NASA Technical Memorandum 107351

Bidirectional Brush Seals

Robert C. Hendricks
Lewis Research Center
Cleveland, Ohio

Jack Wilson
NYMA, Inc.
Brook Park, Ohio

Tom Wu
Lewis Research Center
Cleveland, Ohio

Ralph Flower
Cross Manufacturing, Ltd.
Devizes, England

Prepared for the
Turbo Expo '97
sponsored by the American Society of Mechanical Engineers
Orlando, Florida, June 2–5, 1997

National Aeronautics and
Space Administration

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Robert C. Hendricks
National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135

Jack Wilson
NYMA, Inc.
Brook Park, Ohio 44142

Tom Wu
National Aeronautics and Space Adminsitration
Lewis Research Center
Cleveland, Ohio 44135

Ralph Flower
Cross Mfg. Ltd
Devizes, England

ABSTRACT

Presented is a study of the use of a set of I.D./O.D. bidirectional brush seals to reduce the leakage losses in a wave rotor. Relative to the baseline configuration, data indicate the use of brush seals enhanced wave rotor efficiency from 36 to 45 percent at low leakages (small rotor endwall gap spacings) and from 15 to 33 percent at high leakages (larger endwall gap spacings). These brush seals are capable of sealing positive or negative pressure drops with respect to the axial direction.

Surface tribology for these tests suggested little evidence of grooving although the bristles did wear-in and the rotor surface appeared polished.

INTRODUCTION

If the performance goals for advanced gas turbines are to be achieved by conventional means, increasingly difficult thermal demands on materials need to be realized, and high overall pressure ratios are required. This has led to the suggestions for using nonconventional techniques to help achieve these goals, such as pulsed combustion (Kentfield, 1995) and wave rotor topping cycles (Kentfield 1995, Wilson and Paxson, 1996). Pulsed combustion can result in combustion with a pressure gain. A wave rotor topping cycle can also be thought of as pressure gain combustion, and appears to generate higher pressure ratios than does pulsed combustion currently, and so is more attractive. Fuel intercooling and wave rotors could be used in conjunction in the same engine to enhance performance.

A wave rotor machine uses expansion and shock waves within rotating passages on the rotor to accomplish the work typically done by an axial blade/vane or centrifugal component which may expand or compress the working fluid. In the wave rotor machine both compression and expansion occur and in some cases chemical reactions take place. Advantages cited for the wave rotor include enhanced efficiency, rotor material temperatures less than the peak gas temperatures, lower speed rotation with reduced stress, simple robust construction , and rapid transient response. However the achievement of high efficiency will depend on the ability to control leakage losses by adequate sealing.

Sealing can be a major factor in the wave rotor and the seal is dynamic. The leakage takes place into or out of a passage through the gap between the passage and the end walls of the machine. The leakage flow changes direction, depending on whether the passage is at high or low pressure. These endwall losses can potentially be controlled by seals such as the compliant brush configuration or the close self activating rim or leaf seals. Because brush seals could be incorporated easily into an existing 3-port wave rotor at NASA Lewis Research Center, (Wilson, 1966), bidirectional seals were used. Tests of wave rotor performance with, and without, the brush seals were made. The results of these tests are reported herein.

APPARATUS

A photograph of the wave rotor test rig illustrates the nature of a three-port flow divider(figure 1). There is a single inlet flow, which is separated into two outlet flows, one at higher stagnation pressure than the inlet, and the other at lower stagnation pressure. The gas flow inlet is at the upper right of the photograph and the low pressure fluid exhausts out the duct in the middle left part of the photograph. The high pressure air exhausts from the duct in the center. The rotor itself is a cylinder with axially aligned passages on its circumference. As a passage rotates, the pressure at the ends of the passage fluctuates between the high and low pressures. The cavity surrounding the rotor will be at a pressure close to the inlet pressure. Leakage will take place from the passage to the cavity when the passage is at high pressure and from the cavity to the passage when the latter is at low pressure. Thus the seals must be capable of withstanding pressure reversals with the corresponding flow reversals.

A cross-sectional view of the rotor (Fig. 2) illustrates the cavities and placement of the brush seals. The rotor passage represents that portion of the rotating cylinder containing the working fluid while the inner and outer cavities constitute potential paths for leakage. The movable end wall establishes a gap between the rotor/stator interface and to some extent controls the cavity volumes at the seal-rotor-stator interface.

There are sixty (60) 0.54-in. wide passages spaced about the circumference of the rotor, Fig. 3(a). As the rotor turns the pressure at each end of the passage varies with the highest pressure being about 3 to 4 times that of the lowest. The cavity pressure is nearly the average of the two. The rotor, seals and end plates are illustrated in Fig. 3(b). The function of the seals is to prevent leakage from the cavities into the rotor passage and vice versa (Fig. 2). The leakage may be thought of as composed of radial and circumferential components. Circumferential leakage is from passage to passage where the pressure differences are not large, so this leakage is believed to be less important than radial leakage. While this circumferential leakage could have been moderated somewhat by swirl brakes, only the brush seals were considered, which had little effect on the circumferential leakage. Radial leakage is from a high pressure passage into the cavity, and then from the cavity back into a low pressure passage. By blocking the path from the passages into the cavity brush seals can reduce the radial leakage.

