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SUMMARY OF SHAFT SEAL TESTS
FOR
HIGH-TEMPERATURE, HIGH-PRESSURE WATER APPLICATION

by

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SUMMARY OF SHAFT SEAL TESTS FOR HIGH-TEMPERATURE, HIGH-PRESSURE WATER APPLICATION

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A. Smaardyk

ABSTRACT

This report summarizes the program directed toward the development of an all-metallic seal assembly for use in connection with the design of external control rod drive actuating mechanisms for pressurized, water-cooled reactor systems.

Various material combinations are evaluated based upon performance in seal component assembly tests conducted in dynamic water systems at temperatures up to 450F and pressures up to 2000 psig.

Specifications and drawings of seals are included; test data are presented in tabular and graphical form.

I. INTRODUCTION

In the design of pressurized, water-cooled reactors, the control rods within the reactor vessel may be actuated by external drive mechanisms through individual control rod extension shafts. Each extension shaft then requires a means of sealing at the point where it passes through the pressure vessel.

The purpose of this report is to present the evaluation of various material combinations directed toward the development of a seal assembly for operation at pressures up to 2000 psig and water temperatures up to 450F.

II. DESCRIPTION OF TEST SEALS

Four types of seals were considered for test purposes:

- (1) Packing Seal
- (2) Face-type Seal
- (3) Breakdown Bushing Seal
- (4) Ring Type (labyrinth) Breakdown Seal

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The labyrinth breakdown seal was chosen because it offered the following advantages:

- (1) Low coefficient of friction
- (2) All-metallic seal
- (3) High-temperature resistance
- (4) Minimum end thrust

A. Rotating Seal

The basic seal, illustrated in Fig. 1, consists of a series of rings clamped in a retaining tube or seal sleeve. The seal rings float in annular grooves formed by the spacer rings and diaphragms. The spacer rings are about 3 mils thicker than the seal rings so that the floating rings are not tightly clamped; this provides for radial displacement of the control rod shaft extension such as might occur during assembly or in actual operation.

Figure 2 is an enlarged view of the floating ring assemblies. In operation, the floating rings are held against the diaphragms by the upward pressure forces acting upon them. To allow the seal rings to move radially, it is desirable to minimize the contact pressure at this point. This is accomplished by machining grooves and spot facing on the diaphragm which equalizes the pressure up to the groove. The land (A) formed between the groove and the bore of the diaphragm acts as a face-type seal preventing flow around the floating rings. The upper and lower surfaces of all rings are lapped to assure good fit and close tolerances.

The first test seal assembly had 7 seal rings; therefore, the theoretical pressure drop was approximately 290 psi/ring for a system pressure of 2000 psig. The two rings located above the lantern ring (Fig. 1) served as an atmospheric seal to insure that all leakage was drawn off at the seal drain for recirculation into the system.

Figure 3 is a disassembly of a representative seal. The seal sleeve, containing the seal ring elements, is installed in the seal housing (Fig. 4) which, in turn, is gasketed to the pressure vessel. The seal shaft is located by a radial bearing at the bottom, and the thrust is taken by a set of duplex-mounted, angular contact ball bearings. The shaft is driven by means of a V-belt and a drive pulley located above the thrust bearings.

During early test work, it was found that leakage of steam and water at 450F from the seal was difficult to handle. In addition, corrosion in water at elevated temperatures posed a major problem in the selection of seal component materials.

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FIG. 1
HIGH-PRESSURE BREAKDOWN SEAL

RE-6-11598-A
A. SMAARDYK:L.D., 4-12-54

ρ = DENSITY (lb/cu ft)

$$F = (\text{AREA } A) \times \frac{P_1 - P_2}{2}$$

r = RADIUS SHAFT (in.)

c = RADIAL CLEARANCE (in.)

μ = VISCOSITY (lb-sec/sq in)

L = TOTAL LENGTH OF SEAL RINGS (in.)

