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High Temperature Propulsion System Structural Seals for Future Space Launch Vehicles

Patrick H. Dunlap, Jr. and Bruce M. Steinetz
Glenn Research Center, Cleveland, Ohio

Jeffrey J. DeMange
University of Toledo, Toledo, Ohio

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HIGH TEMPERATURE PROPULSION SYSTEM STRUCTURAL SEALS FOR FUTURE SPACE LAUNCH VEHICLES

Patrick H. Dunlap, Jr. and Bruce M. Steinetz
National Aeronautics and Space Administration
Glenn Research Center
Cleveland, Ohio 44135

Jeffrey J. DeMange
University of Toledo
Toledo, Ohio 43606

ABSTRACT

Durable, flexible sliding seals are required in advanced hypersonic engines to seal the perimeters of movable engine ramps for efficient, safe operation in high heat flux environments at temperatures of 2000 to 2500 °F. Current seal designs do not meet the demanding requirements for future engines, so NASA's Glenn Research Center is developing advanced seals and preloading devices to overcome these shortfalls. An advanced ceramic wafer seal design and two types of seal preloading devices were evaluated in a series of compression, scrub, and flow tests. Silicon nitride wafer seals survived 2000 in. (1000 cycles) of scrubbing at 1600 °F against an Inconel 625 rub surface with no chips or signs of damage. Flow rates measured for the wafers before and after scrubbing were almost identical and were up to 32 times lower than those recorded for the best braided rope seal flow blockers. Canted coil springs and silicon nitride compression springs showed promise conceptually as potential seal preloading devices to help maintain seal resiliency. A finite element model of the canted coil spring revealed that it should be possible to produce a spring out of high temperature materials for applications at 2000+ °F.

INTRODUCTION

High temperature, dynamic structural seals are required in advanced hypersonic engines to seal the perimeters of movable engine ramps for efficient, safe operation in high heat flux environments at temperatures from 2000 to 2500 °F. Seals must be flexible enough to accommodate distorted walls and provide positive, resilient sealing. They also must be sufficiently durable to meet required engine life goals.

NASA Glenn Research Center (GRC) became involved in the development of high temperature structural seals in the late 1980's and early 1990's during the National Aerospace Plane (NASP) program. Researchers at GRC (then called the Lewis Research Center) carried out an in-house program to develop seals for the NASP hypersonic engine and oversaw industry efforts for airframe and propulsion system seal development for this vehicle.[1] Figure 1 shows one of the seal locations in the NASP engine. Seals were needed along the edges of movable panels in the engine to seal gaps between the panels and adjacent engine sidewalls. Seal development efforts undertaken during the NASP program became the basis for current seal development activities at GRC to meet the seal challenges of future hypersonic and reentry vehicles.

Seals developed during the NASP program met many requirements but fell short of leakage, durability, and resiliency goals. Due to program termination the seals could not be adequately matured. To overcome these shortfalls, GRC is currently developing advanced seals and seal preloading devices for the hypersonic engines of future space vehicles as part of NASA's Next Generation Launch Technology (NGLT) program.

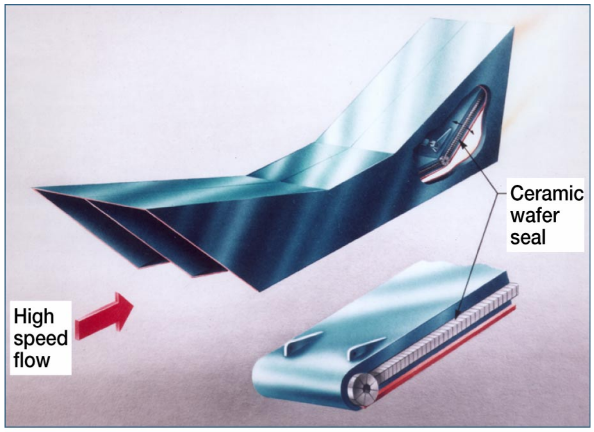

Figure 1.—NASP engine panel edge seals designed to seal the gaps between the movable ramps and adjacent engine sidewalls.

CHALLENGES AND OBJECTIVES FOR SEAL DEVELOPMENT

High temperature structural seals have been identified as a critical technology in the development of future space vehicles. Seals in hypersonic propulsion systems are expected to reach very high temperatures and operate in a chemically hostile environment in which oxidation and hydrogen embrittlement can occur. Analyses of panel-edge seals in the inlet of the NASP engine predicted seal temperatures of up to 2100 °F.[2] However, similar seals in the entrance region of the NASP engine combustor were expected to reach temperatures as high as 4900 °F at Mach 10. Even higher temperatures were possible if the vehicle stayed in scramjet mode for a longer time at higher Mach numbers. Because no existing materials could withstand those temperatures, active cooling of the seals was planned to ensure that they would survive in the combustor and nozzle regions. A cooling system inevitably comes with a weight penalty due to the plumbing lines, tanks, valves, and coolant gas associated with it. Ideally future propulsion system seals would operate at the flowpath temperature without coolant. However, the presence of steam and an oxidizing environment limits uncooled seal temperatures to the upper use temperatures of modern engineering ceramic materials. In regions where gas temperatures exceed material limits, some use of active cooling will be required.

In addition to operating at very high temperatures, seals must also remain resilient after repeated loading to ensure that they stay in contact with their sealing surfaces and restrict the flow of hot gases (Figure 2). Different techniques have been considered to keep seals in contact with their sealing surfaces including pressurized cavities, bellows, and springs behind the seals. As part of this effort, GRC is evaluating novel high temperature seal preloading devices to develop robust, reusable sealing systems that can operate at higher temperatures for longer periods of time while still remaining resilient. The objective is to produce a sealing system that would have a permanent set of less than 20 percent of its stroke at 2000+ °F. These advanced seals and preloading devices will be evaluated and their performance will be demonstrated through testing in simulated environments to ultimately develop seal systems that fill this technology gap.

SEAL DESIGN REQUIREMENTS

Hypersonic engine seals have a demanding set of design requirements as shown in Table 1. For near term applications in X-vehicle demonstrators, seal temperatures of 1600 °F are anticipated. As engine systems are developed for the final vehicles, seal temperatures are expected to increase to as much as 2000 to 2500 °F. To meet engine performance, safety, and life goals, the seals must withstand these extreme temperatures with minimal active cooling to limit the need for complex, heavy seal purge cooling systems. Inlet mass addition caused by purge cooling the seals could affect inlet performance and

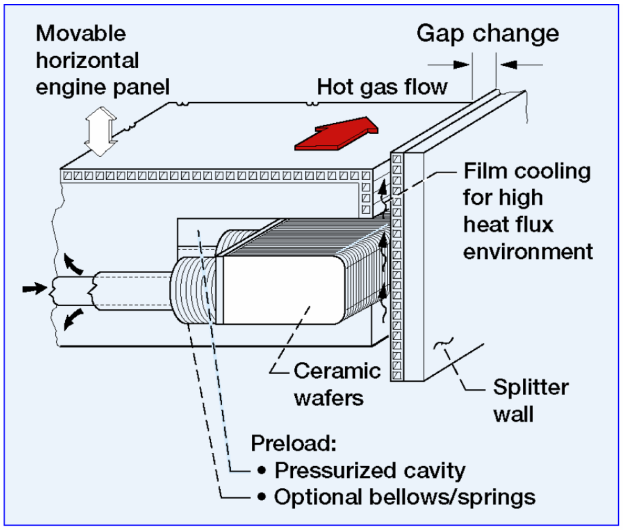

Figure 2.—Schematic of ceramic wafer seal designed to seal the gaps between the edges of movable ramps and adjacent engine sidewalls.

stability thereby increasing the chance of engine unstart. Because of this, hypersonic engine inlet designers would prefer inlet seals that do not require purge cooling. Furthermore, engine seals must limit the leakage of hot, pressurized (~100 psi) gases and unburned propellant into backside cavities to prevent explosive mixtures from forming there. The seals must operate in an oxidizing/steam environment and resist hydrogen embrittlement if hydrogen is used as a propellant. Structural and thermal loads on the engine sidewalls can cause distortions that the seals must accommodate. To stay in contact with the walls, the seals must remain resilient and flexible for multiple heating cycles. The seals will also be rubbed over these distorted, rough walls as the engine panels holding the seals are actuated. The seals must survive the hot scrubbing without incurring increases in leakage due to wear.

Table 1. Hypersonic engine seal design requirements.