BRUSH SEAL REQUIREMENTS

The wave rotor represents an unusual set of operating conditions for a brush seal. In addition to the usual compliance and sealing requirements, the brush must satisfy the following:

1. the brush must be capable of sealing bidirectional flows.
2. it must be capable of sealing pressures to ± 40 psi,
3. withstand surface speeds to 500 fps at temperatures < 350 °F.
4. the brush pair must seal the interface at both the inside and outside diameters,
5. both ends of the wave rotor must be sealed, i.e., matching pairs are required, and
6. they must be retrofit into the existing equipment with minimum modifications.

After some consideration, it was decided that the shielded design (Hendricks et al. 1992) could be modified to provide sealing in both directions, provided a gap was introduced between each sideplate and the bristles.

Figure 4(a) OD seal and Fig. 4(b) ID seal, illustrates the cross section views. The brush was otherwise of standard Cross Mfg. Co. construction. The bristles were 0.0028-in. diameter Haynes 25 AMS 5796 28, at angles 40 to 50° to the interface and inclined in the direction of rotation with suitable antirotation pins (Fig. 4). The rotor inner seal radius was nominally 5.518-in. (5.5167 left and 5.5192 right) after testing and the rotor outer seal radius was nominally 6.380-in. (6.3747 left and 6.3803 right) after testing. In each case 0.010 inch radial interference was built into the as manufactured seal.

The rubbing interfaces were Proxair(Union Carbide) LC-1H CrC coated to 0.006 to 0.010-in. thickness.

Photographs of the OD seal and ID seal are Figs. 4(c) and (d) respectively. The photographs shown herein were taken after the wave rotor testing was completed. Figures 4(e) and (f) represent a closer look at the bristle interfaces. Figures 4(g) and (h) illustrate a sharp view of the 0.028-inch bristles protruding from the seal fence (the fence is the same on both sides, see Figs. 4(a) and (b), and Figs. 4(i) and (j) represent the direct view of the brush bristle interface for the OD and ID seals respectively. As can be seen, the brushes are in good condition, with the exception of a tuft pullout in one location of the ID brush that can be seen in Fig. 4(h).

BRUSH SEAL WEAR-IN

The brush seals were installed by rotating them into position (i.e., in a direction opposite to the rotor rotating direction) and suitable static "O"-rings provided the necessary static seals. The rotor was torqued by hand to set the bristles, followed by a set of break in runs where the speed was incremented 1000 to 7500 rpm at nominal 1 hr intervals (10 hr) no heat and a similar schedule at 120 °F inlet temperature (7.5 hr) with 5 hr additional for 22.5 hr wear-in of the bristles. Once the bristles were set and rubbed into place a borescope examination of the bristles revealed the characteristic powder debris in flow stagnation regions with the remainder being swept away with the flow. Some bristles strayed beyond the pack with those of the inner seal being most susceptible. The photographs, Figs. 4(c) to (j), show little evidence of unusual bristle dispersion with the exception of Fig. 4(h).

OPERATIONS

The system was operated for a total time of 7.5 hr at a speed of 7400 rpm. The rotor average temperature was approximately the inlet temperature, 580R (322 K), with hot gas temperatures to 724 °R (402 K) and cold side temperatures to 511 °R (284 K).

RESULTS

Performance

A measure of the performance of a three-port wave rotor is the efficiency, defined as

$$\eta = \frac{\beta}{(1-\beta)}\left[\frac{\left(P_{hi}/P_{in}\right)^{(\gamma-1)/\gamma}-1}{1-\left(P_{lo}/P_{in}\right)^{(\gamma-1)/\gamma}}\right]$$

where P_{in} is the inlet stagnation pressure, P_{hi} is the stagnation pressure in the high pressure outlet, and P_{lo} is the stagnation pressure in the low pressure outlet, and β is the ratio of mass flow in the high pressure outlet to total mass flow. Higher values of both P_{hi}/P_{in} and P_{lo}/P_{in} will result in higher efficiency. Reducing leakage will create higher values of system pressure.

The wave rotor efficiency as a function of the size of the gap between the end wall and the rotor, for β = 0.37, and P_{lo}/P_{in} = 0.6, is illustrated in Fig. 5(a). It can be seen that the brush seals were effective in increasing efficiency. The mass leakage is approximately proportional to gap size, and in the experiment, the gap size was altered and the associated wave rotor efficiencies measured. To enable a direct comparison, tests were run with and without the brush seals.

The performance of a three port wave rotor is typically described by plotting P_{hi}/P_{in} versus P_{lo}/P_{in} . Such a plot is shown in Fig. 5(b), again for β = 0.37. The maximum efficiency occurs at roughly P_{lo}/P_{in} = 0.6. Again, this figure shows that use of the seals did result in higher pressures, with a corresponding increase in efficiency. On the average, the brushes diminished the leakage by a factor of 2. At a large gap spacing, the brush seals have a very pronounced effect. However, at small spacings, there is less of an effect on efficiency and pressure.

