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The object of the work discussed in this paper was to develop a reliable helium gas shaft seal for use in an electric motor-driven, liquid oxygen pump on a space vehicle. The development effort covered tests on two basically different face seal designs, one with an attached carbon face and the other with a floating lap-fitted carbon face. Several bellows vibration damping devices and various seal material combinations were investigated.

INTRODUCTION

Liquid oxygen is one of the more active cryogenic fluids. Under proper conditions it will react with the common combustible materials, and under certain conditions, such as added energy input, it will react with metallic construction materials. This is an important consideration in the design of equipment for use in liquid oxygen applications. It is especially important in the design of rotating machinery, for example electric motor-driven pumps. In this type of equipment the resultant energy input due to a possible electrical overload or mechanical shock may be sufficient to initiate a mild reaction or even a violent detonation.

Electric motor-driven liquid oxygen pumps have operated successfully, under normal conditions, with all parts completely submerged and wetted by liquid oxygen. But in applications which may present a possible hazard to human life, the safety aspects can be enhanced by additional design precautions. In the electric motor driven pump for instance, even though all materials are selected for maximum compatibility with liquid oxygen, the motor can be enclosed in a helium gas inerted container. A design of this type, of course, will require rotating shaft seals. The selection and testing of a suitable helium seal for use in an electric motor driven liquid oxygen pump for a manned space vehicle was the objective of this investigation.

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PUMP DESIGN

Electric motor-driven, liquid oxygen pumps can be designed with a flooded, canned, or sealed motor. Shaft seals are not required in the first two types of units, but one or more are necessary with the sealed type motor. The latter type of unit is discussed here together with the test work conducted in developing a satisfactory shaft seal.

From the design standpoint, the flooded motor unit is the most simple. All motor components operate in direct contact with the pumped fluid and no seals are required. But, from a safety standpoint, this design could be the most hazardous. While all materials are selected for compatibility with liquid oxygen, combustion is still possible under certain conditions. For instance, in a simulated short circuit test of a motor stator submerged in liquid oxygen, the electrical insulation, part of the copper windings, and iron stator laminations were burned away, as shown in Fig. 1. Combustion of these materials was terminated only when the supply of oxygen was exhausted.

In the canned motor design, usually only the stator laminations, windings and leads are hermetically sealed within a stainless-steel shell, thus preventing direct contact with the liquid oxygen. The rotor, however, is normally still submerged directly in the liquid. In this design the safety aspects of the stator with its electrical insulation are improved. However, the presence of the stainless steel stator shell in the motor air gap reduces the motor efficiency and increases the motor operating current.

For the unit discussed here the required current was increased by approximately 20 percent when a canned stator design was tested. This figure would be further increased if the rotor was also canned.

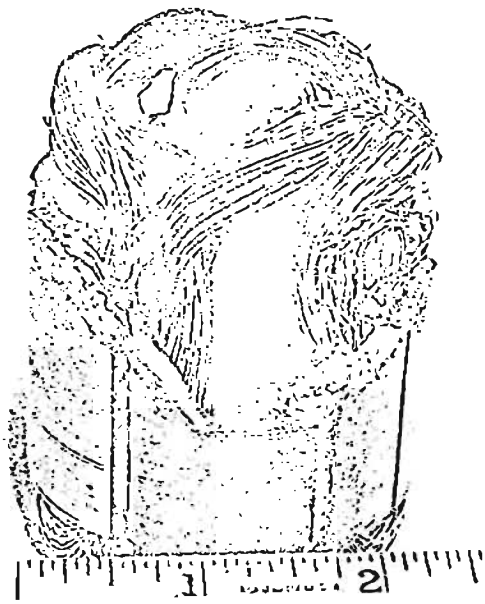

Fig. 1—Electric Motor Stator After Simulated Short Circuit Test In Liquid Oxygen.