DESIGN REQUIREMENT	GOAL
Seal temperature: near term for X-vehicle demonstrator	1600 °F
Seal temperature: longer term for final vehicle engine system	2000 to 2500 °F
Gas temperatures near seal	~5000 °F
Pressure drop across seal	100 psi (ref. 1)
Leakage	Minimize
Heat flux	Up to 2000 Btu/ft^2-sec
Environmental considerations	Oxidizing and steam environment; possible hydrogen embrittlement
Use of cooling: Inlet	Prefer to operate without active cooling
Use of cooling: Combustor and nozzle	Operate with minimal cooling
Flight time	~2500 sec (~42 min)
Reusability	TBD – nominally 10 to 100 cycles
Flexibility	Accommodate engine side wall non-uniformities and distortions
Resiliency	Accommodate seal gap openings and closings
Seal gap size	0.03 in. to 0.15 in.; gap variation of 0.15 in. over 18 in. span (ref. 1)
Sliding speed	2 in./sec
Wear resistance	Withstand scrubbing against rough surfaces

SEAL PRELOADING DEVICE REQUIREMENTS

The high temperature seal preloading devices that are being developed and evaluated would be installed behind the seals to ensure sealing contact with the opposing sealing surfaces (Figure 3). The requirements for these devices are also quite challenging. They must operate in the same environment and temperature as the seals while providing the required stroke (nominally 0.1 in.) with a permanent set of less than 20 percent of that stroke for multiple loading and heating cycles. Complicating this effort further is the limited amount of space available for the preloader behind the seals. The cross sectional area of the device must fit in a space that would be about 0.5 in. wide by about 0.5 in. high. Ideally the device would be about as long as the seal and able to be installed around corners. The device must be stiff enough to support the seal and keep it pressed against the sealing surface but soft enough that it does not apply excessive loads to that surface.

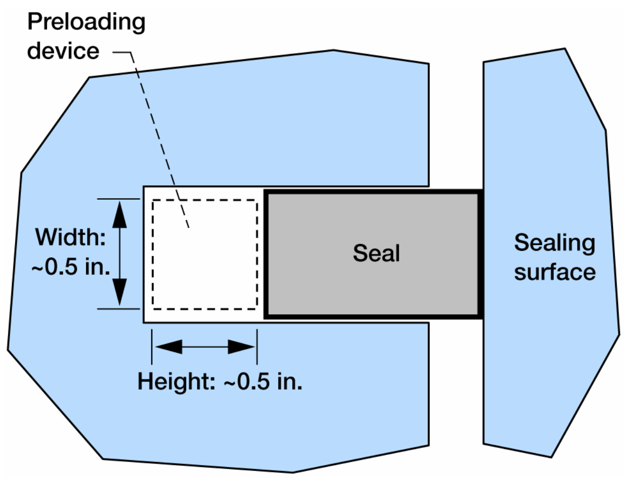

Figure 3.—Cross sectional model of seal showing location and space limitations for preloading device.

TEST APPARATUS AND PROCEDURES

SEAL SPECIMENS

Ceramic wafer seals were originally developed during the NASP program.[2,3] They are composed of a series of thin ceramic wafers installed in a channel in a movable panel and preloaded from behind to keep them in contact with the opposing sealing surface (Figures 2 and 3). Materials that were evaluated for the wafer seals during the NASP program included a cold-pressed and sintered aluminum oxide, a sintered alpha-phase silicon carbide, a hot-isostatically-pressed silicon nitride, and a cold-pressed and sintered silicon nitride.[4] Early wafers were made from silicon carbide and aluminum oxide. The corners of the silicon carbide wafers chipped off during static flow tests. It was believed that these chips originated from inclusions and impurities found near the wafer corners. Aluminum oxide wafers exhibited edge chipping during aggressive dynamic scrubbing and flow tests in a follow-on program. A detailed analytical comparison of all the materials that were considered ranked the advanced silicon nitride ceramics as the most promising material for future consideration.[4]

Given that these tests were performed in the late 1980’s, considerable improvements have been made since then to produce stronger and tougher ceramic materials. Because of these improvements and the high ranking of silicon nitride as a candidate wafer seal material, GRC selected silicon nitride as the best candidate for these seals. The wafers tested in the current study were made of monolithic silicon nitride (Honeywell AS800) and were 0.5-in. wide, 0.92-in. tall, and 0.125-in. thick. They had corner radii of 0.050 in. A series of tests were conducted to characterize the performance of these seals including room and high temperature compression and scrub tests and room temperature flow tests on the seals before and after the scrub tests.

SEAL PRELOADING DEVICE SPECIMENS

Two types of seal preloading devices were evaluated in this study. The first was a canted coil spring produced by Bal Seal Engineering Company, Inc. (Figure 4). These springs have several unique features that could make them very good seal preloading devices. Unlike typical compression springs that generate increasing amounts of force as they are compressed, the force produced by canted coil springs remains nearly constant over a large deflection range. This is an appealing feature for a seal preloading device because it could provide a large amount of stroke and resiliency to a seal without applying excessive loads to the seal or the opposing sealing surfaces. Another advantageous feature of canted coil springs is that they are produced in long, linear lengths that would allow them to be installed in a groove directly behind a seal and potentially around corners. Additionally, the part count would be far lower for a canted coil spring than for a typical compression spring because hundreds of compression springs would have to be lined up behind a long seal to accomplish what only a few canted coil springs could do.

The baseline canted coil springs evaluated in this study were Bal Seal part number 109MB-(84)L-2 and were made of 302 stainless steel (Table 2). Stainless steel springs were used to investigate the feasibility of this seal preloader concept. The work performed herein forms the basis for future studies where high temperature canted coil springs made of refractory materials would be investigated. A series of room temperature compression and flow tests were performed using this stainless steel spring design to evaluate it as a potential preloading device.

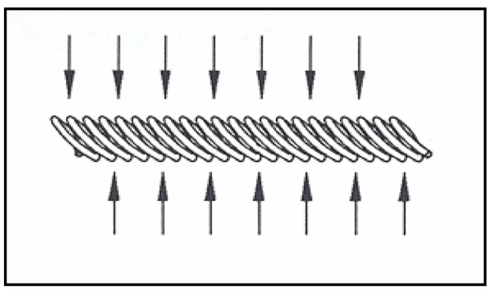

Figure 4.—Schematic of canted coil spring under transverse loading.

Table 2. Design specifications for seal preloading devices.

Spring design	**Material**	**Part number**	**Wire diameter, in.**	**Coil height, in.**	**Coil width, in.**	**Max. deflection,[a] in.**	**Max. load,[a] lbf**
BalSeal canted coil spring	302 stainless steel	109MB-(84)L-2	0.041	0.450	0.508	0.170[b]	
NHK standard compression spring	Silicon nitride	NCS2-025S	0.065	0.815	0.520	0.098	5.5
NHK modified compression spring	Silicon nitride	NCSS-02618DOB	0.065	0.720	0.435	0.043	5.5

[a]Per manufacturer's specifications.
[b]Before coil bind.

Another concept that was evaluated as a potential seal preloading device was a silicon nitride compression spring produced by NHK Spring Co., Ltd. Two different designs were tested: a standard spring and a modified design. Design specifications from the manufacturer for these springs are given in Table 2. Because they are made of silicon nitride, these springs have the potential to be used as high temperature seal preloading devices. Product literature claims that the strength of these springs will not decrease significantly at service temperatures up to 1832 °F and that the strength maintains a level of 29 ksi with a failure probability of 0.1 percent at up to 2192 °F.[5] A series of compression tests were performed on the springs at both room temperature and 2000 °F to evaluate these claims.

COMPRESSION TESTS

Compression tests were performed on the preloading devices and seals using a new state-of-the-art test rig at GRC. This test rig is capable of performing either high temperature seal compression tests or scrub tests at temperatures of up to 3000 °F using different combinations of test fixtures made of monolithic silicon carbide (Hexoloy α-SiC). The main components of this test rig are a servohydraulic load frame, an air furnace, and a non-contact laser extensometer (Figure 5). The load frame has a top-mounted actuator capable of generating a load of 3300 lb over a 6 in. stroke at rates from 0.001 to 8 in./sec. Both cyclic loading compression tests and stress relaxation tests can be performed. Tests can be performed in either load control or displacement control. In displacement control, feedback from the LVDT (linear variable differential transformer) inside the actuator controls actuator movement. When the LVDT is used in its narrowest calibration range of ± 0.5 in., it has an accuracy of ± 0.0017 in.

