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Evaluation of Brush Seals for Limited-Life Engines

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Brush seals are a relatively new concept for replacing labyrinth seals in gas turbine engines. An evaluation was performed to assess the potential of brush seals for limited-life gas turbine engines. A rotating rig was designed and built to test labyrinth and brush seals over a range of simulated engine conditions. An initial set of brush seals was rig-tested to determine leakage and wear performance and identify potential optimum configurations. The measured results showed that brush seals offer significant improvements over labyrinth seals with a factor of three or more reduction in leakage flow. Brush seals exhibit an initial wear-in period but retain significantly reduced leakage over labyrinth seals for times exceeding most limited-life engine applications. Consequently, brush seals offer the potential to precisely meter cooling/leakage air, thereby decreasing parasitic leakage and improving fuel consumption and thrust. Thus, brush seals are a definite candidate for replacing labyrinth seals in gas turbine engines.

Nomenclature

CL	= labyrinth seal radial clearance, in.
D	= disk outer diameter, in.
N	= disk rotational speed, rpm
Pd	= exit air pressure, psia
Pu, Pin	= inlet air pressure, psia
T, Ta	= inlet air temperature, °F or °R
Tm	= seal material temperature, °F
V	= disk surface speed, ft/s
W	= air flow rate, lb/s
Δp	= pressure drop across seal, psid
ϕ	= flow factor, $W\sqrt{T}/Pu$

Introduction

AIR breathing propulsion systems have been an integral part of technology since the first man-controlled powered flight made by the Wright brothers in December 1903. As propulsion systems progressed from reciprocating propeller-driven systems to the first flight of a gas turbine engine on August 27, 1939, performance improvements have continued at an amazing rate. Gas turbine seal approaches have not kept pace with the improvements of aerodynamics or material processing. Losses due to internal-flow systems in small turbine engines can account for up to a 17% loss in power and over a 7% increase in specific fuel consumption (SFC).[1] Changes and improvements in sealing technology have been made, including a rush of government-sponsored optimization during the oil crisis of the 1970s. Nevertheless, the concept of using labyrinth knife-edged seals for air-to-flow sealing continues to be prevalent.

Brush seals are a relatively new, low-leakage sealing concept. They are potential replacements for some or most of the labyrinth seals in a gas turbine engine. Figure 1 shows a schematic diagram of a brush seal. These seals consist of finely packed, inward-pointing metallic or nonmetallic fibers held together by an inner and outer packing washer. The wire fibers are usually made of a metallic alloy with a 0.003-in. diam. The bristles are angled in the direction of rotation to reduce the friction and resulting bristle wear, and run against the rotating or stationary shaft. Figure 2 shows photographs of a brush seal viewed from the side and into the bristles ends.

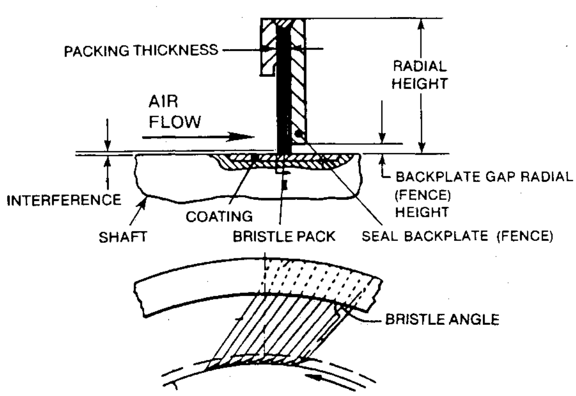

Fig. 1 Schematic diagram of a brush seal.

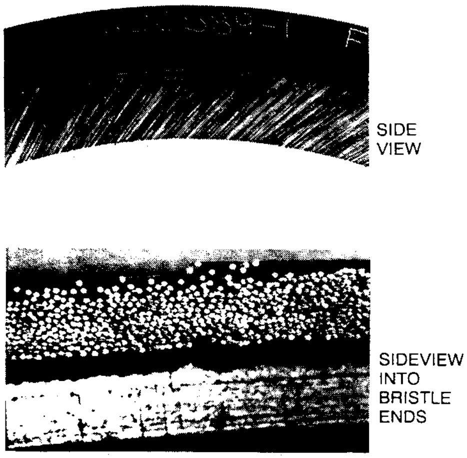

Fig. 2 Photographs of a brush seal.

