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APOLLO SERVICE PROPULSION SYSTEM ROCKET ENGINE
BIPROPELLANT VALVE IMPROVEMENT PROGRAM

Phase I Report

Prepared Under
Contract NAS 9-8317
for
MANNED SPACECRAFT CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
Houston, Texas

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Liquid Rocket Division
Sacramento, California

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I. INTRODUCTION

This Phase I report is submitted in accordance with Exhibit D of Appendix II of Contract NAS 9-8317. In addition to reporting all work accomplished on the completed valve, it includes recommendations for work to be undertaken in the event further development becomes desirable.

The objective of the program was to improve the Apollo Service Propulsion System Rocket Engine bipropellant valve. The scope of work included design studies and subcomponent level development tests to support the primary objectives of the program.

The program had a duration of 18 months (July 1968 through December 1969); see Figure 1. Program work was administered under the Product Improvement and New Applications Section of the Aerojet Apollo Program Department. The program manager was C. E. Teague.

II. SUMMARY

Phase I of the Apollo Service Propulsion System (SPS) Rocket Engine Bipropellant Valve Improvement Program (VIP) started during the first week of July 1968, upon receipt of contractual authority to proceed.

Related experience was reviewed and tradeoff studies were conducted to establish the basic valve concept. The selected concept retained the basic SPS dual propellant passages with redundant dual seal ball valves, but departed significantly from the SPS in other design areas: (1) the valve assembly was completely modular, i.e., individual interchangeable ball and seal assemblies, housings, actuators, and actuation systems; (2) the ball seals were lifted free of the ball in the first few degrees of motion to reduce the wiping action; (3) the drive mechanism incorporated idler gears to ensure that the bores of fuel and oxidizer balls were parallel to the housing bores when in the full

II, Summary (cont.)

open position; and (4) the return spring of the actuators was positioned external to the actuation cavity. The pneumatic actuation control was retained, although it was recognized that an electrical system would offer some advantages.

Following the conceptual design review held at NASA-MSC in early November 1968, the drafting of the detailed engineering drawings was started. First priority was assigned to the valve cartridge, i.e., the module which contains one ball, lift-off cams, and the upstream and downstream seals. As soon as the drawing status permitted a good approximation of the final configuration, a test assembly composed of obsolete SPS components was fabricated to test the feasibility of the concept. The results of these tests showed excellent seal life and unsatisfactory cam cycle life. Consequently, the design was modified to increase the cam bearing width.

At this point, it was decided to depart from the original program plan which had been published at the end of the first program month. This plan called for the Phase I design review and subsequent fabrication of the optimized valve immediately upon completion of the tests conducted with the obsolete SPS hardware. An assessment of the test results indicated that this course of action would be impractical because it had not been possible to maintain the close tolerances required to adequately evaluate the cam and follower assembly. Another unknown was the bellows used in both the upstream and downstream cage assemblies. Consequently, since the cartridge assembly was definitely the most important subassembly and the only one which extended the existing technology, the expenditure of additional effort upon component level tests with preprototype cartridge assemblies before committing to the complete valve assembly was considered advisable.

In consonance with this policy, components were procured in sufficient quantity to allow testing of several configurations in parallel. Included were

II, Summary (cont.)

two types of bellows assemblies, two seal materials and four combinations of cams and cam followers. During the three-month procurement cycle, laboratory tests of candidate cam and follower materials were performed to provide additional backup.

Cartridge testing started during the latter part of May 1969. The initial buildup was completed as planned and the acceptance tests, including proof, leak and pressure relief, were satisfied. However, dry cycle tests showed an actuation torque increase characteristic of cam wear. Consequently, the cartridge was disassembled and the cams lubricated with a thin Teflon film to assist in the initial wear-in. The cartridge was then subjected to 20,000 actuation cycles with the oxidizer (N_2O_4) followed by 5,000 dry cycles. Cam wear characteristics were excellent and seal leakage rates were reduced significantly over those previously experienced with the rubbing seals of the SPS valve although still not meeting the design goal of 10 cc/hr. Following the N_2O_4 tests, a fuel cartridge was assembled for the purpose of testing in the reverse order, i.e., dry before wet. The results of these tests demonstrated the cams to be completely satisfactory, although the leakage characteristics were not as good as those demonstrated during the previous tests. This disparity was probably due to the slightly longer rubbing contact of the fuel seals due to the oxidizer lead.

Following the dry cycle test, flow characteristic tests were conducted to determine transient flow characteristics and water hammer. Results were satisfactory.

Assembly of the two-bore valve started in September 1969. This assembly shown in Figure 27 consisted of one fuel bore and one oxidizer bore, i.e., one half of a prototype design.

II, Summary (cont.)

Testing of the completed valve commenced in November 1969 and terminated in January 1970. Although problems prevented completion of the intended test plan sufficient data were obtained to evaluate the two bore valve configuration.

Throughout the testing, the No. 2 and 4 ball seals of both the fuel and oxidizer valves indicated satisfactory sealing characteristics. These positions are the lifted seal concept with axial displacement absorbed by a convoluted bellows. The No. 1 and No. 3 ball seals, however, had unacceptable and erratic leakage rates during most of the testing. The No. 1 position is of the lift configuration with axial movement through a toroidal bellows. Problems inherent in the fabrication of a toroidal bellows resulted in dimensional changes from the original concept. This may have influenced sealing traits of No. 1 ball seals. The No. 3 seal is a bending type with no axial displacement. There is probably some seal deformation from bending which results in loss of adequate sealing characteristics.

The cams and followers in the oxidizer valve performed satisfactorily during testing. Although there were indications of minor wear patterns occurring in the center portion of the parts, the hardware is considered adequate for its intent. Considerable problems, however, were encountered with the fuel cams and followers. Immediate loss of lubricant (S-122 Teflon Spray) on friction surfaces, upon AeroZINE 50 exposure, resulted in severe galling and abrasion.

The actuators, drive gears, and shaft seal components functioned adequately throughout the program. Minor problems occurred during actuator assembly but they are not considered as having a critical effect on the design.

Although some minor problems are yet to be solved, the basic conclusion is that the valve developed under this contract demonstrated a definite

II, Summary (cont.)

potential for improvement over the existing SPS valve design with respect to leakage, cycle life, and assembly procedures. Further, a cycle life of 20,000 and a maximum leakage rate of approximately 50 cc/hr could be realized with little additional development effort.

III. TECHNICAL DISCUSSION

A. TASKS

The various tasks accomplished are discussed in the same order as that in which they appear in the Program Plan (Report 8317-P) with the addition of the two-bore valve testing.

1. Review of Previous Experience

A comprehensive review of related experience within Aerojet was initiated in the second contract week. The purpose of this review was to identify past problems so that appropriate counter measures could be instituted early in the program.

Although this review considered several valves and valve concepts, the SPS propellant valve experience was the most applicable. Figure 2 tabulates the problems experienced with the SPS valve.

In addition to sources within Aerojet, technical discussions were held with two of the Apollo subcontractors which also utilize ball type propellant valves.

On 23 September 1968, a technical exchange meeting was held at Thompson Ramo Wooldridge (TRW Inc.), the manufacturer of the Lunar Descent Engine bipropellant shutoff valve. Much of the discussion involved TRW's experience with various materials and coatings in the N_2O_4 and A-50 environment. It was of interest to note that TRW used a 4 finish on the balls which was the same as that planned for this program. Another comment made by the TRW representatives was that they doubted the advantages of two seals per ball because of the added complexity. This viewpoint was shared by a number of persons at Aerojet-General and was one of the factors given considerable emphasis during the tradeoff studies.

III, A, Tasks (cont.)

Bell Aerosystems experience with the Lunar Module Ascent Engine bipropellant valve was reviewed at Niagara Falls, New York, 18 October 1968. During the discussions, it developed that the major problems encountered had been in the actuator. When asked about flaking or wear of the main Teflon ball seals, it was reported that this had not been a problem since early in the development program. In discussing ball surface finishes, the Bell representative stated that they used a 4 finish with the requirement of a mirror polish. They had found that a "satin" finish on a 4 surface gave much higher breakaway and running torques due to friction between the balls and seals. A review of the various design features also showed that although two seals are used on each ball, the design concept was quite different from that being considered for the VIP Program, because the ball was allowed to float between the two teflon seals. The main seal is the downstream seal which is fixed in place. The upstream seal is not really a seal in that it does not seal between seal and the body - its primary purpose being to transmit load from Belleville springs through the ball and against the downstream seal. The review of previous experience was concluded at this point with the results applied to the tradeoff studies.

2. Tradeoff Studies

Tradeoff studies were also initiated in the second contract week. The purpose of these studies was to select the optimum design from the various design concepts. As a part of this study, formal review meetings were held at Aerojet which were attended by personnel from Engineering, Test, Quality Control, and Apollo Departments which had specialized applicable knowledge and experience.

The first of these reviews was held during the second program month and resulted in the establishment of design ground rules which would serve to narrow the field of potential designs to receive detailed studies. These were:

III, A, Tasks (cont.)

a. The improved valve should be physically and functionally interchangeable with the existing valve.

b. Weight was important but not an overriding factor.

c. The valve should have improved reliability and fabricability. A design goal was to reduce fabrication costs to one-third that of the existing valve.

d. The complete development of an electrical actuation system was beyond the scope of the present contract. However, consideration should be given to tailoring the design such that electrical actuators could be used on the new valve without requiring changes to the valve assemblies other than replacing the pneumatic actuators.

e. An improved SPS pneumatic actuation system should be used.

f. The existing engine and injector interfaces should not be changed except as required to avoid severely compromising the bipropellant valve design.

g. The optimized bipropellant valve should be functionally interchangeable with the present SPS valve.

h. The valve would use cam lifted ball seals to minimize seal wear. The crank lifted seal design would be a backup configuration, and interchangeability of the two designs would be considered.

III, A, Tasks (cont.)

i. Cartridge modules, installed in quadmonopropellant valves would be used. The cartridge would contain, as a minimum, a ball and two seals, with the capability of leak testing the seals as a subassembly. The cartridges, if not alike, would be noninterchangeable.

j. The valve housing would permit a cartridge to be installed without requiring disassembly of another cartridge. It was also a design goal to permit replacement of a valve cartridge without disassembly of the valve halves, or inboard shaft seals.

k. The actuator housing would be separate from either valve half. The valve halves would be identical, except that the fuel and oxidizer valve-to-actuator interface would be noninterchangeable. The interchangeability between electrical and pneumatic actuation systems would be a consideration in the design of the actuator housing.

l. It was a design goal to provide adjustable valve end position stops.

m. Bipropellant valve assembly handling provisions should be incorporated in the design.

Once the basic ground rules had been established, work on the following specific design concepts was started:

a. Actuation System

- (1) Electrical
- (2) Hydraulic
- (3) Pneumatic

III. A, Tasks (cont)

b. Propellant Valve Configuration

- (1) Ball
- (2) Poppet
 - (a) Cam visor
 - (b) Toggle Visor
 - (c) Axial
- (3) Blade

c. Redundancy

- (1) Quadmonopropellant valve module
- (2) Quadbipropellant valve module
- (3) Single monopropellant valve module
- (4) Cartridge module

d. Seal Configuration

- (1) Cam lift
- (2) Crank lift
- (3) Eccentric ball

The matrix listing the advantages and disadvantages of each of the concepts is included in Figure 3.

Upon completion of the tradeoff studies, another internal Aerojet-General design review was held to select the basic design. The results were a design with replaceable cartridges which included a ball, upstream and downstream cam actuated seals, shaft, and cage assembly. The actuator drive mechanism included idler gears which allowed the valve to be

III, A, Tasks (cont.)

stopped in the full open position and still provide oxidizer lead during initial opening. The actuator design had the spring outside of the pressurized cavity to eliminate spring generated contamination from affecting the actuator seals.

At the end of the fourth program month, a conceptual design review was held at NASA/MSD where the basic valve concept was presented. This meeting resulted in general agreement as to the basic design selection and allowed the start of the detail design effort. Figure 4 is an artist's rendering of the cartridge assembly presented at the design review.

3. Single-Position Valve Tests

The original purpose of the single-position tests was to support the design effort and provide a quick economical method of evaluating design concepts before they were incorporated in the final design. The first tests were to be conducted with surplus or expended hardware from the SPS Program and later tests were to be conducted with breadboard versions of VIP hardware.

The single-position tests were later expanded to include tests with prototype cartridge assemblies. This change was necessary because the cartridge assembly was completely different from the basic SPS design, and as a consequence, it was not possible to evaluate the design to the extent that it would be prudent to commit to the total valve with only limited testing with used hardware. The tests conducted with both types of hardware are summarized in the following paragraphs:

III, A, Tasks (cont.)

a. Single-Position Tests with SPS Hardware

The single-position testing was initiated in mid November 1968. The primary purpose of the first series of tests was to gain insight into the operation of the cam action. The secondary objective was to obtain preliminary data to verify that the reduced rubbing distance of the seal would produce a significant reduction in seal wear and improvement in leakage.

The hardware required for performing these tests consisted principally of the existing Apollo ball valve hardware with some minor modifications which were: (1) the ball was made narrower so that the ball and cams would fit into the existing cage; (2) the cage was shortened to accommodate the seal lift ring; and (3) the shaft was slotted to key the outboard cam to the shaft. New parts, such as the cams, followers, and the seal lifting mechanisms, were designed and procured specifically for these tests. The cam design was recognized as being one of the most challenging aspects of the new components because of the long cycle life required at high loads. Consequently, on the basis of experience obtained from the M-1 Program, the cams and followers were fabricated from type 17-4 stainless steel and hard coated with a flame dispersion hard coating known by the trade name of Colmonoy No. 6. The major constituents of the single position test assembly are shown in Figure 5. An engineering drawing of the assembled cage, with the cams and followers in place, is shown in Figure 6.

(1) Assembly Tests - Cams Coated with Colmonoy No. 6

Four assembly tests were performed to develop a procedure for assembling the hardware in such a manner that desired seal load and lift could be obtained. The results of these tests were:

III, A, Tasks (cont.)

(a) On the first attempt, regular wide Apollo valve balls were used to determine seal standoff and load. The objective was to reduce the chance of scratching the polished sealing surface of the two narrow balls fabricated especially for these tests. During the load tests, the wide balls contacted the bearing ring and bound so that the force gage was not receiving the full force of the seal on the ball. Therefore, to obtain accurate seal load data, the narrow balls designed for the test assemblies had to be used for this test and all subsequent tests.

(b) The seal lift could be predicted within 0.003 or 0.004 in. This variation was attributed to the slight waviness of the shim stock used to set the seal lift dimension.

(c) The cams and followers galled even at relatively low loads.

The galling between the cams and followers was attributed to the apparent softness of the flame spray coating applied to the bearing surfaces of the two parts. The galled areas are shown in Figure 7. These photographs show the parts magnified five times. The coating, Colmonoy No. 6, contains chromium boride and chromium carbide aggregates fixed in a nickel binder. The hardness of the coating was supposed to be R_c 56 to 61. However, the hardness measured on the parts which galled, measured R_c 40 to 59. Because of the apparent discrepancy in the hardness measured, one cam and one follower were sectioned so that the hardness at various depths through the coating could be measured at two locations. These hardness measurements are presented in the table below along with the equivalent compressive strength:

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III, A, Tasks (cont.)

<u>Part</u>	<u>Depth Below Surface, in.</u>	<u>Hardness Measured Rc</u>		<u>Compressive Strength, ksi</u>	
		<u>No. 1</u>	<u>No. 2</u>	<u>No. 1</u>	<u>No. 2</u>
Cam	0.002	51	47	253	225
	0.004	54	50	280	248
	0.006	48	54	230	280
	0.008	40	40	185	185
	0.010	27	20	Below coating	
Follower	0.002	49	59	240	330
	0.004	58	57	315	300
	0.006	58	53	315	268
	0.008	49	54	240	280
	0.010	35	28	Below coating	

Metallographic examination of the cam and the follower revealed that the surfaces underneath the plating in the area of failure had ridges about 0.001-in. high. These could possibly have acted as stress risers under bearing load at that location.

Three microphotographs of the cam and follower are presented for inspection. Figure 8a is 100-X magnification of cross-section samples of the coating in the area of the failure. Figure 8b shows the follower coating with a portion of the cam coating adhering to it. Figure 9 is a 500-X magnification of the coating at the surface of the cam. This photograph shows that the nickel binder is exposed at the surface of the coating. This binder, being softer than the boride and carbide particles, was probably causing the galling.

The Wall-Colmonoy Company was then contacted to determine what the problem with the coating might be and for recommendations for future hardware, since the stress analysis had indicated that the Hertz stress in the cams and followers would be 220,000 psi and the design minimum for the Colmonoy coating was 300,000 psi.

III, A, Tasks (cont.)

On 17 December 1968, the Los Angeles office of the Wall-Colmonoy Corporation agreed with the earlier Aerojet assessment that the coating on the cams and followers was soft. This softness was due to porosity caused by poor furnace fusion and to the thinness of the finished coat.

The main office of the Wall-Colmonoy Corporation in Detroit recommended the use of Colmonoy No. 72 coating for the VIP application on both the cams and followers. Cams and followers with this coating were then ordered.

In addition, as a result of recommendations of the Aerojet-General Materials Department, followers were ordered fabricated from Haynes Stellite 6B alloy as a backup. This alloy was reported to have an ultimate compressive strength of 347,000 psi.

While waiting for the arrival of the new cams and followers coated with the Colmonoy No. 72, a series of tests was conducted to determine the force versus deflection curves for the various parts of the cam lifting mechanism. This data was used in conjunction with the computerized stress analysis of the single-position cam configuration to determine the maximum cam loads to be used in holding the stress level to 185,000 psi, which was the stress predicted as the worst case in the cartridge design. From this data, it was calculated that the pressure on the seal assembly during cam lift-off should be held to a maximum of 65 psig to keep the stress in the cam from exceeding 185,000 psi.

(2) Assembly Tests - Cams Coated with Colmonoy No. 72

After receipt of the Colmonoy No. 72 coated cams and followers, tests were conducted to determine the coefficient of friction

III, A, Tasks (cont.)

between the cams and followers at various loads and stress levels. With the No. 72 coated follower on the No. 72 coated cam, the coefficient varied from 0.20 to 0.24 as the stress was varied from 0 to 185,000 psi. With the Stellite 6B followers on the No. 72 coated cams, the coefficient was essentially constant at 0.23 over the same range of stress.

After this series of tests was completed, assembly and limited cycling tests using both types of followers were conducted with the cams loaded to 185,000 psi. The results of these tests and the conclusions were:

(a) Both cams and followers coated with Colmonoy No. 72.

1 Coefficient of friction, = 0.24

2 No galling

3 Torque increase from 20 to 50-in.-lb after 50 dry cycles. Examination by wipe test of contacting surfaces after disassembly disclosed particulate accumulation sufficient to discolor tissue. Powder analyzed and found to be carbide aggregates which had been pulled from coating binder. Torque returned to original value after the parts were cleaned. During 50 more cycles, the torque again increased from 20 to 50 in.-lb. Disassembly revealed the same conditions.

4 Microscopic examination of cams showed hairline transverse cracks starting in coating.

III, A, Tasks (cont.)

(b) Cams coated with Colmonoy No. 72 and Haynes
Stellite 6B followers:

1 Coefficient of friction, = 0.23.

2 Torque increased from 20 to 45 in.-lb in first 50 cycles. After disassembly and clean-up, torque returned to 20 in.-lb. One hundred additional cycles were run and the torque increased to 42 in.-lb in about 60 cycles and then remained constant. Examination upon disassembly showed followers to be in good condition. Slight scratching on surface showed unequal load distribution.

(c) Conclusions:

Breakdown of coating was probably result of three factors:

1 Coating too thin. Actual was 0.010 in. and recommended is 0.015 to 0.020 in.

2 Unequal loading in test fixture.

3 Voids within coating.

On the basis of the foregoing results, it was decided that while the hard coating process was probably feasible, it was not desirable for this program because of the difficulty of maintaining the desired product control. Consequently, all subsequent effort was devoted to cam designs which did not require the use of coatings to obtain the desired surface hardness, with primary emphasis placed upon obtaining suitable cam material to use with a Haynes Stellite 6B follower.

III, A, Tasks (cont.)

In addition, it was decided at this time, to initiate dry seal testing with the available cams and followers with propellant testing to follow. The purpose of these tests was to evaluate the lift-off seal concept.

(3) Cycle Tests

Three different lipseals were tried in the test assembly before adequate sealing performance was achieved. The first seal leaked 30 cc in 3 min at 175 psig. The second seal leaked more than could be measured by the leak detector (the seal was damaged during assembly). After assembly of the third seal, leakage was reduced to 9 cc in 3 min at 175 psig, and it was decided to commence propellant testing with that unit. This decision was made in recognition of the worn condition of the test assembly and because the leakage was considered acceptable for development tests. Twenty-four hours later the actuator was installed and another leak check revealed that the leak rate had decreased to 2 cc in 3 min at 175 psig. The assembly was then baked for 8 hr at 300°F in an attempt to stress-relieve the lipseal and improve the sealing performance. The leakage did not decrease.

The assembly was then installed in the test bay, and five dry cycles were performed to check out the actuation system. Following the checkout cycles, another leak check was performed with the result that the seal leakage had increased to 32 cc in 3 min. This rate was in excess of that which could be tolerated and the valve was returned to the Controls Laboratory for disassembly and inspection.

Disassembly revealed that the seal was badly scratched in approximately six different areas on the sealing surface. A single metal flake was evident in one of the disfigurations, and the ball

III, A, Tasks (cont.)

was scratched in a corresponding location. A chemical analysis of the particles was made by spectroscopic methods. The results of the examination led to the conclusion that the contamination was introduced into the system externally and was not from the cams or followers.

In subsequent rebuilding the single-position test assembly for the N_2O_4 testing, a total of nine buildups were made before leak-tight seals were obtained. The primary cause of difficulty is the installation of the shaft through the ball with the small Woodruff keys required to hold the cams. This difficulty did not occur in the prototype cartridge design since the shaft is integral with the ball. Once a good assembly was made, the unit went through 10,000 N_2O_4 cycles with no apparent wear on the seal. The test results are shown in Figure 10.

Disassembly after the test revealed that the cam followers had flat spots worn on them that reduced the total seal lift from 0.006 in. at buildup to 0.003 in. after the 10,000 cycles. This wear is believed to be primarily caused by the galling and spalling of the Colmonoy No. 72 coated cams. The lack of wear on the seal surface after 10,000 cycles proved that the theory of lifting the seal away from the ball provided a greatly extended seal life which was the objective of the test series.

b. Oxidizer Cartridge Tests

(1) Assembly

Sufficient hardware was received 21 May 1969, to allow the start of the single position cartridge testing. The initial work consisted of checking out the assembly shop aids and the assembly procedures.

III, A, Tasks (cont.)

The detail parts for a cartridge are shown in Figure 11. The first cartridge assembly was prepared 27 May 1969. The assembled cartridge is shown in Figures 12 and 13. This first assembly was made without cleaning the parts to Level 1 and was intended as an assembly procedure checkout. The unit was operated for one cycle with blueing on the cams to determine if full width contact was occurring between the cams and the followers. It appeared that the contact was good but it was not certain since the blueing smeared to some extent. The unit was then disassembled and all of the parts cleaned to Level 1.

The unit was reassembled 28 May 1969, on the laminar flow bench. It was assembled without the seals and with a light load on the cams for a 1500 cycle run-in of the cams and followers. Figure 14 shows the assembled cartridge with the cam and followers in the proper position through the inspection holes.

(2) Dry Cycle

Cycling tests were then conducted with the cams and followers dry, with aircraft polish on the parts, and with the parts lubricated with FS-1281 and S-122. Cycling with the dry, unlubricated cams and followers resulted in noisy, squeaky operation, a rapid increase in shaft torque, the generation of black deposits, and scratched hardware. A very slight scratching or galling resulted in a significant increase in torque. The aircraft polish was used in an attempt to improve the finish on the cams and followers, but the polish dried out rapidly and permitted scratching of the parts. Cams and followers lubricated with FS-1281 operated smoothly with no noise and low torque. After 500 cycles, operation was still smooth with a moderate cam breakaway torque increase (from 43 to 55 in.-lb).

III, A, Tasks (cont.)

Because FS-1281 tends to wash out in propellant and thick coatings will decompose and flake off, the FS-1281 was replaced by S-122 spray lubricant for all subsequent tests using Stellite 6B and Stellite 19 cams and followers.

(3) Ball Seals

While dry cycling the cartridge assembly to evaluate the cam follower characteristics, a parallel effort was being conducted with the ball seals.

The downstream ball seals were all undersized as received from the supplier. The seal spring (Figure 15) is positioned in the seal so that it does not support the Teflon portion of the seal in any way. Therefore, the natural tendency of the Teflon is to shrink away from the spring. Several seals were temperature cycled on a mandrel in an attempt to resize the seal ID; however, the seals started to shrink immediately after being removed from the mandrel and within a day had returned to approximately their original size. As a permanent solution, a metal ring was designed and fabricated to be installed in the seal ID and became an integral part of the seal. The seal ID is machined to accept the ring, taking into account the amount the seal is undersized. Then, the seal is heated, the ring is cooled, and the two parts are assembled together. Figure 16 shows the ring clearly.

Problems concerning the upstream seal, PN 1156438, were out-of-round condition of the seating surface and low gage dimensions. Seals that were round within 0.002 in. were selected for use and the seats burnished with a minimum removal of material. In addition, the supplier was asked to investigate various machining methods which would result in a more round seal. The low gage dimensions, while to print as received, usually fell out of print after seat burnishing. The gage dimension is used

III, A, Tasks (cont.)

in the calculation of the seal shims in the cartridge; therefore, the shimming compensates for the low gage dimensions and no problem due to this condition has been experienced in cartridge assembly. To allow more rework, it was decided that future seal orders should specify a larger gage dimension to allow more material for burnishing.

