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LMFBR LARGE VALVE DEVELOPMENT

STATIC SODIUM PROOF-OF-PRINCIPLE (POP)
TEST DESIGN AND ANALYSES INTERIM SUMMARY REPORT
FOR THE 12.0-INCH ROTATING OFFSET BALL VALVE INTERNALS

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Contract No. AT-(04-3)-1035 ERDA SAN

Report No. 1839-01(03)ER

April 1975

AC03-76SF01035

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1.0 SUMMARY

This report updates progress made by AMCO to date in the development of the 12-inch rotating offset ball valve (Figure 1), for use in sodium isolation service for the LMFBR Demonstration Plant Intermediate Heat Transport System (IHTS). The work is being performed under Contract No. AT-(04-03)-1035 ERDA SAN. The design incorporates an offset rotating ball in conjunction with a flexible metal lip seal to provide shut-off in liquid sodium at elevated temperatures.

The offset ball design, recommended by AMCO, was selected through a trade study (Reference 1) of the existing state-of-the-art designs utilized in liquid sodium systems. AMCO's conclusion was that the offset ball incorporated the maximum in sodium design features attainable by any single configuration. Some of these features include 1) low flow resistance coefficient at full-open position--important for minimum required pumping power, 2) tight shut-off necessary for isolation service, 3) fast closing time capability for emergency isolation, and 4) rotary motion preferred for use in conjunction with a freeze seal or other shaft sealing methods.

AMCO has also included a sub-housing design to permit in-situ replacement of valve internals as a unit, an Oldham coupling on the shaft system to allow for axial and angular misalignment of the actuator and valve shafts, and a unique dual flexible metal lip seal. The lip seal serves a dual function by providing the dynamic seal with the ball and the static seal with the valve body.

The forthcoming sodium proof-of-principle (POP) test will be conducted at the Liquid Metal Engineering Center (LMEC) in Chatsworth, California. This test will evaluate the valve internals which are shown in Figure 2.

A major innovation that has not been previously utilized in sodium valves is the use of a "Ferrometic"* shaft seal. The design is being investigated for candidacy as a secondary shaft seal for sodium valve design.

The sodium POP test will prove the integrity of the Ferrometic seal to operate in a sodium/argon vapor at a pressure of 10 psig and temperature of 225°F (maximum).

The Ferrometic seal is not intended to replace a primary shaft seal in a sodium valve such as the sodium freeze seal, but is intended to serve as a secondary shaft seal to replace the conventional packing used in existing sodium valves.

*Ferrofluidics Corporation, 144 Middlesex Turnpike, Burlington, Mass. 01803

Figure 2

ITEM	DESCRIPTION	QTY	UNIT
1	COIL	1	EA
2	ACTUATOR	1	EA
3	1.5 IN. DIA. BALL	1	EA
4	1.5 IN. DIA. BALL	1	EA
5	1.5 IN. DIA. BALL	1	EA
6	1.5 IN. DIA. BALL	1	EA
7	1.5 IN. DIA. BALL	1	EA
8	1.5 IN. DIA. BALL	1	EA
9	1.5 IN. DIA. BALL	1	EA
10	1.5 IN. DIA. BALL	1	EA
11	1.5 IN. DIA. BALL	1	EA
12	1.5 IN. DIA. BALL	1	EA
13	1.5 IN. DIA. BALL	1	EA
14	1.5 IN. DIA. BALL	1	EA
15	1.5 IN. DIA. BALL	1	EA
16	1.5 IN. DIA. BALL	1	EA
17	1.5 IN. DIA. BALL	1	EA
18	1.5 IN. DIA. BALL	1	EA
19	1.5 IN. DIA. BALL	1	EA
20	1.5 IN. DIA. BALL	1	EA
21	1.5 IN. DIA. BALL	1	EA
22	1.5 IN. DIA. BALL	1	EA
23	1.5 IN. DIA. BALL	1	EA
24	1.5 IN. DIA. BALL	1	EA
25	1.5 IN. DIA. BALL	1	EA
26	1.5 IN. DIA. BALL	1	EA
27	1.5 IN. DIA. BALL	1	EA
28	1.5 IN. DIA. BALL	1	EA
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47	1.5 IN. DIA. BALL	1	EA
48	1.5 IN. DIA. BALL	1	EA
49	1.5 IN. DIA. BALL	1	EA
50	1.5 IN. DIA. BALL	1	EA
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88	1.5 IN. DIA. BALL	1	EA
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93	1.5 IN. DIA. BALL	1	EA
94	1.5 IN. DIA. BALL	1	EA
95	1.5 IN. DIA. BALL	1	EA
96	1.5 IN. DIA. BALL	1	EA
97	1.5 IN. DIA. BALL	1	EA
98	1.5 IN. DIA. BALL	1	EA
99	1.5 IN. DIA. BALL	1	EA
100	1.5 IN. DIA. BALL	1	EA

VALVE INTERNALS
15" ORBIT BALL
16-J-4042

The seal is being considered because of its

- o In-situ filling capability. The seal may be refurbished without removal and while the valve is flowing sodium. (Filling ports are provided in the seal housing).
- o Zero leakage because of its hermetic seal characteristics.
- o Negligible torque--virtually friction free.

The seal is ideally suited for the sodium POP test because of its inherent low friction performance. For example, the torque measurements, that will be taken during the sodium POP test, to seat and unseat the ball under various temperatures and pressures will be absolute which will permit accurate calculations of valve friction coefficients. These friction coefficients will be used to size other valve operators when scaling to larger or smaller rotating offset ball valves, such as the 24.0-inch valve now in the design phase at AMCO.

The Ferrometic seal operates by focusing a Ferrofluid in the annulus between a series of magnetic pole blocks and the rotating shaft to form a non-rubbing circular seal.

The Ferrofluid is a polyphenyl ether base that was selected for its low vapor pressure (1 Torr at 500°F) and good radiation resistance (in excess of 10^8 rads).

The sodium POP test program is described in Reference 2.

Testing of the unit will include an extended soak time in liquid sodium at 936°F to investigate self-welding problems, cyclic capability, seat leakage (measured at various stages of the test), running torques, and seat-breakaway torques. At the conclusion of the test, the unit will be disassembled and inspected. This test will provide an effective program to determine the performance and reliability of the unit.

The work performed to date is:

- o Design and analysis of the test article for subsequent water testing (Reference 3). The water test valve assembly is shown in Figure 3.
- o Design and analysis of selected valve internals which may be used or modified for follow-on sodium testing. (Figure 2).

GENERAL NOTES

1. TEST MEDIA: WATER
2. PRESSURE:
 - a. OPERATING PRESSURE: 0.132 PSIA
 - b. PROOF PRESSURE: 384 PSIA
 - c. VALVE OPEN: 384 PSIA
 - d. VALVE CLOSED: 384 PSIA
3. DESIGN FLOW RATE: 8000 GPM
4. PRESSURE DROP: 2.5 PSI AT DESIGN FLOW CALCULATED
5. OPERATING TEMPERATURE: AMBIENT
6. LEAKAGE:
 - a. LEAKAGE: 1000 PPM
 - b. LEAKAGE: 1000 PPM
 - c. LEAKAGE: 1000 PPM
7. TEST WATER: 1000 PPM
8. TEST OBJECTIVE:
 - a. MEASURE LEAKAGE VERSUS AP
 - b. MEASURE FLOW VERSUS AP FOR VARIOUS VALVE POSITIONS
 - c. MEASURE FLOW THROUGH VARIOUS VALVE POSITIONS
 - d. MEASURE FLOW VERSUS FLOW AT VARIOUS VALVE POSITIONS
 - e. MEASURE FLOW VERSUS WATER FLOW AT FULL OPEN POSITION

10. CLEANLINESS PER ANSI SP-5014, LEVEL 4

C

B

C

COVERED 16 C 4037
O-RING 2.560
BOLT 1/2 16 UNF 3A + 5/16 16 (16 DEGD) CARBON STEEL
WASHER 1/2 16 UNF 3A + 5/16 16 (16 DEGD) CARBON STEEL
NUT 1/2 16 UNF 3A + 5/16 16 (16 DEGD) CARBON STEEL

ITEM	QTY	DESCRIPTION	UNIT
1	1	16 C 4037	COVERED
2	1	16 C 4037	O-RING
3	1	16 C 4037	BOLT
4	1	16 C 4037	WASHER
5	1	16 C 4037	NUT
6	1	16 C 4037	COVERED
7	1	16 C 4037	O-RING
8	1	16 C 4037	BOLT
9	1	16 C 4037	WASHER
10	1	16 C 4037	NUT
11	1	16 C 4037	COVERED
12	1	16 C 4037	O-RING
13	1	16 C 4037	BOLT
14	1	16 C 4037	WASHER
15	1	16 C 4037	NUT
16	1	16 C 4037	COVERED
17	1	16 C 4037	O-RING
18	1	16 C 4037	BOLT
19	1	16 C 4037	WASHER
20	1	16 C 4037	NUT
21	1	16 C 4037	COVERED
22	1	16 C 4037	O-RING
23	1	16 C 4037	BOLT
24	1	16 C 4037	WASHER
25	1	16 C 4037	NUT
26	1	16 C 4037	COVERED
27	1	16 C 4037	O-RING
28	1	16 C 4037	BOLT
29	1	16 C 4037	WASHER
30	1	16 C 4037	NUT
31	1	16 C 4037	COVERED
32	1	16 C 4037	O-RING
33	1	16 C 4037	BOLT
34	1	16 C 4037	WASHER
35	1	16 C 4037	NUT
36	1	16 C 4037	COVERED
37	1	16 C 4037	O-RING
38	1	16 C 4037	BOLT
39	1	16 C 4037	WASHER
40	1	16 C 4037	NUT
41	1	16 C 4037	COVERED
42	1	16 C 4037	O-RING
43	1	16 C 4037	BOLT
44	1	16 C 4037	WASHER
45	1	16 C 4037	NUT
46	1	16 C 4037	COVERED
47	1	16 C 4037	O-RING
48	1	16 C 4037	BOLT
49	1	16 C 4037	WASHER
50	1	16 C 4037	NUT
51	1	16 C 4037	COVERED
52	1	16 C 4037	O-RING
53	1	16 C 4037	BOLT
54	1	16 C 4037	WASHER
55	1	16 C 4037	NUT
56	1	16 C 4037	COVERED
57	1	16 C 4037	O-RING
58	1	16 C 4037	BOLT
59	1	16 C 4037	WASHER
60	1	16 C 4037	NUT
61	1	16 C 4037	COVERED
62	1	16 C 4037	O-RING
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67	1	16 C 4037	O-RING
68	1	16 C 4037	BOLT
69	1	16 C 4037	WASHER
70	1	16 C 4037	NUT
71	1	16 C 4037	COVERED
72	1	16 C 4037	O-RING
73	1	16 C 4037	BOLT
74	1	16 C 4037	WASHER
75	1	16 C 4037	NUT
76	1	16 C 4037	COVERED
77	1	16 C 4037	O-RING
78	1	16 C 4037	BOLT
79	1	16 C 4037	WASHER
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82	1	16 C 4037	O-RING
83	1	16 C 4037	BOLT
84	1	16 C 4037	WASHER
85	1	16 C 4037	NUT
86	1	16 C 4037	COVERED
87	1	16 C 4037	O-RING
88	1	16 C 4037	BOLT
89	1	16 C 4037	WASHER
90	1	16 C 4037	NUT
91	1	16 C 4037	COVERED
92	1	16 C 4037	O-RING
93	1	16 C 4037	BOLT
94	1	16 C 4037	WASHER
95	1	16 C 4037	NUT
96	1	16 C 4037	COVERED
97	1	16 C 4037	O-RING
98	1	16 C 4037	BOLT
99	1	16 C 4037	WASHER
100	1	16 C 4037	NUT

FIG 3

2.0 INTRODUCTION

The static sodium POP testing of the AMCO (12 x 8 x 12) rotating offset ball valve internals is a continuation of the development program associated with the LMFBR large valve development contract with RRD.

The static sodium POP test article (Figure 2) utilizes valve internals that were modified from the water POP test valve internals shown in AMCO Drawing 16-E-4020 and Figure 3.

The purpose of the sodium POP test is to evaluate the valve internals in sodium for seal leakage, cyclic life, the effect of time exposure in sodium, torque to crack the valve open against a maximum up-stream pressure head, and the Ferrometic seal.

Based on the current planning, LMEC will test the valve internals in Building 032, Vessel T-5 (032-R2-TS) which is scheduled to be available in February 1976 (see Figure 4 for the overall schedule for the sodium POP test).

The test article has been designed to fit into the precision mechanisms in sodium (PMIS)-II T-5 test vessel in accord with LMEC Drawing 7693-PT1400204. Figures 5, 6, and 7 are photographs of the T-5 vessel at LMEC showing the lowering of an unrelated experiment into the vessel. Figures 6 and 7 show water cooling coils attached to the cover to prevent over-heating of the cover seals.

Under direction from RRD, AMCO will design and manufacture the complete test unit as depicted in AMCO Drawing 16-J-4042 and Figure 2. The test insert will consist of the T-5 test vessel cover, the test article (valve sub-body), the support beams, sodium level indicators (thimbles), baffles, actuator to operate the test article, Ferrometic shaft seal sub-assembly, and the cover cooling coils.

AMCO has provided interface connections for LMEC to measure the test parameters. These items are pressure taps, strain gages to measure operating torque, sodium level measurement, leakage measurements, and valve position indicators.

The design has been reviewed by RRD and LMEC. The minutes of the meeting with LMEC are shown in Appendix E1. The LMEC and RRD comments were incorporated into the design so that the manufacturing and planned testing reflects the combined agreements of RRD, AMCO, and LMEC.

AMCO has submitted a request to LMEC for approval in principle (AIP) to perform static in-sodium testing of the test article. LMEC approved the AIP on all objectives with only minimal changes required for the test article. (Appendix E2).

12.0 INCH STATIC SODIUM POP TEST ARTICLE

(1) Schedule could be improved by two months if seals and subhousing fabrication can start during GFY 1975.

FIGURE 4- OVERALL SCHEDULE

FIGURE 4 - Continued

AMCO MILESTONES

1. Complete detail design of POP test article - include AMCO furnished drive mechanism (operator), torque meter drive shafting and coupling, upper shaft seal assembly, and valve internals. January 1975.
2. Purchase order for long lead materials. December 1974.
3. Start interface and liaison with LMEC. August 1974.
- △ 4. Release approval-in-principle document. November 1974.
5. Release test request document. December 1974.
- △ 6. Complete fabrication test article. January 1976.
7. Test article delivery to LMEC. February 1976.

LMEC MILESTONES

1. Response to approval-in-principle document. November 1974.
2. Response to test request. January 1975.
3. Release test proposal. February 1975. ✓
4. Authorization to test. March 1975.
5. Complete POP testing. October 1976.
6. Release test data sheets. November 1976.
7. Topical test report. December 1976.

FIG. 5

12-70
Atomics International
North America

FIG. 6

12-7-70

7741-4061

FIG. 7

12-7-70

Atomic International
North American Rockwell

741-4060

3.0 DESIGN PHILOSOPHY

The philosophy for the design of the rotating offset ball valve water test article was to meet the geometric configuration for flow oriented tests only. The design would include assurances that only the minimum number of modifications would be required for follow-on static sodium testing.

In order to accomplish this goal, elastomeric O-rings were used wherever possible at interface connections in order to reduce the cost of manufacturing and seal development.

In addition, the valve internals were fabricated with minimal amount of machining to reduce cost, thus leaving sufficient material on the parts for subsequent design modification for the static sodium POP test.

Figure 3 shows the assembly drawing of the water POP test valve with emphasis placed on the geometric configuration for the flow oriented tests. The heavy cross-sections noted of the valve internals were subsequently thinned down for the sodium POP test.

4.0 DESIGN AND TESTING HISTORY OF VALVE INTERNALS

The design and testing history to date for the water test internals is presented in References 3 and 4.

Reference 3 discusses the valve design and analysis to support the water POP test as well as calculations to support the sodium test.

Reference 4 presents the water flow test results for valve pressure drop and flow torque as well as some preliminary lip seal leakage test results.

4.1 Lip Seal and Ball Material Couple Evaluation

To date, AMCO has manufactured three lip seals for water and gas testing at room temperature. They are:

- o 304 Cres lip seal
- o Inconel 718 annealed lip seal
- o Inconel 718 heat-treated lip seal

Their configuration is shown in Figure 3.

A 304 Cres ball was also manufactured with a 0.19-inch thick Stellite 6 overlay (after grinding) on the sealing surface. Preliminary leakage and water flow tests were made with the 304 Cres seal on the Stellite 6 overlay. Results of these tests are reported in Reference 4.

AMCO proposes to do further development work on the existing stellite ball as follows based on inputs from the National Friction and Wear Program. The tentative plan for this effort is noted as follows.

- o Relap the stellited ball to a 4AA finish.
- o Lap the 304 seal to the ball to determine if this technique has merit.
- o Statically (ball locked in closed position--no relative motion between ball and seals) evaluate the lapped 304 seal and the two existing Inconel 718 seals for leakage performance under controlled laboratory conditions with filtered gas and/or water.
- o Aluminize or Tribaloy 700 a lapped or as-is Inconel 718 seal (aluminize is a diffusion coating and Tribaloy 700 can be applied by welding or using the D-gun low temperature process).
- o Chrome carbide (CN-1B) the stellited sealing portion of the ball for further static leakage evaluations with the best candidate seal couple. This requires regrounding the ball (removing approximately .005 from the stellited radius) followed by chrome carbiding to approximately .010 thickness with the D-gun process (Union Carbide at Speedway, Indiana). The ball will then be reground down to the original radius and then lapped. After the final leakage evaluations, the ball will then be ready for the static sodium POP tests proposed in the original AIP.

The best candidate material couple for the lip seal will be incorporated into the final static sodium lip seal configuration as shown in Figure 8.

DUAL METAL LIP-SEAL
FIGURE 8

5.0 ANALYSIS TO SUPPORT THE SODIUM POP TEST UNIT IN THE T-5 TEST VEHICLE AT LMEC

A stress analysis was performed on the test unit to substantiate its integrity under the extended duration sodium POP test. Investigations included all support members, pressure containing parts, and bearing surfaces. Parameters used for the design analysis were for a six-month period in sodium at a temperature 936°F. The design pressure in the body primary pressure containing region was 225 psig. Results for the analysis are shown in Appendices A and B.

5.1 Lip Seal Analysis

A finite element analysis of the aged Inconel 718 lip seal, not yet manufactured, (Figure 8 configuration) is being performed using the AMCO/AB5U computer code. Requirements of the ASME Code Section III and Code Case 1331-8 will be adhered to, considering the lip seal as a primary pressure retaining component. Appendix E is a preliminary report covering the work performed to date. The final analysis will be included in this report as a revision when completed. The metal lip seal design shown in Figure 8 performs a dual function. It provides a seal on the ball and another lip seal at the interface connection between the valve sub-body and the valve body-free-standing leg. The two lip seals are integrated into a single part, thus enhancing maintainability by providing a means of in-situ replacement of the valve internals with a minimum of cost and manipulation time.

5.2 Drive Shaft Analysis

A heat transfer analysis was conducted on the drive shaft assembly to investigate the steady state temperature at the Ferrometic shaft seal. The purpose of the analysis is to assure that the temperature does not exceed the limit temperature of 225°F for the Ferrometic seal.

The design solution to the problem if the temperature were too high is to provide cooling coils on the test vessel cover or to force cool air across the seal. The analysis is shown in Appendix B.

The analysis takes into account the convection effect of the sodium on the baffles and vessel cover. The analysis shows that the temperature at the Ferrometic seal would be 190°F which is well below the operating temperature based on the assumption that cooling coils are used on the cover as shown in Figure 7 and on the housing of the Ferrometic seal. Figure 9 shows the temperature gradient across the test article with an average water temperature of 95° through the cooling coils.

