AMF-388-TR-1

COMPONENT FEASIBILITY TEST REPORT
FOR THE AMF ON-POWER REFUELING
SYSTEM FOR THE DOUGLAS POINT PLANT

February 1962

AMF Atomics, Inc.
Greenwich, Connecticut

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COMPONENT FEASIBILITY TEST REPORT
FOR THE AMF ON-POWER REFUELING
SYSTEM FOR THE DOUGLAS POINT PLANT

to the
Savannah River Operations Office, AEC

February 1962

Work performed under Contract No. AT(38-1)-233
Phase I-Task II

AMF Atomics, Inc.
A Division of
American Machine & Foundry Company
Greenwich, Connecticut

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INTRODUCTION

This report describes work performed by AMF Atomics, a Division of the American Machine & Foundry Company, under contract AT(38-1)-233, dated 17 January 1961, with the USAEC, Savannah River Operations Office, during the task specified herein.

The objective of this activity is to engineer, design, fabricate and test a prototype, on-power, refueling machine suitable for operation with the Canadian Douglas Point Plant, CANDU I, heavy-water moderated, power reactor. On-power refueling will permit the loading and unloading of the natural-uranium fuel while the reactor is operating at full power.

This work is being performed under a Cooperative Program between the USAEC and Atomic Energy of Canada Ltd., which covers the development of heavy-water moderated, power reactors. It is administered by the Savannah River Operations Office of the USAEC.

Task II of the AMF On-Power Refueling System Work Program concerned itself with obtaining test data for various system subcomponent parts. The basis for selecting those units requiring test was established through engineering evaluation of existing available information. In this manner it was determined which components or subcomponents evidenced questionable feasibility and programs were initiated to conduct the required tests. Before testing could commence, applicable special or standard test equipment had to be designed or procured, a test facility established and a test stand fabricated and erected.

After firming the above requirements, component testing started. Sections I through VIII of this publication provide documentary evidence of these tests in the form of test reports.

A Table of Contents and a List of Illustrations is presented to provide a ready cross-reference to each test report and its associated graphic coverage.

The high-temperature water supply is the environmental water source for the various tests documented in this report.

City water is the source of supply for this system. It is conditioned as follows:

- a. The water first passes through a filter which removes particulate matter.
- b. The water then passes through a mixed-bed demineralizer which maintains the resistance below 1,000,000 ohms.

FIGURE 1. HIGH-TEMPERATURE WATER SUPPLY

- c. The water then passes through a mono-bed demineralizer for further purification.
- d. The water is finally passed through a polishing filter on the downstream side of the demineralizer which removes any resin carry over.

The water is directed to a storage tank where sufficient lithium hydroxide is added to the medium to maintain a pH of 10. This controlled water is pressure injected into the system to replenish losses incurred by testing or normal system leakage.

Oxygen levels of less than 0.5 PPM are maintained by venting non-condensable gases through system high points.

The test medium (water) is analyzed daily to maintain conditions compatible with the Douglas Point Plant.

The system water is heated in a vessel and circulated from the vessel to the supply manifold. The supply manifold is connected to the various test rigs. The discharged water is collected in a discharge manifold and cooled before being directed to positive-displacement pump. The water is then pumped to the heating vessel and the process repeated.

System pressure is achieved by several methods all contributing to pressure maintenance. This is accomplished through heating and nitrogen pressurization. Pressure losses due to testing and normal system leakage tend to imbalance the pressure stability. This is counteracted by final trim using nitrogen pressurization.

This system was capable of supplying test water at 1500 psig pressure and 585°F for the various tests at a rate of approximately 2 gallons per minute.

FIGURE 2. HIGH-TEMPERATURE WATER SUPPLY INCLUDING TEST STAND

SECTION I

WEATHERHEAD QUICK-DISCONNECT COUPLING TEST

1.0 PURPOSE:

1.1 The purpose of this report is to present and evaluate the results of a test program involving the thermal and mechanical cycling of a commercial hydraulic quick-disconnect coupling assembly.

2.0 SCOPE:

2.1 The scope of this test was to cycle the coupling assembly up to and including the manufacturer's rated parameters proving the feasibility of the disconnect concept, and component reliability.

2.2 The unit tested was a Weatherhead Quick-Disconnect Coupling Assembly. It consists of:

- a. Nipple-Half Assembly (Dwg. No. 405-00127). (See figure 1-1)
- b. Socket-Half Assembly (Dwg. No. 405-00128). (See figure 1-2)

3.0 SUMMARY:

3.1 The coupling halves were successfully tested through the initial test phase. This phase consisted of cycling the coupling from ambient to 1500 psi pressure at room temperature. The coupling failed during the second series of this test program. This series was intended to consist of thermally cycling the coupling from room temperature to 400°F. Failure occurred in the two coupling seal

FIGURE 1-1. NIPPLE-HALF ASSEMBLY

FIGURE 1-2. SOCKET-HALF ASSEMBLY

areas. The failure occurred at a temperature between 340°F and 370°F at 900 psig pressure which was lower than the manufacturer's rating of 400°F.

Inspection revealed a permanent set and loss of resilience in the Viton "O" ring (figure 1-3) causing the failure.

A second failure occurred adjacent to the ball at the KEL-F seal. This failure (figure 1-4) was caused by a metal chip embedded in the seal.

4.0 CONCLUSION:

- 4.1 Test results indicate that the commercial coupling is not adequate for intended service, at elevated temperatures, as a hose-line connection on both adapter and shut-off valves.

5.0 RECOMMENDATIONS:

- 5.1 It is recommended that the Weatherhead Quick-Disconnect Coupling Assembly not be used for the service intended. The coupling has two basic disadvantages. They are:
- a. Its inability to withstand high-temperature, refueling system requirements, and
  - b. Its use of elastomer seals which are subject to radiation deterioration and damage caused by system fluid contamination.

FIGURE 1-3. FAILED "O" RING SHOWING PERMANENT SET

FIGURE 1-4. KEL-F SEAL FAILURE

5.2 It is further recommended that testing be discontinued and another approach to solve the problem of connecting and disconnecting flexible hoses at elevated temperatures be investigated. If a coupling configuration is to be used in the cold portion of the refueling machine hydraulic system, it is recommended that another type be considered. The cost of the subject coupling is high compared to other low-temperature, commercially-available couplings.

6.0 TEST PROCEDURE:

6.1 The coupling was installed in the test stand fixture (figure 1-5). Each coupling half was assembled in the test stand and the joint sealed with metal "O" rings (figure 1-6). The schematic of the piping connecting the test rig to the high-temperature water supply.

6.2 Testing, as outlined in paragraph 2.0, proceeded as follows:

- a. The first series, consisting of pressurizing the coupling with environmental water in increments of 100 psig up to 1500 psig at room temperature, was completed successfully.
- b. The second, or high-temperature series of this program, was not successful. The coupling failed between 340°F and 370°F. Failure (a leak in the joint between the nipple-half and socket-half couplings) occurred as the coupling temperature was lowered. Close inspection of the nipple-half coupling revealed failure in

FIGURE 1-5. TEST STAND FIXTURE

FIGURE 1-6. CLOSE-UP OF COUPLING IN TEST STAND

the two seal areas. The socket-half coupling passed inspection. The cause of leakage was traced to the Viton "O" ring which had set and lost its resilience.

NOTE

This problem was discussed with the Viton representative and Weatherhead Company engineers. The Viton representative and the Weatherhead engineer suggested a post-cure process to improve material performance. Weatherhead offered to supply properly-treated "O" rings.

7.0 TEST DATA:

7.1 This portion of the report provides the data generated during the program. They appear as follows:

- a. Manufacturer's Shop Tests
- b. Delivery Inspection
- c. Pre-Test Inspection
- d. Pre-Test Cleaning
- e. Leakage Acceptance Tests
  - 1. Helium Leak Test
  - 2. Compressed Air Leak Test
- f. Hydrostatic Tests
  - 1. Ambient Temperature
  - 2. Rated Temperature
- g. Final Inspection

7.2 Manufacturer's Shop Tests

Prior to delivery, the coupling assembly was tested by the manufacturer using water, maintained at 60°F to 100°F, as a test medium. The assembly was tested as follows:

- a. Proof-Pressure Test. Water pressure at 4500 psig was applied to the coupled unit for one minute. No evidence of damage or external leakage was permitted.
- b. Leakage Test. Water pressure, in increments of zero to 10, 300, and 3000 psig, was applied to the coupled and uncoupled unit for one minute. No evidence of external leakage was permitted.
- c. Flow Area Inspection. The alignment of the flow balls in the coupled position was visually inspected. Flow passages having less than 90 percent flow area were cause for rejection.
- d. Drying procedure. The coupling assembly and detail parts were dried in a circulating air oven at 150°F.

7.3 Delivery Inspection

The coupling halves were individually wrapped in plastic bags and shipped in separate cardboard containers. Aluminum caps protected the threaded ends of the coupling halves. The couplings were visually inspected and found in satisfactory condition.

7.4 Pre-Test Inspection

Disassembly of the quick-disconnect coupling revealed the following physical defects:

a. Nipple-half assembly.

1. A 20-power lens revealed scratches, scores, and pits in the ball section.
2. The white "O" ring was split and the black "O" ring rough and pitted.
3. "O" ring particles adhering to surrounding metal.
4. Machining chatter marks beyond "O" ring.
5. Rough machine finish.

b. Socket-half assembly.

1. A high-spot on the ball. A mirror finish at the ball center.
2. Scratch marks on ball finish. (These scratches may have been caused by ball rotation).
3. Nicks and scratches on machined parts.
4. Chatter marks on machining.

NOTE

Chatter marks, nicks and scratches were on the outer surfaces of the coupling-half component parts. The balls were scratched, leading one to believe the manufacturer did not maintain a honed or polished sealing surface.

7.5 Pre-Test Cleaning

After test, each coupling half was immersed in demineralized water for 2 to 3 hours, and air dried with a warm air blower. Visual inspection revealed no evidence of oil chips or dirt.

7.6 Leakage Acceptance Tests

7.6.1 Helium Leak Test. The vacuum technique was used. The ball was lifted off the vacuum and helium bypassed the ball and seal. The bypass of helium was verified by the closure of the ball opening. No leakage reading was then obtained on the mass spectrometer.

7.6.2 Compressed Air Leak Test. Each coupling half was pressurized with compressed air at 80 psig. The outer surfaces were swabbed with soap. There was no evidence of leakage. The coupling was assembled and the same technique applied. There was again no evidence of leakage.

7.7 Hydrostatic Tests

7.7.1 Room Temperature. The coupling was pressurized from zero to 1500 psig in increments of 100 psig at room temperatures at intervals of 2 minutes. Results were as follows:

<u>Test Run No.</u>	<u>Initial Conditions</u>	<u>Final Conditions</u>	<u>Remarks</u>
1	Room temperature and pressure	Room temp. -1225 psig	No leakage from coupling
2	Room temperature and pressure	Room temp. -1175 psig	No leakage from coupling
3	Room temperature and pressure	Room temp. -1050 psig	No leakage from coupling
4	Room temperature and pressure	Room temp. -1300 psig	No leakage from coupling
5	Room temperature and pressure	Room temp. -1500 psig	No leakage from coupling
6	Room temperature and pressure	Room temp. -1500 psig	No leakage from coupling

Figure 1-8 Hydrostatic Test Results

NOTE

The first 4 tests were terminated at less than test pressure (1500 psig), due to test rig leakage external to coupling assembly.

7.7.2 Rated Temperature. The unit failed before the manufacturer's rated temperature (400°F to 3000 psig) was attained. The preliminaries of the test were conducted for five hours between 340°F and 370°F at a maximum pressure of 900 psig. The coupling did not leak during this period. As the system cooled, a leak developed in the coupling joint. This leak was caused by the Viton seal failure.

7.8 Final Inspection

7.8.1 The socket-half coupling was tested at 1500 psig at room temperature. No leakage was evident. The nipple-half coupling leaked at a rate of 122 drops per minute with a gravity head of 20 feet of water. A visual inspection of the disassembled nipple-half coupling revealed failure of the Viton "O" ring.