In the sealed motor design all motor parts operate within a housing inerted with pressurized helium gas. This design presents a minimum safety hazard. An example of this design is shown in Fig. 2. This is an electric motor driven liquid oxygen pump unit for use on a space vehicle. The helium pressurized motor is separated from the pumped fluid by a helium seal and a liquid oxygen seal operating in a back-to-back arrangement with a common overboard vent between them.

The pump unit is driven by a one horsepower electric motor operating at 11,000 rpm from a three-phase A.C., 400 Hz, power source at a supply voltage of approximately 40 V.R.M.S. line-to-line. The unit is approximately 12" long, 4" in diameter and has an integral mounting flange 10 inches in diameter. Weight of the unit is approximately 15 pounds.

On the space vehicle the unit is flange mounted in a bottom opening of a liquid oxygen supply tank and, except for the outside face of the flange, is totally submerged in liquid oxygen at −297°F. This cryogenic cooling permits a motor design of smaller size and weight and of improved efficiency due to the reduced copper losses in the stator windings. Normally, the motor cavity is inerted with helium gas at a pressure of approximately 50 psig, but this pressure can go as high as 80 psig, which is limited by the motor cavity relief valve.

SEAL CONSTRUCTION

Because of the cryogenic operating environment, elastomeric sealing elements are not usable. Therefore, an all metal welded bellows seal design is employed as shown in Fig. 3. This is a cartridge type seal which is shrink fitted directly into the aluminum pump housing. Static sealing is provided by the seal cartridge shrink fit in the pump housing and by the metallic bellows.

A loose or unattached carbon face piece is used with this seal. The back side of the carbon face piece is lap-fitted to the bellows end plate to provide an effective static seal at this point. The dynamic or operating surface of the carbon face is of the gas face type consisting of two concentric lands. The inner land is continuous and performs the pressure sealing function, while the outer is a segmented bearing land which serves to reduce seal face pressure. Rotation of the carbon face piece is prevented by slots, in the O.D. of the carbon face, which engage with radial keys located in the I.D. of the seal cartridge.

Compared to a seal having an integral type carbon face piece, the loose face piece type seal has the following advantages:

1. Seal face distortion due to differential thermal contraction of seal materials is minimized.
2. Vibration damping is achieved by friction between the face piece and keys.

The carbon face piece operates against a rotating ring clamped axially on the shaft and statically sealed to the shaft by aluminum compression gaskets.

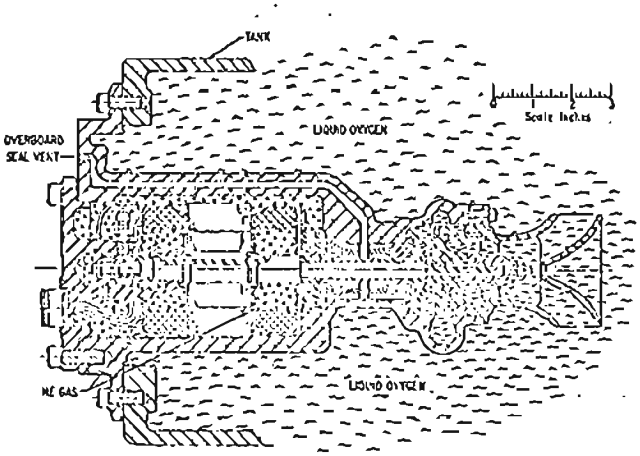

Fig. 2—Liquid Oxygen Pump With Helium Inerted Motor.

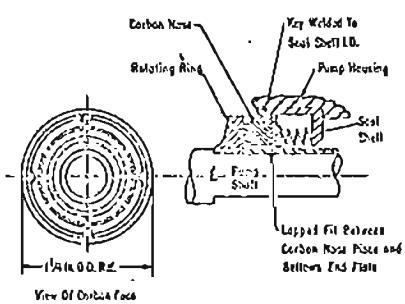

Fig. 3—Helium Bellows Seal With Loose Carbon Face.