Surface Tribology

While the bristles wore significantly, and some tufts were disheveled to the point of permitting rivering, Figs. 4(i) and (j), both sets of brushes did not appear to deteriorate further with time, albeit, the testing time was 19 hr 54 min and not thousands of hours operating at nominal interferences of 6 and 9 mils (radial) for the OD and ID brushs seals respectively.

Attempts to quantify the surface wear were unsuccessful. From surface measurements we know, however, that the CrC interface coating wore less than 0.001-in., on all four interfaces. Optically however, polishing/ burnishing of the interfaces could be readily observed, Fig. 6(a) shows the polished track on the OD of the exhaust side and Fig. 6(b) shows the track on the inlet side. A closer examination of these tracks reveals “skipping” or changes in hardness which may represent tool marks of the parent machining operation or changes in the CrC coating. In Figs. 6(c) and (d), these types of marks are seen as bands extending beyond the brush wear path implying a precondition of the surface by machining. Figure 6(d) shows “spottiness” of the surface that may be related to bristle wearing. Figures 7(a) to (c) show similar patterns. No definitive tests were undertaken during the break-in stage or during the testing; thus one can only conclude that the bristles wore significantly, and the rotor coating showed little evidence of tracking other than being highly polished. Some powder debris is shown in Fig. 8.

SUMMARY

A set of inner and outer brush seals capable of bidirectionally restricting flows have been successfully fabricated and tested. The brush seals represented an extension of the side plates with sufficient gap to permit compliance.

Tests were run on a 3-port wave rotor with the baseline (gap control only) and the brush seal configurations in which the endwall gaps were varied and the wave rotor efficiencies were measured.

The wave rotor efficiency improvement due to the brush seals was more pronounced for the larger endwall gaps (from 15 to 33 percent) and to a lesser extent for the small endwall gaps (from 36 to 45 percent) where the leakages become quite small. On the average the leakages with brush seals were half those of the gap controlled baseline configuration.

For the limited test time and operating conidtions, the rotor surface appeared polished with little evidence of grooving although the bristles did wear-in.

ACKNOWLEDGMENT

The authors would like to thank Paulette Adams, Len Bellisario, Cleve Horn III, Rick Senyitko, and Hal Weaver who significantly contributed to the success of the project.

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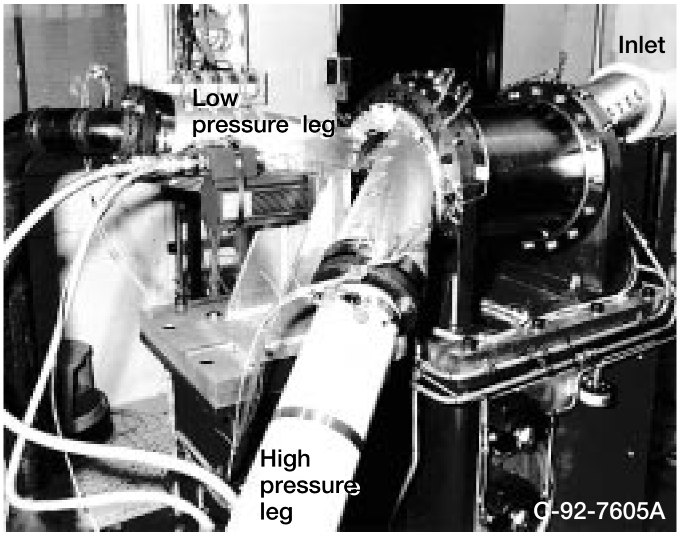

Figure 1.—Wave rotor test rig.

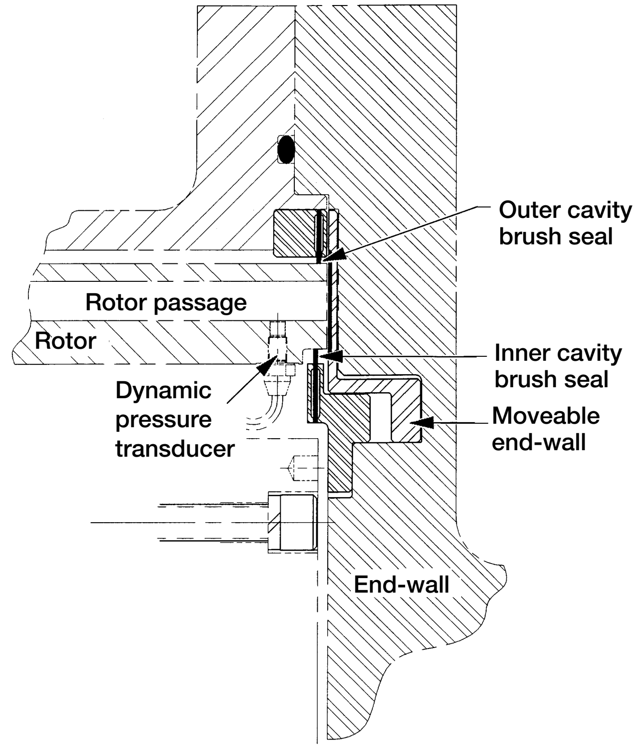

Figure 2.—Rotor cross section.

