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NASA

ADVANCED TECHNOLOGY FOR SPACE SHUTTLE AUXILIARY PROPELLANT VALVES

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THE MARQUARDT COMPANY
13555 Saticoy Street
Van Nuys, California 91409

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

NASA Lewis Research Center
Contract NAS3-14349
R. Grey, Project Manager

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FINAL REPORT

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May 23, 1973

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Cleveland, Ohio
R. Grey, Project Manager

FOREWORD

This report is submitted by The Marquardt Company in accordance with the requirements of NASA Contract NAS3-14349. The work was administered by the NASA Lewis Research Center, Cleveland, Ohio, with Mr. R. Grey as the NASA Technical Project Manager.

This program was performed by the engineering department of The Marquardt Company at the Van Nuys facility. The Project Manager was Mr. H. Wichmann. Other contributors to this program were Messrs. D. Phillips, A. Malek, I. Dickens, E. Benz, R. Alger, R. Dunlop, W. Newhouse and R. Dickinson.

ABSTRACT

Valves for the gaseous hydrogen/gaseous oxygen Shuttle Auxiliary Propulsion System are required to feature low leakage over a wide temperature range coupled with high cycle life, long term compatibility and minimum maintenance. In addition, those valves used as thruster shutoff valves must feature fast response characteristics to achieve small, repeatable minimum impulse bits. These valve technology problems were solved by developing unique valve components such as sealing closures, guidance devices, and actuation means and by demonstrating two prototype valve concepts. One of the prototype valves was cycled over one million cycles without exceeding a leakage rate of 27 scc's per hour at 450 psia (310. N/cm^2) helium inlet pressure throughout the cycling program.

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SUMMARY

The objective of NASA Contract NAS3-14349 was the development of valve technology for pulse modulated gaseous hydrogen/gaseous oxygen thrusters for the Space Shuttle Auxiliary Propulsion System. These thruster valves are required to operate for one million cycles over a ten-year period with zero maintenance, very fast response, very low leakage, and over a wide temperature range. The program included the tradeoff of various valve concepts and their subcomponents; conceptual design, analysis, fabrication, and test evaluation of sealing closures; design layouts of candidate flightweight valves; detail design, fabrication, and test evaluation of two test fixtures featuring a variety of valve subcomponents; and culminated in the successful demonstration of one million cycles with one of these test fixtures. A design layout of the recommended flightweight valve design was also prepared.

The initial part of the program included two parallel tasks entitled, "Valve Subcomponent Analysis" and "Conceptual Design and Sealing Closure Evaluation." The valve subcomponent analysis and conceptual design task served to identify potential valve concepts and their subcomponents such as guidance devices, actuators, linkages, and sealing closures. Since the ability to meet the 100 scc per hour of helium at 450 psia (310.3 N/cm^2) inlet pressure maximum leakage requirement over a wide temperature range (-260 to + 390° F) (111 to 472°K) under fast response conditions and, therefore, under high impact loads for one million cycles was considered a severe requirement, an analytical and experimental sealing closure evaluation program was performed. Ten (10) sealing closure concepts featuring a wide variety of materials such as polyimides, gold plating, Inco 718, ceramics, teflon, and Custom 455, and including both spherical and flat sealing closure interfaces were conceived. Of these ten (10) concepts, seven (7) were selected for fabrication and test evaluation in the Rapid Screening Tester which was specifically designed for this program. The Rapid Screening Tester fully simulated valve guidance techniques by featuring metallic flexure guidance and had a stroke capability of one inch. The Rapid Screening Tester further had the capability of varying both static and dynamic loads at the sealing closure interface to simulate actual valve conditions. Sealing closure interface criteria such as static load, dynamic load, surface finish, maximum allowable scrubbing distance during mating, etc. were generated by means of analytical leakage model. Sealing closure evaluation tests with the Rapid Screening Tester included cycling of each sealing closure to 100,000 cycles at -260°F (111°K). Four (4) of the seven (7) sealing closures successfully passed the sealing closure screening tests and were considered suitable for incorporation into the test fixture designs of this program. These four (4) sealing closures were identified as a flat polyimide seat, spherical polyimide seat, flat tungsten carbide seat, and gold-plated lip seat. Both the flat polyimide seat and the gold-plated lip seat were subsequently incorporated into the test fixtures that were evaluated

during this program. Also, the analytical leakage model was updated at the end of the sealing closure evaluation program to agree with the experimental data obtained during this program.