Seal materials are as follows:

Seal cartridge including bellows	—718 Stainless Steel
Carbon face piece	—P5N carbon
Rotating ring	—Hard chrome plate on 440C Stainless Steel (Annealed)

The 400 series stainless steel is used in preference to a 300 series because of its higher thermal conductivity. The chrome plate thickness is 0.0015-0.005" as plated and 0.001" minimum after lapping.

SEAL CHARACTERISTICS

Significant seal characteristics are listed in Table 1.

The seal face pressure of 10 psi consists of 7.5 psi due to bellows spring pressure and 2.5 psi resulting from the 55% seal hydraulic overbalance at 50 psig helium gas operating pressure. The maximum allowable seal friction torque of 10 oz. in. is governed by the motor torque remaining after all other pump requirements have been satisfied. It is influenced to a large extent by the motor starting current limit which governs the motor torque capability.

The maximum permissible seal leakage rate is 25 standard cubic inches per minute (SCIM) of helium gas at a motor cavity pressure of approximately 50 psig. Actual seal leakage experienced during testing is about 2 SCIM dynamically and 20 SCIM statically, i.e., with the unit non-operating. It is interesting to note that the dynamic leakage is much lower than the static leakage.

The transition from the dynamic to the static leakage rate takes place in approximately 10 to 40 seconds after the pump has come to rest following power shut-off. The seal leakage increases to the peak static value at which it remains for a period of 30 seconds to 3 minutes. The leakage then slowly decays to some rate between the seal dynamic and maximum static rates. This characteristic is repeatable on successive pump tests.

The pump unit operating life requirement is 10 hours which is made up of duty cycles each consisting of 20 minutes of operation followed by a soak time of not less than 5 minutes. Wear rates of seal components experienced during test are as follows:

P5N Carbon Face Piece	0.00005 in./hr
Chrome Plate on Rotating Ring	0.000025 in./hr

These wear rates were determined from three tests with a total run time of approximately 30 hours. The rates indicate that the seal will easily surpass the required life requirement.

TABLE 1—HELIUM SEAL CHARACTERISTICS

1.	Seal Operating Speed, RPM	11,000
2.	Surface Speed, ft/min.	2300
3.	P-V Factor, PSI Ft/Min.	23,600
4.	Operating Medium	Helium Gas
5.	Pressure, PSID	50-80
6.	Temperature, °F	−297
7.	Seal Deflection (installed), Inch	.040-.050
8.	Axial Load, Lbs.	2.5
9.	Face Area, in²: Sealing Land	0.19
	Bearing Land	0.14
10.	Hydraulic Overbalance, %	55
11.	Face Pressure (Total), PSI	10
12.	Friction Torque (Maximum), oz in.	10
13.	Friction HP	0.10
14.	Face Flatness, Helium Light Bands	1-2
15.	Run-Out (Rotating Face), T.I.R., inch	0.0005

DEVELOPMENT TESTS

Numerous tests were performed to develop a seal combination that would meet the required life, leakage, and torque requirements. The tests were conducted on several seal design variations and on various combinations of seal face and mating ring materials. Variations in seal face unit loading were accomplished by varying the bellows spring load, seal face width, and hydraulic overbalance. For test purposes, the seals were installed in the LO_2 pump previously discussed.

The tests were performed with the unit submerged in liquid oxygen and with the motor cavity inerted with helium gas at pressures from 5 to 130 psig. The tests consisted of repeated operating cycles of 20 minutes duration. After each operating cycle, the electrical power to the unit was shut off and the unit was allowed to soak for a minimum of five minutes before restart. Static seal leakage was measured before and after every run, and dynamic leakage during each run.

SEAL CONFIGURATIONS TESTED

Two basically different types of bellows seals were tested with the design variations shown in Fig. 4 and Table 2. Initial tests were performed with a seal having an integral carbon face press fitted in an end plate welded to the seal bellows. Later tests were done with a seal having a separate unattached floating carbon nose piece statically sealed to the bellows end plate by a lapped fit as previously described.

