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Hot Dynamic Test Rig for Measuring Hypersonic Engine Seal Flow and Durability

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ABSTRACT

A test fixture for measuring the dynamic performance of candidate high-temperature engine seal concepts has been developed.[¶] The test fixture has been developed to evaluate seal concepts under development for advanced hypersonic engines, such as those being considered for the National Aerospace Plane (NASP). The fixture can measure dynamic seal leakage performance from room temperature up to 840 °C (1550 °F) and air pressure differentials up to 0.7 MPa (100 psi). Performance of the seals can be measured while sealing against flat or engine-simulated distorted walls. In the fixture two seals are preloaded against the sides of a 0.3 m (1 ft) long saber that slides transverse to the axis of the seals, simulating the scrubbing motion anticipated in these engines. This report covers the capabilities of this text fixture along with preliminary data showing the dependence of seal leakage performance on high temperature cycling.

INTRODUCTION

A long term NASA goal is the development of a single-stage-to-orbit, reusable space vehicle. One approach being investigated is the National Aerospace Plane (NASP) shown in Fig. 1, that is powered by hydrogen-fueled ramjet-scramjet engines. To optimize engine performance over the full operating range, articulating panels, such as the nozzle panel shown in Fig. 2, are employed to provide required flowpath geometry.

Structural panel seals are required to seal the many linear feet of gaps between the movable engine panels and the stationary splitter walls[1] (Fig. 2). These panel edge seals serve one of two purposes. In engine regions where heating loads exceed seal temperature limits the seals serve to limit purge coolant flow, such as helium. In less severe areas of the engine, the seals serve to minimize parasitic leakage.

[¶]Rig installed at NASA Lewis Research Center.

The simultaneous requirements to operate hot while sealing against distorted adjacent sidewalls requires advanced seal design concepts combined with high-temperature materials technology. The performance of these key mechanical components must be evaluated prior to costly engine testing. This requires advanced test techniques which will be described in this paper.

Only preliminary estimates of the seal design criteria have been defined to-date. Those criteria related to the design of this test fixture include:

1. Limit seal purge coolant flow rates to that required for seal thermal survival in the high engine heating rates. Transpiration cooling flow-rate calculations performed by Steinetz indicate helium flow rates of approximately 0.013 kg/m-s (0.009 lb/ft-s) are required to prevent the engine inlet seals from exceeding an operating temperature of 815 °C (1500 °F).

2. Operate hot at temperatures consistent with: minimizing coolant resources and surviving engine cycle life (i.e., sliding durability).

3. Conform to and seal against distorted adjacent sidewalls. The distortion can be as great as 4 mm over 0.46 m span (0.15 in. over an 18 in. span)[1] - or approximately 0.010 mm deflection per millimeter of span.

4. Seal gaps up 1.6 mm (0.065in.) for 13 mm (0.5 in.) diameter seals or gaps up to 4 mm (0.16 in.) for 25 mm (1 in.) diameter seals.

5. Sustain minimal seal structural damage over engine life (estimated sliding distance is 120 m (400 ft)).

6. Maintain material stability and abrasion resistance in the air/hydrogen engine environments.

PREVIOUS WORK

A high temperature static test fixture was developed[2] to measure seal leakage rates under engine-simulated gas temperatures (up to 815 °C, 1500 °F) and pressures (up to 0.7 MPa, 100 psi). This test fixture was used to select amongst four candidate seals those seals having leakage flow near or below the flow goal. Leakage test revealed that the ceramic wafer seal and braided rope seal structures shown in Fig. 3 had relatively low leakages[3] and were able to conform and seal engine simulated distorted walls.

Preliminary scrubbing tests were performed using a simple seal scrubbing fixture to evaluate relative durability performances of competing braided rope seal architectures and materials[4]. In these tests, seal were preloaded and scrubbed against a reciprocating rub plate in a furnace operating at 700 °C (1300 °F). Long term durability benefits of hybrid (ceramic core/metal sheath) seals were demonstrated over the all-ceramic seal designs. These scrubbing tests combined with pre- and post-leakage flow measurements identified seal design features necessary for good durability and leakage. From these tests a matrix of seal architectures was developed for subsequent testing in the current high temperature dynamic seal test rig.