During the initial months of the Advanced Technology for Space Shuttle Auxiliary Propellant Valves Program, valve subcomponent analysis and conceptual design for application to both a 20 psia (13.8 N/cm^2) and a 400 psia (276 N/cm^2) inlet pressure were performed. However, as a result of other gaseous hydrogen/gaseous oxygen propulsion system programs sponsored by NASA, it became apparent that the 400 psia (276 N/cm^2) inlet pressure system was the preferred application and consequently valve design layouts and test fixture evaluation performed during this program consisted of only those concepts that were directly applicable to the high pressure requirement. The valve subcomponent analysis and conceptual design concluded that a poppet-type valve utilizing the gaseous propellant supply pressure for actuation was the most promising type of concept for this application. Within this type of concept, several arrangements are possible such as double pressure actuation, pressure actuation to open and spring return, or vent to open and spring return, and all of these with or without pressure balancing of the poppet. These possible arrangements were explored in more detail and preliminary design layouts of three of these concepts were prepared. Two of these concepts were finally chosen for detail evaluation and detail design drawings of test fixtures of these concepts were subsequently accomplished.

Two separate test fixtures were detail designed, fabricated, and test evaluated to permit a comparison of various valve subcomponents such as guidance devices -- flexures vs. sliding fits, dynamic seals -- teflon jacketed sliding seals vs. bellows, static seals -- teflon jacketed seals vs. welded joints, and sealing closures. The test fixtures were designed such that the moving elements were of a flightweight configuration, but the housings were boilerplate and featured several joints which could be readily disassembled to permit inspection of all inner parts of the valve assembly.

Each of the two test fixtures was cycled 100,000 cycles with gaseous nitrogen at an operating pressure of 400 psia (276 N/cm^2) and over a temperature range of -260 to +390^oF (111 to 472^oK). Leak checks of all static and dynamic seals, as well as response tests and pressure drop tests, were performed periodically to determine test fixture performance degradations. Both test fixtures successfully achieved the 100,000 cycle requirements. Post test inspection and teardown of these test fixtures did, however, indicate several problem areas. These included the wear-out of the sliding teflon jacketed dynamic seals, excessive leakage of the pilot valves, and the loosening of a locknut and some resultant damage from this loosening in one of the test fixtures. However, sealing closure leakage performance, which was considered the most severe requirement, was excellent for both of these test fixtures.

Upon completion of the test fixture evaluation program, additional funding was received from NASA to permit the extended life cycling of the more promising of the two test fixtures. During this extended life cycling, a bellows as well as a welded static seal failure and some minor operational problems with the pilot valves, were experienced. However, after these items were repaired, over one million cycles were demonstrated successfully with the selected test fixture. Maximum leakage rate through the main valve during this cycling program never exceeded 27 scc's per hour of helium at 450 psia (310.3 N/cm^2) inlet pressure.

Based upon the test results of the test fixture evaluation program and the extended cycling program, a final flightweight valve design layout was prepar d.

In conclusion, the Advanced Technology for Space Shuttle Auxiliary Propellant Valves Program successfully generated and demonstrated the technology required by valves featuring very low leakage over a wide temperature range in combination with fast response and very high cycle life. The program further resulted in the demonstration of components which, through a choice of materials and design concepts, appear to have the capability of meeting the ten-year life and zero maintenance requirement of the Space Shuttle.

INTRODUCTION

The valve technology program described in this report was performed in support of the Space Shuttle Program of the National Aeronautics and Space Administration. The Space Shuttle vehicle is designed to provide low cost transportation to earth orbit to support a variety of missions, including logistic resupply of a space station. To achieve maximum cost effectiveness, the Space Shuttle will be designed for up to one hundred (100) flights (reuses) over a ten-year operational life time and be capable of relaunch within two weeks after earth landing. The system will be designed to minimize required post flight refurbishment, maintenance and checkout, and for simplicity and ease of maintenance when required. An Auxiliary Propulsion System (APS) is required for attitude control of the orbiter stage during all phases of the mission. As originally conceived, the APS was to utilize the hydrogen/oxygen propellant combination. At the outset of the program described in this report, both low pressure (20 psia) (13.8 N/cm^2) and high pressure (400 psia) (276 N/cm^2) gaseous propellant operation was under consideration. The APS studies have indicated a wide range of engine duty cycles from small pulse widths to long continuous firings. The pulse mode operation of rocket engines of the size (1500 lbs thrust) (6672 N) propellant phase, and service life that is required for the APS had never before been accomplished. Therefore, the valve design philosophy and accepted design practices of past applications had to be extrapolated and new approaches developed for the propellant valves required to control the pulsing of the engines to achieve the high cycle life, low leakage, and fast response requirements of the Space Shuttle APS.