A bellows seal with an integral carbon face and no vibration damper was tested first. Excessive leakage, carbon wear and chipping of the carbon face at the O.D. and premature bellows failure were experienced with this seal. After removal from the pump, the seal was subjected to vibration tests at ambient temperature and found to have a broad natural resonant frequency range, which included the unit operating speed. An attempt was made to shift this resonant frequency band by changing the number of bellows convolutions to 7 and also to 11 from the original 9 convolutions. These changes did not

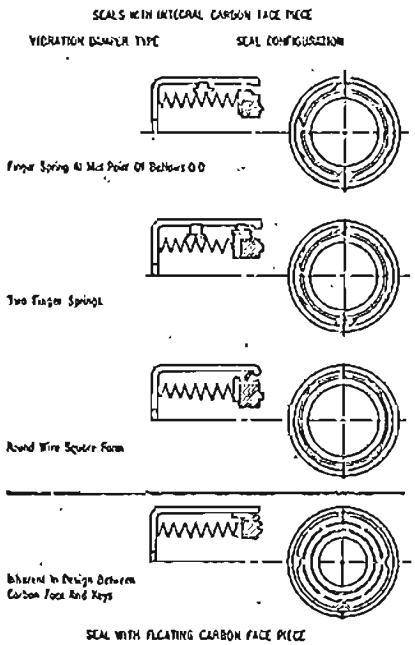

Fig. 4—Seal Configurations Tested.

prove effective, so a vibration damper spring was added to the seal.

The vibration damper consisted of a flat steel spring encircling the bellows O.D. approximately at the mid-point of its axial length. The spring applied a distributed force acting radially inward at the bellows O.D. This was a finger type spring with the fingers extending outward and reacting against the I.D. of the seal case.

A vibration test of this seal at ambient temperature indicated that this spring was not very effective in damping out vibration. Close visual examination of the seal revealed that there was very little physical interaction between the spring and the bellows. This was confirmed by the presence of very little hysteresis in the load versus deflection calibration of this seal performed at room temperature.

The seal design was then further modified to include an additional spring acting around the O.D. of the seal nose retainer plate which is welded to the bellows. Vibration tests of this seal indicated no natural resonance in the operating speed range. However, operational tests of the seal within the unit still resulted in excessive leakage and chipping of the carbon nose at the face O.D. A load versus deflection calibration of this seal exhibited a very wide hysteresis. This may have caused hanging up of the carbon nose relative to the mating ring with the consequent poor performance.

A round wire damper spring of approximately square configuration was installed in the seal acting between the seal nose retainer O.D. and the seal case I.D. This spring proved effective in damping the seal when it was subjected to a vibration test at room temperature. An operational test of the seal within the unit showed the leakage to be within acceptable limits. But addition of the round

Table 2—SEAL VIBRATION AND LEAKAGE CHARACTERISTICS

Vibration Damper	Resonant Frequency of Bellows Seal (Room Temperature)	Remarks
Seal with Integral Carbon Face Piece		
None	173 to 190 Hz	Excessive leakage, premature bellows failure, resonant frequency range includes operating speed of 183 cps.
Finger Spring at Mid-Point of Bellows O.D.	120 to 205 Hz	Excessive Leakage, Insufficient Damping
Finger Spring at Bellows Mid-Point & at Carbon Face O.D.	None between 20–800 Hz	Excessive Leakage, High Hysteresis Calibration Curve
Round Wire at Carbon Face O.D.	560 Hz	Low Leakage and Adequate Damping
Seal with Floating Carbon Face Piece		
Inherent in Design	153 to 205 Hz with no torsional load.	
	None between 20–800 Hz with a torsional load of 8 oz. in. applied to the carbon.	Low Leakage and Adequate Damping. Friction damping arises between the slots at the carbon face O.D. and the keys at the seal case I.D.

wire damper spring increased the seal spring rate and made installation within the pump fairly critical. For a seal load of 2.5 lbs. the seal had to be installed with an initial deflection of 0.010 to 0.012 inch. Further improvement was, therefore, considered desirable.

