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NEW CIRCUMFERENTIAL SEAL DESIGN CONCEPT USING SELF-ACTING LIFT GEOMETRIES

by William F. Hady and Lawrence P. Ludwig

Lewis Research Center
Cleveland, Ohio 44135

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16. Abstract

Seal operating temperatures, leakage (pressurizing gas flow), torque, and wear of a conventional circumferential shaft seal were measured and compared to those of a conventional seal modified to have self-acting lift geometries. Both seal types had a 6.67-cm (2.625-in.) diameter bore and were operated at a sliding velocity of 46 m/sec (150 ft/sec) with differential pressures ranging from 0 to 70 N/cm^2 (0 to 100 psi). Results of this investigation show that the self-acting seal operated at lower bulk carbon temperatures with half the torque and approximately one-tenth the wear of the conventional seal. Seal leakage of the self-acting seal was of order of 0.18×10^{-4} m^3/sec (0.04 scfm) for pressures above 42 N/cm^2 (60 psi) which is well within the accepted range for gas turbine engine applications.

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Lewis Research Center

SUMMARY

Experimental studies were made to compare seal operating temperatures (due to self-generated heat), leakage (flow of pressurizing gas through the seal), torque, wear, and oil leakage of a conventional circumferential seal to those of a conventional seal modified to have self-acting lift geometries. Both seal types had a 6.67-centimeter (2.625-in.) diameter bore and were operated at a sliding velocity of 46 meters per second (150 ft/sec) with the differential pressure ranging from 0 to 70 newtons per square centimeter (0 to 100 psi). Results of this investigation show that the self-acting seal operated with lower bulk carbon seal ring temperatures than the conventional seal; this lower temperature is related in part to the flow of the pressurizing gas through the seal as well as the amount of coolant oil flow supplied to the inside of the cantilevered seal runner. Measured torque values indicate that the self-acting seal developed approximately half the torque (over the entire differential pressure range of this investigation) of the conventional seal. In the lower pressure range (7 to 28 N/cm^2 (10 to 40 psi)) torque values are of the order of 0.113 millimeter-newton (1.00 in.-lb). Total accumulated running time at various speeds to 75 meters per second (250 ft/sec) and pressure differentials to 70 newtons per square centimeter (100 psi) was of the order of 120 hours with the accumulated wear of the self-acting seal being one-tenth the wear of the conventional seal. Pressurizing gas flow through the self-acting seal was approximately twice that of the conventional seal; the maximum flow rate of 4.25×10^{-4} cubic meter per second (0.9 scfm) is still, however, within the accepted range for gas turbine engine applications. Leakage of coolant oil through the self-acting seal was not detected throughout this investigation. The seal runner remained dry with no visible traces of oil seepage within the sealing area.

INTRODUCTION

Shaft seals are a problem area in advanced gas turbine engines for aircraft. The high seal sliding speeds and pressures of these advanced engines can cause high wear and excessive heat generation in conventional rubbing contact seals. Labyrinth seals are, therefore, sometimes used because their operation (without rubbing contact) provides a degree of reliability in that there are few seal failures. However, they have several disadvantages: high gas leakage into the bearing sump which has an impact on engine efficiency, and in addition, the passage of sufficiently large entrained debris that can reduce bearing life (ref. 1).

Recent developments in self-acting (gas bearing) face seals promise a seal with the speed capability of a labyrinth seal and the low leakage capability of a contact seal (refs. 2 to 6). This same self-acting principle can also be applied to circumferential shaft riding seals commonly used in engines for helicopters. Just as the self-acting (gas bearing) effect has extended the operating capability of the conventional face seal, it should also extend the operating capability of the conventional shaft riding seal.

Current shaft riding seals are limited to about 58 newton per square centimeter (85 psi) pressure differentials at speeds of 73 meters per second (240 ft/sec) (ref. 7). But advanced engines will require seals for pressure differentials to 138 newtons (200 psi) and speeds to 152 meters per second (500 ft/sec) (ref. 8). Thus a considerable gap exists between current capability and advanced requirements.

The objective of this study is to (1) compare seal operating temperatures (due to self-generated heat), leakage (pressurizing gas flow), and torque of self-acting shaft riding seals to that of conventional shaft riding seals, and (2) investigate wear of both seal types.