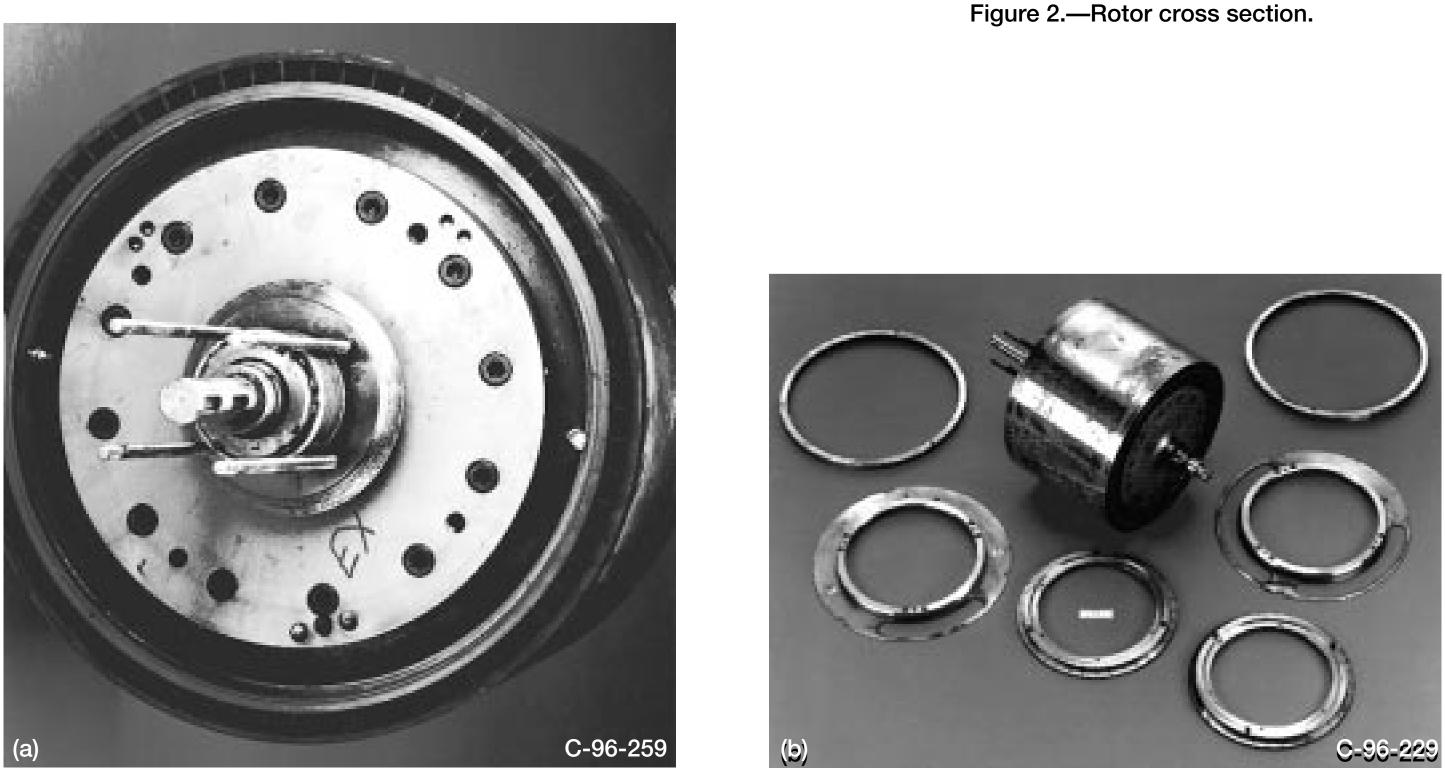

Figure 3.—(a) View of 60-passage rotor configuration. (b) Rotor, end flanges, and bush seals.

