Counterrotating Intershaft Seals for Advanced Engines

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Feasibility for application of a counterrotating intershaft carbon seal to high thrust-to-weight military gas-turbine engines was demonstrated through a series of rig tests. The noncontacting high-speed (800 ft/s) seal incorporating hydrodynamic lift geometry with spiral grooves in the seal plates similar to the one designed by NASA-Lewis Research Center.[1] Seal air leakage and carbon wear rates were determined for each of two configurations in rig screening evaluations of 10-h duration. One seal assembly was subjected to an additional 50-h endurance testing. For both configurations, seal air leakage was approximately one-third the leakage that would occur with a conventional labyrinth seal. The wear rate was low for the configuration tested a total of 60 h.

Introduction

TO meet the thrust-to-weight ratio and cost of ownership goals of advanced engines, rotor systems technology development has been directed toward counterrotating rotors with a straddle-mounted high rotor coupled to the low rotor through an intershaft bearing, as shown in Fig. 1. Rotor speeds in advanced engines will be of the order of 20% higher than those of current state-of-the-art production military engines. The counterrotating shafts, coupled with the high rotational speed of the shafts, presents a potential problem in effective bearing compartment sealing. Candidate seal configurations were investigated in this program for the critical sealing portion of an intershaft bearing compartment.

Design studies have shown that a noncontracting carbon seal has the advantages of reduced leakage through a counterrotating intershaft bearing compartment, the possible elimination of the breather pipe system, a smaller capacity de-oiler, reduced compartment seal leakage, lower oil consumption, and weight reduction. The seal concept must operate at advanced engine conditions at a relative surface speed between the carbon and the seal plates in the range of 800 ft/s. The trend of increasing seal rubbing velocities can be seen in Fig. 2.

The intent of this program was to demonstrate, by testing, a low-leakage, low-wear intershaft seal for advanced engine application where breather flows are to be minimized. Seal design and endurance testing was sponsored and monitored by the Naval Air Propulsion Center. Fabrication and screening tests were funded by Pratt & Whitney Aircraft.

Approach

Due to the stringent operating requirements to be imposed upon the intershaft seal, i.e., surface speeds twice the current state-of-the-art, it was necessary to conduct a preliminary seal design study to define a number of candidate concepts. Six preliminary concepts were generated, and these were reduced to two viable concepts. Final design evaluation selected a piston-ring carbon seal trapped between two spiral grooved steel runners. A variation in the sealing of the butt gap of the carbon seal provided an alternate configuration. Seal components were fabricated by an established manufacturer of high-speed circumferential/face type seals. Screening test programs of 10-h duration determined leakage and short-term wear data. A 50-h endurance program provided additional wear data. All data were reviewed, and the design system was upgraded for the next generation of seal designs.

Design Studies

A preliminary design study was conducted to evaluate different carbon seal geometries as a replacement for an advanced engine intershaft labyrinth seal. The operating conditions present difficult design requirements from the standpoint of package dimensioning, cooling, and shaft relative velocities. Three different sealing concepts were investigated. These included 1) wear-tolerant seals, 2) seals attached to one shaft using lift-off geometry (Rayleigh pads or spiral grooves) to prevent rubbing on the other shaft, and 3) seals that rotate at an intermediate speed between the two shafts. For the three sealing concepts, several geometries were studied. The study consisted of basic mechanical and analytical evaluation. The mechanical aspects considered included packaging, assembly, and primary loading effects. Analytical evaluation included pressure balance, leakage, and heat generation. The pressure balance evaluation made use of computer programs for rotating seal dam and Rayleigh pad loads.

The configuration selected, shown in Fig. 3, consists of a split carbon ring coupled to the low-rotor seal land by the friction developed through centrifugal loading of the carbon element. The rotational seal is formed between the carbon and two axial face seats, which are located forward and aft of the carbon and which rotate with the high-rotor hub. The seal seats contain seal dams and shallow spiral grooves, which develop hydrodynamic lift to prevent rubbing during relative axial translations of the two rotors.

The primary factors that influenced this design are high operating speed and small geometric envelope. The engine configuration dictated that the seal be located between two counterrotating shafts. The resulting high relative speeds precluded the use of a conventional contacting seal. The geometric constraints prevented the use of typical face seal accessories such as springs, guides, and oil cooling holes. These factors necessitated a unique seal configuration.