SEAL RIG DESIGN CRITERIA AND OBJECTIVES

A test fixture has been installed to address the following important engine seal development issues:

1. Quantify the change in seal leakage flow rates and evaluate seal durability as a function of engine simulated sliding conditions at temperatures up to 815 °C (1500 °F) and pressures up to 0.7 MPa (100 psi).

2. Evaluate seal leakage performance as a function of sealed gap dimension, and applied preload. Identify limits of gap-size and applied preload to prevent seal roll-out, under the simulated transverse seal sliding motion.

3. Measure the change in seal leakage sealing against either flat or engine simulated distorted sidewalls.

4. Assess dependence of seal leakage on seal movement in the seal channel caused by seal-to-sidewall interactions.

5. Establish seal design guidelines identifying seal architecture, material, and installation techniques required for the best balance of low leakage durable seal performance.

6. Evaluate seal-leakage gas property dependence measuring air leakage rates (flowpath) and helium leakage rates (candidate seal coolant).

7. Evaluate back pressure (i.e., gas density) effects on seal leakage.

TEST FIXTURE DESCRIPTION

A primary consideration in designing the test fixture was to simulate the scrubbing motion the seals will experience in the engine, in order to examine seal wear mechanisms and examine conditions leading to seal roll-out. In the engine, the seals will be scrubbed transverse to their axes. In the test rig two seals are held in intimate contact with a reciprocating saber as shown in Fig. 4. The saber moves vertically through a 50 mm (2-in.). stroke. Sealing both sides of the saber with the candidate test seals virtually eliminates the need for other supporting seals that must operate at the full operating temperature.

End-Leakage Control

Unlike circular seals, linear seals unavoidably have two ends. Treatment of the ends is critical to obtaining accurate measurement of the seal's leakage. Two steps were taken to virtually eliminate end leakage in the test rig. To seal the ends of the reciprocating saber, rub blocks made of Inconel X-750 material are employed. These rub blocks are machined flat and parallel and rub against both ends of the reciprocating saber. They are preloaded against the ends of the saber through pneumatically-actuated, hermetically-sealed bellows axial preloaders (Fig. 4), developed in Ref. 2. The rub blocks were machined to nominal dimensions and hand-lapped in place for a precise fit with their mating rub-block cavities. This approach virtually eliminates leakage past the rub-blocks.

Another feature controlling seal end-leakage, demonstrated in previous rig designs,[2] is the technique of "building-in" the ends of the seal into the test rig. The seal test zone and saber are nominally 0.3 m (12 in.) long. The seals and their cavities are made 25 mm (1 in.) longer than the test zones on both ends.

In the ends, there is no test gap that the seals must seal. The face of the seal, rig and side of the rub block all lie in the same plane. The seal is firmly preloaded (with special loaders) laterally against the side of the rub block virtually eliminating leakage in this region. The leakage follows the path of least resistance which is the center 0.3 m (1 ft) test zones (Fig. 4). If there is any trace leakage from the end cavities, its effects are minimized by testing 0.3 m (1 ft) lengths of seals and calculating an average leakage rate in terms of leakage rate per unit seal length.

Rig Heating

Metered, pressurized air is supplied to a plenum machined into the rig bases upstream of the seal (Fig. 4). In-line electric resistance air heaters are screwed into the bases. Hot, pressurized air exhausting from the heaters flows through the mesh heat exchanger to the seals. A maximum of four air heaters can be attached to the rig--each delivering 0.009 kg/s (0.02 lb/s) of air at temperatures up to 815 °C (1500 °F) and pressures up to 0.7 MPa (100 psi). Each heater is controlled with a digital controller with temperature feedback.

The mesh heat exchanger serves two purposes. The mesh diffuses the incoming air to the full length of the seals. Second, under low flow conditions heat conducted to the mesh from the surface heaters further heats the incoming air to the desired temperature. This allows the electric air heaters to run cooler, thereby extending their lives.