ΔP = PRESSURE DROP (psi)

Q = LEAKAGE

$$Q = \frac{\pi r c^3}{6 \mu L} \Delta P \text{ (in}^3\text{/sec)}$$

$$= \left(\frac{\pi r c^3}{6 \mu L} \right) \Delta P \left(\frac{3600}{1728} \right) \rho \text{ (lb/hr)}$$

FLOATING RING

SPACER RING

FIG. 2
FLOATING RING ASSEMBLY

NOTE: NUMBERS INDICATE SEQUENCE OF SEAL RINGS FROM TOP DOWN

FIG. 3
DISASSEMBLED SEAL COMPONENTS

Figure 4
SHAFT SEAL ASSEMBLY

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To reduce the seal and water temperature, thereby reducing the corrosion problem, a seal assembly with a water-cooled jacket was constructed as shown in Fig. 5. In addition, the following modifications were incorporated:

1. The seal discharge, after passing through a heat exchanger for cooling, is reintroduced directly at the bottom of the seal at a lower temperature (200F) than the pressure vessel (450F).
2. The seal sleeve was eliminated to reduce the housing diameter and corresponding pressure vessel opening.
3. The effective shaft sealing length was increased by combining the functions of the spacer and diaphragm rings into one ring, without increasing the over-all height of the seal.
4. The number of seal rings was increased from 7 to 12 rings, resulting in less pressure drop per ring and consequent lower theoretical bearing force per ring.
5. A valve seat was installed below the bottom seal ring and an engaging seat machined on the seal shaft. Through application of pressure from a hand wheel at the upper end, the valve seat can be engaged to close off the pressure vessel completely. The downward movement of the shaft, through the ball thrust bearings, was made possible by grinding suitable relief on the thrust bearing lands.
6. A carbon face-type atmospheric seal was substituted for the two upper atmospheric seal rings.

B. Reciprocating Seal

In an effort to determine the feasibility of the labyrinth ring-type seal for reciprocating motion, a test was conducted in which the reciprocating motion was imparted to the seal shaft by a ball bearing lead screw. The mechanical arrangement of this reciprocating seal is shown in Fig. 6. The seal shaft of $1\frac{1}{4}$ -in. diameter was successfully operated over a 28-in. stroke for 72,000 strokes. The test setup was then used to observe the static leakage over relatively long periods of time.

Another reciprocating seal test was conducted on a $\frac{1}{2}$ -in. diameter shaft operating over a 6-in. stroke. The shaft was actuated by an eccentric crank operating the horizontally sliding shaft. The seal was tested successfully for over 10,000 hr during which time it completed 18 million strokes. The seal is shown in Fig. 7 and the test apparatus in Fig. 8.

Figure 6
SEAL FOR 1-1/4 IN. DIAMETER RECIPROCATING SHAFT

Figure 7
WATER-COOLED 1/2 IN. DIAMETER RECIPROCATING
DRIVE SHAFT SEAL

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Figure 8
WATER-COOLED RECIPROCATING SEAL TEST APPARATUS

III. TEST APPARATUS AND RESULTS

The test apparatus used for oscillating and rotating seals is shown schematically in Fig. 9 and pictorially in Fig. 10. Simulated reactor vessel pressure was maintained by a three-plunger, high-pressure, reciprocating injection pump, feeding distilled water to a nitrogen-pressurized surge tank. All seals were tested at pressures usually ranging between 1700 and 2000 psi. Liquid level in the surge tank was controlled by an adjustable relief valve in a bypass line of the feed pump. Water leakage from the seal discharge was collected in a sump tank, filtered, and reintroduced into the system by the high pressure reciprocating pump. Typical leakage variations during test are plotted for rotating seals in Fig. 11 and for a reciprocating seal in Fig. 12.

Temperatures up to 450F were maintained by four 3-kw immersion heaters in the seal tank and additional external heaters clamped on the outside wall of the seal tank.

Twenty-nine seal combinations were evaluated in 30 tests using various shaft and ring materials. Various drive mechanisms were used to simulate control rod drive notions of interest.

The original design called for a seal operating at a moderate speed of 150 cpm and oscillating through an arc of 150°, therefore, the greater part of the tests were carried out at moderate speeds oscillating up to 300 cpm and rotating up to 500 rpm. Tests F-27 and F-28 were conducted at 5200 rpm and Test F-29 at 3500 rpm in an effort to determine whether this type of seal could be operated at a higher speed. These tests showed that higher speeds are feasible when compatible materials are used, such as in Test F-29. It is felt that larger clearances should be allowed for higher speed operation.

The test data are summarized in Appendix B. The measurements shown are those obtained by averaging inspection results before and after test. A typical inspection report sheet is shown in Fig. 13.