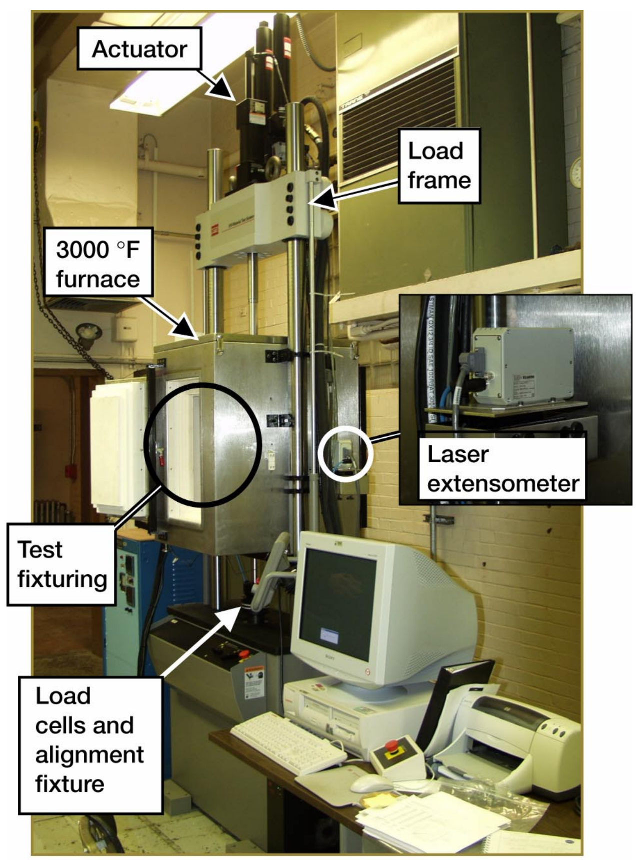

Figure 5.—Photograph of hot compression and scrub test rig showing main components: load frame, high temperature furnace, and laser extensometer.

The box furnace has a working volume that is 9 in. wide by 14 in. deep by 18 in. high. Test fixtures are configured inside the furnace so that the stationary base for each test setup sits on top of a loading rod on a load cell below the furnace. Two different load cell ranges are available, 500 lb or 3300 lb, depending on the seal that is being tested and the loads that are expected during a test. The 500 lb load cell has an accuracy of ± 0.15 lb (± 0.03 percent of full scale), and the accuracy of the 3300 lb load cell is ± 2.64 lb (± 0.08 percent of full scale). The load cells are used to measure compressive loads applied to the seals during a compression test or frictional loads on the seals during scrub testing.

Compression tests were performed inside the furnace using the test set up shown in Figure 6. These tests were performed to determine the resiliency and stiffness of the preloading devices and to generate load versus displacement (i.e., linear compression) data. Test specimens were installed into a holder that rested on the stationary base described above. A movable platen attached to the actuator was translated up and down to load and unload the test specimens. The laser extensometer was used to measure the amount of compression during testing (Figure 5). The laser system has a measurement range of up to 2 in. and an accuracy of ± 0.00025 in. Additional details for this test rig can be found in the paper by Dunlap, et al.[6]

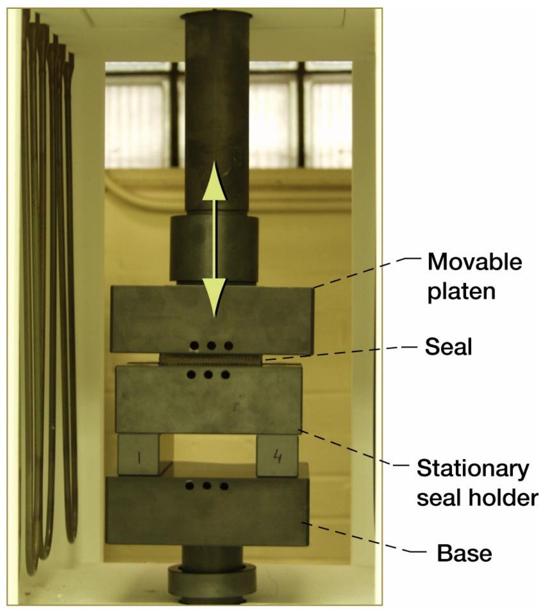

Figure 6.—Photograph of hot compression test fixture setup.

Compression tests were conducted at room temperature on the canted coil spring by itself and with a set of 31 wafer seals on top of a canted coil spring to see how the seals and spring performed together. A thin ceramic shim was placed in between the wafers and the spring to support the wafers during these tests. Tests were conducted on individual silicon nitride compression springs at both room temperature and at 2000 °F. Tests were also performed with 31 wafer seals on top of a set of silicon nitride springs (modified spring design) to see how they performed together. These tests were performed at 1600 °F so that the results would correspond with the hot scrub tests that were performed on the wafers at 1600 °F. Four springs were placed below the wafers on 1.15-in. centers. A thin load transfer element (0.02-in.-thick silicon carbide) was placed between the springs and the wafers to distribute the load from the four springs to the wafers. Primary and repeat tests were conducted for each test case. The exact amount of compression applied in each test is summarized in the Results and Discussion section.

Test specimens were typically loaded and unloaded for a total of 20 cycles for each test. The silicon nitride compression springs, however, were tested for 10 cycles. Each load cycle consisted of loading a test specimen at a rate of 0.001 in/sec to the specified amount of compression, holding at that compression level for 30 to 60 sec., and then unloading at 0.001 in/sec to the starting point. There was no hold time after the specimen was unloaded between load cycles. At the start of each test, the movable platen was lowered until it was in contact with the test specimen. For the tests on the canted coil springs, “contact” was defined when there was a load of 1 lb on the test specimen (or 0.25 lb/in. for a 4-in. specimen). A load of 0.1 lb was used to define contact for the tests on the NHK springs.

SCRUB TESTS

The main test rig that was used for the compression tests was also used to perform scrub tests on the seals using a different set of test fixtures (Figure 7). Tests were performed both at room temperature and at 1600 °F to evaluate seal wear rates and frictional loads as the seals were scrubbed against Inconel 625 rub surfaces. The rub surfaces had an average surface roughness before testing of 6 μin. in the scrubbing direction and 3 μin. in the transverse direction. The seals were installed in grooves in two stationary seal holders on either side of a pair of movable rub surfaces. The rub surfaces were assembled in a holder that was connected through the upper load train to the actuator. The gaps between the rub surfaces and the seals were set by spacer shims in front of and behind the seal holders. A gap size of 0.125 in. was used for these tests.

Four silicon nitride compression springs (modified spring design) were installed in the bottom of each seal groove to keep the wafer seals preloaded against both rub surfaces. As with the compression tests, the springs were installed on 1.15-in. centers below the wafers, and a load transfer element was placed on top of the springs to support the wafers and distribute the load from the springs. Thirty two wafers were installed into each seal holder to fill the 4-in.-long seal grooves. The amount of compression on the seals and springs (0.030 in.) was set through an interference fit between the seals and the rub surfaces resulting in a preload of about 2 lb per inch of seal.

During these tests, the seals were held in place in the holders while the rub surfaces were scrubbed up and down against them. For each load cycle a triangle wave was used with a stroke length of 1 in. in each direction and a stroke rate of 2 in./sec. There was no hold time between scrub direction changes. The seals were subjected to 1000 scrub cycles at 1 Hz for a total scrub length of 2000 in. for each test. Frictional loads were measured by the load cell under the furnace below the test fixture base. Seal wear rates were determined by examining the condition of the seals before and after each test and by measuring seal weight changes and changes in flow rates.

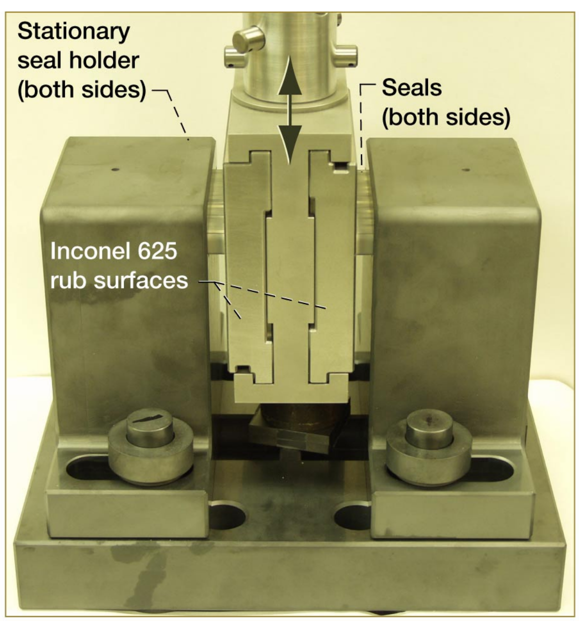

Figure 7.—Photograph of hot scrub test fixture setup.

The test temperature, rub surface material, number of scrub cycles, and scrubbing distance used for these tests were all based on requirements for a seal application in a candidate rocket-base combined cycle (RBCC) demonstrator engine. Seal temperatures in this demonstrator engine were predicted to be about 1600 °F, as compared to seal temperatures in the final vehicle engine system of 2000 to 2500 °F. In the demonstrator engine, the seals would be scrubbed against cooled Inconel 625 engine panels over a distance of 18 in. per mission. During engine checkout before each flight, the seals could be subjected to an additional 18 in. of scrubbing. After 25 missions this would result in a total seal scrubbing distance of 900 in. A safety factor of 2.2 was applied to achieve the 2000-in. total scrubbing distance that was used for these tests.