The self-acting seal was made from a conventional circumferential seal by machining the self-acting (gas bearing) geometry into the wear pad area of the carbon segments. Except for this extra machining, the seals are physically identical. Seal operating temperatures, leakage, and torque were measured at pressure to 70 newtons per square centimeter (100 psi), a shaft speed of 46 meters per second (150 ft/sec), and with undercooling of the cantilevered runner. In addition, both seals were operated at 46 meters per second (150 ft/sec) and 35 newtons per square centimeter (50 psi) for 50 hours.

APPARATUS

Circumferential Seal Test Apparatus

A schematic of the circumferential seal test apparatus is shown in figure 1. A variable-speed motor drives the test shaft through a 1:6 speed increaser. The test seal is mounted on a precision tool slide for axial positioning and is externally pressurized with dry air at room temperature. The pressurizing gas flow or leakage through the seal is measured by a series of rotometers; this procedure is one basis for comparing the performance of the conventional and self-acting seals. The torque generated by the seal was measured by a rotating torque sensor and monitored on a digital voltmeter. Bulk carbon temperatures were measured by thermocouples located at the interface of the static or secondary sealing surface of the carbon elements and the lapped end plates (see fig. 1). In earlier attempts to obtain bulk carbon temperature, measurements were made with the thermocouple embedded in the carbon. This method, however, restricted the radial movement of the segments and resulted in excessive segment end wear. A comparison of these two temperature measurement methods indicated that there was only a 2.8 to 5.6 K (5° to 10° F) temperature variance between the two methods, with the interface method indicating a lower bulk carbon temperature. Cooling oil at room tempera-

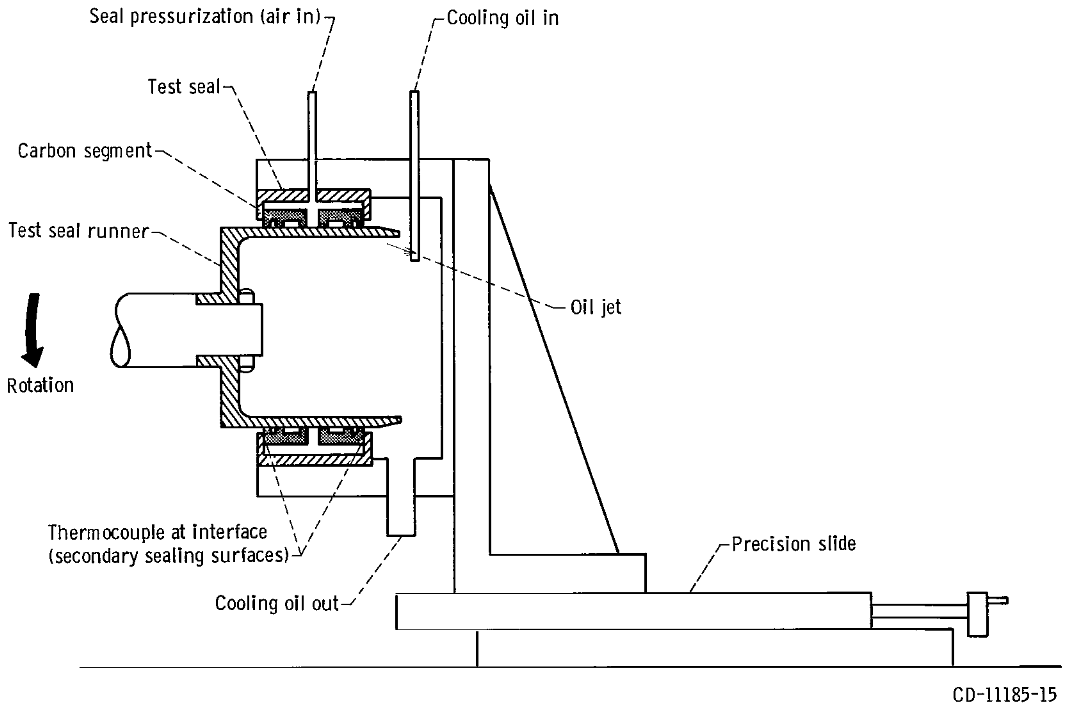

Figure 1. - Schematic of circumferential seal test apparatus.

ture was supplied to the inside of the seal runner through an oil jet (see fig. 1). Maximum oil flow used during these tests was 100 cubic centimeters per minute (7.3 (7.3 in.^{3}/min). (This low oil flow is comparable to that which can be expected as splash in a bearing cavity.)