The spiral grooves were designed following the NASA configuration described in Ref. 1. The analysis included predicting the spiral groove load capacity curves and optimizing the spiral groove geometry for steady-state conditions. A computer program was used to predict the performance of spiral groove, gas-lubricated (compressible fluid) face seals for either inward or outward pumping spiral groove configurations. An upstream, inward pumping configuration pumps the fluid radially from the outer region of the seal toward the center. A downstream, outward pumping configuration

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pumps the fluid radially from the inner region of the seal toward the outside. Input variables to the computer program are spiral groove geometric data, misalignment angles, speed, fluid viscosity, temperatures, and pressures. Outputs used were fluid film lift force, film thickness, and pressure distribution.

Spiral grooves were selected because of their uniform circumferential pressure distribution and lift capability. They were located in the hard-faced seat faces to prevent their elimination during rub conditions with the carbon seal ring. Two seal ring configurations were selected.

Configuration 1 (see Fig. 3) is a conventional straight butt gap cut design with pressure balance featured on the OD to reduce net normal load transfer to the OD seal land. This geometry is easily manufactured but has two primary problems: large end gap at operating temperatures due to alpha mismatch, and lack of ring rigidity. Full ring carbons were considered to eliminate these problems but were found not feasible due to the low thermal expansion coefficient of the carbon. The carbon works loose from the OD seal land as a result of rotational and thermal effects. Attempts to incorporate a low alpha OD seal land material were critical to this type of design, however such a material that could withstand the stresses was not available, and hence this approach was abandoned. Configuration 2 utilizes a basic split ring but adds a sealing plate at the gap in order to close off the leakage path and provide alignment between the two ends of the ring. This configuration closely approximates that of a full ring.

The bore of the seal ring is grooved in the center, and the axial ring thickness is minimized to reduce the mass and the resulting centrifugal normal loading of the element. The radial height was maximized to the limits of the geometric envelope, which provided a maximum lift/weight ratio for a fixed carbon ring axial thickness.

In the preliminary design, pressure balance features were incorporated in the forward and aft axial seats. These provided high-pressure air to an OD seal dam in the aft seat to minimize net pressure forces. Because of the porting configuration, simultaneous low clearance gaps had to be maintained at each axial face. Lack of wear tolerance and manufacturing limitations precluded the use of these features. In the final design, an inward pumping groove is incorporated in the forward seat and an outward pumping groove in the aft seat. These take advantage of the high-pressure supply air to maximize lift during operation. The ring is free to assume an equilibrium position between the two axial seats.

Figure 4 shows the seal hardware components and their interrelationship to the configuration.

Materials Selection

For the seal ring, existing carbon graphite compounds were selected because of

1) Low density—results in low normal load transfer to OD seal land.
2) Low friction coefficient—this, combined with low density, mentioned above, results in low sliding force (against steel) requirements for a given ring size.
3) Good wear characteristics—based on face seal experience.
4) High-temperature oxidation resistance.
5) Availability.

Material for the seal land and seat components was selected to ensure compatibility with mating hardware. Chrome carbide plasma sprays have been specific on potential wear surfaces.

Test Facility Description

All testing was conducted at the Orenda Division of Hawker Siddeley Canada, Toronto, Ontario. Orenda has a twin-rotor

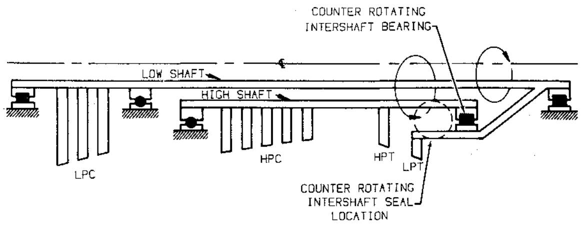

Fig. 1 High-speed rotor system.

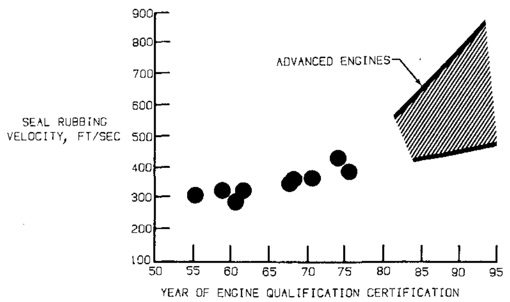

Fig. 2 Mainshaft seal speeds.

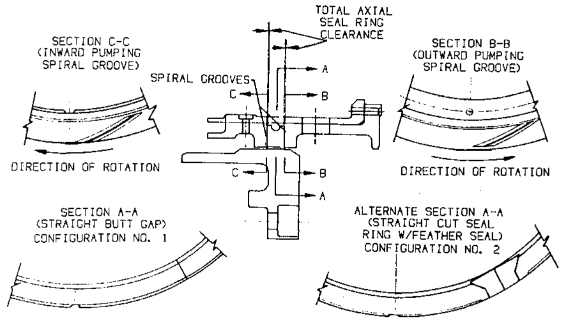

Fig. 3 Counterrotating intershaft seal design.

