness and density is 0.055 in.
- performance characteristics of brush seals manufactured by two different vendors were similar.

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