A photograph of the circumferential seal with the self-acting geometry used in these tests is shown in figure 2. The seal has a bore diameter of 6.67 centimeters (2.625 in.) and is approximately 1.9 centimeters (0.750 in.) long. The seal consists of two segmented carbon rings held together by circumferential compression springs. Each ring

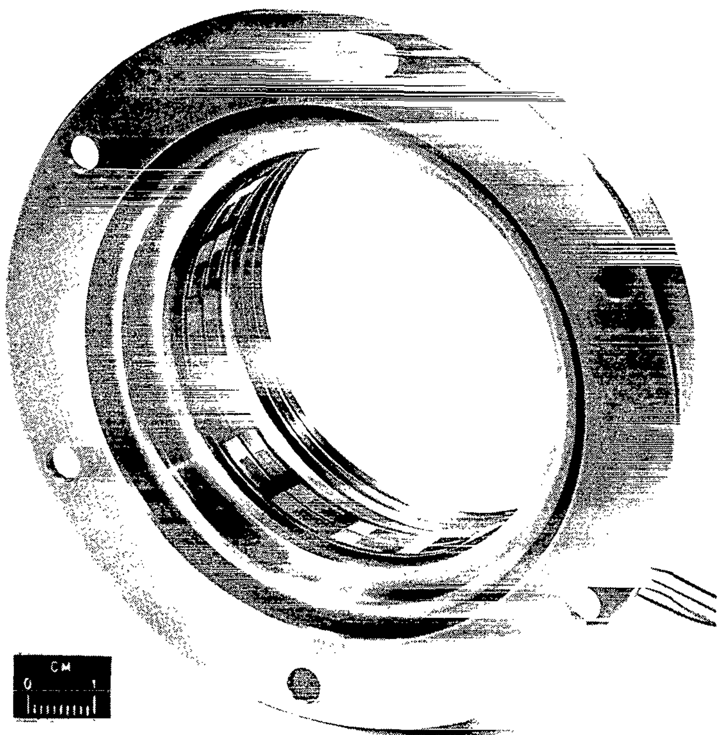

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Figure 2. - Circumferential test seal with self-acting lift geometery (two carbon segmented rings). Recess pad depth, 0.0025 centimeter (0.001 in.); seal after 125 hours of testing at shaft sliding velocities to 75 meters per second (250 ft/sec) and pressure differentials to 70 newtons per square centimeter (100 psi).

consists of three segments. Six small compression springs provide an axial force that holds the secondary scaling surface of the carbons in contact with the lapped end pieces.

Figure 3 is a developed view of one of the carbon segments showing details of the self-acting geometries that were machined into the inside diameter of each segment. Also indicated in this figure is the direction of rotation of the seal runner with respect to the carbon segments.