The actual leakage measurements, taken at frequent intervals during the test, were not constant, probably due to the random orientation of the rings on the shaft. This corroborates the theoretical deviation due to eccentrically located rings as discussed in Appendix A.

Generally, the theoretical leakage obtained through the Hagen-Poiseuille formula agreed with the measured leakage during the early stages of the test. As the rings wore slightly and the shaft grooved somewhat, accurate average measurements were difficult to obtain and theoretical approximations were more difficult to establish. Measured and calculated leakage rates for selected seals are compared in Table I.

Measurements of starting torque were made to ascertain the amount of stick friction in the rotating-type seal. A torque arm was fastened on the seal shaft drive pulley and, by means of a spring scale, the breakaway torque was measured. The results are given in Table II.

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FIG. 9
SCHEMATIC DIAGRAM OF
LABYRINTH-TYPE SEAL
TEST EQUIPMENT

RE-8-12635-A

FIG. 10
SEAL TEST APPARATUS

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RE-7-12603-A

FIG. 11
LEAKAGE VS OPERATING TIME,
 $1\frac{1}{4}$ IN. DIA. ROTATING SHAFT SEALS

RE-7-12603-A

A. SMAARDYK:L.R.D.,3-10-54

RE-7-12604-A

FIG. 12
LEAKAGE VS OPERATING TIME.
1/2 IN. DIA. RECIPROCATING SHAFT SEAL

RE-7-12604-A
A. SMAARDYK; L.R.D., 3-10-54

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SEAL TEST F-23

SHAFT: ARMCO 17-4 PH, CHROME PLATED
 SEAL RINGS: STELLITE 3
 DIAPHRAGMS: K-MONEL
 LANTERN: K-MONEL

TEMPERATURE: 200 F
 PRESSURE: 2000 psi
 HOURS OF OPERATION: 13,878
 ROTARY
 CONTINUOUS
 SPEED: 385 rpm

EFFECTIVE SEAL LENGTH: 2.2815 in.

LEAKAGE, lb/hr
 INITIAL: 150
 FINAL: 200

RE-7-12789-A

INITIAL INSPECTION
 9-12-51

FINAL INSPECTION
 10-16-53

ALL MEASUREMENTS ARE IN INCHES

<u>SEAL RINGS</u>		
	<u>INNER DIA.</u>	<u>WIDTH</u>
TOP 1.	1.2503	0.3267
2.	1.2504	0.3263
3.	1.2503	0.3259
4.	1.2503	0.3265
5.	1.2503	0.3264
6.	1.2503	0.3262
7.	1.2502	0.3245
8.	1.2502	0.3255
9.	1.2503	0.3265
AVERAGE = 1.2503

<u>SEAL RINGS</u>		
	<u>INNER DIA.</u>	<u>WIDTH</u>
1.	1.2505	0.3266
2.	1.2505	0.3260
3.	1.2505	0.3260
4.	1.2504	0.3265
5.	1.2504	0.3263
6.	1.2504	0.3261
7.	1.2505	0.3247
8.	1.2505	0.3258
9.	1.2506	0.3262
AVERAGE = 1.2505		

AVERAGE RING WEAR = 0.0002

<u>SHAFT</u>
1.2492
1.2492
1.2492
1.2492
AVERAGE = 1.2492

<u>SHAFT</u>
1.2490
1.2490
1.2490
1.2490
AVERAGE = 1.2490

AVERAGE SHAFT WEAR = 0.0002

REMARKS:

SHAFT WAS IN EXCELLENT CONDITION. RINGS AND DIAPHRAGMS WERE COVERED WITH A LIGHT BROWN OXIDE.

FIG. 13
 TYPICAL INSPECTION REPORT

RE-7-12789-A
 A. SMAARDYK: B.M., 4-14-54

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Table I

COMPARISON BETWEEN MEASURED AND
CALCULATED SEAL LEAKAGE

Test No.	Leakage, lb/hr			
	Initial		Final	
	Measured	Calculated*	Measured	Calculated*
F-8	50	60	300	1000+
F-10	90	100	175	600
F-12	30	75	220	375
F-14	100	130	250	500
F-16	70	35	100	60
F-18	15	8	4	8
F-21	90	65	275	210
F-23	150	130	200	250
F-29	350	15	350	300
F-30	25	17	50	not measured

*Calculated leakages obtained from Fig. 14.