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The purpose of this article is to describe a project which has been conducted to evaluate replacing labyrinth seals with brush seals in limited-life engines. The project involved developing a rotating bench rig to test brush seals and obtaining initial data to determine seal performance, reliability, and wear characteristics before testing in technology demonstrator engines.

Background

Modern aircraft gas turbine engines have many sealing locations: 1) between the stationary and rotating parts of the bearings, compressor, and turbine; 2) over the blades of the compressor and turbine; 3) between stationary components; and 4) throughout the internal cooling flowpath. Large turbofan engines may have well over 50 sealing locations. As gas turbine engines increase cycle pressure ratio and operate at higher temperatures, gas path sealing starts playing a more important part in overall engine performance and efficiency. Continued use and wear of these engines has a deleterious effect on sealing due to erosion and wear. Large turbofan engines, for instance, have an average increase in SFC of over 1% per year. The results in a small engine are even greater.

Labyrinth seals have been the standard air-to-air sealing technique for over 50 yr. Labyrinth seals create a flow restriction which depends primarily on the magnitude of radial clearance, between the rotating and stationary parts, and the number of knife teeth. The inherent clearance, coupled with a marked degradation of the seal over time due to shaft excursions and thermal growth, results in a significant performance loss. Future advanced propulsion systems must make more effective use of cooling air to improve SFC and power. Brush seals offer the potential of reduced leakage, exactly metering internal flow cooling, and improved control of engine rotor thrust loads. Performance improvements are very mission-dependent and engine-specific. However, these improvements in small engines can be as much as a 7% decrease in SFC and up to 17% increase in power.

Replacing labyrinth seals with brush seals is a relatively new concept. The first U.S. engine test of a brush seal-type concept was in the GE J-47 in 1955. This test was unsuccessful, and the idea of using a compliant, interference seal was abandoned for nearly 30 yr. The first successful engine test of a brush seal was in the Rolls Royce RB-199 in 1983. The engine was qualified with brush seals but they were subsequently removed. Since then, brush seal technology has progressed slowly with few articles published on the subject.[2-4] Currently, brush seals are being used in a few applications, e.g., the IAE V2500 commercial engine. Future plans for several U.S. and European engines, however, include using brush seals throughout the internal flowpath. To aid the development efforts, various investigation projects of brush seals, such as the one described in this article, have been conducted.

Brush Seal Evaluation Project

The subject brush seal project was initiated in September 1988. The overall objectives were to evaluate performance improvements, wear characteristics, and cost impacts associated with replacing labyrinth seals with brush seals, and to determine optimum brush seal configurations for selected potential applications. This project included a study to determine potential locations and corresponding conditions where brush seals could be applied in limited-life engines, design and procurement of a rotating rig to test brush seals, and the testing of several candidate brush seal designs. The rig was designed to have the capability to simulate conditions of both subsonic and supersonic limited-life engines. The testing in this project, however, only included simulations of subsonic applications.

This project follows a cost and time savings approach to maturing a technology such as brush seals, i.e., bench testing, which later transitions to engine demonstrations.

Engine Study

In the engine study effort, several limited-life engines were considered. These engines are representative, state-of-the-art, limited-life engines. The results of the study are given in Fig. 3. This figure shows a cross section of a limited-life engine with seven possible brush seal locations and corresponding engine parameter ranges. The temperature ranges shown in this figure are fairly large, i.e., the lower temperatures corresponding to subsonic applications and the upper ones to supersonic. Two prime locations were identified in the study as the best places to initially apply brush seals. These locations were along the engine shaft at the compressor backface and the turbine frontface, i.e., "B" and "C" in Fig. 3.

The engine study efforts also included choosing a representative engine cycle to follow in part of the seal testing. The cycle chosen consisted of an initial maximum power condition (100% engine speed) for 10 min, corresponding to launch at altitude, followed by a cruise power condition (85% engine speed) for 35 min, corresponding to cruise operation at sea level.