<u>FLOW TESTS</u>

Room temperature flow tests were performed in a linear flow fixture shown schematically in Figure 8. The flow fixture was designed so that seals of different diameters could be tested in removable cartridges that are inserted into the main body of the test fixture. Seals can be tested in this fixture with different seal gaps and under different amounts of linear compression.

Seals were tested in a groove in the seal cartridge shown in Figure 8. Tests were performed on the wafer seals with four silicon nitride springs (modified spring design) mounted on 1.15-in. centers preloading the wafers from behind using the load transfer element. Thirty wafers were used for each flow test so that the total seal length was 3.75 in. in the 0.502-in.-wide groove. Preload was applied to the wafers and springs through an interference fit between the seal and the cover plate. All tests were performed with a 0.135-in. seal gap and with the springs under nominally 0.030 in. of compression.

During testing, flow meters upstream of the flow fixture measured the amount of flow that passed through the test seal. The maximum capacity flow meter that was used had a range of 0 to 26.5 standard cubic feet per minute (SCFM) with an accuracy of 1 percent of full scale. A pressure transducer (0 to 100 psid, accuracy 0.074 percent of full scale) upstream of the test seal measured the differential pressure across the seal with respect to ambient conditions, and a thermocouple measured the upstream temperature.

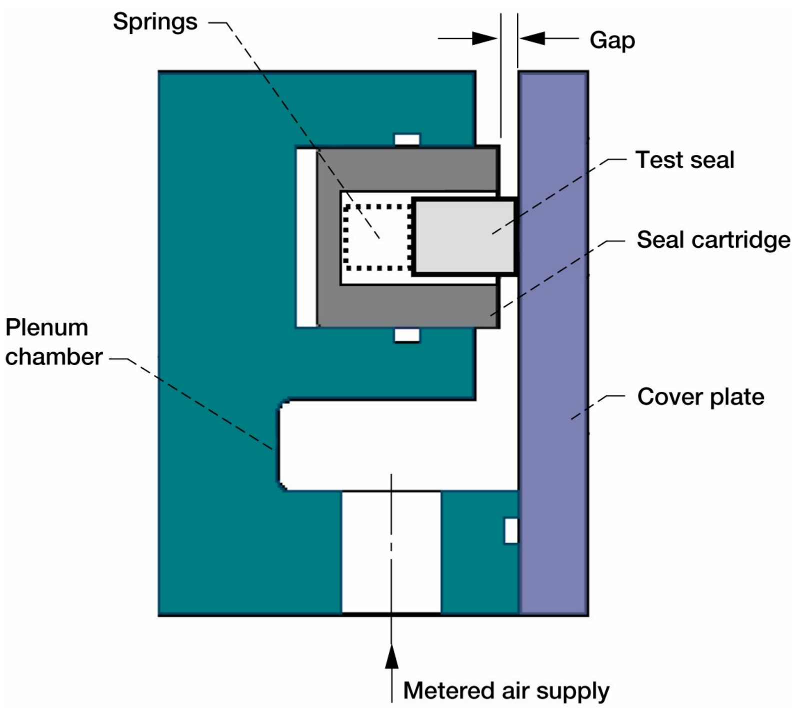

Figure 8.—Cross section schematic of flow fixture.

Tests were performed on as-received seals and on seals that had been scrub tested at 1600 °F. Primary and repeat tests were performed for each test case. More detailed descriptions of the hardware and procedure used to perform these tests can be found in the paper by Dunlap, et al.[6]

RESULTS AND DISCUSSION

COMPRESSION TEST RESULTS: CANTED COIL SPRINGS

In the room temperature compression tests on the canted coil springs, the springs were loaded to a linear compression of 0.220 in. A representative plot of the results is shown in Figure 9 for cycles 1 and 20 of a test. During these tests, there was little hysteresis in the data as the loading and unloading portions of the curves were similar. The springs became slightly stiffer with load cycling as cycle 20 reached a higher peak load than the peak for cycle 1. There were no signs of permanent set or relaxation in the springs after 20 load cycles.

The initial portion of the loading curve showed a gradual increase in force vs. linear compression up to a deflection of about 0.060 in. where the load leveled off at about 6 lbf/in. At this point, the curve flattened out and the force remained nearly constant until the spring deflection reached about 0.170 in. (38 percent of free height) and the coils began contacting each other. Over this 0.110 in. deflection range, the load slowly rose from 6 to 7 lbf/in. The force on the spring rose sharply beyond deflections of 0.170 in. This unique force vs. deflection curve is typical of a canted coil spring.[7] The large deflection range in which the load remained nearly constant makes canted coil springs appealing as seal preloading devices because they could provide a large amount of stroke and resiliency to a seal without applying excessive loads to the seal or the opposing sealing surfaces.

COMPRESSION TEST RESULTS: SEALS + CANTED COIL SPRINGS

Figure 9 also shows representative results for a room temperature compression test performed using a set of 31 wafer seals on top of a canted coil spring. This test was performed using a maximum

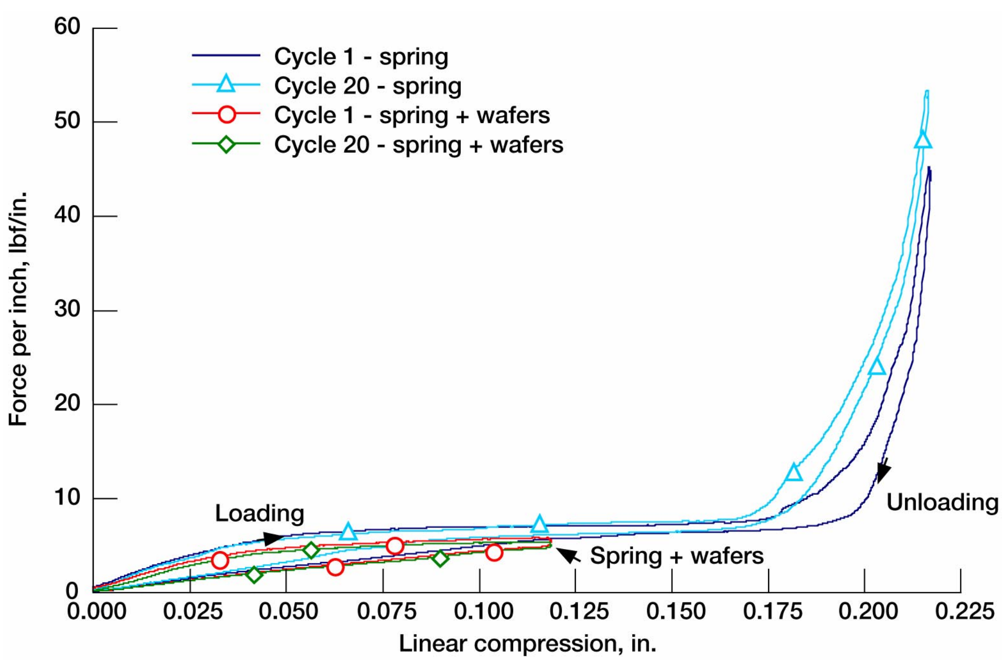

Figure 9.—Room temperature load versus linear compression data for canted coil spring by itself and for wafer seals on top of spring. (Note: load cycles other than 1 and 20 removed for clarity. Symbols are provided for identification only and do not represent actual data points.)

compression of about 0.120 in. The loading and unloading curves for this test were very similar to those for the spring by itself, and there was little hysteresis in the curves. The results of this test also showed no permanent set or loss of resiliency as load cycles 1 and 20 were almost identical.

This series of tests on stainless steel canted coil springs demonstrated the initial feasibility of using this type of spring as a seal preloading device. The authors recognize that the springs would have to be made out of a different material for applications at 2000+ °F.

COMPRESSION TEST RESULTS: SILICON NITRIDE COMPRESSION SPRINGS

Figure 10 shows the results of the compression tests performed on the silicon nitride compression springs. For the tests performed on the springs by themselves, both designs were loaded to a linear compression that was about 85 percent of the maximum deflection specified by the manufacturer to avoid breaking the springs (Table 2). The standard springs were subjected to a linear compression of 0.083 in., and the modified spring design was compressed 0.036 in.