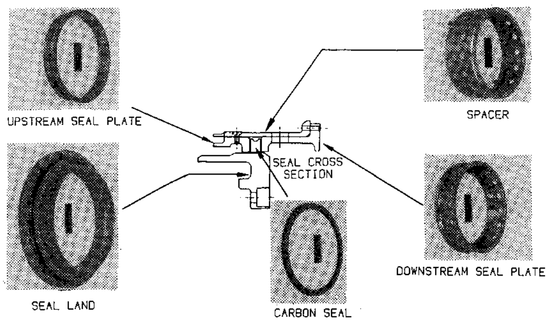

Fig. 4 Intershaft seal hardware.

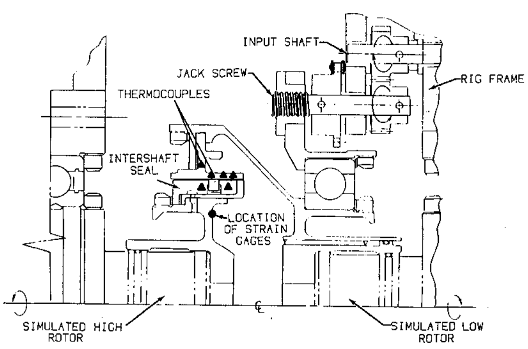

Fig. 5 Intershaft seal installed in test rig.

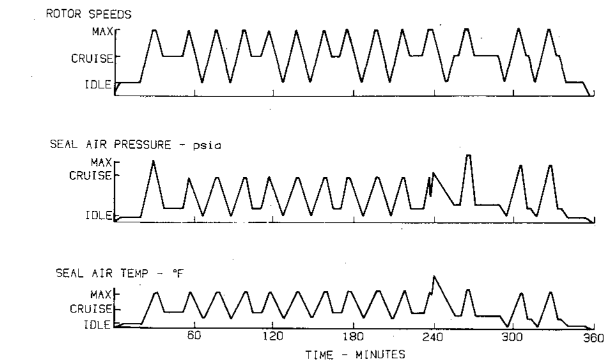

Fig. 6 Counterrotating intershaft seal endurance test run program.

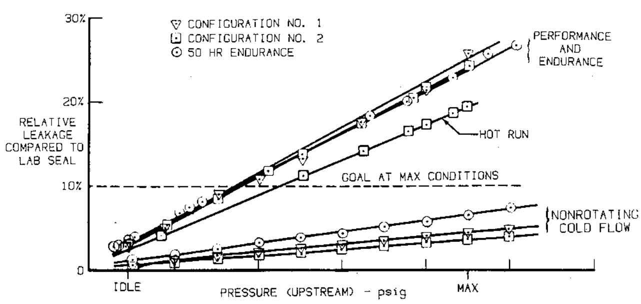

Fig. 7 Leakage vs upstream pressure.

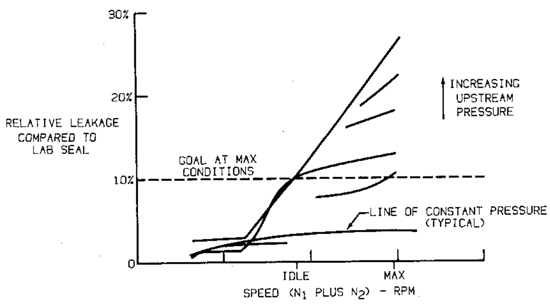

Fig. 8 Relative leakage vs combined speed—configuration 1

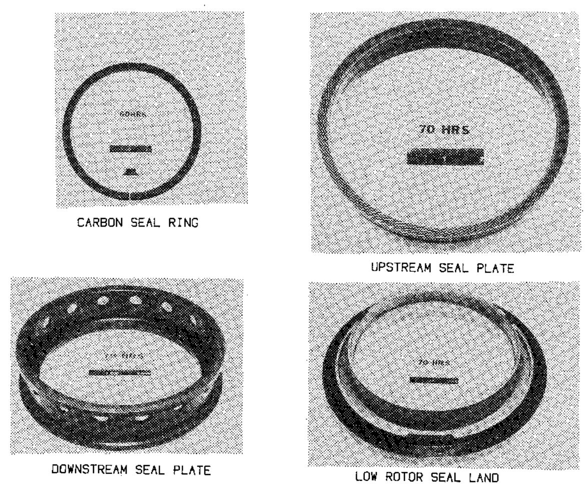

Fig. 9 Seal components after 50-h endurance test.

seal rig that was modified to accept the intershaft seal hardware, simulate relative axial thermal growth of engine rotors, operate at advanced engine conditions, and provide rotating instrumentation. The rig test section and adaptive hardware are shown in Fig. 5.