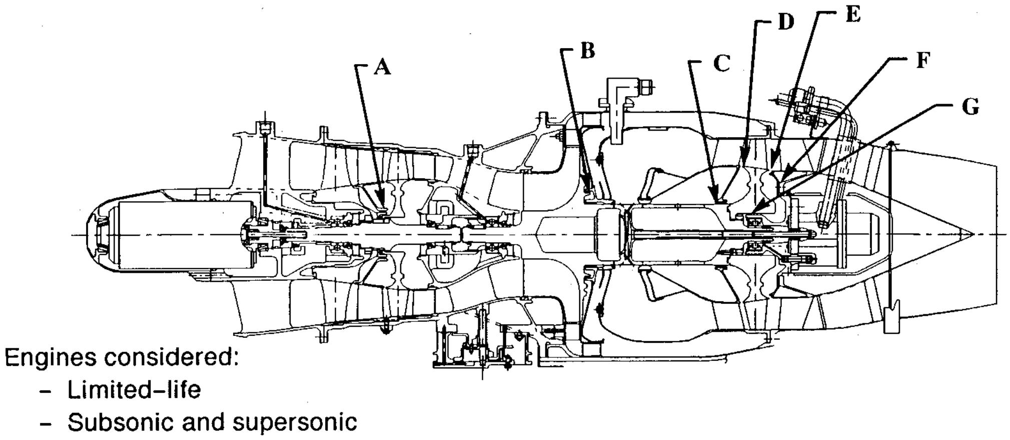
Rig Design

Figure 4 is a cross section of the brush seal test rig designed and built in this project. The rig is driven by a facility air turbine which rotates the rig up to 40,000 rpm. This equates to a surface speed of nearly 900 ft/s for the 5.10-in.-diam test disk which approximately models maximum engine conditions determined in the engine study (see Fig. 3). The rig has four air supplies, one for the heated test air and the other three to provide cooling and buffering air to protect the bearings and static structure. The bearings are lubricated and cooled by a facility powered circulating lubrication system. The housing walls are insulated and cooled by convection to the ambient room air. A bypass valve located at the bottom of the rig allows extra air to flow through the rig while data are not being acquired to speed rig heat-up and to help maintain operating temperature. The rig is designed to operate at a maximum temperature of 1500°F and maximum pressures between 65–200 psia depending upon the temperature level. The axial location of the seal holder is adjustable so that up to four different individual brush seals can be tested with a given disk. This is a cost savings feature in that several individual brush seals can be tested with a given disk surface coating. The holder is designed to accommodate up to three brush seals installed at one time running against the relatively wide disk so that multiple brush seals in series can be tested. A labyrinth seal ring which fit into the rig was also fabricated so that flow performance comparisons between labyrinth and brush seals could be made.

The rig contains instrumentation for measuring several parameters including: 1) shaft rotational speed, 2) inlet and exit pressures, 3) flow rate, 4) air temperatures, and 5) various rig monitoring temperatures, pressures, and vibrations. Key parameters had multiple instrument probes so that more accurate, representative data could be acquired. The flow rate is calculated from pressure drop and temperature measurements across an upstream orifice. The rig data acquisition system allows real-time continuous display of the parameters on a monitor and simultaneous storage on disk.

Test Approach

As part of the brush seal project, an initial test program was conducted using the newly developed rotating brush seal rig. The purpose of the initial testing was to evaluate current brush seal designs as applied to subsonic, limited-life gas turbine engines. The maximum test temperature was approximately 600°F which encompassed the temperatures of most of the subsonic engines for locations "B" and "C" as defined in Fig. 3. The initial testing included 10 brush seals and one reference labyrinth seal. In choosing a test matrix, various



	A	B	C	D	E	F	G
Ta, F	305	490-920	500-1200	660-1250	1180	1260	1180
Tm, F	305	550-920	535-1220	900-1450	1200	1290	1190
Pin, psia	59	80-300	81-380	35-340	325	275	325
Δ P, psid	19	35-145	42-155	0.4-55	48	100	310
V, ft/sec	720	570-915	570-915	1185-1890	785	1185	500
W, lb/sec	minimal	.064-.140	.039-.780	.039-.780	.735	.735	.152
P/Pin	0.320	0.34-0.48	.110-.570	.0055-.160	.150	.360	.955

Fig. 3 Engine study results.