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SECTION II

GRAYLOC CLAMP AND SEAL TEST REPORT

1.0 PURPOSE:

1.1 The purpose of this report is to present and evaluate the results of a test program which involved mechanical and thermal cycling of a commercial clamp assembly and seal ring.

2.0 SCOPE:

2.1 The scope of this test was to cycle the clamp assembly and seal ring up to and including Douglas Point Plant environmental conditions in order to substantiate the clamping and sealing concepts and establish component reliability. Test criteria used in reaching the test results are as follows:

- a. Determine the ability of seals to withstand repeated cycling at incremental pressures from atmospheric to 1500 psig.
- b. Determine the ability of seals to withstand repeated cycling at 1500 psig pressure at temperatures from 180° to 568°F.
- c. Determine wear between clamping and sealing surfaces.
- d. Determine leakage through seal under following conditions:
  - 1) Atmospheric pressure to 1500 psig.
  - 2) Temperatures from 180° to 568°F at 1500 psig pressure.

2.2 The unit tested was a standard Grayloc Clamp Assembly, Model number GR20 having a 316 stainless steel seal ring and hub.

3.0 SUMMARY:

The test program was divided into two series, a hydrostatic test and a thermal test.

3.1 The hydrostatic series, consisting of testing the clamp and seal at room temperatures and pressures to 1500 psig, was completed successfully.

3.2 The thermal series, consisting of testing the clamp and seal at 1500 psig pressure and temperatures from room ambient to 575°F, was successful for 40 thermal-mechanical cycles.

4.0 CONCLUSION:

4.1 As a result of this test, the 316 stainless steel seal ring was found incapable of withstanding the repeated mechanical cycling of the 700-cycle refueling machine design requirements.

4.2 This conclusion, however, is limited to the single combination (316 SS seal ring and 316 SS hubs). Part of the failure may be attributed to galling which occurred between two similar materials under wear-test conditions.

5.0 RECOMMENDATIONS:

5.1 It is recommended that the use of 316 stainless steel be discontinued and a more compatible combination of materials be selected.

5.2 It is also recommended that investigations be made of the stress capabilities and relaxation rate of substitute seal ring material at elevated temperatures. The massive failure experienced during

test was due to the 316 SS seal ring taking a permanent set.

It is recommended that 410 SS be used in lieu of 316 SS and that a similar test be run.

5.3 It is further recommended that the method of seal ring and hub attachment be changed. The seal ring should be retained and guided, rather than held firm. This method should preclude an initial seal ring preset which might occur under the present method of attachment.

5.4 The test setup is available and may be used again as required.

6.0 TESTING:

6.1 Test Setup

The seal ring was mounted in a test rig (figures 2-1 and 2-2) and the rig attached to a test stand. High-temperature water was piped to the test rig as shown in flow system schematic diagram (figure 2-3). The seal ring was attached to one clamp half with set screws, preloading the seal ring. (This method is according to the original reference design of the valve, clamp and seal-ring assembly.)

The reference seal ring is contained within the hub-half of the adapter or shutoff valve, and positioned in the mating hub which is either the reactor end fitting or the outboard end of the adapter valve, as applicable. To simulate the above conditions, the second Grayloc hub-half was mounted in the test rig above the first hub (with attached seal). The clamp held both hubs and seal ring securely. The relative flexibility of the tubing,

FIGURE 2-1. GRAYLOC CLAMP AND SEAL TEST RIG

FIGURE 2-2. GRAYLOC CLAMP AND SEAL TEST RIG SHOWING HUB AND SEAL

FIGURE 2-3. TEST SCHEMATIC DIAGRAM

however, permitted compatible assembly expansion. The clamp hubs were joined by the locking clamps. The seal ring was positioned in the second hub.

6.2 Test Procedure

6.2.1 After each thermal cycle, the clamps were separated. The hubs enclosing the seal ring were forced open, separating both clamp halves. This simulated both the machine operation and thermal operating conditions which the seal ring and sealing surfaces would experience during normal refueling machine operation.

6.2.2 The clamp halves were opened and sealed by a cylinder and through a linkage system. The linkage system provided a positive mechanical lock when the air cylinder and linkage were in the down or clamp-locked position. This locking method precluded the possibility of the rig opening upon loss of air pressure.

6.2.3 The test medium (water) was admitted to the test rig and test assembly through valve V-1 in the pump discharge line (figure 2-3). Air was vented by opening valve V-2. After the system was vented, valves V-1 and V-2 were closed. Valve V-3 (connected to the high-temperature water supply) and valve V-4 (connected to the discharge manifold) were opened, permitting water circulation in test apparatus.

6.2.4 The test was conducted from five to twelve minutes at test temperatures (figure 2-4). Leakage was determined by swabbing the seal ring joint with white, moisture-absorbent paper. The test apparatus was isolated from the main high-temperature supply by closing valves V-3 and V-4. Water was drained from the assembly through Valve V-5.

.0 TEST DATA:

Test data, generated during test program, is presented as follows:

- a. Delivery Inspection Report
- b. Cleaning Procedures
- c. Hydrostatic Tests - Ambient Temperature
- d. Hydrostatic Tests - Thermal Conditions
- e. Final Inspection Report

7.1 Delivery Inspection Report

The Grayloc clamp, hubs and seal ring came assembled as part of the test assembly. The seal ring was removed from the rig and placed in a lathe to determine any out-of-round condition of the sealing surface. A dial indicator gage showed no runout.

7.2 Cleaning Procedure

The internal surfaces of the test assembly were degreased with trichloroethylene followed by a detergent cleaning. The surfaces were then rinsed with demineralized water and dried with a hot air blower.

The surfaces were inspected with a white, lint-free cloth. No traces of dirt were apparent.

7.3 Hydrostatic Tests - Ambient Temperature

The entire assembly, including the seal ring and hubs, was pressure tested at 2250 psig for a period of four hours. No sign of leakage or drop in pressure in the closed hydraulic test setup was evident.

7.4 Hydrostatic Tests - Thermal Conditions

The assembly was tested at various temperatures and pressures up to the maximum of 1500 psig and 530°F. After each cycle, the assembly was mechanically disassembled and the seal ring lifted from its seat. The temperature within the test assembly stabilized in two minutes at the lower temperatures (350°F and below) and in four minutes at the higher temperatures (500°F). Figure 2-4 tabulates the results of this test.

<u>Test Run No.</u>	<u>Initial Condition</u>	<u>Final Condition</u>	<u>Test Duration</u>	<u>Remarks</u>
1	Room Temp and Pressure	950 psig, 335°F	25 minutes	Bolts tightened and water leakage stopped.
2	"	1000 psig, 375°F	25 minutes	No visible leakage.
3	"	1000 psig, 300°F	20 minutes	No visible leakage 6 mechanical cycles.
4	"	1000 psig, 300°F	15 minutes	No visible leakage.
5	"	900 psig, 410°F	5 minutes	No visible leakage.
6	Room Press & 280°F	950 psig, 325°F	5 minutes	"
7	" & 180°F	950 psig, 320°F	10 minutes	"
8	" & 180°F	1000 psig, 320°F	8 minutes	"
9	" & 110°F	1000 psig, 320°F	8 minutes	"
10	" & 200°F	1000 psig, 320°F	10 minutes	"
11	" & 204°F	950 psig, 320°F	10 minutes	"

FIGURE 2-4. THERMAL TEST DATA (SHEET 1 OF 3)

<u>Test Run No.</u>	<u>Initial Condition</u>	<u>Final Condition</u>	<u>Test Duration</u>	<u>Remarks</u>
12	Room Press & 120°F	900 psig, 340°F	10 minutes	No visible leakage.
13	" & 210°F	950 psig, 341°F	10 minutes	"
14	" & 210°F	925 psig, 339°F	12 minutes	"
15	" & 70°F	1300 psig, 335°F	5 minutes	Slight leak, tightened bolts.
16	" & 100°F	1350 psig, 350°F	5 minutes	No visible leakage.
17	" & 225°F	1400 psig, 350°F	5 minutes	"
18	" & 220°F	1475 psig, 355°F	7 minutes	"
19	" & 220°F	1500 psig, 360°F	8 minutes	"
20	" & 232°F	1475 psig, 375°F	7 minutes	"
21	" & 235°F	1375 psig, 370°F	7 minutes	"
22	" & 235°F	1200 psig, 370°F	6 minutes	"
23	" & 235°F	1100 psig, 385°F	6 minutes	"
24	" & 230°F	1050 psig, 385°F	5 minutes	"
25	" & 105°F	1300 psig, 405°F	5 minutes	Slight weep.
26	" & 235°F	800 psig, 405°F	5 minutes	Tightened clamp, slight weep.
27	" & 250°F	900 psig, 425°F	5 minutes	Adjusted clamp, slight weep.
28	" & 280°F	550 psig, 430°F	3.5 minutes	Slight weep.
29	" & 170°F	1000 psig, 435°F	4 minutes	Slight weep.
30	" & 205°F	1125 psig, 435°F	4 minutes	Adjusted clamp, slight weep.
31	" & 220°F	1175 psig, 460°F	4 minutes	Slight weep.
32	" & 280°F	1125 psig, 450°F	4 minutes	Adjusted clamp, slight weep.
33	" & 270°F	1075 psig, 445°F	3 minutes	Adjusted clamp, 5 drips of water per min.

FIGURE 2-4. THERMAL TEST DATA (SHEET 2 OF 3)

<u>Test Run No.</u>	<u>Initial Condition</u>	<u>Final Condition</u>	<u>Test Duration</u>	<u>Remarks</u>
34	Room Press & 330°F	1050 psig, 455°F	4 minutes	Adjusted clamp, slight weep.
35	" & 100°F	1550 psig, 280°F	10 minutes	No sign of leakage.
36	" & 180°F	1000 psig, 300°F	5 minutes	No sign of leakage.
37	" & 100°F	875 psig, 405°F	6 minutes	Slight leak.
38	" & 260°F	900 psig, 410°F	5 minutes	Adjusted clamp, slight leak.
39	" & 250°F	950 psig, 420°F	4 minutes	"
40	" & 100°F	900 psig, 420°F	4 minutes	Adjusted clamp, no sign of leakage.
41	" & 90°F	1500 psig, 500°F	6 minutes	Seal leaked.
42	" & 90°F	1550 psig, 528°F	6 minutes	Excessive leakage.
43	" & 90°F	1500 psig, 360°F	2 minutes	Seal leaked even with nuts torqued to 180 ft-lb. 20 ft-lb required to seal clamps initially. Test stopped because seal leaked excessively.
44	" & 190°F	1425 psig, 360°F	2 minutes	"
45	" & 215°F	1325 psig, 420°F	2 minutes	"
46	" & 150°F	400 psig, 420°F		"

FIGURE 2-4. THERMAL TEST DATA (SHEET 3 OF 3)

7.5 Final Inspection

The assembly was pressure tested to 1500 psig at a temperature of 525°F, for a period of 32 minutes. Steam appeared at the clamped joint. The bolts were tightened to 180 foot-pounds torque, but the joint continued to leak.

FIGURE 2-5. SEAL RING DEFORMATION

The clamp assembly was dismantled and inspection revealed seal ring deformation (figure 2-5).

The seal ring and hub showed galling of the seal surfaces (see figures 2-6 and 2-7).

The seal ring was removed from the test apparatus and the test program stopped.

FIGURE 2-6. SEAL RING GALLING

FIGURE 2-7. HUB GALLING

SECTION III

HYDRAULIC RAM POSITION - INDICATION TEST REPORT

1.0 PURPOSE:

The purpose of this report is to document the results of a test program involving two methods of remotely locating and indicating the position of a hydraulic ram within a hermetically-sealed pressure tube.

2.0 SCOPE:

2.1 The scope of this test is to determine the feasibility of two different position indicating systems for the hydraulic ram within the AMF On-Power Refueling Machine.

2.2 One system utilized a proximity sensor, a soft iron slug, and an audible annunciator (figure 3-1), and magnetic actuators. The second system used a magnetic-actuated switch. Two switches were tested; one, a reed-type glass switch (figure 3-2) and, the other, an epoxy-coated ball switch (figure 3-3). Each system was tested using four magnet (figure 3-4) and keeper (figure 3-5) combinations (a soft iron ring used to shape the magnetic flux field and to hold the magnet in place on the ram). These magnet and keeper combinations were selected to suit design configurations of the various rams.