Table II

STICK FRICTION OF ROTATING SEALS

Seal Test No.	Torque Required, ft-lb
F-22	1.9
F-15	0.85
F-14	0.18
F-17	1.3
F-21	0.8

The seal materials were evaluated on the basis of increase in leakage, time of operation, wear resistance and corrosion resistance. In Table III are listed the composition and mechanical properties of selected materials.

Combinations of the selected materials also exhibited good compatibility and high resistance to galling and wear in separate Falex Wear Tests.

Supplementary corrosion data, pertinent to the materials selected, are summarized in Table IV. The evaluations are based on design criteria for a specific application; therefore, in extrapolating the data for conditions other than set forth in the table, each engineer or designer has the opportunity to determine for himself a conclusion which suits his particular problem.

IV. CONCLUSIONS

In general, the tests demonstrated successfully the feasibility of sealing control rod extension shafts for reactor application.

The seal material combinations which exhibited the best performance are listed in the order of preference in Table V.

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Table III

COMPOSITION AND MECHANICAL PROPERTIES OF THE SUCCESSFUL SEAL MATERIALS

(Data obtained from manufacturers' publications)

Material	Nominal Composition, %															Coeff of Therm Exp, in./in./F $\times 10^{-6}$	Tensile Strength, psi $\times 10^{-3}$	Hardness	
	Cr	Ni	Co	C	W	Mn	Si	Cu	Fe	S	Al	Mo	P	B	Others			BHN	R
Armco 17-4 PH	16.50	4.25		0.05				4.0	Bal							6.1	135-165	372-426	
Colmonoy 6	13-20	65-75												2.75-4.75	Fe } 10 Si } max C }	8.7			C-56-62
Ti Carbide		23													Mo ₂ C5; TiC 72	3.25	140*		C-71
K-Monel		66		0.15		0.75	0.5	29.0	0.9	0.005	2.75					7.8	90-110	140-180	
Stellite 3	29-33	3 max	Bal	2.2-2.7	9.5-13.5	0.5 max	1.5 max		3 max					0.05-0.20		6.85	85		C-60
Stellite 6	26-31	3 max	Bal	0.9-1.4	3.5-5.5	1 max	1.5 max		3 max							9.4	105		C-55
Type 304	18-20	8-10		0.08 max		2 max	0.75 max									9.7	90	150	
Type 440C	16-18			0.95-1.20		0.50	0.50			0.03 max		0.50	0.03 max			6.0	110	220	
Vascaloy Ramet 166	27-32		45-50	2-4	14-19	1-3									Nb 2.5-3.5	7.5	65		C-53-59
Vascaloy Ramet 171	20-22		60-65	0.3 max	13-17		0.2-0.5									7.5	300-350		C-28-30

* Transverse Rupture Strength

Table IV

CORROSION OF SEAL MATERIALS IN WATER

Medium: High-Purity Distilled Water at Temperatures Indicated

Material	Evaluation*			
	200F	300F	400F	500F
Armco 17-4 PH	Very Good	Very Good	Very Good	Very Good
Type 304	Very Good	Very Good	Very Good	Very Good
Type 440C	Very Good	Very Good	Very Good	Good
Stellite 3	Very Good	Very Good	Very Good	Very Good
Stellite 6	Very Good	Very Good	Very Good	Very Good
Vascoloy Ramet 166	Very Good	N.A.	N.A.	Fair
Vascoloy Ramet 171	Very Good	N.A.	N.A.	Good
Colmonoy 6	N.A.	N.A.	N.A.	Fair
Inconel	Very Good	Very Good	Very Good	Very Good
Firthite H.T. 77 (Titanium Carbide)	Very Good	N.A.	N.A.	N.A.
K-Monel	Very Good	Very Good	Very Good	Very Good
Chrome Plate	Very Good	Very Good	Very Good	Fair

*Very Good:

Appearance: No apparent corrosion products; clean tarnish film.
 Weight Change: $<0.3 \text{ mg}/(\text{cm}^2)(\text{mo})$

Good:

Appearance: No apparent corrosion products; clean tarnish film.
 Weight Change: $>0.3 \text{ mg}/(\text{cm}^2)(\text{mo})$

or

Appearance: Some apparent corrosion products on surface.
 Weight Change: $<0.3 \text{ mg}/(\text{cm}^2)(\text{mo})$