Spring constants for both spring designs are shown in Figure 10 both at room temperature and at 2000 °F. The modified spring design had a spring constant of 65 lbf/in. at room temperature and 58 lbf/in. at 2000 °F, indicating that the springs were slightly less stiff at high temperatures. The elastic modulus of silicon nitride at 2000 °F is about 5 percent lower than it is at room temperature which helps explain this behavior. The standard spring design showed a different type of loading behavior, though. Its load versus linear compression curve at room temperature had two different regions. In the linear compression range up to about 0.040 in., the standard spring had a spring constant of about 28 lbf/in. From 0.040 in. to 0.083 in., the spring became stiffer with a spring constant of 46 lbf/in. This type of behavior did not occur during the test at 2000 °F, though, as the spring constant remained at 28 lbf/in. throughout the test.

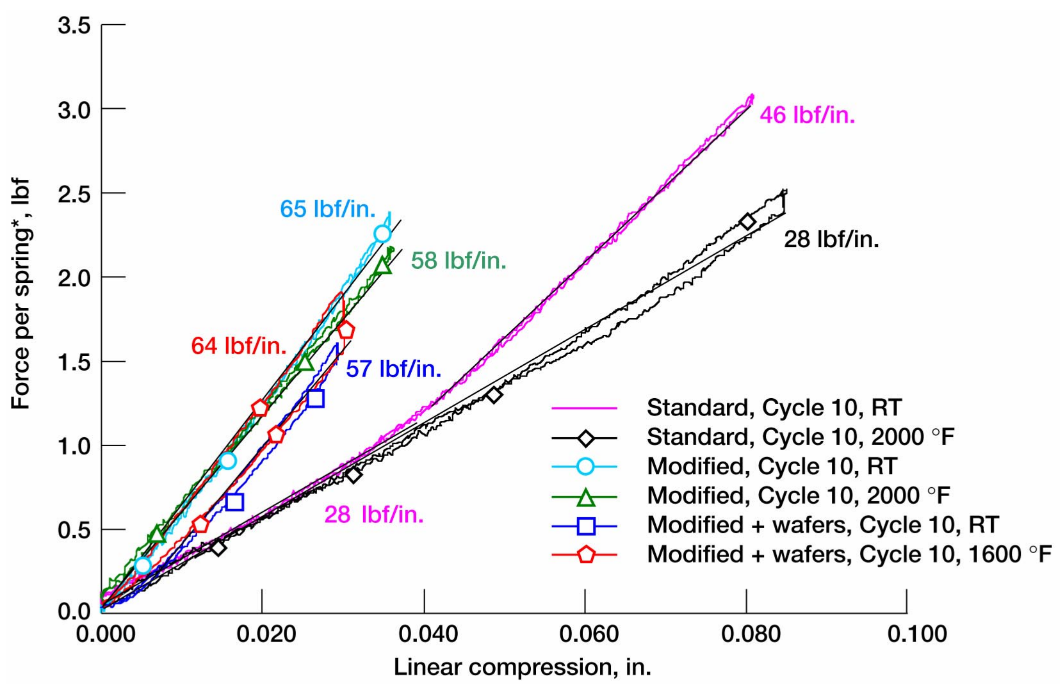

Figure 10.—Load versus linear compression data for silicon nitride compression springs by themselves and for wafer seals on top of springs. (Note: load cycles other than 10 removed for clarity. Symbols are provided for identification only and do not represent actual data points.) *Four springs were tested under the wafer seals, while only one spring was used for other tests.

For all of the tests performed on the silicon nitride springs by themselves, there was very little hysteresis in their load versus linear compression data. In each load cycle, the loading and unloading portions of the curve were almost identical. There was also no permanent set or relaxation in these springs either at room temperature or at 2000 °F. For clarity, Figure 10 only shows the curves for cycle 10 of each test because they were almost identical to the curves for all other load cycles.

COMPRESSION TEST RESULTS: SEALS + SILICON NITRIDE COMPRESSION SPRINGS

In addition to showing data for the silicon nitride compression springs by themselves, Figure 10 also shows the results for the compression tests performed on a set of 31 wafer seals supported by the load transfer element and four silicon nitride springs (modified design). These tests were performed up to a maximum compression of 0.030 in. and at room temperature and 1600 °F to correspond to test conditions during scrub testing.

The room temperature tests generated a spring constant of 57 lbf/in. that was close to the spring constant for the spring by itself at room temperature. For the test at 1600 °F, the spring constant during the loading portion of the curve increased slightly to 64 lbf/in. This was similar to the spring constant for the spring by itself at 2000 °F.

The main difference between the tests performed at room temperature and 1600 °F was that there was virtually no hysteresis for the room temperature tests but a small amount for the tests at 1600 °F (Figure 10). It is possible that during the high temperature test, there was some small amount of friction between the wafers and the side walls of the seal groove that caused this hysteresis as the wafers and springs were unloaded during each load cycle. Although there was some hysteresis at 1600 °F, there was no permanent set or relaxation in these springs for any of the tests. This is to be expected because the wafers are solid blocks of silicon nitride, and the springs by themselves did not exhibit any permanent set during testing up to 2000 °F. These results show that the silicon nitride springs show promise for use as high temperature seal preloading devices.

The tests performed on the silicon nitride compression springs were “proof of concept” tests that showed that it is possible for a spring-like device to operate at high temperatures while still maintaining its load-bearing capabilities and resiliency. The specific spring designs that were evaluated did not meet all of the requirements described earlier for potential seal preloading devices in that they were taller than the requirements and did not provide the desired 0.1 in. of stroke. It is also likely that heavier duty springs would be required for applications in which higher loads are anticipated. Although these springs did not meet all of the requirements, their designs have not yet been optimized.

SCRUB TEST RESULTS

Peak frictional loads during the down stroke of each scrub cycle are presented in Figure 11 for both the room temperature scrub test and the test performed at 1600 °F. During the room temperature scrub test, the frictional loads started around 6 lbf at the beginning of the test and gradually rose as the test proceeded until they reached about 15.5 lbf by the end of the test. The seals were installed so that the springs behind them provided a load against the Inconel 625 rub surfaces of about 2 lbf/in. over both 4-in. seal lengths. This resulted in a normal load of 16 lbf during testing. Based on this normal load, the friction coefficient during the room temperature scrub test rose from about 0.4 to almost 1.0 by the end of the test.

After the room temperature scrub test was completed, the seals and rub surfaces were inspected for signs of damage. Figure 12 shows what the seals looked like before and after scrubbing and what the Inconel 625 rub surface looked like after the test. The seals showed little if any damage after testing. Wear debris from the rub surface can be seen on some of the wafers in locations that correspond to areas on the rub surface that were worn during the test. None of the wafers were chipped or broken during testing, and the total weight of both wafer sets before and after testing was almost identical.

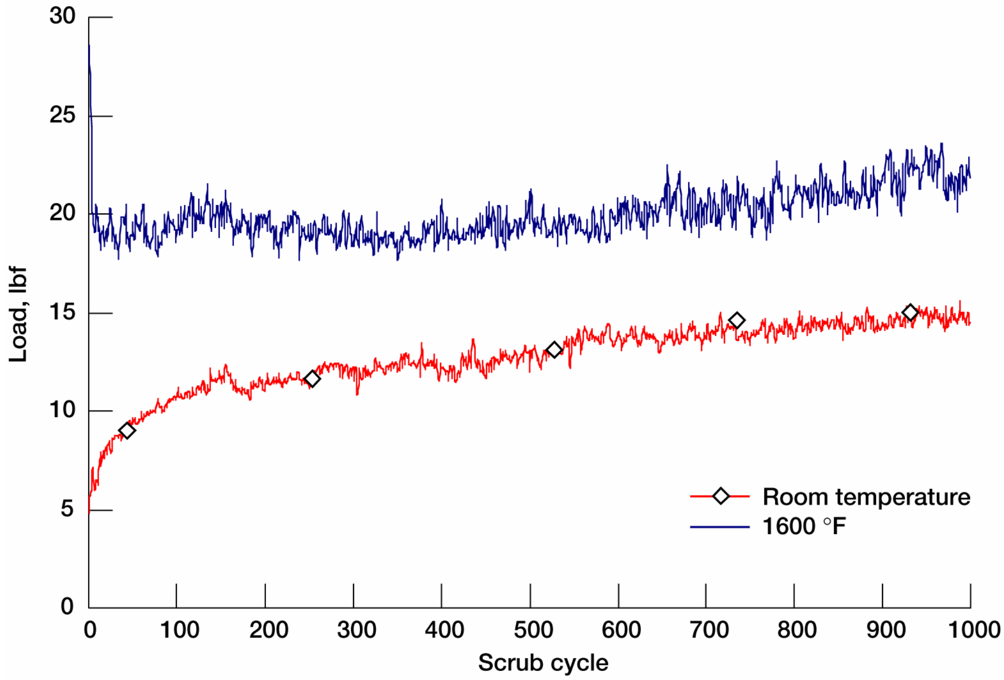

Figure 11.—Peak frictional loads during down stroke of each scrub cycle for scrub tests performed at room temperature and 1600 °F. (Note: symbols are provided for identification only and do not represent actual data points.)