The rig has two shaft assemblies on which separate mechanical seals or a combined seal may be mounted. In some test arrangements the seal is mounted on a support assembly held between the two shaft assemblies with a seal runner plate carried on and driven by each shaft assembly. Each shaft is driven by its own 40-hp variable-speed electric motor and can be rotated in either direction at speeds up to 17,000 rpm.

Each shaft is mounted on three precision ball bearings, two of the bearings paired in tandem to accommodate the seal thrust loads, the third bearing serving to steady the shaft and to control the end float of the shaft assembly. A toothed pulley located between the single bearing and the tandem pair is driven through a 3-in. wide timing belt by the variables speed electric motor.

The rig is composed of two halves, each of which houses one complete rotor system. The two halves are mounted on a lathe bed to permit easy separation for inspection or disassembly. The process of rig separation and removal of a support assembly mounted seal specimen for examination can normally be performed in about 2-3 h, including cool-off time.

The rig is provided with its own lubrication supply system and sump ventilation system and is capable of supplying cooling oil to seal runner plates at a controlled temperature.

The rig configuration is such that heated, pressurized air can be supplied to the space inside the seal. External instrumentation facilities provide accurate control, measuring, and recording of the quantity of air supplied to the seal, as well as its pressure and temperature.

A seven-channel Lebow slip ring was used for the rotating thermocouples and strain gages. Figure 5 shows the locations of rotating instrumentation. Only high-rotor instrumentation was used during the screening and endurance test programs. An additional thermocouple installation available in the low-rotor land was not used during this program.

During the cyclic operation of an engine, the rotor axial thermal growth differs between the high and low rotors. This operational condition was built into the Orenda rig adaptive hardware. In addition, strain gages were installed to measure the amount of force required to move the carbon seal (centrifugally loaded into the bore of the low-rotor land) when it is "pushed" by the hydrodynamic film of the counterrotating high-rotor seal plates. Figure 5 shows the general arrangement of this jackscrew mechanism.

Test Program

The objective of the 10-h screening tests was to determine leakage rates, determine short-term wear rates, and measure actuation forces. The performance test program consisted of a series of speeds from advanced engine idle to full rotor speeds. Seal upstream air temperature and pressures were representative of projected seal environment. Relative axial motion excursions and friction measurements were taken during these tests.

The endurance test program was developed from an advanced tactical engine typical mission and is shown in Fig. 6. Acceleration and deceleration rates were expanded to accommodate the limits of the test facility. Speeds and pressure changes were made simultaneously. Air inlet temperature did lag considerably due to the relatively low seal leakage rate. After each "throttle movement" was made, the seal was actuated to an axial position simulating the relative thermal growth of the engine rotors. The seal system was actuated approximately 300 times during the test. Actuation rates exceeded those that the engine would experience thermally during operation.

At the beginning of each test and at regular intervals during the testing, the gap between the carbon and high-rotor seal plates was measured with feeler gages. This gave an average wear rate and is accurate within 0.0005 in.

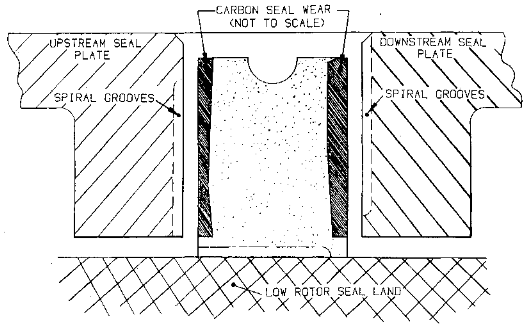

Fig. 10 Carbon and seal plate relative position showing carbon wear area.

Test Results

Leakage

Figure 7 shows the leakage flow data vs upstream pressure for both screening tests and the endurance test. The difference between the nonrotating cold flow data and the rotating performance/endurance test data suggests an internal change in flow area as a function of speed. This may be due in part to relatively loose radial fits on the low- and high-rotor adaptive hardware required to minimize seal plate distortion and, in part, to an air gap that occurs during operation between the carbon seal and the seal plate due to the hydrodynamic film. This operational condition accounts for some of the difference between the rotating and cold flow leakage rates.