A basically different type of bellows seal was tested next. This seal employed a separate unattached carbon face piece sealed statically to the bellows end plate by a lapped fit. No added vibration damping devices were required with this seal. Leakage and wear were repeatedly within acceptable limits. Because of a lower seal spring rate, installed seal deflection is approximately 0.040 to 0.050 inch for a seal load of 2.5 lbs. making installation non-critical. This seal is presently being used in production liquid oxygen pumps. At approximately 50 psig helium pressure static leakage of this seal is from 3 to 20 SCIM and dynamic leakage is about 2 SCIM.

SEAL FACE PRESSURE

One of the more important seal parameters is the face pressure. Statically, it is due to the bellows spring load and hydraulic unbalance. During seal operation, hydrodynamic loads and thermal distortions also affect the face pressure.

The spring load must be adequate to enable the seal face to follow, and to maintain contact with, the seal rotating ring with its inherent out-of-squareness. When a seal is to be operated at a single pressure only, the spring load alone could be used, with a hydraulically balanced seal, to achieve acceptable seal performance.

In a hydraulically balanced seal the hydraulic forces tending to load and unload the seal face are equal and the face pressure is due to the bellows spring load only. Under this condition, assuming a triangular hydraulic face pressure distribution, 50 percent of the seal face area is outside and 50 percent is inside the bellows mean effective diameter. Such a seal is said to have an overbalance of 50 percent. The mean effective or equivalent piston diameter is approximately equal to the average geometric diameter of the bellows. In a seal with a 70 percent overbalance, 70 percent of the seal face area is outside of the mean effective diameter. In this seal, the total face pressure consists of the pressure due to the spring load and 70 percent minus 50 percent or 20 percent of the seal operating pressure. Theoretically, hydraulic seal overbalance should not be necessary, but practically it compensates for seal face mechanical and thermal distortions and manufacturing imperfections in face flatness. When the seal must operate over a range of pressures, it must be hydraulically overbalanced sufficiently to keep the leakage within acceptable limits at the highest pressure.

During the development tests, the seal bellows spring loads were varied from approximately seven to two pounds, seal hydraulic overbalance from 70 to 46 per cent, and seal face areas from 0.10 to 0.39 square inches. This resulted in seal face pressures from 40 to 10 psi.

The higher values of seal spring load and overbalance produced higher face pressures. The higher face pressures resulted in low initial leakage, but presented considerable wear and friction torque problems, and eventually high leakage due to seal face scoring. At the other seal face load extreme, while wear and friction torque were low, very little sealing was achieved. At overbalances of 50 per cent or less, seal leakage was very erratic. Best over-all results were obtained with a seal spring load of 2.5 lbs., an overbalance of 55 per cent, and a resultant seal face pressure of 10 psi. Seal leakage, friction torque and face wear were within acceptable limits.

TABLE 3—PROPERTIES OF SEAL MATERIALS

Material	Thermal Conductivity BTU—IN / Hr F° — FT²	Thermal Expansion In./In./F° @ 70°F	Hardness Knoop
Chrome	460	3.4×10^{-6}	1000
304	108	9.6	261 (*Rc* 22)
440C	168	5.6	251 (*Rc* 20)
Tungsten Carbide (K801) (Nickel Binder)	714	2.53	2200 to 2400
BECU (Berryleo 25)	650–800	9.4	400 (Rc 37)
P5N (Pure Carbon Co.)	60	2.7	Scleroscope 90
G39 (U. S. Graphite Co.)	3–90	1.5	Scleroscope 100
P2003 (Pure Carbon Co.)	220 approx.	2.5	Scleroscope 80
Silicon Carbide	720	1.88	2480