2.3 Objectives of the test program were to:

- a. Determine capability of various magnet and keeper combinations to actuate switching devices to indicate ram position at discrete intervals.

FIGURE 3-1. PROXIMITY SENSOR, AUDIBLE ANNUNCIATOR AND SOFT IRON SLUG

FIGURE 3-2. REED-TYPE GLASS SWITCH

FIGURE 3-3. EPOXY-COATED BALL SWITCH

M-1

ALNICO V

M-2

ALNICO V

M-3

ALNICO V

M-4

ALNICO V

FIGURE 3-4. MAGNET CONFIGURATIONS

FIGURE 3-5. KEEPER CONFIGURATIONS

b. Determine optimum magnet and keeper arrangement for a particular ram configuration.

c. Determine the most feasible switching arrangement, both at room ambient conditions and at Douglas Point Plant operating temperature.

2.4 The first system of position indicating equipment tested utilized a proximity sensor (Honeywell Model No. CSX-14089), a soft iron slug, and an audible annunciator. The second system utilized two similarly functioning units, a reed-type glass switch (C. P. Clair Company), and an epoxy-coated ball switch (Aerotec Industries).

3.0 SUMMARY:

3.1 Due to the erratic nature of actuation of the proximity sensor and the epoxy-coated ball switch, it was not possible to obtain reliable quantitative data.

3.2 The following repeatable air gap actuation limits were measured using the reed-type glass switch:

	<u>Room Temperature Tests</u>				<u>560° to 600°F Tests</u>			
Magnetic-Keeper Combination	1-1	1-2	3-3	3-4	1-1	1-2	3-3	3-4
Minimum Air Gap	1.86	1.59	1.66	1.72	1.75	1.59	1.81	1.56
Maximum Air Gap	2.56	2.25	1.97	2.00	2.38	2.22	1.88	1.81

4.0 CONCLUSION:

4.1 Test results above indicate that the reed-type glass switch was not effected by environmental changes, and worked well under all conditions. It may be used to locate a ram along discrete points of travel within

a pressure vessel and to provide positive indication at a remote location. Points of glass-switch actuation and de-actuation vary depending upon magnetic-field intensity, direction of ram travel, and the gap between the switch and the magnet. For best results the glass switch should be positioned parallel (figure 3-6) to the longitudinal axis of the ram and magnet with the thin side adjacent to the tube.

4.2 The proximity sensor does not provide reproducible results when subject to slight temperature fluctuations. It also actuates and de-actuates in only one direction. It is expensive and requires complex circuitry to provide remote indications. The magnetic field interaction of each sensor, when placed close together, causes false points of actuation and de-actuation.

4.3 The design of the epoxy-coated ball switch made it very sensitive to vibration. As a result, the switch did not actuate or de-actuate with reproducible results.

5.0 RECOMMENDATIONS:

5.1 The following recommendations are made as a result of the test program:

- a. Use the reed-type glass switch as means of providing position indication. Do not use the epoxy-coated ball switch or proximity sensor schemes.
- b. Indications within two inches from the end points of ram travel may be better attained using another method. A limit switch concept is suggested.

FIGURE 3-6. GLASS SWITCH POSITION

- c. Use non-magnetic materials to preclude the possibility of false indications.
- d. Protect switches with aluminum shields and install in place with an aluminum mounting device.
- e. Locate the switch a minimum of 0.25 inch from the pressure tube.

NOTE

The switch may be mounted anywhere around the tube periphery when using ring-type magnets.

6.0 TEST:

6.1 Test Equipment

- a. Test Rig (figure 3-7) - Fabricated to simulate a ram within a hermetically-sealed pressure tube.
- b. Magnets and Keepers (figure 3-8) - Mounted on the ram and placed within the simulated pressure tube.
- c. Flash Light and Circuit (figure 3-9) - Used to establish actuation and de-actuation points.
- d. Model SE120 Chromolex Strip Heaters (120V, 500W) - Used to establish temperature requirements for test.
- e. Non-magnetic Shim Stock - Used to establish gap between switches and pressure tube.

6.2 Test Setup

- a. The test rig was supported at each end with a suitable bracket.

FIGURE 3-7. POSITION INDICATOR TEST RIG

FIGURE 3-8. CUTAWAY SHOWING MAGNET AND KEEPER TEST RELATIONSHIP

FIGURE 3-9. FLASHLIGHT CIRCUIT

- b. The Chromalex heaters were secured at each end of the test rig and connected to a suitable power source.
- c. Combinations of magnets and keepers were attached to the simulated ram and placed within the tube.
- d. The epoxy-coated ball switch and the reed-type glass switch were wired into the flashlight circuit and placed flush against, or at a specified distance from (using non-magnetic shim stock) the simulated tube (see figure 3-6). Vellum, used to chart the points of actuation and de-actuation, was placed under the switch being tested.

6.3 Test Procedure

6.3.1 Proximity Sensor

The proximity sensor was placed adjacent to the tube and moved manually. Points of actuation and de-actuation were determined by audible noise from the sensor device. Slight temperature changes caused premature actuation. Considering these factors, the test of this device was discontinued.

6.3.2 Epoxy-Coated Ball Switch

The epoxy-coated ball switch was placed under the tube with the pole end up and the longitudinal axis perpendicular to the tube. The switch was moved manually.

NOTE

Positioning of this switch is critical due to internal construction.

6.3.2.1 Cold Testing - Weak magnetic flux field and general switch sensitivity to vibration precluded the gathering of recordable data for this test.

6.3.2.2 Cold Testing after Heat Cycling - The test vessel, including magnets and keepers, was heat cycled from 560°F to 500°F, 16 hours a day for five days. Again, weak magnetic flux fields and general switch sensitivity to vibration precluded the gathering of recordable data.

6.3.2.3 Heat Cycling - The vessel was tested at 560°F to 600°F. No recordable data was obtained due to weak magnetic flux fields and general switch sensitivity.

6.3.3 Reed-Type Glass Switch

6.3.3.1 Cold Testing (see figures 3-10 and 3-11) - Magnets M-2 and M-4 were too weak to actuate the switch. Test data was obtained from Magnets M-1 and M-3 used in the following keeper combinations: K-1 and K-2 with M-1, and K-3 and K-4 with M-3. The results of testing are as follows (all dimensions are in inches, all decimal dimensions having a _ 0.05 inch tolerance):

(Magnet and Keeper Combinations)

	<u>1-1</u>	<u>1-2</u>	<u>3-3</u>	<u>3-4</u>
Minimum Air Gap Between Tube Wall and Switch	1/4	1/4	Flush	Flush
Minimum Air Gap Actuation Limits	1.86	1.86	1.81	1.81

(A) SWITCH CONTACT LIMITS AT ROOM TEMPERATURE

FIGURE 3-10. REED-TYPE GLASS SWITCH COLD TEST RESULTS

FIGURE 3-11. FLUX FIELDS AT ROOM TEMPERATURE CONDITIONS

(Magnet and Keeper Combinations)

	<u>1-1</u>	<u>1-2</u>	<u>3-3</u>	<u>3-4</u>
Maximum Air Gap Between Tube and Wall Switch	1-1/2	1-1/4	1/2	1/2
Maximum Air Gap Actuation Limits	2.56	2.22	1.97	2.00

FIGURE 3-12. REED-TYPE GLASS SWITCH COLD TEST RESULTS

6.3.3.2 Cold Testing after Heat Cycling (see figures 3-13 and 3-14) - The test vessel was heated from room ambient to a temperature between 560° and 600°F in intervals of one hour per cycle for 16 hours over a period of five days. Magnets M-1 and M-3 were used in combination with keepers K-1, K-2, K-3 and K-4. The results of testing were as follows (all dimensions are in inches, all decimal dimensions having a .005 inch tolerance):

(Magnet and Keeper Combinations)

	<u>1-1</u>	<u>1-2</u>	<u>3-3</u>	<u>3-4</u>
Minimum Air Gap Between Tube Wall and Switch	1/4	1/4	Flush	Flush
Minimum Air Gap Actuation Limits	1.94	1.59	1.66	1.72
Maximum Air Gap Between Tube and Wall Switch	1-1/2	1-1/4	3/4	1/2
Maximum Air Gap Actuation Limits	2.53	2.25	1.97	1.84

FIGURE 3-15. REED-TYPE GLASS SWITCH COLD TEST
AFTER HEAT CYCLING RESULTS

(B) SWITCH CONTACT LIMITS AT ROOM TEMPERATURE
AFTER HEAT CYCLING OF TEST APPARATUS

FIGURE 3-13. REED-TYPE GLASS SWITCH COLD TEST AFTER HEAT CYCLING RESULTS

FIGURE 3-14. FLUX FIELDS AFTER THERMAL CYCLING

6.3.3.3 Hot Testing (see figure 3-16) - Hot testing was accomplished at a temperature of 560° to 600°F using the same magnet and keeper combinations tested in the previous reed-type glass switch tests. The results were as follows (all dimensions are in inches, all decimal dimensions have a _ 0.05 inch tolerance):

(Magnet and Keeper Combinations)

	<u>1-1</u>	<u>1-2</u>	<u>3-3</u>	<u>3-4</u>
Minimum Air Gap Between the Wall and Switch	1/4	1/4	Flush	Flush
Minimum Air Gap Actuation Limits	1.75	1.59	1.81	1.56
Maximum Air Gap Between Tube Wall and Switch	1-1/2	1-1/4	1/2	1/2
Maximum Air Gap Actuation Limits	2.38	2.22	1.88	1.81

FIGURE 3-17 REED-TYPE GLASS SWITCH HOT TEST RESULTS

7.0 TEST DATA:

The following test data, a compilation of pertinent test results, was obtained by testing the reed-type glass switch in combination with magnets M-1 and M-3 and keepers K-1, K-2, K-3 and K-4. All dimensions are in inches and all decimal dimensions are toleranced _ 0.05 inch.

7.1 Magnet M-1 and Keeper K-1

1. Flush position - not recommended as switch actuates and de-actuates twice in one direction.
2. 1/4 in. gap - from 2.09 min hot to 2.19 max cold.

(C) SWITCH CONTACT LIMITS AT 580°F.

FIGURE 3-16. REED-TYPE GLASS SWITCH HOT TEST RESULTS

3. 3/4 in. gap - from 2.28 min hot to 2.41 max cold.
4. 3/4 in. gap - from 2.38 min hot to 2.56 max cold.
5. 1 in. gap - from 2.34 min hot to 2.53 max cold.
6. 1-1/4 in. gap - from 2.16 min hot to 2.44 max cold.
7. 1-1/2 in. gap - from 1.75 min hot to 2.16 max cold.
8. 1-3/4 in. gap - no actuation at this point.

7.2 Magnet M-1 and Keeper K-2

1. Flush position - not recommended as switch actuates twice in one direction.
2. 1/4 in. gap - from 1.88 min hot to 1.93 max cold.
3. 1/2 in. gap - from 2.00 min cold to 2.13 max cold.
4. 3/4 in. gap - from 2.13 min hot to 2.2 max cold.
5. 1 in. gap - from 2.09 min hot to 2.22 max cold.
6. 1-1/4 in. gap - from 1.81 min cold to 2.13 max cold.
7. 1-1/2 in. gap - no actuation at this point.

7.3 Magnet M-3 and Keeper K-3

1. Flush position - from 1.81 min hot to 1.94 max cold.
2. 1/4 in. gap - from 1.84 min hot to 1.97 max cold.
3. 1/2 in. gap - from 1.84 min hot to 1.97 max cold.

4. 3/4 in. gap - actuate only cold after heat cycling 1.66 to 1.66.

5. 1 in. gap - no actuation.

7.4 Magnet M-3 and Keeper K-4

1. Flush position - from 1.69 min hot to 1.88 max cold.

2. 1/4 in. gap - from 1.67 min hot to 1.88 max cold.

3. 1/2 in. gap - from 1.56 min hot to 1.91 max cold.

4. 3/4 in. gap - no actuation.