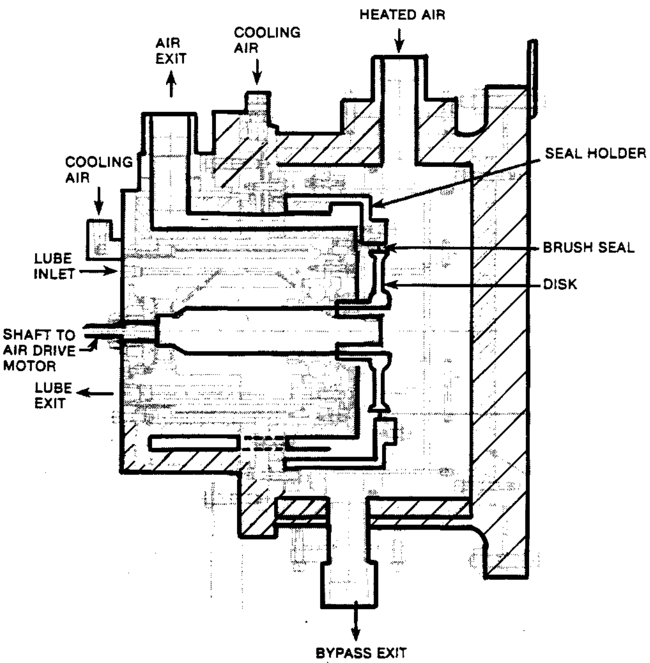


Fig. 4 Brush seal rig cross section.

seal parameters were considered including 1) seal construction variables, i.e., initial seal/disk interference, backplate radial gap (fence) height, radial bristle length, packing thickness, bristle angle, bristle diameter, bristle material, and bristle surface finish operation; 2) disk surface treatment; 3) seal offset; 4) number of seals in series; and 5) operating temperature. The parameters varied in the testing are listed in Table 1 and were selected as an initial set to determine potential optimum configurations. Brush seal configuration no. 1 was the baseline brush seal. This seal was recommended for limited-life applications by personnel from Cross Manufacturing Co. (1938) Ltd. of Devizes, Wilts, England.[5] Cross was the major supplier of brush seals for this project. The other configurations in Table 1 had a variation in one or more of the seal/disk parameters to see the effect. Configuration no. 9 corresponded to a seal with no special shaft surface treatment and no final grinding operation of the bristle inner diameter. This configuration was an attempt to minimize the cost of a brush seal and mating shaft surface preparation for limited-life applications where fabrication costs are important. Configurations no. A1 and no. A2 were supplied for EG&G Sealol Engineering Products Division of Warwick, Rhode Island and Textron Turbo Components of Walled Lake, Michigan, respectively.

The test procedure for the brush seal included 1) an ambient temperature performance check, 2) a cycle simulation, and 3) evaluations of performance and wear. The ambient temperature performance check consisted of measuring the seal flow for fixed upstream and downstream pressures at two moderate rotational speeds, i.e., 15,000 and 25,000 rpm. This was done to insure that the seal was properly installed, but the run times were limited to minimize seal wear. The cycle simulation modeled the cycle established in the engine study and was generally the first test performed after the rig reached the desired temperature level. The cycle simulation consisted of setting the desired pressure conditions at a lower speed (15,000 rpm), accelerating the rig to a designated maximum speed condition (35,000 rpm) and holding that condition for 10 min, decreasing the speed to 85% of the maximum speed (30,000 rpm), resetting the pressure conditions, and holding that condition for 35 min. Subsequently, performance data were acquired by setting the speed at a desired value, lowering the pressure drop across the seal to nearly zero, and taking data points at successively higher pressure drops to a maximum of 115 psid (rig limit), followed by data points at successively lower pressure drops decreasing to nearly zero again. Wear evaluations were made by running the rig for a series of 1-h time intervals at a higher speed (35,000 rpm) and making a single point performance check between intervals. In the initial brush seal test over 14 h of test time were accumulated on the single seal with most of the wear occurring in the first 4 h. For this reason and the fact that 14 h is considerably longer than many limited-life applications, the wear evaluations for subsequent seals were limited to about 4–8 h.