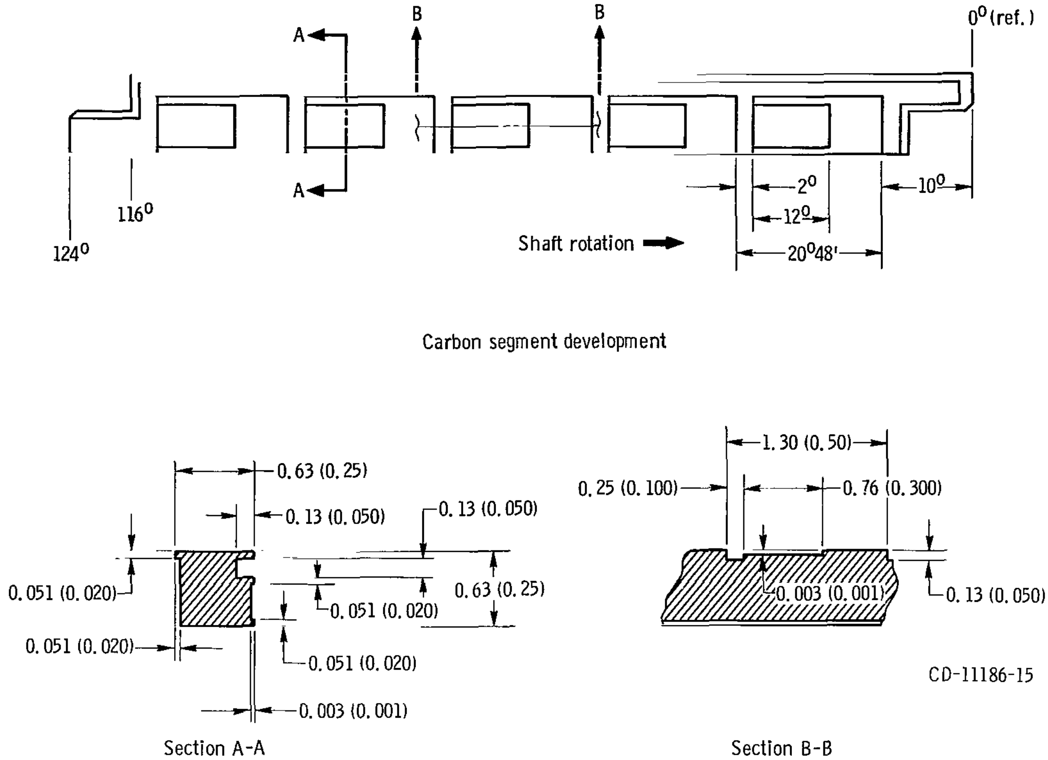

Figure 3. - Details of self-acting lift geometry machined into bore of carbon segments. Dimensions in centimeters (in.).

RESULTS AND DISCUSSION

A series of experiments was made to compare the performance of a conventional shaft riding seal and a modified seal with self-acting lift geometry. The seals have a nominal 6.67-centimeter (2.625-in.) bore and were operated at a shaft velocity of 46 meters per second (150 ft/sec), pressure differentials to 70 newtons per square centimeter (100 psi), and oil cooling, 100 cubic centimeters per minute (7.3 in.^{2}/min) to the inside of the cantilevered runner.

Figure 4 shows the bulk carbon temperatures as functions of differential pressure. Three sets of curves show the effect of no oil cooling, 50 cubic centimeter per minute (3.6 in.^{3}/min) oil cooling, and 100 cubic centimeter per minute (7.3 in.^{3}/min) oil cooling. From the curves it can be seen that the self-acting seal operates at lower carbon temperatures for all three cases. With oil cooling, the curves for the self-acting seal suggest that the seal is operating without rubbing contact at the lower differential pressures. Rubbing contact appears to start at about 21 newtons per square meter (30 psi) for a coolant flow of 50 cubic centimeters per minute (3.6 in.^{3}/min) and about

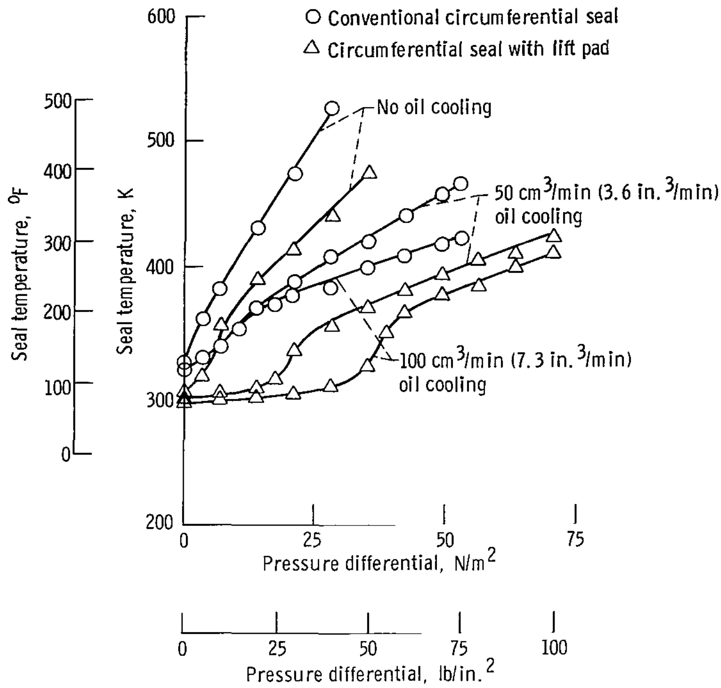

Figure 4. - Bulk carbon seal temperature as function of pressure ΔP. Shaft sliding velocity, 46 meters per second (150 ft/sec); room temperature, 293 K (78° F).

35 newtons per square meter (50 psi) for a coolant flow of 100 cubic centimeters per minute (7.3 in.^{3}/min). The lower oil flow results in a higher bulk carbon temperature that, in turn, causes greater distortions with the seal. The effect of oil cooling can also be seen for the conventional seal; however, these curves indicate that the seal probably maintained contact with the shaft throughout the entire differential pressure range. It should be noted that, because of rubbing contact, the conventional seal operates at a relatively higher temperature than the self-acting seal with a zero pressure differential. In the case of the conventional seal at zero differential pressure, the approximate 4.45-newton (1-lb) radial force from the circumferential compression springs causes the segments to contact the shaft; but in the self-acting seal the segments are lifted off the shaft by the self-acting lift forces.