Fair:

Appearance: Corrosion products on surface.
 Weight Change: $>0.3 \text{ mg}/(\text{cm}^2)(\text{mo})$

N.A.: Data not available

Table V

PREFERENCE OF SEAL MATERIAL COMBINATIONS

Shaft	Seal Rings	Diaphragms
Armco 17-4 PH, Chrome Plated	Stellite 3	K-Monel
Type 440C, Chrome Plated	Stellite 3	K-Monel
Colmonoy 6 on Inconel	Titanium Carbide	Titanium Carbide
Colmonoy 6 on Inconel	Cast Colmonoy 6	Cast Colmonoy 6
Vascoloy Ramet 171	Vascoloy Ramet 166	Vascoloy Ramet 166

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APPENDIX A

SEAL THEORY

Leakage

The method¹ for calculating leakage flow through an annulus of length $L = c$ and of width equal to $2\pi r$ is given by:

$$Q = \frac{2\pi r c^3}{12\mu L} \Delta P$$

where

Q = leakage flow, cu in./sec

r = radius of shaft, in.

c = radial clearance between ring and shaft, in.

μ = viscosity, lb-sec/sq in.

L = total length of seal rings, in.

ΔP = pressure drop through seal rings, psi

The flow in pounds per hour for a fluid with density ρ is given by:

$$Q = \left(\frac{\pi r c^3}{6\mu L} \right) (\Delta P) \left(\frac{3600}{1728} \right) \rho$$

Assuming a shaft diameter of 1.25 in., a viscosity (μ) of 0.2×10^{-3} lb/ft-sec for water at 200F or $0.2 \times 10^{-3} \times 2.1585 \times 10^{-4}$ lb-sec/sq in., a seal length of 2.3 in., water density of 60 lb/cu ft, and a pressure drop $\Delta P = 2,000$ psi, the curve in Figure 14 can be obtained. The curve shows that leakage of 100 lb/hr is attainable with seal rings having a radial clearance in the order of 0.0005 in.

Type of Flow

With a leakage of 100 lb/hr, one can determine whether the flow is laminar, such as was assumed for the shaft size used in Figure 14.

$$Re = \frac{VD\rho}{\mu}$$

where

Re = the Reynolds number

ρ = density, lb/cu ft

V = velocity, ft/sec

μ = viscosity, lb/ft-sec

$D = \frac{\text{cross-sectional area}}{\text{wetted perimeter}}, \text{ ft}$

¹H. F. P. Purday, "An Introduction to the Mechanics of Viscous Flow," (New York: Dover Publications, Inc., 1949).

FIG. 14
CALCULATED LEAKAGE VS RADIAL CLEARANCE
FOR A $1\frac{1}{4}$ IN. DIA. SHAFT SEAL

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RE-7-12363-A

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The radial clearance corresponding to 100 lb/hr is 0.0005 in. Assuming a water temperature at 200F:

$$V = \frac{\text{Volume flow}}{\text{Clearance area}}$$

$$= \frac{100}{(60)(3600) \left(\frac{(\pi)(1.25)(0.0005)}{144} \right)}$$

$$= 33.3 \text{ ft/sec}$$

$$D = \frac{4\pi dc}{2\pi d} = 2c = 0.85 \times 10^{-4} \text{ ft}$$

$$Re = \frac{(33.3)(0.85 \times 10^{-4})(60)}{0.0002} = 850$$

Leakage Deviation

Deviation of leakage flow through narrow rectangular channels caused by eccentricity of shaft in bushing is given by:

$$Q = \frac{\Delta P}{12\mu L} \int_0^W c^3 dy$$

Considering a shaft of diameter $2r$, working in a bushing with mean radial clearance c_0 and eccentricity ϵ , the maximum clearance is $c_0 + \epsilon$, and the clearance at any point on the circumference is

$$c = c_0 \left(1 + \epsilon \cos \frac{y}{r} \right)$$

The ratio of leakage flow under pressure of the eccentrically located shaft to leakage flow of the concentrically located shaft is given by:

$$\frac{\int_0^\pi (1 + \epsilon \cos \theta)^3 d\theta}{\int_0^\pi d\theta} = 1 + \frac{3}{2}\epsilon^2$$

For example, consider a mean radial clearance of 0.0005 in. and an eccentricity ϵc of 0.0002 in. then

$$\epsilon = \frac{0.0002}{0.0005} = 0.4$$

and the leakage ratio equals $1 + \frac{3}{2}\epsilon^2$ or 1.24.