High watt-density conduction heaters are strapped to the top and bottom of the test rig. Four 3.5 kW heaters are used to ramp the rig temperature up to the desired test temperature using a digital ramp-soak controller. Due to the efficiency of thermal-conduction, these surface heaters supply most of the heat to the rig during operation. Employing surface heaters on the top and bottom minimize the thermal gradients and any unnecessary thermal distortions through the 15 cm (6 in.) high test fixture.

To achieve the high test temperatures, the test rig is insulated with high temperature, low conductivity board insulation. At leat 5 cm (2 in.) thick alumina insulating board is fitted closely around the outside of the rig.

Saber Actuation

The saber moves vertically between the two test seals. The saber is guided by a set of linear bearings maintaining precise alignment to the seals and seal holders. The bearings have a high preload to provide the necessary stiffness and prevent any out of plane or sideways motion of the saber.

The saber crank system is driven by a 2.2 kW (3 hp) variable speed DC motor. This motor was chosen for exceptional low speed torque. An external blower permits very slow operation without overheating. The motor drives through a 100:1 double reduction gearbox. A slip clutch is mounted between the gearbox and crank to ensure the torque does not exceed a user selected value, in the event something becomes jammed. An eccentric crank provides the desired reciprocating motion.

A connecting rod connects the crankshaft to the saber through the linear-bearing carriage assembly. A stroke of 5 cm (2 in.) was selected for these initial tests. The average linear saber speed is adjustable from 0 to 5 cm/sec (2 in./sec). Tests will be performed at 2.5 cm/sec (1 in./sec) to simulate engine seal sliding speeds. The position of the saber is measured with a linear variable differential transformer (LVDT). Cycle counting is performed with a cycle counter mounted on the idler end of the crank shaft.

Seal Drag Measurement

Load cells are mounted in the saber support assembly to measure seal drag loads over the simulated hot sliding duty cycle. The seal drag loads are important for engine designers for structural design considerations of the adjacent engine walls and for sizing panel actuation systems.

Load cells are mounted between the two pairs of linear bearings (Fig. 5) so that none of the off-axis loads caused by the connecting rod are transmitted to the load cells. These off-axis loads would lead to inaccuracies in the load readings. To prevent thermal damage of the load cells, cooling air is purged over the cells keeping below 70 °C (160 °F), well below their 200 °C (400 °F) temperature compensation rating. Seal friction coefficients are evaluated by dividing the seal drag load by the normal load applied to the seals. Seal drag loads are determined as the average load for the upstroke and downstroke motions of the saber. Seal drag loads are corrected for the saber weight and the pressure-induced loads acting upward on the saber.

Comparison tests performed with the rub blocks in and out of contact with the saber indicated that under steady-state conditions, rub block loads were a small fraction of the seal drag loads. This is expected when considering the contact width of the rub blocks is 4 percent of the seal contact length.

Test Condition Measurement

Leakage Measurement—Leakage rates are measured upstream of the in-line heaters. Leakage is measured in this manner for several reasons. Measuring the mass flow prior to heating to 815 °C (1500 °F) precludes the need to pre-cool the gas before measuring it with room temperature flowmeters. Eliminating the need to capture the leaked gas and then pre-cool it saves considerable expense and complexity. Measuring the leakage flow upstream of the seal also gives a conservative estimate of the actual seal leakage rate. The leakage rate that is measured includes the seal leakage (both through and around the seals) and any minor parasitic leakage mentioned earlier.

Three flow meters (Teledyne Hastings HFM-201) are arranged in parallel in a flow bench and are switch-selectable to measure air flows in the following ranges: 0–0.009 kg/s, 0–0.018 kg/s; 0–0.036 kg/s (0 to 0.02 lb/s, 0.04 lb/s, 0.08 lb/s).

Helium flow is supplied by a pair of K-bottles plumbed into the system to evaluate seal coolant leakage rates. Helium flow rates up to 0.005 kg/s (0.01 lb/s) can be evaluated.