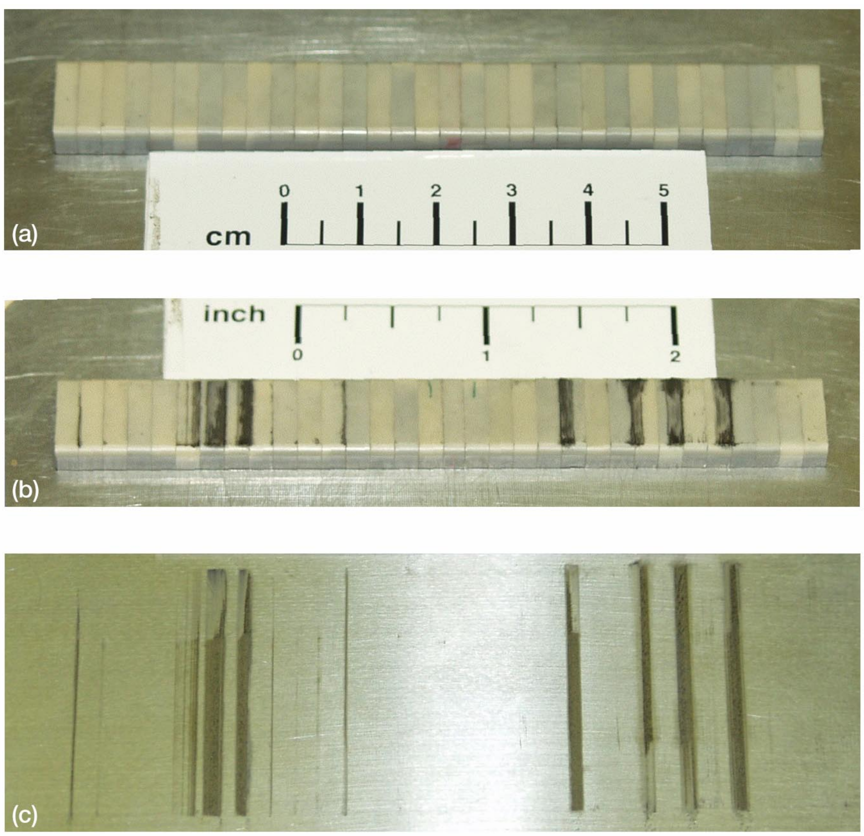

Figure 12.—Photographs of seals (a) before and (b) after room temperature scrub test and (c) Inconel 625 rub surface after testing.

Before this scrub test, the average surface roughness of the rub surfaces was about 6 μin. in the scrubbing direction and 3 μin. in the transverse direction. After testing, the surface roughness had risen to a range of 6 to 43 μin. in both directions. This increase in surface roughness during testing likely contributed to the increase in frictional forces as the test proceeded.

Frictional loads recorded during the down stroke of each scrub cycle for the test performed at 1600 °F are also shown in Figure 11. While the loads started lower and then gradually increased during the room temperature test, they behaved somewhat differently during the hot test. The frictional loads recorded during the first few scrub cycles were as high as 28.5 lbf before quickly dropping to around 20 lbf by the fifth cycle. From there they rose very slowly to a peak of about 23.5 lbf by the end of the test. The seals in this test were installed in the same manner as those for the room temperature test resulting in a total normal load of about 16 lbf. This means that the friction coefficient for this test ranged from about 1.25 during the fifth scrub cycle to about 1.5 by the end of the test.

The high loads recorded for the first few scrub cycles were likely influenced by the relatively long heat up time before the furnace temperature reached 1600 °F. To avoid subjecting the silicon carbide test fixtures to thermal shock and large temperature gradients, the furnace was heated at a rate of 400 °F per hour until it reached 1600 °F after 4 hours. During this time, the rub surfaces oxidized in the areas around the seals. Surface roughness measurements after the test in areas of the rub surface that were not scrubbed were in the range of 10 to 23 μin. as compared to less than 6 μin. before testing. For the first few scrub cycles of the test, the seals scrubbed over this slightly rougher oxide layer and generated higher frictional forces. After the first few cycles, the scrubbing may have smoothed the oxide layer and transferred some of it to the wafers resulting in lower frictional loads later in the test. This could explain why the frictional forces dropped off after the first few scrub cycles and leveled off.

Figure 13 shows photographs of the seals before and after 1600 °F scrubbing and the Inconel 625 rub surface after the test. As with the room temperature test, the seals showed few signs of damage,

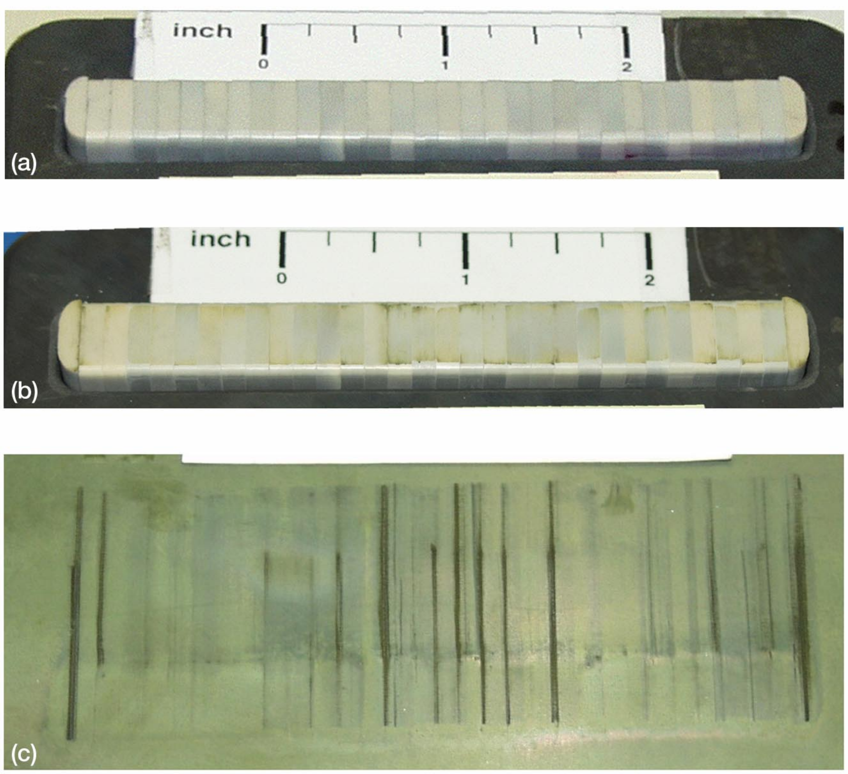

Figure 13.—Photographs of seals (a) before and (b) after 1600 °F scrub test and (c) Inconel 625 rub surface after testing.

and none of them were chipped or broken during the test. The total weight of both wafer sets did not change before and after testing. Some wear debris can be seen on the wafers in locations that correspond to areas on the rub surface that were worn during the test. The surface roughness of the rub surface increased during the test to a range of 6 to 34 μin. in both directions. This is comparable to what was measured after testing for the rub surfaces used for the room temperature scrub test.

Overall, the silicon nitride wafers performed well during these tests. They survived both tests completely intact with no signs of chipping or other damage. Silicon carbide wafer seals tested during the NASP program were much more damage-prone and chipped during static flow testing even without scrubbing. The silicon nitride wafers appear to be much more robust and damage-resistant.

FLOW TEST RESULTS

Flow test results for the wafer seals before and after scrub testing are presented in Figure 14. These tests were performed with four silicon nitride springs installed behind the wafers to keep them preloaded against the cover plate. Flow rates for the wafers before and after scrubbing were almost identical in both cases. This is consistent with the observation that the wafers were not damaged during either scrub test. These results are encouraging because they show that the seals are still effective at blocking flow even after 1000 scrub cycles at 1600 °F.

Figure 14 does show a difference in flow behavior between the wafers scrubbed at room temperature and those scrubbed at 1600 °F. Between these two test sequences, it was noticed that the wafers were not as uniform height-wise (i.e., 0.92-in. dimension) as desired. The wafers that were flow tested before and after the room temperature scrub tests exhibited height variations of approximately 0.001 in. To determine the effects of tighter dimensional height control, a new set of wafers was individually measured and sorted so that their heights varied by only 0.0005 in. As shown in Figure 14, this improvement in control of wafer heights resulted in about a three-fold reduction in flow rates.