It is also noted that the air temperature entering the seal (rotating thermocouple) greatly lagged behind the air temperture entering the rig. The data shown in Fig. 7 as "hot run" were a series of abnormally log static points, where the seal was allowed to reach steady-state temperatures (within the limits of the test program). The figure shows that, as the seal is allowed to heat up, flow area is reduced.

The plot shown in Fig. 8, using data from configuration 1, shows the effect of speed and pressure on leakage. At constant speeds the leakage is approximately proportional to the upstream pressure as expected for sonic flow. Also, at a constant pressure, the leakage increases with speed, suggesting an increase in flow area due to rotation.

The seal rig was pressure-checked prior to the start of the program and at several intervals during testing. Nonrotating cold flows were conducted prior to each test day and at any time the rig was opened up for inspection or hardware changes. There was no measurable rig leakage (adaptive hardware, shafting, and all external plumbing) during the endurance tests.

Friction Force and Wear Measurement

Mechanical relative axial actuation of the low-rotor land with respect to the high-rotor seal plates was conducted from idle to maximum speed conditions. Unexpectedly low frictional forces were measured by strain gages. This was confirmed since during actuation, the amount of physical force required to move the seal was very low. This apparently lower than predicted friction coefficient between the carbon seal, and low-rotor land can be utilized by reducing the lift force required of the hydrodynamic lift geometry.

During the first 5 h of testing on configuration 1, there were events when the carbon contacted the downstream seal plate. This was due to off-design operation or an upstream pressure increasing rate greater than the developing hydrodynamic film rate. These occurrences caused high initial wear on configuration 1. Subsequently the wear rates of the two configurations were similar.

Thermocouples embedded in the seal plates showed a rapid temperature increase, sometimes as high as 700°F. Once the pressure was reduced, the carbon would recenter and seal temperatures would slowly return to the original steady-state level of approximately 250°F. This demonstrates sensitivity to engine pressure transients.

The general condition of the seal hardware after the 50-h endurance test was excellent, as shown in the component photographs of Fig. 9.

Selected dimensional inspection was conducted on all seal components before and after testing. Pretest circumferential traces show very uniform spacing and depth of the spiral grooves on the downstream seal plate. Traces after 70 h of testing show a small amount of carbon deposited on the surface and in the grooves.

A composite of the seal components and the wear pattern on the carbon surface is shown in Fig. 10. The majority of the preferential wear is on the downstream face and does match the coning of the seal plate. There is much less wear on the upstream face, but a two-step wear is noticed. The coned seal plate will wear the inner diameter portion of the carbon face where possibly a cocking effect may cause the outside diameter portion to wear as shown.

Cumulative carbon wear from the endurance test (configuration 2) was 170% of goal wear limit. Advanced seal design concepts will address reduction of wear rate and assess the effect of wear on seal function, if any.

Conclusions and Recommendations

Test results indicate that the counterrotating seal is suitable for engine demonstration in the counterrotating intershaft position of advanced engines. Results have provided valuable insight into the direction of further seal development.

1) Demonstrated 800 ft/s operation at advanced engine conditions.

2) Leakage is one-third of lab seal operating under similar conditions.

3) Carbon seal contact with the high-rotor seal plates and the resulting wear are dependent upon the rate at which upstream pressure is increased. Typical engine rates cause contact. Some pressure cross bleed may be appropriate.

4) The force required to move the carbon axially along the bore of the low-rotor land was an order of magnitude lower than anticipated, suggesting that carbon and land vibrations dramatically decrease the friction coefficient. Less spiral groove lift force may be required.

5) Leakage rates increase with speed at constant pressure, indicating a flow area change such as an interference fit opening between the seal components and the rig hardware.

6) Wear is moderate but the patterns indicate seal plate coning and possible carbon cocking. More attention to component thermal-structural dimensional control is warranted.

7) No thermal or shell analysis was done on the seal component. The wear patterns indicate that some coning occurred. This could be a major cause of leakage. Rig radial fits may have opened up and contributed to the leakage.

Recommendations: 1) Laboratory measurement of carbon-to-steel friction coefficient in a vibratory environment should be made to confirm the low values measured by the test results. 2) Continued seal development and testing should be conducted to address the transient pressure sensitivity, leakage increase with speed, and carbon wear pattern.

Acknowledgments

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References

[1]Dirusso, E., "Design Analysis of a Self-Acting Spiral Groove Ring for Counter-Rotating Shafts," NASA Technical Paper No. 2142, May 1983.