SECTION IV

BALL VALVE SEAL TEST REPORT

1.0 PURPOSE:

The purpose of this report is to present, document and evaluate the results of a test program conducted to determine the feasibility and reliability of the ball seal principle, as embodied in the Reference Adapter Valve design for the AMF On-Power Refueling System for the Douglas Point Plant.

2.0 SCOPE:

2.1 The scope of this test was to cycle the ball valve under simulated system operating conditions against silver-impregnated seals of different material composition in order to determine the ability of seals to preclude leakage.

2.2 The seal materials selected for test purposes are as follows:

- a. Silver-impregnated carbon-graphite material, Grade B1-02
(supplied by Electro-Nite Carbon Company, Philadelphia, Pennsylvania). (See figure 4-1.)
- b. 430 stainless steel, 30 per cent dense, impregnated with silver
(supplied and developed by Armour Research Foundation, Illinois Institute of Technology, Chicago, Illinois). (See figure 4-1.)

2.3 The seals were tested in an environment of demineralized water at a pH of 9.5 to 10.5 at 1500 psig pressure and a temperature of 568°F.

3.0 SUMMARY:

At the conclusion of the first series of tests on the silver impregnated carbon-graphite seal, the average leakage rate was 6.0 cc per hour. The

FIGURE 4-1. SILVER IMPREGNATED SEALS: CARBON-GRAPHITE AND 430 STAINLESS STEEL

range of leakage values decreased from 28.8 to 1.5 cc per hour with repeated lapping of the seal material.

At the conclusion of the second series of tests on the silver impregnated stainless steel seal, the average leakage rate was 23.0 cc per hour. The range of leakage values decreased from 51.5 to the 23.0 cc per hour with repeated lapping of the seal material.

These leakage values may be compared with the maximum permissible average leakage rate at the ball valve seal of approximately 300.0 cc per hour.

4.0 CONCLUSION:

4.1 The results of this test indicate that both the silver impregnated carbon-graphite seal and the silver-impregnated stainless steel seal satisfy the sealing requirements of the ball-seal combination of the Reference Adapter Valve.

4.2 Although leakage rates at the ball seal may normally be expected to increase during the estimated life span of the adapter valve as a result of wear, it is not anticipated that they will ever exceed the maximum permissible rate of 300 cc per hour at any time during adapter valve life.

4.3 The silver impregnated carbon-graphite seal leakage rate was lower than that obtained for the silver impregnated stainless steel seal. It should be noted that lapping was performed on both seals. More extensive lapping was done on the carbon-graphite seal, however, and it is assumed that if comparable lapping was performed on the silver-impregnated stainless steel seal the leakage rate of this seal would be considerably reduced.

4.4 The silver-impregnated stainless steel seal is stronger and easier to handle than the silver impregnated carbon-graphite seal. This was indicated when one of the carbon-graphite type seals cracked because of mishandling.

5.0 RECOMMENDATIONS:

5.1 Because of the current commercial inavailability of the silver-impregnated stainless steel seal material, it is recommended that the silver impregnated carbon-graphite seal continue to be used as the Reference Adapter Valve ball valve seal.

5.2 The silver-impregnated stainless steel seal should be considered as a substitute for the carbon-graphite seal because of its strength and sealing capability.

5.3 If the silver-impregnated stainless steel material becomes more readily available, further tests should be made on this type of seal to determine the amount of lapping and burnishing necessary to obtain optimum sealing capability.

6.0 TEST PROCEDURE:

6.1 Test Apparatus (See figures 4-2, 4-3, and 4-4)

6.1.1 The basic test vehicle consisted of a two-inch commercially available 316 stainless steel ball valve altered to accept seals fabricated from different materials, and a load ring. Loads were applied to the seal through use of a ram and force multiplying-linkage system. This system multiplied the weight placed on the hanger arm by a mechanical advantage of 100. The valve and test rig was insulated to retain heat.

FIGURE 4-2. DISASSEMBLED 2-INCH COMMERCIAL BALL VALVE AND SEALS

FIGURE 4-3. DISASSEMBLED BALL VALVE AND TEST RIG

FIGURE 4-4. BALL VALVE SEAL TEST RIG SCHEMATIC

6.1.2 Leakage measuring apparatus consisted of an Ehrlenmeyer flask filled with a desiccant (Military Specification MIL-D-3464).

Dry nitrogen, pressurized to 2.5 psi, was directed through the test chamber, through the leakage measuring apparatus and discharged to the atmosphere. Leakage was determined by:

- a. Accurately weighing the dry flask and desiccant prior to each test run.
- b. Connecting the dry flask and desiccant to the test chamber for test runs.
- c. Weighing the flask and partially saturated desiccant after each test run had been performed for a pre-determined time interval.

The difference in desiccant weight before and after each test run represented ball valve seal leakage during that particular test run.

A thermocouple was placed in the environmental chamber outlet port to monitor the test package temperature.

6.2 Test Procedure

The following procedure was used for both series of tests:

- a. The ball valve ball was rotated to the closed position.
- b. The test fluid, under specified temperature and pressure, was introduced to the environmental chamber and circulated through the test rig.

- c. The ball valve seal was loaded by placing varying weights on the test rig hanger arm. Each weight constituted a separate test run.
- d. The desiccant, dehydrated prior to each test run by oven-heating at 350°F for 15 to 30 minutes, was placed in the flask. The flask was weighed before and after each test run to determine leakage values.

Corollary procedures, performed on the ball valve seals, were also followed to ensure that leakage readings would be as optimal as possible and reflect the true sealing capabilities of the two sealing materials that were tested. The techniques used to reduce surface irregularities of the seals were lapping and burnishing.

The lapping process performed on the silver impregnated carbon-graphite seals consisted of rotating the ball valve ball through a 90-degree arc (from OPEN to CLOSED position) while under various force loads and with the test rig dry. Force applied and frequency of rotation of the ball from open to closed position was varied to develop the most efficient lapping procedure. It was determined that an initial weight on the test rig hanger arm of five pounds and completion of 12 rotations, with the weight progressively reduced thereafter, resulted in the most significant diminution of leakage past the seal. No abrasive was used since this seal material was soft enough to be self-abrading when in frictional contact with the stainless-steel ball.

After each lapping, the ball valve was disassembled and the seal and parts were washed in trichloroethene and rinsed in demineralized water. Prior to disassembly, the seal, valve body, ball, and test rig were marked with orientation marks to ensure proper alignment when re-assembled. Care was taken to preserve these orientation marks.

This lapping process proved to be quite involved and extensive trials were required before the most effective sealing surface was obtained.

Lapping was performed on three different silver impregnated carbon-graphite seals. The first seal was inadvertently dropped and broken after it had been lapped to a degree that resulted in improved sealing characteristics. After lapping and testing for eight to ten cycles, the second seal developed a chip and then a crack which rendered it unserviceable (figure 4-5). The third seal was successfully lapped and remained in service for the remainder of the test program.

The burnishing process applied to the silver-impregnated stainless steel seals differed from the lapping technique because of the nature of the seal material. A process similar to that developed for lapping was first tried but proved to be too slow and tedious to produce comparable results.

The ball valve ball was then placed in a triple-jaw lathe chuck and rotated at approximately 75 rpm. The seal was held manually against the rotating ball for 15 to 20 minute intervals, periodically removed and washed in kerosene. Linseed oil was the only abrasive used at first and then a mixture of diamond powder and linseed oil to hasten the process.

FIGURE 4-5. SILVER-IMPREGNATED CARBON-GRAPHITE SEAL CRACKED DURING TEST

This burnishing technique was continued for a total of eight to ten hours and resulted in more effective sealing capability. Repetition of the entire burnishing process was carried out four to six times to obtain minimal leakage values for this type of seal (see figure 4-6).

A major problem encountered as burnishing progressed was the tendency of the hard-seal material (430 stainless steel) to score the softer (316 stainless steel) valve ball. This problem was solved by using one ball for burnishing purposes exclusively and obtaining another for use in the test rig. Another problem area involved avoidance of slurring the softer silver content of the seal metal composite over the entire sealing surface area during burnishing. This problem was essentially solved through manual control of the force applied to achieve the burnishing effect and by the use of jeweler's rouge as an abrasive in the final stages of the burnishing procedure.

7.0 TEST DATA:

TEST SERIES 1 - Silver Impregnated Carbon-Graphite Seal

Lapping operations were completed twice prior to commencement of this test run.

<u>Test No.</u>	<u>Pressure psi</u>	<u>Temperature of Valve °F*</u>	<u>Ram Force on Seal</u>	<u>Leakage cc/hr</u>
1	1400	490	150	4.3
	1350	505	250	2.8
	1300	515	350	5.3
	1250	515	500	2.8
	1250	510	600	3.6
	1200	515	750	4.1
	1175	515	850	0.7
	1100	550	1000	0.9
	1075	530	1250	0.4
	800	510	1500	0.0

*Temperature readings shown have been adjusted to allow for radiant heat loss.

FIGURE 4-6. SILVER IMPREGNATED STAINLESS STEEL SEAL
ON COMPLETION OF LAPPING AND BURNISHING

<u>Test No.</u>	<u>Pressure psi</u>	<u>Temperature of Valve °F</u>	<u>Ram Force on Seal</u>	<u>Leakage cc/hr</u>
2	1325	545	150	14.2
	1300	550	250	19.8
	1300	550	350	12.8
	1300	550	500	8.5
	1300	550	000	5.1
	1300	550	750	6.9
	1275	545	850	7.4
	1275	545	1000	3.0
	1275	545	1250	5.1
	1275	540	1500	5.1
	1275	540	1500	2.8
	1250	540	1250	3.0
	1250	545	1000	4.2
	1250	545	850	1.2
	1200	550	750	2.8
	1050	550	600	2.5
	825	540	500	5.7
	500	530	350	4.7
	225	495	250	10.2
	0	440	150	-
3	1375	470	100	7.9
	1350	475	150	10.9
	1350	475	250	11.0
	1375	470	350	5.5
	1400	470	500	9.0
	1425	465	600	8.2
	1375	460	750	4.8
	1350	455	850	4.6
	1225	450	1000	5.1
	1000	450	1250	3.8
	675	450	1500	4.5
	375	450	1650	8.2
4	1375	520	100	8.3
	1375	545	150	13.0
	1275	535	250	23.2
	1275	555	350	17.5
	1300	555	500	4.2
	1325	550	600	10.3
	1375	550	750	8.5
	1375	550	850	8.6
	1400	550	1000	7.0

The seal used during this test run was found to be cracked on post-run examination and was discarded.

<u>Test</u> <u>No.</u>	<u>Pressure</u> <u>psi</u>	<u>Temperature</u> <u>of Valve °F</u>	<u>Ram Force</u> <u>on Seal</u>	<u>Leakage</u> <u>cc/hr</u>

4 (continued)

1450	547	1250	5.7
1475	550	1500	5.3
1475	555	1650	6.3
1450	560	1500	6.0
1450	563	1250	5.4
1450	563	1000	4.9
1475	530	850	1.9
1500	480	750	1.9
1500	540	600	1.9
1500	535	500	1.9
1500	530	350	1.8
1475	525	250	1.8
1375	510	150	1.7
1325	500	100	4.1

The new seal placed in service for this and subsequent test runs in this series was carefully and extensively lapped, utilizing experience gained in previous handling of this type of seal.

5	1400	515	100	14.8
	1400	520	150	6.8
	1400	525	250	2.0
	1400	525	350	0.0
	1400	530	500	0.1
	1375	530	600	0.1
	1375	530	750	0.1
	1400	525	850	0.7
	1450	525	1000	1.0
	1450	520	1250	0.7

Some variation in leakage values obtained during this test run and during test run 7 resulted from minor deficiencies in the leakage collection system which were corrected prior to the beginning of test run 8.