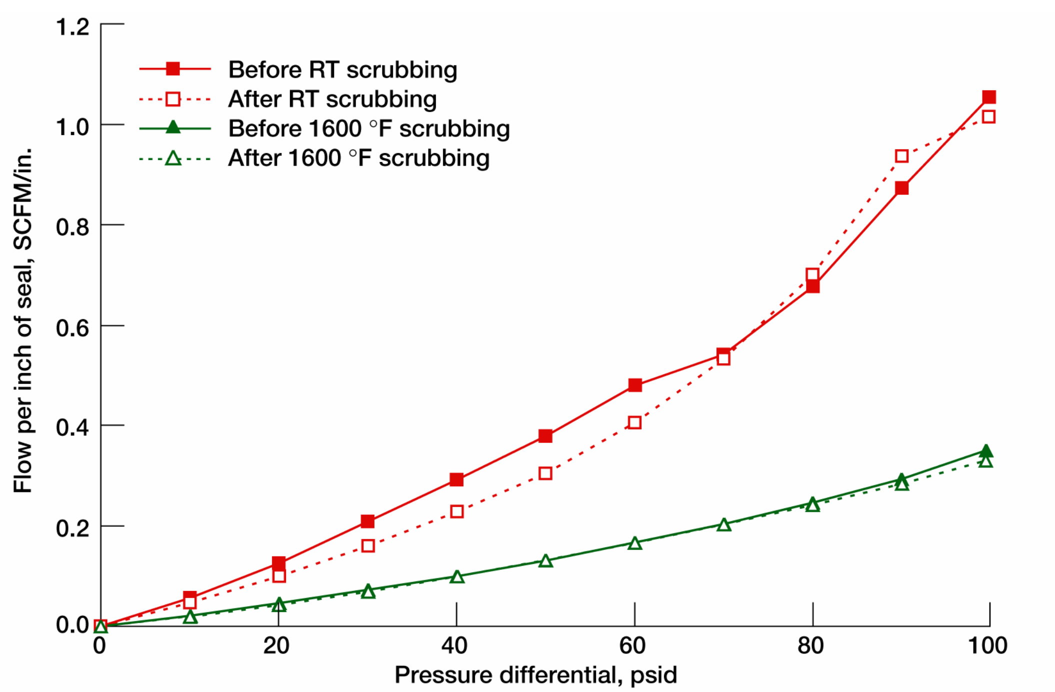

Figure 14.—Wafer seal flow rates versus pressure differential before and after scrub testing for a gap size of 0.135 in. Results shown for seals scrub tested at room temperature (nominal 0.001-in. wafer height tolerance) and 1600 °F (nominal 0.0005-in. wafer height tolerance).

The wafer seal flow rates presented in Figure 14 are much lower than those for braided rope seals tested under the same conditions.[6] For example, at a pressure differential of 100 psid the flow rate for a 0.6-in.-diameter braided rope seal with a core of uniaxial ceramic fibers (AC1 design) was 11 SCFM/in. with the seal under 20 percent compression. For the same setup, a 0.565-in.-diameter seal with a core composed of smaller braided rope seals braided together (BC1 design) had a flow rate of 22 SCFM/in. Both of these flow rates are much higher than those for the wafer seals. Compared to the flow rates at 100 psid for the wafers scrub tested at room temperature, the flow rate for the AC1 design was over 10 times higher and the rate for the BC1 design was 21 times higher. If the results for the wafers scrub tested at 1600 °F are used as the basis for comparison, the AC1 design rate was 32 times higher and the BC1 rate was 64 times higher. In all cases, the wafer seals were much more effective at blocking flow than the braided rope seals were.

CANTED COIL SPRING ANALYSES

In this study, tests were performed on canted coil springs made of stainless steel to evaluate their performance at room temperature and assess the feasibility of using this type of spring as a seal preloading device. While the results of these tests were promising, the springs will need to be made out of high temperature materials for future applications in which the temperature of the seals and preloading devices will reach 2000 to 2500 °F. Researchers at GRC have several efforts currently underway to develop high temperature canted coil springs for use as seal preloading devices. In one of these efforts, a series of analyses is being performed to determine if canted coil springs made of coated refractory metals can provide the required stroke and load at high temperatures.[8]

A finite element model of a canted coil spring was created using ANSYS (Figure 15). The model was three-dimensional, and each cross section through the spring was composed of 48 eight-noded solid brick elements. A total of five spring coils were included in the model so that the force versus displacement of the spring could be accurately modeled while maintaining reasonable solution times.

Different test cases were run in which the bottom of the spring was constrained while prescribed displacements were applied at the top. During each run, the two cut faces at the ends of the spring were constrained in the X direction, the bottom node of each coil was constrained in the Y direction, and the

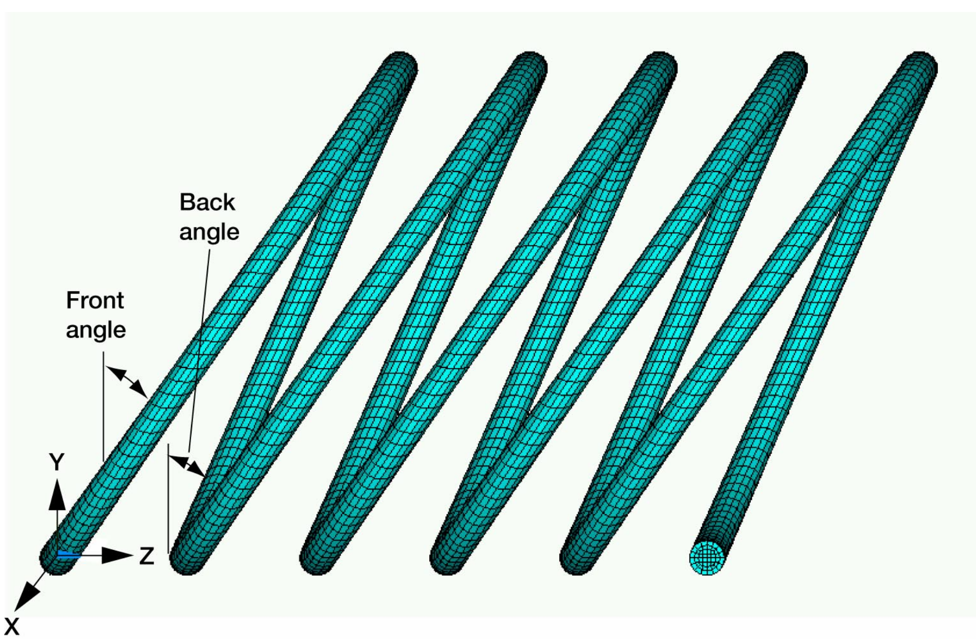

Figure 15.—Finite element model of canted coil spring.

center node of each coil at the bottom was constrained in the Z direction. Displacements were applied to the top node of each coil and the resulting forces were monitored.

To validate this modeling approach, the canted coil spring described in Table 2 was modeled, and load versus displacement curves were generated and compared to the compression test results at room temperature for this spring. The analysis was run to a displacement of 0.140 in. This was more than half way into the flat portion of the loading curve shown in Figure 9. Figure 16 shows that the analytical predictions for load versus displacement compared favorably to the experimental test results, and there was good agreement in both the curve shapes and magnitudes.

After validating the model against experimental data for the stainless steel canted coil spring, the model was used to simulate the performance of other spring materials and geometries. Refractory metals were evaluated for spring applications above 2000 °F including Mo-47.5Re (molybdenum rhenium) and Mo-0.5Ti-0.08Zr (TZM). At high temperatures, these materials would be coated to prevent oxidation. Coating materials under consideration include rhodium and iridium. Analyses were conducted at elevated temperatures including 2000, 2300, and 2500 °F using material properties at those temperatures. Other spring geometries beyond the baseline stainless steel design were also evaluated in an effort to keep the maximum predicted stresses below each material's high temperature yield strength while still providing a stroke of at least 0.1 in. Parameters that were varied included wire diameter, spring front and back angles, and coil height and width.

Figure 17 shows example stresses predicted for an analysis case run on a spring made of TZM using material properties at 2000 °F. Several parameters were varied for this case as compared to the baseline stainless steel spring design shown in Table 2. The wire diameter was decreased to 0.025 in., and the coil height and width were decreased to 0.4375 and 0.490 in., respectively. The front and back angles were increased to 40 and 28.5°, respectively. Figure 17 shows that this geometry resulted in a maximum predicted stress of 67.9 ksi at a deflection of 0.1 in. This stress is below the yield strength of 72 ksi for TZM at 2000 °F, so it appears that the spring would be able to provide the desired deflection at 2000 °F without yielding the TZM wire material. While the stresses and deflection for this design were acceptable, the loads generated were lower than the design criteria. This design produced a force of about 0.6 lb/in. at a deflection of 0.1 in. as compared to the goal of 2 lb/in. Although this load was below the goal, this spring design has not yet been optimized.