Pressure Measurement.—The pressure differential applied across the seal is evaluated using pitot static pressure taps immediately upstream of the seal. Gage pressure measurements are used since the seal vents to atmospheric conditions and the exiting flow velocities are low. The pressures are measured using solid-state capacitance type transducers. Pressure is supplied to the transducers using suitably long (over 15 cm) tubing, to prevent high temperatures from reaching the transducer. Measurements are taken at multiple axial stations so an accurate average pressure differential is obtained.

Pressures are also measured in the seal cavity behind the seal to determine fluid forces exerted by the simulated engine chamber pressure on the backside of the seal. Pressure supplied to the lateral preload bellows is also measured from which the seal contact stress is calculated.

Back Pressure.—The effects of engine-chamber back-pressure on seal leakage are measured by restricting the flow out of the top of the rig (above seal in Fig. 4.) A special picture frame collar is bolted in place around the saber with hand valves to control the back pressure in the cavity above the seals. Room temperature back pressures up to 0.5 MPa (70 psia) can be tested.

Temperature Measurement.—Gas temperatures impinging on the seal are measured using fast-acting thermocouples just upstream of the seal. The thermocouple beads are inserted in the gas stream to measure true gas temperature. For averaging purposes, multiple thermocouples are used along the length of each of the 0.3 m (1 ft seals). Thermocouples are also placed at the exhaust of the heaters. Temperature readings from these thermocouples are used in independent feedback control circuits for each of the heaters.

Key hardware temperatures such as the seal, the Inconel metal bellows, and the rig bulk temperature are also measured using thermocouples. In all cases, type K (Chromel-Alumel; 1090 °C) thermocouples are used. Wherever thermocouples, pitot static pressure taps, or bellows supply tubes are inserted into the pressurized rig special high temperature fittings are used to prevent parasitic leakage. These fittings are made by Conax Co. and use a proprietary fitting design with magnesium-oxide (lava) type glands capable of 980 °C operation.

All sensors are continuously monitored with a personal computer data acquisition system. The host platform is a 486 PC with a LabMaster analog-to-digital converter board. Lab Tech Notebook XE software is used to record and display the data.

Seal Preload and Measurement.—An important parameter requiring investigation is the seal preload required to adequately seal the pressurized gas. Both lateral preload (e.g., transverse to the seal axis) and axial preload (e.g. rub-block) are measured in the rig. Lateral preload is applied using a series of hermetically-sealed, welded-leaf, flexible Inconel 718 metal bellows (Fig. 4). These 0.5 in. diameter bellows are mounted on 2.5 cm (1.0 in). centers and are pressurized from a common manifold. In-line with each of the bellows pressure supply tube is a hand valve (Fig. 6) that can be used to select the number of active bellows.

A thin (1.5 mm; 0.06 in.) strip of Inconel is placed between the nose of the sealed bellows and the back of the candidate seal. This strip distributes the preload to portions of the seal between the bellows. An average seal contact pressure is determined by pro-rating bellows pressure (as measured in the manifold supply) by the ratio of the bellows area to the backside seal area. If all of the bellows are active the average contact pressure is the bellows supply pressure divided by 6.5.

Fixed Preload.—In certain area of the engine, replacing the "active" preload system with a fixed preload is desirable if seal performance does not suffer. To evaluate the effect of a fixed preload on leakage and durability, the bellows preload system described is replaced with a pre-machined seal crush bar that is installed behind the seal. These crush bars are preloaded against the seals with load-cell instrumented jacking-screws to monitor seal prelaod with cycling. Early durability studies[4] showed the drop off in seal preload with cycling in a fixed preload set-up, but could not measure leakage. The current fixture will enable leakage measurement over the course of seal cycling to determine acceptability of seal performance.

Gap/Wall Condition

Uniform Gaps.—The seals must seal several different gap conditions at various locations in the engine. The gap in the test zone (Fig. 4) is changed by selecting different saber thicknesses. The thicker the saber the smaller the gap presented to the seal. Gaps that can be tested for the 1.3 cm (0.5 in.) seals are 0.8 mm (0.03 in.) and 1.6 cm (0.065 in.).