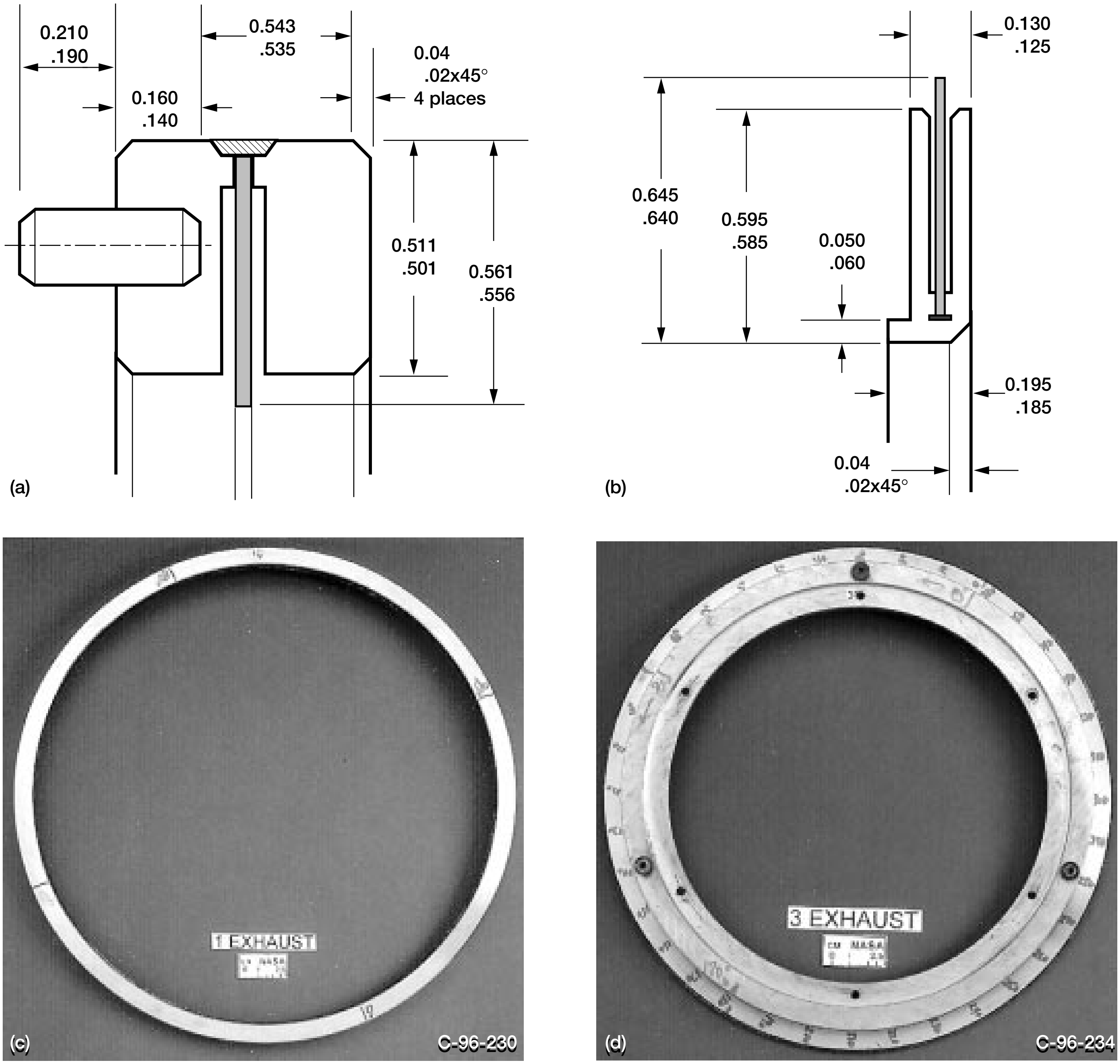

Figure 4.—Details of the seal cross section and photographs of brush seals taken after testing. (a) Outer brush seal cross section. (b) inner brush seal cross section. (c) OD seal. (d) ID seal.