(4) Propellant Cycling

The oxidizer cartridge was assembled with Stellite 19 cams and Stellite 6B followers, both lubricated with FS-1281. Special tooling was fabricated to facilitate assembly. The tools are shown in Figure 17. Wet cycle testing was initiated in the Components Evaluation Laboratory on 12 June 1969. Torque checks and leakage checks were conducted after 64 and 150 cycles. After 150 cycles, the No. 1 ball seal had excessive leakage. The valve was disassembled; particles of white contamination were found throughout the valve, the ball was scratched in the area of the No. 1 seal, and a metallic particle was embedded in the No. 2 seal. The white particles were decomposed silicone grease from the excessive use of FS-1281 during buildup. The ball was scratched by the No. 1 seal lift-off ring contacting the ball. The metallic particle appeared to be from the welded bellows assembly. These problems were resolved by machining 0.010 in. from the lift-off ring ID, by substituting S-122 lubricant for FS-1281 on the cams and followers and eliminating its use on Raco seals, and by supercleaning the bellows assembly.

The oxidizer valve was reassembled using new ball seals, S-122 cam and follower lubricant, the remachined lift-off ring, and the supercleaned bellows assembly. The wet cycling test was restarted from zero cycles. The torque and leakage results are tabulated in Figures 18 and 19. After 1500 cycles, the shaft torque increased from a previous high of

III, A, Tasks (cont.)

60 in.-lb to 100 in.-lb in 500 cycles. The cartridge was disassembled to the extent required to remove the cams and followers to determine the reason for the relatively high torque increase. The cams and followers were in good condition with only light scuff marks on the wear surfaces. The cams and followers were lightly polished on the lapping tools, sprayed with S-122, and reassembled into the valve. No other part was altered or changed.

Testing was resumed with cycle 1501 and continued through 20,000 wet cycles. Following the conclusion of the wet cycle testing, the cartridge was disassembled to permit examination of the cams. The cams were in good condition.

The dry cycle cartridge was then reassembled with no additional lubricant and testing resumed with 5,000 dry cycles. The test results are shown in Figures 20 and 21 and the ball showing the sealing surface after test is shown in Figures 22 and 23.

c. Fuel Cartridge Tests

A fuel cartridge was also assembled and tested. The purpose of the tests was to demonstrate that dry cycling could be accomplished prior to wet cycling as had occurred with the N_2O_4 assembly and also to evaluate the effect of the increased wiping action associated with the fuel cam (9° vs 6° for the oxidizer cam). The results of these tests are shown in Figures 24 and 25. It will be noted that the results were not as good as those obtained with the oxidizer cartridge but were still significantly better than had been possible with a nonlifting ball seal.

Following dry cycling, water flow tests including Kw and seal bite were conducted. No anomalies were disclosed.

III, A, Tasks (cont.)

Cartridge testing was terminated at this point because the test results had demonstrated conclusively the improved potential of the seal lift-off/cartridge assembly and readiness to commit to full-scale valve testing.

4. Optimized Design

This task consisted of the design and drafting effort associated with the actual making of engineering drawings/specifications, etc. A total of 70 engineering drawings were completed and are listed in Figure 26. Also included in this task category, were the friction and wear tests of various cam and follower materials which were conducted in the materials laboratory to back up the single-position tests. The test setup could not duplicate the cam and follower conditions closely enough to give quantitative data on cycle life but it did provide comparative data to indicate which material combination should be the best. The combinations tested and their comparative ratings are as follows:

CAM	<u>Specimen Materials</u>		<u>Rating</u>	<u>Remarks</u>
		<u>Follower</u>		
Colmonoy 72		Stellite 6B	Comparison Base	Produced black powder same as actual cam tests
Stellite 6B		Tungsten Carbide ⁽¹⁾	Fair	---
Stellite 6B		Stellite 19	Fair	---
Stellite 6B		Aluminum Oxide ⁽²⁾ (99.5%)	Fair	---
Stellite 6B		Aluminum Oxide ⁽³⁾ (99.9%)	Better	---
Stellite 19		Tungsten Carbide	Superior	---

(1) Carboloy 320, a cemented carbide (General Electric Company)

(2) Wearbox 995, a cemented aluminum oxide (Western Gold & Platinum Company)

(3) Carboloy 0-30, a cemented aluminum oxide (General Electric Company)

III, A, Tasks (cont.)

The tests showed that the best results would be expected using cams of Stellite 19 and a follower with tungsten carbide contact surfaces bonded in place. Rework of existing Stellite 6B followers to braze tungsten carbide inserts to the tips was completed; however, these followers were never tested because of the success experienced with the Stellite 19 cam and Stellite 6B follower in the cartridge tests.

5. Test Plan

Three test plans were prepared and submitted: one for the single-position tests with SPS hardware, one for the cartridge tests, and one for the two-bore valve assembly which is included in the Appendix.

6. Valve Timing Rig Studies

The original purpose of these studies was to investigate the feasibility of a device for timing the bipropellant valve actuation system which would provide a substantial reduction in the total number of valve ball cycles during buildup. Because of the excellent cycle life capability demonstrated by the cam lift seal, this device is no longer needed, consequently, no effort was expended on this task.

7. Design Guide

A draft copy of a design guide was submitted in June 1969. The comments of the NASA Technical Monitor were received in July 1969 and were incorporated.

III, A, Tasks (cont.)

8. Two-Bore Valve Tests

a. Assembly

The two-bore valve configuration is shown in Figure 27. During cartridge buildup it was noted that the fuel and oxidizer toroidal bellows, shown in Figure 28, exceeded design tolerances with respect to centering and parallelism to its cartridge housing half. To compensate for the centering error a portion of the seal flange inside diameter was increased. This allowed the ball seal and seal flange to be centered to the cartridge housing. It was assumed that the bellows flange would be parallel after the cartridge halves were assembled.

Pertinent ball seal assembly data is shown in Figure 29.

Actuator assembly was hindered by a dimensional error in one of the components of both assemblies. Schedule restraints did not allow investigation to determine which components were in error. The problem, however, was corrected by the addition of a shim between the piston and piston secondary seal.

Armalon thrust washers were added between opposing oxidizer and fuel drive gears. Washers were also placed on each side of the idler gears. Although not a print requirement it was deemed necessary in order to eliminate metal-to-metal contact on rotating surfaces.

b. Test

(1) Dry Cycle

Testing started in November 1969 with the initial acceptance test which was to be followed by 500 dry cycles. However, at the

III, A, Tasks (cont.)

end of 200 cycles, excessive oxidizer ball seal leakage was noted from positions number 1 and 3. Testing was terminated and it was concluded that either teflon generation or other contaminants were causing the problem. The valve was flushed with level 1 water until the particulate count was within level 1 acceptable limits. Particulate examination indicated teflon flakes and metal. Leakage rates after flushing were acceptable.

Testing was resumed; however, after a total of 500 cycles oxidizer seals, numbers 1 and 3, and fuel number 3 had unacceptable leakage rates.

The cartridges were removed and disassembled. Apparent cause of leakage was metallic contamination which scratched seals and balls, teflon generation, and rust contamination caused by inadequate dehydration after the water flushing. Rebuilding of the oxidizer half consisted of replacing all ball seals, removing scratches from the balls, and having the cartridge housing supercleaned. Fuel half rework involved replacing the numbers 1, 2, and 3 ball seals, removing scratches from the balls, supercleaning the cartridge housings, and re-lapping one upstream cam and its followers. The cam and followers were scratched and galled resulting from metallic contamination. Rebuild ball seal data is presented in Figure 30.

It was decided that, prior to resuming testing, the ball seals should be subjected to 500 wear-in cycles and simultaneously be flushed with level 1 alcohol to remove any contamination generated or inherent in the valve assembly.

At the completion of the wear-in cycles and alcohol flushing, excessive leakage from the fuel number 2 ball seal precipitated the removal of the fuel upstream cartridge assembly. It was then noted that the

III, A, Tasks (cont.)

alcohol had removed all the lubricant (Teflon Spray S-122) from the loading surfaces of the cams and followers. This resulted in replacement of one oxidizer upstream follower, re-lapping of fuel downstream cams and followers, and re-lubrication of all cams and followers.

The re-acceptance test was started and excessive leakage occurred from the fuel 1, 3, and 4 positions. Corrective action consisted in removing a sliver of metal from the number 4 seal and rotating, 180 degrees, one-half of the upstream cartridge housing. The cartridge housing was rotated because it appeared that there was non-uniform ball-to-seal contact. No apparent damage was noted on the number 3 seal.

Cartridges were installed and re-acceptance testing resumed. The 500 dry cycles test was repeated. Leakage rates and functional data are presented in Figures 31, 32, 33 and 34. Leakage rates were in excess of the design goal of 10 cc/hour, but were not of sufficient magnitude to warrant a termination of testing.

(2) Wet Cycle

Wet cycle testing started in December 1969. At the completion of the first 250 cycles the fuel half had excessive leakage from the numbers 1, 2, and 3 ball seals as shown in Figure 31. The fuel half was subsequently removed from the oxidizer valve for failure analysis.

The following conditions were noted upon teardown of the fuel valve: all cams and followers were galled. Galling occurred in the center portions of the loading surfaces. No damage was noted on the number 1 seal. One metallic contaminate was found on the number 2 seal. The number 3 seal had one minor scratch. Damage was not apparent on the number 4 seal.

III, A, Tasks (cont.)

The upstream fuel ball had a scratch corresponding to the number 2 seal contaminate. The lubricant had been removed from the loading surfaces of the cams and followers.

Rework of the fuel half consisted of the following: cams and followers were re-lapped or replaced as required. A Microseal 100-1 coating was applied to all cams and followers in order to reduce the coefficient of friction between cam and follower. Glass filled teflon ball seals were installed in the numbers 1 and 2 positions.

During the fuel valve rework an additional 500 wet cycles was conducted on the oxidizer half. Subsequent leak checks resulted in excessive leakage from the number 1 ball seal as shown in Figure 32.

The fuel valve was re-installed on the oxidizer half and testing resumed with an additional 500 wet cycles on oxidizer and 500 dry cycles on the fuel valve. Post cycle leak checks indicated excessive fuel leakage from the numbers 1, 2, and 3 ball seals. Number 1, 3, and 4 ball seals on the oxidizer half were also leaking excessively.

Further wet cycling was completed comprising a total of 1500 wet on the oxidizer half and 500 wet on the fuel half. Post cycle checks indicated excessive leakage past the fuel number 3 and 4 ball seals and also past the number 3 oxidizer seal.

Post cycle leakage, functional, timing, and seal back pressure relief data are shown in sequence in Figures 31, 32, 33, 34, and 35.

III, A, Tasks (cont.)

c. Posttest Teardown

The following conditions were observed during post wet cycle teardown and evaluation:

Fuel Cams and Followers - All parts showed signs of severe wear and galling in the centers as shown in Figure 36. There was no indication of lubricant on the loading surfaces. The Microseal was removed from the worn areas.

Oxidizer Cams and Followers - Minor wear similar to that shown in Figure 37 was noted on all parts. Some lubricant (very little) remained on loading surfaces. The minor wear was in the centers of the cams and followers.

Fuel Ball Seals

Number 1 - no visible damage

Number 2 - small piece of metal on sealing surface (probably from cams and followers)

Number 3 - minor scratches and small metal chips on sealing surface

Number 4 - minor scratches

Oxidizer Ball Seals

Number 1 - minor scratches

Number 2 - minor scratches

Number 3 - no apparent damage

Number 4 - three deep scratches on sealing surfaces.

Other pertinent ball seal data is presented in Figures 38 and 39. There was no visible indication of wear or damage on the actuator components, drive gears, shafts, or shaft seal components.

IV. CONCLUSIONS AND RECOMMENDATIONS

The basic conclusion of the effort is that the modular bipropellant valve assembly concept with cam lifted seals is definitely feasible. The concept of the modular cam lifted seal design is considered adequately demonstrated and any further effort would be of a design refinement category. The two-bore valve actuators and associated gearing performed satisfactorily during testing. Those areas which would require further development in order to meet all the design objectives are development of the fuel cams and cam followers to prevent galling in AeroZINE 50 and development of an improved upstream seal design and/or use of a filled Teflon seal material to minimize seal flaking.

The basic conclusion is substantiated by several subordinate conclusions in the following areas:

A. SEALING

1. The lifted seal concept results in a significant improvement in both life and sealing capability over the SPS rubbing seal design because of its reduced contact time and increased seating loads.

2. Axial seal displacement is preferable to the bending seal concept because the load on the cam is reduced and the seal is not deformed. Some permanent deformation and resultant loss in sealing characteristics probably occurs due to bending. This deformation is, possibly, indicated by the minimal change in seal face width corresponding to an increase in gage dimension (ref: Figures 30 and 39). The No. 3 seal, bending type, did not perform satisfactorily during two-bore valve testing. Test results demonstrated superior performance of downstream seals. Development of an improved upstream seal through the use of a redesigned bellows or filled Teflon seal material is one of the two primary outstanding design refinements. Filled Teflon seals that would be considered for evaluation would include calcium fluoride, boron nitride, carbon and graphite and borosilicate glass.

IV, A, Sealing (cont.)

3. Dual ball seals add to reliability, but also add complexity and cost. Tradeoff studies would be required for specific application before committing to either the single or redundant seal concept.

4. Additional effort is required to establish run-in procedures. Tests with both the oxidizer and fuel cartridges showed a reduction in leak rate after the initial cycling, leading to the conclusion that run-in cycles should be accomplished as a part of the assembly procedure prior to acceptance test. However, the run-in cycling accomplished on the two-bore valve assembly did not provide the desired results, indicating further development of the run-in procedures is necessary. Run-in cycle rate is critical. The leakage results obtained with the oxidizer cartridge assemblies when compared with those obtained with fuel cartridge indicate that the slightly longer rubbing interval, e.g., 6° oxidizer vs 11° fuel, results in a marked decrease in the number of cycles before Teflon builds up on the ball and excessive leakage is experienced. Based on the single position test results, the Aerojet-General Material Department recommended a reduced cycle rate from the 6 cycles/min to approximately 2 cycles/min to reduce the temperature buildup which is hypothesized as the cause of the premature failure of the oxidizer seals after only 2,000 dry cycles. Opening and closing rates remained unchanged at approximately 0.5 sec throughout the program.

5. Initial run-in of cams is critical and a light film of Teflon is beneficial while the cams and followers are seating in. The run-in methods used to date do not provide sufficient contact between the cams and cam followers and, in the case of fuel valve cartridges, this reduced contact area leads to excessive cam and cam follower galling during valve cycling with propellants. The combination of a Stellite-19 cam and Stellite-19 cam follower appears to be the most resistant to wear and galling during valve cycling. However, further investigation must be made into the feasibility of lapping

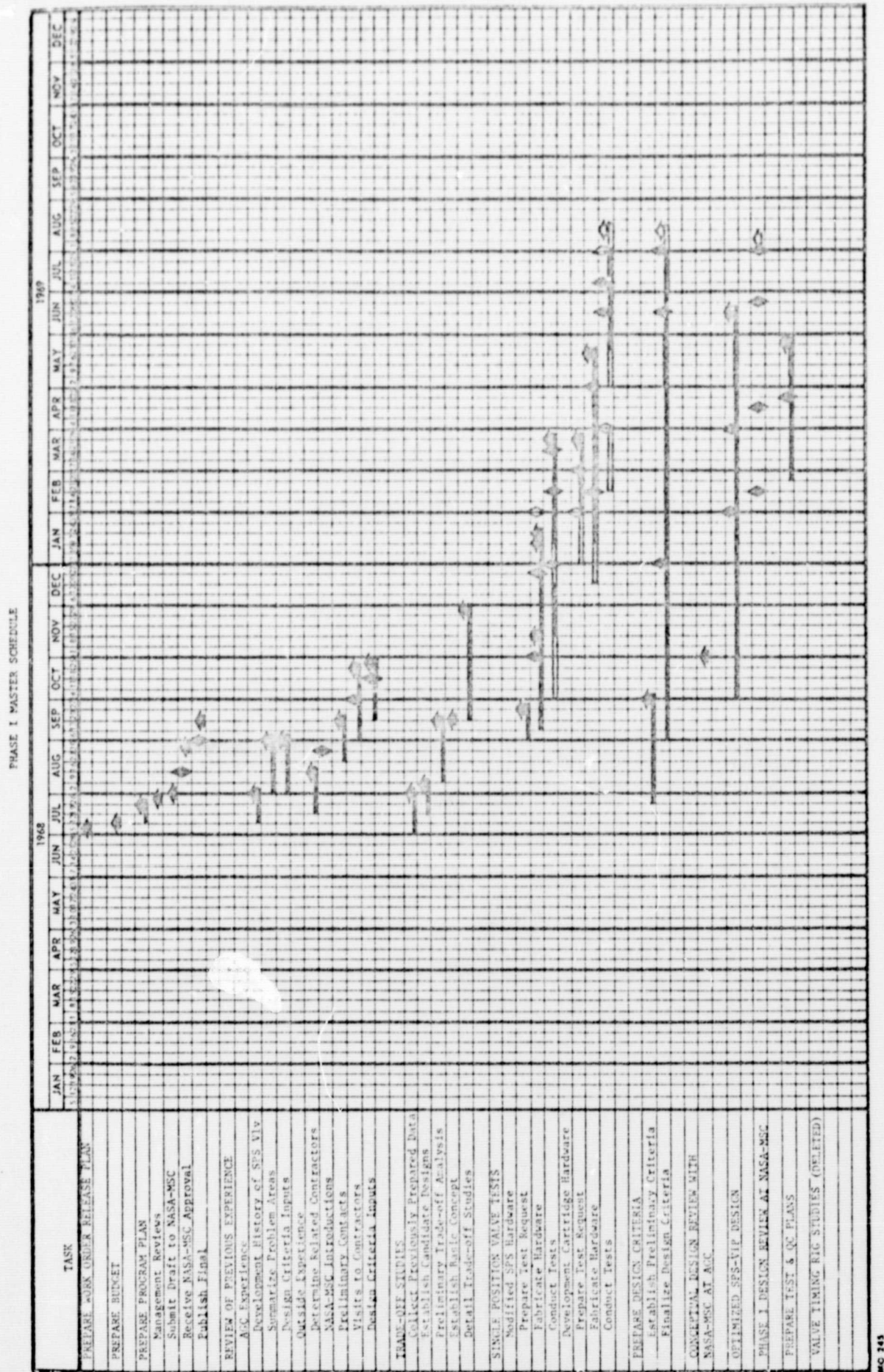
IV, A, Sealing (cont.)

the cam to the cam follower, resulting in matched set usage. Assuring adequate cam contact area is available and development of a lubricant or anti-galling material remains as a desired design refinement for AeroZINE 50 service.

B. ASSEMBLY

Assembly procedures must be refined. An excessive number of builds have been required prior to obtaining required leak rates. This is a typical development problem which can only be solved by increased familiarity with the hardware and the evolution of improved assembly methods and tooling.

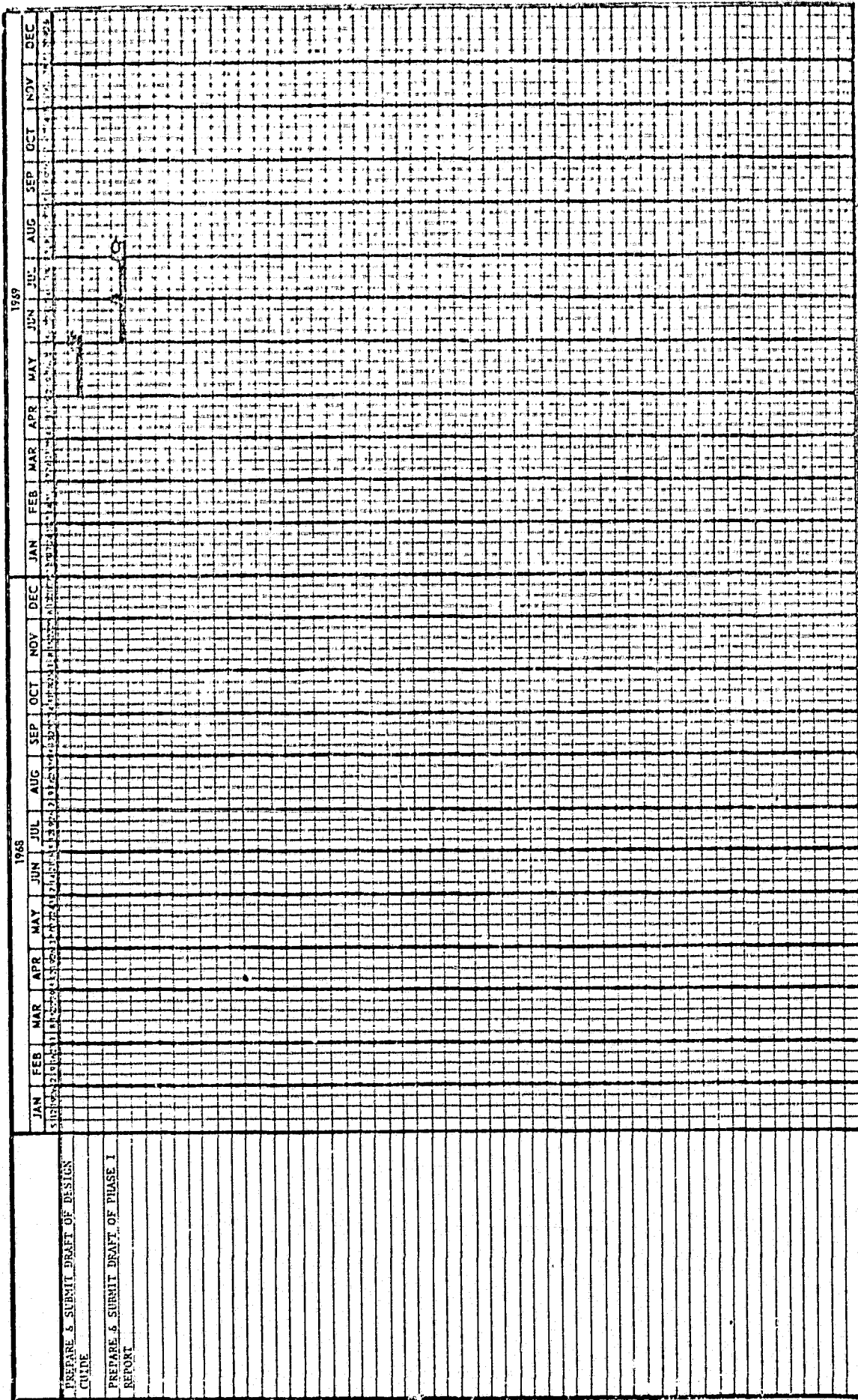




Master Schedule

Figure 1, Sheet 1 of 3

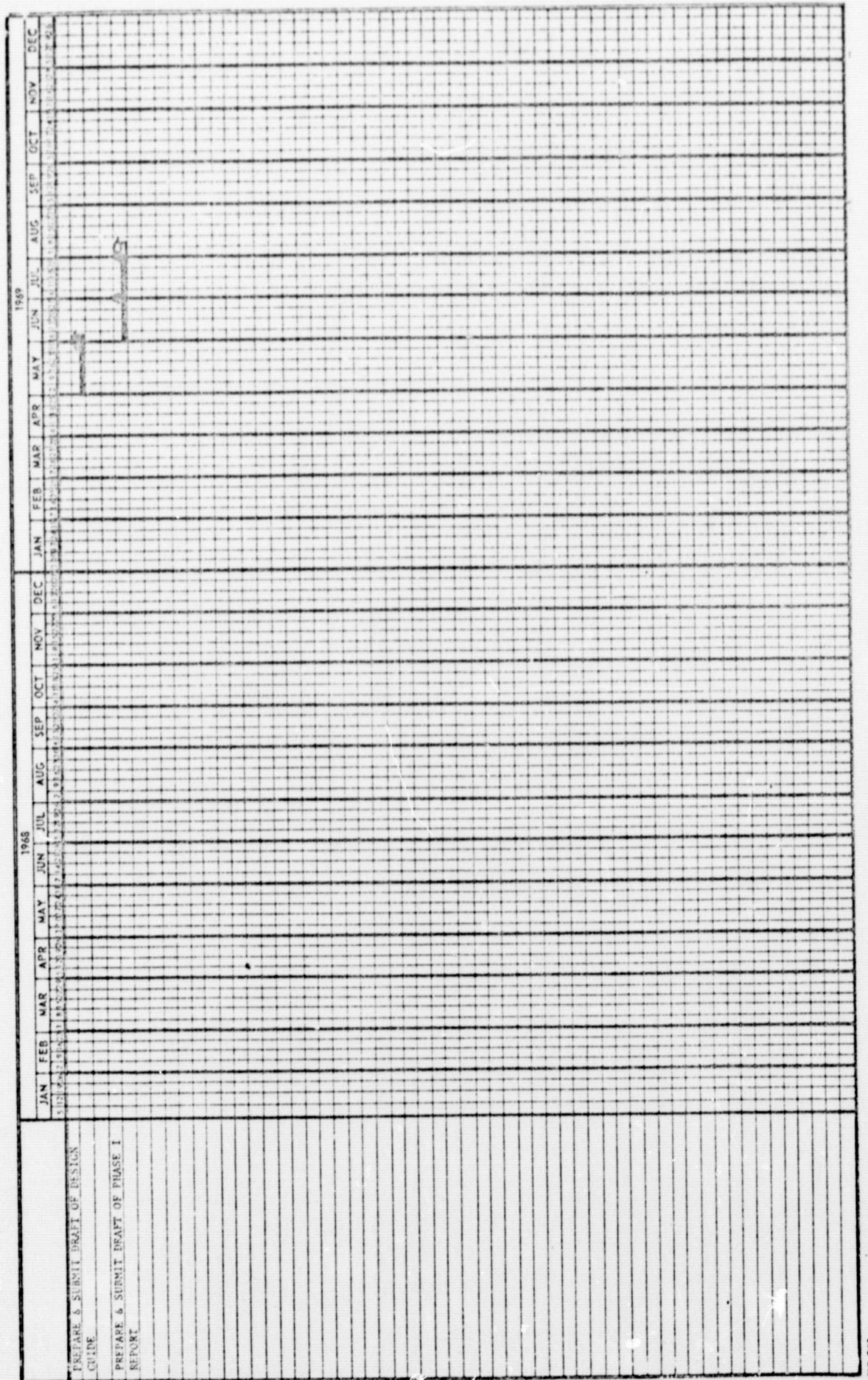
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Master Schedule