TABLE 4—SUMMARY OF SEAL TEST RESULTS

	Seal Face Material	Rotating Ring Material	Damper Spring Type	Spring Load Lbs.	% Over Balance	Face Area Inch²	Total Face Press. psi	Run Time Hr	Run Time Min	Wear Rate: Seal Face In./Hr	Wear Rate: Rotating Ring In./Hr	Seal Operating Press. psig	Leakage Std In.³/Min. Static	Leakage Std In.³/Min. Dynamic	REMARKS
Integral Carbon Face Type Seal															
1	G39 Carbon	Chrome on 304	None	3.2	70	0.18	20	4	44	0.004	(1)	5			Heavy transfer film, high torque, wear and leakage
2		Chrome on 440C	None	1.8	70	0.18	18	1	01	(2)	(1)	40 5	150 90	4200 1300	High leakage, seal lift off
3		P2003 Carbon	Round Wire	2.6	70	0.18	41	3	40	0.00015	(2)	130	35	22	High wear
4		P5N Carbon	Round Wire	1.3	55	0.18	17	2	27	0.0008	0.010	130	36	30	Very high rotating ring wear, early seal failure
5	P2003 Carbon	Chrome on 440C	Round Wire	2.6	70	0.18	41	4	00	(2)	(2)	130	225	130	Light wear, erratic seal, high leakage
6		P2003 Carbon	Round Wire	3.1	70	0.18	44	4	30	(1)	(1)	130	54	36	P2003 carbon is hygroscopic and not suitable for cryogenic use.
7		P5N Carbon	Round Wire	1.8	46	0.10	13	8	05	(2)	(2)	130	320	220	High leakage
8	P5N Carbon	P5N Carbon	Round Wire	2.0	55	0.13	22	1	32	0.0043	0.012	130	87	44	Very high wear and high torque
9		Chrome on 440C	Round Wire	1.7	50	0.11	16	15	47	0.00006	(1)	130	81	30	Damper spring malfunctioned Good wear and leakage
10		Aluminum Oxide LA-2 [3]	Round Wire	3.2	50	0.18	18	2	16	0.0014	(2)	5 130	3 35	15 33	Selective wear caused conical projections and high wear and high leakage
11		Chromium Carbide LC-IC [3]	Round Wire	4.9	50	0.18	28	1	55	0.0011	0.000008	5 130	590 28	170 100	Selective wear caused conical projections and high wear and high leakage
12		Silicon Carbide	2 Flat Springs	3.7	55	0.16	31	2	50	0.0005	(2)	130 95	270 200	260 60	Very high wear, erratic and high leakage (rough surfaces)
13	Tungsten Diselenide	Chrome on 440C	Round Wire	3.6	50	0.18	21	16	40	0.0003	0.000003	130	330	300	Low mechanical strength, dimensional instability, high leakage and wear
14	Silver Teflon	Chrome on 440C	Round Wire	4.6	55	0.15	37	5	00	0.0004	0.000004	130	1000	300	Soft material, low mechanical strength, very high leakage
Floating Carbon Face Type Seal															
1	MY10K	Chrome on 440C	Inherent In Design	6.9	70	0.35	30	0	20	(1)	(1)	50	40	20	High seal torque, restart impossible
2				2.7	70	0.38	17	1	50	0.00042	0.000048	50	20	3	High torque at operating pressure Unit started at lower pressure
3				2.0	68	0.39	14	4	41	0.00032	0.000021	50	81	10	High torque and high leakage
4	P5N Carbon	Chrome on BECU	Inherent In Design	2.5	55	0.33	10	6	00	0.00045	0.000008	54	40	2	Leakage was not significantly better than present design
5		Tungsten Carbide		2.5	55	0.33	10	5	00	0.00002	(2)	54	20	2	Leakage was 5 scim for first four hours, then increased to 20 scim
6		Chrome on 440C		2.5	55	0.33	10	5	00	0.00005	0.000025	54	20	2	Low wear, low and repeatable leakage

(1) No Data (2) No Measurable Wear (3) Flame Plated

MATERIALS TESTED

To minimize seal distortion and consequent leakage, it is desirable to use materials having, as nearly as possible, similar expansion characteristics and maximum heat conductivity.