The rig rotational speeds were selected based on matching engine study results and rig vibrational limitations. The maximum speed obtainable with the disks used was 40,000 rpm which corresponded to a disk surface speed of 890 ft/s for the 5.1-in.-diam disk. The highest speed of 35,000 rpm was utilized in most of the testing since the corresponding disk sur-

Table 1 Test matrix for initial brush seal testing

Configuration no.	Initial interf, in.	Fence height, in.	Brushes in series	Surface finish (bristle)	Shaft treatment	Maximum temperature, °F
1	0.005	0.030	1	Ground	Al O	600
2	0.005	0.030	1	Ground	Al O	400
3	0.001	0.030	1	Ground	Al O	600
4	0.005	0.020	1	Ground	Al O	600
5	0.005	0.030	1	Ground	Al O	600
7	0.005	0.030	1	Ground	Chr/Carb	600
8	0.005	0.030	2	Ground	Al O	600
9	0.005	0.030	1	None	None	600
A1	Similar to seal configuration no. 1 except supplied by EG&G Sealol				Al O	600
A2	Similar to seal configuration no. 1 except supplied by Textron				Al O	600
Ref.	Reference 4-knife labyrinth seal with initial clearance of 0.008 in.				Al O	600

Al O = Aluminum oxide (METCO P105-13)
Chr/Carb = Chrome carbide (METCO P81VF-10)
Parameters held constant:

Seal radial height	= 0.450 in.	Bristle diameter	= 0.0028 in.
Seal offset	= 0.000 in.	Disk outer diameter	= 5.100 in.
Bristle radial height	= 0.286 in.	Seal plate material	= Nimonic 75
Bristle angle	= 45 deg	Bristle material	= Haynes 25
Bristle packing thick	= 0.027 in.		

face speed of 780 ft/s encompassed most of the engines studied for locations "B" and "C" in Fig. 3, and this speed gave a margin of safety in operating the rig.

The reference labyrinth seal was tested by acquiring performance data at ambient 400 and 600°F temperatures for various rotational speeds and pressure conditions.

Test Results

In this article, representative results obtained for the seals tested will be presented. Additional results and data analyses will be presented in later publications.

Figure 5 shows measured performance test data for the reference labyrinth seal at 30,000 rpm and 600°F compared to calculated results. The y-axis parameter is flow factor and on the x axis it is a pressure parameter which approximately linearizes the data. This pressure parameter is derived from porous-wall flow equations[6] ignoring the laminar flow terms. The seal-calculated results were obtained using a labyrinth seal flow code[7] assuming radial clearances of 0.005–0.007 in. The labyrinth seal had a cold-build clearance of about 0.008 in. At 30,000 rpm, the radial clearance would have decreased to about 0.006 in. based on the rig structural analysis. The agreement between the measured and calculated results in Fig. 5 validates this measurement approach.

Figure 6 is a plot of flow factor and disk rpm vs time obtained in the engine simulation testing of configuration no. 1 brush seal with an air temperature ranging between 550–600°F. The points shown are quasisteady state points. The line plotted show approximately how the flow and speed varied between points. The flow factor varies during the mission simulation as follows:

1) During the first 10 min the flow factor drops suddenly as the speed is increased to 35,000 and the disk grows in diameter because the bristles are held in place by the pressure drop across the seal; flow factor then increases slowly during the 10 min as the seal bristles slowly adjust with the pressure drop maintained across the seal.

2) During the subsequent 35 min the flow factor initially jumps up as the disk diameter decreases with the sudden decrease in speed and the bristles do not have time to react with the pressure drop maintained; the flow gradually decreases as the bristles respond and eventually reaches a steady-state level (in about 7 min).