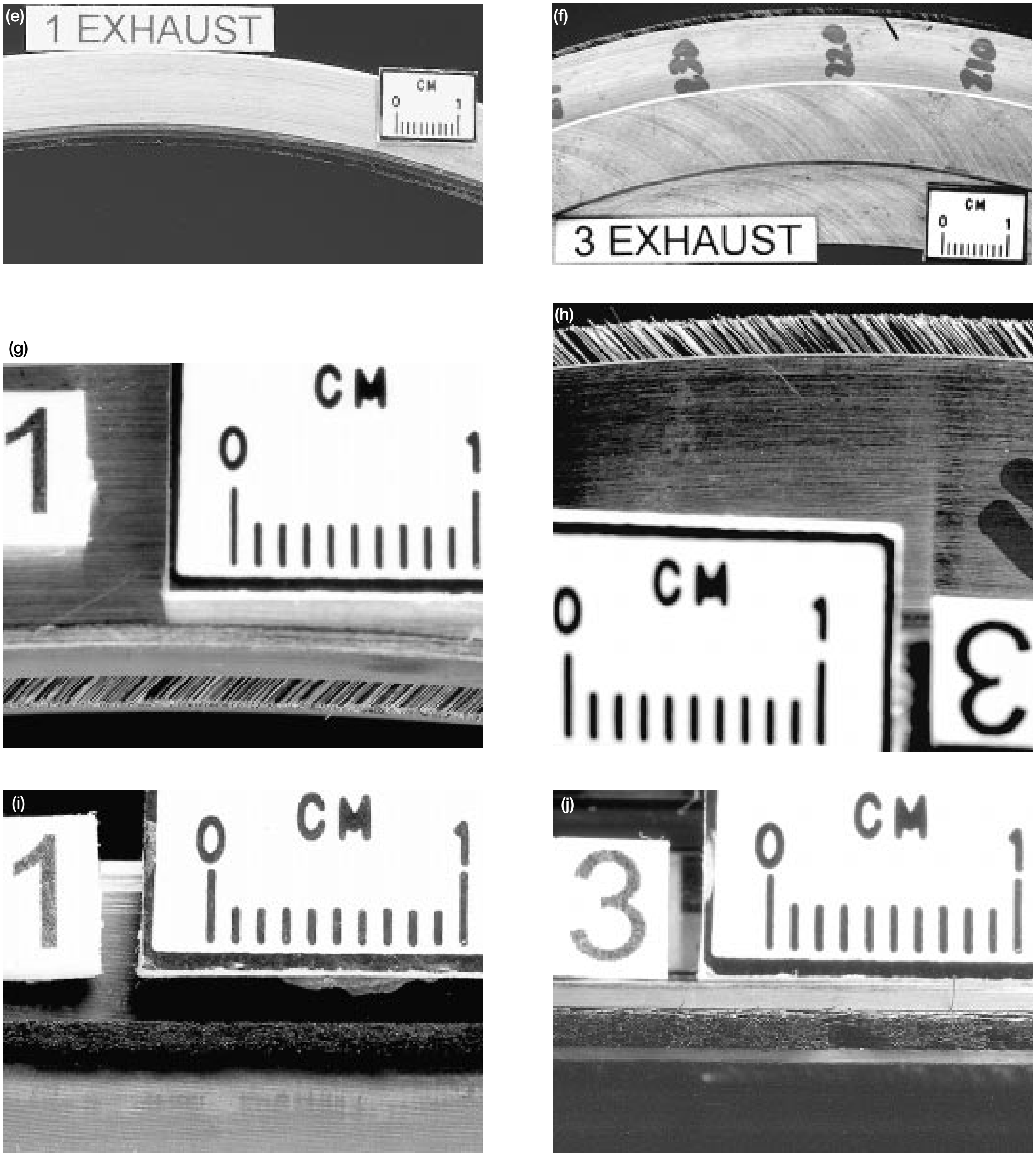

Figure 4.—Concluded. (e) OD seal close up. (f) ID seal close up. (g) OD seal bristles and side plate fence. (h) ID seal bristles and side plate fence. (i) OD seal, view of bristle tips. (j) ID seal, view of bristle tips.

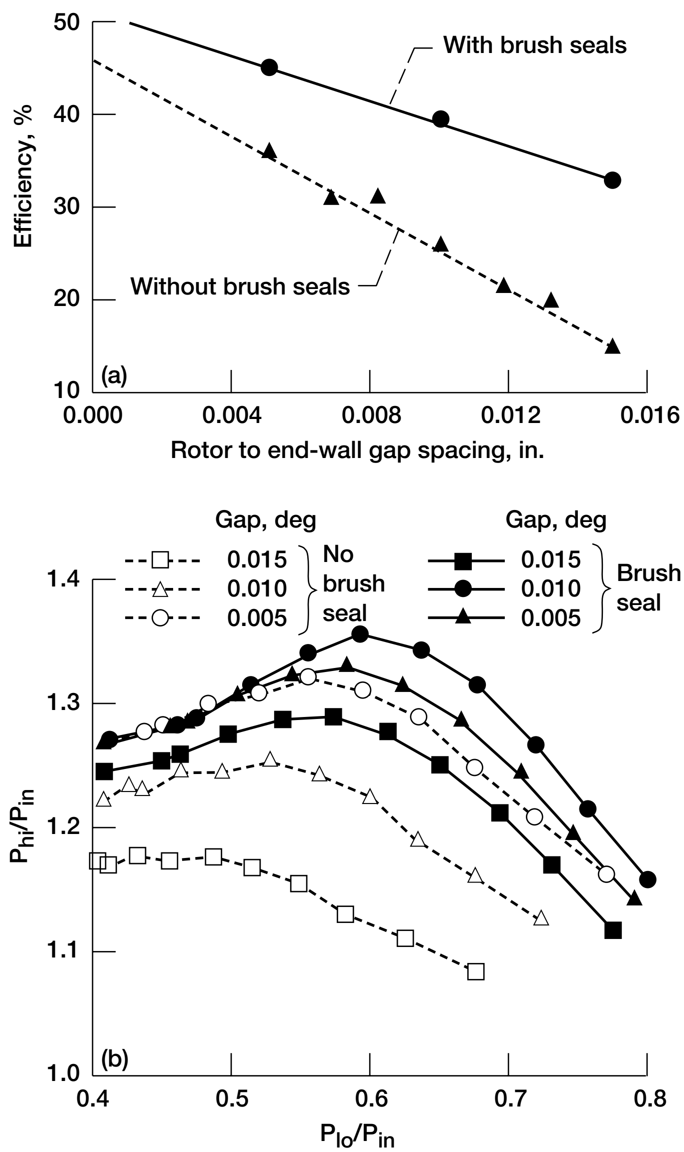

Figure 5.—(a) Efficiency vs gap for the wave rotor with and without brush seals. (b) High pressure/inlet pressure ratio vs low pressure/inlet pressure ratio with rotor to end wall gap as a parameter.