6	1200	475	1250	28.8
	1225	485	1000	25.7
	1275	495	850	26.4
	1300	500	750	24.4
	1350	505	600	22.4
	1375	510	500	22.8
	1375	510	350	1.3
	1350	515	250	13.3
	1325	520	150	15.2

<u>Test No.</u>	<u>Pressure psi</u>	<u>Temperature of Valve °F</u>	<u>Ram Force on Seal</u>	<u>Leakage cc/hr</u>
7	1100	525	100	27.2
	1000	500	150	34.7
	1000	450	350	-
	1000	310	350	16.3
	1025	315	300	10.0
	1050	315	600	6.6
	1100	300	750	2.7
	1125	315	850	4.8
	1150	315	1000	2.1
	1150	315	1250	7.7
8	1375	505	1750	10.2
	1375	505	1500	4.4
	1400	500	1250	0.7
	1450	510	1000	0.2
	1450	515	850	0.0
	1475	520	750	0.0
	1500	520	600	0.0
	1475	505	500	0.2
	1450	510	350	0.4
	1475	515	250	0.1
	1475	525	150	0.4
	1450	515	100	0.5

Final lapping of the seal was performed prior to the beginning of this test run.

9	1200	485	1000	14.5
	1200	495	1500	7.4
	1200	490	2500	2.7
	1200	485	3500	1.9
	1200	490	5000	0.0
	1200	485	6000	0.0
	1225	485	7500	0.1
	1250	495	8500	0.2
	1250	490	10000	0.0
	1275	490	12500	6.6
	1275	490	15000	0.2
	1300	490	17500	0.2
	1300	485	17500	0.0
	1300	490	15000	0.0
	1300	485	12500	0.2
	1325	485	10000	0.0
	1325	490	8500	0.0
	1325	475	7500	0.2
	1350	475	6000	0.2
	1350	475	5000	0.0
	1300	470	3500	0.0
	1350	460	2500	0.0
	1350	460	1500	0.4
	1350	460	1000	0.1

TEST SERIES 2 - Silver Impregnated Stainless Steel Seal

Extensive burnishing was performed on the seal prior to the first test run including reduction of high spots caused through use of a triple jaw chuck to hold the seal.

Some variations obtained during test runs 3 and 5 are attributable to inaccuracies of the leakage collection system. Additional precautions were taken to ensure dehydration of the desiccant and accuracy of weighing.

<u>Test No.</u>	<u>Pressure psi</u>	<u>Temperature of Valve °F</u>	<u>Ram Force On Seal</u>	<u>Leakage cc/hr</u>
1	1200	455	3500	21.8
	1050	515	3000	24.6
	1025	540	3600	35.4
	1000	540	3850	37.0
	1000	535	4000	36.2
	1000	525	4400	51.5
2	1300	485	1750	28.8
	1300	495	1500	26.3
	1350	490	1250	40.0
	1375	500	1000	31.3
	1400	500	850	28.1
	1450	505	750	33.7
	1450	510	600	33.4
	1450	515	500	39.5
	1450	510	350	40.2
	1475	510	250	38.8
	1500	515	150	37.8
	1475	520	100	40.7
	1450	520	100	42.3
	1450	510	150	39.7
	1450	500	250	43.8
	1450	505	350	40.8
	1425	505	500	37.6
	1400	510	600	44.8
	1350	505	750	43.0
	1325	500	850	17.4
3	1225	490	1000	26.4
	1175	480	1500	31.8
	1275	495	2500	25.8
	1325	490	3000	16.8
	1350	490	3500	27.6
	1375	495	4000	26.4
	1400	500	4500	25.6
	1450	495	5000	33.6
	1450	505	5500	32.2

<u>Test No.</u>	<u>Pressure psi</u>	<u>Temperature of Valve °F</u>	<u>Ram Force On Seal</u>	<u>Leakage cc/hr</u>

3 (continued)

1450	505	6000	30.0
1450	505	6500	24.7
1450	510	6900	7.1
1450	510	6900	4.8
1350	515	6500	2.6
1325	520	6000	0.6

4	1500	475	2500	32.9
	1500	480	3000	38.3
	1500	480	3500	30.7
	1475	480	4000	33.4
	1500	480	4500	20.0
	1525	480	5000	17.2
	1500	480	4500	14.1
	1500	480	4000	18.2
	1500	480	3500	14.6
	1500	480	3000	13.0
	1500	480	2500	14.3

5	1375	455	2500	7.1
	1400	455	3000	1.9
	1425	485	3500	1.1
	1450	500	4000	3.5
	1500	515	4500	-
	1500	500	5000	1.3
	1500	540	4500	0.5
	1500	565	4000	0.8
	1500	570	3500	0.1
	1500	530	3000	0.5
	1500	550	2500	2.1

6	1250	485	2000	17.8
	1200	480	3000	6.6
	1150	485	4000	4.0
	1100	490	5000	0.1
	1050	500	4000	0.5
	1000	500	3000	0.9
	1025	505	2000	9.9

8.0 FINAL INSPECTION:

At the conclusion of ball valve seal test operations, the test rig and ball valve components were disassembled and examined. All parts, with the exception of the O-rings, were found to be in operable condition. The tested seals were still serviceable and seal surfaces had maintained the high polish developed during lapping and burnishing operations.

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SECTION V

SERVICE LINE FLEXURE TEST REPORT

1.0 PURPOSE:

The purpose of this test report is to present, document and evaluate the feasibility of using high-pressure rubber hose as service lines for the AMF On-Power Refueling Machine for the Douglas Point Plant.

2.0 SCOPE:

The scope of the test performed was to determine the minimum force required to bend a pressurized high-pressure rubber hose about its minimum bend radius. The objective of this test was to determine the feasibility of using a cable takeup reel configuration for the Refueling Machine Utility Distribution System through investigating the forces required to coil a service line around a simulated takeup reel.

The rubber hose used to simulate the service line was a 0.75 inch ID Very-High-Pressure Rubber Hose, Aeroquip Part No. 1508-12.

3.0 SUMMARY:

Test results indicate the force required to bend a pressurized high-pressure hose will not present any significant problems in the design of a cable takeup reel device for the Refueling Machine Utility Distribution System. The maximum force required to bend the hose was 13 pounds. This force was exerted to bend the hose from 75 to 90 degrees around its minimum bend radius of 9 inches.

4.0 CONCLUSION:

A cable takeup reel device may be used for the Refueling Machine Utility Distribution System. No problems should be encountered related to the force required to bend the high-pressure hose around the takeup reel.

5.0 RECOMMENDATIONS:

5.1 It is recommended that the radius of the cable takeup reel be designed slightly larger than the minimum bend radius of the applicable service line. This will reduce stress on the hose and increase hose life; also, it will further reduce the already low force required to bend the hose.

6.0 TEST:

6.1 Test Setup

The test was conducted using a 0.75 inch ID high pressure rubber hose. The hose was clamped about a 90 degree arc of the circumference of an 18-inch diameter plywood disc. The plywood disc was used to simulate the minimum bend radius of hose.

The plywood disc was mounted in a fixed position with an 18-inch length of hose extending from the hose clamp, tangent to the disc circumference. The extended length of hose was capped at one end. This hose was filled with water and connected to a nitrogen cylinder. (See figure 5-1).

Using a pressure regulator in conjunction with the nitrogen cylinder, the hose was pressurized to 1500 psig at ambient temperature. A 40-pound capacity Chatillon scale was attached to the end of the extended hose length.

The weight of the extended length of hose was sufficient to bend the hose 30 degrees around the disc. The force required to bend the hose from 30 to 90 degrees around the disc was determined and recorded.

6.2 Test Procedure

6.2.1 Using the test setup shown in figure 5-1, force was applied through the scale sufficient to bend the hose from 30 to 90 degrees in 15 degree increments.

6.2.2 At each 15 degree increment the force was applied perpendicular to the extended length of hose.

6.2.3 At each 15 degree increment the extended length of hose was diminished since the hose was being wrapped around the disc (see figure 5-1).

6.2.4 At each 15 degree increment the perpendicular force applied and the extended length of hose was recorded.

FIGURE 5-1. SERVICE LINE FLEXURE TEST SETUP

7.0 TEST DATA:

Figure 5-2 summarizes the test data.

RUN NO. 1

<u>Pressure/Psig</u>	<u>Bend Angle of Hose/Degrees</u>	<u>Total Force Req'd To Bend Hose/Lb.</u>	<u>Extended Length of Hose/Inches</u>
1. 1500	15	Own Weight	18
2. 1500	30	Own Weight	16
3. 1500	45	2	14
4. 1500	60	4	12
5. 1500	75	8	10
6. 1500	90	13	8

RUN NO. 2

1. 1500	15	Own Weight	18
2. 1500	30	Own Weight	16
3. 1500	45	2	14
4. 1500	60	4	12
5. 1500	75	8	10
6. 1500	90	13	8

RUN NO. 3

1. 1500	15	Own Weight	18
2. 1500	30	Own Weight	16
3. 1500	45	2	14
4. 1500	60	4	12
5. 1500	75	8	10
6. 1500	90	13	8

RUN NO. 4

1. 1500	15	Own Weight	18
2. 1500	30	Own Weight	16
3. 1500	45	2	14
4. 1500	60	4	12
5. 1500	75	8	10
6. 1500	90	13	8

FIGURE 5-2. SERVICE LINE FLEXURE TEST RESULTS (SHEET 1 OF 2)

RUN NO. 5

<u>Pressure/Psig</u>	<u>Bend Angle of Hose/Degrees</u>	<u>Total Force Req'd To Bend Hose/LB.</u>	<u>Extended Length of Hose/Inches</u>
1. 1500	15	Own Weight	18
2. 1500	30	Own Weight	16
3. 1500	45	2	14
4. 1500	60	4	12
5. 1500	75	8	10
6. 1500	90	13	8

FIGURE 5-2. SERVICE LINE FLEXURE TEST RESULTS (SHEET 2 OF 2)

SECTION VI

GOODRICH RADIAL SEAL TEST REPORT

1.0 PURPOSE:

The purpose of this report is to present, evaluate and document the results of testing a radial seal which could be used as a reactor pressure tube end closure matching the existing design of the end fittings for the Douglas Point Plant.

2.0 SCOPE:

2.1 The scope of this test was to cycle the radial seal assembly up to and including Douglas Point Plant environment conditions to substantiate sealing concepts and establish component reliability. Test criteria used in reaching the test results are as follows:

- a. Obtain leakage rates through the radial seal at room temperatures and with the upstream side pressurized from zero to 1500 psig and from 1500 psig down to zero.
- b. Obtain leakage rates through the radial seal under operating pressure and temperature.
- c. Perform life cycling consisting of 50 mechanical and a minimum of 50 thermal cycles.

3.0 SUMMARY:

The initial test series resulted in very little leakage at pressures greater than 300 psig. After specific adjustments were made to the seal assembly to preclude hub deformation, additional tests were performed resulting in leakage in varying degrees and interface brinelling. Repeated thermal and

mechanical cycling tests revealed good sealing characteristics. The seal was subjected a total of; 7 mechanical cycles, 20 thermal cycles and 50 pressure cycles. The test objectives were not fully realized, however, due to a development change in the existing design of the end fitting. The data therefore became obsolete, as a result, the tests were discontinued.

4.0 CONCLUSIONS:

4.1 Test results indicate that under the following conditions:

a) adequate clearance between mating surface and seal, b) established torque settings for both sides of seal ring, and c) compatible materials, i.e., mating surface as hard as seal ring, the radial seal should withstand system temperature and pressure requirements.

4.2 Tests indicate that the seal assembly can withstand repeated mechanical and thermal cycles.

4.3 Brinelling occurs when soft materials are used as the seal seat.

4.4 Sealing characteristics are not impaired by seal deformation. This is evidenced by the use of a seal which incurred a growth of 0.002-inch on the OD of the 3.248-inch seal diameter (during its first test cycle). Its sealing capability was not affected in subsequent cycles.

4.5 Minor torque gap adjustments are required to preclude excessive brinelling between contact surfaces of different hardnesses.

5.0 RECOMMENDATIONS:

5.1 It is recommended that this radial seal design be referenced should an end fitting closure be required as part of the AMF On-Power Refueling System. Optimization of the design would require additional test effort to be performed.

5.2 It is further recommended that to maintain proper sealing conditions under repetitive mechanical cycling the mating surface be hardened to a minimum of Rockwell C40 to approximate the hardness of the ring.