These flow factor variations demonstrate that brush seal flow leakage will react to speed changes that are normal in

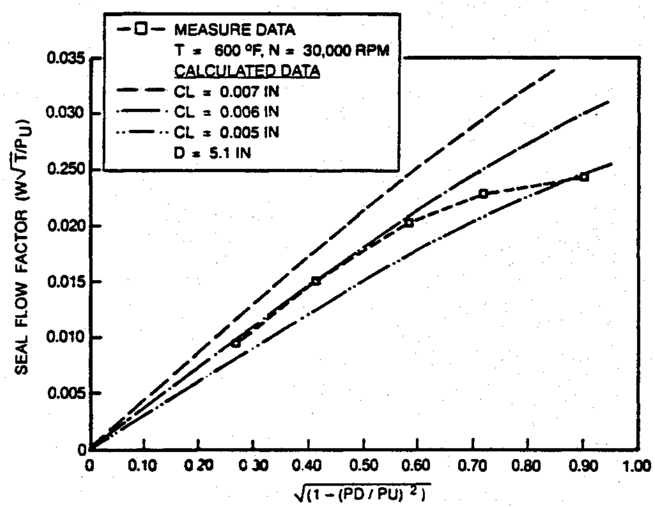


Fig. 5 Comparison of measured and calculated flow performance for the labyrinth seal.

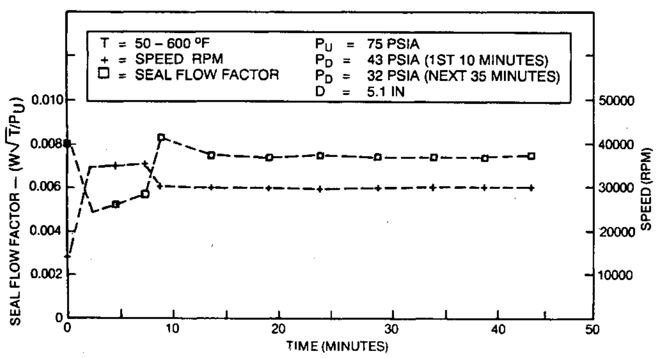


Fig. 6 Engine simulation cycle results for brush seal configuration no. 1.

engine operation, but maximum brush seal leakage is still much lower than that of a labyrinth seal.

Figure 7 is a performance plot obtained for the configuration no. 1 brush seal at approximately 30,000 rpm and 500–600°F air temperature. The points shown are stabilized ones taken 3–6 min after establishing a pressure drop condition. The first data point obtained was for a very low pressure drop

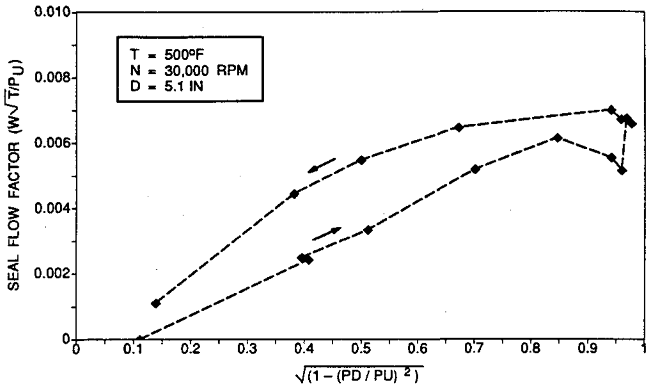


Fig. 7 Measured flow performance for brush seal configuration no. 1.

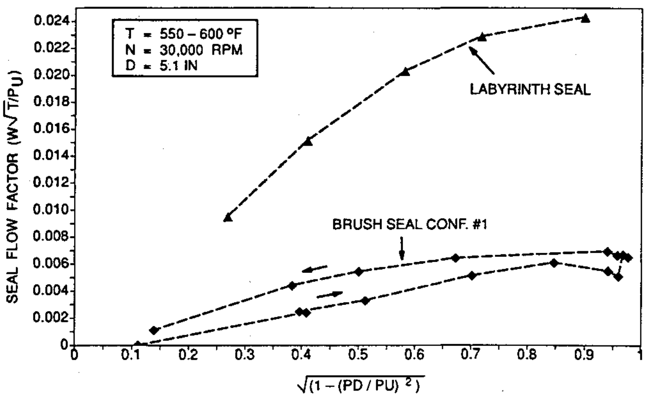


Fig. 8 Comparison of flow performance results for the labyrinth seal with brush seal configuration no. 1.