The cooling coils will be installed on the test vessel cover by AMCO prior to shipping to LMEC.

5.3 Ball Analysis

The stress analysis on the ball was performed for the water POP test and is presented in Reference 3. This analysis was based on design temperature of 965°F and a design pressure of 300 psi and is well within the operating limits of the sodium test requirements. No machining of the ball is anticipated for the static sodium test. However, it is planned that the ball sealing surface (Stellite 6) will be coated with chrome carbide (CN-1B) applied by the detonation gun process (D-gun) at approximately 400°F under controlled conditions by Union Carbide Corporation.

The chrome carbide coating on the ball was based on an AMCO study (Reference 5) to select the optimum seal and ball material combination which would best meet the valve leakage criteria, in addition to having low wear, friction, and self-welding characteristics in liquid sodium.

The Stellite 6 overlay on the 304 Cres ball at the sealing surface is under high stress at elevated temperatures due to the differential coefficient of expansion of the two materials.

There is a concern that surface cracking could occur during sodium testing if the ball is shocked thermally. Based on the technique used for applying the Stellite and the temperature duration of post-stress relieving, it was concluded that the Stellite 6 overlay integrity could only be assured if thermal shocking of the ball was avoided. This conclusion also accounted for the fact that the maximum operating temperature (936°F) was well enough below the stress relief temperature (1200°F). (Appendix E3). The maximum test pressure will be 225 psig.

AMCO does intend to observe this potential problem area closely during the forthcoming sodium POP test.

6.0 MATERIAL USED IN THE TEST ARTICLE

The materials used for the POP test article are depicted in Appendix C. This list of materials was used to purchase raw material and to aid in establishing manufacturing costs.

Appendix E5 is a report by AMCO to establish the the testing required to upgrade materials of the offset ball valve water test article to RDT Standards. Although it will not be necessary to upgrade the material for sodium testing at LMEC (Reference F1), the report shows the purchase specification for the materials and the non-destructive examinations performed on them.

7.0 MANUFACTURING OF THE TEST ARTICLE

The fabrication and assembly of the test article is scheduled for completion on 1 January 1976. A Parts List for the test article is shown in Appendix D.

AMCO will be responsible for the fabrication of the complete test insert. Present planning requires that AMCO will manufacture the test article in-house except for the Ferrometic seal and shaft sub-assembly (AMCO Drawing 16-J-4041) which will be supplied by the Ferrofluidic Corporation because of the specialized requirements of this seal. To date, manufacturing has not been initiated for this seal assembly.

Cost estimates of the 304 Cres raw material were initiated approximately eight months ago along with a review of material delivery time.

8.0 STATIC SODIUM POP TEST REQUIREMENT DOCUMENTS

The static sodium POP test requirement documents are noted as follows:

- o A request for approval in principle (AIP) was submitted to LMEC in November 1974 and is presented in Appendix E1.
- o The AIP outlines the objectives and needs of the test program as specified in the "Procedure for LMFBR Component and Instrument Testing at LMEC." (Reference 6). The information is used by LMEC for a scoping cost estimate and for schedule information. This scoping has been completed by LMEC and the AIP approved (Appendix F1 and F2).
- o The test request (Reference 2) is a detailed expansion of the AIP objectives, outlining all steps necessary to accomplish and document the test program. The test request information will be used by LMEC to form the basis of their 189 Operational Cost to RRD. In addition, LMEC will prepare a step-by-step procedure from this test request for performing the static sodium POP test.

9.0 CONCLUSIONS AND RECOMMENDATIONS

The drawings and analysis for the sodium POP test article have been completed along with a cost estimate for the fabrication phase including a long lead material procurement list. Based on the analysis and the water POP test results, the sodium POP test article is now ready for fabrication and testing.

Based on this work, the following action is recommended:

- o Prepare alternate lip seal concepts which will include the investigation of lip seal geometries and hard-surfacing material combinations based on the latest inputs from the RRD National Friction and Wear Program. A tentative plan for lip seal development for the near future is given in Section 4.1.
- o Complete fabrication of test article for the static sodium POP test.
- o Upgrade static sodium POP test article for liquid or gas thermal transient testing.
- o Consider the offset ball valve as an LMFBR Demonstration Plant IHTS drain valve. The in-line offset ball has a lower pressure drop than the angle globe valve and may be more adaptable to the system (IHTS) geometry which uses two drain valves in series.
- o Initiate shaft lip seal feature test to eliminate a freeze seal and reduce valve stem length. See Figure 10 for the schematic of the sodium feature test proposed. This was one of the candidate shaft seal system features presented at the RRD/AMCO meeting at Fullerton on 21 October 1974.
- o Consideration should be given to extending the sodium POP test for one year from the currently proposed 6 months.

SCHEMATIC -
SHAFT (METAL) LIP SEAL
SODIUM POP TEST

FIG. 10

10. REFERENCES

1. AMCO Report No. 1773-01(01)ER, "Trade Study Fast Acting Valve, dated November 1973.
2. AMCO Report No. 1839-01(04)TRD, Test Request Document, "LMFBR Large Valve Development - Static Sodium Proof-of-Principle (POP) Test of AMCO 12.0-Rotating Offset Ball Valve Internals."
3. AMCO Report No. 1773-01(09)ER, "LMFBR Large Valve Development Task II - Summary Report," dated June 1974.
4. AMCO Report No. XXXXX-"LMFBR Large Valve Development Task II - IHTS Isolation Valve Internals Interim Water Test Report," dated January 1975.
5. AMCO Report No. 1773-01(06)TM, "Selection of Seal Material for the Offset Ball Valve," dated 5 February 1974.
6. LMEC Letter No. 74LMEC-DRF-2012, "Procedure for Testing Activities at LMEC," dated 3 April 1974.

APPENDIX A

STRESS ANALYSIS STATIC SODIUM POP TEST ASSEMBLY

APPENDIX A
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REFERENCES

1. LMFBR Large Valve Development, Task II - IHTS Isolation Valve Internals, Summary Report No. 1773-01(09)ER, June 1974
2. AMCO Letter No. 74:6460:027, "Preliminary Stress Evaluation of Sodium Test Lip Seal
3. Telecon Report - T. O'Callaghan/D. DeKlotz / Dave Nulton, dated 10-12-74 - Intermediate Loop Isolation Valve (24.0 Inch)

APPENDIX A

STRESS ANALYSIS STATIC SODIUM POP TEST ASSEMBLY

1.0 SUMMARY

The stress analysis of the sodium POP test internals and accompanying structural support members are presented to assure the basic structural adequacy of the unit. The analyses for the valve internals were previously verified and are presented in Reference Report 1. However, design changes were necessary to convert the unit from water to sodium service necessitating additional valve internals to facilitate the testing in sodium.

The existing valve internals were either redesigned, down graded or used as is for various reasons. The design of the unit is based on 25,500 in-lbs of torque which was calculated in Reference 1. The torque was based on the valve closing against 300 psig which was the requirement at that time.

Accordingly, the valve has been stressed using 25,500 in-lbs to size the ball, its shafts and couplings. However, the valve closing pressure has been reduced from 300 psig to 10 psig thus reducing the torque requirements significantly.

The stress calculations are all based on the 25,500 in-lbs and do not reflect the latest low closing pressure requirement since 90% of the test article has already been manufactured. Also the possibility that the valve still might have to close against 300 psi still exists.

The test article operating time is based on the operating requirement for the 24 inch prototype valve which is "30 to 60 seconds opening or closing time". The valve is to close against a pump decay flow of 3,300 GPM corresponding to 10 psi pump pressure.

2.0 STATUS-EXISTING VALVE INTERNALS

- o Sub-Body - Drawing 16-J-4049. Remachined to remove external material.

- o Ball - Drawing 16-E-4020. Downgraded by reducing the operating torque of the shaft from 25,500 in-lbs to 22,700 in-lbs due to the 304L CRES material operating in sodium at 936 °F. Reference 1, Appendix E, Page 1.
- o Lip-Seal - Drawing 16-E-4045. Redesign from a retained seal to a floating double lip-seal. Analysis shown in Reference 2.
- o Coupling - Drawing 16-E-4035. Use as is, not to be in contact with sodium. Analysis shown in Reference 1, Appendix E.
- o Key - Drawing 16-E-4057. Use as is, not to be in contact with sodium. Analysis shown in Reference 1, Appendix E.
- o Bushing - Drawing 16-E-4048. I.D. remachined to accept sleeve Drawing 16-E-4051.
- o Stop-Pin - Drawing 16-D-4046. Sized based on cantilever beam loading. Load established from operating torque 25,500 in-lbs.

3.0 STATUS-NEW VALVE INTERNALS

- o Pin Lock - Drawing 16-E-4050. Stress analyzed to operate in sodium @ 300 psi and 936 °F. Calculated stress 4,954 psi, material 304 CRES.
- o Coupling - Drawing 16-E-4052 and Drawing 16-D-4053. Stress and analysed to operate in sodium @ a maximum torque of 25,500 in-lbs and 965 °F.
- o Bushing - Drawing 16-C-4051. Analyzed for fit into sleeve. Drawing 16-E-4048. Accounting for thermal coefficient of expansion between the two materials at sodium temperatures 965 °F.
- o Shaft-Seal- Assembly - Drawing 16-E-4041. Shaft was sized based on maximum operating torque 25,500 in-lbs @ 936 °F, the seal was established by Ferrofluidic Company.

4.0 STATUS TEST ARTICLE STRUCTURAL SUPPORT MEMBERS AND SODIUM LEVEL INDICATORS

- o Support - Drawing 16-E-4059. Sized for bending and torsional stress to resist actuation torsional loads.
- o Cover Sub- Drawing 16-E-4043. Sized to operate in Sodium @ Housing 300 psi and 936 °F.
- o Vessel - Drawing 16-E-4054. Size of vessel cover was established by LMEC T-5 vessel, Drawing No. 7693-PT14100204. The strength of the cover has been proven by previous LMEC testing.
- o Tube Sodium Leakage Indicator - Drawing 16-E-4058. Sizing of tube assembly is based on internal proof pressure of 300 psi @ 936 °F.
- o Tube Sodium Level Indicator - Drawing 16-E-4064. Unit was sized based on the vessel cover gas pressure 15 psi @ 965 °F.
- o Actuator Support Yoke - Drawing 16-E-4055. Sized to withstand maximum operating torque 40,000 in-lbs ambient temperature.

5.0 MATERIAL PROPERTIES

	<u>304 & 304H</u>			<u>304L</u>		
	R.T.	936 °	965 °	R.T.	936 °	965 °
Temp.						
S _u (ksi)	75			70		
S _y (ksi)	30	16	15.8	25	13.7	13.5
S _o & S _m (ksi)	20	14.4	14.2	16.6	12.3	12.1
α (10 ⁻⁶) in/in °F	9.1	10.2	10.25	9.1	10.2	10.5
ν Poisson Ratio	.26	.30	.31	.26	.30	.31
E(X10 ⁶ psi)	27.3	23.1	22.8	27.3	23.1	22.8
G(X10 ⁶ psi)	10.8	8.8	8.8	10.8	8.8	8.8

	Aged Inconel 718			Annealed 718
Temp	R.T.	936 °	965 °	R.T.
S_u (ksi)	185	161	160	145
S_y (ksi)	150	132.6	132.3	60
$S_o = S_m$ (ksi) ⁽¹⁾	61.6	53.6	53.3	40
S_t (ksi)				
10 ² cycles		106.8		
10 ⁴		84.6		
10 ⁵		74.0		
S_a (ksi) ⁽²⁾	105.7			
E (ksi)	29,000	25,000		
α (in/in °F)		7.96 x 10 ⁻⁶		

(1) - S_o assumed equivalent to S_m as normally derived. Per Code Section III (i.e., lesser of $S_u/3$ or $2S_y/3$).

(2) - S_a based on 1.25 S_e at 10⁴ cycles.

	Stellite 6B		
Temp.	R.T.	936 °F	965 °F
S_u (ksi)	154	130	130
S_y (ksi)	92.6	68	68
$S_o = S_m$ (ksi) ⁽¹⁾	51.33	43.33	43.33
E (psi)	30.4 x 10 ⁶	30.4 x 10 ⁶	30.4 x 10 ⁶
α (in/in °F)	7.6 x 10 ⁻⁶	8.38 x 10 ⁻⁶	8.43 x 10 ⁻⁶
ν Poisson Ratio	0.27	0.27	0.27

(1) - Lesser of $S_u/3$ or $2S_y/3$.

6.0 STRESS REQUIREMENTS

* For Shear Stress

$$S_{s_a} = 0.6 S_m \quad \text{for 304 @ 965 } S_{s_a} = 8520 \text{ psi}$$

** For Bending Stress

$$S_{b_a} = 1.5 S_m \quad \text{for 304 @ 965 } S_{b_a} = 21.3 \text{ ksi}$$

$$304 \text{ S.S. @ } 965^\circ \text{F} \quad S_m = 14.2 \text{ ksi}$$

For Bearing Stress

$$S_{br} = 1.5 S_y$$

* Code Case 1592 par. 3227.2(a) $S_{s_a} = .6 S_{m_t}$

$$S_{m_t} \text{ for 304 at } 965^\circ \text{F} = 14.2 \text{ ksi}$$

**Code Case 1592 para. 3227.1 (b) $S_{b_a} = 1.5 S_o$

$$S_o \text{ for 304 at } 965^\circ \text{F} = 14.2 \text{ ksi}$$

Report Title STRESS ANALYSIS OF VALVE INTERNALS 12" BALL VALVE				Program	
				Work Order	
Prepared By <i>[Signature]</i>	Date 2-6-75	Checked By <i>[Signature]</i>	Date 3/21/75	Revision	Date

7.0 ANALYSIS

7.1 SUB-BODY

CALCULATE STRESS AT SECTION A-A FIG. 1

$$A = 1.187 \times 1.50 = 1.78 \text{ IN}^2 \quad \text{STRESS AREA DUE TO } F/2 \text{ LOAD}$$

(300 PSI PRESSURE) (LIP SEAL LOAD)

$$S_y = \frac{F + F_{LS}}{2(2A)} = \frac{15,843 + 3000}{4 \times 1.78} = 2646 \text{ PSI}$$

OK LOW

ALLOW $S_{sa} = 8520 \text{ PSI}$ FOR 304 CRES AT 965°F

S_s IS THE SHEAR OUT STRESS NORMALLY OCCURS @ 40° OFF THE LOAD VECTOR $F/2$ AND IS CONSIDERED A NON-COMBINING STRESS

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
AB		TSK	3/21/75		

CALCULATE THE HOOP STRESS IN THE SUB-BODY ASSUMING THAT THE BALL THRUST LOAD ACTING THRU THE BUSHINGS TO THE *SUB-BODY IS DISTRIBUTED AS A SINUSOIDAL PRESSURE OVER THE DIAMETER.

SECTION C-C
OF FIG. 1

$$\sigma = \left(\frac{2}{\pi} \right) \frac{F/2}{A}$$

THE STRESS DUE TO THE
SINUSOIDAL LOADING

$$\sigma = \frac{(15,843 + 3000)}{\pi * 1.78} = 3369 \text{ PSI}$$

$$\text{ALLOW } S_m = 14,200 \text{ PSI}$$

*THIS ANALYSIS WAS PERFORMED IN REFERENCE 1 APPENDIX E AND IS BEING RECALCULATED BECAUSE OF THE SMALLER SECTION AREA.

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. Kuntz	4/9/75	DB	4/11/75		

THE PRESSURE LOAD IS TRANSFERRED BY THE SHAFT TO THE SUB-BODY. THE SUPPORT FOR THE SUB-BODY CONSISTS OF THE TWO LOCK PINS. THE FOUR POINT LOAD ON THE ASSEMBLY CREATES BOTH A BENDING MOMENT AND TWIST. THESE FORCES TEND TO SEPARATE THE BALL AND THE SEAL.

TRANSVERSE FOUR-POINT LOADING OF A CURVED BEAM. (FROM "RINGS AND ARCUATE BEAMS" BY ALEXANDER BLAKE)

CONSERVATIVE ANALYSIS
ASSUME PIN ACTS AS A
SUPPORT 90° FROM
THE LOAD

$$P = 18.850^*/2 = 9425^*$$

$$R = 7.21"$$

$$M = Pr(\sin \theta - \cos \theta)/2$$

$$T = Pr(\sin \theta + \cos \theta - 1)/2$$

DEFLECTION AS A FUNCTION OF θ

$$y = \frac{PR^3}{8EI} \left\{ 2(\theta - 1) \sin \theta + (2\theta + 2 - \pi) \cos \theta + \pi - 2 \right. \\ \left. + \frac{EI}{GK} [(6 + 2\theta - \pi) \cos \theta - 2(3 - \theta) \sin \theta + \pi + 4\theta - 6] \right\}$$

ANGLE OF TWIST AS A FUNCTION OF θ

$$\beta = \frac{PR^2}{8EI} (1 + \frac{EI}{GK}) [2(1 - \theta) \sin \theta + (\pi - 2 - 2\theta) \cos \theta]$$

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				Work Order	
Prepared By <i>F.S. Kiefer</i>	Date 3/26/75	Checked By <i>DB</i>	Date 4/11/75	Revision	Date

$D_{max} = 1.8"$

CALCULATION OF THE TORSIONAL
CONSTANT 'K'. FOR THE CROSS
SECTION.