Figure 1, Sheet 2 of 3

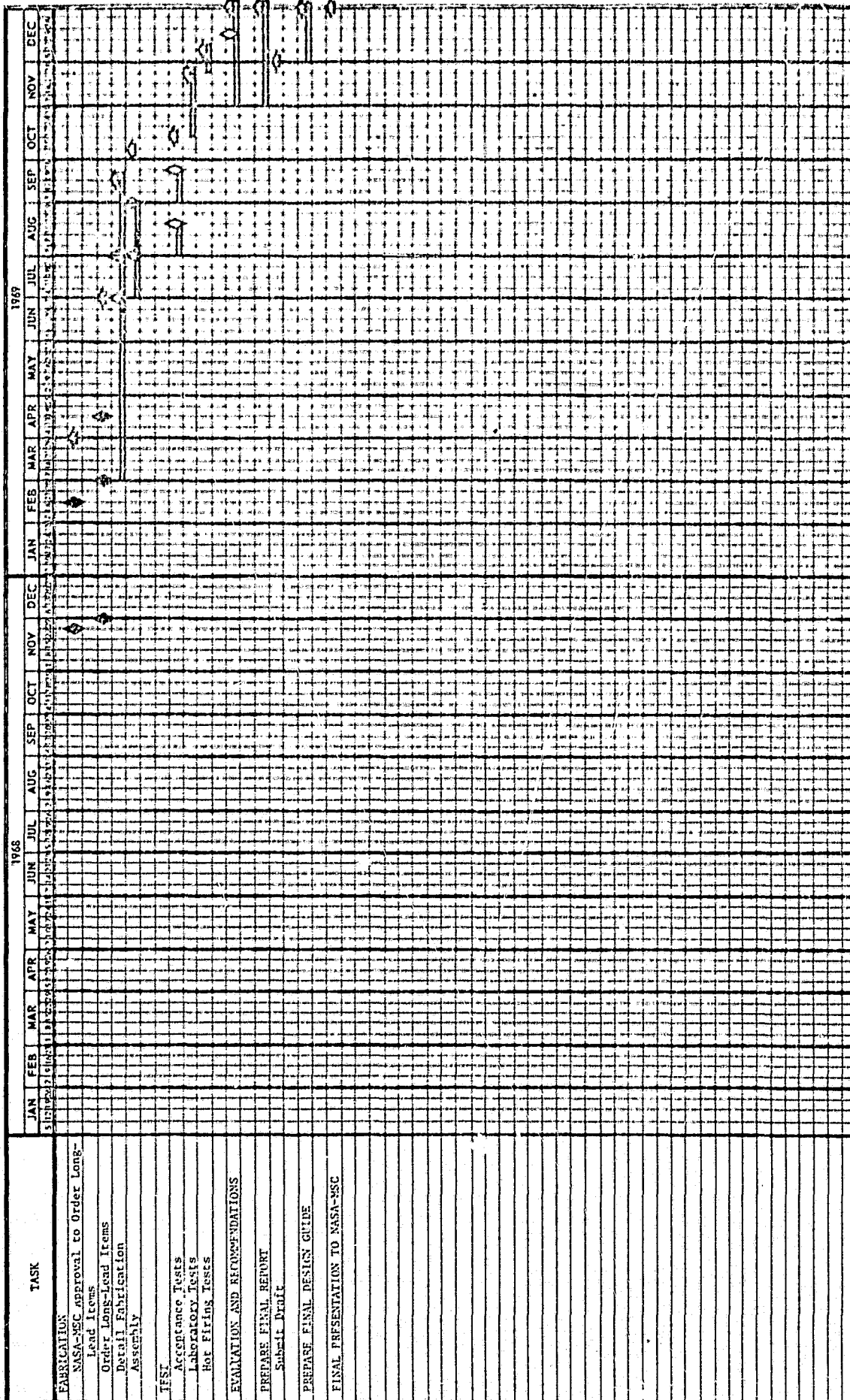
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Master Schedule

Figure 1, Sheet 2 of 3

PHASE II MASTER SCHEDULE



MO 345

Master Schedule

Figure 1, Sheet 3 of 3

1. Downstream Seal Leakage

a. Leakage past ball

- wear
- seal deformation
- seal sticking in cage
- improper contact: area and load
- bad seal-to-ball fit
- marginal ball/seal overlap
- sharp edge on ball
- orange peel finish on ball
- mechanic error
- handling damage
- downstream salts

b. Leg leakage

- seal leg foldover
- improperly staked seal retainer
- seal sticking in cage
- contamination
- handling damage
- Teflon "hairs" on seal leg

2. Upstream Seal Leakage

a. Leakage past ball

- wear
- improper contact area and load
- contamination
- bad seal to ball fit
- bite, excessive deflection in partially closed position
- handling damage

b. Leg leakage

- cage distortion

3. Shaft Seal Leakage

- excessive deflection of shaft during actuation under pressure
- omniseal had insufficient compliance
- inadequate seal surface on shaft

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4. Valve Pressure Drop Variation - critical tolerance stackup and oxidizer lead requirement
5. Shaft Seal Drain Manifold
 - a. Static seal leakage - poor installation design, installation of 4 seals at once
- inadequate seal lead in chamfer
- welded assembly too flexible and easily damaged
 - b. Poor welds - difficult weld required (thin to thick section)
6. Ball Rub - lack of tolerance control
7. Gearbox
 - a. Potential accumulation of vapors - no positive vent
 - b. Roller galling - CRES roller on CRES rack and on CRES bolt with microseal lubrication
8. Actuator
 - a. Actuator piston leakage - spring rubbing produced contamination
 - b. O-Ring blown out of groove - poor face seal design
9. Actuator Piston
 - a. Leakage of piston seal - improper matching of coefficient of expansion
- Eccentricity of piston to shaft
 - b. Aluminum piston galled - Aluminum on aluminum with microseal
10. Pneumatic Tank Contamination - heat treat methods during fabrication
- AM 350 not rust resistant
11. Actuation System
 - a. Initial Block I had insufficient force margin - high ball torque and low regulator outlet pressure
 - b. Valve timing out of spec - piston binding
- inadequate lube
12. Regulator
 - a. Lock-up out of spec - long piston stroke
- main seat leakage

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12.. Regulator (cont.)

- b. Flow rate out of spec - improper stroke adjustment
- c. Internal leakage - contamination from tank and handling

13. Three-Way Valve

- a. Internal leakage - insufficient air gap
- b. External leakage - wrong O-ring on connector
- poor seal on solenoid
- c. Response time - improper spring setting
- d. Moisture in solenoid area - poor O-ring groove design by vendor

14. Two-Way Valve

- a. Leakage - Kel-F seal failures (seat deformation)
- contamination
- extruded O-ring
- b. Response time - extruded O-ring
- improper spring setting
- c. Moisture in solenoid area - poor O-ring groove design by vendor

16. Relief Valve

- a. Spring hang-up - improper cover design

17. Tube Weld Defects

- poor restrictor design for welding

18. Propellant Line to Valve
Flange Leakage

- too low force spring load in Raco seal
- inadequate valve seal surface finish
- improper torque on flange bolts
- contamination and damage caused by improper procedures in installing valve

19. Bipropellant Valve/Injector
Header Leakage

- installation damage
- low spring load Raco seal
- improper finish
- No. 4 seal cage movement

20. Housing Proof Failures

- increased proof pressure required due to heat soakback

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- 21. Harness Insulation Resistance Failures - poor connector design to exclude moisture
- 22. Harness Connector Separation - overtorquing of connector

Method of Actuation				Valve Configuration			
Type	Advantages	Disadvantages	Summary	Type	Advantages	Limiting Types	Summary
<u>Electrical</u>	1. No seals required 2. Repeatable timing 3. Fewer components 4. No decontamination required	1. Does not meet present engine interfaces 2. Requires more development. 3. Impact on system power requirements.	The electrical actuation was selected as the most desirable type for an improved valve. However, due to the long and expensive development required it was determined that the work could not be accomplished within the constraints of the present program. On this basis pneumatic actuation was selected but the actuator design is to be made so that electric actuation could be applied at a later date with no modification required to the valves or valve housings.	<u>Ball</u>	1. Retains some flow characteristics to prevent vapor. 2. Low pressure drop. 3. Proven maximum life of previous experience. 4. Can use two seals per ball.	1. Low sealing force available. 2. High sealing force available on the primary failure to allow for leak. 3. Ball to ball contact is not a problem. 4. No other variations in the ball design. 5. No other variations in the ball design. 6. No other variations in the ball design. 7. No other variations in the ball design. 8. No other variations in the ball design. 9. No other variations in the ball design. 10. No other variations in the ball design.	The ball type valve was selected with the primary failure to allow for leak. Ball to ball contact is not a problem. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design. No other variations in the ball design.
<u>Pneumatic</u>	1. Uses propellant, no third fluid required. 2. No limit on fluid available for actuations.	1. Pressure decay during start transient requires large area piston and large volume of fuel. Decay can also result in erratic valve timing. 2. More cavities to be decontaminated.		<u>Toggle Visor</u>	1. Minimum wiping of seal. 2. Low pressure drop. 3. High seal life. 4. Valve flow wiping for ease of redundant arrangement.	1. Seal per housing. 2. High seal life. 3. Low pressure drop. 4. Valve flow wiping for ease of redundant arrangement. 5. High seal life. 6. Low pressure drop. 7. Valve flow wiping for ease of redundant arrangement.	1. Seal per housing. 2. High seal life. 3. Low pressure drop. 4. Valve flow wiping for ease of redundant arrangement. 5. High seal life. 6. Low pressure drop. 7. Valve flow wiping for ease of redundant arrangement.
<u>Pneumatic</u>	1. High force margin available. 2. Repeatable timing. 3. No decontamination required. 4. No existing interfaces. 5. Maximum utilization of SPS experience.	1. Third fluid must be carried. 2. Additional sealing required. 3. Requires least development.		<u>Ball</u>	1. High seal life. 2. Minimum wiping of seal. 3. Seal complex of pipe types.	1. High pressure drop. 2. Seal per housing. 3. High seal life. 4. Valve flow wiping for ease of redundant arrangement.	1. High pressure drop. 2. Seal per housing. 3. High seal life. 4. Valve flow wiping for ease of redundant arrangement.
				<u>Plate</u>	1. High sealing force. 2. Low pressure drop.	1. High sealing force. 2. Low pressure drop.	1. High sealing force. 2. Low pressure drop.

Bipropellant Valve Tradeoff Chart

Type	Redundancy Configuration		Summary	Drawings	Cost Consideration		Summary
	Advantages	Disadvantages			Advantages	Disadvantages	
Quandmono-propellant Valve	1. Permits assembly/disassembly of one valve without affecting another. 2. Can be adjusted, cycled, and leak tested independent of other valves.	1. Probable increased size and weight. 2. Additional external leak paths. 3. More interface alignment problems.	The cartridge module approach was selected with the housing to be in the quad mono-propellant configuration. This configuration provides the best accessibility to individual modules and still requires the least space and weight external seals.	Can Hic Full	1. Good potential for considerable improvement of chamber cooling capability due to the fact that the cartridge module is isolated from the main body of the engine and can be cooled by the main body cooling system. 2. The cartridge module is isolated from the main body of the engine and can be cooled by the main body cooling system.	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	
Quandmono-propellant Module Valve	1. Any valve can be disassembled without disassembling another. 2. There are fewer internal static seals than cartridge concept. 3. Spacing between valve bores can remain same as QIS.	1. Most complex of all types. 2. Requires additional external static propellant seals for interface between the two valve and actuator units. 3. May not meet existing QIS injector and feedline interfaces.		Quandmono Hic Full	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	
Single Mono-propellant Valve	1. Each module less complex than above types. 2. Can be adjusted, cycled, and leak tested independent of other valves.	1. Requires more flanges than cartridge concept. 2. Probable increased size and weight. 3. More interface alignment problems.		Single Hic Full	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	
Cartridge Module	1. Most compact of the designs. 2. Fewer flanges and static seals. 3. Each valve mechanism is identical and completely independent of the another. 4. Each cartridge will house and support both the ball and shutoff seals therefore cartridges can be assembled, cycled, and leak tested prior to assembly in the valve housing. 5. Shaft seal problems are virtually eliminated by use of universal linkage between shafts of actuator and cartridge.	1. Increased space required between the upper and lower bores and between the balls possibly requiring redesign of present injector and/or propellant feedline interfaces.		Cartridge Hic Full	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	1. The high cost of the cartridge module approach is a major consideration for the primary reason of least total weight. 2. The cartridge module approach is a major consideration for the primary reason of least total weight.	

Bipropellant Valve Tradeoff Chart

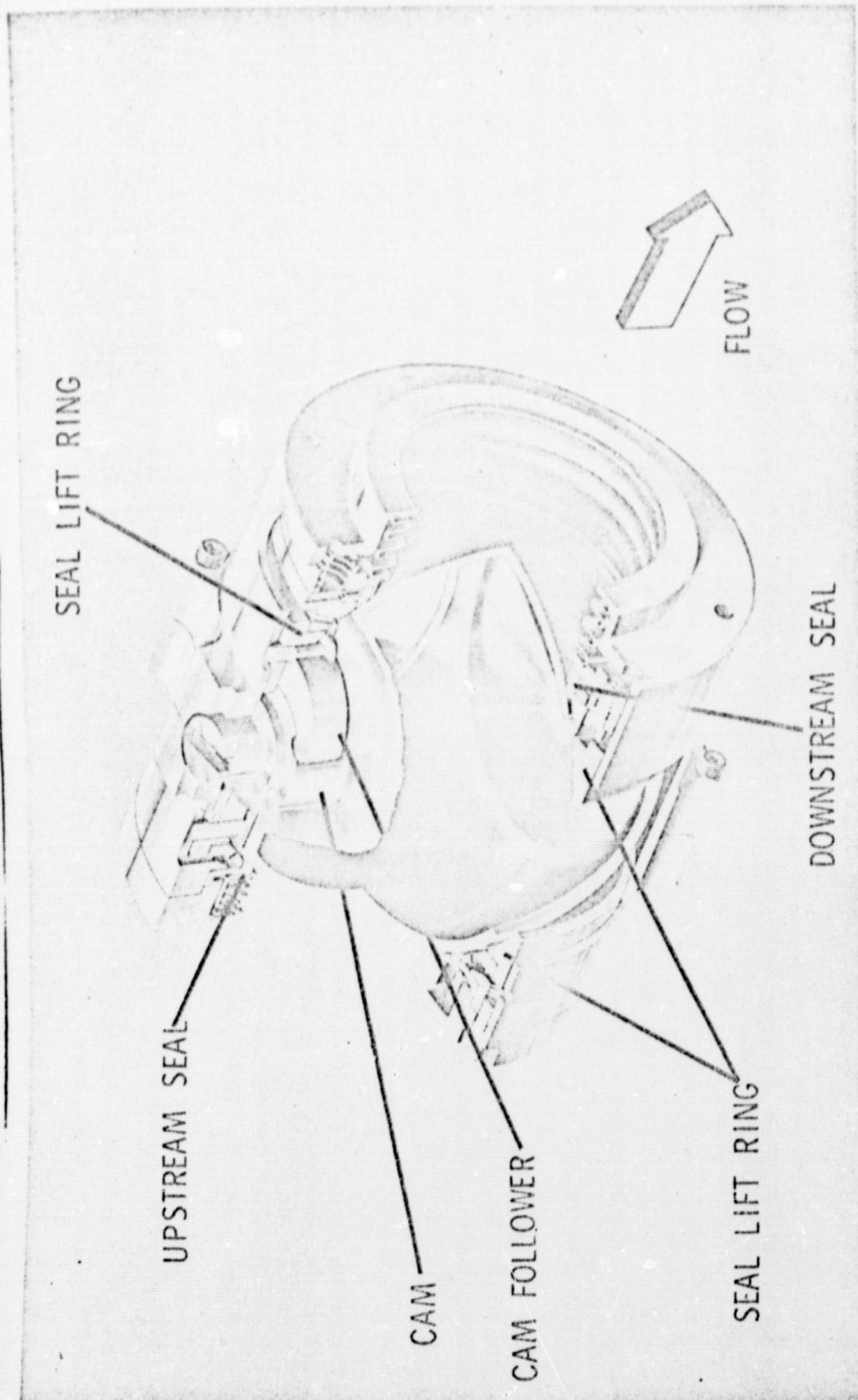
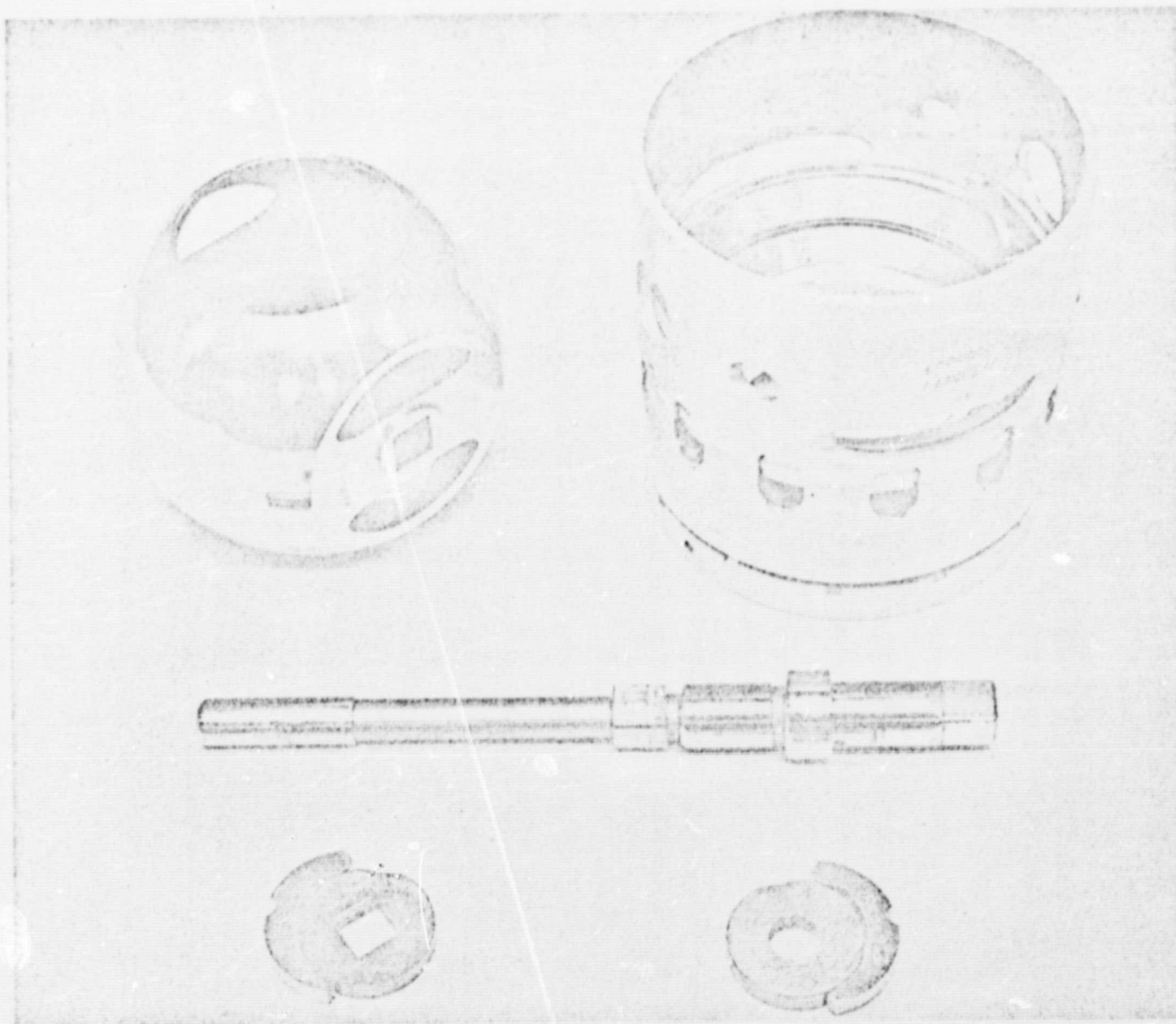


Figure 4

Cartridge Assembly

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Major Components of Single-Position Test Assembly

Figure 5

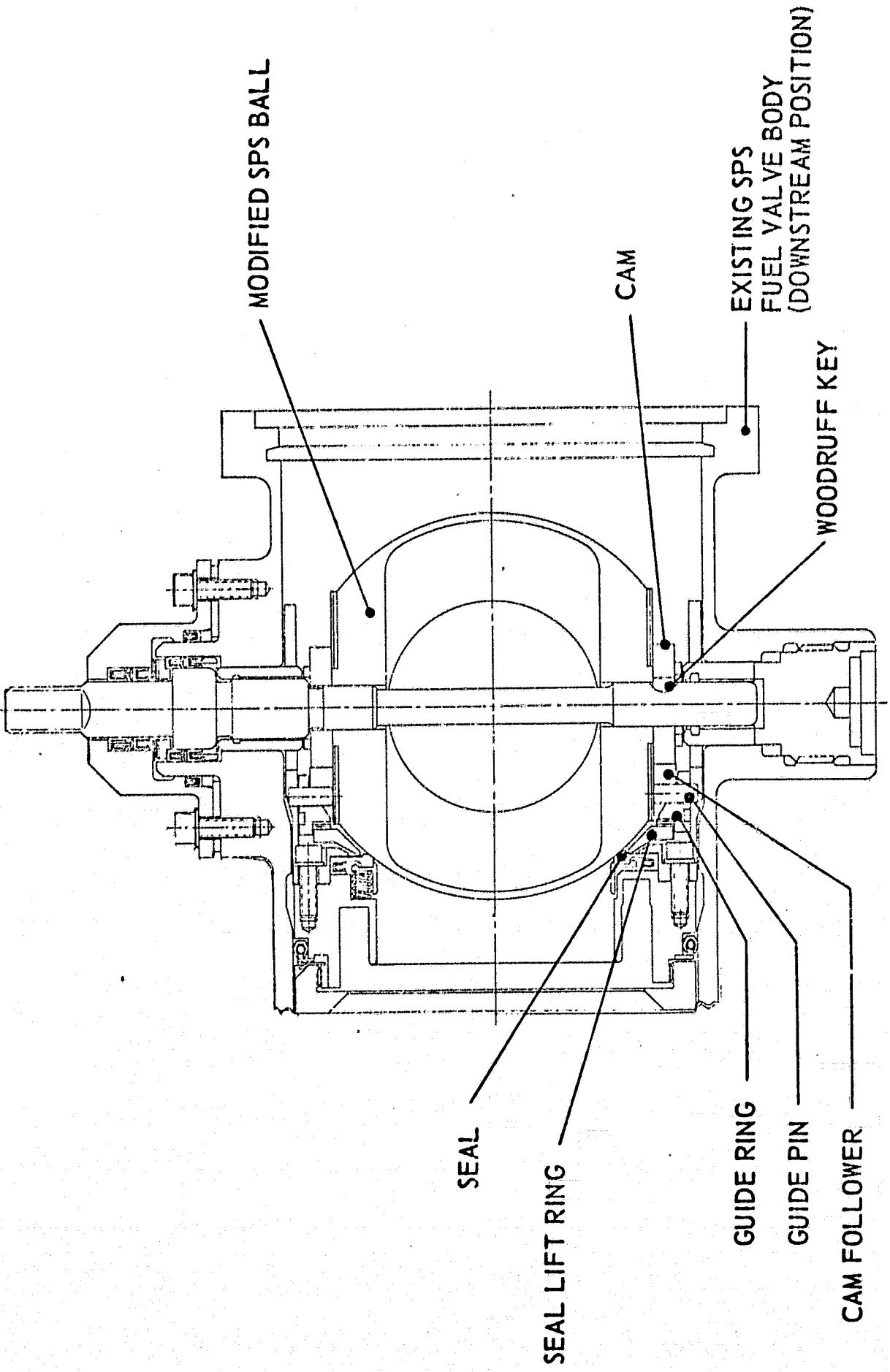


Figure 6

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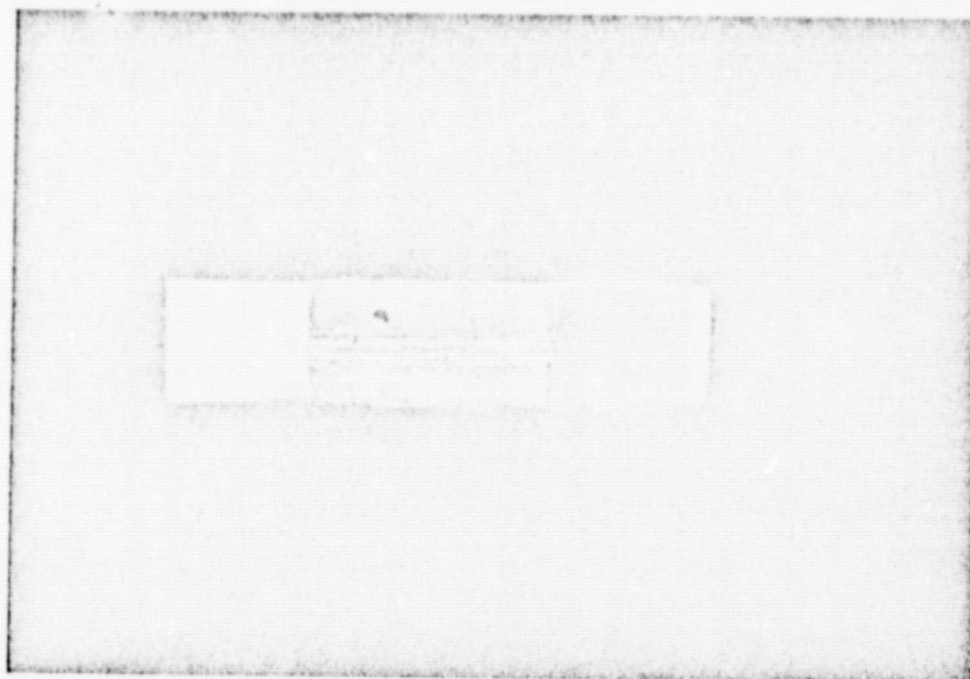


Figure 7a. Galled Surface of P/N 115920-2 Cam. Indentations are a Result of Surface Hardness Measurements. Magnification 5X