A comparison of the gas leakage, or flow of the pressurization gas through the test seals, is shown in figure 5. These experiments were run at 46 meters per second (150 ft/sec) with an oil coolant flow of 100 cubic centimeters per minute (7.3 in.^{3}/min). The data for the self-acting seal show that the gas leakage is relatively high (2.4 to 4.3×10^{-4} m^3/sec (0.5 to 0.9 scfm)) until a pressure of 28 newtons per square meter (40 psi) is reached. Above this pressure the leakage decreases rapidly and reaches a low leakage (0.19×10^{-4} m^3/sec (0.04 scfm)) at about 42 newtons per square meter

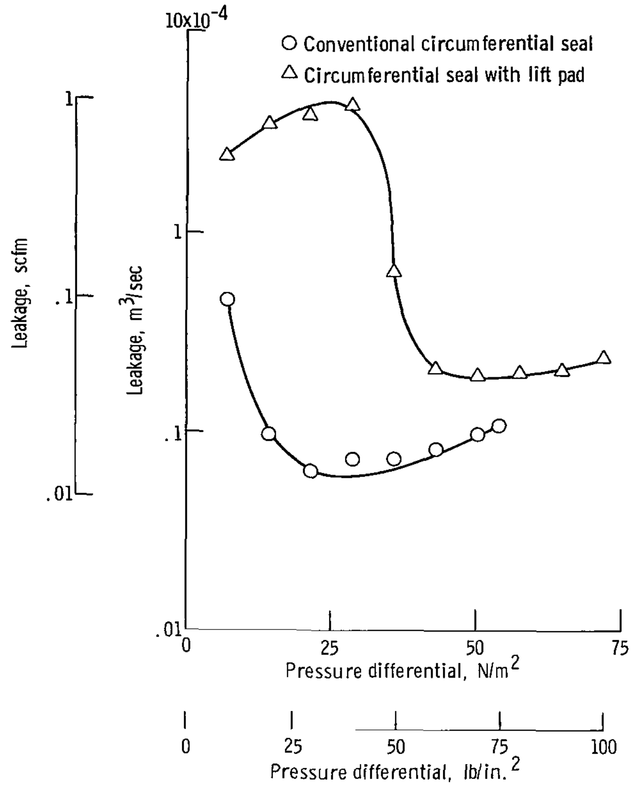

Figure 5. - Leakage of pressurization gas through seal as function of pressure ΔP. Shaft sliding velocity, 46 meters per second (150 ft/sec); room temperature, 300 K (78° F).

(60 psi). The leakage then gradually increases to a value 0.24×10^{-4} cubic meter per second (0.05 scfm) at 70 newtons per square meter (100 psi). This slight increase in leakage at the higher pressures is thought to be caused by thermal distortions within the seal (associated with the temperature increase noted in figure 4 at pressures above 28 N/m^2 (40 psi)). The conventional seal, however, maintains a low leakage (less than 0.14×10^{-4} m^3/sec (0.03 scfm)) over the pressure range of 14 to 53 newtons per square meter (20 to 75 psi). At 7 newtons per square meter (10 psi) the leakage of approximately 0.47×10^{-4} cubic meter per second (0.1 scfm) is probably the result of a small amount of lift due to shaft rotation. Thermal distortions that result in axial coning of the carbon segments or warpage of the lapped end pieces is thought to contribute to increased leakage at the higher pressures. The leakage rate shown for the self-acting seal is within the acceptable range for application to gas turbines, that being 4.72×10^{-4} cubic meter per second (1 scfm).

Seal torque as a function of differential pressure is compared in figure 6. Over the entire differential pressure range the torque of the conventional seal is generally twice

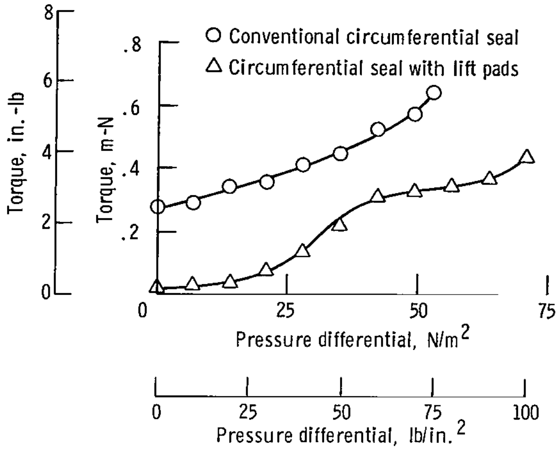

Figure 6. - Seal torque as function of pressure ΔP. Shaft sliding velocity, 46 meters per second (150 ft/sec); room temperature, 300 K (78° F).