CONSIDERED AS AN L-SECTION

$$K = K_1 + K_2 + \alpha D^4$$

$$\begin{aligned} K_1 &= ab^3 \left[\frac{1}{3} - .21 \frac{b}{a} \left(1 - \frac{1}{12} \left(\frac{b}{a} \right)^4 \right) \right] \\ &= (11)(1.8)^3 \left[\frac{1}{3} - .21 \left(\frac{1.8}{11} \right) \left(1 - \frac{1}{12} \left(\frac{1.8}{11} \right)^4 \right) \right] \\ &= [5.632] (.318) = \underline{1.79} \end{aligned}$$

$$\begin{aligned} K_2 &= cd^3 \left[\frac{1}{3} - .105 \frac{d}{c} \left(1 - \frac{1}{12} \left(\frac{d}{c} \right)^4 \right) \right] \\ &= (1)(2.5)^3 \left[\frac{1}{3} - .105 \left(\frac{2.5}{1} \right) \left(1 - \frac{1}{12} \left(\frac{2.5}{1} \right)^4 \right) \right] \\ &= (15.625) [.124] = \underline{1.94} \end{aligned}$$

$$\begin{aligned} \alpha &= \frac{d}{6} (.07 + .076 \frac{r}{b}) \quad r = 0 \\ &= (2.5/6) (.07) = .219 \end{aligned}$$

$$K = 1.79 + 1.94 + (.219)(1.8)^4 = 3.73 + 2.30$$

$$\underline{\underline{K = 6.03}}$$

Report Title				Program	
				Work Order	
Prepared By <i>P.S. Kaufman</i>	Date 3/26/75	Checked By <i>[Signature]</i>	Date 4/11/75	Revision	Date

CALCULATION OF FLEXURAL CENTER POSITION

$$I_1 = \frac{1}{12} (9.75) (.8)^3 = .416$$

$$I_1 + I_2 = 2.239$$

$$I_2 = \frac{1}{12} (1.4) (2.5)^3 = 1.823$$

$$e_x = \frac{1}{2} (1.4) \left[\frac{1.823}{2.239} \right] = .57''$$

$$e_y = \frac{1}{2} (9.75) \left[\frac{.416}{2.239} \right] = .91''$$

CALCULATE CENTER OF BENDING

BEAM 1	$X_{1c} = 0$	$Y_{1c} = 5.5$	$A_1 = 7.8$
--------	--------------	----------------	-------------

$$A_T = 11.3$$

BEAM 2	$X_{2c} = .5$	$Y_{2c} = 0$	$A_2 = 3.5$
--------	---------------	--------------	-------------

$$\bar{X} = \frac{A_1 X_{1c} + A_2 X_{2c}}{A_1 + A_2} = \frac{(3.5)(.5)}{11.3} = .155''$$

$$\bar{Y} = \frac{A_1 Y_{1c} + A_2 Y_{2c}}{A_1 + A_2} = \frac{(5.5)(7.8)}{11.3} = 3.80''$$

$$(I_x)_{xx} = \frac{1}{12} (.8)(8.5)^3 + (7.8)(5.5 - 3.8)^2 = 40.94 + 22.54 = 63.48$$

$$(I_z)_{xx} = \frac{1}{12} (1.8)(2.5)^3 + (3.5)(3.8)^2 = 2.34 + 50.54 = 52.88$$

$$\underline{\underline{I_{xx} = 116.36 \text{ in}^4}}$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
<i>RS Kufner</i>	4/9/75	<i>DR</i>	4/11/75		

CALCULATION OF BALL & SEAL SEPARATION

USING: $E = 23.1 \times 10^6 \text{ LB/IN}^2$ $G = 8.8 \times 10^6 \text{ LB/IN}^2$
 $P = 9425 \text{ LB}$ $R = 7.21 \text{ IN}$
 $I = 116.36 \text{ IN}^4$ $K = 6.03$

At 0° : $\gamma = 0$
 $\beta = \frac{PR^3}{8EI} (\pi - 2) + \frac{PR^3}{8KG} (\pi - 2) = 2.601 \times 10^{-5} + 1.315 \times 10^{-3}$
 $= 1.344 \times 10^{-3} \text{ — } .077^\circ$

At $\pi/2$: $\gamma = \frac{PR^3}{8EI} (2\pi - 4) - \frac{PR^3}{8KG} (4\pi - 12) = 3.751 \times 10^{-4} + 4.713 \times 10^{-3}$
 $= .00509 \text{ IN}$

$\beta = \frac{PR^3}{8EI} (2 - \pi) + \frac{PR^3}{8KG} (2 - \pi) \text{ — } -.077^\circ$

At $\pi/4$: $\gamma = \frac{PR^3}{8EI} (\pi - 2) + \frac{PR^3}{8KG} (2\pi - 6) \text{ — } .00254 \text{ IN}$

$\beta = 0$

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				Work Order	
Prepared By <i>F.S. Knifur</i>	Date 4/9/75	Checked By <i>JS</i>	Date 4/11/75	Revision	Date

$$Y - \text{DEFLECTION} \quad Y_T = Y + l_y (1 - \cos \theta) + l_x (\sin \theta)$$

$$X - \text{DEFLECTION} \quad X_T = l_y (\sin \theta) + l_x (1 - \cos \theta)$$

TOTAL SEPARATION IN Y DIRECTION
(PRE-LOAD DIRECTION)

$$S_T = Y_T + \left\{ R_{LS} - \sin \left[\cos^{-1} \left(\frac{4.4582 - X_T}{R_B} \right) R_B \right] \right\}$$

R_{LS} - RADIUS OF LIP SEAL 4.20 "
 R_B - RADIUS OF BALL 6.125 "
 l_y - Y-DISTANCE FROM FLEXURAL
 CENTER TO LIP66 "
 l_x - X-DISTANCE FROM FLEXURAL
 CENTER TO LIP 2.51 "

$$\text{At } 0^\circ : Y_T = 0 + .66 (9.0 \times 10^{-3}) + 2.51 (1.344 \times 10^{-3}) = 3.37 \times 10^{-3} \text{ in}$$

$$X_T = .66 (1.344 \times 10^{-3}) + .66 (9.0 \times 10^{-3}) = .89 \times 10^{-3} \text{ in}$$

$$S_T = 3.37 \times 10^{-3} + \left\{ 4.2 - \sin \left[\cos^{-1} \left(\frac{4.4582 - .00089}{6.125} \right) 6.125 \right] \right\}$$

$$= 3.37 \times 10^{-3} + \{ 4.2 - 4.20095 \} = \underline{.00242 \text{ in}} \quad \left\{ \begin{array}{l} \text{AGAINST} \\ \text{PRELOAD} \end{array} \right.$$

$$\text{At } 90^\circ : Y_T = .00509 - .00337 = .00172$$

$$X_T = .00089$$

$$S_T = .00172 + \{ 4.2 - 4.19906 \} = \underline{.00266 \text{ in}} \quad \left\{ \begin{array}{l} \text{AGAINST} \\ \text{PRELOAD} \end{array} \right.$$

$$\text{At } 45^\circ : \beta = 0 \quad \therefore S_T = Y$$

$$S_T = \underline{.00254 \text{ in}} \quad \left\{ \begin{array}{l} \text{AGAINST} \\ \text{PRELOAD} \end{array} \right.$$

TO BE CONSERVATIVE THE
PRE LOAD SHOULD BE ABLE
TO TOLERATE A DECREASE
OF .003 UNDER MAX LOAD.

Report Title				Program	
				Work Order	
Prepared By <i>J.S. Kaufman</i>	Date 3/19/75	Checked By <i>28.</i>	Date 4/11/75	Revision	Date

7.2 BUSHING RETAINING BOLT

THE MOMENT M_t TO THE BUSHING IS RESISTED BY FOUR BOLTS RETAINING THE BUSHING TO THE SUB-BODY (FIG. 1).

SIZE BOLTS

$$M_t = (F/2 + F_L/2) L$$

$$M_t = (15,843/2 + 3000/2) * .51 = 4805 \text{ IN-LB}$$

PIVOT ABOUT POINT (A)

$$2A_b (2.16^2 + 4.84^2) = 2A_b (28.1)$$

SECTIONAL AREA AT MINOR DIA 3/8-24 BOLTS

$$A_b = .0809 \text{ IN}^2$$

Report Title				Program	
				Work Order	
Prepared By R.S.K	Date 3/19/75	Checked By [Signature]	Date 4/11/75	Revision	Date

AXIAL LOAD PER BOLT (DUE TO MOMENT)

$$F_B = M_T (4.84) / 2 (28.1) = 414 \text{ #}$$

$$\sigma_B = F_B / A_B = 414 / .0809 = \underline{5.12 \text{ KSI}}$$

BOLT TORQUE 100-105 FT-LB

AXIAL LOAD ON BOLT DUE TO TORQUE

$$P_B = T_B / .2 D_B = 105 / .2 (3/8) = 1400 \text{ #}$$

STRESS ON BOLT

$$\sigma_B = P_B / A_B = 1400 \text{ #} / .0809 = 17.3 \text{ KSI}$$

ALLOWABLE STRESS, (.8)(S_y) = 24 KSI

24 KSI IS GREATER THAN
17.3 KSI ∴ OK.

BOLT MATERIAL 304

CODE CASE 1592 TABLE 1-19.12

S₀ - MAXIMUM ALLOWABLE STRESS INTENSITY AT 950°F

5.3 KSI

5.3 KSI IS GREATER THAN 5.12 KSI ∴ OK.

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
AS		J.S.K	3/4/75		

7.3 LOCK PINS

FIG 2 SHOWS HOW THE LOCK PINS HOLD THE COVER TO THE SUB-BODY.

THE PINS ARE IN PURE SHEAR AND ARE SUPPORTED IN THEIR ENTIRE LENGTH BY A CLOSE FITTING HOLE. THE FORCE ACTING ON THE PINS IS PRODUCED BY THE PRESSURE $P = 300 \text{ PSI}$ ACTING ON THE PROJECTED AREA OF THE LIP SEAL

THE FORCE IS EXPRESSED AS

$$F_c = P \frac{\pi D_c^2}{4} = 300 * \pi * \frac{9.562^2}{4} = 21,545 \text{ \#}$$

THE PINS LOCK THE COVER TO THE SUB-BODY IN FOUR PLACES. THE SHEAR AREAS ACROSS THE PINS ARE SHOWN IN SECTION A-A OF FIG. 2 WHICH IS IN EFFECT A DIAGONAL SECTION THRU THE SHAFT OF THE PIN.

Fig-2

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
AB		T.S.K.	3/21/75		

THE SHEAR AREA MAY BE EXPRESSED AS
A 45° DIAGONAL ACROSS THE PINS

$$A_{\text{SHEAR}} = \pi d_c^2 / \cos 45^\circ$$

SHEAR STRESS ON PINS & PLACES

$$S_s = \frac{F_c}{4 \pi d_c^2} = \frac{21,545}{\pi \cdot 0.875^2 \cdot 0.707} = 6334 \text{ PSI}$$

FOUR PLACES $4 \cos 45^\circ$

$$\text{ALLOW } S_{SA} = 8520 \text{ PSI}$$

304 CRS

FIG 1, 2 - CALCULATE PULL OUT SHEAR
BETWEEN PIN AND SUB-BODY

$$S_s = \frac{F_c}{A_{\text{P}} L_{\text{PIN}}} = \frac{21,545}{4 \times 1.06 \times 7.4} = 686 \text{ PSI}$$

DOUBLE SHEAR
2 PINS

AREA SUB-BODY

ANALYZE PULL OUT STRESS ON PINS
AT 75°F
ASSUMING SODIUM IS WEDGED BETWEEN
PIN AND HOLE FOR THE FULL LENGTH OF
THE PIN $L = 10.0$ "

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
		F.S.K	3/21/75		

SHEAR STRENGTH FOR SODIUM 75°F

$$\sigma_s = 80 \text{ PSI} * 2 = 160 \text{ PSI}$$

SAFETY FACTOR

THE FORCE TO PULL PIN

$$F_p = \sigma_s * \pi d_c L = 160 * \pi * .875 * 10 = 4398 \#$$

STRESS ON PIN

$$\sigma_{pin} = \frac{F_p}{\frac{\pi d_c^2}{4}} = \frac{4398 * 4}{\pi * .875^2} = 7314 \text{ PSI}$$

SELECT PIN MATERIAL - 304 CRES

ALL STRESSES ARE LOW.

7.4 LEAK TUBE

CALCULATE HOOP STRESS OF THE OUTER AND INNER TUBE DUE TO TEST PRESSURE $P = 300$ SEE FIG 1 FOR DIMENSIONS OF TUBE.

OUTER TUBE

$$S = \frac{pY}{t}$$

$$p = 300 \text{ PSI}$$

$$Y = \frac{1.315}{2} =$$

$$t = .133$$

$$S = \frac{300 * .66}{.133} = 1483 \text{ PSI LOW}$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
<i>JB</i>		<i>J.S.K</i>	<i>3/21/75</i>		

INNER TUBE

$$P = 300$$

$$r = \frac{840}{2} = .42$$

$$S = \frac{300 \times .42}{.109} = 1155 \text{ PSI} \quad t = .109$$

7.5 OLDHAM COUPLING

COUPLING: THE COUPLING DESIGN
TORQUE $T = 25,500 \text{ IN-LB}$.

CALCULATE THE STRESS TO SIZE THE
COUPLING.

THE SIZE OF THE COUPLING DIMENSIONS
IS SHOWN IN FIGURE 1

MATL: 304H CRES

ANALYSIS:

THE TONGUES ON THE CENTRAL PIECE
OF THE COUPLING ENGAGE THE GROOVES
ON THE CENTRAL END PIECES WHICH
ARE ATTACHED TO THE SHAFT.

THESE TONGUES $1.00" \times 1.125" \times 5.25"$ LG
ARE SUBJECTED TO THE FULL
TORSIONAL STRESS AND ARE TREATED AS
SHORT RECTANGULAR SECTIONS.

Report Title				Program	
				Work Order	
Prepared By <i>J.S. Kinfan</i>	Date 3/25/75	Checked By <i>DP</i>	Date 4/11/75	Revision	Date

APPROXIMATION OF SHEAR STRESS FOR SHALLOW RECTANGULAR BEAM

$$\sigma_s = T_c / J$$

$$J = wt^3/12 + tw^3/12 = 14.2$$

$$\sigma_s = (25,500)(5.25/2) / 14.2 = 4718 \text{ PSI}$$

MAX LOAD

$$V_{max} = \sigma_s t = 5307 \text{ LB/IN}$$

$$M_b = V l / 2 = 5307 * .5 = 2654 \text{ IN-LB/IN}$$

MAX MOMENT

$$\sigma_b = 6 M_b / t^2 = \frac{6 * 2654}{1.125^2} = 12,580 \text{ PSI}$$

BENDING STRESS

Report Title				Program	
				Work Order	
Prepared By <i>T.S. K...</i>	Date 3/25/75	Checked By <i>...</i>	Date 4/11/75	Revision	Date

AXIAL STRESS COMBINED WITH SHEAR CASE
(3) TABLE II FORMULAS FOR STRESS AND STRAIN,
ROARK.

$$\text{PRINCIPAL STRESSES } \frac{1}{2}\sigma_b \pm \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \sigma_s^2}$$

MAXIMUM STRESS INTENSITY

$$2 \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + \sigma_s^2} = 2 \sqrt{(12,580/2)^2 + (4718)^2} = \underline{\underline{15,725 \text{ psi}}}$$

ALLOWABLE 304 AT 936°F

$$1.5 S_m = 1.5 (14.4 K) = 21,600 \text{ psi}$$

Report Title				Program	
				Work Order	
Prepared By <i>[Signature]</i>	Date 3-2-75	Checked By <i>[Signature]</i>	Date 3/2/75	Revision	Date

7.6 OLDHAM COUPLING KEY

REFERENCE TEXT - DESIGN OF MACHINE
ELEMENTS FOURTH EDITION BY FAIRES.
PAGE 282

$$S_s = 2T / bLD \text{ SHEAR}$$

$$T = 25,500 \text{ IN-LB}$$

$$t = b = .625$$

$$L = 2.00$$

$$D = 2.50$$

$$S_s = \frac{2 * 25,500}{.625 * 2.00 * 2.50} = 16,320 \text{ PSI} \quad \text{SHEAR}$$

Report Title				Program	
				Work Order	
Prepared By <i>[Signature]</i>	Date 3-2-75	Checked By <i>J.S.L.</i>	Date 3/4/75	Revision	Date

TWO KEYS ARE BEING USED THEREFOR TWO
STRESS PER KEY

$$S_{s/KEY} = 16,320 / 2 = 8160 \text{ PSI}$$

ALLOWABLE FOR 30XCRS @ 936°F $S_s = 60\% S_m = .6 (14,400) = 8640 \text{ PSI}$

$$S_c = 4 T / t L D \quad \text{COMPRESSION}$$

$$S_c = 4 * 25,500 / (.625 * 2 * 2.5) = 32,640 \text{ PSI FOR}$$

TWO KEYS

$$\text{FOR ONE KEY } S_{c/KEY} = 32,640 / 2 = 16,320 \text{ PSI}$$

ALLOWABLE FOR 30XCRS @ 936°F $S_{c_allow} = S_m * 1.5 = 14,400 * 1.5 = 21,600 \text{ PSI}$

CONCLUSION BASED ON THE ANALYSIS

THE KEYS ARE SAFE. THE STRESSES
ARE BELOW THE ALLOWABLES.

7.7 BALL OVERLAY (THERMAL EXPANSION)

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. Kaufman	3/26/75	DG.	4/11/75		

DISCUSSION OF THE AFFECT DUE TO THE DIFFERENCE IN THE COEFFICIENTS OF THERMAL EXPANSION BETWEEN THE BALL AND THE OVERLAY.

MATERIAL PROPERTIES

	304		STELLITE	
	70°	936°	70°	936°
S_u (KSI)	75		154	130
S_y (KSI)	30	16.0	92.6	68
α ($10^{-6}/^{\circ}F$)	9.1	10.2	7.6	8.38
ν	.26	.3	.27	.27
E (10^6 PSI)	27.3	23.1	30.4	30.4

THE STELLITE (OVERLAY) WAS APPLIED TO THE BALL AT A TEMPERATURE OF 1800° F. THE BALL WAS THEN COOLED TO 70° F, ($\Delta T = -1730^{\circ}F$), PUTTING THE STELLITE IN COMPRESSION. THE STRESS DUE TO THE CHANGE

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. Kaufman	3/28/75	ED	4/11/75		

IN TEMPERATURE OF -1730°F YIELDS
THE STELLITE BUT NOT THE BALL:

USING α AT 936°F & S_y, E, ν AT 70°F
 $\nu_s = \nu_b \approx .27$

$$\sigma_{\text{STELLITE}} = \frac{E_s \Delta \alpha \Delta T}{(1-\nu) \left[1 + \frac{E_s t_s}{E_b t_b} \right]} = 118 \text{ KSI} > 92.6 \text{ KSI}$$

COMPRESSION

$$t_s = .19" \quad t_b = 1.904$$

$$\sigma_{\text{BALL}} = \frac{E_b \Delta \alpha \Delta T}{(1-\nu) \left[1 + \frac{E_b t_b}{E_s t_s} \right]} = 11.8 \text{ KSI} < 30 \text{ KSI}$$

TENSION

SUBJECTING THE OVERLAY TO STEADY STATE
TEMPERATURES BELOW THE STRESS RELIEF
TEMPERATURE (1800°F) WILL TEND TO RELIEVE
THE COMPRESSIVE STRESS THAT EXISTS AT ROOM TEMP:

$$\text{AT } T = 936^{\circ} \quad \Delta T = 936^{\circ} - 1800^{\circ} = -864^{\circ}\text{F}$$

$$\sigma_{\text{STELLITE}} = \frac{E_s \Delta \alpha \Delta T}{(1-\nu) \left[1 + \frac{E_s t_s}{E_b t_b} \right]} \approx 57.9 \text{ KSI} < 68 \text{ KSI}$$

COMPRESSION

$$\sigma_{\text{BALL}} = \frac{E_b \Delta \alpha \Delta T}{(1-\nu) \left[1 + \frac{E_b t_b}{E_s t_s} \right]} \approx 5.8 \text{ KSI} < S_m = 14.2 \text{ KSI}$$

TENSION

AEROJET MANUFACTURING COMPANY

7.8 BUSHINGS

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SUBJECT THERMAL EXPANSION ANALYSIS OF SUB-HOUSING SUB-ASSEMBLY		REV.	NO.
		WORK ORDER	DATE 10-30-74
PREPARED BY JFP	CHECKED BY <i>[Signature]</i>	APPROVED BY	APPROVED BY SCALE <i>#</i>

SHAFT 304 L
O.D. 2.500/2.502
2.502 ACTUAL

THE ANALYSIS WAS PERFORMED TO ESTABLISH THE TOLERANCE OF THE O.D. AND I.D. OF THE BUSHING TO ASSURE THAT EXCESSIVE HOOP STRESSES ARE NOT INDUCED DUE TO THE HIGH SODIUM TEMPERATURES.

DESIGN CONDITIONS $\alpha_s = 11.25 \times 10^{-6} / F^\circ$
 $\alpha_b = 8.33 \times 10^{-6} / F^\circ$
 $T = 935^\circ F$ $T_a = 70^\circ F$ $\Delta T = 866^\circ F$

$$\Delta \text{SHAFT} = \alpha_s D_s \Delta T_s \quad \Delta \text{BUSHING} = \alpha_b D_b \Delta T_b$$

DESIRE .0005 TO .0010 CLEARANCE AT DESIGN TEMP. (DIA.)