Distorted Gaps.—To test seals against simulated wall distortions, the flat saber is replaced with one with a pre-machined surface simulating the engine wall distortion. Sabers with both inward and outward bulges have been designed that simulate the engine wall distortion on an inch per inch-of-span basis.

Surface Condition.—Seal friction and wear characteristics can also be assessed against engine wall material or surface roughness conditions. This is accomplished by replacing the saber plate with sabers made of any of the candidate engine materials with anticipated engine wall surface roughnesses.

DESIGN FOR HIGH TEMPERATURE SERVICE

Designing test fixtures for elevated temperature operation requires attention to be paid to certain design elements not often required for conventional design. For instance the rig must be properly sized to meet safety criteria of high temperature pressure vessels. Also allowances must be made for the significant growths that will occur as the fixture heats to the operating temperatures.

Stress Analysis.—In sizing the test fixture, a finite-element stress analysis of the test rig was performed. The loads used in the finite element model included a 0.7 MPa (100 psi) seal preload pressure bearing against the front wall, and a 0.7 MPa (100 psi) simulated engine pressure applied to the "wetted" surfaces upstream of the test seal. These represent the maximum engine pressure and seal preload envisioned for the test sequence. A vertical force of 570 kg (1250 lb) was imposed on the top surface of the seal channel representing a maximum anticipated seal drag load. The maximum Von

Mises stress found using the MARC finite element code was 39 MPa (5700 psi) at the bottom of the plenum pocket machined into the fixture.

The stress found above was compared to the allowable strength as recommended by the ASME Boiler and Pressure Code. In Refs. 5 and 6, the design stress is the lesser of one-third the tensile strength at operating temperature or two-thirds the yield strength at operating temperature. The first criterion is the more conservative of the two resulting in an allowable design stress of 130 MPa (18 ksi) (e.g., 1/3 of 390 MPa (55 ksi) tensile strength) for Inconel X-750 at 815 °C (1500 °F).[7] This allowable stress is significantly greater than the maximum stress calculated for the test fixture. Hence it was concluded that the rig was properly sized. Comparing the design stress to the Von Mises stress, a factor of safety of 3.2 is found.

In addition to having high yield and ultimate strengths at temperature, Inconel X-750 has a very high creep rupture strength. At 815 °C (1500 °F), its 1000 hr creep rupture strength of 140 MPa (20 ksi)[7] ranks with the best of the high temperature metals. By comparison this creep rupture strength is almost four times that of Inconel 600 and five times that of 304 series stainless steel. These features in addition to its excellent oxidation resistance make Inconel X-750 an excellent material for the high temperature fixture.

Large Scale Thermal Growth.—Calculations predicted that the 0.6 m (22-in.). Inconel fixture heated to 815 °C (1500 °F) would grow over 6.4 mm (0.25 in.). This is the growth measured when the rig reaches operating temperature. To accommodate thermal growth of this magnitude special features were incorporated into the rig:

> Rig Tie-down.—Ignoring thermal growth will normally result in unforgiving hardware failures because the thermal strain energy will be released in one way or another. To allow the rig to grow unimpeded, slotted feet were used on both ends of the rig. Light tension on the end-bolts used to secure the rig to the table allowed the rig to expand and

contract without binding during a temperature cycle. Vertical loads were reacted through tie-down bolts located on rig centerline.

Piping Manifold.—A flexible piping manifold system was implemented in the rig. The manifold allows the heaters to move axially with the rig growth mentioned above without placing bending loads and unnecessary stresses on the hot heater pipes. Oversized clearance holes are made in the bench top for the heaters to allow rig growth. The manifold also allows the pipes to grow axially.

Axial Preload System.—The axial preload system actuates the two end rub blocks. The rub blocks seal the leakage path at the saber ends. The pneumatic actuators are bracketed to the base of the rig. This guarantees that the axial load will not be affected by the rig expansion.

TEST FIXTURE DEMONSTRATION

The design features incorporated in the test fixture allow a broad range of candidate engine seal concepts to be tested. The test rig is easily configured to test the ceramic wafer seal and the braided ceramic rope seal.