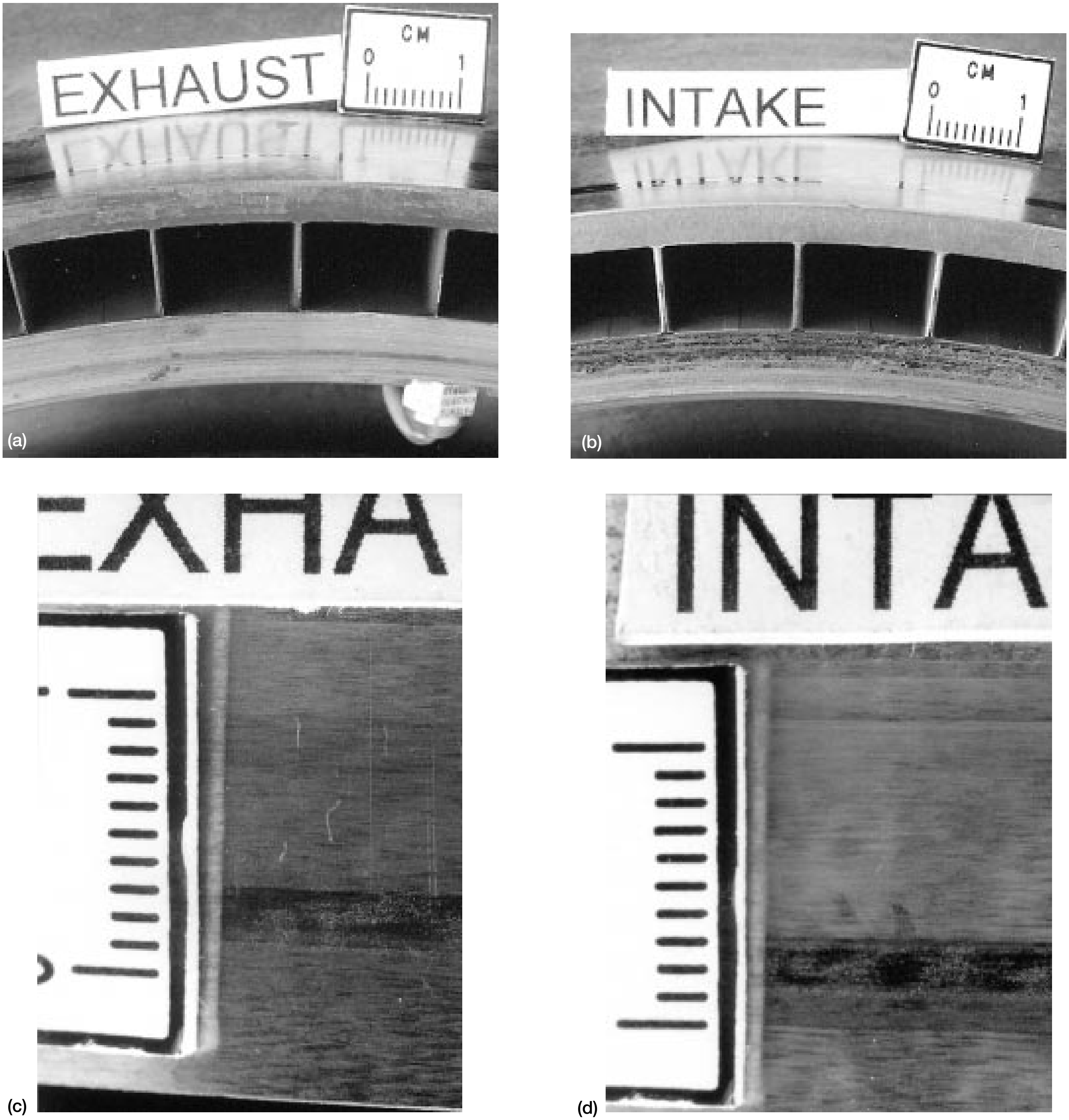

Figure 6.—(a) OD seal wear track seen as polished surface, exhaust side. (b) OD seal wear track seen as polished surface, inlet side. (c) Wear track closeup, exhaust side. (d) Wear track closeup, inlet side.