— SHAFT - BUSHING —

$$\Delta_s = \alpha_s D_s \Delta T_s = (11.25 \times 10^{-6})(2.502)(866) = .0244$$

$$D_s' = D_s + \Delta = 2.5264$$

FOR BUSHING

AT DESIGN TEMP. (936°F)

BUSHING DIA = 2.5268/2.5273

AEROJET MANUFACTURING COMPANY

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SUBJECT THERMAL EXPANSION ANALYSIS OF SUB-HOUSING SUB-ASSEMBLY		REV.	NO.
		WORK ORDER	DATE 10 30 74
PREPARED BY JFP	CHECKED BY P.S.K.	APPROVED BY	APPROVED BY
			SCALE H

$$D_b = D'_b / (1 + \alpha_b \Delta T)$$

$$D_b = D'_b / [1 + (8.33 \times 10^{-6} \times 866)] = D'_b / 1.0072$$

$$D_b = 2.5087 / 2.5092$$

DESIGN AS 2.509 +.001/-.000

BUSHING I.D.	T _{amb}	T _{Na}
MIN.	2.509	2.527
MAX.	2.510	2.528

SHAFT O.D.		
ACTUAL	2.502	2.527

MIN. CLEARANCE		
ACT. SHAFT / MIN. BUSH	.007	.000
MAX. CLEARANCE		
ACT. SHAFT / MAX BUSH.	.008	.001

— BUSHING — SUB-HOUSING —

$$\alpha_b = 8.33 \times 10^{-6} \quad \alpha_{SH} = 11.25 \times 10^{-6}$$

I.D. SUB-HOUSING - 2.750/2.752

DESIRE .0010 MIN. CLEARANCE AT ROOM TEMPERATURE

SUB-HOUSING I.D.	T _{amb}	T _{Na}
MIN.	2.750	2.7768
MAX.	2.752	2.788

BUSHING O.D.

MIN.	2.748	2.7678
MAX	2.749	2.7688

AEROJET MANUFACTURING COMPANY

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SUBJECT THERMAL EXPANSION ANALYSIS OF SUB-HOUSING SUB-ASSEMBLY			REV.	NO.
			WORK ORDER	DATE 10 30-74
PREPARED BY JFP	CHECKED BY P.S.K.	APPROVED BY	APPROVED BY	SCALE 1X

BUSHING-SUB-HOUSING (CONT)

MIN. CLEARANCE	T_{amb}	T_{max}
MAX BUSHING/MIN. SUB-HSING	.001	.008
MAX. CLEARANCE		
MIN. BUSHING/MAX. SUB-HSING	.004	.011

BUSHING (UPPER)

ROOM TEMP DIMENSIONS

Report Title BUSHING STRESS ANALYSIS				Program	
FOR: 12" BALL VALVE				Work Order	
Prepared By JFP	Date 11-27-71	Checked By J.S.K.	Date 3-25-71	Revision	Date

SUB-HOUSING SHAFTS — CALCULATE CONTACT STRESS

D_1 = HOUSING I.D. AT 936°F = 2.7788 (MAX)

D_2 = BUSHING O.D. AT 936°F = 2.7678 (MIN)

ASSUME SIDE THRUST IS DIVIDED EQUALLY BETWEEN LOWER & UPPER BUSHINGS

$$F_b = \frac{1}{2} F_T = \frac{1}{2} (15,843) = 7921 \#$$

LENGTH OF BUSHINGS ARE EQUAL AT 2.375"

LOAD / LINEAR INCH OF LENGTH OF BUSHING = P

$$P = 7921 \# / 2.375 \text{ INCHES} = 3335 \#/\text{IN}$$

Report Title BUSHING STRESS ANALYSIS				Program	
For: 12" BALL VALVE				Work Order	
Prepared By JFP	Date 11-27-74	Checked By P.S.K.	Date 3-25-75	Revision	Date

FROM ROARK TABLE XIV CASE 6.

MAXIMUM CONTACT STRESS

$$S_{c_{max}} = 0.798 \sqrt{\frac{P \frac{D_1 - D_2}{D_1 D_2}}{\left[\frac{1 - \nu_1^2}{E_1} + \frac{1 - \nu_2^2}{E_2} \right]}}$$

TEMP. = 936°F

$$\begin{array}{l} 304 \\ 5.5 \end{array} \left\{ \begin{array}{l} \nu_1 = 0.3 \\ E_1 = 23.1 \times 10^6 \text{ PSI} \end{array} \right. \quad \begin{array}{l} \text{STELLITE} \\ 6B \end{array} \left\{ \begin{array}{l} \nu_2 = 0.27 \\ E_2 = 30.4 \times 10^6 \text{ PSI} \end{array} \right.$$

$$S_{c_{max}} = 0.798 \left\{ 3335 \left[\frac{2.78}{(2.78)} \frac{-2.77}{(2.77)} \right] / \left[\frac{1 - (0.3)^2}{23.1 \times 10^6} + \frac{1 - (0.27)^2}{30.4 \times 10^6} \right] \right\}^{1/2}$$

$$S_{c_{max}} = 0.798 \left\{ 3335 * .001244 / 3.94 \times 10^{-8} + 3.05 \times 10^{-8} \right\}^{1/2}$$

$$S_{c_{max}} = 0.798 \left\{ 6.197 \times 10^7 \right\}^{1/2} = \underline{6281 \text{ PSI}} \quad \text{OK}$$

$$\text{ALLOW } \underline{S_{va}} = 1\frac{1}{2} S_y = 1.5 * 15,800 = 23,700 \text{ PSI}$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
TS Kiefer	4/9/75	LD	4/11/75		

7.9 SUPPORT MEMBERS

THE TORQUE ON THE SHAFT IS REACTED BY THE TWO SUPPORT MEMBERS. THE TORQUE BOTH BENDS AND TWISTS THE MEMBERS. THE BOX BEAM IS USED, INSTEAD OF AN OPEN CHANNEL, BECAUSE OF ITS GREATER STIFFNESS.

ASSUME ROTATION ABOUT POINT A.

TWO BOX BEAMS $6 \times 3\frac{3}{8} \times \frac{3}{8}$

Report Title				Program	
				Work Order	
Prepared By <i>TS Kiefer</i>	Date 4/9/75	Checked By <i>h</i>	Date 4/11/75	Revision	Date

$$a = 3.375''$$

$$b = 6.0''$$

$$t = t_1 = .375''$$

HOLLOW RECTANGLE

CALCULATE TORSION CONSTANT (K)

$$K = \frac{2tt_1(b-t)^2(a-t_1)^2}{bt + at_1 - t^2 - t_1^2} = \frac{2(.375)^2(5.625)^2(3.0)^2}{(.375)(9.375) - 2(.375)^2}$$

$$K = 24.76 \text{ IN}^4$$

CALCULATE MOMENT OF INERTIA (I_{xx})

$$I_{xx} = \frac{ba^3 - dc^3}{12} = \frac{(6)(3.375)^3 - (5.25)(2.625)^3}{12}$$

$$I_{xx} = 11.31 \text{ IN}^4$$

Report Title				Program	
				Work Order	
Prepared By <i>JS Kuntz</i>	Date 4/9/75	Checked By <i>DJ</i>	Date 4/11/75	Revision	Date

ANGLE OF TWIST : $\theta = T_B L / K_G \leftarrow \{ \text{PURE TORSION} \}$

DEFLECTION : $\delta = F_B L^3 / 3EI \leftarrow \{ \text{SIMPLE SUPPORT AT TEST COVER} \}$
 NO SLOPE AT VALVE COVER

COMPATIBILITY REQUIREMENT

$$\delta = 5 \sin \theta$$

$$C_2 F_B = 5 \sin (C_1 T_B)$$

FORCE BALANCE

$$T = 2(T_B + 5F_B)$$

$$F_B = (T - 2T_B) / 10$$

WHERE : $C_1 = L / K_G \left(\frac{1}{1.25 - L_B} \right)$
 $C_2 = L^3 / 3EI \left(\frac{1}{16} \right)$

(CONSERVATIVELY ASSUMES
ONE END OF BEAM IS
PINNED)

COMBINED

$$(T - 2T_B) (C_2 / 50) = \sin (C_1 T_B)$$

WITH : $E = 23.1 \times 10^6 \text{ PSI}$ $G = 8.8 \times 10^6 \text{ psi}$

$T = 25,500 \text{ IN-LB}$ $L = 44 \text{ IN}$

THEN : $C_1 = 2.019 \times 10^{-7}$ $C_2 = 1.087 \times 10^{-9}$

$$5.544 \times 10^{-2} - 4.348 \times 10^{-6} (T_B) = \sin [2.019 \times 10^{-7} (T_B)]$$

$T_B = 12,180 \text{ IN-LB} \rightarrow \theta = C_1 T_B \left(\frac{180}{\pi} \right) = .141^\circ$

$F_B = 115 \text{ LB} \rightarrow \delta = C_2 F_B = .0125''$

SHAFT DISPLACEMENT UNDER DESIGN LOAD

UNDER DESIGN LOAD THE
TEST MUST TOLERATE A
SHAFT MOVEMENT OF .03''

$$\delta_s = (11.15) (\sin \theta) = .0274''$$

Report Title				Program	
				Work Order	
Prepared By <i>P.S. Kaufman</i>	Date 4/11/75	Checked By <i>FX</i>	Date 4/11/75	Revision	Date

STRESS ON SUPPORT MEMBER

(ASSUME CHANNEL AT TEST FIXTURE COVER)

$$S_s = \frac{G \theta C}{L}$$

$$\text{WHERE } C = \frac{D}{1 + \frac{\pi^2 D^4}{16 A^2}} \left[1 + 1.5 \left(\frac{\pi^2 D^4}{16 A^2} - D/2r \right) \right] = .437$$

$$D = .439 \quad A = 2.11 \quad r = 0$$

$$S_s = (8.8 \times 10^6) (.00246) (.437) / (.44) = 215 \text{ } ^\circ\text{B}/\text{IN}^2$$

$$S = M_c / I = (6920) (.9) / 3.6 = 1730 \text{ } ^\circ\text{B}/\text{IN}^2$$

$$\text{MAX STRESS INTENSITY} = 2 \sqrt{(.25)^2 + (S_s)^2} = 1784 \text{ } ^\circ\text{B}/\text{IN}^2$$

LOW

BENDING STRESS ON SUPPORT ANGLE

ASTM A36

$$S_y = 36 \text{ KSI}$$

$$S_{ab} = 32 \text{ KSI}$$

$$S = M_c / I$$

$$S = (6920) (2.2) / 9.06$$

$$S = 1677 \text{ } ^\circ\text{B}/\text{IN}^2$$

LOW

ASSUMED CONFIGURATION

$$I = bh^3/12$$

$$I_1 = (6)(3.75)^3/12 + Ad^2 = 3.83$$

$$I_2 = (7.5)(3)^3/12 + Ad^2 = 5.25$$

$$\bar{x} = A_1 x_{1c} + A_2 x_{2c} / A_T = 1.48$$

$$I_T = 9.06 \text{ IN}^4$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. KINTNER	9/2/75	AP	4/11/75		

TEST FIXTURE BOLT STRESSES & TORQUE VALUES

W_1 ~ COVER WEIGHT AT DISTANCE $l_1 = 2 \text{ IN}$ 286"

W_2 ~ BALL & HARDWARE WEIGHT AT DISTANCE $l_2 = 9.5 \text{ IN}$ 234"

W_1 & W_2 NOT USED IN TORQUE CALCULATION (SMALL)

Report Title				Program	
				Work Order	
Prepared By <i>H.S. Kaufman</i>	Date 4/10/75	Checked By <i>L.D.</i>	Date 4/11/75	Revision	Date

BOLT MATERIAL

ASTM - A453 GR 651 (CLB)

 $S_y = 60 \text{ KSI}$ $S_y = 36 \text{ KSI AT } 950^\circ$

- 1] SUPPORT BEAM TO SUB HOUSING COVER (ITEM 29, 16-J-4042)
8 REQD $\frac{3}{4}$ "-10 UNC (AT 936°F)

$$\text{SHEAR : } F_s = (w_1/2 + w_2/2)/4 = 65 \text{ LB}$$

$$\text{TENSION : } F_T = F_s/4 + T_B/2(3.14) = 2060 \text{ LB}$$

$$\sigma_T = 2060 / .302 = 6.82 \text{ KSI}$$

TORQUE 70-75 FT-LB

$$\text{TENSION : } B_T = (70)(12)/(2)(\frac{3}{4}) = 5600^\circ$$

$$\text{FRICTION FORCE : } (.15)(5600) = 840^\circ$$

- 2] SUPPORT BEAM TO ANGLE
8 REQD 1"-8 UNC (AT $\sim 300^\circ\text{F}$)

$$\text{SHEAR : } F_s = (w_1/2 + w_2/2 + w_3) = 90 \text{ LB}$$

$$\text{TENSION : } F_T = F_s/4 + T_0/2(3.14) + [(F_s)(48) + w_1/2(2) + w_2/2(9.5)]/2(2.75) \\ = 3360 \text{ LB}$$

$$\sigma_T = 3360 / .551 = 6.10 \text{ KSI}$$

TORQUE 270-275 FT-LB

$$\text{TENSION : } B_T = (270)(12)/(2)(1) = 16,200 \text{ LB}$$

$$\text{FRICTION FORCE : } (.15)(16,200 \text{ LB}) = 2430 \text{ LB}$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
F.S. Kaufman	4/11/75	AG	4/11/75		

3] ANGLE TO SODIUM POT COVER
8 REQD 1" - 8 UNC

$$\text{SHEAR} : 12,180 / 2(3.4) = 1790 \text{ LB}$$

$$\text{TENSION} : [(F_B)(48) + W/2(2) + W/2(9.5)] / 2(3.4) = 1020 \text{ LB}$$

$$\text{MAX STRESS INTENSITY} \rightarrow 2 \sqrt{[(S_1/2)^2 + (S_2)^2]}$$

$$2 \left\{ [(1020)/(2 \times .55)]^2 + [1790/.55]^2 \right\}^{1/2} = 6.76 \text{ KSI}$$

TORQUE 270-275 FT-LB

$$B_T = 16,200 \text{ LB}$$

$$\text{FRICTION FORCE} = 2430 \text{ LB}$$

7.10 SODIUM LEAK TUBE - VOLUME DISPLACEMENT

Report Title		LEAK TUBE EVACUATION FOR		Program	
TEST PROCEDURE				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
JFP	11-22-74	A.S. Kiffin	3/31/75		

TO INCREASE THE SODIUM LEVEL IN THE LEAK TUBE TO ALLOW FOR ADDITIONAL TEST VOLUME, THE PRESSURE ABOVE THE SODIUM MAY BE DECREASED TO ALLOW THE TANK PRESSURE TO FORCE THE COLUMN UP.

SODIUM LEVEL SHOULD BE KEPT BELOW BAFFLE LEVEL TO PRECLUDE THE POSSIBILITY OF FREEZING THE SODIUM IN THE TUBE.

MAX DESIGN LEVEL = 12" ABOVE TANK LEVEL

FOR A COLUMN OF SODIUM 12" HIGH, PRESSURE REQUIRED EQUALS

$$P = h \gamma = (1 \text{ FT}) (\gamma)$$

FOR SODIUM AT $\sim 900^\circ\text{F}$, $\gamma = 52 \text{ #/ft}^3$

$$P = 52 \text{ #/ft}^3 \cdot 1 \text{ FT} = \underline{\underline{.36 \text{ PSI}}}$$

Report Title <u>LEAK TUBE TESTING</u>				Program	
				Work Order	
Prepared By <u>JFP</u>	Date <u>11-12-74</u>	Checked By <u>F.S. K.</u>	Date <u>3/31/75</u>	Revision	Date

FOR LEAK TESTING WITH THE WELL-PRESSURIZING TUBE

I.D. OF PRESSURIZING TUBE (I.D.) = 1.049
O.D. OF WELL TUBE (O.D.) = .840

AREA OF SODIUM SURFACE = $.7854(O.D.^2 - I.D.^2) = .31$ SQ. IN.

FOR 1" Δh Volume = .31 cu in. = 5.08 c.c.

AT $T = 936^\circ F$, ASSUME PIPE TEMP. = $T = 936^\circ F$

$\alpha = 10.2 \times 10^{-6}$ IN/IN $^\circ F$ ($\Delta T = 866^\circ F$)

$D' = \text{DIAMETER AT } 936^\circ F = D(1 + \alpha \Delta T)$

$I.D.' = 1.0583$ $O.D.' = .8474$

AREA' = .316 IN 2 .

FOR 1" Δh AT T , $V = 5.17$ c.c

CONVERSION FACTOR FOR LEAK CHECK AT TEMPERATURE = $936^\circ F$

FOR Δh = 1" LEAK = 5.17 c.c ; 1 c.c / .193" Δh.

7.11 COVER - SUB BODY

Report Title END PLATE ANALYSIS

Program

12" BALL VALVE SODIUM TEST

Work Order

Prepared By JFP

Date

11-13-74

Checked By R.S.K.

Date

3/31/75

Revision

Date

$$P_0 = 225 \text{ PSIG @ } 936^\circ\text{F}$$

$$A_P = 10.5''$$

Report Title <i>END PLATE ANALYSIS</i>				Program	
12" BALL VALVE SODIUM TEST				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
<i>JFP</i>	<i>11-13-74</i>	<i>R.S.K.</i>	<i>3/31/75</i>		

ASSUME THE FOLLOWING MODEL

FROM ROARK TABLE X CASE 2

$$\text{MAX } S_r = S_t = -\frac{3W}{2\pi m t^2} \left[m + (m+1) \log \frac{a}{r_0} - (m-1) \frac{r_0^2}{4a^2} \right]$$

$$t = 2.5"$$

$$a = 9"$$

$$r_0 = 5.25"$$

Report Title END PLATE ANALYSIS				Program	
12" BALL VALVE SODIUM TEST				Work Order	
Prepared By JFR	Date 11/13/74	Checked By P.S.K.	Date 3/21/75	Revision	Date

$$W = P_o \pi r_o^2$$

For 304 S.S. TEMP = 936°F. $\nu = 0.3$

$$m = \frac{1}{\nu} = 3.33$$

$$\text{MAX } S_r = S_t = - \frac{3P_o \pi r_o^2}{2\pi m t^2} \left[m + (m+1) \log \frac{a}{r_o} - (m-1) \frac{r_o^2}{4a^2} \right]$$

$$\text{MAX } S_r = S_t =$$

$$- \frac{3(225)(5.25)^2}{2(3.33)(2.5)^2} \left[3.33 + (4.33) \log \frac{9}{5.25} - (2.33) \frac{(5.25)^2}{(4)(9)^2} \right]$$

$$= -447 [3.33 + 2.33 - 1.198] = -447 (5.462)$$

$$= \underline{\underline{2442 \text{ PSI}}}$$

TORK-MOR DIMENSIONS AND RATINGS

ACTUATOR SELECTION BASED ON 25/500 IN-LB. TORQUE

THEORETICAL TORQUE

SINGLE VANE—280° ROTATION.

DOUBLE VANE—100° ROTATION.