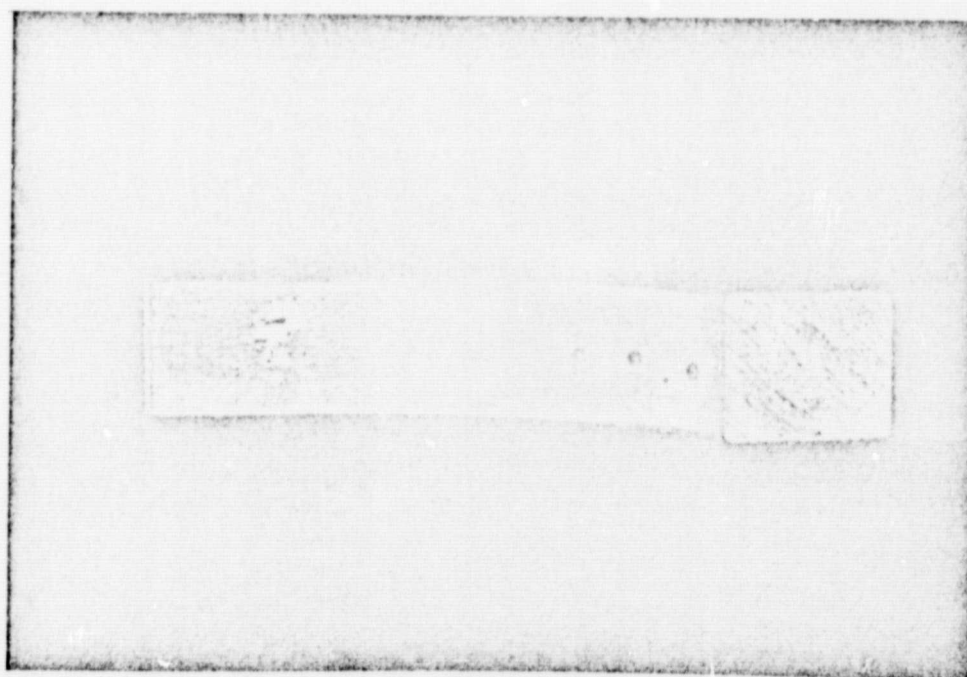


Figure 7b. Galled Surface of P/N 115917 Cam. Follower. Magnification 5X

Galled Surfaces of Cam and Follower

Figure 7

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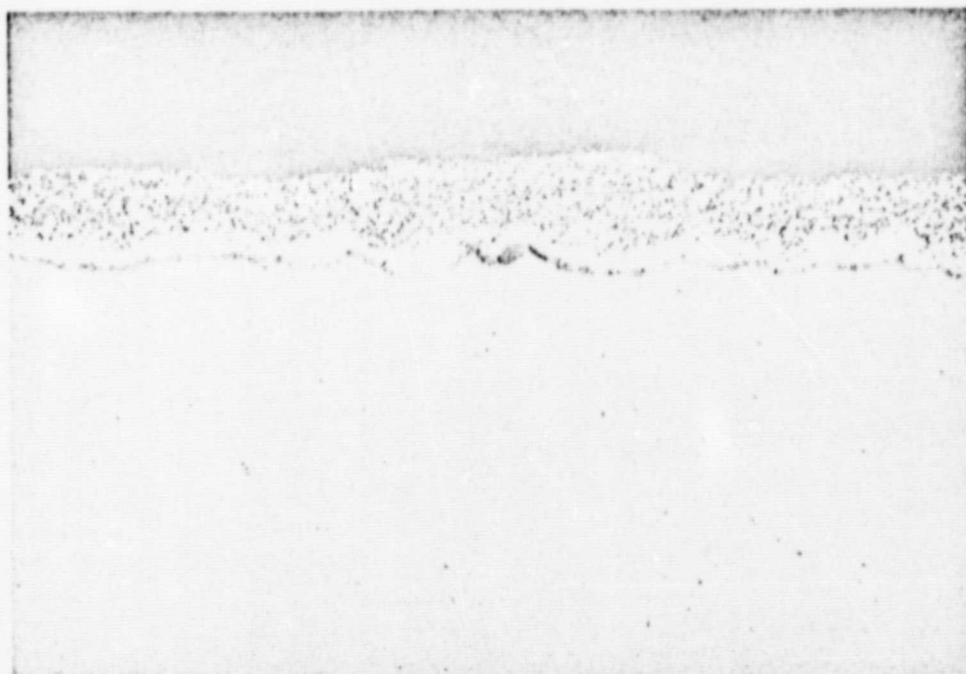


Figure 8a. Cross Section of Colmonoy Coating. On Follower. Raised Portion is Coating Transferred from Cam. Note Roughness of Parent Metal Beneath Galled Surface. Magnification: 100X

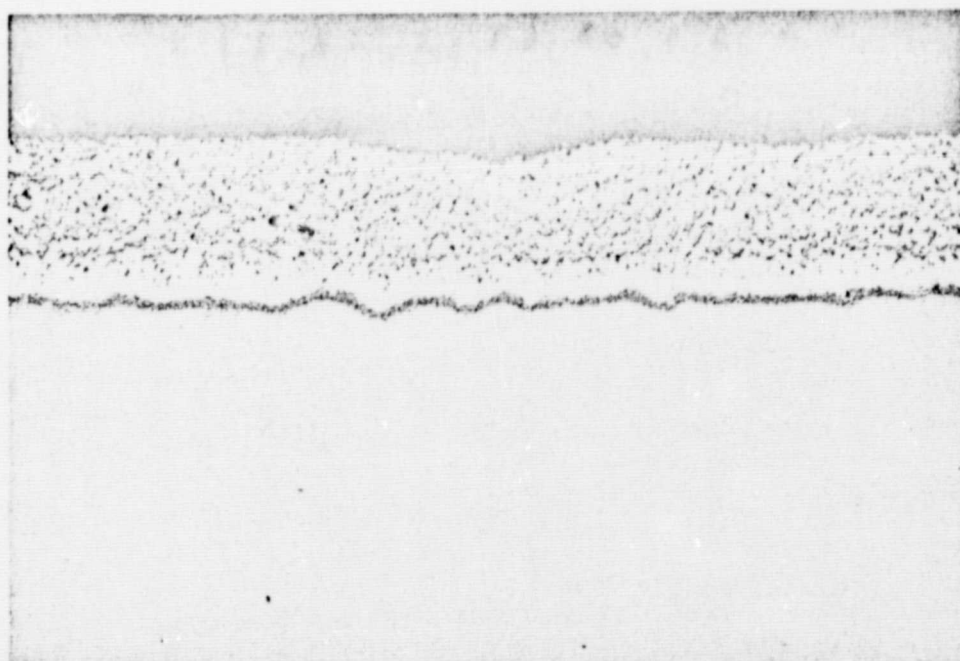
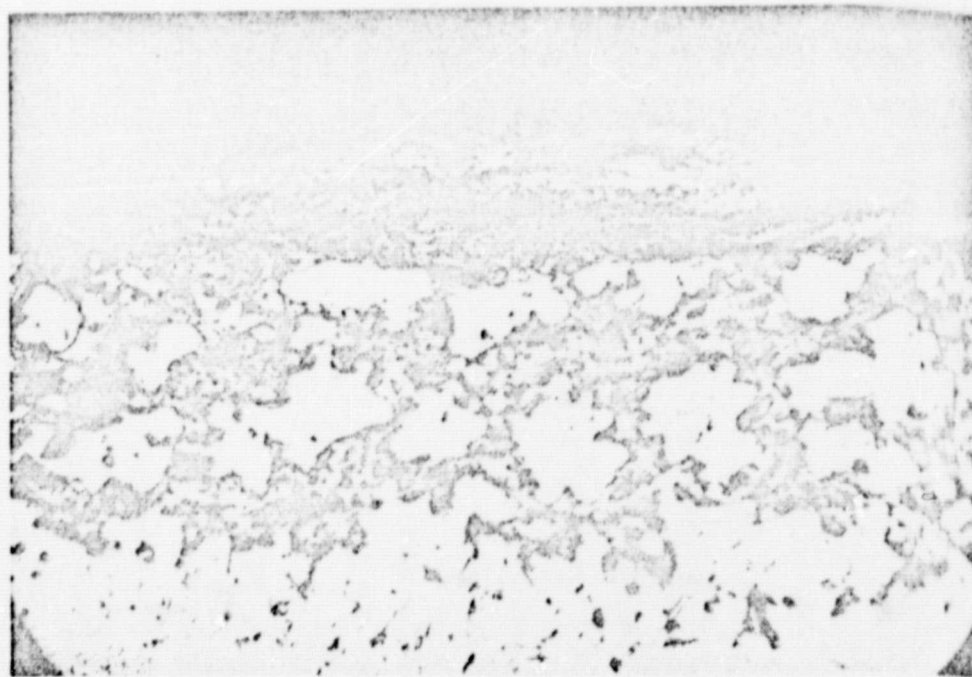


Figure 8b. Cross Section of Colmonoy Coating on Cam. Depression is Due to Transfer of Material to Follower. Magnification: 100X

Cross-Sections of Coatings on Cam and Follower

Figure 8

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Cross-Sectional View of Colmonoy Coating on Follow-on
Etchant: 10% Oxalic Acid Magnification: 500X

Figure 9

DRY CYCLE TEST - LEAK RATE IN SCC/HR GN₂

INLET PRESS	CYCLES*										
	0	10	25	50	100	150	200	250	300		
2	140	60	20	30	40	30	30	30	10		
75	0	0	0	0	0	0	0	0	0		
175	0	0	0	0	0	0	0	0	0		
2	0	0	0	0	0	0	0	0	0		

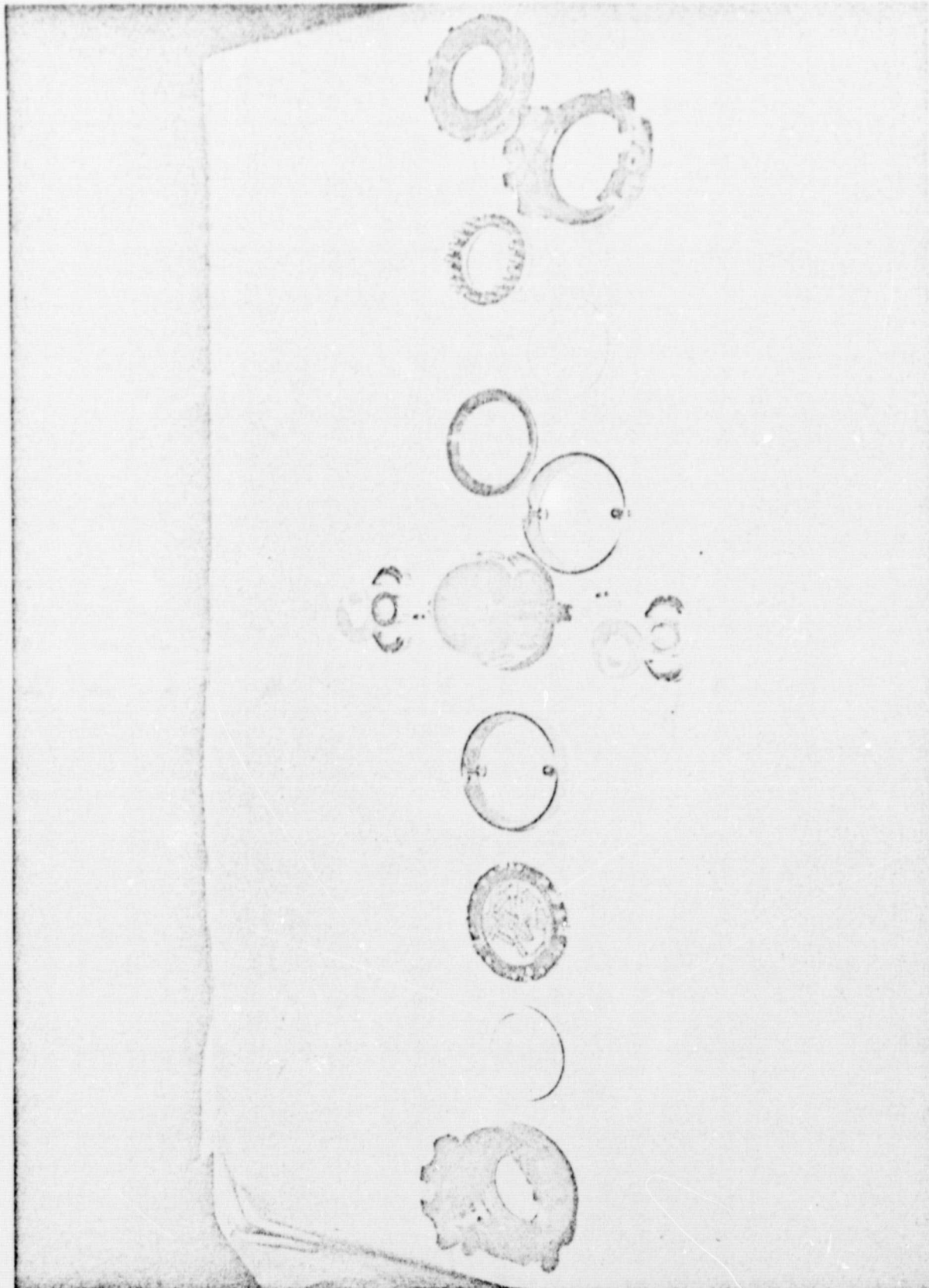
N₂O₄ CYCLE TEST - LEAK RATE IN SCC/HR GN₂

INLET PRESS	CYCLES*										
	0	64	250	500	1000	2000	3000	4000	5000	6000	10,000
2	0	10	100	480	300	0	3840	30	2280	0	0
75	0	0	0	10	40	24	160	300	360	56	20
175	0	0	10	20	30	10	190	160	360	40	24
2	0	0	0	0	0	0	0	0	0	0	0