6.0 TESTING:

6.1 Test Setup

The seal assembly (figure 6-1) was installed in the test rig (figure 6-2). The test rig was connected to the high-temperature water supply.

6.2 Test Procedure:

6.2.1 Hydrostatic Test at Room Temperature and Pressure Gradients:

The front nut (figure 6-1) was tightened to 250-inch-pounds torque. This series was performed 15 times. The test rig was pressurized from zero to 1500 psig and then depressurized to zero. In each cycle, the seal leaked steadily from zero to 300 psig during pressurization, and from 300 psig to zero during depressurization.

6.2.2 Hydrostatic Test at Operating Pressure and Operating Temperature:

The radial seal was subject to 15 thermal cycles at pressures averaging 1250 psig and temperatures of 550°F. The results of this test are tabulated below:

FIGURE 6-1. RADIAL SEAL ASSEMBLY

FIGURE 6-2. RADIAL SEAL TEST RIG

<u>Thermal Cycle</u>	<u>Leakage Reading/in parts per million</u>	<u>Average Pressure psig</u>	<u>Initial Temp/°F (Note 1)</u>	<u>Test Tem/°F</u>	<u>Time Interval Minutes</u>
1	30	1325	100	520	5
	100				5
2	140	1300	415	510	5
	180				5
3	200	1300	305	510	5
	250				5
4	320	1300	460	505	5
5	640	1300	290	395	15
	660				5
6	17	1100	340	505	5
7	17	1375	450	485	15
	57				15
	77				5
	82				5
8	77	1450	420	530	10
	62				15
9	107	1400	355	465	30
	93				15
	57				10
	47				5
	37				5
	27				5
	17				5
10	47	1200	360	585	5
	52				5
	97				5
11		1300	260	520	
12		1275	400	515	
13		1275	280	410	Note 2
14		1300	290	498	
		0	100		

NOTE 1. Low temperatures obtained by normal radiant and convective cooling when high-temperature water supply was shut down.

NOTE 2. No leakage recordable.

6.2.2.1 Post-Test Inspection:

The seal was removed from the test rig (figures 6-3, 6-4 and 6-5) and inspection revealed the following:

FIGURE 6-3. RADIAL SEAL IN TEST HUB

FIGURE 6-4. RADIAL SEAL PARTIALLY REMOVED FROM TEST HUB

FIGURE 6-5. RADIAL SEAL DISMANTLED

- a. The seal ring OD (3.248 inches) enlarged 0.002 inch as a result of a permanent set.
- b. Hub brinelling occurred resulting in an indent of 0.001 on the hub ID (figure 6-6).

6.2.3 Hydrostatic Tests at Room Temperature and Pressure Gradients:

Prior to testing, the following changes were made to preclude the brinelling which occurred previously:

- a. Sleeves and tension springs were removed from the back side of seal and replaced by the original retaining nut.
- b. No tightening torque was applied to the forward nut. It was assembled finger-tight.
- c. In the absence of a tightening torque value, the force required to move the seal in the hub was determined by placing weights on top of the seal. This caused the hub to slide in the sealing hub. A weight of 25 pounds was required.

6.2.3.1 The results of the test under these conditions is tabulated below:

<u>Press Cycle Number</u>	<u>Leakage</u>	<u>Pressure-psig</u>
1	none visible	0-1600-0
2	slight	0-600
	none visible	600-1600-450
	slight	450-0
3	none visible	0-1600-0
4	none visible	0-1600-0
5	none visible	0-1600-0
6	none visible	0-1600-0
7	none visible	0-1600-0
8	none visible	0-1600-0

FIGURE 6-5. SEAL HUB SHOWING BRINELLING

<u>Press Cycle Number</u>	<u>Leakage</u>	<u>Pressure-psig</u>
9	none visible	0-1600-0
10	none visible	0-1600-0
11	none visible	0-1600-0
12	none visible	0-1600-0
13	slight	0-1050
	none visible	1050-1600
	none visible	1600-300
	slight	300-150
	none visible	150-0
14	none visible	0-1600-0
15	none visible	0-1600-0
16	none visible	0-1600-0
17	none visible	0-1600-0
18	none visible	0-1600-0
19	slight	0-1350
	none visible	1350-1600
	none visible	1600-1400
	slight	1400-1350
	none visible	1350-400
	slight	400-0
20	slight	0-825
	none visible	825-1600
	none visible	1600-1050
	slight	1050-800
	none visible	800-450
	slight	450-0

6.2.3.2 In order to further reduce leakage, a torque was applied to the forward nut which increased the force necessary to move the seal. The weight was increased to 55 pounds. Test results under this condition are outlined below:

<u>Press Cycle Number</u>	<u>Leakage</u>	<u>Pressure-psig</u>
1	none visible	0-1600-0
2	none visible	0-1600-0
3	none visible	0-1600-0
4	none visible	0-1600-0
5	none visible	0-1600-0
6	none visible	0-1600-0
7	none visible	0-150
	slight	150-500
	none visible	500-1600
	none visible	1600-0
	slight	0
8	slight	0-300
	none visible	300-1600
	none visible	1600-50
9	slight	0-800
	none visible	800-1600
	none visible	1600-0
10	slight	0-1600
	slight	1600-1350
	none visible	1350-50
	slight	50-0

6.2.4 Hydrostatic Test at Operating Pressure and Temperature Gradients:

6.2.4.1 This test series was accomplished under conditions in paragraph 6.2.3 with a weight of 55 pounds applied to the seal. Test results were as follows:

<u>Thermal Cycle</u>	<u>Leakage</u>	<u>Average Pressure Reading-psig</u>	<u>Initial Temper- F</u>	<u>Test Temper-^oF</u>	<u>Time Interval-Minutes</u>
1	none visible	1200	100	430	30
	none visible				30
2	slight	0	100	100	5

6.2.4.2 In order to further reduce leakage and create better sealing characteristics, the following modifications were made prior to test continuation:

- a. The torque on the forward nut was released.
- b. A gap of 0.0015 inch was established between the backup ring and spacer.

6.2.4.3 The first series of tests accomplished under the above conditions was conducted under room temperature conditions and pressure gradients starting at 1500 psig and decreasing to 300 psig. The results of this test series follows: (All temperatures at room ambient).

COLD

<u>Pressure Cycle</u>	<u>Leakage</u>	<u>Pressure psig</u>	<u>Time Interval-Minutes</u>
1	none visible	1500	10
2	none visible	1350	5
3	none visible	1100	5
4	none visible	900	5
5	none visible	700	5
6	none visible	500	5
7	slight	450	5
8	slight	400	5
9	steady stream	350	5
10	steady stream	350	5
12	slight	400	5
13	steady stream	350	5
14	steady stream	300	5

6.2.4.4 The second series of tests was accomplished in two thermal cycles.

- a. The first thermal cycle was conducted at pressures ranging from 300 to 1350 psig and from room temperature. The results are outlined below:

HOT

<u>Leakage</u> <u>Run No.</u>	<u>Leakage</u>	<u>Pressure</u> <u>psig</u>	<u>Temp/°F</u>	<u>Time</u> <u>Interval-minutes</u>
1	none visible	950	425	20
2	none visible	1100	425	10
3	none visible	1350	425	10
4	none visible	1350	425	20
5	none visible	1050	435	10
6	none visible	950	460	10
7	none visible	750	470	10
8	none visible	500	465	10
9	slight	350	450	10
10	steady stream	300	400	10

b. The second thermal cycle was performed at pressure ranging from 1100 to 1500 psig and from room temperatures to 575°F and back down to room temperature. The leakage results at environmental conditions are outlined below:

<u>Leakage</u> <u>Run No.</u>	<u>Leakage</u>	<u>Pressure</u> <u>psig</u>	<u>Temp/°F</u>	<u>Time</u> <u>Interval-minutes</u>
1	none visible	1100	450	15
2	none visible	1200	475	15
3	none visible	1300	500	15
4	none visible	1400	525	15
5	none visible	1500	575	15
6 Note 1.	88 PPM	1525	575	10
7	60 PPM			
8	42 PPM	1525	575	10

Note 1. Leak rate in parts per millions as indicated on a hygrometer.

Note 2. Testing was concluded at this point as it was learned that the sealing hub was not compatible with the Douglas Point Plant reactor end fitting.

7.0 FINAL INSPECTION

Final inspection of the radial seal revealed the following:

- a. The seal ring had a permanent set of 0.002 inch.
- b. The hub brinelled to a depth of 0.001 inch.
- c. The remaining components were in good condition.

1.0 PURPOSE:

2.0 SCOPE:

2.1 Base-Line Tests

2.2 Reliability Tests

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- a. Determining leakage across the valve seat with the test medium at 1500 psi and 550°F on the upstream side of the valve seat and room conditions on the downstream side of the valve.
- b. Determining the forces required to open the valve disc with pressure differentials of 150, 100, and 50 psi across the valve seat.
- c. Determining leakage across the valve seat during thermal cycling of the upstream side of the valve from 150°F to 550°F and back to 150°F maintaining the test medium in the liquid state.

2.3 The valve tested (figure 7-1) was a 4-inch sliding disc valve manufactured by the Okadee Company, Chicago, Illinois. The valve is fabricated of 316 stainless steel, contains Stellite seats, and Grayloc hubs.

3.0 SUMMARY:

- 3.1 The average leakage past the valve disc at operating pressure and temperature was 1.12 cc per hour. Thirty-eight ten-minute leakage tests were conducted. The leakage values ranged from a high of 8.05 cc per hour to a low of 0.04 cc per hour. The minimum acceptable leakage rate for this valve was specified to be 300 cc per hour.

FIGURE 7-1. 4-INCH SLIDING DISC VALVE

3.2 The average torque required to open the valve disc at pressure differentials of 150, 100, and 50 psi was 8866, 6453, and 3456 inch-pounds, respectively. Ten tests were conducted at each pressure differential, with the high and low torque values recorded below:

<u>Pressure Differential</u>	<u>High Torque Reading/In.-Lb</u>	<u>Low Torque Reading/In.-Lb</u>
150 psi	9990	6480
100 psi	7290	5400
50 psi	4050	2700

3.3 Due to the inability of the stainless steel gasket to maintain a leak-tight valve joint, a composite stainless steel-asbestos was used. No visible leakage occurred thereafter.

3.4 The valve housing was scored on the downstream side.

4.0 CONCLUSIONS:

4.1 The valve may be used as part of an adapter valve for the AMF On-Power Refueling Machine if gross leakage past the valve seat at differential pressures lower than 400 psig can be accepted.

4.2 At differential pressures exceeding 400 psig, the valve seals effectively.

4.3 The thermal cycling of the valve did not cause leakage from the valve's sealing surfaces.

4.4 Stainless steel gaskets supplied with the valve were not effective at high temperatures.

4.5 Score marks on the valve housing were a result of the preload condition.

5.0 RECOMMENDATIONS:

5.1 It is recommended that this valve be considered for use on the alternate adapter valve. The pressure loaded sliding disc concept contained the fluid within acceptable leakage rates.

5.2 Use stainless steel-asbestos gaskets on future applications.

5.3 Eliminate the preload scoring condition by providing harder wear surfaces.

6.0 TEST:

6.1 Test Setup

The valve was mounted in a high-temperature water circulation system. The test medium was demineralized to a pH of 9.5 to 10.5 (see figures 7-2 and 7-3). The valve was tested under static and dynamic pressure conditions. Static pressure tests were conducted at room water temperature and dynamic pressure tests at high-temperature fluid conditions.

6.2 Test Procedure

6.2.1 Base-Line Testing

6.2.1.1 Preliminary Valve Leak Test

The top Grayloc Clamp and a 316 stainless steel seal ring were assembled on the valve. The valve disc was closed. The clamp nuts were tightened to 180-foot-pounds torque. The flange body nuts were tightened to 320 foot-pounds torque. The valve was submerged in water and pressurized with nitrogen. Leakage developed at the clamp and

FIGURE 7-2. 4-INCH SLIDING DISC VALVE TEST RIG

FIGURE 7-3. CLOSE-UP OF TEST RIG

past the valve when the pressure reached 200 psig. The clamp nuts were tightened to 200 foot-pounds torque and the valve was repressurized. Leakage continued past the valve disc until 400 psig pressure was reached. At this pressure, leakage stopped. The tests were continued until 1500 psig was reached.