MODEL	DIMENSIONS				TORQUE-IN-LBS. AT PRESSURE			CAP. FREE AIR CU. FT.	CAP. OIL GAL
	A	B	C	D	100#/in	500#/in	1000#/in		
S-3-3 DS-3-3	3	4-3/4	1-3/4	.750	140	700		.005	.037
S-4-2	4-1/2	4-1/4	3	1.000	300	1,500	3,000	.010	.077
S-4-4	4-1/2	6	3	1.000	600	3,000	6,000	.018	.13
S-4-6	4-1/2	8	3	1.000	900	4,500	9,000	.027	.20
S-7-4	8	7	5-1/2	1.750	2,000	10,000	20,000	.061	.46
S-7-7	8	10	5-1/2	1.750	3,500	17,500	35,000	.107	.80
S-10-5	12-1/4	10	9	3.000	6,000	30,000	60,000	.188	1.40
S-10-8	12-1/4	13	9	3.000	9,700	48,500	97,000	.301	2.25
S-10-12	12-1/4	17	9	3.000	14,500	72,500		.452	3.38

MODEL	DIMENSIONS				TORQUE-IN-LBS. AT PRESSURE			CAP. FREE AIR CU. FT.	CAP. OIL GAL
	A	B	C	D	100#/in	500#/in	1000#/in		
DS-3-3 DS-3-3	3	4-3/4	1-3/4	.750	280	1,400		.003	.026
DS-4-2	4-1/2	4-1/4	3	1.000	600	3,000	6,000	.007	.055
DS-4-4	4-1/2	6	3	1.000	1,200	6,000	12,000	.013	.098
DS-4-6	4-1/2	8	3	1.000	1,800	9,000	18,000	.019	.147
DS-7-4	8	7	5-1/2	1.750	4,000	20,000	40,000	.043	.32
DS-7-7	8	10	5-1/2	1.750	7,000	35,000	70,000	.076	.57
DS-10-5	12-1/4	10	9	3.000	12,000	60,000	120,000	.134	1.00
DS-10-8	12-1/4	13	9	3.000	19,000	95,000	190,000	.215	1.61
DS-10-12	12-1/4	17	9	3.000	29,000	145,000		.322	2.41

Splined shafts extra — consult factory for price.

Series 3 and 4 have double end shaft, standard.

Series 7-10 double end shaft available at additional cost.

ADJUSTABLE TAPER LOCK FACE MOUNTED STOP

PATENT APPLIED FOR

An external stop must be used under heavy load or high speed operation. This face mounted, adjustable taper lock stop on the load arm prevents shock load to internal components of the actuator. It is readily adjustable to any degree of rotation the job may require on either single or double vane actuators. We will gladly advise proper stops for your use.

MODEL	A	B
S-3-3	3	13/8
S-4	4 1/2	1 5/8
S-7	8	2 1/8

HEAVY DUTY MODELS

TORK-MOR Heavy Duty actuators are of the same basic design as the standard models, but of heavier construction to handle pressures up to 2,000 p.s.i.

All Heavy duty models are equipped with husky roller bearings. Exclusive design molded seals are backed-up with teflon to assure longer, efficient seal life and all materials used in building these models are carefully selected for ample strength and safety margin.

For dimensions see page 4

SINGLE VANE—(HEAVY DUTY) —280° ROTATION.

DOUBLE VANE—(HEAVY DUTY) —100° ROTATION.

MODEL	DIMENSIONS				TORQUE-IN-LBS AT PRESSURE					CAP. FREE AIR CU. FT.	CAP. OIL GAL
	A	B	C	D	100# /in	500# /in	1000# /in	1500# /in	2000# /in		
HS-7-4	B	8-3/4	5-1/2	2,500	1,000	5,000	10,000	15,000	20,000	.033	.24
HS-7-7	B	11-3/4	5-1/2	2,500	1,750	8,750	17,500	26,250	35,000	.057	.43
HS-10-5	12-1/4	11	9	4,000	4,500	22,500	45,000	67,500	90,000	.139	1.04
HS-10-8	12-1/4	14	9	4,000	7,200	36,000	72,000	108,000	144,000	.222	1.66
HS-10-12	12-1/4	18	9	4,000	10,750	53,750	107,500	161,250	215,000	.334	2.49

MODEL	DIMENSIONS				TORQUE-IN-LBS AT PRESSURE					CAP. FREE AIR CU. FT.	CAP. OIL GAL
	A	B	C	D	100# /in	500# /in	1000# /in	1500# /in	2000# /in		
HDS-7-4	B	8-3/4	5-1/2	2,500	2,000	10,000	20,000	30,000	40,000	.022	.18
HDS-7-7	B	11-3/4	5-1/2	2,500	3,500	17,500	35,000	52,500	70,000	.041	.31
HDS-10-5	12-1/4	11	9	4,000	4,500	22,500	45,000	67,500	90,000	.099	.74
HDS-10-8	12-1/4	14	9	4,000	14,400	72,000	144,000	216,000	288,000	.159	1.19
HDS-10-12	12-1/4	18	9	4,000	21,500	107,500	215,000	322,500	430,000	.238	1.78

Splined shafts extra — consult factory for price.

Series 7-10 double end shaft available at additional cost.

LIMIT SWITCH ACTIVATOR

In many instances Tork-Mor actuators can be used to operate valves or switches at any desired degree of rotation or sequence of operation. This is accomplished by limit switches operating solenoid valves or cams operating valves.

PART NO.	A	B	C	D	E	F	G	H	J
F-51-S3	3/4	1-1/8	7/8	3/4	1/2	1/4	5/8	3/16	5/8
F-51-S4	1	1-1/2	1-3/8	1	7/8	3/8	1-1/8	1/4	1
F-51-S7	1-3/4	2-1/4	1-3/4	1	7/8	3/8	1-1/8	1/4	1
F-51-HS7	2-1/2	3-1/4	2-3/8	1	1	1/2	1-1/8	1/4	1
F-51-S10	3	3-3/4	2-5/8	1	1	1/2	1-1/4	1/4	1
F-51-HS10	4	4-3/4	3-3/8	1	1-1/4	1/2	1-1/4	1/4	1

Special models built on request

7.13 ACTUATOR SUPPORT FRAME

Report Title ANALYSIS OF UPPER MOUNTING FRAME

Program

FOR: 12" BALL VALVE

Work Order

Prepared By

JAP

Date

11-20-74

Checked By

J.S.K

Date

3/31/75

Revision

Date

STRESS DUE TO TORQUE ON FRAME

A: $\frac{3}{8}$ MAX. SIZE SCREW - $13\frac{1}{4}"$ ϕ B.C.B: $\frac{5}{8}"$ SIZE SCREWS - $16"$ ϕ B.C.

ALL MATERIAL CARBON STL.

SA-307 GRADE B

$$S_a = 7 \text{ KSI}$$

$$S_{sa} = (1.6) 7 = 4.2 \text{ KSI}$$

Report Title: ANALYSIS OF UPPER MOUNTING FRAME				Program	
				Work Order	
Prepared By: <i>T.S. Kirtan</i>	Date: 4/3/75	Checked By: <i>DR</i>	Date: 4/11/75	Revision	Date

TWO HOLES

D = 10"

MAXIMUM EXPECTED TORQUE ~ 25,500"

$$\sigma_s = F/A_s$$

$$A_s = r_m (\pi - 2\phi) t$$

$$\phi = \sin^{-1} (5/6.625) = 49^\circ$$

$$A_s = 6.625 (180^\circ - 98^\circ) .75 = 7.11 \text{ in}^2$$

$$2F\ell = T$$

$$F = T/2\ell = 25,500 / 2(6.625) = 1925 \text{ LB}$$

$$\sigma_s = 1925 / 7.11 = 271 \text{ LB/in}^2$$

LOW

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. Kirtner	4/11/75	[Signature]	4/11/75		

ACTUATOR FRAME - BOLT STRESSES & TORQUE VALUES

ASTM - A307 GR B

$S_u = 60 \text{ KSI}$

$S_y = 33 \text{ KSI}$

SAE GRADE 1

MAX TORQUE :

$3/8 - 16 \text{ UNC}$	-	12 FT-LBS
$5/8 - 11 \text{ UNC}$	-	60 FT-LBS
$3/4 - 10 \text{ UNC}$	-	102 FT-LBS

- 1] ACTUATOR TO ACTUATOR FLANGE (ITEM 35, 16-J-4042)
4 REQD $3/4 - 10 \text{ UNC}$

$$\text{SHEAR FORCE } F_s = T_f / 4R = 25,500 / (4)(4.725) = 1350^*$$

$$G_s = F_s / A_s = 1350 / .302 = 4.47 \text{ KSI}$$

TORQUE 100-102 FT-LBS

$$\text{FRICTION FORCE} = (.15) \left[\frac{100(12)}{(.2)(3/4)} \right] = 1200^* \quad \left| \begin{array}{l} \text{SLIP MAY} \\ \text{OCCUR} \end{array} \right.$$

- 2] ACTUATOR FLANGE TO ACTUATOR SUPPORT (ITEM 36)
8 REQD $3/8 - 16 \text{ UNC}$

$$\text{SHEAR FORCE } F_s = 25,500 / (6)(6.625) = 481^*$$

$$G_s = 481 / .0678 = 7.10 \text{ KSI}$$

TORQUE 11-12 FT-LBS

$$\text{FRICTION FORCE} = (.15) \left[\frac{(11)(12)}{(.2)(3/8)} \right] = 264^* \quad \left| \begin{array}{l} \text{SLIP MAY} \\ \text{OCCUR} \end{array} \right.$$

- 3] ACTUATOR SUPPORT TO SODIUM POT COVER (ITEM 33)
8 REQD $5/8 - 11 \text{ UNC}$

$$\text{SHEAR FORCE } F_s = 25,500 / (6)(6) = 398^*$$

$$G_s = 398 / .202 = 1.97 \text{ KSI}$$

TORQUE 58-60 FT-LBS

$$\text{FRICTION FORCE} = (.15) \left[\frac{(58)(12)}{(.2)(5/8)} \right] = 835^*$$

7.14 SHAFT STRESS CALCULATIONS

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
R.S. Kufner	3/19/75	PJ	4/11/75		

CHECK SHAFT - STRESS DUE TO TORSION &
BENDING MOMENT

$$T = 25,500 \text{ IN-LB}$$

$$F/2 = 9,420 \text{ LB}$$

$$S = M/I_c = (F/2)(.89)/I_c = 8389/I_c$$

$$I_c = \frac{1}{4} \pi R^4 = (\frac{1}{4})(3.14)(1.25)^4 = 1.539$$

$$S = 5465 \text{ PSI}$$

SHEAR

$$S_s = \frac{2T}{\pi r^3} + \frac{F/2}{\pi r^2} = \frac{(2)(25,500)}{\pi (1.25)^3} + \frac{(9,420)}{\pi (1.25)^2}$$

$$= 8310 + 1920 = 10,230 \text{ PSI}$$

COMBINED STRESS:

$$S_p = \frac{1}{2} S \pm \sqrt{(\frac{1}{2} S)^2 + (S_s)^2} = 4268 \pm 10,590$$

$$\text{MAX INTENSITY} = \underline{\underline{21.2 \text{ KSI}}}$$

$$\text{ALLOW} = 1.5 S_m = 21.3 \text{ KSI}$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. Linfear	4/11/75	hsp	4/11/75		

SHAFT WELD ANALYSIS (T = 25,500 IN-LB)

TORSIONAL SHEAR STRESS IN THE WELD IS NEAR MAXIMUM IN THE 45° CONICAL SECTION ASSUMING A UNIFORM DISTRIBUTION OF PRIMARY SHEAR STRESS. THE TORSIONAL MOMENT CAN BE FOUND BY INTEGRATION, AND THE EQUATION REARRANGED TO CALCULATE THE PRIMARY SHEAR.

$$T = \int_{r_1}^{r_0} (2\pi r)(\sqrt{2} dr) S_s r = 2\sqrt{2} \pi S_s \int_{r_1}^{r_0} r^2 dr$$

$$T = \frac{2}{3} \sqrt{2} \pi S_s (r_0^3 - r_1^3)$$

$$S_s = \frac{1.5 T}{\sqrt{2} \pi (r_0^3 - r_1^3)}$$

$$\text{WHERE } r_1 = \frac{r_0 + r_c}{2}$$

UPPER TRUNNION : 5/8 WELD

$$r_0 = 1.25 \quad r_1 = .9375$$

$$S_s = (1.5)(25,500) / [\sqrt{2} \pi (1.25^3 - .9375^3)] = \underline{7625 \text{ PSI}}$$

LOWER TRUNNION 1/2 WELD

CONSERVATIVELY ASSUME T = 25,500/2 = 12,750 IN-LB

$$r_0 = 1.25 \quad r_1 = 1.$$

$$S_s = \underline{4520 \text{ PSI}}$$

FOR 304 AT 936°

$$S_{sa} = \underline{8520 \text{ PSI}}$$

Report Title				Program	
				Work Order	
Prepared By	Date	Checked By	Date	Revision	Date
J.S. Kumpf	4/16/75	DP	4/17/75		

7.15 STOP PIN

THE STOP PIN IS DESIGNED TO TAKE THE FULL DESIGN TORQUE OF 25,500 IN-LB IN BOTH THE OPENED & CLOSED POSITION.

PIN DIMENSIONS : DIAMETER - 1.25" LENGTH 3"
 THREAD - 1.375-12 UNC ENGAGED .85"
 PITCH DIAMETER - 1.3209"
 CONTACT WITH BALL ≈ 1.2"
 $I = \pi D^4 / 64 = .12 \text{ IN}^4$

CLOSED

BENDING MOMENT

$$\sigma_B = (T/2.5)(2.4)(\frac{1.375}{2}) / .120$$

$$\sigma_B = 5 T \text{ } ^{\circ}\text{B}/\text{IN}^2$$

$$\sigma_B = (5)(25.5) = \underline{127.5 \text{ KSI}}$$

STOP PIN MUST BE MADE OF ICONEL 718

$$S_Y = 136.2 \text{ KSI (AT } 936^{\circ}\text{F)}$$

OPENED

THREAD SHEAR

$$\text{CONSERVATIVE } A = \frac{\pi D_p h}{2} = 1.74 \text{ IN}^2$$

$$S_s = F/A = (T/5.125) / 1.74 = (.112) T = \underline{2.86 \text{ KSI}}$$

ALLOWABLE - 304 SUB-HOUSING AT 936°F

$$S_{sa} = (.6)(S_Y) = 9.6 \text{ KSI}$$

APPENDIX B

SODIUM POP TEST THERMAL ANALYSIS

INTER-OFFICE MEMO

TO: D. Pangburn
FROM: D.E. Knittle
SUBJECT: Sodium Isolation Valve POP Test - Steady State Thermal Analysis
DISTRIBUTION: J.E. DeKlotz, M.L. Nelson, T. O'Callaghan, J. Perschler,
V.A. Smith, J.F. Soppet

75:6460:013

DATE: 23 January 1975

DEK:pc

Reference: 1. AIP to Dr. G.W. Cunningham Reactor R&D AEC from
S.G. Harbison AMCO, Request for Approval POP Test
LMFBR Large Sodium Valve Development Program
74:6460:18, dated 4 November 1974

1.0 SUMMARY

This memorandum deals with a steady-state thermal analysis of the intermediate loop sodium isolation valve POP test which was done using the SINDA computer code. All of the assumptions made to simplify the thermal analysis were conservative but one (infinite conductance from cooling duct to sodium tank cover surface).

The Ferrometic seal and 'O' ring cover seal temperatures can be maintained below the allowed limits by using active cooling, see Tables 1 and 2 and Figure 3. The active cooling should consist of placing cooling coils on the top of the sodium tank cover and flowing water through the coils at ≥ 100 gal/Hr. The minimum inlet water temperature allowed depends upon the location of the innermost cooling coil as given in Table 1. The cooling coils must be thermally bonded to the top of the sodium tank cover. This can be done using heat conducting putty as shown in Figure 1. The water will remove about 5000 Btu/Hr of heat from the sodium.

The heat load which must be removed by the cooling system which is also the heat loss from the sodium reservoir can be reduced with the addition of radiation baffles between the liquid sodium and the tank cover.

2.0 INTRODUCTION

In the sodium valve POP test it will be necessary to maintain the seals below their allowed temperature limits which are as follows:

Ferrometic Seals	225 °F
Sodium Tank Cover Seals	500 °F

This thermal analysis was undertaken to determine the most inexpensive active or passive cooling or combination thereof which will maintain the seals below their allowed temperature limits.

3.0 DISCUSSION

3.1 Thermal Model General Description

The test article and its supporting structure were divided into nodes as shown in Figure 2. All of the solid material is 304 stainless steel except the shaft which is Inconel 718. Thermal conductivity of all of the solid material was considered to vary as a function of temperature. See Figure 4.

3.2 Assumptions Used in the Analysis

The analysis was based upon the following assumptions:

1. The liquid sodium temperature was 936 °F.
2. The ambient air temperature was 100 °F.
3. The argon cover gas temperature was 936 °F.
4. The temperature of the argon cover gas remains constant as it flows along the shaft (a conservative assumption). (This assumption was used in all cases except in Case 8.)
5. The argon cover gas was free to flow along the bottom of the sodium tank cover (a conservative assumption). (This assumption was used in all cases except Case 8.)
6. The emissivity of the sodium, the bottom of the sodium tank cover and of all of the radiations baffles was 0.8 (a conservative assumption which assumed contamination of all surfaces).
7. The argon cover gas at atmospheric pressure fills all of the gaps below the cover seals.
8. The heat conductance from the cooling coil surface to the top of the sodium tank cover is infinite. (Not a conservative assumption but if good thermal contact is established between the coils and the surface a reasonable one.)

3.3 Heat Equations Used (See Nomenclature in Section 3.8)

3.3.1 Conduction.

Axial

$$G = \frac{AK}{L} \quad (\text{Eq. 1})$$

Radial

$$G = \frac{2\pi lK}{\ln(D_2/D_1)} \quad (\text{Eq. 2})$$

Series Conductors

$$G = \frac{1}{\sum_{i=1}^n \frac{1}{G_i}} \quad (\text{Eq. 3})$$

Parallel Conductors

$$G = \sum_{i=1}^n G_i \quad (\text{Eq. 4})$$

3.3.2 Natural Convection

$$\frac{h_c L}{k_f} = C \left[\frac{L^3 \rho_f^2 g \beta_f \Delta t}{\mu_f} \left(\frac{C_p \mu}{k} \right)_f \right]^n \quad (\text{Eq. 5})$$

Type of Surface	Flow Regime	C	n	Equation
Vertical Surface	Laminar	0.59	1/4	5a
Vertical Surface	Turbulent	0.13	1/3	5b
Horizontal Plate	Laminar	0.54	1/4	5c
Hot Plate facing upward Cold Plate facing downward	Turbulent	0.14	1/3	5d
Horizontal Plate	Laminar	0.27	1/4	5e
Hot Plate facing downward Cold plate facing upward	Turbulent	0.07	1/3	5f

3.3.3 Forced Convection

Outside Pipes

$$\frac{h D}{k_f} = 0.0239 \left(\rho_f \frac{DV}{\mu_f} \right)^{0.805} \quad (\text{Eq. 6})$$

Inside Pipes

$$\frac{h D}{k_f} = 0.023 (N_{Re})^{0.8} (N_{Pr})^{0.4} \quad (\text{Eq. 7})$$

3.3.4 Radiation

$$Q = \sigma F A (T_1^4 - T_2^4) \quad (\text{Eq. 8})$$

Where T_1 and T_2 are in degrees Rankin

3.4 Source Heat Paths

The following heat paths from the liquid sodium and argon cover gas were considered.

3.4.1 Conduction Heat Paths.

Shaft & Support Brackets

The temperature of the shaft and support bracket was considered to be at the liquid sodium temperature at or below the sodium level. Where the shaft was divided radially into 2 nodes, conduction between the nodes on the same level was accounted for using Equation 2. The support bracket was thermally attached to the sodium tank cover as shown in Figure 2.

3.4.2 Convection Heat Paths.

Natural convection to the shaft, the support brackets and the underside of the sodium tank cover from the argon cover gas was included in the model (except in Case 8). The temperature of the argon cover gas was kept constant at the source temperature (936 °F) which is a conservative assumption since the temperature of the argon gas will fall as it gives up heat to the solid surfaces.

The natural convection equation used for the shaft and support bracket was that for laminar flow on vertical surfaces (i.e. equation 5a). The film coefficients were reduced by one half above the start of the radiation baffles to account for the restricted flow passage. The equation used for the bottom face of the sodium tank cover was for a cold plate facing downward (i.e. equation 5d).