*CAMS LUBRICATED WITH FS-1281 AND DC-11 TO FACILITATE TESTING - NOT SATISFACTORY FOR FLIGHT

Figure 10

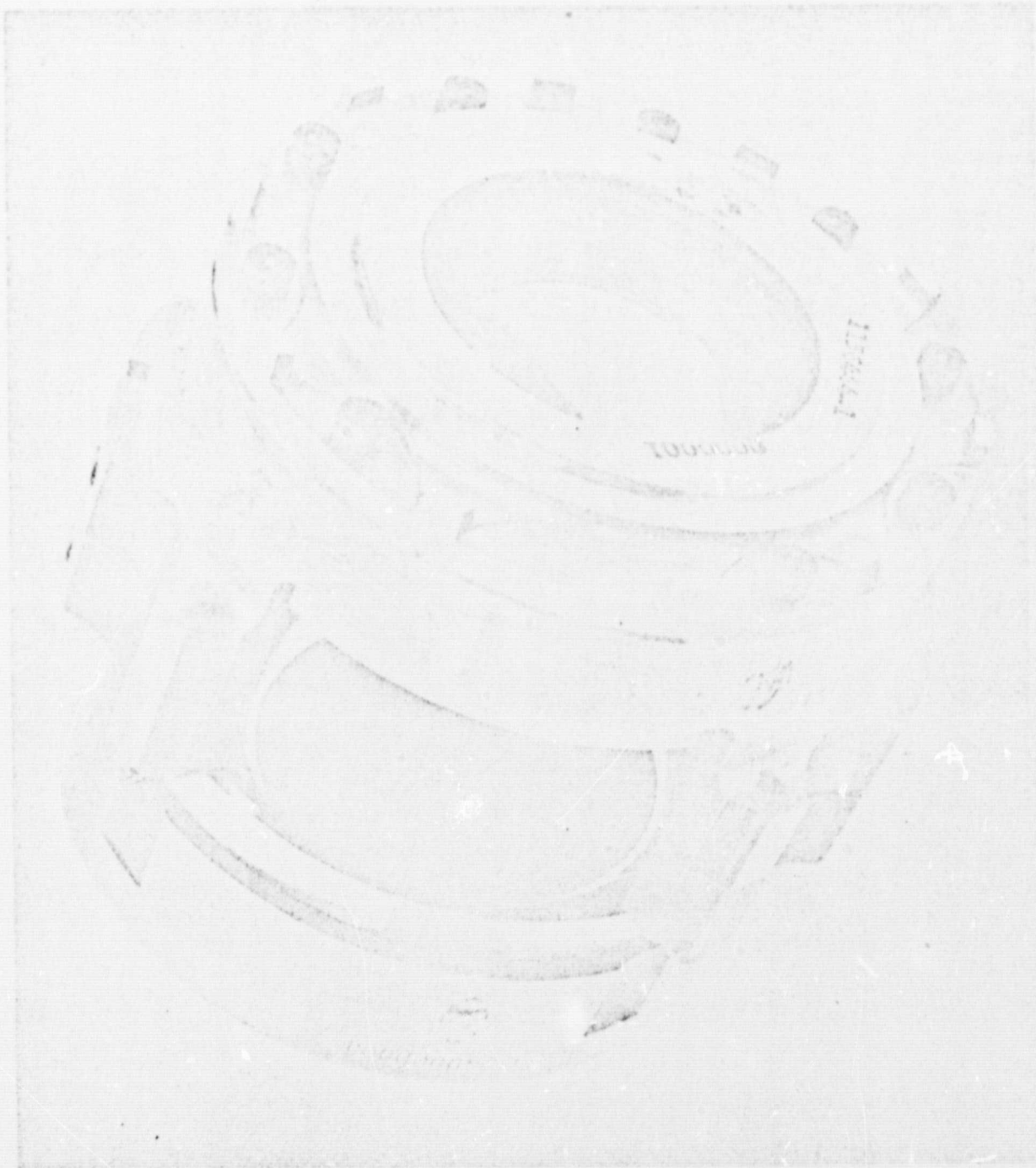
Report 8317-PI



Disassembled SPS-VIP Cartridge

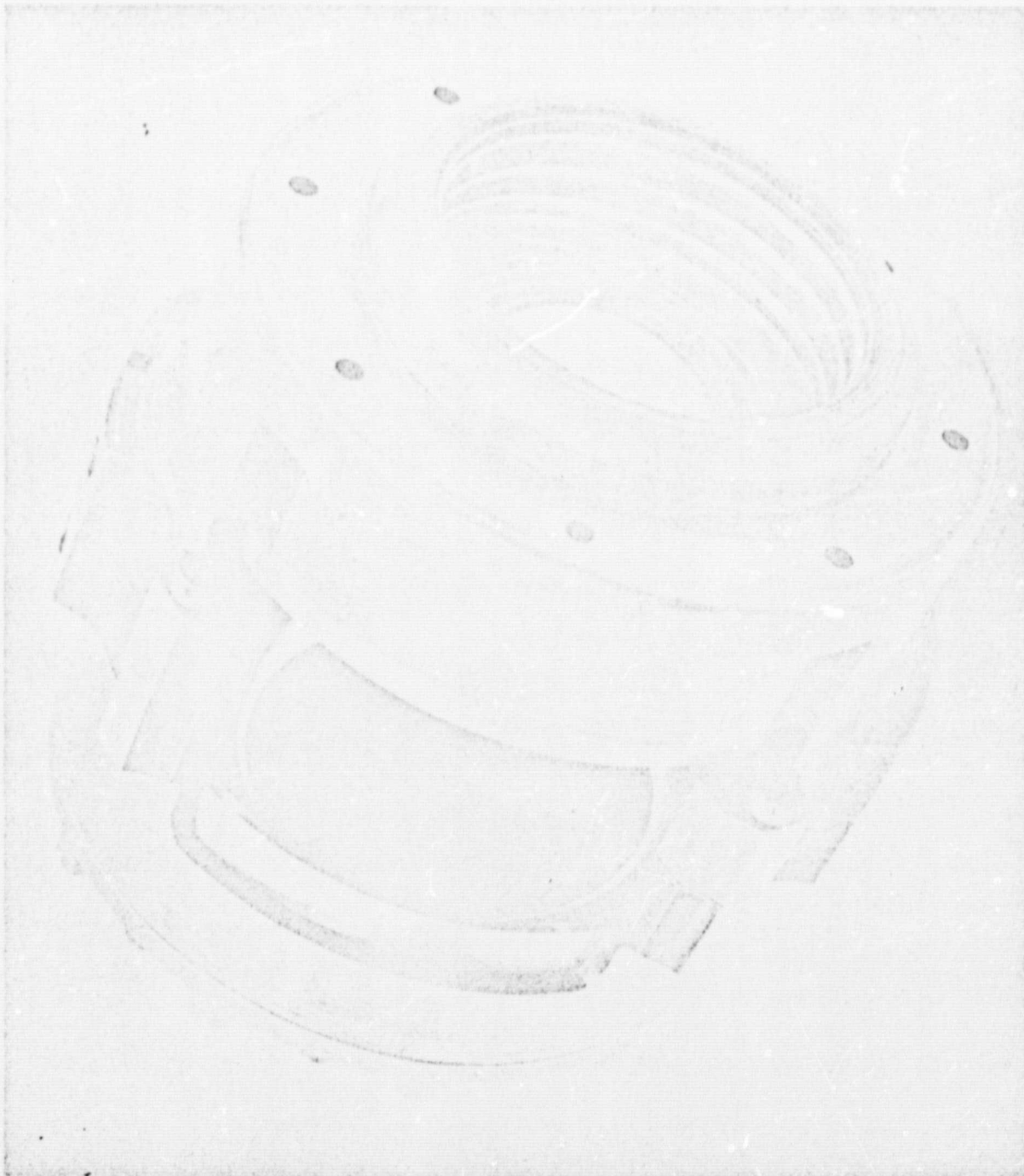
Figure 11

Report 8317-PI



Upstream View of Assembled Cartridge

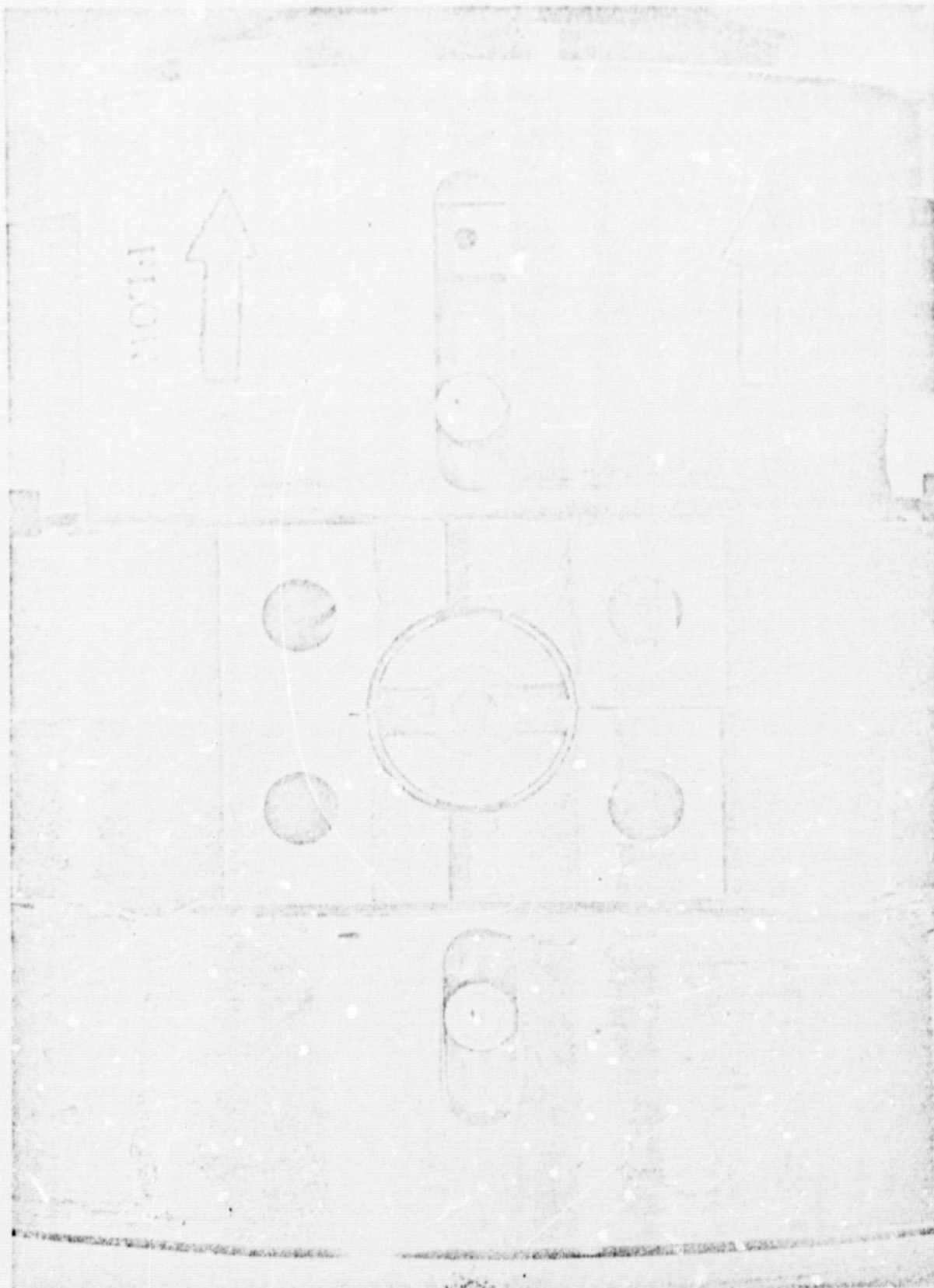
Report 8317-PI



Downstream View of Assembled Cartridge

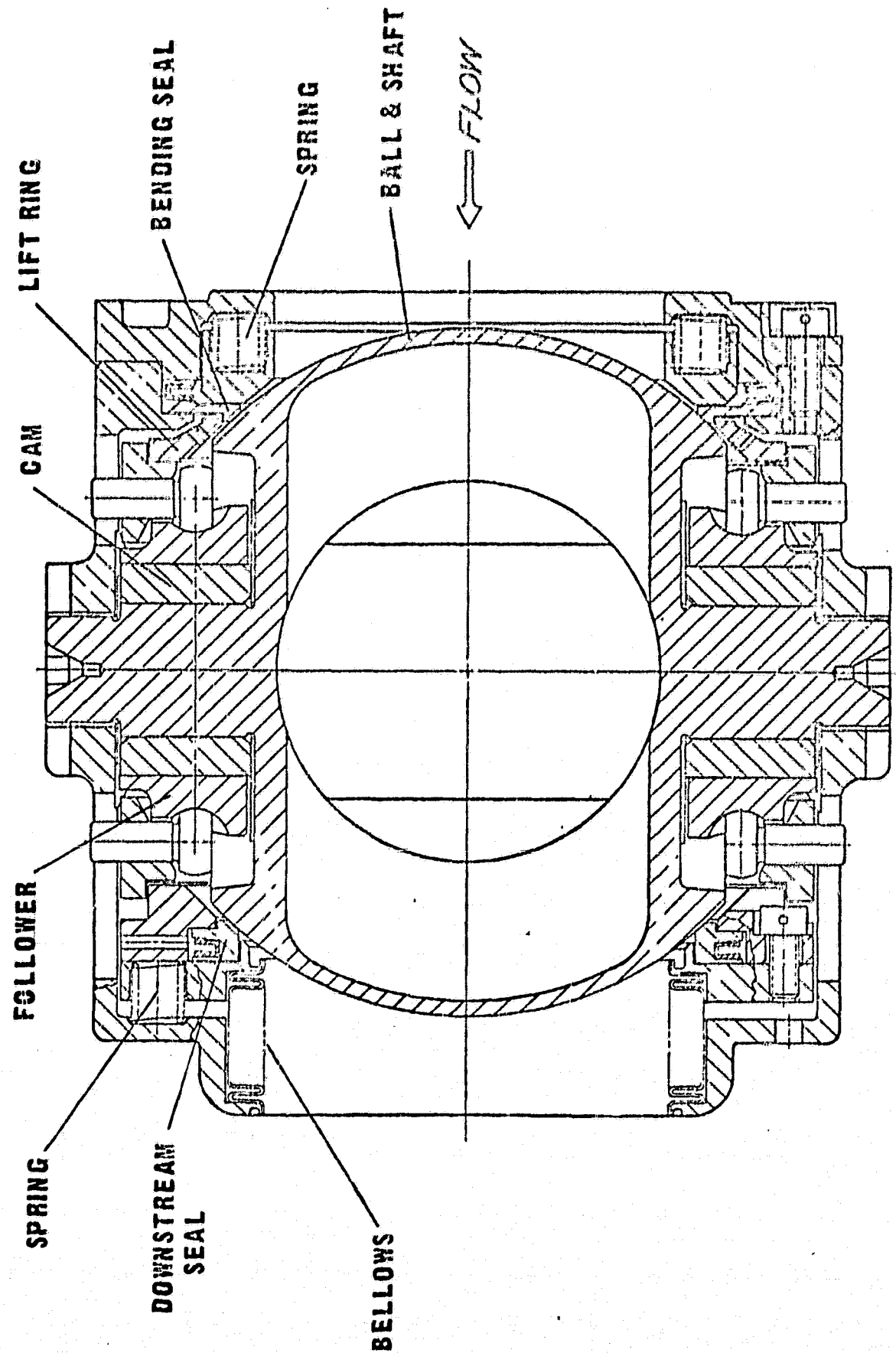
Figure 13

Report 8317-PI



Cartridge Assembly Showing Cam and Actuator Contact Through Inspection Ports

Figure 14



Cartridge Assembly - Bending Seal

Figure 15

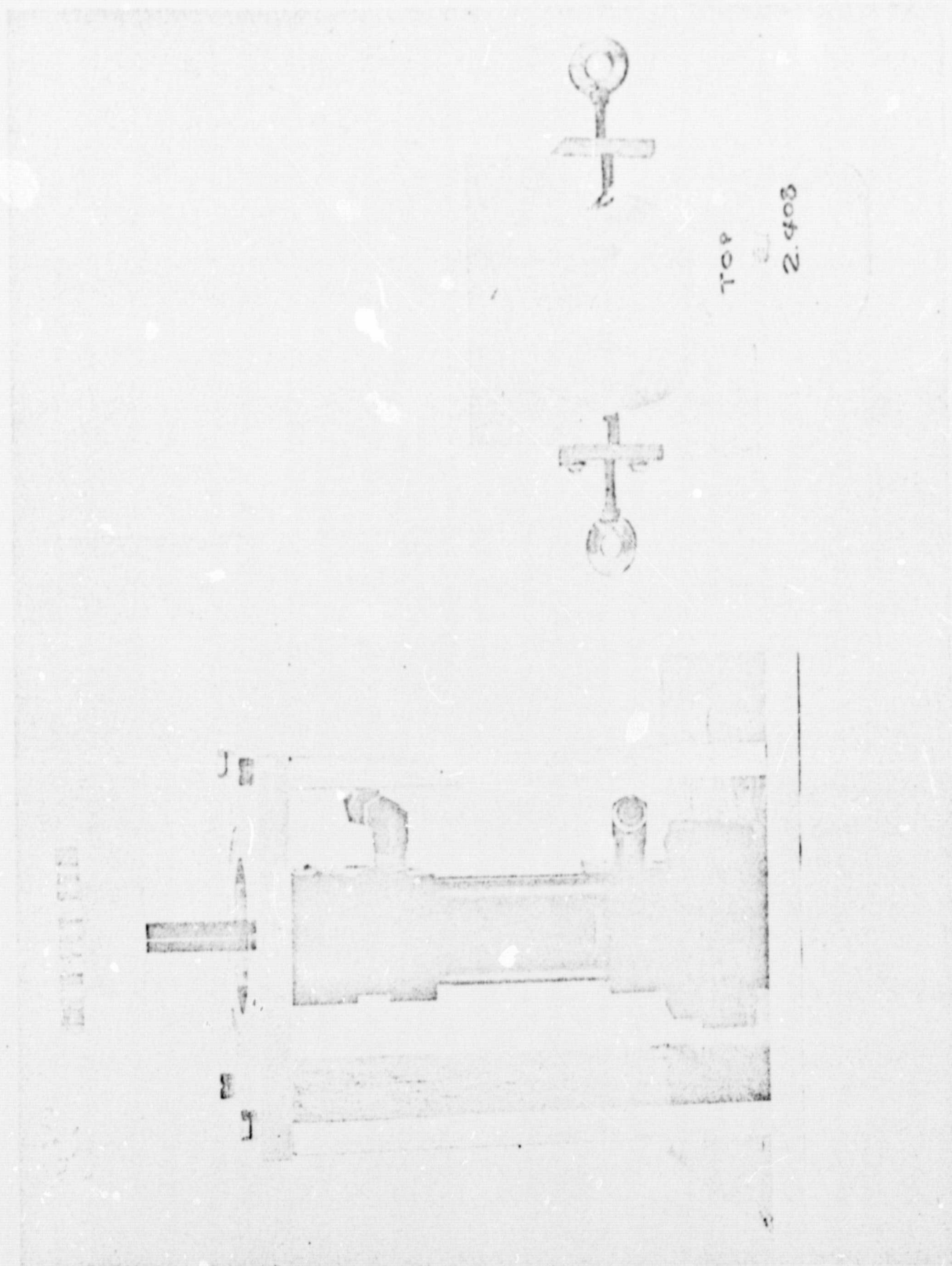
Report 8317-PI



Downstream Seal Showing ID Ring

Figure 16

Report 8317-PI



Cartridge Assembly Tooling

Figure 17

0

Report 8317-PI

<u>No. of Wet Cycles</u>	<u>Bearing and Ball Seals before Cam Lift-off Breakaway/Run</u>	<u>Cam Lift-Off and Opening Breakaway/Run</u>	<u>Valve Closing Breakaway/Run</u>
Acceptance	25/25	48/25	30/32
64	35/30	50/35	35/32
150	45/45	55/42	42/40
250	35/32	60/42	40/45
500	45/42	60/50	45/48
1000	35/35	50/30	35/30
1500	27/27	100/85 ⁽¹⁾	80/75
After rework	28/28	42/30	30/28
1550	40/38	60/38	35/30
2000	35/35	70/55	50/50
2500	40/40	70/55	45/45
3500	38/38	82/60	58/58
4750	38/38	70/60	56/56
7500	35/35	65/58	52/52
10,000	30/30	65/58	52/52
15,000	42/42	58/55	54/54
20,000	40/40	60/55	55/55
20,020 ⁽²⁾	35/35	80/60	62/55

(1) Cams and followers were relapped after 1500 cycles.

(2) Last 20 wet cycles of series conducted with 225-psig inlet pressure.
All other cycles with 163-psig inlet pressure.

Oxidizer Valve Shaft Torque (in.-lb) - Wet Cycle Test Series

Figure 18

Report 8317-PI

No. of Wet Cycles	Ball Seal No. 1				Ball Seal No. 2					Shaft Seals					
	GN ₂ Leak Test Pressures, psig														
	2	75	175	240	2	2	75	175	240	2	2	75	175	240	2
Acceptance	0	0	0	-	0	0	0	16	-	0	0	0	0	-	0
64	0	0	0	-	0	0	0	0	-	0	0	0	0	-	0
150	0	0	0	-	0	96	36	10	-	0	0	0	0	-	0
250	0	0	10	-	0	0	0	0	-	0	0	0	0	-	0
500	0	70	144	-	0	0	0	10	-	0	0	0	0	-	0
1000	0	42	72	-	0	0	0	0	-	0	0	0	0	-	0
1500	0	0	0	-	0	0	0	0	-	0	0	0	0	-	0
After rework	0	18	120	-	0	0	0	60	-	0	0	0	0	-	0
1550	0	0	0	-	0	0	30	100	-	0	0	0	0	-	0
2000	0	0	88	-	10	0	30	46	-	0	0	0	0	-	0
2500	0	0	0	-	0	0	0	10	-	0	0	0	0	-	0
3000	0	0	28	-	0	0	0	0	-	0	0	0	0	-	0
4750	0	0	16	-	0	0	0	0	-	0	0	0	0	-	0
7500	0	0	0	16	0	0	0	0	16	0	0	0	0	0	0
10,000	0	0	8	0	0	0	0	0	20	0	0	0	0	0	0
15,000	0	16	110	255	0	0	0	14	16	0	0	0	0	0	0
20,000	0	120	160	208	0	0	0	0	0	0	0	0	0	0	0
20,020 ⁽¹⁾	0	630	876	1080	0	0	0	40	28	0	0	0	0	0	0

(1) Last 20 wet cycles of series conducted with 225-psig inlet pressure. All other cycles with 163-psig inlet pressure.

Oxidizer Valve Wet Cycle Leakage Rates (cc/hr)

Figure 19

Report 8317-PI

<u>No. of Dry Cycles</u>	<u>Bearing and Ball Seals before Cam Lift-Off Breakaway/Run</u>	<u>Cam Lift-Off and Opening Breakaway/Run</u>	<u>Valve Closing Breakaway/Run</u>
(1)	45/35	74/65	75/65
50	26/26	55/60	50/55
100	40/40	45/30	30/30
200	35/35	45/35	30/30
300	40/36	50/40	33/33
500	34/34	50/45	35/35
750	33/33	50/45	36/36
1000	33/33	50/48	40/40
1500	33/33	65/65	55/55
2000	33/32	65/65	55/55
3000	33/33	62/62	53/52
4000	33/32	70/70	65/65
5000	33/32	70/70	60/60

(1) Torque test after 20,020 wet cycle test series and partial valve disassembly for cam, follower, and ball seal visual inspection.

Oxidizer Valve Shaft Torque (in.-lb) - Dry Cycle Test Series

Figure 20

Report 8317-PI

No. of Dry Cycles	Ball Seal No. 1				Ball Seal No. 2				Shaft Seals			
	GN ₂ Leak Test Pressure, psig											
	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>
(1)	0	50	40	0	0	20	120	0	0	0	0	0
50	0	0	156	0	0	0	0	0	0	0	0	0
100	0	70	510	0	0	J	0	0	0	0	0	0
200	0	76	450	0	0	0	0	0	0	0	0	0
300	0	50	246	0	0	0	0	0	0	0	0	0
500	0	70	330	0	0	0	0	0	0	0	0	0
750	0	60	330	0	0	0	0	0	0	0	0	0
1000	0	56	330	0	0	0	0	0	0	0	0	0
1500	0	170	456	0	0	0	0	0	0	0	0	0
2000	0	130	750	0	0	0	0	0	0	0	0	0
3000	0	76	225	0	0	0	20	0	0	0	0	0
4000	0	390	1080	10	0	20	68	0	0	0	60	0
5000	0	384	1140	10	0	24	84	0	0	0	0	0

(1) Leak test after 20,020 wet cycle test series and partial valve disassembly for cam, follower, and ball seal visual inspection.

Oxidizer Valve Dry Cycle Leakage Rates (cc/hr)

Figure 21

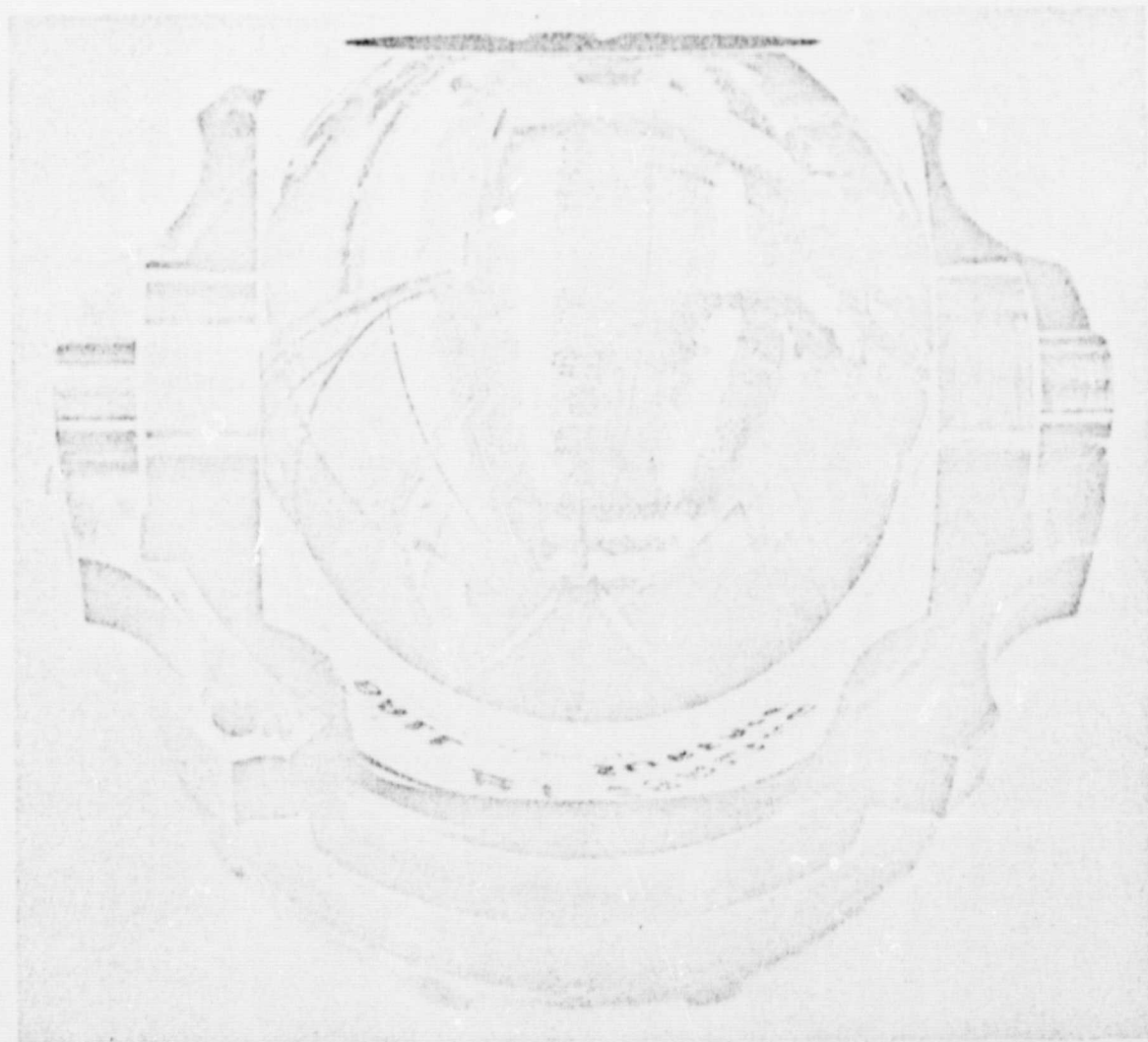
Report 8317-PI

No. of Dry Cycles	Ball Seal No. 1			Ball Seal No. 2			Shaft Seals					
	GN ₂ Leak Test Pressure, psig			GN ₂ Leak Test Pressure, psig			GN ₂ Leak Test Pressure, psig					
	2	75	175	2	75	175	2	75	175	2	75	175
(1)	0	50	40	0	0	20	120	0	0	0	0	0
50	0	0	156	0	0	0	0	0	0	0	0	0
100	0	70	510	0	0	0	0	0	0	0	0	0
200	0	76	450	0	0	0	0	0	0	0	0	0
300	0	50	246	0	0	0	0	0	0	0	0	0
500	0	70	330	0	0	0	0	0	0	0	0	0
750	0	60	330	0	0	0	0	0	0	0	0	0
1000	0	56	330	0	0	0	0	0	0	0	0	0
1500	0	170	456	0	0	0	0	0	0	0	0	0
2000	0	130	750	0	0	0	0	0	0	0	0	0
3000	0	76	225	0	0	0	20	0	0	0	0	0
4000	0	390	1080	10	0	20	68	0	0	0	60	0
5000	0	384	1140	10	0	24	84	0	0	0	0	0

(1) Leak test after 20,020 wet cycle test series and partial valve disassembly for cam, follower, and ball seal visual inspection.

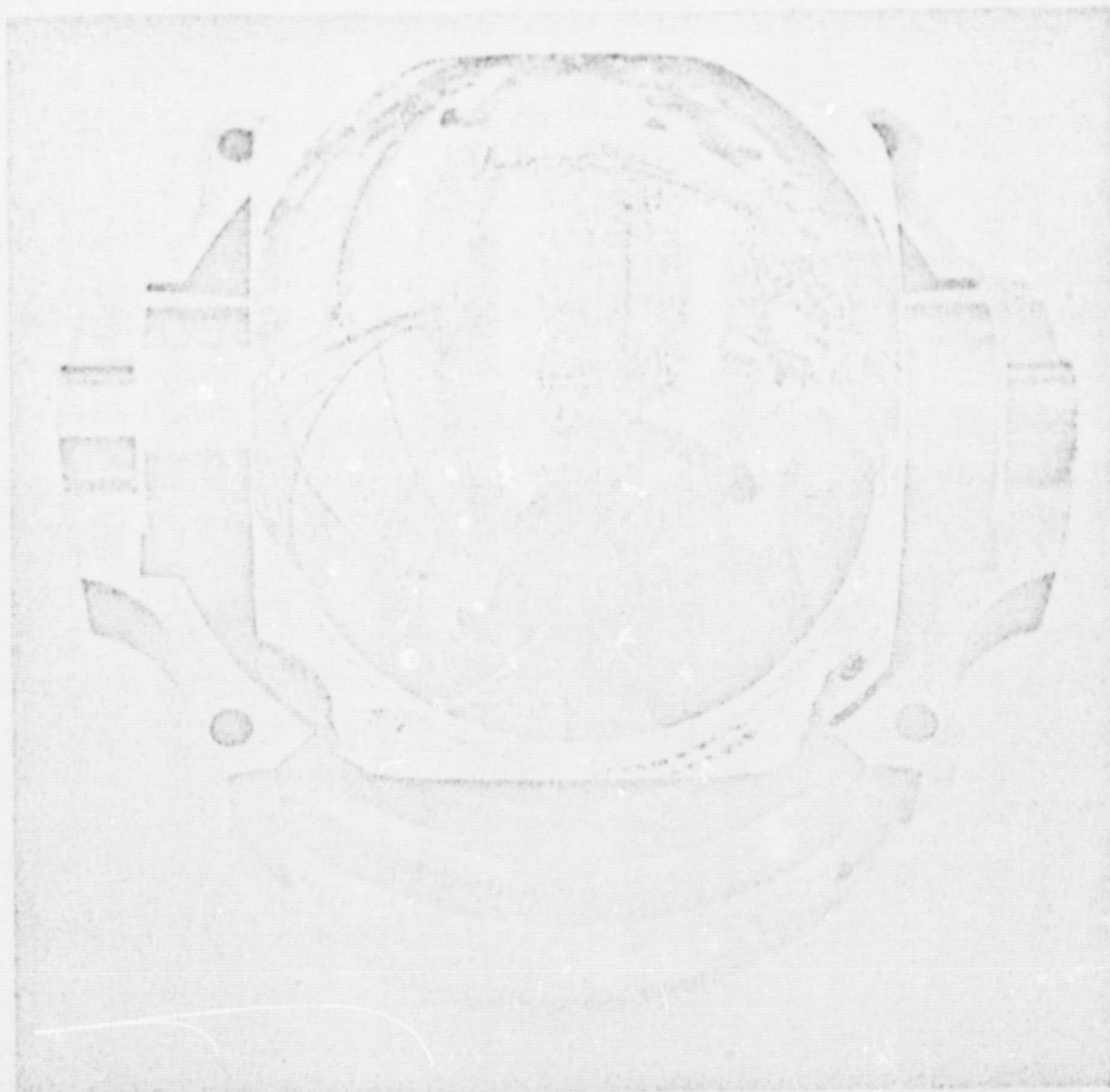
Oxidizer Valve Dry Cycle Leakage Rates (cc/hr)

Figure 21



Oxidizer Valve Posttest - No. 1 Seal
Sealing Surface on Ball

Figure 22



Oxidizer Valve Posttest - No. 2 Seal
Sealing Surface on Ball

Figure 23

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<u>No. of Dry Cycles</u>	<u>Bearing & Ball Seals Before Cam Lift-Off Breakaway/Run</u>	<u>Cam Lift-off and Opening Breakaway/Run</u>	<u>Valve Closing Breakaway/Run</u>
10	28/26	43/29	26/29
50	40/30	46/27	24/26
100	44/34	47/28	25/27
200	46/29	47/30	27/29
300	45/35	48/27	26/28
500	44/33	45/26	22/24
1000	41/31	42/25	21/24
2000	37/31	40/22	18/20
2010	44/32	42/25	22/25
3000	42/35	45/30	29/31
4000	41/32	60/45	38/40
5000	41/35	105/100	75/80

Fuel Valve SN X-2 Torque (in.-lb) Dry Cycle Test

Figure 24

Report 8317-PI

No. of Dry Cycles	Ball Seal No. 1				Ball Seal No. 2				Shaft Seals			
	GN ₂ Leak				Test Pressure, psig							
	<u>2</u>	<u>75</u>	<u>MS</u>	<u>2</u>	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>
10	0	40	64	0	0	32	0	0	0	0	0	0
50	0	68	120	0	0	32	0	0	0	0	0	0
100	0	54	82	0	0	22	0	0	0	0	0	0
200	0	20	28	0	0	26	0	0	0	0	0	0
300	0	14	22	0	0	12	0	0	0	0	0	0
500	0	24	26	0	0	30	2	0	0	0	0	0
1000	0	16	26	0	4	44	30	0	0	0	0	0
2000	0	252	360	0	2	264	1080	2	0	0	0	0
2010	0	112	228	0	0	152	290	0	0	0	0	0
3000	6	960	1320	0	0	38	130	0	20	80	70	0
4000	90	4800	9600	62	34	1032	3600	20	280	8	0	0
5000	90	4800	10400	32	80	3660	10800	110	30	2400	3600	50

Fuel Valve SN X-2 Dry Cycle Leakage Rates (cc/hr)

Figure 25

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No. of Dry Cycles	Ball Seal No. 1				Ball Seal No. 2				Shaft Seals			
	GN ₂ Leak				Test Pressure, psig							
	<u>2</u>	<u>75</u>	<u>MS</u>	<u>2</u>	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>	<u>2</u>	<u>75</u>	<u>175</u>	<u>2</u>
10	0	40	64	0	0	32	0	0	0	0	0	0
50	0	68	120	0	0	32	0	0	0	0	0	0
100	0	54	82	0	0	22	0	0	0	0	0	0
200	0	20	28	0	0	26	0	0	0	0	0	0
300	0	14	22	0	0	12	0	0	0	0	0	0
500	0	24	26	0	0	30	2	0	0	0	0	0
1000	0	16	26	0	4	44	30	0	0	0	0	0
2000	0	252	360	0	2	264	1080	2	0	0	0	0
2010	0	112	228	0	0	152	290	0	0	0	0	0
3000	6	960	1320	0	0	38	130	0	20	80	70	0
4000	90	4800	9600	62	34	1032	3600	20	280	8	0	30
5000	90	4800	10400	32	80	3660	10800	110	30	2400	3600	50

Fuel Valve SN X-2 Dry Cycle Leakage Rates (cc/hr)

Figure 25

Report 8317-PI

<u>Drawing</u>	<u>Title</u>
1156401	Flange Inlet
1156402	Flange Outlet
1156404	Washer, Spacer, Seal
1156405	Sleeve, Spacer, Seal
1156407	Drive Shaft, Oxidizer Valve
1156408	Drive Shaft, Fuel Valve
1156409	Coupling, Drive Shaft
1156411	Bearing, Shaft, Inboard
1156412	Bearing, Shaft, Outboard
1156413	Ring, Retaining Shaft Seal
1156414	Ring, Spacer, Inboard Shaft Seal
1156416	Ring, Spacer, Outboard Shaft Seal
1156418	Ring, Separator, Shaft Seals
1156419	Key, Woodruff, Drive Shaft
1156425, A & B	Cover - Drive Shaft
1156426, A	Cover, Valve Body
1156427	Shaft, Drive, Lever
1156428 (3 Sheets)	Body Bipropellant Valve Single Bore
1156430 (4 Sheets)	Cartridge Test Assembly
1156431	Cam Follower, Dual Link
1156432	Flange - Seal Retainer Cartridge
1156433	Ring - Guide Cartridge
1156434	Pin, Straight, Headed
1156435, A	Bearing - Thrust Armalon
1156436	Washer - Thrust Polytetrafluoroethylene Nonmetallic
1156437, A	Seal, Ball Valve - Downstream
1156438	Seal, Ball Valve - Upstream Envelope Drawing
1156439	Ring-Seal Support
1156440, A,B,C, (3 Sheets)	Housing, Cartridge - Downstream Envelope Drawing

List of Drawings

Figure 26, Sheet 1 of 3

Report 8317-PI

<u>Drawing</u>	<u>Title</u>
1156441, A	Cartridge Housing, Upstream
1156442	Shim
1156443	Ring - Liftoff Seal Cartridge
1156444	Flange Cartridge
1156445 (2 Sheets)	Ball, Bipropellant Valve
1156446	Ring - Seal & Spring Cartridge
1156447, A,B,C	Ball Bipropellant Valve
1156448	Cam, Valve
1156449, A	Seal, Ball Valve - Upstream Envelope Drawing
1156450 (3 Sheets)	Cartridge Assembly, Ball Valve
1156451, A	Housing Assembly, Cartridge - Upstream
1156452	Flange, Bellow - Cartridge
1156453, A & B	Cartridge Housing, Upstream
1156454	Bellows, Toroidal - Envelope Drawing
1156455	Pin, Cartridge Housing
1156650	Plug, Alignment Shaft Key
1156651	Seal Assembly Aid 0.500 ID Raco
1156652	Retainers Ball Seal Valve 
1156653	Retainers Ball Seal Valve 
1156654	Support Ring Ball Seal Valve 
1156461	Body, Actuator
1156462	Flange - Bipropellant Valve Pneumatic Actuator
1156463, A	Cam Follower, Dual Link
1156464	Piston - Bipropellant Valve Pneumatic Actuator
1156465	Gear, Idler - Bipropellant Valve Pneumatic Actuator
1156466	Shaft, Gear - Bipropellant Valve Pneumatic Actuator
1156467	Bearing, Flange
1156468	Seal, Molded - Bipropellant Valve Pneumatic Actuator
1156469	Bearing, Sleeve
1156471	Retainer, Spring - Bipropellant Valve Pneumatic Actuator

List of Drawings (cont.)