that of the self-acting seal. At the lower pressures, torque values for the self-acting seal are of the order of 0.113 meter-newton (1.0 in.-lb). Around 28 newtons per square meter (40 psi) the torque increases more rapidly (with associated drop in leakage (fig. 5) and an increase in carbon temperature (fig. 4) and levels off again above 42 newtons per square meter (60 psi). The maximum torque measured for the self-acting seal was 0.43 meter-newton (3.8 in.-lb) at 70 newtons per square meter (100 psi). This lower torque (of the self-acting seal) means that the operating temperature and carbon wear rate will be much lower than that of a conventional seal. At zero pressure differential the conventional seal had a higher torque value (0.26 m-N (2 in.-lb)) than the self-acting seal (0.02 m-N (0.15 in.-lb)). This is caused by the radial load imposed on the seal by the circumferential compression springs.

Coolant oil leakage, or seepage, was not observed for the self-acting seal for all of the run conditions of these experiments. The fact that the seal runner remained dry is attributed to the flow of pressurization gas into the sump cavity. Oil leakage was observed, however, with the conventional seal at the lower differential pressures during startups but ceased when a pressure differential of 24 newtons per square meter (35 psi) was reached. Accumulated leakage for 20 starts totaled less than 100 cubic centimeters (7.3 in.^{3}).

In addition to the series of tests reported in figures 4 to 6, a 50-hour endurance test was run on both seal configurations at 46 meters per second (150 ft/sec) shaft velocity and 31 netwons per square meter (50 psi) pressure differential. Tabulated data are as follows:

	Conventional seal	Self-acting seal
Experiment duration, hr	50	50
Shaft sliding velocity, m/sec (ft/sec)	46 (150)	46 (150)
Pressure differential, N/cm^2 (psi)	31 (50)	31 (50)
Operating temperature, K (^oF)	394 (250)	347 (165)
Leakage, m^3/sec (scfm)	0.09×10^{-4} (0.02)	0.18×10^{-4} (0.04)
Torque, m-N (in.-lb)	0.43 (3.8)	0.23 (2.00)

Wear measurements for the endurance tests are inconclusive because the same seals were used for all the experiments to date. An overall wear comparison, however, shows that conventional seal wore approximately 2.5×10^{-3} to 3.8×10^{-3} centimeter (0.001 to 0.0015 in.) and that the self-acting seal were approximately 0.25×10^{-3} to 0.50×10^{-3} centimeter (0.0001 to 0.0002 in.) (measured as a decrease in carbon thickness). Total accumulated running time at various speeds to 75 meters per second (250 ft/sec), pressure differential to 70 netwons per square meter (100 psi), and several seal runner designs are of the order of 120 hours for the modified seal with self-acting lift geometries.

SUMMARY OF RESULTS

Seal operating temperatures, leakage (pressurizing gas flow), torque, and wear of a conventional circumferential seal were measured and compared to those of a conventional seal modified to have self-acting lift geometries. Both seal types had a 6.67 centimeter (2.625-in.) diameter bore and were operated at a sliding velocity of 46 meters per second (150 ft/sec) with pressures ranging from 0 to 70 newtons per square centimeter (0 to 100 psi). Results of the seal operation revealed the following:

1. The self-acting seal operates at a significantly lower bulk carbon temperature and torque as compared to the conventional seal. This was taken as evidence that the self-acting geometry is providing a lift force to reduce the contact load on the shaft.

2. The torque, temperature, and leakage all indicate a definite range in which the self-acting seal operates without sliding contact. For the 46 meter per second (150 ft/sec) sliding speed, this noncontact operation was from 0 to 21 newtons per square centimeter (0 to 30 psi).

3. Seal leakage of the self-acting seal was of the order of 0.18×10^{-4} cubic meter per second (0.04 scfm) for pressures above 42 newtons per square centimeter (60 psi) which is well within the accepted range for gas turbine engine applications.

4. The wear of the self-acting seal was about one-tenth that of the conventional seal.

5. Oil cooling of the seal runner had a very significant effect on carbon seal bulk temperature.

6. Oil leakage through the seal was not observed with the self-acting seal. Seal runners remained dry throughout experiments when coolant oil was supplied.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, March 21, 1972,
132-15.

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