Figure 15 shows the leakage flow ratio plotted vs. eccentricity ratio ϵ ; it may thus be seen that variations in flow due to eccentricity are to be expected.

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A. SMAARDYK:F.S., 3.17.54

FIG. 15
LEAKAGE VARIATION WITH RING ECCENTRICITY

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RECAPITULATION OF SHAFT SEALS TESTS

Medium: Distilled water at temperature and pressure noted

Legend

Type of Drive
 R = Rotary
 O = Oscillatory
 Re = Reciprocating

Units

rpm
 cpm
 rpm

Test No.	Components and Material			Test Conditions					Test Results					Remarks	
	Shaft	Seal Rings	Diaphragms	Type of Drive	Speed	Temp, F	Press, psi	Time, hr	Leakage, lb/hr		Average Radial Clearance, in.		Wear, in.		
									Initial	Final	Initial	Final	Shaft		Ring
F-1	Armco 17-4 PH, age hardened	Armco 17-4 PH, age hardened	Type 416	R	---	450	2000	---	---	---	0.0004	Seized	Galled	Galled	Immediate failure due to galling.
F-2	Armco 17-4 PH, age hardened	Armco 17-4 PH, age hardened	Type 416	R	---	450	2000	---	19.3 static	---	0.0005	Seized	Galled	Galled	Same components as test F-1. Shaft was polished, seal rings opened up 0.0002 in.; seal ring thickness decreased 0.0004-5 in. Immediate failure due to galling of rings and shaft.
F-3	Armco 17-4 PH, age hardened & chrome plated	Armco 17-4 PH, age hardened	Type 416	R	150	450	2000	1.7	10	46	0.00075	Seized	Galled	Galled	Shaft from above tests, chromed. New seal rings; diaphragms from above. Seized with scoring on shaft and rings.
F-4	Armco 17-4 PH, age hardened & chrome plated	Type 440C	Type 416	R O	400 150	450 350	2000 2000	432 493	70	160	0.0047	---	0.0003	---	Same shaft as above with new chrome. Type 416 steel corroded badly.
F-6	Type 304, Stellite 1 overlay	Stellite 3	Type 440C	O	300	290	2000	5.25	100	100	---	Seized	Galled	Galled	Failure due to galling of shaft and rings.
F-7	Type 440C	Type 440C	Type 440C	O	300	200	2000	2.5	40	40	---	Seized	Galled	Galled	Rings seized on shaft causing failure.
F-8	Armco 17-4 PH	Armco 17-4 PH with carbon inserts	Type 440C, re-worked from previous seal	R	590	450	2000	175	50	300	0.0003	0.0010	None	0.0015	Increased leakage due to excessive graphite wear. Slight corrosion in 440C.
F-9	Type 440C, re-worked from Test 7	Type 440C from Test 7	Type 304, chrome plated	O	300	70	2000	0.05	75	75	0.0004	0.0004	None	None	Chrome peeled from diaphragms, causing seizure.
F-10	Type 440C from Test 9	Type 440C from Test 9	Type 440C from Test 8	O	300	350	1900	6.1	90	175	0.0004	Seized	Galled	Galled	440C rings seized on shaft. Lapped surfaces galled.
F-11	Hastelloy C	Stellite 3	Type 304	O	300	100	500	0.008	---	---	0.00055	Seized	Galled	Galled	All rings seized on shaft. Diaphragm pressure faces galled. Material corroded badly. Test discontinued.
F-12	Ontario Tool Steel	Ontario Tool Steel	Ontario Tool Steel	O	300	350 450	2000 2000	Total 252	30	220	0.0003	0.0006	0.0005	None	Failure due to seizing of shaft. Pressure faces of rings are galled.
F-13	Hastelloy C from Test 11	Stellite 3 from Test 11	Type 440C	O	135	240	1900	0.25	50	102	0.00065	Seized	Galled	Galled	Corrosion present on Type 416. Discontinued to test other materials.
F-14	Type 440C, chrome plated from Test 10	Type 440C	Type 416	O R R	143 480 480	450 2000 300	2000 2000 2000	4417 183 841	100	250	0.0004	---	---	---	Shaft corroded at ring contact areas.
F-15	Hastelloy C, chrome plated from Test 13	Stellite 3, re-finished	Type 440C, re-finished	R R Static	190 480 480	200 2000 2000	2000 2000 2000	3874 193 2029	10	---	0.00045	---	0.0001	0.0007	

*Operated 1 min/week.