Figure 26, Sheet 2 of 3

Report 8317-PI

<u>Drawing</u>	<u>Title</u>
1156472	Ring Half, Retainer - Bipropellant Valve Pneumatic Actuator
1156473	Cover, Spring - Bipropellant Valve Pneumatic Actuator
1156474	Potentiometer, Translatory, Valve Position Indicating
1156475	Guide, Pushrod-Self-Aligning, Bipropellant Valve Pneumatic Actuator
1156476	Retainer, Threaded - Pneumatic Actuator
1156477	Body, Fluid Passage Bolt - Cluster Fitting, Pneumatic Actuator
1156478	Bolt, Fluid Passage - Cluster Fitting, Pneumatic Actuator
1156479	Shim-Bipropellant Valve Pneumatic Actuator
1156484	Spring, Machined - Bipropellant Valve Pneumatic Actuator
1156485	Spring, Inner - Bipropellant Valve Pneumatic Actuator
1156486	Spring, Outer - Bipropellant Valve Pneumatic Actuator

List of Drawings (cont.)

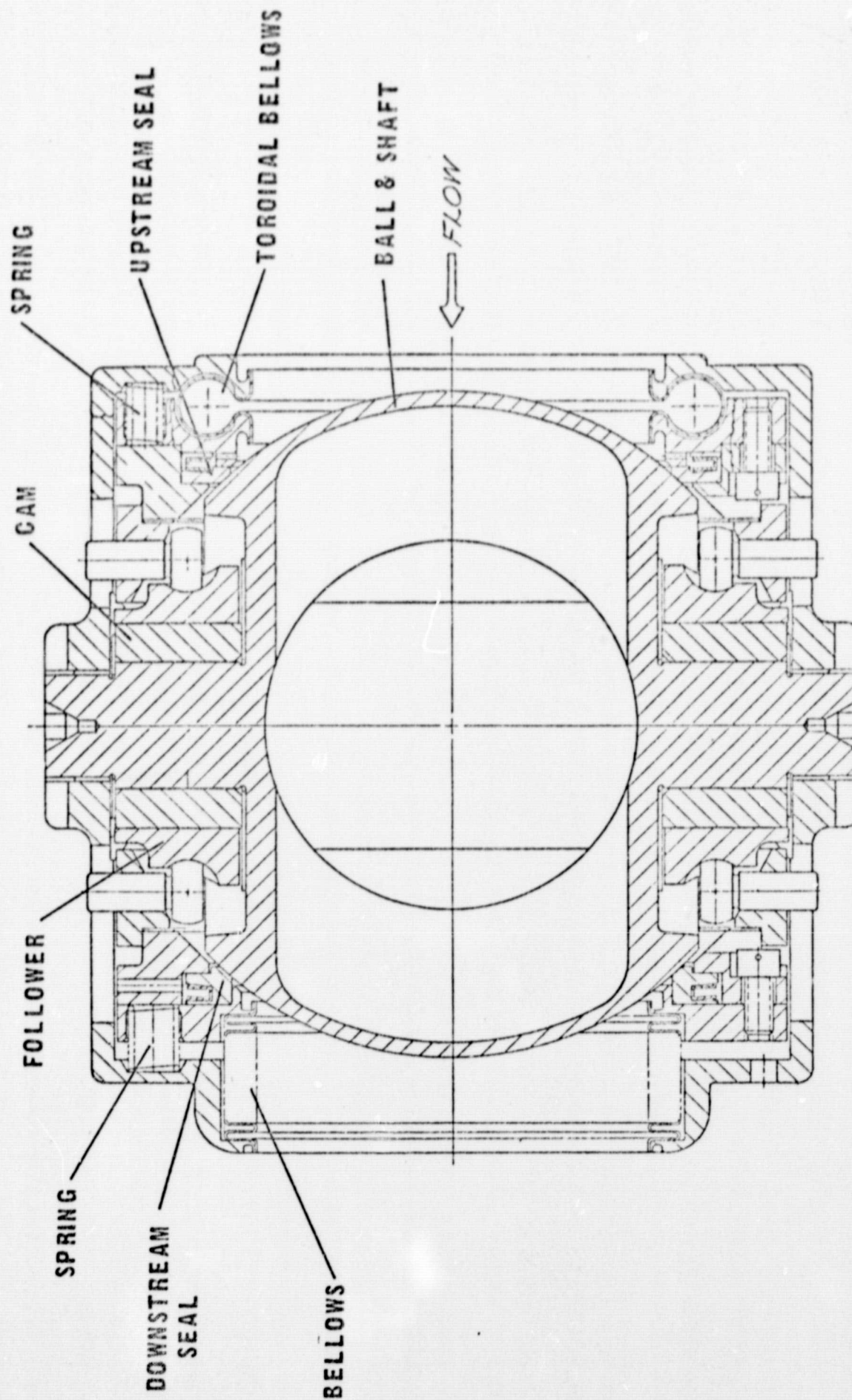
Figure 26, Sheet 3 of 3

Report 8317-PI

<u>Position</u>	<u>Cam Material</u>	<u>Follower Material</u>	<u>Seal Type</u>	<u>Bellows Type</u>
<u>Fuel</u>				
Upstream	Stellite 19	Stellite 19	No. 1 - Lift Off No. 2 - Lift Off	Toroidal Convoluted
Downstream	Stellite 19	Stellite 6B	No. 3 - Bending No. 4 - Lift Off	Convoluted Convoluted
<u>Oxidizer</u>				
Upstream	Stellite 19	Stellite 6B	No. 1 - Lift Off No. 2 - Lift Off	Toroidal Convoluted
Downstream	Stellite 19	Stellite 19	No. 3 - Bending No. 4 - Lift Off	Convoluted Convoluted

Two-Bore Valve Configuration

Figure 27



Toroidal Seal

Figure 28

Report 8317-PI

	<u>Face Width (in.)</u>	<u>Gage Dim. (in.)</u>	<u>Seal Load (lb)</u>	<u>Seal Lift (in.)</u>	<u>Cam Lift (in.)</u>	<u>Clearance (in.)</u>
<u>Fuel Seals</u>						
No. 1	0.075	1.178	96.0	0.010	0.014	0.004
No. 2	0.072	1.213	70.5	0.013	0.018	0.005
No. 3	0.075	1.212	98.0	0.013	0.018	0.005
No. 4	0.070	1.214	100.0	0.014	0.019	0.005
<u>Oxidizer Seals</u>						
No. 1	0.080	1.1785	110.0	0.017	0.022	0.005
No. 2	0.080	1.212	84.5	0.010	0.015	0.005
No. 3	0.070	1.212	91.5	0.012	0.017	0.005
No. 4	0.073	1.210	92.5	0.013	0.018	0.005

Ball Seal Data - Initial Buildup

Figure 29

Report 8317-PI

	<u>Face Width</u> (in.)	<u>Gage Dim.</u> (in.)	<u>Seal Load</u> (lb)	<u>Seal Lift</u> (in.)	<u>Cam Lift</u> (in.)	<u>Clearance</u> (in.)
<u>Fuel Seals</u>						
No. 1	0.070	1.180	86.0	0.009	0.014	0.005
No. 2	0.065	1.214	85.0	0.008	0.014	0.006
No. 3	0.075	1.214	85.0	0.012	0.018	0.006
No. 4	0.070	1.214	100.0	0.014	0.019	0.005
<u>Oxidizer Seals</u>						
No. 1	0.072	1.179	109.0	0.016	0.020	0.004
No. 2	0.074	1.213	70.0	0.011	0.017	0.006
No. 3	0.070	1.213	94.0	0.012	0.017	0.005
No. 4	0.076	1.212	78.0	0.015	0.019	0.004

NOTE: Number 1 and 2 fuel seals, filled TFE, were installed after the first 250 wet cycles. Fuel Number 3 seal was installed after the water flush. Fuel Number 4 was not replaced. All oxidizer seals were replaced after the water flush.

Ball Seal Data - Rebuild

Figure 30

Event	Seal No. 1			Seal No. 2			Seal No. 3			Seal No. 4			Shaft Seals		
	2	75	175	2	75	175	2	75	175	2	75	175	2	75	175
	(Test Pressure, psig)														
Acceptance Test	0	10	40	0	16	24	0	0	0	9	0	0	0	0	0
Post 50 Dry Cycles	0	10	30	0	25	45	0	0	0	10	0	0	0	0	0
Post 100 Dry Cycles	0	8	12.5	0	20	10	0	0	0	0	0	0	0	0	0
Post 200 Dry Cycles	0	14	30	0	52	52	0	0	0	0	0	0	0	0	0
Post 300 Dry Cycles	0	10	40	0	50	60	0	0	80	25	0	0	0	0	0
Post 500 Dry Cycles	0	0	30	0	35	20	0	0	0	0	0	0	0	0	0
Acceptance Test	0	0	52	0	0	0	0	0	10	66	0	0	0	0	0
Post 50 Dry Cycles	0	47	115	0	75	15	0	0	20	40	0	0	0	0	0
Post 100 Dry Cycles	0	0	0	0	30	0	0	0	0	5	0	0	0	95*	0
Post 200 Dry Cycles	0	8	54	0	10	4	0	0	0	3	0	0	0	0	0
Post 300 Dry Cycles	0	8	30	0	25	0	0	0	0	0	0	0	0	25*	0
Post 500 Dry Cycles	0	0	22	0	5	2.5	0	0	7	12.5	0	0	0	20*	2.5* 2.5* 5*
Leakage from upstream primary shaft seals.															
Fuel Valve Leakage Data (cc/Hour)															

Figure 31 (Sheet 1 of 2)

Event	Seal No. 1			Seal No. 2			Seal No. 3			Seal No. 4			Shaft Seals		
	2	75	175	2	75	175	2	75	175	2	75	175	2	75	175
	(Test Pressure, psig)														
Post 250 Wet Cycles	0	210	820	96	0	220	325	13	0	52	130	11	2	1	45
Post 500 Wet (Oxid Only)															
Post Fuel Half Rebuild	0	0	0	0	0	20	10	0	0	60	210	0	0	70	60
Post 1000 Wet Oxid - 500 Dry Fuel	0	120	360	0	0	70	92	0	0	177	312	0	0	74	36
Post 1500 Wet Oxid - 500 Wet Fuel	0	0	0	0	0	8	32	0	0	201	249	0	0	468	1200
Leakage from upstream primary shaft seals.															

Fuel Valve Leakage Data (cc/Hour)

Event	Seal No. 1			Seal No. 2			Seal No. 3			Seal No. 4			Shaft Seals		
	2	75	175	2	75	175	2	75	175	2	75	175	2	75	175
	(Test Pressure, psig)														
Acceptance Test	0	0	0	0	0	7	0	0	8	0	0	0	0	0	0
Post 50 Dry Cycles	0	0	0	0	0	33	0	0	17	18	0	0	0	0	0
Post 100 Dry Cycles	0	0	0	0	8	0	0	0	120	336	0	12	8	0	0
Post 200 Dry Cycles	0	75	260	0	12	28	0	0	125	420	0	11	5	0	0
Post 300 Dry Cycles	0	30	150	0	7	21	0	0	52	102	3	0	1	0	0
Post 500 Dry Cycles	0	0	23	0	10	25	0	0	68	168	0	10	90	0	0
Acceptance Test	0	0	7	0	15	15	0	0	0	0	0	18	12	0	0
Post 50 Dry Cycles	0	37	52	0	25	30	0	0	25	100	0	8	8	0	0
Post 100 Dry Cycles	0	17	40	0	0	0	0	0	23	70	0	20	20	0	0
Post 200 Dry Cycles	0	12	35	0	19	27	2	0	15	33	0	10	15	2	0
Post 300 Dry Cycles	0	50	50	0	15	25	0	0	15	32	0	20	12	0	0
Post 500 Dry Cycles	0	25	38	0	16	20	0	0	22	31	0	25	10	0	0

Oxidizer Valve Leakage Data (cc/Hour)

Figure 32 (Sheet 1 of 2)

Event	Seal No. 1			Seal No. 2			Seal No. 3			Seal No. 4			Shaft Seals		
	2	75	175	2	75	175	2	75	175	2	75	175	2	75	175
	(Test Pressure, psig)														
Post 250 Wet Cycles	0	0	0	0	0	1	0	0	0	7	0	0	0	0	0
Post 500 Wet Cycles (Oxid Only)	8	55	112	0	20	36	0	0	13	47	0	1	0	0	0
Post 1000 Wet Cycles	0	40	90	0	10	20	0	0	21	192	0	0	52	44	0
Post 1500 Wet Cycles	7.5	44	74	0	13	24	0	0	24	252	0	0	30	26	13

Oxidizer Valve Leakage Data (cc/Hour)

Figure 32 (Sheet 2 of 2)

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Event	Actuator	Initial Movement	Full Open	Event	Actuator	Initial Movement	Full Open
Acceptance Test	1 2	56 60	98 100	Post 300 Dry Cycles	1 2	60 63	102 100
Post 50 Dry Cycles	1 2	- -	- -	Post 500 Dry Cycles	1 2	61 63	104 103
Post 100 Dry Cycles	1 2	65 68	104 105	Post 250 Wet Cycles	1 2	59 64	153 145
Post 200 Dry Cycles	1 2	63 65	105 103	Post 500 Wet Cycles (Oxid Only)	1 2	77 68	143 131
Post 300 Dry Cycles	1 2	53 55	96 93	Post 1000 Wet Oxid - 500 Dry Fuel	1 2	63 65	152 138
Post 500 Dry Cycles	1 2	62 64	110 104	Post 1500 Wet Oxid - 500 Wet Fuel	1 2	58 65	173 157
Acceptance Test	1 2	60 65	105 105				
Post 50 Dry Cycles	1 2	69 75	108 108				
Post 100 Dry Cycles	1 2	67 66	108 106				
Post 200 Dry Cycles	1 2	58 60	99 102				

Actuator

Figure 33

Event	Actuator	Opening	Closing	Event	Actuator	Opening	Closing
Acceptance Test	1 2	0.745 0.525	0.380 0.600	Post 300 Dry Cycles	1 2	0.680 0.435	0.360 0.615
Post 50 Dry Cycles	1 2	0.765 0.525	0.375 0.600	Post 500 Dry Cycles	1 2	0.685 0.475	0.360 0.620
Post 100 Dry Cycles	1 2	0.790 0.570	0.350 0.550	Post 250 Wet Cycles	1 2	0.816 0.598	0.378 0.612
Post 200 Dry Cycles	1 2	0.722 0.490	0.380 0.595	Post 500 Wet Cycles (Oxid Only)	1 2	0.704 0.468	0.352 0.582
Post 300 Dry Cycles	1 2	0.685 0.465	0.385 0.610	Post 1000 Wet Oxid - 500 Dry Fuel	1 2	0.707 0.487	0.382 0.604
Post 500 Dry Cycles	1 2	0.725 0.508	0.400 0.590	Post 1500 Wet Oxid - 500 Wet Fuel	1 2	0.794 0.554	0.403 0.615
Acceptance Test	1 2	0.710 0.510	0.365 0.620				
Post 50 Dry Cycles	1 2	0.705 0.505	0.350 0.590				
Post 100 Dry Cycles	1 2	0.730 0.520	0.360 0.605				
Post 200 Dry Cycles	1 2	0.690 0.480	0.350 0.600				

Valve Travel Time (sec)

Figure 34

Event	Fuel No. 1 Seal	Fuel No. 2 Seal	Fuel No. 3 Seal	Oxid No. 1 Seal	Oxid No. 2 Seal	Oxid No. 3 Seal
Acceptance Test	120	129	57	45	128	129
Post 50 Dry Cycles	115	6	55	49		
Post 100 Dry Cycles	112	8	78	48	123	86
Post 200 Dry Cycles	104	123	52	40	131	86
Post 300 Dry Cycles	100	6	55	47	130	88
Post 500 Dry Cycles	98	17	48	49	136	84
Acceptance Test	108	100	62	79	60	55
Post 50 Dry Cycles	108	137	57	75	60	57
Post 100 Dry Cycles	105	145	54	76	60	52
Post 200 Dry Cycles	111	15	54	75	60	55
Post 300 Dry Cycles	112	115	54	70	60	53
Post 500 Dry Cycles	112	125	55	70	61	51

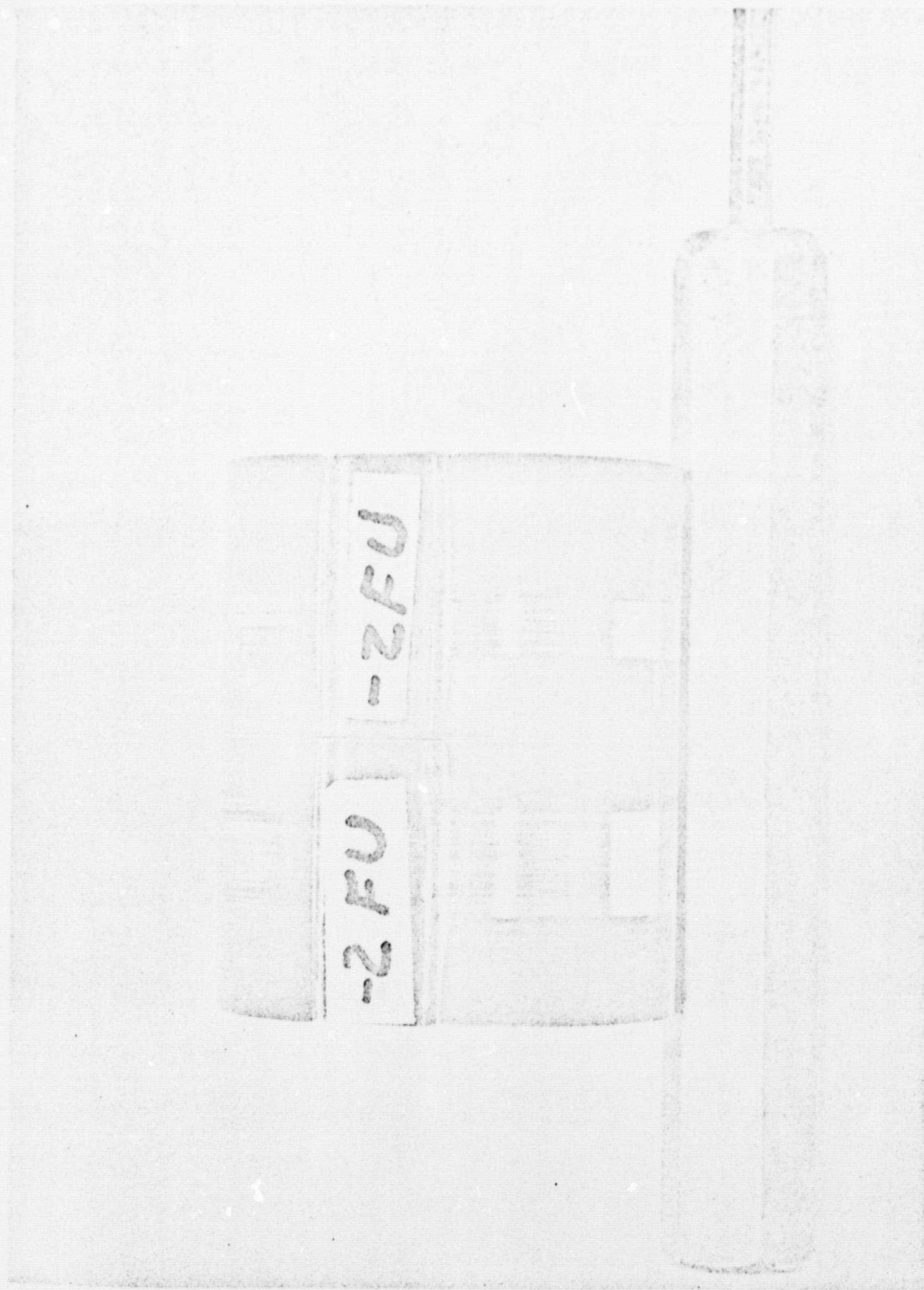
Ball Seal Back Pressure Relief Data (psig)

Figure 35 (Sheet 1 of 2)

Event	Fuel No. 1 Seal	Fuel No. 2 Seal	Fuel No. 3 Seal	Oxid No. 1 Seal	Oxid No. 2 Seal	Oxid No. 3 Seal
Post 250 Wet Cycles	95	169	41	66	43	29
Post 500 Wet Cycles (Oxid Only)	-	-	-	93	56	62
Post 1000 Wet Oxid - 500 Dry Fuel	90	50	60	87	45	27
Post 1500 Wet Oxid - 500 Wet Fuel	90	58	46	72	43	55

Ball Seal Back Pressure Relief Data (psig)

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Fuel Cams after Wet Cycle nesting

Figure 36



Oxidizer Followers after Wet Cycle Testing

Figure 37

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	Fuel Upstream		Fuel Downstream	Oxid Upstream	Oxid Downstream
Pre-Acceptance Test	Bearing	36	36	34	36
	Breakaway	71	50	36	43
	Run	21	28.5	14.5	23
Post Rebuild after Dry Cycles and Water Flush	Bearing	31.5	23	39	21.5
	Breakaway	36	33	64	40
	Run	21.5	20	36	21.5
Post Assembly after Alcohol Flush	Bearing	30	35	35	25
	Breakaway	35	37	47	30
	Run	25	12	40	17
Post 250 Wet (A-50) Cycles	Bearing	29	21	-	-
	Breakaway	86	114	-	-
	Run	78	100	-	-
Post 500 Wet Fuel Cycles and 1500 Wet		55.5	31	49	31
		92	123	68	62
		80	93	68	43

Torque, Cartridge Installed In Housing (in.-lb)

Figure 38

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	Face Width in.	Gage Dim. in.	Seal Load lb	Seal Lift in.	Cam Lift in.	Clearance in.
<u>Fuel Seals</u>						
No. 1	0.070	1.180	85	0.013	0.018	0.005
No. 2	0.072	1.212	83	0.012	0.012	0.0
No. 3	0.075	1.2185	90	0.008	0.013	0.005
No. 4	0.076	1.2113	80	0.019	0.022	0.003
<u>Oxidizer Seals</u>						
No. 1	0.077	1.178	86	0.008	0.010	0.002
No. 2	0.078	1.211	75	0.017	0.023	0.006
No. 3	0.065	1.2186	90	0.012	0.017	0.005
No. 4	0.078	1.2111	87	0.016	0.024	0.008

Ball Seal Data - Posttest

Figure 39

Appendix A

September 1966

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BIPROPELLANT VALVE IMPROVEMENT PROGRAM
APOLLO SERVICE PROPULSION SYSTEM

PHASE II TEST PLAN
SINGLE BORE BIPROPELLANT VALVE TESTING PROGRAM

PREPARED UNDER
CONTRACT NAS 9-8317

FOR

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FIGURE LIST

	<u>FIGURE</u>
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REFERENCE:

Report 8317-T, Test Plan, "Single Cartridge Testing Program,"
dated 25 March 1969

I. INTRODUCTION

This test plan outlines the Phase II test program for the SPS-VIP bipropellant valve assembly. Testing will be conducted on a single bore bipropellant valve of the P/N 1156400-1 configuration. The basic objective of this series of tests is to evaluate the cartridge assembly developed during the Phase I single position tests (Reference (a)) in an improved valve assembly. The valve assembly will consist of two single bore housings, one fuel and one oxidizer, with two cartridge assemblies installed in each bore. Valve actuation will be accomplished by use of two actuators of an improved design. Each actuator will operate one fuel ball and one oxidizer ball through a gear mechanism. Actuation pressure will be supplied to each actuator from a workhorse actuation system consisting of a solenoid operated 3-way pilot valve, an accumulator, and related plumbing and adapters. The assembled valve will represent one half of a complete SPS bipropellant valve assembly.

Primary test objectives will be to obtain the following:

- a. Additional ball seal and shaft seal cycle life data.
- b. A comparison between the bending seal design and the toroidal bellows seal design and selection of the better seal for the final valve design.
- c. Additional cam and cam follower cycle life data and selection of materials for the final valve design.
- d. Interaction of cartridge assemblies under rated flow conditions and the effect of cartridge replacement upon valve flow characteristics.
- e. An evaluation of the improved actuator design.
- f. An evaluation of the actuator-to-valve interface and the adequacy of the gear system for oxidizer valve and fuel valve relative position control.

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- g. Demonstration of the adequacy of the valve design under hot firing conditions and the establishment of required valve times.
- h. A determination of the adequacy of valve decontamination procedure.
- i. Overall design performance data under environmental conditions (temperature, vibration, and static firing).