3.4.3 Radiation Heat Paths.

Heat transfer only by radiation was considered for the radiation baffles using equation 8. The emissivity of all surfaces was considered to be 0.8^1 and a view factor of 1 was used. Radiation from the liquid sodium source through the radiation baffles to the lower face of the sodium tank cover was considered. The presence of the argon cover gas between the radiation baffles was ignored.

3.5 Sink Heat Paths to Air.

The following heat paths to the surrounding air at 100°F were considered.

3.5.1 Natural Convection.

Natural convection to the air from the seal assembly housing surface using equation 5a and from the top face of the sodium tank cover using equation 5c were considered.

3.5.2 Radiations.

Radiation to the air from the seal assembly housing and the top face of the sodium tank cover were considered using equation 8. The view factors for these surfaces were taken as 0.5 and the emissivities as 0.4.

3.6 Water Cooling

Heat transfer to the cooling water was considered using a constant film coefficient of $h = 1000 \text{ Btu/Hr Ft}^2 ^\circ\text{F}$ based on equation 7 assuming a flow rate of 1000 lbm/Hr of water. The average temperature of the cooling water was assumed to be 5°F higher than the inlet cooling water temperature (e.g. a rise of 10°F from inlet to outlet). The temperature of the inlet cooling water used for the various cases is given in Table 2. Conduction between the cooling tube wall and the surface node was considered to be infinite. This was the only non-conservative assumption made in the analyses and its affect on the accuracy of the results is negligible if a good heat path is assured in the method used to bond the cooling tube coils to the cover surface.

¹ The use of $\epsilon = 0.8$ assumes that all of the surfaces of the radiation baffles are contaminated since the emissivity range of stainless steel is 0.3 to 0.6.

3.7 Conduction Heat Paths

Equations 1 and 2 were used to calculate the conduction conductors in the axial and radial direction respectively. Thermal conductivity was considered as a function of the average temperatures of the connected nodes. Gaseous conduction was considered to be the predominant mode of heat transfer in the argon filled gaps between the shaft and the surrounding metal surfaces.

3.8 Nomenclature

3.8.1 Latin Symbols.

$A \sim$ cross sectional area, surface area $\sim \text{ft}^2$

$C \sim$ constant

$C_p \sim$ specific heat - $\text{Btu/Lbm } ^\circ\text{F}$

$D \sim$ diameter $\sim \text{ft}$

$F \sim$ gray body view factor $\sim$ dimensionless

$g \sim$ acceleration of gravity $\sim \text{ft/Hr}^2$

$G \sim$ conduction

$h \sim$ film coefficient $\sim \text{Btu/HrFt}^2 \text{ } ^\circ\text{F}$

$i \sim$ th item

$k \sim$ thermal conductivity $\sim \frac{\text{Btu}}{\text{Hr-Ft-}^\circ\text{F}}$

$L \sim$ path length between nodes $\sim \text{ft}$

$\ell \sim$ thickness of node $\sim \text{ft}$

$L \sim$ length $\sim \text{ft}$

$n \sim$ number of items; exponent

$NG_r \sim$ Grashoff No.

$NP_r \sim$ Prandtl No.

$NR_e \sim$ Reynolds No.

$V \sim$ Velocity $\sim \text{ft/hr}$

3.8.2 Greek Symbols.

$\beta \sim$ coefficient of volumetric expansion $\sim \frac{1}{^{\circ}\text{R}}$

$\Delta t \sim$ temperature difference $\sim ^{\circ}\text{F}$

$\epsilon \sim$ infra red emissivity $\sim$ dimensionless

$\mu \sim$ dynamic viscosity $\sim \frac{\text{lbm}}{\text{ft-hr}}$

$\pi \sim$ Pi

$\rho \sim$ density $\sim \text{lbm/ft}^3$

$\sigma \sim$ Steffan Boltzman constant $\sim 0.1713 \times 10^{-8} \frac{\text{Btu}}{\text{Hr Ft}^2 ^{\circ}\text{R}^4}$

3.8.3 Subscripts.

2 $\sim$ outside

1 $\sim$ inside

f $\sim$ film

i $\sim$ i th item

4.0 RESULTS

The maximum temperatures for the Ferrometic seals, and the maximum temperatures for the upper most radiation baffle and the lower face of the sodium tank cover, which establish the maximum cover seal temperatures for the cases considered, are presented in Table 2. In addition, the temperature profiles for Cases 5 and 8 (where the innermost cooling coil was placed on the seal assembly housing with (Case 5) and without (Case 8) heating from the argon cover gas are presented in Figure 3.

4.1 Active Cooling Systems.

4.1.1 Water Cooling.

The active cooling system can consist of coils of ducting attached to the top of the sodium tank cover. Cold water passing through the ducting can cool the top of the cover to a temperature near the water temperature. If no other active cooling system is to be used and the inlet water temperature exceeds (or can exceed) 62°F , then at least one of the ~~cooling~~ water ducting coils must be placed inside the access

*This number includes a 10°F analytical error margin.

cylinder weldment in thermal contact with the cover and/or the seal assembly housing. The coils must be thermally connected to the top cover.

An acceptable method using heat conducting putty is shown in Figure 1.

Figure 1. Cooling Duct Attachment.

In the following table the maximum allowable water temperature is defined (including a 10 °F error margin) as a function of the location of the innermost cooling coil and the use of additional active cooling or additional heat barriers if any.

4.1.2 Forced Air Cooling.

It can be seen in Table 1 that forced air cooling of the seal assembly housing will allow the use of ≤ 85 °F inlet water and the placing of the innermost cooling water coil outside the seal access housing weldment. It is not, however, sufficient of itself (in the absence of the water cooling system) to keep either the Ferrometic seals or the sodium tank cover seals below their allowed temperature limits. Most of the heat which must be removed from the tank cover comes by radiation from the sodium and by convection from the cover gas (the cover gas path can be partially blocked however).

4.2 Passive Cooling.

4.2.1 Radiation Barriers.

Doubling the number of radiation barriers¹ will reduce the heat loss from the sodium

¹ It was assumed in the analysis that all of the radiation barriers are contaminated and, therefore, have a high emissivity $\epsilon = 0.8$.

TABLE 1

COOLING WATER INLET TEMPERATURE REQUIREMENTS

Location of Innermost Cooling Water Coil	Additional Active or Passive Cooling or Heat Barriers*	Maximum Allowable Water Inlet Temp. °F
Outside Access Cylinder Weldment	None	62 °F
Outside Seal Assembly Housing	None	107 °F
On the Seal Assembly Housing	None	150 °F
Outside Access Cylinder Weldment	8 Radiation Barriers Added	70 °F
Outside Access Cylinder Weldment	Forced air @ 10 mph in Transverse Flow over Seal Assembly Housing	85 °F

* In addition to 7 radiation barriers and the cooling water coils on the top surface of the sodium tank cover.

reservoir by nearly $1/2$ (or from $3600 \frac{\text{BTU}}{\text{Hr}}$ to $2000 \frac{\text{BTU}}{\text{Hr}}$). This heat loss is, however, only significant if

- 1) One cannot permit a $3600 \frac{\text{BTU}}{\text{Hr}}$ heat loss from the sodium tank
- or
- 2) No (or an inadequate number of) cooling water coils were used.

Otherwise the cooling water can easily remove the higher heat load resulting from the use of fewer radiation barriers and keep the cover seals below their allowed maximum temperature.

4.3 Argon Gas Heating.

It can be seen by comparing Table 2, Case 5 (argone gas heating) and Case 8 (no argon gas heating) that the presence of argon gas at the source temperature (936°F) along the shaft contribute greatly to increasing the temperature of the Ferrometic seal (by 40°F)¹. There is no reasonable way of eliminating this source of heat but its effect can be reduced by reducing the opening. The real world will be some where between Cases 5 and 8 but the system should be designed such that the boundary conditions considered in Case 5 can be accounted for to assure that the maximum Ferrometic seal temperature is not exceeded. Case 5 can be considered as conservative because in the thermal model the temperature of the source (the argon gas) was held constant at 936°F along the entire heat path.

In the actual case the temperature of the gas will decrease as it passes downward since its thermal capacity is small and it gives up heat to the shaft as it flows.

5.0 CONCLUSIONS AND RECOMMENDATIONS

Recommended Cooling.

In conclusion the Ferrometic seal and sodium tank cover seals can be maintained well below their allowed limits if

- 1) Cooling coils are placed on the top of the sodium tank cover insuring good conduction contact with that surface (see Figure 1).

¹ This is true even through the film coefficient is small $h < 0.7 \frac{\text{BTU}}{\text{Ht Ft}^2 ^{\circ}\text{F}}$ since the ΔT is large.

TABLE 2
RESULTS

Case No.	Cooling Water Inlet Temp.	Location of Innermost Cooling Water Coil	Forced Air Cooling of Seal Ass'y. Housing @ 10 mph.	Number of Radiation Baffles	Max. Allowable* Cooling Water Temp. to Keep Ferritic Seal $\leq 225^{\circ}\text{F} \sim ^{\circ}\text{F}$	Maximum Ferrometic Seal Temperature $\sim ^{\circ}\text{F}$	Temperature ** Margin for Ferrometic Seals	Max. Temp. of Lower Face of Sodium Tank Cover $\sim ^{\circ}\text{F}$	Max. Temp. of Uppermost Radiation Baffle ($\epsilon = 0.8$) $\sim ^{\circ}\text{F}$
1	95	Outside Seal access Housing Weldment	No	7	72	238	-13		
2	70	Outside Seal access Housing Weldment	No	7	72	224	+1	176	353 Ave. + 76 Max. above Average
3	60	Outside Seal access Housing Weldment	No	7	72	218	+7	170	351 Ave. + 78
4	95	Outside Seal Ass'y Housing	No	7	117	213	+12	164	359 Ave. + 40
5	95	On Seal Assembly Housing	No	7	160	189	+36	143	359 Ave. + 23
6	70	Outside Seal access Housing Weldment	No	15	80	216	+9	166	257 Ave. + 76
7	95	Outside Seal access Housing Weldment	Yes	7	95	225	+ 0	194	361 Ave. + 74
8***	95	On Seal Assembly Housing	No	7	200	148	+77	117	354 Ave. + 17

* Does not necessarily represent the cooling water inlet temperature used to produce the maximum ferritic seal temperature given but it is that which will keep the maximum ferritic seal temperature $\leq 225^{\circ}\text{F}$.

** A 10°F margin is suggested (i.e. a calculated maximum seal temperature $\leq 215^{\circ}\text{F}$) to insure that the maximum seal temperature is not exceeded.

***Heating from the argon cover gas was ignored on all surfaces.

- 2) The innermost cooling coil be placed inside the seal access housing weldment.

The remaining coils should be evenly distributed on the sodium tank cover with the outermost near the outer radius.

- 3) The cooling water inlet temperature be maintained at or below the following:
- a) 150 °F if the innermost coil is placed on the seal housing
 - b) 107 °F if the innermost coil is placed outside the seal housing
- 4) The minimum water flow rate be not less than 100 gal/Hr (830 lbM/Hr).
- 5) A minimum number of radiation barriers can be used (4 to 7) if a heat loss of 5000 BTU/Hr or less from the sodium reservoir can be accepted. More radiation heat barriers should be used if a lower heat loss is desired.

D.E. Knittle
Analytical Development & Support

**VALVE INTERNALS -
SODIUM POD TEST
SCHEMATIC**
NODE IDENTIFICATION

PORTION SHOWN
INSIDE PHANTOM
LINE TO BE FURN-
ISHED BY AMCO

BOUNDARY NODES	
NO.	DESCRIPTION
999	SODIUM & ARGON
998	AIR
997	WATER

FIGURE 2

TEMPERATURE DISTRIBUTIONS

WITH & WITHOUT COVER GAS HEATING

ROTARY ACTUATOR

PORTION SHOWN -
INSIDE PHANTOM
LINE TO BE FURN-
ISHED BY AMCO

FIGURE 3

THERMAL CONDUCTIVITY
VS
TEMPERATURE

DKNITTLE
19 JAN. 75

FIGURE 4

APPENDIX C

MATERIALS LIST SODIUM POP TEST ARTICLE

ACTUAL DIMENSIONS	TITLE	MATERIAL			PART NO.
<p>(1)</p> <p>18.00 DIA x 5.12</p>	COVER, SUB HSG	304			16-E-4043
<p>(1)</p> <p>11.60 x 8.09 x 1.90 O.D. x I.D.</p>	LIP SEAL	INCONEL 718			16-J-4045
<p>(1)</p> <p>15.20 x 10.50 x .80 O.D. x I.D.</p>	RETAINER, SEAL	304			16-E-4047
<p>(2)</p> <p>1.00 DIA x 15.3</p>	PIN, LOCK	304			16-E-4050
<p>(2)</p> <p>3.25 x 2.50 x 2.40 O.D. x I.D.</p>	BUSHING	TRIBALCOY 700			16-E-4051

ACTUAL DIMENSIONS	TITLE	MATERIAL		PART NO.
(1) 5.25 DIA x 9.70	COUPLING	304		16-E-4052
(4) .625 SQ x 1.75	KEY	INCONEL 718		16-E-4053
(1) 42.0 DIA x 2.0	COVER, SODIUM POT (1)	304		16-E-4054
(1) 14.00 O.D. x .75 WALL x 14.80 PIPE (1) 18.00 O.D. x 12.50 I.D. x 1.00	ACCESS CYLINDER WELOMENT	CARBON STEEL		16-E-4055

ACTUAL DIMENSIONS

TITLE

MATERIAL

PART NO.

(1)

FLANGE,
ACTUATOR
MOUNT

CARBON
STEEL

16-E-4056

(2)

KEY

INCONEL
718

16-E-4057

.625
SQ X 3.50

(1)

THIMBLE
WELDMENT

304

16-E-4058

1" PIPE SCH 40 X 49.50
(1.315 O.D. X .133 WALL)

(1)

1/2" PIPE SCH 40 X 48.75
(.840 O.D. X .109 WALL)

(1)

1/4" PIPE SCH 40 X 14.00
(.540 O.D. X .088 WALL)

(1)

1.062 DIA X 1.50

ACTUAL DIMENSIONS	TITLE	MATERIAL			PART No.
<p>(4)</p> <p>3 x 3 x 3/8 ANGLE x 48.75 LONG</p> <p>(1)</p> <p>3.75 x 1.50 x 48.0</p>	SUPPORT BEAM WELDMENT	304			16-J-4059
<p>(1)</p> <p>10.60 DIA x .12</p>	SPACER, LIP SEAL	304			16-E-4062
<p>(1)</p> <p>3.90 DIA x 1.00</p>	SPACER, COUPLING	CARBON STEEL			16-E-4063
<p>(1)</p> <p>1/2" PIPE SCH 40 x 48.75</p>	SODIUM POT LEVEL INDICATOR	304			16-E-4064

ACTUAL DIMENSIONS	TITLE	MATERIAL		PART NO.
(15) 34.50 DIA X .062 (4) 3/8" PIPE SCH 40 X 12.25 (.675 O.D. X .091 WALL)	BAFFLE WELDMENT	304		16-E-4065
(2) 6 x 6 x 3/8 ANGLE X 6.0 LONG (2) 3/4 x 3.0 x 7.0	ANGLE WELDMENT	CARBON STEEL		16-E-4066

BASIC PARTS LIST

APPENDIX D

BASIC PARTS LIST

DRAWING NUMBER
16-E-4042
SHEET 1 OF 1

REV.	DATE	REVISION	BY	CHECKED

TITLE VALVE INTERNALS,
12" OFFSET BALL

DRAWN BOYLAN DATE 12/20/74
CHECKED
APPROVED

SIZE	DRAWING NUMBER	REV	1	2	3	4	5	6	7	8	PART NAME	NO REQ'D	ITEM No.
	16-E-4041										SHAFT SEAL AS-1	1	3
	16-F-4043					X					COVER SUB H-1	1	4
	16-J-4044					1					SUB HOUSING	1	5
	16-J-4045					✓					LIP SEAL	1	6
	16-E-4046					✓					PIN, BALL STOP	1	7
	16-E-4047										TUBE	1	8
	16-E-4048										SLEEVE, UPPER	1	9
	16-E-4049										SLEEVE, LOWER	1	10
	16-E-4050										PIN, LOCK	2	11
	16-C-4051					✓					E-12113	2	12
	16-E-4052-1					✓					COUPLING	2	13
	16-E-4052-2					~					COUPLING	1	14
	16-E-4053					✓					KEY	4	15
	16-E-4054					✓					COVER, SODIUM POT	1	16
	16-E-4055					✓					ACCESS, WELDMENT	1	17
	16-E-4056					✓					FLANGE, ACTUATOR MOUNT	1	18
	16-E-4057					✓					KEY	2	19
	16-E-4058					✓					THIMBLE, WELDMENT	1	20
	16-E-4059					✓					SUPPORT BEAM, WELDMENT	1	21
	16-E-4062					✓					SPACER, LIP SEAL	1	22
	16-E-4063					✓					SPACER, COUPLING	1	23
	16-E-4064					✓					SODIUM LEVEL INDICATOR	1	24
	16-E-4065					✓					BAFFLE WELDMENT	1	25
	16-E-4066					✓					ANGLE WELDMENT	1	26

TODAY'S NUMBER

SHEET

OF

[illegible]

[illegible]

APPENDIX E

AMCO CORRESPONDENCE

APPENDIX E

AMCO CORRESPONDENCE

TABLE OF CONTENTS

- E1 AMCO Letter No. 75:6460:179, "AMCO Meeting with LMEC on 12.0-Inch Sodium POP Test Valve, dated November 1974.
- E2 AMCO (AIP) No. 74:6460:18, "Proof of Principal Test AMCO 12.0-Inch Rotating Offset Ball Valve Internals," dated 4 November 1974.
- E3 AMCO Letter No. 74:6700:023, "Proposed Weld and Process Sequence for 12.0-Inch Offset Ball Valve Components, Drawing No. 16-E-4020," dated 20 March 1974.
- E4 IOM 75:6460:027, D Pangburn from C F Reimers, "Preliminary Stress Evaluation of Sodium Test Lip Seal," dated 2/17/75
- E5 AMCO Report No. 74:6460:116, "Upgrading of Offset Ball Valve Materials for Subsequent Sodium POP Test," dated 4 June 1974.

INTER-OFFICE MEMO

AEROJET MANUFACTURING COMPANY

APPENDIX E/

TO: T. O'Callaghan

DATE: 11 November 1974
74:6460:179

FROM: J. E. DeKlotz

SUBJECT: AMCO meeting with LMEC on 12 Inch Sodium POP Test Valve

DISTRIBUTION: J. DeKlotz, L. E. DeWall, M. Nelson, J. Perschler, J. Soppet,
S. Harbison

Reference: Draft AIP

Enclosure: (1) Minutes of Meeting AMCO/LMEC

A meeting was held at LMEC Santa Susana, California, Bldg. 468. The purpose of the meeting was to discuss and review the Draft AIP submitted to LMEC by AMCO, (see Ref. (1)); and to review the sodium POP test valve design, AMCO Drawing 16-J-4042.

The AIP was submitted to LMEC to satisfy the agreements and commitments established at a meeting with RRD on the AMCO large valve development program, 21 October 1974.

The meeting was successful whereby the Draft AIP was reviewed with only minor changes. A comment by LMEC concerning the AIP was that it was one of the better ones they have seen.

The problems and questions regarding the design details of the sodium POP test unit were also resolved and will be incorporated into the design of the sodium POP test valve.

The minutes of the meeting are presented in Enclosure (1).

A handwritten signature in dark ink, appearing to read "J. E. DeKlotz".

J. E. DeKlotz
Supervisor
Design Development

AMCO Large Valve Development Meeting Minutes
with LMEC, 31 October 1974, at LMEC

The meeting was held at LMEC Santa Susana, California, Building 468.