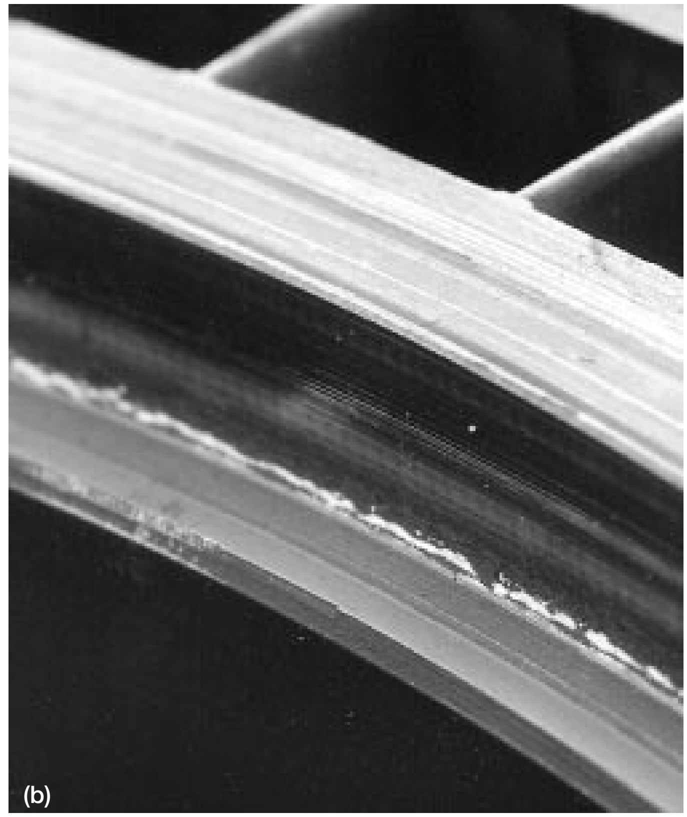

Figure 7.—(a) ID seal wear surface. (b) ID seal wear surface band marks. (c) ID seal "spotty" surface.

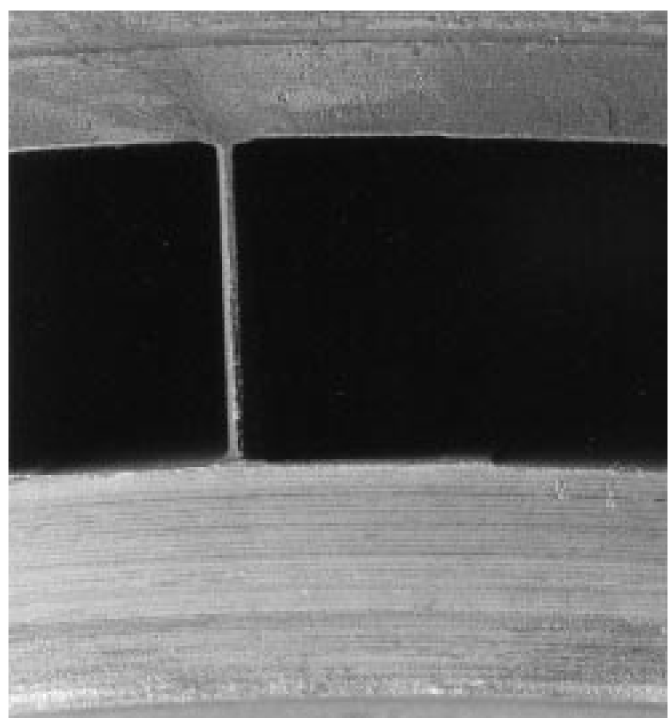

Figure 8.—Exhaust port powder bristle debris.

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. AGENCY USE ONLY (*Leave blank*)	2. REPORT DATE	3. REPORT TYPE AND DATES COVERED
	January 1997	Technical Memorandum

4. TITLE AND SUBTITLE

Bidirectional Brush Seals

5. FUNDING NUMBERS

WU–233–1B–1B

6. AUTHOR(S)

Robert C. Hendricks, Jack Wilson, Tom Wu, and Ralph Flower

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Lewis Research Center
Cleveland, Ohio 44135–3191

8. PERFORMING ORGANIZATION REPORT NUMBER

E–10519

9. SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)

National Aeronautics and Space Administration
Washington, DC 20546–0001

10. SPONSORING/MONITORING AGENCY REPORT NUMBER

NASA TM–107351

11. SUPPLEMENTARY NOTES

Prepared for the Turbo Expo '97 sponsored by the American Society of Mechanical Engineers, Orlando, Florida, June 2–5, 1997. Robert C. Hendricks, NASA Lewis Research Center; Jack Wilson, 2001 Aerospace Parkway, Brook Park, Ohio 44142 (work performed under NASA Contract NAS3–27186); Tom Wu, NASA Lewis Research Center; Ralph Flower, Cross Manufacturing, Ltd., Devizes, England. Responsible person, Robert C. Hendricks, organization code 5000, (216) 977–7507.

12a. DISTRIBUTION/AVAILABILITY STATEMENT

Unclassified - Unlimited
Subject Categories 07 and 20

This publication is available from the NASA Center for AeroSpace Information, (301) 621–0390.

12b. DISTRIBUTION CODE

13. ABSTRACT *(Maximum 200 words)*

A study of flow rate and pressure drop through a set of I.D./O.D. bidirectional brush seals is presented for both bench testing and wave rotor rig testing. Data indicate the use of brush seals enhanced performance from 3% to 12% depending on leakages; at low leakages, the relative performance is similar to that for close clearance cylindrical seals; however, at higher relative leakage rates the performance increases to 12% were noted. These brush seals are capable of sealing positive or negative pressure drops with respect to the axial direction.

14. SUBJECT TERMS

Brush seals; Wave rotor; Tribology

15. NUMBER OF PAGES

11

16. PRICE CODE

A03

17. SECURITY CLASSIFICATION OF REPORT	18. SECURITY CLASSIFICATION OF THIS PAGE	19. SECURITY CLASSIFICATION OF ABSTRACT	20. LIMITATION OF ABSTRACT
Unclassified	Unclassified	Unclassified	