In addition, during the course of the program, effort will be directed toward developing valve assembly techniques, the establishment of tooling requirements, and the design and fabrication of necessary assembly tooling in the form of shop aids.

Throughout the program, leak test data will be analyzed to establish the sealing capability of the valve design. Ball seals, actuation shaft seals, actuator piston seals, and piston shaft seals will be evaluated for ability to maintain a leak rate no greater than 10 cc per hour of gaseous nitrogen across each seal or each set of shaft seals. Actuation and timing data will be monitored and analyzed during the test program to determine when significant galling or wear of parts is encountered. All components will be subjected to a critical visual examination at the completion of testing with additional dimensional, chemical or physical analysis conducted as necessary. Pretest and post-test photographs will be taken of cams and cam followers as a means of documenting the effects of the test program.

II. TESTS TO BE PERFORMED

The tests to be conducted are tabulated below. It is intended to conduct these tests in the order listed. Any deviation from this test sequence will be documented in the applicable test request supplement. Functional and leak tests will be conducted during the life cycle test series and temperature tests at the test intervals described in Section V. Photographs will be taken of all test setups and of any damage incurred during testing. The valve test schedule is presented in Figure 1.

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<u>TEST NO.</u>	<u>TEST TYPE</u>	<u>REFERENCE PARAGRAPH</u>	<u>TEST LOCATION</u>
1	Acceptance	VA	MCL
2	Life Cycle, Series 'A) Decontamination Functional and Leak	VC3 VH VB	CEL TAC CEL
3	Temperature Functional and Leak	VD VB	CEL CEL
4	Vibration Functional and Leak	VE VB	VL CEL
5	Life Cycle, Series (B), Decontamination Functional and Leak	VC4 VH VB	CEL TAC CEL
6	Flow Test	VF	MCL
7	Acceptance Hot Firings Decontamination Functional and Leak	VA VG VH VB	MCL TAC TAC MCL
8	Posttest Examination	VI	MCL

LEGEND: MCL - Mechanical Controls Laboratory
 CEL - Components Evaluation Laboratory
 VL - Test Area "G" Vibration Laboratory
 TAC - Test Area "C"

III. HARDWARE

One bipropellant valve, P/N 1156400-1, will be utilized in the development evaluation. This assembly will include four cartridge assemblies, P/N 1156450, two actuator assemblies, P/N 1156460-1, and a workhorse actuation system. All assembly and disassembly will be accomplished in the Mechanical Controls Laboratory, Building 2002. Photographs will be taken of any damaged hardware after testing and of all cams and followers prior to test.

IV. HARDWARE CLEANLINESS

Cleanliness of all parts, subassemblies, cartridges, and valve assemblies will be maintained equivalent to AGC-46652, Level 1; however, they will not be certified as such. All assembly work will be accomplished in laminar flow benches by personnel wearing Nylon gloves. Balls, ball seals, and all soft goods will be cleaned in a detergent solution in accordance with AGC-46843. The remaining hard parts will be cleaned by a degrease wash in Trichlorethane, a GN_2 purge, and a final cleaning with Freon in a sonic cleaner. All parts, whether new or for reuse, except new parts which were cleaned to Level 1 by the supplier, will be cleaned in accordance with these procedures. All parts will be sealed in clean bags of Nylon "C" material until needed for assembly.

Test fluids will be filtered through 5-micron nominal, 18-micron absolute, filters prior to entering the valve. Valve assembly service ports which must be vented to atmosphere during testing will be protected by 5-micron nominal, 18-micron absolute, filters. Metallic protective closures will be installed on all ports when the valve is not in use.

V. DESCRIPTION OF TESTS

A. ACCEPTANCE TEST

The acceptance test will consist of a proof test, a functional test, a back pressure relief test, and a leak test as follows:

1. Proof Test

All proof pressure tests will be performed in a safe, remote area with gaseous nitrogen as the test fluid. No evidence of permanent distortion or damage will be acceptable. Pressure applied to more than one port at a time will be increased and decreased simultaneously at all ports.

V.A.1. Proof Test (continued)

a. Valve Housing Proof Test:

The inlet port, outlet port, and all purge ports in each propellant bore will simultaneously be pressurized to 800 ± 10 psig for five minutes minimum.

b. Upstream (No. 1) Seal Assembly Proof Test:

The two valve inlet ports will be pressurized to 315 ± 10 psig for five minutes minimum. No. 1 purge ports (OP-1, FP-1) open. (Figure 1 outlines the nomenclature to be used with the valve assembly to identify ball seal, purge, drain port, and cartridge locations).

c. Oxidizer No. 3 Seal Assembly Proof Test:

The No. 3 seal is the primary shut-off for the oxidizer bore during valve closure. The inlet port and No. 1 and No. 2 purge (OP-1 and OP-2) in the oxidizer bore will be pressurized to 500 ± 10 psig for five minutes minimum with the No. 3 purge port (OP-3) open.

d. Downstream (No. 4) Seal Assembly Proof Test:

The No. 4 seals must withstand the pressure generated when propellant trapped in the valve assembly due to a temperature increase (the No. 3, No. 2, and No. 1 seals relieve pressure into the inlet line). The inlet port and all three purge ports in each propellant bore will be pressurized to 500 ± 10 psig for five minutes minimum with the outlet ports open.

V.A.1. Proof Test (continued)

a. Valve Housing Proof Test:

The inlet port, outlet port, and all purge ports in each propellant bore will be simultaneously pressurized to 800 ± 10 psig for five minutes minimum.

b. Upstream (No. 1) Seal Assembly Proof Test:

The two valve inlet ports will be pressurized to 315 ± 10 psig for five minutes minimum with No. 1 purge ports (OP-1, FP-1) open. (Figure outlines the nomenclature to be used with the valve assembly to identify ball seal, purge drain port, and cartridge locations).

c. Oxidizer No. 3 Seal Assembly Proof Test:

The No. 3 seal is the primary shutoff of the oxidizer bore during valve closure. The inlet port and No. 1 and No. 2 purge (OP-1 and OP-2) in the oxidizer bore will be pressurized to 500 ± 10 psig for five minutes minimum with the No. 3 purge port (OP-3) open.

d. Downstream (No. 4) Seal Assembly Proof Test:

The No. 4 seals must withstand the full pressure generated when propellant trapped in the valve expands due to a temperature increase (the No. 3, No. 2, and No. 1 seals relieve pressure into the inlet line). The inlet port and all three purge ports in each propellant bore will be pressurized to 500 ± 10 psig for five minutes minimum with the outlet ports open.

e. Slave Actuation System-to-Valve Assembly Proof Test:

This test will verify the integrity of the system-to-valve joints, the actuators, and the actuation system. With the actuation system pilot valves open, the actuation system inlet port will be pressurized to 700 ± 10 psig for five minutes minimum. CAUTION: The bipropellant valve will open during this test.

2. Functional Test

The functional test will be performed in two segments, an actuation test and a timing test, as follows:

a. Actuation Test:

The intent of this test is to determine changes in the breakaway and operating torques of each actuator and ball valve assembly. With one pilot valve energized with 22-28 vdc, increasing gaseous nitrogen pressure will be applied to the actuation system inlet port until the applicable actuator piston moves to the open position as verified by the potentiometer trace. Pressures corresponding to initial actuator piston movement and full travel will be measured and recorded. This test will be conducted on both actuators. The pressures recorded will be analyzed to determine that the actuator is working properly and that no large change in cartridge torque or actuator friction has occurred.

b. Timing Test:

This test will verify the proper selection of timing orifices during valve assembly and will identify valve timing changes from test to test. The test fluid will be gaseous nitrogen. All potentiometers will be energized with 5.0 ± 0.1 vdc and the actuation system inlet port will be pressurized to 210 ± 10 psig. The two actuation system pilot valves will be energized simultaneously with 22-28 vdc. A minimum of one second later the two pilot valves will be de-energized simultaneously. The actuation system inlet pressure, the

V.A.2.b. Timing Test (continued)

energization and de-energization signals, and the potentiometer outputs will be measured and recorded.

The actual opening and closing times will be recorded for three complete opening and closing cycles. These data will be compared with previous tests to see if significant changes have occurred which would indicate any malfunction of the seals, cams, or actuators. Any discrepancies will be analyzed to determine if they are caused by the valve or by the test actuation system. If erratic timing is caused by the actuation system, then corrections will be made to the appropriate part to bring the valve actuation times back to the previous times. If the valve is found to have caused the shift in timing, then appropriate corrective design action will be taken.

Main valve opening and closing times shall be as tabulated below:

<u>Function</u>	<u>Actuator</u>	<u>Time (milliseconds)</u>	
		<u>New Valve</u>	<u>Used Valve</u>
Opening travel Time	1	725 $\pm$ 75	725 $\pm$ 100
	2	500 $\pm$ 75	500 $\pm$ 100
Closing travel Time	1	375 $\pm$ 25	375 $\pm$ 50
	2	575 $\pm$ 75	575 $\pm$ 100

3. Back Pressure Relief Test

This test will verify that ball seals No. 1, No. 2, and No. 3 in each propellant bore will relieve pressure when pressurized in the reverse (outlet port to inlet port) direction. Gaseous nitrogen pressure will be applied to the ball seal at a pressure rise rate of 5 psi per second maximum until a minimum of 60 cc/minute reverse leakage flow is measured past the seal being tested. The minimum

V.A.3. Back Pressure Relief Test (continued)

pressure that is required to give the specified leakage will be recorded. The No. 3 seal will be tested by pressurizing the outlet port and the No. 3 purge port and measuring leakage at the No. 2 purge port. To test the No. 2 seal, the outlet port and No. 3 and No. 2 purge ports will be pressurized and the leakage will be measured at the No. 1 purge port. The No. 1 seal will be tested by pressurizing the outlet port and all three purge ports and measuring leakage at the inlet port.

The back pressure relief pressure data will be analyzed to determine that the seals will relieve at a low enough pressure to insure that heat soakback on an engine would not cause pressures in the valve higher than the proof pressure.

4. Leakage Test

All leakage tests will be conducted using gaseous nitrogen as the pressurizing medium. Internal leakage will be measured using leakage test meter E-1015466 or equivalent water displacement type leakage measurement apparatus. When a water displacement type apparatus is used, water will be expelled from each measurement tube prior to measuring leakage. External leakage will be checked by use of liquid leak detector per Specification MIL-L-25567, Type I, or equivalent.

Ball seals and ball actuation shaft seals will be leak tested at 2, 75, 175, and 2 psig. The No. 1 ball seal will be tested by pressurizing the inlet port and measuring leakage from the No. 1 purge port. The inlet port and No. 1 purge port will be pressurized to test the No. 2 ball seal and leakage will be measured at the No. 2 purge port. The No. 3 ball seal will be tested by pressurizing the inlet port and No. 1 and No. 2 purge ports and measuring leakage from the No. 3 purge port. The inlet port and all three purge ports will be pressurized to test the No. 4 ball seal and leakage will be measured at the outlet port.

V.A.3. Back Pressure Relief Test (continued)

pressure that is required to give the specified leakage will be recorded. The No. 3 seal will be tested by pressurizing the outlet port and the No. 3 purge port and measuring leakage at the No. 2 purge port. To test the No. 2 seal, the outlet port and No. 3 and No. 2 purge ports will be pressurized and the leakage will be measured at the No. 1 purge port. The No. 1 seal will be tested by pressurizing the outlet port and all three purge ports and measuring leakage at the inlet port.

The back pressure relief pressure data will be analyzed to determine that the seals will relieve at a low enough pressure to insure that heat soakback on an engine would not cause pressures in the valve higher than the proof pressure.

4. Leakage Test

All leakage tests will be conducted using gaseous nitrogen as the pressurizing medium. Internal leakage will be measured using leakage test meter E-1015466 or equivalent water displacement type leakage measurement apparatus. When a water displacement type apparatus is used, water will be expelled from each measurement tube prior to measuring leakage. External leakage will be checked by use of liquid leak detector per Specification MIL-L-25567, Type I, or equivalent.

Ball seals and ball actuation shaft seals will be leak tested at 2, 75, 175, and 2 psig. The No. 1 ball seal will be tested by pressurizing the inlet port and measuring leakage from the No. 1 purge port. The inlet port and No. 1 purge port will be pressurized to test the No. 2 ball seal and leakage will be measured at the No. 2 purge port. The No. 3 ball seal will be tested by pressurizing the inlet port and No. 1 and No. 2 purge ports and measuring leakage from the No. 3 purge port. The inlet port and all three purge ports will be pressurized to test the No. 4 ball seal and leakage will be measured at the outlet port.

V.A.4. Leakage Test (continued)

The upstream ball actuation shaft primary seals will be leak tested during the No. 2 ball seal leak test and the downstream shaft primary seals (and upstream shaft primary seals in combination) will be leak tested during the No. 4 ball seal leak test. After completion of the No. 4 ball seal leak test, the inlet port, outlet port, and all three purge ports will be pressurized to 175 psig to check for external leakage of the housing seals.

The actuators will be leak tested at 50, 210, and 350 psig. The two actuation system pilot valves will be energized with 22-28 vdc and the leak test pressure will be applied to the actuation system inlet port. Internal leakage will be measured at the vent port in the actuator spring cover. The measured leakage will be the combined leakage of the piston head seal and piston shaft seals.

CAUTION: The bipropellant valve will open during the 210 and 350 psig tests.

The ball actuation shaft secondary seals will be leak tested by pressurizing the inlet port, outlet port, and all three purge ports in each bore and the primary shaft seal drain port to 10 ± 1 psig. Leakage will be measured at the actuator spring cover vent port.

Actuator external leakage will be checked by pressurizing the actuator spring cover vent port to 15-20 psig.

Each leak test will be conducted for a six minute minimum time period. All leakages will be measured and leakage rates will be recorded in cc/hour. Leakage test requirements will be as follows:

- a. Ball seal internal leakage: 10 cc/hr.
- b. Ball actuation shaft primary seals internal leakage: 10 cc/hr/shaft.

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V.A.4. Leakage Test (continued)

- c. Actuator piston and shaft seal internal leakage: 10 cc/hr/ piston or shaft.
- d. Ball actuation shaft secondary seals internal leakage: 10 cc/hr/shaft
- e. External leakage: None allowed

Additional ball seal leak tests at 240-245 psig will be conducted as specified below:

- a. Life Cycle Series A: Post 500 dry cycles
Post 1000 wet cycles
- b. Temperature: Posttest functional and leak.
- c. Vibration: Posttest functional and leak.
- d. Life Cycle Series B: Posttest functional and leak.
- e. Flow Test: Posttest functional and leak.
- f. Hot Firings: Posttest functional and leak.

The leakage data obtained from all tests will be analyzed to determine the effects of cycle life on leak rate, the effect of cam operation on leak rates, and to determine what leak rate can be consistently maintained throughout the life of a valve.

B. FUNCTIONAL AND LEAK TEST

A functional and leak test will be performed after each significant cycling test phase or after each environmental test to determine the effect of the test on valve operation. The test will consist of the functional test, back pressure relief test, and leakage test portions of the acceptance test. The test requirements will be considered flexible and may be changed or additional tests may be added by test request supplement if required by changing developmental conditions.

C. LIFE CYCLE TEST

1. Dry Cycle:

A dry cycle will consist of opening and closing the bipropellant valve with no propellants or pressure upstream of the valve utilizing the test setup and instrumentation of Figure 3. Each cycle will consist of simultaneously energizing the two pilot valves with 22-28 vdc with 210 ± 10 psig gaseous nitrogen pressure supplied to the actuation system inlet port, then, after a one second minimum delay, of simultaneously de-energizing the pilot valve.

2. Wet Cycle:

The wet cycle test will consist of cycling the bipropellant valve with propellants (N_2O_4 and A-50) supplied to the valve inlet port test plates at 178 ± 5 psig, using the test setup and instrumentation of Figure 4. Oxidizer will be supplied to the valve through 1/2 inch lines and the fuel supply will utilize 1/4 inch flow lines. Each wet cycle will consist of opening the bipropellant valve as described in the dry cycle test, permitting propellant flow for a minimum of one second, then closing the valve as described in the dry cycle test. After completion of the specified number of wet cycles, the valve will be drained, purged with gaseous nitrogen, and aspirated as required to dry the propellant bores. The valve will then be subjected to a zero psig leak test at the inlet port, outlet port, and three purge ports of each propellant bore. If leakage greater than 2 cc/15 minutes is measured during the zero psig leak test, the purge and aspiration will be repeated. After successful completion of the zero psig leak test, the functional and leak test will be performed.

c. Life Cycle Test (continued)

3. Test Series A

This test series will consist of 500 dry cycles followed by 1000 wet cycles. A functional and leak test will be conducted after 50, 100, 200, 300, and 500 dry cycles and after 64, 250, 500, and 1000 wet cycles. At the completion of the test series, the valve will be decontaminated as described in Paragraph VH and a functional and leak test will be conducted.

4. Test Series B

This test series will be conducted in the following sequence:

- a. 500 dry cycles (functional and leak test after 500 cycles).
- b. 4000 wet cycles (functional and leak test after 500, 1000, 2000, and 4000 wet cycles).
- c. 1000 dry cycles (functional and leak test after 500 and 1000 cycles).
- d. 5000 wet cycles (functional and leak test after 2500 and 5000 cycles).
- e. 1000 dry cycles (functional and leak test after 1000 cycles).
- f. 5000 wet cycles (functional and leak test after 5000 cycles).
- g. Repeat (e.) above.
- h. Repeat (f.) above.
- i. Repeat (e.) above.
- j. 20 wet cycles with an inlet pressure of 225^{+5}_{-0} psig followed by a functional and leak test.
- k. Decontamination per Paragraph VII followed by a functional and leak test.

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C.4. Test Series B (continued)

Completion of the life cycle testing, test series A and B, will result in a total of 5000 dry cycles and 20,020 wet cycles. Functional and leak test data will be available for the following increments in the dry cycle testing: 50, 100, 200, 300, 500, 1000, 1500, 2000, 3000, 4000, and 5000 cycles. For wet cycle testing, functional and leak test data will be available for the following increments: 64, 250, 500, 1000, 1500, 2000, 3000, 5000, 7500, 10,000, 15,000, 20,000 and 20,020 cycles.

D. TEMPERATURE TEST

The valve will be installed in a temperature chamber and the valve environmental temperature will be raised to $145 \pm 5^{\circ}\text{F}$. After the valve temperature has stabilized at $145 \pm 5^{\circ}\text{F}$ for a minimum of four hours, a functional and leak test will be conducted while the $145 \pm 5^{\circ}\text{F}$ valve temperature is maintained. A second functional and leak test will be conducted after the valve has returned to ambient temperature. Data from the two functional and leak tests will be compared to determine the effect of elevated temperature on valve operation. The test setup and instrumentation will be as shown in Figure 5.

E. VIBRATION TEST

The bipropellant valve will be subjected to the vibration levels representative of launch and engine firing as shown in Figure 6. The launch vibration test will be conducted first. The valve will be in the closed position for this test. The vibration will be applied for a duration of 15 minutes along each of the three mutually perpendicular axes. The oxidizer and fuel inlet ports will be pressurized with gaseous nitrogen to 163 ± 4 psig. Ball seal leakage will be monitored for each valve half throughout the vibration testing at the PO-1 and

C.E. Vibration Test (continued)

The firing vibration test will be conducted with the valve initially in the open position. The vibration will be applied for a duration of 20 minutes along each of the three mutually perpendicular axes.

The oxidizer valve inlet port will be pressurized with gaseous nitrogen to 143 ± 4 psig and the fuel valve inlet port will be pressurized with gaseous nitrogen to 132 ± 4 psig. During the last 5 minutes of vibration in each axis, the valve will be closed and the pressure at the valve outlet ports will be reduced to atmospheric pressure. Leakage past the valve will be monitored at the outlet port during the last three minutes of vibration in each axis using a water displacement method.

Both valve actuators will be operated during vibration testing with 210 ± 5 psig gaseous nitrogen. A functional and leak test will be conducted at the conclusion of the vibration testing to determine if valve operation has been affected by the vibration environment.

F. FLOW TEST

The valve flow characteristics will be investigated in water flow tests using the test setup and instrumentation of Figure 7. The valve will be functionally tested and the relative position of the balls in each bore established. The pressure drop across the fuel and oxidizer valve bores will then be established for approximately eight equal increments of controlling ball travel while the in-service ball is in its normal relative position based on functional data. Testing will be repeated with the upstream and downstream cartridges reversed to demonstrate the interchangeability of the cartridge design. Using the K_w curves established from the preceding tests, the flow characteristics of the bores will be examined and compared with the existing SPS valve bore characteristics. Since initial flow data indicates a significant increase in K_w will be obtained

F. Flow Test (continued)

for the improved valve, new orifices will be established to control steady state flow, and the valve timing will be increased to provide a similar start transient. The establishment of new valve times may require the use of the engine computer model. Once the proper timing is established, the valve will again be flow tested using the new valve times and flow control orifices. During this testing, the valve will be opened, steady state flow obtained, and the valve closed in order to simulate the flow conditions of an engine firing. Valve inlet and outlet pressures and the pressure in each bore between the ball valves will be monitored throughout the simulated firing along with flow rates and valve position. A total of five simulated firings will be accomplished. The valve will be dehydrated at the completion of flow testing and subjected to a functional and leak test.

G. HOT FIRING TESTS

At the conclusion of the cycling and environmental tests, the valve will be subjected to one hot firing test while installed on an Apollo engine. The test will be conducted on engine test stand C-3 in the Sacramento liquid rocket test area. Valve flow performance will be verified by a transient computer model run prior to the test firings. After the firing test, transient and steady state data will be compared to present SPS data to verify satisfactory valve-engine integration. The instrumentation utilized will be of the type and accuracy specified in AGC Specification 46846. The test parameters specified will be nominal conditions for an SPS engine firing as follows:

1. Propellant Temperature
 - a. Fuel: +40 to +100°F
 - b. Oxidizer: +40 to +100°F

G.1. Hot Firing Tests (continued)

2. Propellant Pressures

- a. Fuel (lockup): 178 ± 4 psia
- b. Oxidizer (lockup): 178 ± 4 psia
- c. Fuel (run): 169 ± 4 psia
- d. Oxidizer (run): 162 ± 4 psia

3. Mixture Ratio

$1.60 \pm .02:1$

4. Flow Rate

Oxidizer: 40.7 lbs/sec
Fuel: 25.3 lbs/sec

5. Test Duration

5 seconds nom.

6. Combustion Chamber

Steel

7. Valve Parameters Monitored

- a. Inlet pressures
- b. Cavity (between ball valve) pressures
- c. Valve position
- d. Valve temperature
- e. Actuation pressure
- f. Pilot valve position
- g. Propellant flow rates

H. DECONTAMINATION

At the conclusion of test series A and series B and the hot firing test series, the valve will be decontaminated to flush all traces of propellants from the bores. Flushing will be accomplished with Freon in the oxidizer valve and alcohol in the fuel valve. Decontamination will be conducted in accordance with the procedures of the Apollo engine assembly acceptance test specification AGC-46846. Detailed test procedures will be prepared prior to wet cycle

H. Decontamination (continued)

testing. The adequacy of the decontamination procedure will be determined as a part of the posttest examination.

I. POSTTEST EXAMINATION

During the posttest examination, the valve halves will be separated and the actuators removed. Each valve half will be examined and carefully disassembled. During the valve disassembly, parts will be examined for evidence of salting or similar indications that the decontamination procedure was not adequate. Parts will be checked with moist litmus paper as a further check of the decontamination process. Valve half disassembly will be accomplished as follows: Each cartridge will be removed and opening and closing torques measured. Seal and cam lifts will be measured during the torque measurement. The cartridge halves will be separated and seal loads measured. The cams and followers will be examined and photographed. Ball seal face width will be measured. Spring loads will be measured. The drive shafts and associated parts will be removed. All parts will be examined for evidence of damage, wear, or inadequate decontamination. Parts will be photographed as necessary to document their posttest condition. Dimensional inspection and materials analysis will be conducted as necessary based on the noted parts condition.

Both actuators will be tested to determine the pressure required for initial movement and to achieve full travel. Each actuator will be disassembled and examined for evidence of damage or wear. All parts will be photographed as necessary to document their posttest condition and additional inspection and materials analysis conducted as required.

I. Posttest Examination (continued)

All data obtained during posttest examination will be compared to similar assembly data and the variations, if any, analyzed for significance.