6.2.1.2 Hydrostatic Test

The valve was tested at 2250 psig pressure. No leakage past the valve disc, the clamp, or the valve disc stem was evident at this pressure. Leakage, however, was evident at the valve body flange stainless steel gasket (figure 7-4). Continuous pumping was required to maintain 2250 psig pressure within the valve. There was no evidence of valve body damage at this proof pressure (figures 7-5 and 7-6).

6.2.1.3 Pressure Testing at Room Temperature and Pressure Gradients

The stainless steel gaskets were removed, coated with Permatex, and replaced. The valve was tested, with the valve disc closed, at pressures of 200, 500, 750, and 1000 psig. The valve was filled with water maintained at room temperature. Leakage test results through the valve body gasket joint are as follows:

FIGURE 7-4. LEAKAGE THROUGH GASKET

FIGURE 7-5. STELLITE SEATS ON DISC AND BODY BEFORE TESTING

FIGURE 7-6. VALVE BODY AFTER HYDROSTATIC TEST

- a. At 300 psig - leakage past valve body flange metal gasket at the rate one drop every two seconds.
- b. At 500 psig - leakage past valve body flange metal gasket at rate of one drop per second.
- c. At 750 psig - leakage past valve body flange metal gasket at rate of one drop per second.
- d. At 1000 psig - a continuous stream of water leaked past valve body flange metal gasket.

Leakage across the valve disc occurred as a steady stream of water until 350 psig pressure was reached. Beyond this pressure, no leakage was evident.

Further testing provided basically the same results.

The valve disc did not seal properly at pressures less than 350 psig. The valve body flange metal gaskets leaked.

The first series of tests, conducted at five-minute intervals, resulted as follows:

<u>Pressure/psig</u>	<u>Leakage at Gasket cc/hr</u>	<u>Leakage Past Valve Disc</u>
300	None	Steady Stream
500	None	None
700	None	None
1000	None	None
1200	108	None
1250	132	None
1300	204	None
1350	228	None

and pressure control and other data to determine leakage

<u>Pressure/psig</u>	<u>Leakage at Gasket cc/hr</u>	<u>Leakage Past Valve Disc</u>
1400	288	None
1425	288	None
1475	360	None
1500	912	None

After this test series, the valve body flange metal gaskets were removed and inspected. Two of the three gaskets contained surface high spots ranging from 0.0002 to 0.0006 inch.

As a result of this damage and the leakage history of these gaskets, they were replaced with stainless steel-asbestos gaskets. The valve with the new gaskets was tested at room temperature and pressure gradients, with the valve disc closed.

This series of tests, also conducted in five-minute intervals, resulted as follows:

<u>Pressure/psig</u>	<u>Leakage at Gasket cc/hr</u>	<u>Leakage Past Valve Disc</u>
300	None	Steady Stream
500	None	None
700	None	None
1000	None	None
1200	None	None
1250	None	None
1300	None	None
1350	None	None
1400	None	None
1450	None	None
1500	None	None
*1600	None	None

*This test lasted 20 minutes.

6.2.1.4 Pressure Testing at Temperature Gradients and Static Pressure

Thermocouples were attached to the valve to record the inlet and outlet temperatures. The test medium was circulated through the valve and maintained at 1500 psig. The temperature was increased from room temperature to 550°F at a rate not exceeding 20°F per minute. The valve disc was closed. The valve body flange nuts were tightened to 320 foot-pounds torque; the clamp nuts to 200 foot-pounds torque. The lower clamp was not installed to facilitate the observation of valve disc leakage. The valve was tested at 1475 psig pressure to 560°F. Leakage past the valve disc was observed only at pressures less than 350 psig. Beyond this pressure, leakage stopped.

6.2.2 Reliability Testing

Reliability testing consisted of three separate test series.

The series were as follows:

- a. Thermal Cyclic Tests - This series required the valve be pressurized to 1500 psig and the temperature of the test medium be increased from room temperature to 550°F in increments not exceeding 20°F per minute. Leakage past the valve disc was observed and recorded throughout this series.

b. Valve Actuation Torque Requirements for Differential Pressures

This test series was required to determine and record the force required to open the valve disc under pressure differentials (between the upstream and downstream sides of the valve) of 150, 100, and 50 psig.

- c. Leakage Test - This test series was required to determine and record the leakage past the valve disc with pressure at 1500 psig and test medium temperature at 550°F.

6.2.2.1 Thermal Cyclic Tests

- a. Twenty-five thermal cyclic tests were conducted. Thirteen tests were conducted with the valve disc closed with the test medium circulating through the upstream side of the valve. No visible leakage was observed past the valve disc.

The results of one test of this series is shown below.

The subject test was completed in 62 minutes, 40 seconds. Readings were taken every 80 seconds.

<u>Pressure</u>	<u>Inlet Water Temperature</u>	<u>Outlet Water Temperature</u>	<u>Leakage Observed</u>
1500	250	205	None
1500	275	230	None
1450	300	275	None
1400	325	300	None
1375	350	320	None
1350	370	340	None
1300	390	355	None
1250	405	375	None
1200	415	390	None
1150	425	395	None

Low Pressure and Low Temperature Tests			
<u>Pressure</u>	<u>Inlet Water Temperature</u>	<u>Outlet Water Temperature</u>	<u>Leakage Observed</u>
1100	435	400	None
1050	440	405	None
1000	445	410	None
950	450	415	None
900	455	420	None
900	460	420	None
850	470	425	None
850	480	440	None
850	485	445	None
800	490	450	None
800	497	455	None
750	505	465	None
700	515	475	None
700	520	480	None
700	525	480	None
1200	535	490	None
1250	550	500	None
1250	560	505	None
1300	550	490	None
1300	545	450	None
1300	530	425	None
1300	505	400	None
1325	475	365	None
1325	425	325	None
1325	360	285	None
1350	325	255	None
1400	300	245	None
1400	290	240	None
1400	265	225	None
1400	245	215	None
1450	225	200	None
1450	205	175	None
1500	190	160	None
1500	180	150	None
1500	165	130	None
1500	150	125	None
1500	145	120	None

NOTE: Temperature was controlled; consequently, pressure was allowed to vary between 700 and 1500 psig.

- b. The remaining tests of this series were conducted with the valve disc open and the test medium was

circulated through the valve. No leakage was observed past the valve body gaskets and the disc actuating handle packing gland. The results of one test of this series are shown below:

The subject test was completed in 46 minutes, 40 seconds. Readings were taken every 80 seconds.

<u>Pressure</u>	<u>Inlet Water Temperature</u>	<u>Outlet Water Temperature</u>	<u>Leakage Observed</u>
1200	325	255	None
1250	340	275	None
1250	360	290	None
1300	375	310	None
1300	385	310	None
1300	295	315	None
1300	405	315	None
1300	415	320	None
1300	425	330	None
1300	435	340	None
1300	445	345	None
1300	455	350	None
1350	470	365	None
1400	480	365	None
1450	485	370	None
1450	490	375	None
1450	495	375	None
1450	500	375	None
1450	500	375	None
1475	500	375	None
1450	500	375	None
1400	475	390	None
1400	455	360	None
1350	420	325	None
1325	375	285	None
1300	325	255	None
1250	290	220	None
1200	265	200	None
1150	245	175	None
1100	225	155	None
1050	210	140	None
975	200	125	None
900	190	115	None
850	180	105	None
800	175	100	None

NOTE: Temperature was controlled; consequently, pressure was allowed to vary between 700 and 1500 psig.

- c. During the thermal cyclic test series, the valve body flange stainless steel gaskets failed. These gaskets were replaced with 0.125 inch Flexitallic stainless steel-asbestos gaskets.

NOTE

The substitution of stainless steel-asbestos gaskets for stainless steel required the removal of 0.019 inch stock from inner seal shoulder.

6.2.2.2 Valve Actuation Torques for Differential Pressures

Prior to this test series, the valve was reassembled using stainless steel-asbestos gaskets. The valve body flange nut torque was reduced to 170 foot-pounds as a result of the change in gasket material. This test series was conducted to determine and record the force needed to open the valve disc at 550°F temperature and pressure differentials of 150, 100 and 50 psi between the upstream and downstream sides of the valve. A LUG-ALL cable device and a 400-pound Chatillon scale were used to determine this force. The cable was secured to the scale and the scale was connected to the end of the valve disc actuating handle. The other end of the cable was secured to a fixed point. As the cable was tightened, the force applied to the valve handle was recorded on the scale (see figure 7-7). The data obtained from this test is shown below:

FIGURE 7-7. TEST RIG SHOWING SCALE AND CABLE

150 PSI DIFFERENTIAL

Pressure		Valve Temp. °F	Chatillon Scale Reading/Lb	Scale Reading Converted to in-lb Torque
Above Disc	Below Disc			
1000	850	490	330	8910
1000	850	490	370	9990
1000	850	495	330	8910
950	800	495	280	7560
875	725	490	370	9990
800	550	485	330	8910
650	500	480	330	8910
600	450	480	240	6480
575	425	475	370	9990
450	300	475	370	9990

100 PSI DIFFERENTIAL

975	875	480	260	7020
975	875	490	240	6480
975	875	490	250	6750
1000	900	490	230	6210
1000	900	485	200	5400
1000	900	480	200	5400
1000	900	480	250	6750
1000	900	475	250	6750
1000	900	475	270	7290
1000	900	480	240	6480

50 PSI DIFFERENTIAL

1025	975	475	100	2700
1025	975	480	140	3780
1025	975	480	130	3510
1050	1000	480	110	2970
1050	1000	480	120	3240
1050	1000	495	140	3780
1075	1025	480	150	4050
1075	1025	480	130	3510
1075	1025	485	120	3240
1100	1050	485	140	3780

6.2.2.3 Leakage Tests

This series was required to determine the leakage past the valve disc. Leakage was determined using a nitrogen purge (at 6 psi) through the downstream portion of the

valve, through an outlet into a vented desiccant bottle. The bottle was weighed before and after each test to determine the amount of moisture (past the valve disc) absorbed by the desiccant. Test results are shown below. All tests were performed at ten-minute intervals.

<u>Pressure Above Valve Disc/psig</u>	<u>Water Temp Above Valve Disc/°F</u>	<u>Leakage cc/hr</u>
1450	470	8.05
1450	485	1.64
1450	475	0.45
1450	480	0.04
1450	475	0.17
1450	475	0.06
1450	475	0.05
1525	440	3.31
1525	470	3.50
1500	495	1.99
1500	505	0.79
1500	505	0.80
1475	510	0.79
1475	505	0.84
1475	505	0.54
1490	500	0.31
1500	495	0.19
1490	490	0.21
1480	485	0.11
1475	480	0.19
1475	480	0.15
1470	475	0.17
1525	480	6.60
1525	510	0.44
1525	520	0.27
1525	525	0.34
1525	525	0.35
1525	525	0.32
1500	520	0.32
1450	510	0.35
1425	500	0.32
1400	490	0.35
1350	480	4.70
1325	480	1.98
1300	480	0.41
1300	480	0.37
1300	480	0.48
1300	480	0.40

7.0 FINAL INSPECTION:

The valve was disassembled and inspected after all tests were completed. No wear was evident on the valve seats and disc (see figures 7-6, 7-7 and 7-8). The downstream side of the valve casing revealed score marks (caused by the backside of the spring loaded disc) on the housing (see figures 7-7, 7-9, and 7-10). Scoring of the valve housing, however, did not impair the sealing properties of the valve as they were remote to the valve seat.

FIGURE 7-8. STELLITE SEAT ON BODY SIDE OF VALVE

FIGURE 7-9 STELLITE SEAT AT SLIDING DISC AFTER MECHANICAL AND THERMAL CYCLING

FIGURE 7-10. CLOSE-UP OF VALVE BODY HOUSING SCORE MARKS

SECTION VIII

MATERIAL WEAR TEST - TEST REPORT

1.0 PURPOSE:

The purpose of this report is to present, evaluate, and document the results of a test program involving material wear under conditions similar to those encountered in AMF On-Power Refueling Machine operation.