The purpose of the meeting was to discuss and review the AIP submitted to LMEC by AMCO on 22 October 1974 under contract number AT-(04)-1035AEC SAN, and to review the sodium POP test valve design AMCO Drawing 16-J-4042.

ATTENDEES:

Joe Perschler	AMCO - Engineer
V. A. Smith	AMCO - Engineer
L. E. DeWall	AMCO - Engineer
F. Poucher	LMEC - Manager, Operations General Test
E. (Vern) Ferguson	LMEC - Manager Systems Engineer for General Test X5829
J. T. Nagamatsu	LMEC - Director LMEC Construction Program (12 M. W. Pumping Facility) - Program Manager of General Test X6345
D. R. Binns	LMEC - Project Manager of General Test X 6133
J. C. Robison	LMEC - Contracts X6335
Ted Johnson	LMEC - Engineer, Mechanical Components X6410
Bud Conners	LMEC - Manager of Components
Mr. Wagner	LMEC - Manager Materials

A. Initial comment by Mr. Johnson (LMEC) concerning AIP was that it was very good, one of the better ones he had seen, and that there were only minor changes LMEC would recommend for the final draft.

1. Figure 1, Date #6 milestones should be 1976 instead of 1975. Also date 4 should be November rather than October.
2. In description of opening and closing of valve with ΔP across ball, Paragraph 3.0, delete "closing" requirement under ΔP . This cannot be performed and is not required.
3. General comment (Ted Johnson), "The test justification section of the AIP be bolstered." - No elaboration.

In respect to commitments by LMEC, none could be obtained formally although Mr. Ferguson stated that, with the minor changes noted, the AIP was acceptable to LMEC and therefore the revised AIP could be submitted to LMEC and RRD for approval and funding authorization. Mr. Smith (AMCO) gave a November 4, 1974 date for final AIP mailing.

B. The remainder of the meeting was concerned primarily with the actual design details of the POP test unit. Agreement was reached on exact interface line of AMCO test unit to LMEC facility. Other information forwarded to AMCO was:

1. Sodium level would be determined via a hand held probe method if leakage was not too rapid as to make accurate measurement difficult.

E/

After given a general idea of expected leak rate of 0.3 cc/min and frequency of checks, LMEC believed the method was acceptable. AMCO will provide (2) one inch sch. 40 thimbles through the sodium pot cover to a required depth for measuring the sodium level in the pot and the lip seal leakage and LMEC will provide the probe. LMEC added measurement took approximately 30 seconds and the sodium level change should be small in this time period.

2. LMEC will check the sodium temperature by measuring from the outside of the tank. LMEC stated this would give an accurate reading within $\pm 10^{\circ}\text{F}$. AMCO agreed that it would probably be accurate enough. With immersion probe could get to $\pm 5^{\circ}\text{F}$.
3. Mr. Wagner, (LMEC), stated carbon steel material would be acceptable in the sodium bath, noting that it must be designed for lower stress allowables and that it must be at least sandblasted clean. He noted the stress analysis would be difficult since ASME coding did not give any information for carbon steel at temperatures over 800°F . Mr. Smith mentioned AMCO desire to use carbon steel for cost and procurement reasons. LMEC suggested that cost savings realized in difference between stainless and carbon steels may be offset by larger sizes required due to lower strength. This did not eliminate the lead time of the stainless, however. (LMEC will want stress analysis on all parts subjected to high pressure 225 psia.) -- Mentioned again in #6 w/o pressure.

4. Concerning the cover design of the sodium pot, the temperature would be a prime concern. Since sealing was enacted via a silicon "O"-ring, the temperature must be less than 200°F. This was previously accomplished by fabricating baffles to reduce radiative heat flux from the bath to the cap. Also, LMEC would like (2) two 1/4 inch NPT plugged holes in the cap for auxiliary access (venting system, etc.). If any additional thermocouples were required, they would be supplied by AMCO. LMEC prefers the chrome L - alumel type. Lifting lugs or eye bolts must be provided on the pot cover for handling. Any bolting in the pot must be lockwired or techweld to prevent intrusion into LMEC loop. Threads not exposed to sodium may have a coat of anti-galling substance (neolube or molykote).
5. Discussion pertaining to purity of sodium was discussed. LMEC wanted to know the justification for the purity level of 5 ppm oxygen as specified in the AIP. Mr. DeWall (AMCO) stressed that prime objective was to simulate actual LMFBR conditions; LMEC agreed this was no problem as long as sodium was continuously flowing in the pot during the test.
6. LMEC did not require material certifications but suggested AMCO should have for our own purposes. To which we noted affirmatively. LMEC mentioned they may desire review of some stress calculations for areas of the test unit which might directly affect the safety of their loop, to which AMCO agreed. AMCO noted a hydrostatic test with water would be conducted prior to sodium testing. This test

6. (continued)

would be at a pressure greater than the sodium test to compensate for stress allowable differences, due to the test being at 965°F and the water hydrotest at room temperature.

7. Concerning the ferrofluidics stem seal, AMCO must check fluid for compatibility with sodium. Also a thermal study should be made to ensure stem temperature does not exceed seal allowable due to conduction from the sodium. LMEC has cooling water available if required, but it would add to test costs.
8. LMEC would provide gas bottles for pressurization for leak test. AMCO should design an expansion loop into the leak-pressurization tube to the valve.
9. For stem torque test, in lieu of strain gage method proposed by AMCO, LMEC suggested simpler in-line torquemeter such as manufactured by LeBow.
10. LMEC would like the following drawings:
 - o Instr. and Elect. Schematic
 - o Test article mechanical layout including materials selected.
 - o Test article outline drawing showing all mech. elect., and instrument interfaces, locations, rigid dimensions.

LMEC to review and concur officially the preliminary and final documents.

LMEC confirmed they would provide all information concerning tank size to AMCO by 7 November 1974 (telecon) with formal submittal approximately two days later. This information would allow AMCO to complete design of test unit.

11. LMEC suggested that AMCO remove the birdcage for the sodium test.

12. Mr. Bud Conners outlined a sequence of events. he felt should be accomplished prior to testing the component at LMEC, although AMCO had provided a similar list in the AIP Figure 1.

- o Preliminary AIP for comment
- o LMEC comments to AMCO
- o Final AIP to AEC & LMEC
- o LMEC response to AEC
- o AEC approval and funding authorization
- o AMCO submit T.R. to AEC & LMEC
- o LMEC initiate report for T.P.
- o AEC approval T.R.
- o LMEC accepts T.R.
- o AMCO submit preliminary interface DOL for LMEC review

A short tour of LMEC facilities was given by Mr. Johnson, for AMCO personnel.

INTER-OFFICE MEMO

TO: J. E. DeKlotz

FROM: R. J. Glumace

SUBJECT: Review of Thermal Transient Test Article Body Drawing 16-E-4019

DISTRIBUTION: L. R. Bird, H. Derow, J. L. Kortenhoeven, M. L. Nelson,
T. O'Callaghan, V. A. Smith, S. R. Speer

References:

1. H. Derow, Memo to those listed, approximately 11 January 1974, not dated.
2. J. C. Torneo (AMCO), "AMCO Proposal FA-73S897D for the Continuation of the LMFBR Large Valve Development Program," Letter to W. E. Sweeney, (ARD), December 19, 1973.
3. W. E. Sweeney, "Agreements and Commitments 12/11/73 and 12/12/73 Meeting," FPV-912 to J. C. Torneo, December 28, 1973, (CAR 60).
4. W. E. Sweeney, "Preliminary Design Requirements," FPV-890, to J. C. Torneo, December 5, 1973, (CAR 59).

Reference 1 requested a review in detail of the conceptual design of the valve body flange test article, Drawing 16-E-4019. The following (1) summarizes the requirements pertinent to the body/cover/seal weld design and (2) provides a review of the design assuming this to be a portion of milestone 15 of the Work Plan (Reference 2). It is understood that the information on the drawing is only sufficient for a structural evaluation and that added analytical and test system input is required prior to completion of the design.

A. Requirements for Review of "Thermal Transient Test Article Body," Drawing 16-E-4019

The review is based on Work Scope submitted by Reference 2, Task III.

1. Basic Requirements:

- (a) Design the test article which consists of the maintenance opening flange/cover configuration for the spherical body defined in Task III (body, bolted flange assembly and canopy seal).
- (b) A structural analysis will be conducted to determine flange configuration.
- (c) Body forgings shall be suitable for subsequent sodium testing per agreements of December 12, 1973 meeting (Reference 3).

74:6460:18
15 January 1974

Page 2

- (d) Purchase of forgings is to consider upgrading to RDT Standards to the maximum practical extent (Reference 3).

2. Design Requirements

Following are pertinent requirements from Task II submitted by Reference 4.

Value size (in)	12*
Design Temperature (°F)	965
Design Pressure (psig)	300
Material	304SS
Design Life (year)	30
End Connection	Sch 30
RDT Standard	RDT E1-18T
Design Flow (lb/hr)	4.3X10 ⁶
Normal Operating Temperature (°F)	936
Normal Operating Pressure (psig)	132
Duty Cycles (transients)	TBD

*From 14 inches by Reference 2.

3. Additional Requirements

- (a) The valve design is shown on Drawing 16-E-4010, dated 11 January 1974.
- (b) The valve body is made from forging Drawing 16-E-4018.

B. Review of Design

The following are general comments since no back-up is submitted for the design (see Ala and Alb above) nor is the drawing complete.

1. An analysis and/or evaluation must be performed to back up design (Ala, Alb, Alc, above).
2. Materials, including weld electrodes and rods, should have chemical analysis by supplier and internal (AMCO) analysis (overcheck) to meet requirements (1c and 1d above).
3. Dimensions and tolerances are missing throughout. As this is a detail/weldment drawing, dimensions before and after welding must be shown. Consider code requirements for weld alignment.

4. Unless the test intends to use different necks a specific neck length should be selected.
5. The access diameter must permit the removal of internals since this test article could be reworked and used for subsequent sodium testing.
6. Following are comments on flanges/seal joint.
 - (a) The bolt circle should be designed to a minimum diameter. The ID of seal should clear the body by .062 inch nominal. The bolt should be as close to final seal weld as practical, considering possible distortion of threads when welding seal to body. Accommodations for threaded inserts (repair purposes) should be provided in the body flange.
 - (b) The bolt material has a different coefficient of thermal expansion than the body and cover. Will another thermal compensator device (sleeve) be used? It is noted that FFTF primary loop valves use 316SS bolts.
 - (c) The number of bolts should be shown.
 - (d) The seal design may be changed per AMCO Report No. 1773-01 (02)ER (to be issued). The flange design should change accordingly. A seal drawing should be prepared and sufficient material ordered to qualify seal welding.
 - (e) The drain hole should be .25 inch diameter. It can serve the following purposes: back-up gas for welding, provide draining, assure sodium behind seal (heat transfer), and release trapped gas when loading the valve with sodium. Perhaps locating the hole on the top would be better since chips would not fall into hole when removing seal weld, and gas would not be trapped when loading the valve with sodium. Draining the cavity may not be as important.
 - (f) Torsion and horizontal shear loads cannot be transmitted into the canopy seal. Provisions for these loads must be made.
 - (g) Bolt torques and sequence of installation should be defined.
7. The configuration of the head (rather flat) may cause problems. Hand calculations comparing flat and elliptical heads might be done prior to detail analysis.

74:6460:18
15 January 1974

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8. The interface with the test system should be considered.
9. Weld callout and weld preparations should be shown.
10. Holes penetrating the body (e. g. drain) will need subsequent plugging for sodium testing.

R. J. Glumace

R. J. Glumace
Design Development

RJG:mk

INTER-OFFICE MEMO

APPENDIX E3

TO: M. L. Nelson

DATE: 20 March 1974
74:6700:023

FROM: W. D. Gaw

SUBJECT: Proposed Weld and Processing Sequence for 12 Inch Offset Ball Valve Components, D/N 16-E-4020

DISTRIBUTION: H Bray, H Derow, S D Driscoll (2), V Smith, File

1. Per the meeting of 3/18/74 with H. Bray and H. Derow in attendance, the following general welding and assembly sequence is suggested:

1.1 Outer Housing Fabrication; no X-ray and no stress relief by agreement:

- (1) Weld rough machined flanges to cones: VW-1 and VW-2; use EB insert (308) and manual GTAW with 308L filler wire (carbon steel flange to 304 SS cone). Visual inspect each pass and PT OD of completed weld per NB5350 and rework as required.
- (2) Machine 16-E-4029-6 (cyl.) to length and machine weld preparations. Pi tape 16-E-4029-6 (cyl.) at both ends and determine av. ID. Machine OD of 16-E-4029-3 (ring) and weld ledge of 16-E-4029-2 (ring) to fit average ID of 16-E-4029-6 (cyl.). Pi tape 16-E-4029-7 (cyl.) and determine average ID. Machine bell hole in 16-E-4029-6 (cyl.) to average of ID of 16-E-4029-7 (cyl.).
- (3) Weld 16-E-4029-3 (ring) to inlet cone: VW-3; use manual GTAW with option of manual GMAW (M-2 gas), 308L filler wire (carbon steel ring to 304 SS flange). Visual inspect completed weld and rework as required.
- (4) Weld 16-E-4029-3 (ring) to 16-E-4029-6 (cyl.); VW-4; use automatic sub-arc, L-40 wire and L-80 flux, 150°F preheat, MT inspect completed weld per NB-5340.
- (5) Machine inlet cone and 16-E-4029-3 for sub-housing (detail 2) interface and final machine flange face and ID of inlet cone.
- (6) Weld 16-E-4029-2 (ring) to 16-E-4029-6 (cyl.); VW-5; use automatic sub-arc, L-40 wire, and L-80 flux, 150°F preheat, MT inspect completed weld.
- (7) Weld 16-E-4029-7 (cyl.) to 16-E-4029-8 (flange) using E-7024 electrode. MT inspect completed weld. Machine and drill and tap 16-E-4029-8 (flange) after this weld is completed.

20 March 1974

74:6700:023

L. Nelson

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- (8) Layout in weld lab, flamecut and trim 16-E-4029-7 cyl. to fit 16-E-4029-6 (cyl.) bell hole with 3/32-1/8 gap. Locate 16-E-4029-7 (cyl.) to 16-E-4029-6 (cyl.) tack, and weld complete using E 7016/E 7018 electrode. Back grind ID as required. MT and weld ID. MT inspect completed weld.

1.2 Weld Assembly of Offset Ball:

- (1) Assemble shafts to bores in rough machined ball by shrink fitting and apply VW-11 and VW-12 fillet welds; use manual GTAW root pass and 308L wire. Complete weld using E-308-16 electrode. Grind finished welds smooth and PT inspect per NB-5350.
- (2) Locate ball/shaft weldment on weld positioner with groove for stellite facing up. Centralize, insulate, and attach to weld positioner faceplate. Preheat to 800°F/1000°F and hold 800°F/1000°F IPT. Apply stellite 6 rod by manual GTAW process per VW-10. Immediately after stellite application is completed, place ball in slope controlled S/R furnace at 1000°F. Increase furnace temperature to 1200°F and hold for 8 hours. Slow cool at 20°F/hr to about 300°F, then open furnace and final slow cool in asbestos blankets.

2. The above plan, particularly 1.1 sequence alters the discussed sequence of 3/18 for practical reasons. If further information is required, please contact the writer.

W. D. Gaw

INTER-OFFICE MEMO

TO: D. Pangburn
FROM: C.F. Reimers
SUBJECT: Preliminary Stress Evaluation of Sodium Test Lip Seal
DISTRIBUTION: M. Chen, J. DeKlotz, L. DeWall, D. Knittle, T. O'Callaghan, J. Perschler, V.A. Smith

DATE: 17 February 1975
CFR:pc
75:6460:027

A. SUMMARY

This initial report presents some results of the preliminary stress analysis of the sodium test lip seal pertaining to:

- (1) Evaluation of assembly interference of the Lip Seal and D - Seal.
- (2) Pressure differential loading effects on the assembly and the seal weldment considering relative thermal expansions of the Lip Seal and Sub-Housing.
- (3) Finite Element analysis of the Lip Seal using the AMCO/AB5U axisymmetric computer program.

Some minor modifications were made to the sodium test configuration as a result of the preliminary evaluation (Items 1 and 2 above) which are discussed below.

B. DESCRIPTION OF LIP SEAL

The Lip Seal being developed for sodium test checkout (Figure 1) is identical to the original water test article in the area of contact and flexible conical ring section. On the present configuration, however, the conical ring section extends on into a heavy cylindrical section that is clamped into the Sub-Housing on assembly and a secondary lip seal (D-seal) is integrally attached to the cylinder to seal the back side. Inconel 718 material is being used for all seal components (Lip Seal, D-Seal and connector pins).

C. MODEL DESCRIPTION

The finite element model of the lip seal extends half way into the integral support cylinder. This model is shown in Figure 2. Five elements were used through the thickness for a total of 311 elements 381 nodes. Axial support was provided at the base tangency point (node 280 at a radius of 4.800 inches). This was found to be the actual bearing point for most load cases as evidenced by the computed deflections. Preload displacements were applied through a quadrilateral ring element aligned normal to the ball at the contact point. This element was relatively rigid in the axial-radial

planes but flexible in the hoop direction. Pressure loading was applied from the contact point, on around the tip of the lip seal and extending the full length of the lip seal and cylinder. The reaction pressure of the D-Seal was applied as a uniform tensile pressure on the upper cross section of the cylinder (nodes 376 to 381).

Material properties used:

Modulus of Elasticity:

$$E = 25 \times 10^6 \text{ psi}$$

Poisson's Ratio:

$$\nu = 0.3$$

Relative Thermal Contraction:

$$\alpha = (7.9 - 10.2) 10^{-6} = -2.3 \cdot 10^{-6} \text{ in/in/}^{\circ}\text{F}$$

(Inconel 718 relative to 304 S/S)

The six load cases imposed were:

- (1) Design pressure = 300 psi
- (2) Operating pressure = 225 psi, Axial preload deflection = .0125 inches
- (3) Same as (2) plus temperature differential = $936^{\circ}\text{F} - 70^{\circ}\text{F}$
- (4) Same as (3) considering axisymmetric friction (i.e. coefficient of friction at the ball to lip seal contact = 0.5 acting radially outward tangent to the ball)
- (5) Lip Seal unsupported by ball, operating pressure = 225 psi
- (6) Same as (5) plus temperature differential = $936^{\circ}\text{F} - 70^{\circ}\text{F}$

Surface stresses were computed by running the AB5U code immediately followed by the UP8U isostress plot program which calculates extrapolated surface stresses.

D. RESULTS OF COMPUTER ANALYSIS

The results to date from the finite element analysis has shown that the flexibility and stress responses of the sodium test lip seal essentially match those of the optimized water test configuration. There appears to be some margin remaining for improvement in flexibility of the Inconel 718 Seal if warranted by leakage or tolerance considerations. The present stress results are also within range for A286 alloy which might be considered for a back up material.

Maximum stresses for the six load cases are presented in Table 1. Typical deflection and isostress plots are shown in Figures 3 to 6 for the operating pressure and temperature load case (with ball axial preload displacement of .0125 inches).

The room temperature preload force (P) produced by an axial ball displacement of .0125 inches can be determined by reference to the ball contact loads for cases (1) and (2), (see Table 1).

$$P = R_{(2)} - R_{(1)} \quad \frac{225 \text{ psi}}{300 \text{ psi}}$$

$$P = 3162 - 2456 \frac{225}{300} = \underline{1320 \text{ lb.}}$$

It is evident from Table 1 that the ball contact increases significantly (from 3162 to 4028 lb) at operating temperature due to the contraction of the Inconel 718 relative to the 304 S/S Sub-Housing. Also axisymmetric friction at the ball contact can further increase the contact load to 4741 lb., (see Case 3) assuming a friction coefficient of 0.5 acting to resist the inward relative contraction of the Lip Seal.