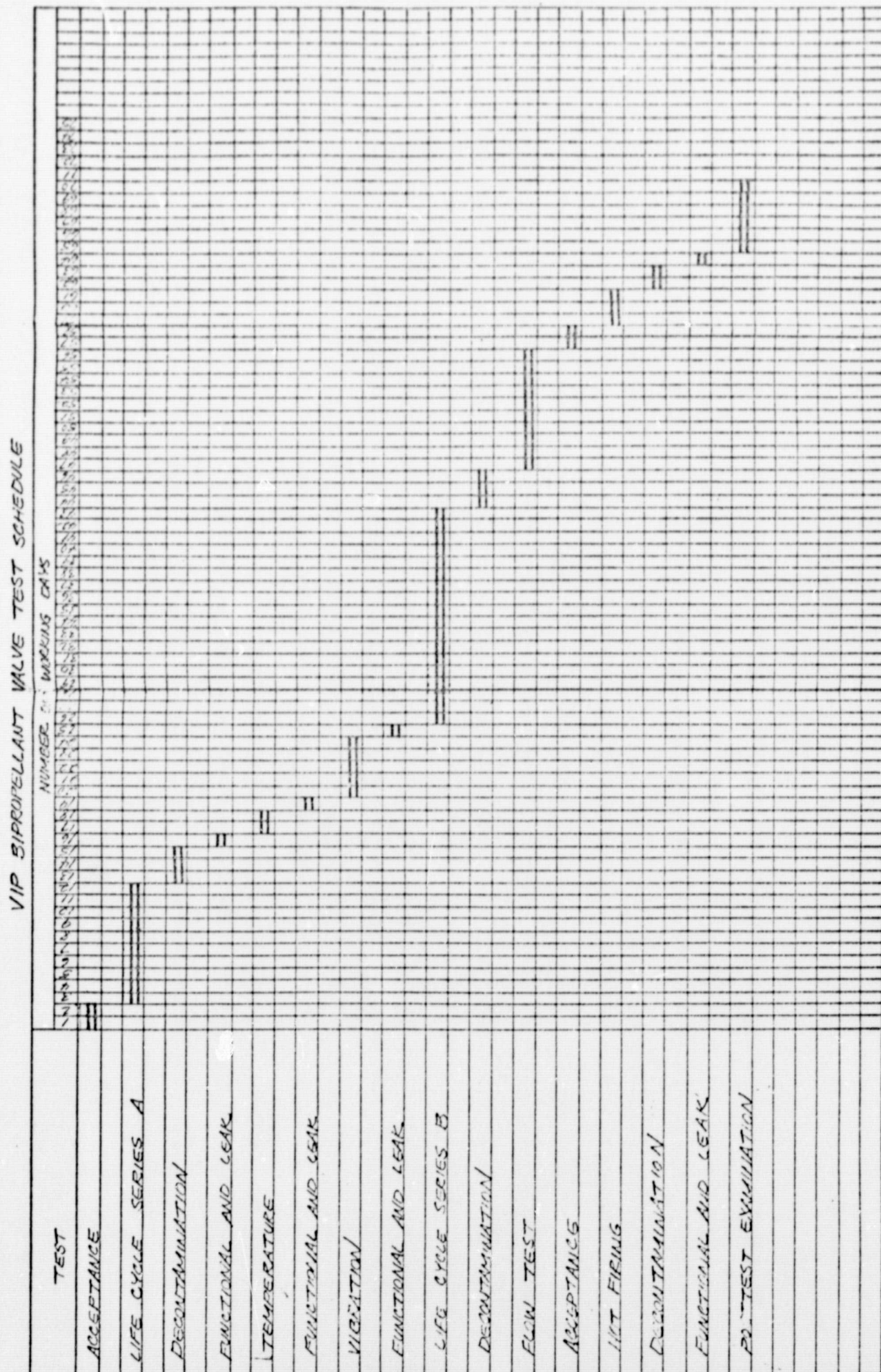
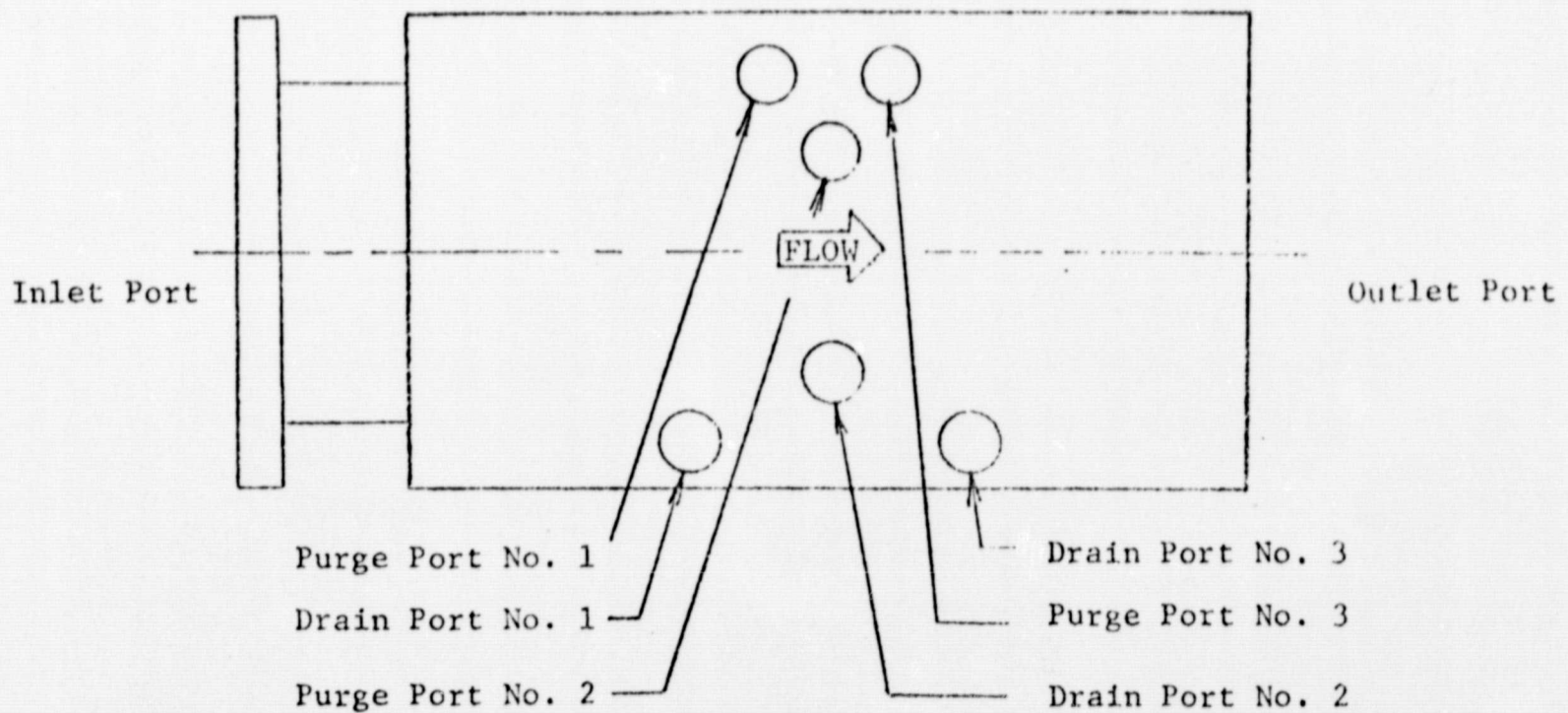


Figure 1



O = Oxidizer
F = Fuel

P = Purge Port
D = Drain Port

S = Seal

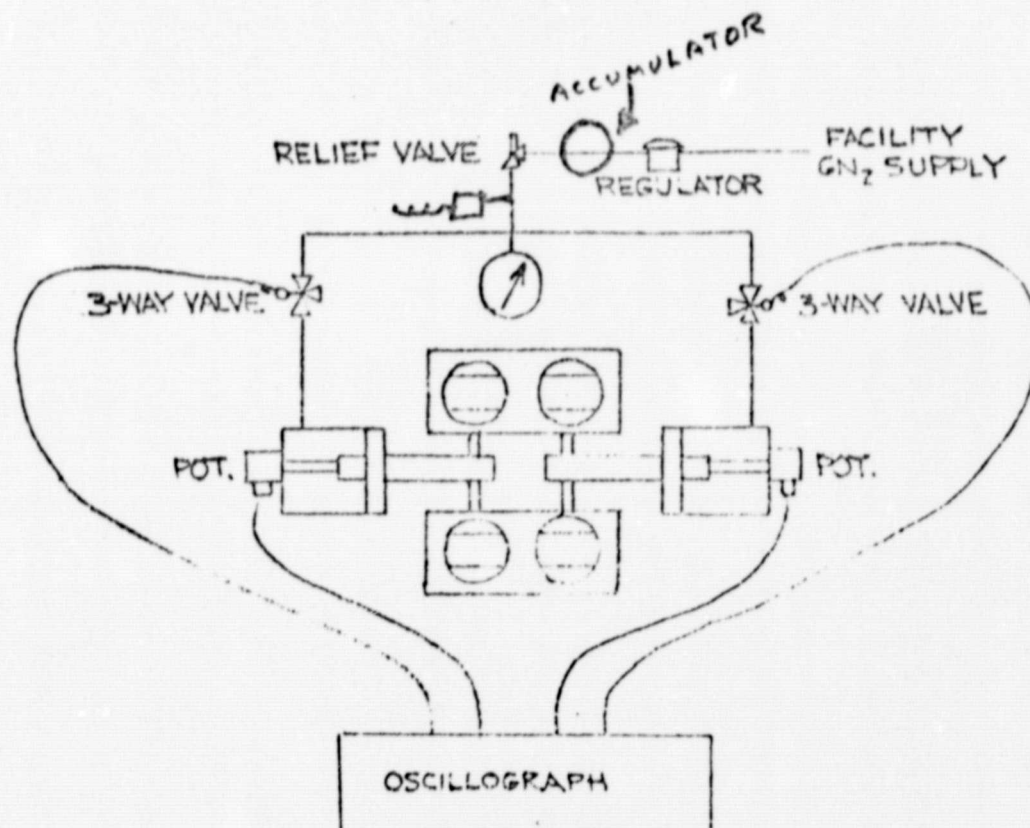
All parts or ball seals are numbered from the inlet port. Cartridges will be labeled "upstream" or "downstream."

Examples: OS-1 = lower oxidizer seal No. 1
FS-2 = upper fuel purge port No. 2

VIP Bipropellant Valve Nomenclature

Figure 2

DRY CYCLE TEST SCHEMATIC AND INSTRUMENTATION LIST



<u>PARAMETER</u>	<u>INSTRUMENTATION</u>
VALVE POSITION	POTENTIOMETER
ACTIONATION PRESSURE	VISUAL GAGE, 1/2% OF FULL SCALE ACCURACY
SOLENOID VALVE POS.	CURRENT TRACKER

Figure 3

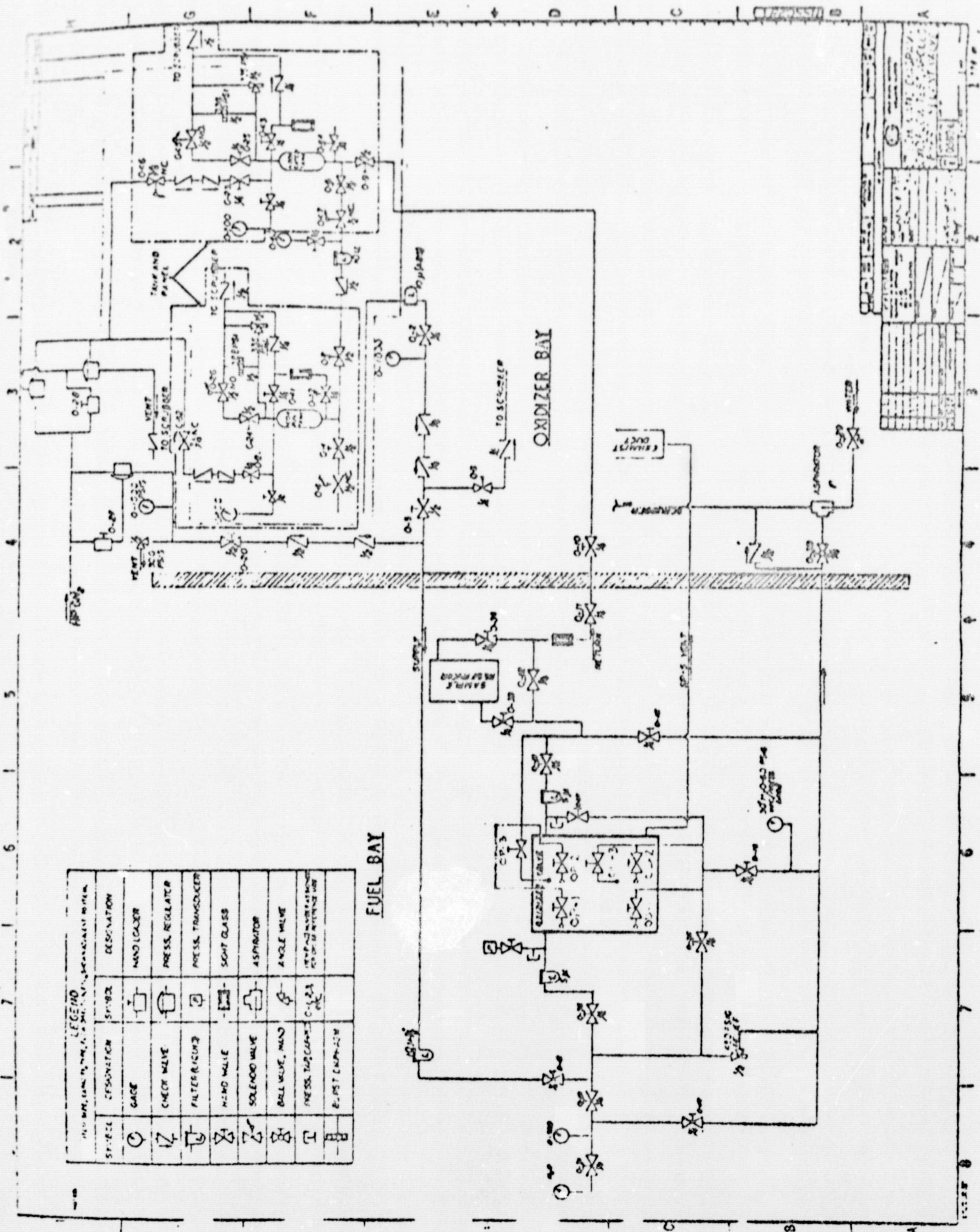
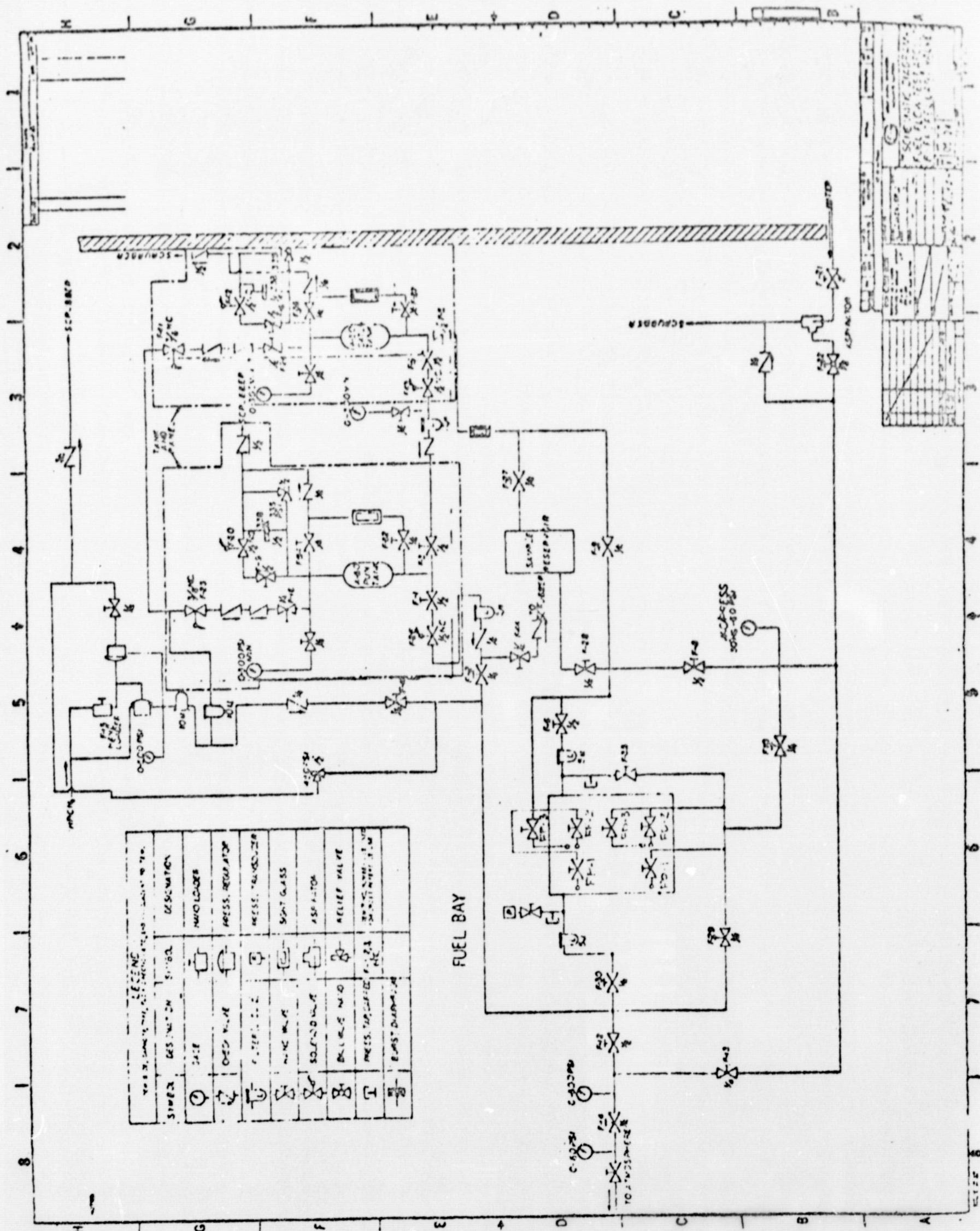


Figure 4, Sheet 1 of 3

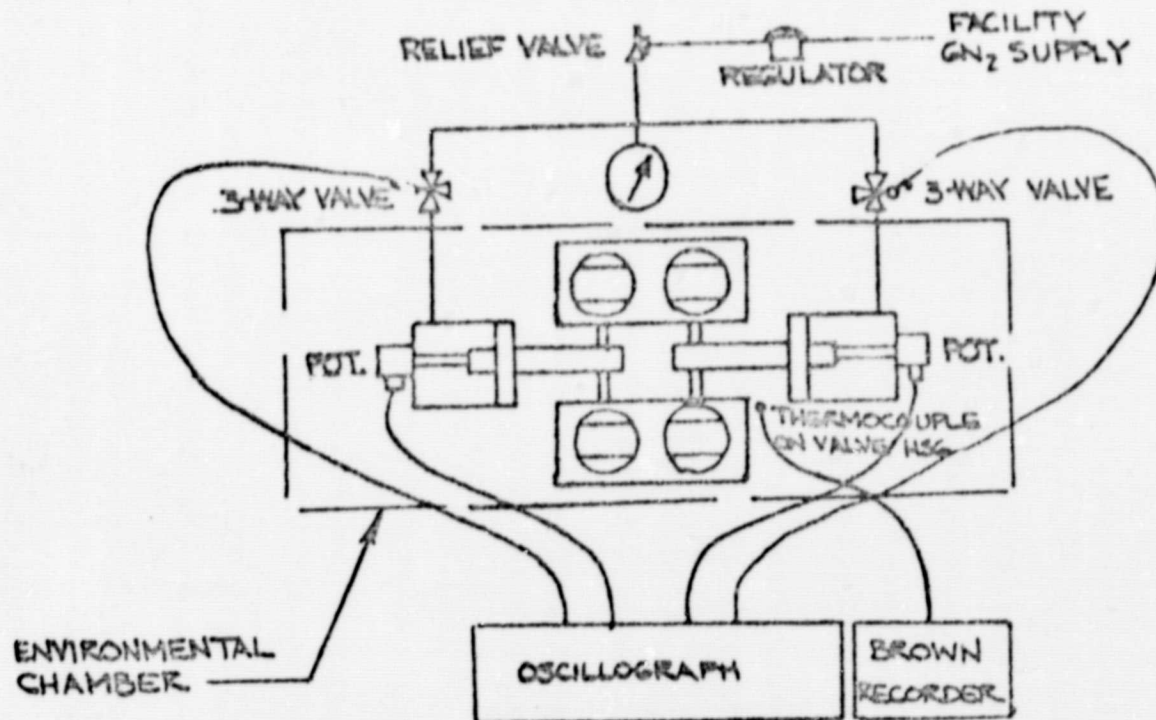


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INSTRUMENTATION LIST

<u>Parameter</u>	<u>Instrumentation</u>
Fuel inlet pressure	0-500 psi visual gage, 1/2% of full scale accuracy.
Oxidizer inlet pressure	0-500 psi visual gage, 1/2% of full scale accuracy.
Actuation pressure	0-500 psi visual gage, 1/2% of full scale accuracy.
Valve position	Potentiometer output recorded on oscillograph.
Solenoid valve position	Current trace recorded on oscillograph.

TEMPERATURE TEST SCHEMATIC AND INSTRUMENTATION LIST



<u>PARAMETER</u>	<u>INSTRUMENTATION</u>
VALVE POSITION	POTENTIOMETER
ACTION PRESSURE	VISUAL GAGE, 1/2% OF FULL SCALE ACCURACY
SOLENOID VALVE POS.	CURRENT TRACE
VALVE TEMPERATURE	BROWN RECORDER

Figure 5

VIP BIPROPELLANT VALVE VIBRATION REQUIREMENTS

LAUNCH SIMULATION

<u>FREQUENCY (CPS)</u>	<u>SPECTRAL DENSITY - G^2/CPS</u>
10-90	Linear increase from 0.0026 to 0.25 G^2/cps at 3 decibels per octave.
90-250	At .025 G^2/cps .
250-2000	Linear decrease from 0.025 to 0.003 G^2/cps at 3 decibels per octave.

Excitation level may be reduced by a multiplication factor of $10/w$ with a lower limit of .001 G^2/cps where w = weight.

FIRING SIMULATION

<u>FREQUENCY (CPS)</u>	<u>SPECTRAL DENSITY - G^2/CPS</u>
5 - 35	Linear increase from 0.00375 G^2/cps to 0.05 G^2/cps at 4 db per octave.
35 - 2000	Flat at 0.05 G^2/cps .

The following tolerances and limits shall be applied to the input spectrum:

Maximum Analysis Filter Bandwidth:

<u>Frequency Range (cps)</u>	<u>Bandwidth (cps)</u>
10 - 100	5
100 - 200	10
200 - 500	20
500 - 2000	50

Amplitude Tolerance: ± 3 db ± 100 percent or -50 percent on G^2/cps

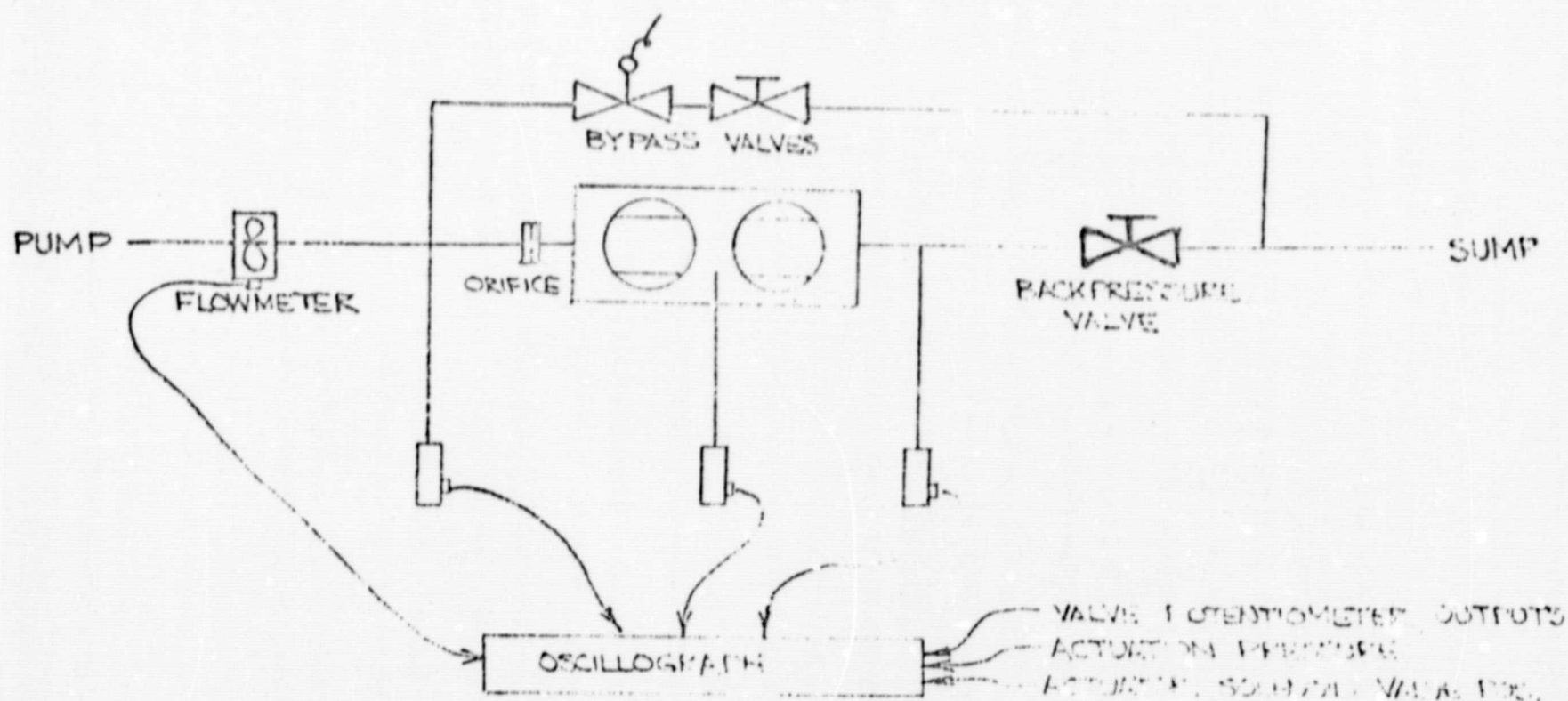
Frequency Tolerance: ± 5 percent or 2 cps, whichever is greater.

Electrical Noise Floor: The electrical noise floor of the vibration measurement system, throughout the zero to 2000 cps frequency range, shall not exceed 5 percent of the vibration test levels expressed in gravity rms.

Figure 6

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FLOW TEST SCHEMATIC AND INSTRUMENTATION LIST



NOTE: ONLY ONE HALF OF VALVE SHOWN. ACTUATORS DELETED FOR CLARITY

PARAMETER

FUEL INLET PRESSURE
FUEL CAVITY PRESSURE
FUEL OUTLET PRESSURE
FUEL HALF ΔP
OXID INLET PRESSURE
OXID CAVITY PRESSURE
OXID OUTLET PRESSURE
OXID HALF ΔP
FUEL FLOW RATE
OXID FLOW RATE
ACTUATION PRESSURES
VALVE POSITION
PISTON VALVE POSITION

INSTRUMENTATION

TRANSDUCER, 0-500 PSI
TRANSDUCER, 0-500 PSI
TRANSDUCER, 0-500 PSI
TRANSDUCER, 0-10 PSI
TRANSDUCER, 0-500 PSI
TRANSDUCER, 0-500 PSI
TRANSDUCER, 0-500 PSI
TRANSDUCER, 0-10 PSI
PISTON FLOWMETER
PISTON FLOWMETER
TRANSDUCER, 0-500 PSI
POTENTIOMETER
CURRENT TRACE

Figure 7