Seal Specimen.—For purposes of demonstrating the high temperature capability of the test rig, the ceramic rope seal depicted in Fig. 2 was installed and tested.

Test Results.—Leakage rates for the ceramic rope seal sealing 840 °C (1550 °F) air are shown versus pressure drop in Fig. 7 after 120 m (400 ft) of sliding. In this figure, the hot leakage rates are normalized by the leakage rate before cycling.

These results show a reduction in leakage with cycling. However, the seal was badly damaged during the test having worn completely through the outer sheath (Fig. 8) and was difficult to remove from the fixture. These test indicated that rub-tolerant seals, such as a Lewis developed hybrid seal (flow resistant ceramic core overbraided with a superalloy sheath), are required to withstand the punishing engine sliding conditions.

SUMMARY

A hot dynamic test fixture for evaluating the performance of advanced hypersonic engine seals has been installed and successfully checked-out. The test fixture will be used to assess the durability and flow change of candidate seals as a function of simulated duty cycle. The fixture can subject seals to simulated temperatures up to 840 °C (1550 °)F and pressures up to 0.7 MPa (100 psi). Furthermore, seal performance in sealing either straight or distorted sidewalls can be measured. The sensitivity of leakage performance to either active or fixed preload can also be evaluated.

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Note: Mention of manufacturers is made only for reference purposes and does not constitue a product endorsement by NASA or the U.S. government.

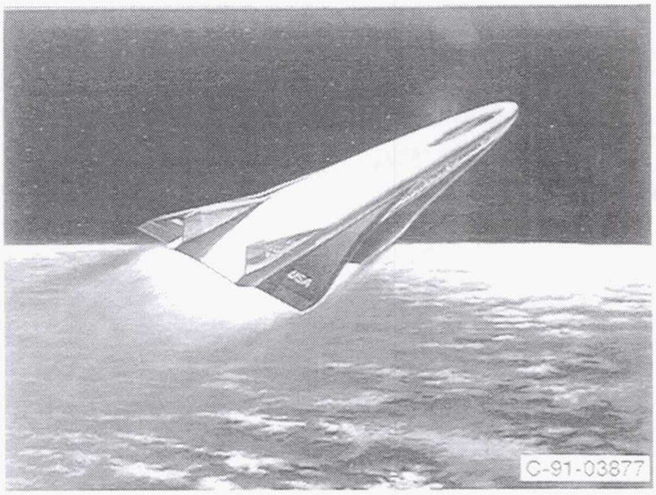

Figure 1.—National Aerospace Plane.

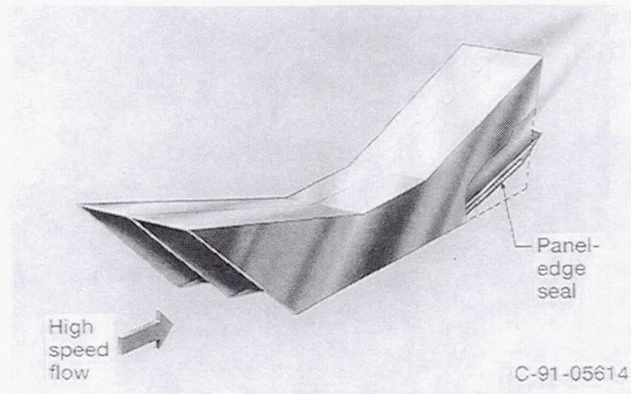

Figure 2.—Hypersonic engine panel-edge seal.