(on the left of the lower curve in Fig. 7). This was done to insure that the bristles are in a relaxed position. In setting off this point the flow factor could be seen to suddenly decrease as the pressure drop was lowered to nearly zero. As the pressure drop was increased for subsequent data points the flow factor followed the lower curve. Then the pressure drop was sequentially decreased from the maximum value and the flow factor followed the upper curve. The resulting plot in Fig. 7 is a classical "hysteresis" curve. The difference between the upper and lower portions of the curve is not due to seal wear, because the flow factor would eventually drop to its initial value as the pressure drop was decreased to nearly zero again. Some of the flow variation between increasing and decreasing pressure drop is due to rig speed variations between points. As the pressure drop is varied the rig bearing load changes, this affects the speed of the rig that is being driven by an air turbine. The air pressure to the turbine is adjusted as the pressures are changed, but speed fluctuations up to a 1000 rpm are common. The effect of rig speed variations is minimized by not taking data points until the seal flow factor appeared to stabilize (after 3–6 min). Although there is quite a bit of variation in the data from increasing and decreasing pressure ratios for the configuration no. 1 brush seal, the comparison in Fig. 8 shows that the flow factor level for the brush seal is less than one-third that of the reference labyrinth seal.

The brush seals tested exhibited a two-phase wear-in period. The initial phase lasted for less than 30 min and was observed not such much from the flow characteristics, but from the heat-up of the brush seal outer ring and seal holder as indicated by rig thermocouples. The seal ring and holder would heat up 200–300°F above the seal air temperature during the initial testing of each seal (usually during the brief ambient temperature performance check), but the heating up would disappear within 30 min and not reoccur during the remainder of that seal's test. Qualitatively, it appeared that the flow factor did not vary significantly during the initial wear-in period.

The brush seals tested showed differing wear characteristics in the second wear-in phase. Figure 9 is a plot of flow factor as a function of accumulated run-time for two brush seal configurations for a given set of pressure and speed conditions. The data points plotted were specifically chosen from the recorded data so that they occurred after the seal pressure drop had been reduced to nearly zero and then increased to about 50 psia, i.e., the lower part of the performance curve (Fig. 7). This was done to insure that consistent results could be obtained. The data for the configuration no. 1 seal reveals that most of its wear occurred in the first 4 h and the total increase in flow factor was about 45%. The data for configuration no. 3 in Fig. 9 shows that its leakage changed very little over the 8 h of testing. Configuration no. 3 seal is the same as configuration no. 1 except that its initial interference was 0.001 in. instead of 0.005 in. The smaller initial interference would explain the less wear, but not the lower level of leakage. Such a difference is more likely due to seal-to-seal variations since the wear characteristics of the other brush seals more closely matched that of configuration no. 3. Even though the wear characteristics of the seals differed, the leakages after the seals wore were still considerably below that of the reference labyrinth seal. For example, the leakage rate of configuration no. 1 at the end of testing, was less than one-third (about 30%) of that of the labyrinth seal, and configuration no. 3 was about one-sixth.

The lack of wear of brush seal configuration no. 3 can be shown at other pressure drop conditions in the performance plot given in Fig. 10. The hysteresis curve in this figure is for the seal after it had been run for about 3.6 h. The other data are for the seal after an additional 3.9 h of running. The latter data were for an abbreviated performance check, but demonstrate that the additional 3.9 h of operation had no apparent effect on seal performance over the whole pressure ratio range.

An additional observation can be made relative to the data shown in Figs. 7 and 10. This observation is that the 0.030-

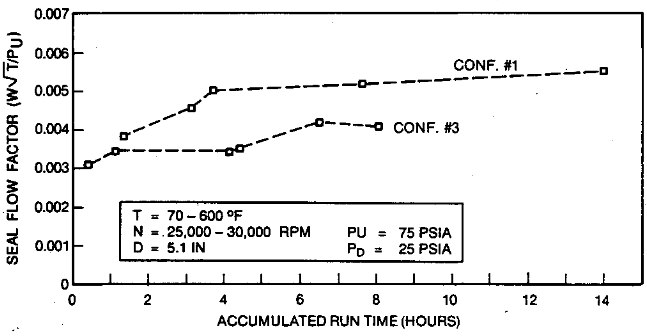


Fig. 9 Comparison of the effect of wear on flow performance for brush seal configurations nos. 1 and 3.