2.0 SCOPE:

The scope of this test was to cycle various material combinations, and their relative surface preparation, under simulated and accelerated environmental conditions. Material combinations tested were those intended for use on the machine on which no applicable reference wear data is available. The combinations consisting of a solid sleeve and a split sleeve were:

<u>Solid Sleeve</u>	<u>Split Sleeve</u>
a. 316L SS (hard chrome ID)	17-4 PH SS (nitrided)
b. 316L SS (hard chrome ID)	17-4 PH SS (hardened)
c. 316L SS (honed ID)	304L SS (hard chrome)
d. 316L SS (honed ID)	17-4 PH SS (nitrided)

The temperature range of the tests was to be 550°F. The test medium was to be demineralized water with a pH of 9.5 to 10.5. Bearing pressures of 8 to 16 psi were specified and were to be duplicated in the test apparatus.

3.0 SUMMARY:

3.1 The combination of hard-chrome plated 316L stainless steel and nitrided 17-4 PH stainless steel was tested for 30,000 cycles. The 17-4 PH stainless steel split sleeves attained a high polish in localized areas along the path of travel. Scuff marks appeared on the solid sleeve chromed surface in small isolated areas, starting at the sleeve ends and increasing in size as testing progressed.

3.2 The combination of hard-chrome plated 316L stainless steel and hardened 17-4 PH stainless steel was tested for 30,000 cycles. The 17-4 PH split sleeve acquired a dull-buffed finish in localized areas and scratch marks along the path of travel. Scuff marks appeared on the solid sleeve chromed surface in small areas, starting at the ends and increasing in size as testing progressed.

3.3 The combination of honed 316L stainless steel and chromed plated 304L stainless steel was tested for six cycles and again at 2000 cycles.

3.3.1 After six cycles of the initial test the specimens seized, causing damage to the test apparatus.

3.3.2 After repairing the test apparatus, a second identical combination was tested for 2000 cycles. Upon completion of the test and removal of specimens, galling was noted on both the split and solid sleeves, with the heavier concentration on the split sleeves.

3.4 The combination of 316L stainless steel with the honed ID and nitrided 17-4 PH stainless steel was tested for 3000 and 30,000 cycles. Very little wear was experienced in the initial states of testing. Slight wear and burnishing occurred on the split sleeve and minor scuffing occurred on the solid sleeve ID after 30,000 cycles.

4.0 CONCLUSION:

4.1 All test specimens either galled, scuffed, or seized during test.

4.2 Combinations of 316L stainless steel hard-chromed or honed on the inside diameter in contact with hardened 17-4 PH stainless steel withstood 30,000 cyclic tests, and would be feasible for system application.

4.3 The combination of honed 316L stainless steel in contact with the hard-chrome plated 304L is not feasible because of severe galling.

5.0 RECOMMENDATIONS:

5.1 The combinations of 316L stainless steel hard-chrome or honed in contact with 17-4 PH stainless steel appears satisfactory for the purpose intended.

5.2 It is recommended that the thickness of chrome plating applied to the 304L stainless steel split sleeve be increased above that (0.0004 to 0.0006 inch) used in testing. Heavier plating and more stringent control of plating quality may further reduce the wear condition obtained from samples tested.

6.0 TESTING:

6.1 Test Setup

The wear-test apparatus (figure 8-1) consists of a pressure vessel designed to maintain the test medium of 1500 psig pressure and 600°F. The test medium is heated externally in the high-temperature water supply. A thermocouple, associated with a thermowell, records the internal test medium temperatures. The test specimens, consisting of solid and split sleeves, are mounted within the test vessel. The solid sleeves are stationary while the split sleeves, mounted on a reciprocating shaft, are dynamic. A 1/2-HP motor, through a gear reducer and an eccentrically mounted cam follower, imparts a total travel of 1.0 inch to the split sleeves, at the rate of 58 cpm. The split sleeves are secured to the shaft and within the solid sleeve by means of a dowel, and are held apart by two springs (figure 8-2).

6.2 Machine Finishes and Surface Treatment

6.2.1 The following machine finishes and surface treatments were applied to the test specimens, prior to testing:

- a. The solid sleeves were machined and honed to a 16 to 24 micro-inch finish.
- b. The split sleeves were machined and polished to a 11 to 18 micro-inch finish.
- c. The 17-4 PH stainless steel nitrided sample was soaked for one hour in a salt bath maintained at 950°F. Upon removal it was quenched in oil.

FIGURE 8-1. WEAR TEST APPARATUS

FIGURE 8-2. WEAR TEST SCHEMATIC DIAGRAM

- d. The 17-4 PH stainless steel sample, hardened to Rockwell C38 to 40, was heat soaked for one hour at 900°F. Upon removal it was air cooled.
- e. The 304L and 316L stainless steel samples were initially flash-copper plated and then hard-chrome plated to a thickness of 0.0004 to 0.0006 inch.

6.2.2 Specimen Material Composition

The following is the certified material composition of the specimens tested:

	<u>Element</u>	<u>PCT</u>
a. 304L Stainless Steel	Carbon	0.023
	Manganese	1.620
	Phosphorus	0.012
	Sulphur	0.013
	Silicon	0.620
	Chromium	18.740
	Nickel	10.250
	Molybdenum	0.120
	Copper	0.070
	Cobalt	0.13
	Iron	Balance
b. 17-4 PH Stainless Steel	Carbon	0.038
	Manganese	0.240
	Phosphorus	0.020
	Sulphur	0.015
	Silicon	0.570
	Chromium	16.180
	Nickel	4.140
	Columbium	0.220
	Copper	3.380
	Tantalum	0.020
	Iron	Balance
c. 316L Stainless Steel	Carbon	0.022
	Manganese	1.700
	Phosphorus	0.010
	Sulphur	0.021
	Silicon	0.500
	Chromium	16.480
	Nickel	11.670
	Molybdenum	2.520
	Copper	0.100
	Cobalt	0.100
	Iron	Balance

6.3 Test Procedure

The test apparatus was purged to preclude test medium contamination during testing. The test specimens were assembled into the test rig. The medium temperature was maintained and monitored. The motor was energized. Tests proceeded as follows:

- a. The 304L hard-chrome plated stainless steel split sleeve was tested against a 316L solid sleeve with a honed ID at 550°F and calculated bearing pressure of 18.1 psi. The specimens seized after six cycles, damaging the test rig. The test rig was repaired and second test using a second set of materials commenced. After 2000 cycles at 520°F, the second specimen showed severe galling, with a calculated bearing pressure of 3.5 psi.
- b. The 17-4 PH nitrided stainless steel split sleeve was tested against 316L solid sleeve with a hard-chrome plated ID at 550°F at a calculated bearing pressure of 14.6 psi. The specimens were cycled 30,000 times and showed a high polish in local areas on the split sleeves, and scuff marks in small isolated areas on the solid sleeves.
- c. The nitrided 17-4 PH stainless steel split sleeve was tested against a honed 316L stainless steel solid sleeve. Temperatures ranged from 375°F to 585°F. Two series were conducted; one for 3000 cycles with a bearing pressure of 17.9 psi; the other for 30,000 cycles with a bearing pressure of 7.7 psi. Wear was minor after all tests.

7.0 TEST DATA:

See figure 8-3 for test data tabulation.

NOTES For Figure 8-3

1. Due to material loss from solid sleeve outer periphery during disassembly from sleeve holder, the loss in sample weight was not representative of ID wear.
2. One split sleeve lost 0.016 grams weight; the other gained 0.029 grams.
3. Normal PH during tests was 9.5 to 10.5. This test was conducted at a pH of 6.9.
4. Bearing pressure was calculated from contact surface areas; the load from the compressed springs in the test rig.

MATERIAL WEAR TEST NO.	WEAR MATERIALS		CYCLES	TOTAL TRAVEL/INCHES	WEIGHT LOSS/GRAMS		TEMPERATURE °F	ACTUAL BEARING PRESSURE/psi (Note 4)	REMARKS
	Solid Sleeve	Split Sleeve			Solid Sleeve	Split Sleeve			
1	316L SS (HD-CHR ID)	17-4 PH (Nitrided)	30000	30000	Note 1	0.0168	510 to 550	14.6	High polish in localized areas on split sleeves; scuff marks in isolated areas on solid sleeves. (See figures 8-4, 8-5, 8-6).
2	316L SS (HD-CHR ID)	17-4 PH (HDN)	30000	30000	Note 1	0.009	490 to 595	4.17	Dull buffed finish in localized areas. Minor scratches on split sleeves. Scuff marks on solid sleeves. (See figures 8-7, 8-8).
3	316L SS (Honed ID)	304L SS (HD-CHR)	6	6	Note 1	Note 2	550	18.1	Severe galling and scoring. Specimens seized. Test rig damaged. (See figures 8-9, 8-10).
			2000	2000	Note 1	0.1299	520 to 550	3.5	Severe galling and scoring. (See figures 8-11, 8-12, 8-13).
4	316L SS (Honed ID)	17-4 PH (Nitrided)	1000	1000	0.019	0.008	530 to 540	17.9	Considerable wear.
		(TOTAL)	2000	2000	0.003	0.003	450 to 540	17.9	Wear reduced after initial misalignment. Scuffing at ends of solid sleeve. High polish on split sleeve. Scuffing at ends of solid sleeve caused by initial misalignment, (See figures 8-14, 8-15).
			3000	3000	0.022	0.011	450 to 540	17.9	
5	316L SS (Honed ID)	17-4PH (Nitrided)	30000	30000	Note 1	0.111	375 to 585	7.73	Slight wear and burnishing on split sleeves. Scuffing on solid sleeve ID. (See figures 8-16, 8-17).
6	316L SS (Honed ID)	17-4 PH (HDN)	2000	2000	0.0617	0.0338	550 to 585	4.88	Scoring and galling on solid and split sleeves Note 3. (See figure 8-18).

Figure 8 - 3.

TABLE OF MATERIAL-WEAR TEST RESULTS

FIGURE 8-4. 17-4PH SS NITRIDED SPLIT SLEEVE - LOCALIZED HIGH POLISH

FIGURE 8-5. 316L SS HARD CHROMED SOLID SLEEVE -- ISOLATED SCUFF MARKS

FIGURE 8-6. 316L SS HAPD CHROMED SOLID SLEEVE - ISOLATED SCUFF MARKS

FIGURE 8-7. 316L SS HARD CHROMED SOLID SLEEVE - SCUFF MARKED,
17-4PH SS HARDENED SPLIT SLEEVE - BUFFING, LOCALIZED MINOR SCRATCHES

FIGURE 8-8. 316L SS HARD CHROMED SOLID SLEEVE - SCUFF MARKED, 17-4PH SS HARDENED
SPLIT SLEEVE - BUFFED FINISH, MINOR LOCALIZED SCRATCHES

FIGURE 8-9. 316L SS HONED SOLID SLEEVE - GALLED AND SCORED

FIGURE 8-10. 316L SS HONED SOLID SLEEVE, 304L SS HARD CHROMED SPLIT SLEEVE-GALLED AND SCORED

FIGURE 8-11. 304L SS HARD CHROMED SPLIT SLEEVE - GALLED AND SCORED

FIGURE 8-12. 304L SS HARD CHROMED SPLIT SLEEVE - GALLED AND SCORED

FIGURE 8-13. 316L SS HONED SOLID SLEEVE, 304L SS HARD CHROMED SPLIT SLEEVE - GALLED AND SCORED

FIGURE 8-14. 316L SS HONED SOLID SLEEVE-SCUFFING

FIGURE 8-15. 17-4PH SS NITRIDED SPLIT SLEEVE-SLIGHT LOCALIZED WEAR AND BURNISHING

FIGURE 8-16 316L SS HONED SOLID SLEEVE, 17-4PH SS NITRIDED SPLIT SLEEVE-
SLIGHT WEAR AND BURNISHING, SCUFFING ON SOLID SLEEVE

FIGURE 8-17. 316L SS HONED SOLID SLEEVE-SCUFFING

FIGURE 8-18. 316L SS HONED SOLID SLEEVE, 17-4PH SS HARDENED SPLIT SLEEVE - GALLED AND SCORED