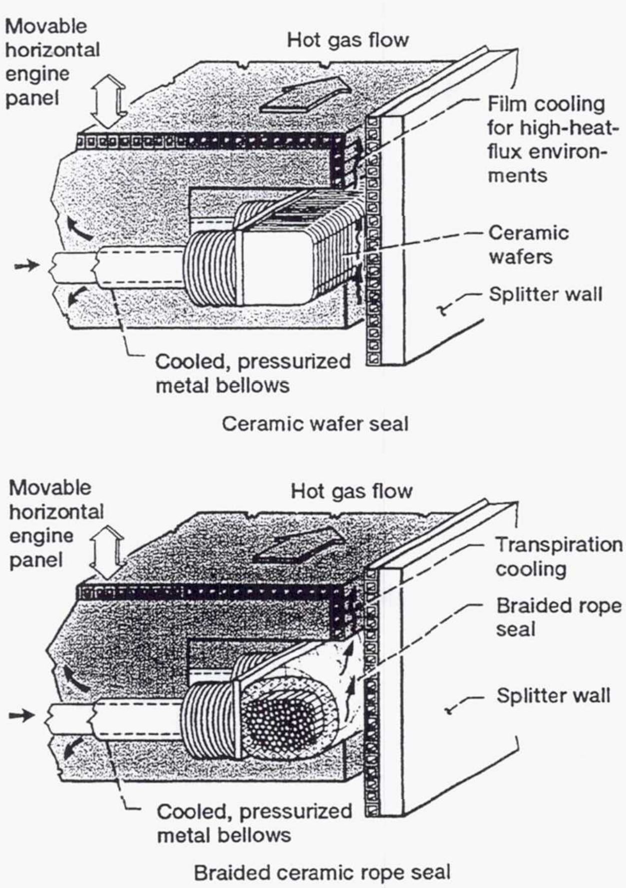

Figure 3.—Seal concepts under development.

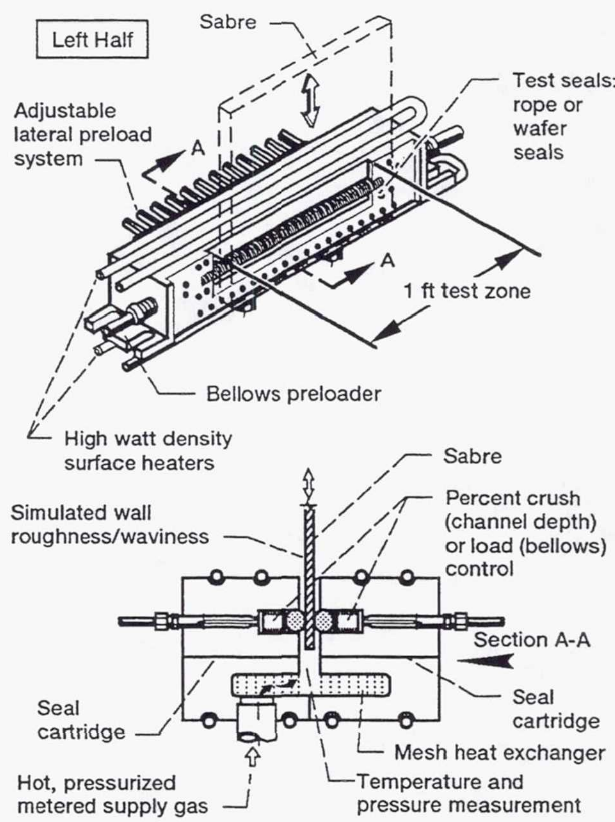

Figure 4.—Schematic of seal rig showing the seal holders, hot gas path, and saber motion.

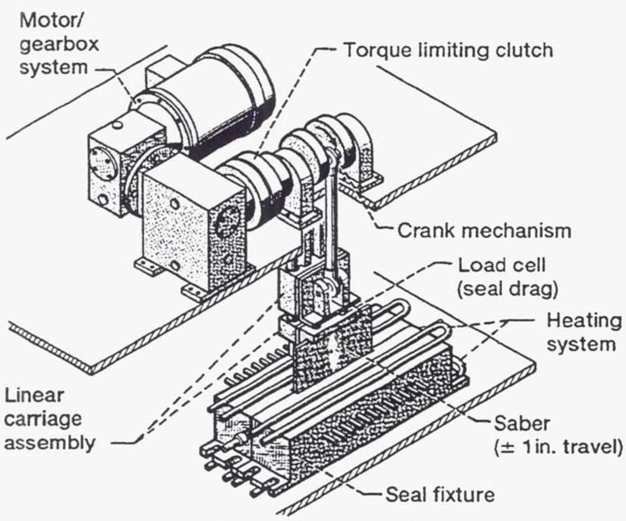

Figure 5.—Schematic of actuation system.