Listed in Table 3 are thermal expansion, conductivity, and hardness for several seal materials. Other properties such as film laying characteristics, friction and wearing qualities, must be determined by actual test.

Various combinations of carbon face and mating ring materials were tested as summarized in Table 4. Initial tests were performed using a seal with an integral carbon nose with a G39 carbon face material operating versus several different mating ring materials. Carbon film transfer onto the mating ring was heavy and leakage, wear, and torque were generally high and not acceptable.

A P2003 graphite material with a chemical salt impregnation operating versus several mating ring materials generally resulted in high leakage. Also, it was discovered in the course of the program that this material was hydroscopic and, therefore, not suitable for cryogenic use. Moisture attracted to it resulted in freezing between the seals and mating rings within the pump.

Next, a P5N carbon graphite face material with a chemical salt impregnation was operated against a P5N mating ring. This resulted in high friction, torque, wear and leakage, and confirmed similar results obtained with other carbon versus carbon combinations in this pump unit. The P5N nose piece was also tested versus flame plated mating rings of aluminum oxide and chromium carbide. Carbon nose wear and seal leakage were high. This appeared to be due to selective wear of the flame plated materials resulting in sharp conical surface projections that abraided the carbon nose material. The P5N was also operated against a silicon carbide mating ring. Wear and leakage were high and sealing was erratic.

In an attempt to reduce seal friction, two non-carbon seal face materials were tested. One was a high temperature material consisting largely of tungsten diselenide solid lubricant and the other was silver-teflon composition. Both resulted in high wear and leakage. The tungsten diselenide material had the undesirable property of becoming soft, dimensionally unstable and wearing excessively after being exposed to liquid oxygen. The silver-teflon material, due to the fibrous nature of the embedded teflon particles, was difficult to polish to the high degree of surface finish necessary for good sealing. An attempt to run-in the material did not improve its sealing characteristics.

An MY10K carbon face piece with an antimony additive was tested versus hardened (*Rc* 55) 440*c* stainless steel. This material produced a very heavy transfer film on the mating ring. The carbon face was polished. Resultant leakage was low but seal friction torque was excessively high.

Tests were also conducted with a P5N carbon seal face operating versus mating rings of hard chrome plate on beryllium copper and also versus tungsten carbide with a nickel binder. In both cases the mating rings and carbon faces were very lightly scored across the areas of contact. A very light carbon transfer film was present on the mating rings. Leakage and wear results were approximately the same as obtained with P5N versus hard chrome on 440*c* stainless steel.

The best and most consistent results were obtained with a P5N carbon face operating versus a mating ring of hard chrome plate on annealed 440*c* stainless steel. Wear, leakage, and friction torque were within acceptable limits and were repeatable. This material combination has been qualified and is presently being used in a liquid oxygen pump on a space vehicle.

CONCLUSIONS

From the work described herein the following conclusions were reached:

1. The best seal combination consisted of a P5N carbon face operating versus a rotating ring of hard chrome plate on annealed 440*c* stainless steel with a hydraulic overbalance of 55%, a face pressure of 10 psi and a spring load of 2.5 lbs.
2. The over-all performance of the floating carbon face type seal was superior to the integral face type seal.
3. The floating carbon face seal was found to have the following advantages:
   a. Seal face distortion due to the differential contraction between the carbon face and the stainless steel bellows end plate was eliminated.
   b. Adequate axial and torsional vibration damping was achieved by friction between the carbon face and the seal stationary keys with no consequent increase in bellows spring rate.
   c. Refinishing and replacement of the seal carbon face could be done without removal of the seal from the pump housing.
   d. The lapped bellows end plate did not require refinishing during the life of the unit.