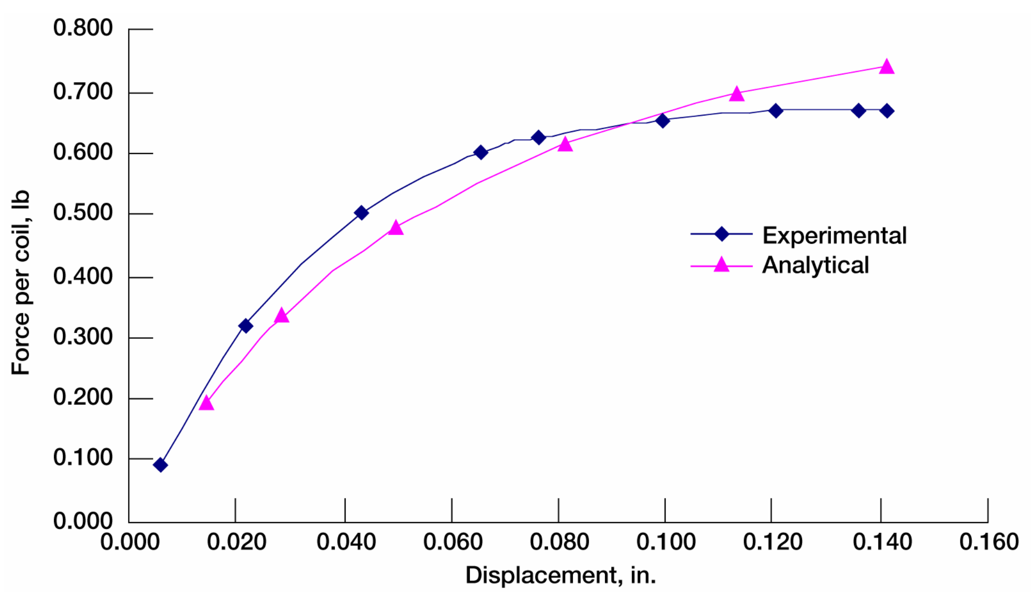

Figure 16.—Analytical predictions for canted coil spring load versus displacement compared to experimental results.

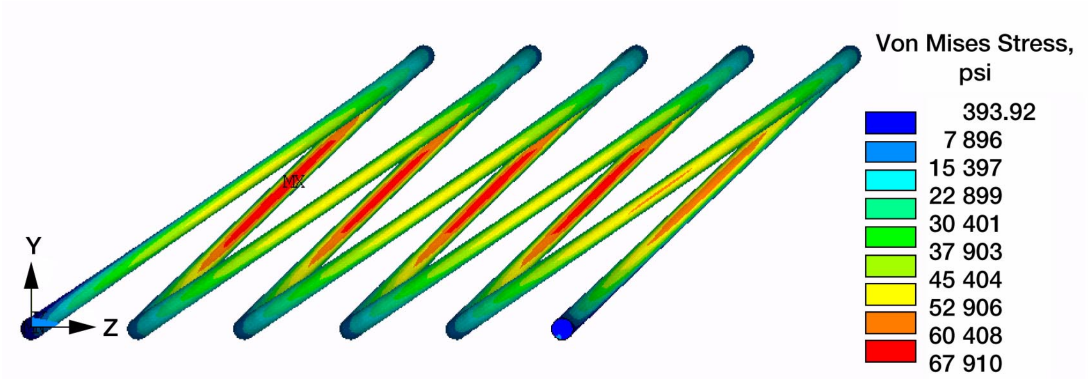

Figure 17.—Stresses predicted for a representative TZM canted coil spring at 2000 °F for 0.1 in. deflection.

SUMMARY AND CONCLUSIONS

NASA GRC developed a variety of high temperature structural seals during the NASP program including ceramic wafer seals. However, those seals fell short of some design goals and could not be adequately matured due to program termination. Requirements for future advanced hypersonic engines are similarly very demanding, and state-of-the-art seals do not meet these requirements. To address these shortfalls, GRC is developing advanced seals under NASA's NGLT program. Shortfalls that were investigated in the current study included a loss of seal resiliency with load cycling at high temperatures, seal durability, and seal flow blocking ability. In an effort to address these shortfalls, modern ceramic wafer seal materials and two types of seal preloading devices were evaluated in a series of compression, scrub, and flow tests. Based on the results of these tests, the following conclusions were made:

1. Silicon nitride wafer seals were more robust and damage-resistant than wafers that were tested during the NASP program. There were no signs of damage after the silicon nitride wafers were scrubbed against Inconel 625 rub surfaces at room temperature and 1600 °F for 2000 in. of scrubbing (1000 cycles). None of the wafers were chipped or broken, and the total weight of each wafer set before and after testing was almost identical. Friction coefficients between the wafers and the Inconel 625 rub surfaces were rather high, though, so work must be done to find ways to lower these frictional forces.

2. Silicon nitride wafer seals were excellent at blocking flow even after 1000 scrub cycles at 1600 °F. Flow rates for the wafers before and after scrubbing were almost identical and were up to 32 times lower than those recorded for the best braided rope seal flow blockers.

3. Canted coil springs are promising seal preloading devices. Room temperature compression tests performed with wafer seals on top of a spring showed that the spring met the stroke requirement with no permanent set or loss of resiliency for 20 load cycles. These feasibility tests were performed on springs made of stainless steel. High temperature materials will need to be used for applications at 2000+ °F.

4. A finite element model of a canted coil spring has been validated against experimental test results with good agreement in both the load versus displacement curve shapes and magnitudes. Use of this parametric model to evaluate high temperature wire materials and geometries produced at least one canted coil spring design that met stroke requirements with stresses below the material yield strength at 2000 °F.

5. Silicon nitride compression springs also show promise as high temperature seal preloading devices. After repeated loading at temperatures up to 2000 °F the springs showed little hysteresis and excellent resiliency.

FUTURE WORK

The results of these tests indicate that a series of silicon nitride wafer seals on top of a high temperature canted coil spring could provide a durable, resilient sealing system that blocks flow effectively at 2000+ °F. More work needs to be done, though, to investigate seal and preloading device combinations that ultimately satisfy all of the seal requirements. The authors plan to investigate other wafer shapes and sizes to see if those changes affect seal durability and frictional forces. Longer scrub tests will also be performed at high temperatures to examine seal durability for more than 1000 scrub cycles. Additional work is planned to optimize the design of the high temperature canted coil spring prior to producing prototypes for evaluation.

REFERENCES

1. Steinetz, B.M., DellaCorte, C., Machinchick, M., Mutharasan, R., Du, G., Ko, F, Sirocky, P.J., and Miller, J.H., “High Temperature Dynamic Engine Seal Technology Development,” NASA TM–105641, April 1992.

2. Steinetz, B.M., “Evaluation of an Innovative High Temperature Ceramic Wafer Seal for Hypersonic Engine Applications,” NASA TM–105556, February 1992.

3. Steinetz, B.M. and Sirocky, P.J., “High Temperature Flexible Seal,” United States Patent 4,917,302 issued April 17, 1990.

4. Steinetz, B.M., “Evaluation and Ranking of Candidate Ceramic Wafer Engine Seal Materials,” NASA TM–103795, May 1991.

5. Silicon Nitride Ceramic Spring product literature, NHK Spring Co., Ltd., KK9502-6-1P, pp. 3–4.

6. Dunlap, P.H., Steinetz, B.M., DeMange, J.J., and Taylor, S.C., “Toward an Improved Hypersonic Engine Seal,” NASA TM–212531, AIAA–2003–4834, July 2003.

7. Canted Coil Spring product literature, Bal Seal Engineering Company, Inc., Catalog No. 3.1E, p. 2.

8. Final report for Base Package of Contract NAS3–03114 with Refractory Composites, Inc.

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

Durable, flexible sliding seals are required in advanced hypersonic engines to seal the perimeters of movable engine ramps for efficient, safe operation in high heat flux environments at temperatures of 2000 to 2500 °F. Current seal designs do not meet the demanding requirements for future engines, so NASA's Glenn Research Center is developing advanced seals and preloading devices to overcome these shortfalls. An advanced ceramic wafer seal design and two types of seal preloading devices were evaluated in a series of compression, scrub, and flow tests. Silicon nitride wafer seals survived 2000 in. (1000 cycles) of scrubbing at 1600 °F against an Inconel 625 rub surface with no chips or signs of damage. Flow rates measured for the wafers before and after scrubbing were almost identical and were up to 32 times lower than those recorded for the best braided rope seal flow blockers. Canted coil springs and silicon nitride compression springs showed promise conceptually as potential seal preloading devices to help maintain seal resiliency. A finite element model of the canted coil spring revealed that it should be possible to produce a spring out of high temperature materials for applications at 2000+ °F.

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