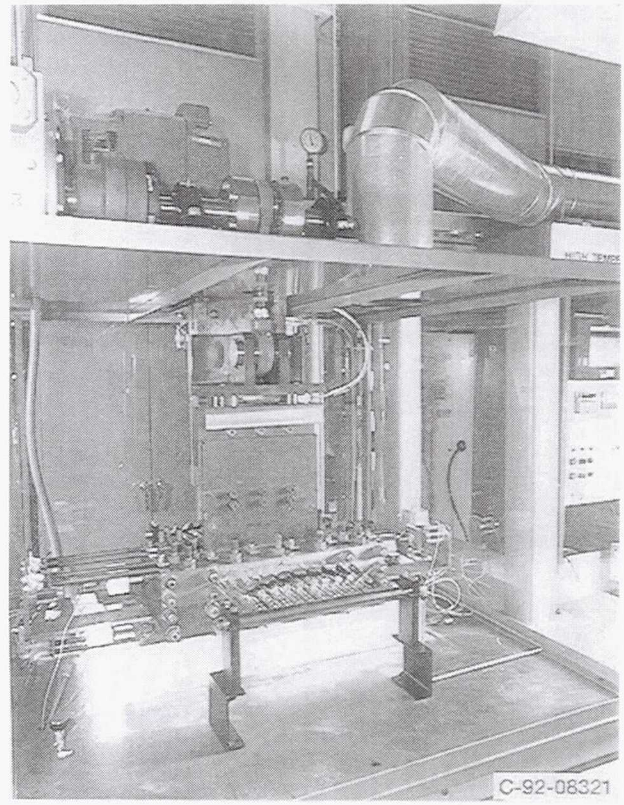

Figure 6.—Photograph of high temperature dynamic seal rig.

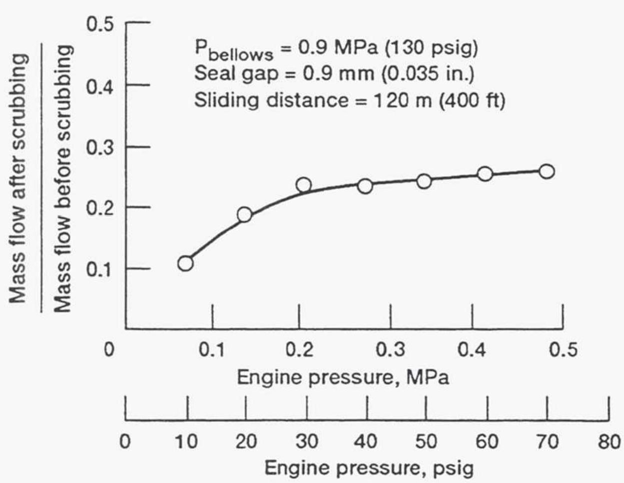

Figure 7.—Normalized seal leakage data at high temperature (840 °C; 1550 °F) showing decrease in leakage with cycling.

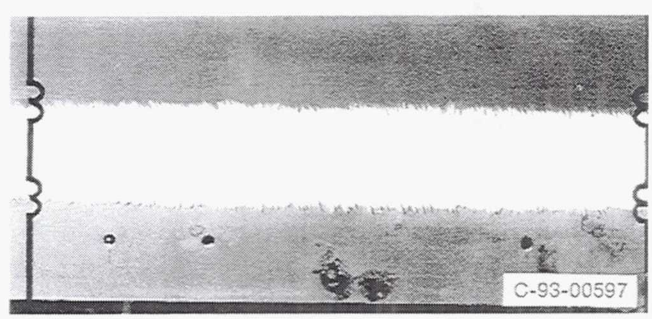

Figure 8.—Severely damaged braided ceramic rope seal in test rig (center portion shown) after hot scrubbing test demonstrating need for rub-tolerant seals.

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13. ABSTRACT ***(Maximum 200 words)***

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