E. IMPROVEMENTS TO LIP SEAL DESIGN AS RESULT OF EVALUATION

Analyses completed so far have led to seal design improvements including:

- (1) The base of the Lip Seal was thickened by adding a taper between the Lip Seal cone and the cylinder. The intent was to develop an effective cantilever support for the sodium test Lip Seal representative of that provided by the clamping ring in the water test article. This preserves the original optimized seal configuration as was subsequently proven in the finite element analyses.
- (2) The outer ring on the secondary Lip Seal (D-Seal) was extended to insure a uniform interference fit when pressed into the Lip Seal cylinder. The preliminary analysis indicated that the original stub-ring would ride the cylinder edge resulting in an undesirable rotation of the D-Seal and a tendency for the D-Seal to back out when axial clamping forces were released.
- (3) Eight connector pins spaced uniformly around the circumference were added to the D-Seal ring to cylinder assembly to provide a positive connection between the two components. During heat up to operating temperatures a relative contraction of .004 inches in the 2 inch long Inconel cylinder occurs relative to the 304 S/S Sub-Housing. The pins were added when it was determined that the interference fit friction forces were insufficient to prevent the two seals acting as separate pistons from overstraining the seal weld during pressure cycling.

F. PLANNED WORK SCOPE

Structural analyses that remain to be completed include:

- (a) Finite element analysis of D-Seal
- (b) Non-symmetric analysis of Lip Seal for eccentric friction loads using AB3U (local distortion effects of the (8) pin attachment on the sealing surface can also be investigated)

- (c) Buckling analysis of unsupported (no Ball contact) Seal using the BOSOR program.
- (d) Creep-relaxation of the Lip Seal preload using the CREEP-PLAST code for the extended sodium test load condition.

C.F. Reimers
Analytical Development & Support

Enclosures:

- 1. Table 1. Maximum Lip Seal Stresses and Reaction Loads.
- 2. Figure 1. Sodium Test Lip Seal Layout.
- 3. Figure 2. Finite Element Computer Model of Lip Seal.
- 4. Figure 3. Deflection Plot For Operating Pressure and Temperature.
(Load Case 3)
- 5. Figure 4. Tangential Stress for Operating Pressure and Temperature.
(Load Case 3)
- 6. Figure 5. Maximum Principal Stress for Operating Pressure and
Temperature. (Load Case 3)
- 7. Figure 6. Minimum Principal Stress for Operating Pressure and
Temperature. (Load Case 3)

Table 1. Maximum Lip Seal Stresses and Reaction Loads.

Case	Element/Radius (in)		Stress ^① (psi)	Allowable ^② (psi)	Ball Contact Load (lb)
(1) Design pressure (300 psi) Run No. 2/1/75 (19:02:18)	2 81	4.047 4.303	$\sigma_e = 14280$ $\sigma_b = 42910$	$1.5S_o = 80400$	2456
(2) Operating Pressure (225 psi) Ball Preload = .0125" Run No. 1/28/75 (21:25:55)	2 87	4.047 4.313	$\sigma_e = 48730$ $\sigma_b = 51930$	$Sq = 105700$	3162
(3) Same as (2) plus $\Delta T = 936^\circ - 70^\circ F$ Run No. 1/29/75 (21:56:33)	2 87	4.047 4.313	$\sigma_e = 73650$ $\sigma_b = 65900$	$Sq = 105700$	4028
(4) Same as (3) with Ball-Seal Friction Coefficient = 0.5 Run No. 2/1/75 (20:36:28)	2 91	4.047 4.338	$\sigma_e = 75550$ $\sigma_b = 66290$	$Sq = 105700$	4741
(5) Operating Pressure (225 psi) No Ball Contact Run No. 2/1/75 (20:06:20)	2 192	4.047 4.681	$\sigma_e = -42310$ $\sigma_b = 35740$	③	$\delta_{\perp}^0 = -.0130''$
(6) Same as (5) plus $\Delta T = 936^\circ - 70^\circ F$ Run No. 2/1/75 (20:16:22)	2 192	4.047 4.681	$\sigma_e = -42310$ $\sigma_b = 35740$	③	$\delta_{\perp}^0 = -.0190''$

① Local hoop stresses are designated as σ_e , meridional bending stresses as σ_b . (Both represent stress intensities).

② Allowable stresses listed for Aged Inconel 718 are from AMCO Report 1773-01(09)ER, Task II-IHTS Isolation Valve Internals Summary Report, June 1974, Appendix D.

③ Represents buckling condition. Allowable stresses to be determined using the BOSOR shell code.

Figure 1. Sodium Test Lip Seal Layout.

Figure 2. Finite Element Computer Model Of Lip Seal.

Figure 3. Deflection Plot For Operating Pressure and Temperature.
(Load Case 3)

CONTOUR VALUES OF
H30P STRESS
(KSI)

- 1 = 5
- 2 = 10
- 3 = 15
- 4 = 20
- 5 = 25
- 6 = 30
- 7 = 35
- 8 = 40
- 9 = 45
- 10 = 50
- 11 = 55
- 12 = 60
- 13 = 65
- 14 = 70
- 15 = 75
- 16 = 80
- 17 = 85
- 18 = 90
- 19 = 95

$$M_A \times \sigma = 73650$$

Figure 4. Tangential Stress For Operating Pressure And Temperature.
(Load Case 3)

CONTOUR VALUES OF
MAX. PRINC. STRESS
(KSI)

1 = 5
2 = 10
3 = 15
4 = 20
5 = 25
6 = 30
7 = 35
8 = 40
9 = 45
10 = 50
11 = 55
12 = 60
13 = 65
14 = 70
15 = 75
16 = 80
17 = 85
18 = 90
19 = 95

Figure 5. Maximum Principal Stress For Operating Pressure And Temperature.
(Load Case 3)

Figure 6. Minimum Principal Stress For Operating Pressure and Temperature.
(Load Case 3)

TO: J. L. Kortenhoeven

DATE: 4 June 1974

74:6460:116

FROM: R. J. Glumace

SUBJECT: Upgrading of Offset Ball Valve Material for Subsequent Sodium
POP Test (Task 2)

DISTRIBUTION: J E DeKlotz, H Derow, M Nelson, T O'Callaghan, V A Smith, Files

SUMMARY

The purpose of this memorandum is to establish the testing required to upgrade materials of the AMCO offset ball valve water test article. The materials for the ball, stems (journal bearings), sub-housing, and the lip seal require upgrading, to the intent of the applicable RDT material standards, since they may be used for subsequent sodium proof-of-principal (POP) testing.

A review of material procurement specifications and other available information resulted in a recommendation of the following additional material tests:

- Check analysis
- Tensile tests
- Radiographic tests
- Corrosion tests (304 stainless steel only)

INTRODUCTION

The materials which required upgrading to the applicable RDT Standards were agreed to by AMCO and WARD. Upgrading of the stainless steel weld rod and the hardsurfacing rod materials was not considered. Manufacturing processes upgrading following material procurement (e.g., heat treatment, welding, machining) was not considered.

The information used for establishing the recommended tests was the following:

- Material procurement specifications
- Valve detail drawings
- Vendor-supplied data
- Commercial material specification
- Applicable RDT Standards
- AMCO Quality Assurance records.
- Manufacturing status

DISCUSSION

For materials which require upgrading, Table 1 identifies the part, the material, the applicable ASME Boiler and Pressure Vessel Code or Aerospace Material Specification (AMS), and the applicable RDT material standard. Table 2 identifies the testing and/or documentation required by the applicable RDT Standards.

TABLE 1

MATERIALS TO BE UPGRADED

Item No.	Part No.	Desc.	Mat'l.	Mat'l. Spec.	Applic. RDT Standard
1	16-E-4021-1	Ball	304H	SA-182(1)	M2-2T
2	16-E-4021-2	Shaft, Idler	304L	SA-479	M7-3T
3	16-E-4021-3	Shaft, Drive	304L	SA-479	M7-3T
4	16-E-4022	Sub-housing	304H	SA-182	M2-2T
5	16-E-4023	Lip Seal	Inconel 718	AMS5562C(2)	RDT M2-15T(3)

(1) "SA" is ASME Boiler and Pressure Vessel Code Section II
Material Specification.

(2) "AMS" is Aerospace Material Specification.

(3) Material is similar to M2-15T material. Test requirements
should be identical.

TABLE 2

REQUIREMENT PER RDT STANDARDS	Requirements Performed ⁽¹⁾				
	Item No.				
	1	2	3	4	5
		(4)	(4)		
Ladle Analysis	Yes	Yes	Yes	Yes	Yes
Check Analysis	No	No	No	No	Yes ⁽²⁾
Tensile Test	Yes	Yes	Yes	Yes	Yes
Corrosion Test	Yes	No	No	Yes	N/R ⁽³⁾
Heat Treat Certifications	Yes	No	No	Yes	Yes
Ultrasonic Examination	No	No	No	No	No
Hardness Tests	Yes	Yes	Yes	Yes	Yes
Forging Drawing	No	N/R	N/R	No	No
Qualification Verification Program (F2-4T)	No	No	No	No	No

(1) Required by Purchase Order and/or submitted by supplier.

(2) Not submitted by the supplier

(3) N/R indicates "Not Required."

(4) Q/C paperwork misplaced. Performance based on Purchase Order Requirements.

4 June 1974
74:6460:116

J. L. Kortenhoeven

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Since vendors were not qualified to the RDT Qualification Standard (RDT F2-4T) nor was all testing performed as required by the RDT Standards, the following testing is recommended for component materials (items numbers refer to Table 1).

a) Ball and Sub-Housing Material (Items 1 and 4)

These are listed together since they are from the same heat of material.

- o Each part shall be radiographed to the extent possible as limited by the part configuration.
- o A tensile test shall be performed using material from the same heat as the parts (e.g., 16-E-4029 -4 and -5 cone material).
- o A check analysis shall be performed from the broken tensile test specimen.
- o A corrosion test shall be performed per ASTM A262, Practice B, for detecting the susceptibility to intergranular attack.

b) Shafts Material (Items 2 and 3)

These are listed together since they are manufactured from the same bar and heat.

- o Since these are welded to the ball forming an assembly, they shall be radiographed to the extent possible in conjunction with a) above.
- o A tensile test shall be performed using material from the same heat (preferably the same bar).
- o A check analysis shall be performed from the broken test specimen.
- o A corrosion test shall be performed per ASTM A262, Practice B, for detecting the susceptibility to intergranular corrosion.

c) Lip Seal Material (Item 5)

- o The seal shall be radiographed.
- o A check analysis shall be performed from a broken test specimen, if it is available, or using forged material from the same lot.

J. L. Kortenhoeven

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Following are additional comments on the materials requiring upgrading:

- * ◦ The Quality Assurance paperwork for the shafts (Items 2 and 3) has been misplaced (available when upgrading was previously investigated). It is desirable to have this paperwork for record purposes.
- The lip seal forgings (Item 5) were received annealed as specified on the face of the lip seal drawing. Tensile specimens were not ordered which could be used for verifying the tensile property of the forgings after final heat treatment by AMCO. Tensile specimens may be made from the part forging or a spare forging. Material for tensile specimens should be included when heat treatments are performed by AMCO.

R. J. Glumace

R. J. Glumace
Design Engineering

* Q/A paperwork located in PFTF files 7-26-74
Changed to sodium valve files (J. Levin)
R. J. Glumace 7-29-74

APPENDIX F

LMEC AND RRD CORRESPONDENCE AND ASSOCIATED REFERENCES

APPENDIX F

LMEC AND RRD CORRESPONDENCE AND ASSOCIATED REFERENCES

TABLE OF CONTENTS

- F1 ERDA SAN Letter dtd 23 January 1975, "AMCO Large Valve Development; Request for Approval in Principle of 12-Inch POP Test; RRD Approval."
- F2 LMEC Letter No. 74LMEC-DRF-8177-AMCO 12.0 Inch Rotating Offset Ball Valve Static Test at LMEC-Scoping Cost Estimates, dated 11 December 1974.

RECEIVED JANUARY 27, 1975

UNITED STATES
ENERGY RESEARCH AND DEVELOPMENT ADMINISTRATION
WASHINGTON, D.C. 20545

JAN 23 1975

APPENDIX F I

R. D. Thorne, Manager, SAN

AMCO LARGE VALVE DEVELOPMENT; REQUEST FOR APPROVAL IN PRINCIPLE
OF 12 INCH POP TEST; RRD APPROVAL

- Ref: 1) AMCO ltr 74:6460:18, Harbison to Cunningham, "Request for Approval in Principle of POP Test of AMCO 12 Inch Rotating Offset Ball Valve Internals," dated 11/4/74
- 2) AMCO ltr 74:6460:19, Harbison to Cunningham, "Long Lead Material Procurement," dated 11/4/74
- 3) LMEC ltr 74LMEC-DRF-8177, Cochran to Campbell, "AMCO 12 Inch Rotating Offset Ball Valve Static Test at LMEC - Scoping Cost Estimate," dated 12/11/74

Reference 1) forwarded an AMCO request for Approval in Principle (AIP) of a static in-sodium test of a 12 inch offset rotating ball valve at LMEC. By reference 2), AMCO recommended procurement of minimum long lead materials which are required for fabrication of the 12 inch valve test article. By reference 3), LMEC provided a cost estimate and confirmed schedule for the proposed test, and requested that funding be made available for commencement of work in the second quarter of FY 1976.

RRD has reviewed the reference documents and approves the AMCO AIP request, subject to the following comments:

1. Section 1 - Thermal Transients - Confirming telephone discussions between RRD (Nulton) and AMCO (O'Callaghan, DeKlotz) on January 7, 1975, AMCO should consider a thermal shock test of the 12 inch test article following the Proof of Principle (POP) testing described in reference 1). This test could be conducted in gas in the LMEC Thermal Transient Facility (TTF), which is scheduled to be available in September of 1976.

Within two weeks of receipt of this memorandum, AMCO should provide RRD with a recommendation regarding thermal shock testing of the POP test article, and indicate the extent and cost of modifications to the test article to accommodate such a test. AMCO should also indicate in this submittal the date by which a separate or revised AIP will be submitted for the additional testing. For the purpose of considering the thermal transient tests, and for initial scoping, the number and description of thermal transients should be in accordance with the enclosure to this memorandum (CRBR IHX E Specification E953032, Appendix B). These transients include those provided informally to AMCO (DeKlotz) by RRD (Nulton), in November of 1974. Final determination of the exact shape and number of transients to be imposed would be made based upon discussion among RRD, AMCO and GE prior to actual testing at LMEC.

JAN 23 1975

2. Section 2 - "Ferrometic" Shaft Seal - RRD agrees with AMCO that the ferrometic seal shows a number of potential benefits over stem seals currently in use. However, RRD believes that prior to the application of such a seal to the POP test article, AMCO should study the problems associated with chemical compatibility of sodium vapor with the ferrofluid, with the maximum temperature and pressure at which the seal can operate, with the type and configuration of cooling system required to limit the seal to its maximum operating temperature, with assuring that the ferrofluid will not leak into the system sodium, and with the consequences of such a leak if it should occur. In view of these potential problem areas, AMCO should design a freeze seal for the test article in parallel with their efforts on the "ferrometic" seal. Prior to manufacture and assembly of the seal, AMCO should provide RRD with the results of their studies on the ferrometic seal, and a recommendation regarding which seal should be used on the test article.
3. Section 4 - Mechanical Cycles - The number of open-close cycles to which the POP test article should be subjected should substantially exceed the number of cycles to be experienced in actual plant applications. Seat leakage tests should be performed at various stages in the mechanical cycling tests and the results compared to previous leak performance. This approach will provide valuable wear and seat leakage data for subsequent valve design and future valve applications.

With regard to reference 2), RRD approves the AMCO request to proceed with the procurement of long lead material which is required for the POP test article manufacture.

RRD also concurs with the LMEC cost estimate and schedule (reference 3) for performing the proposed POP test. LMEC should be authorized to proceed with the work necessary to prepare for the proposed test. The necessary funds for LMEC services should be requested in the FY 1976 LMEC budget submission.

If there are any questions regarding the content of this memorandum, please contact Mr. D. Nulton of the TC staff or Mr. G. Hanlon of the SP staff.

G. W. Cunningham, Assistant Director
for Engineering & Technology, RRD

RRD:TC:005

cc: Director, SCRPO
Director, PMIC, SAN
Director, EM, SAN
J. C. Cochran, VP, LMEC
Director, Construction Programs, LMEC
J. DeKlotz, Supervisor of Des. Dev., AMCO, w/encl.

Liquid Metal Engineering Center

Operated for
the U.S. Atomic Energy Commission by *Atomics International Division*
Rockwell International Corporation

December 11, 1974

In reply refer
74LMEC-DRF-8177

Mr. D. B. Campbell, Director
So. California Reactor Program Office
U. S. Atomic Energy Commission
P. O. Box 1446
Canoga Park, California 91304

Subject: AMCO 12-Inch Rotating Offset Ball Valve Static Test at
LMEC - Scoping Cost Estimate

- References: (1) AMCO letter, "Draft AIP, AMCO 12-Inch Rotating
Offset Ball Valve POP Test," dated October 23, 1974
(DRF-7400)
- (2) AMCO letter, "Request for Approval in Principle,
POP Test of AMCO 12-Inch Offset Ball Valve
Internals," dated November 4, 1974 (DRF-7599)
- (3) AMCO letter, "Long Lead Procurements," dated
November 4, 1974 (DRF-7765)
- (4) 74LMEC-DRF-7619, "LMEC Testing of AMCO 12"
Valve," dated November 7, 1974

Dear Mr. Campbell:

By Reference (2), Aerojet Manufacturing Company (AMCO) submitted a Request for Approval in Principle (AIP) to perform static in-sodium testing of a 12-inch rotating offset ball valve at LMEC. Review of the AIP by LMEC has been accomplished and based upon the proposed test article delivery and test schedule, testing could be accomplished in our Rig 032-R2-T5. All objectives outlined in the AIP can be satisfied with only minimal changes to the test rig.

The estimates for the test were based upon the following assumptions:

1. Testing to be per Reference (2).
2. Schedule
 - a) Test article delivery Feb. 1976
 - b) Test completion Oct. 1976
 - c) Test report Dec. 1976
3. Testing in Rig 032-R2-T5.
4. Receiving inspection at LMEC for identity and damage only.
5. The test article will directly interface with 032-R2-T5.
6. Instrumentation for monitoring valve performance will be installed by AMCO on the test article prior to test article delivery.
7. Instrumentation requirements to be nominal.
8. Argon and water will be required by the test article.
9. Disassembled test article will be returned to AMCO upon completion of post test visual inspection.

\$33,200 will be required for engineering services in support of the proposed testing. This amount will be required in 189 No. 13913, Component Testing Support for FY 1976. In addition, included in the appropriate operating budgets will be costs for the manpower, materials and services required to receive and inspect, handle the test article, operate Rig 032-R2-T5 during test setup and testing, disassembly, cleaning and visual inspection subsequent to test and preparation for return shipment to AMCO.

LMEC will require funding for the proposed test commencing in the second quarter of GFY 1976 and such costs will be included in the FY 1976 proposed 189a's.

Please contact Mr. J. T. Nagamatsu of my staff should you have any further questions.

THIS IS SUBMITTED FOR INFORMATION.

Sincerely yours,

J. C. Cochran
Vice President
Liquid Metal Engineering Center

cc: Director, RRD, HQ
Asst. Director, LMFBR Support Facilities, RRD, HQ
Chief, Engineering & Planning Br., RRD, HQ
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