RECAPITULATION OF SHAFT SEAL TESTS (Contd.)

Test No.	Components and Material				Test Conditions						Test Results				Remarks
	Shaft	Seal Rings	Diaphragm	Type of Drive	Speed	Temp, F	Press, psi	Time, hr	Leakage, lb/hr		Average Radial Clearance, in.		Wear, in.		
									Initial	Final	Initial	Final	Shaft	Ring	
F-16	Vascoloy Ramet 171	Vascoloy Ramet 166	Vascoloy Ramet 166	O	175	130	2000	2094	70	100	0.00043	0.0005	0.00015	None	Test discontinued.
F-17	Type 440C, chrome plated	Type 440C	Type 440C	R	115	300	2000	3003	25		---		---	---	Test discontinued.
F-18	Colmonoy 6 on Inconel	Cast Colmonoy 6	Cast Colmonoy 6	O	175	88	2000	927	15	4	0.0003	0.0003	None	None	Transferred to reciprocating test F-20.
F-19	Type 440C	Type 440C	Type 440C	Re	20	125	2000	211	40		0.0003		0.0002	0.0019	1 2-inch reciprocating seal; 6 in. stroke. Very little shaft wear. Rings corroded and worn to severe eccentricity.
F-20	Colmonoy 6 on Inconel, from Test 18	Cast Colmonoy 6, from Test 18	Cast Colmonoy 6, from Test 18	Re	1.4	100	2000	641	21		0.0003		None	0.0003	1-1 1/4-inch reciprocating seal. External lead screw failed twice. Continued as a static test to observe leakage only.
F-21	Type 440C, chrome plated	Stellite 3	K-Monel	R	116-490	200-450	2000	11800	90	275	0.0004	0.0007	None	0.0006	Test discontinued.
F-22	Colmonoy 6 on Inconel	Cast Colmonoy 6	Type 440C, heat treated	O	175	110	2000	336	30		0.0003				Excellent, except for light corrosion on diaphragms.
F-23	Arco 17-4 PH, chrome plated, from Test 8	Stellite 3	K-Monel	R	480*	200	2000	3786		250	0.0005	0.0007	0.0001	0.0003	Shaft was clean but rings had brown oxide coating.
F-24	Type 440C, chrome plated	Stellite 3	Stellite 6	R	29	200	2000	10700	50	140	0.00045	0.0012	0.0001	0.0014	1/2-inch reciprocating seal. Shaft in excellent condition. Rings worn oval, but not corroded.
F-25	Vascoloy Ramet Alloy 171	Vascoloy Ramet Alloy 166	Vascoloy Ramet Alloy 166	R	458	200	2000	60	80	250	0.0004	0.0004	Galled	0.0002	Shaft galled, causing seizure.
F-26	Type 304 with Stellite 1 overlay	Stellite 3	Type 440C	R	355	96	2000	0.2	25	25	0.00037	Seized	Galled	Galled	Seal rings seized on shaft.
F-27	Type 302, chrome plated	Type 440C	Type 440C	R	5200	200	2000	153	350	280	0.0004	Seized	Galled	Galled	Seal failed when rings seized on shaft. Chrome plating on shaft not to specifications.
F-28	Vascoloy Ramet Alloy 171	Vascoloy Ramet Alloy 166	Vascoloy Ramet Alloy 166	R	5200	200	2000	217	150	150	0.0007	Seized	Galled	Galled	Galling on shaft and seal rings caused shutdown.
F-29	Colmonoy 6 on Inconel	Titanium Carbide	Titanium Carbide	R	3500	125	2000	2767	350	350	0.00035	0.0009	0.001	0.0001	Shaft worn 0.001 in. Rings in excellent condition.
F-30	Type 440C, chrome plated	Type 410	K-Monel	R	163	100	2000	3095	25	50	0.00037	Seized	Galled	Galled	Seal rings and diaphragms were frozen to shaft. Corrosion products present on all parts, probably originating from Type 410 seal rings.