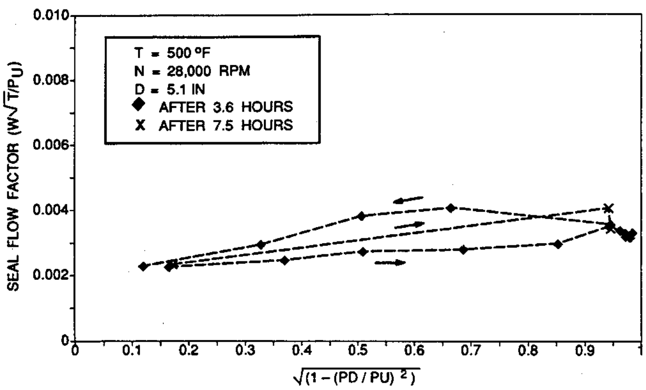


Fig. 10 Measured flow performance for brush seal configuration no. 3 after 3.6 and 7.5 h of testing.

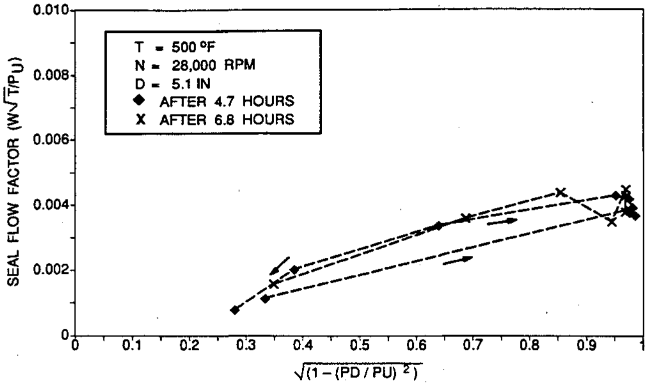


Fig. 11 Measured flow performance for brush seal configuration no. 9 after 4.7 and 6.8 h of testing.

in. gap seal backplate in configurations nos. 1 and 3 allows pressure drops up to 115 psid with little change in flow characteristics. Thus, a single brush seal can provide low leakage even for fairly high pressure drop applications with a backplate gap sufficiently large to allow shaft excursions of over 0.025 in. Such excursions are common for hard landings or quick accelerations.

Another important set of results was obtained for brush seal configuration no. 9, which did not have the final grinding operation of the bristle diameter nor a disk surface coating. Figure 11 is a performance plot for that configuration with two sets of data: 1) one after nearly 5 h of testing; and 2) the other 2 h later. These data reveal that this configuration sealed and wore as well as the other ones that had the additional fabrication steps, i.e., final seal grinding operation and disk surface treatment.

Conclusions

The results presented demonstrate that the rotating seal rig developed in this project is a viable tool for evaluating brush seals in simulated conditions of actual subsonic limited-life engines. Specific conclusions from the data are as follows:

1) Brush seals offer a significant improvement over conventional labyrinth seals with a factor of 3 and more reduction of leakage flow.

2) Brush seals have an initial wear-in period during which the leakage rate can vary, but they retain a significantly reduced leakage over labyrinth seals for time periods corresponding to limited-life engines and probably much longer.

3) Brush seal leakage performance follows a hysteresis curve and leakage can vary significantly for brief periods of time as the seal reacts to engine speed changes, but the leakage rate is still significantly less than that for labyrinth seals.

4) Brush seals can survive shaft excursions of over 0.025 in., common for hard landings (primarily for man-rated applications) or quick accelerations, without any performance loss (such excursions would cause excessive wear for labyrinth seals).

5) Steps examined to reduce the cost of brush seals did not have a detrimental effect on sealing performance for running times representative of limited-life engines.

These rig results show that brush seals have a definite potential for replacing labyrinth seals in gas turbine engines. The improved continuous leakage restriction with brush seals vs labyrinth seals would allow the exact metering of cooling/leakage air, resulting in a dramatic decrease of parasitic leakage and the resulting performance loss. Additional efforts must be made, however, to assess and reduce acquisition costs if brush seals are to be a viable replacement for labyrinth seals in lower cost, limited-life engines.

Acknowledgments

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