The performance requirements of the APS valves are presented in Table I. During the performance of the Space Shuttle Auxiliary Propellant Valves Program (NAS3-14349) described in this report, it became apparent that the high propellant pressure auxiliary propulsion system was more promising that the low propellant pressure approach. Consequently, the work on the low pressure valve concepts was terminated after initial tradeoff studies and sealing closure evaluations and the work was subsequently concentrated on the high pressure system. The program, as originally contracted by NASA, included the demonstration testing of two prototype valves to 100,000 actuations each. At the completion of these 100,000 cycle tests, the NASA-Lewis Research Center provided additional funding to permit life cycle testing of one of the prototype valves through one million cycles. Thus, it was possible to demonstrate exceptionally high cycle life capability.

The Space Shuttle Auxiliary Propellant Valves Program consisted of five (5) principal tasks: an extensive valve subcomponent analysis and conceptual design tradeoff study; the preparation of valve preliminary design layouts; the design,

TABLE I-PERFORMANCE REQUIREMENTS

1.	Valve Type	Single or Bipropellant
2.	Propellants	Hydrogen Oxygen
3.	Operating Temperature Range	200°R to 850°R (111 to 472°K)
4.	Propellant Temperature Range	
	Hydrogen	200°R to 800°R (111 to 444.5°K)
	Oxygen	250°R to 800°R (139 to 444.5°K)
5.	Propellant Pressures at Valve Inlet	
	High Pressure System	400 ± 50 psia (276 ± 34.5 N/cm^2)
	Low Pressure System	20 ± 5 psia (13.8 ± 3.5 N/cm^2)
6.	Pressure Drop (Maximums):	
	High Pressure	
	Fuel	5 psi at 0.69 pps and 540°R (3.5 N/cm^2 at 0.313 Kg/sec and 300°K
	Oxidizer	5 psi at 2.76 pps and 540°R (3.5 N/cm^2 at 1.25 Kg/sec and 300°K)
	Low Pressure	
	Fuel	1 psi at 1.14 pps and 540°R (0.69 N/cm^2 at 0.52 Kg/sec and 300°K)
	Oxidizer	1 psi at 2.86 pps and 540°R (0.69 N/cm^2 at 1.3 Kg/sec and 300°K)
7.	Opening and Closing Response	10-15 milliseconds; total response less than 30 ms signal to open
8.	Internal Leakage	100 scc/hr. with Gaseous Helium at operating pressure and temperature for Items 4 and 5
9.	External Leakage	1 x 10^{-6} scc/sec with Gaseous Helium at operating pressures and temperature per Items 4 and 5
10.	Operating Life (Goal)	1,000,000 Cycles and 10 Years
11.	Maintenance	There shall be no maintenance of components during the design life period.
12.	Size and Weight	Design to minimum
13.	Proof Pressure	1.5 Times Operating Pressure
14.	Burst Pressure	1.33 Times Proof Pressure
15.	Failure Criteria	The valve shall fail safe close under normal operating conditions.

fabrication, and the thorough feasibility testing of sealing closures and subcomponents; the preparation of detailed flightweight valve design [illegible]ts based on the established design criteria and the test results from the previous tasks; and the preparation of various reports. This program approach is shown schematically in Figure 1.

Task I consisted of the conceptual design and analysis of valve subcomponents such as fluid shutoff devices, actuators, linkages and related supporting parts. These devices were sized for application to the gaseous hydrogen and gaseous oxygen shutoff valves for thrusters operating at 1500 lbs thrust and at inlet pressures of either 20 psia (13.8 N/cm^2) or 400 psia (276 N/cm^2). The types of shutoff devices and supporting parts considered include the following subcomponents:

Shutoff Devices		Supporting Parts
Poppet Blade or Gate Diaphragm Ball Butterfly Spool	with	Hard and soft seals Sliding, flexure and ballbearing guidance Single piece and slotted concepts 60° and 90° rotation Impact dampers Retractable seals and force loaded seals, Soft and Metallic dynamic seals Balanced and unbalanced concepts

The actuators, actuator details, and linkages that were analyzed during the subject program consisted of the following devices:

Actuators		Actuator Details
Solenoid Linear electric torque motor Rotary electric torque motor Double pressurized piston Spring return piston	including	Sliding flexure and ballbearing guidance Impact dampers Bellows, diaphragms and dynamic seals Using propellant supply pressure Using 3000 psi pneumatic or hydraulic supply pressure
Vane-type rotary actuator		

With Linkages

Bell crank or cam

Ballbearing or friction screw

Rack and pinion

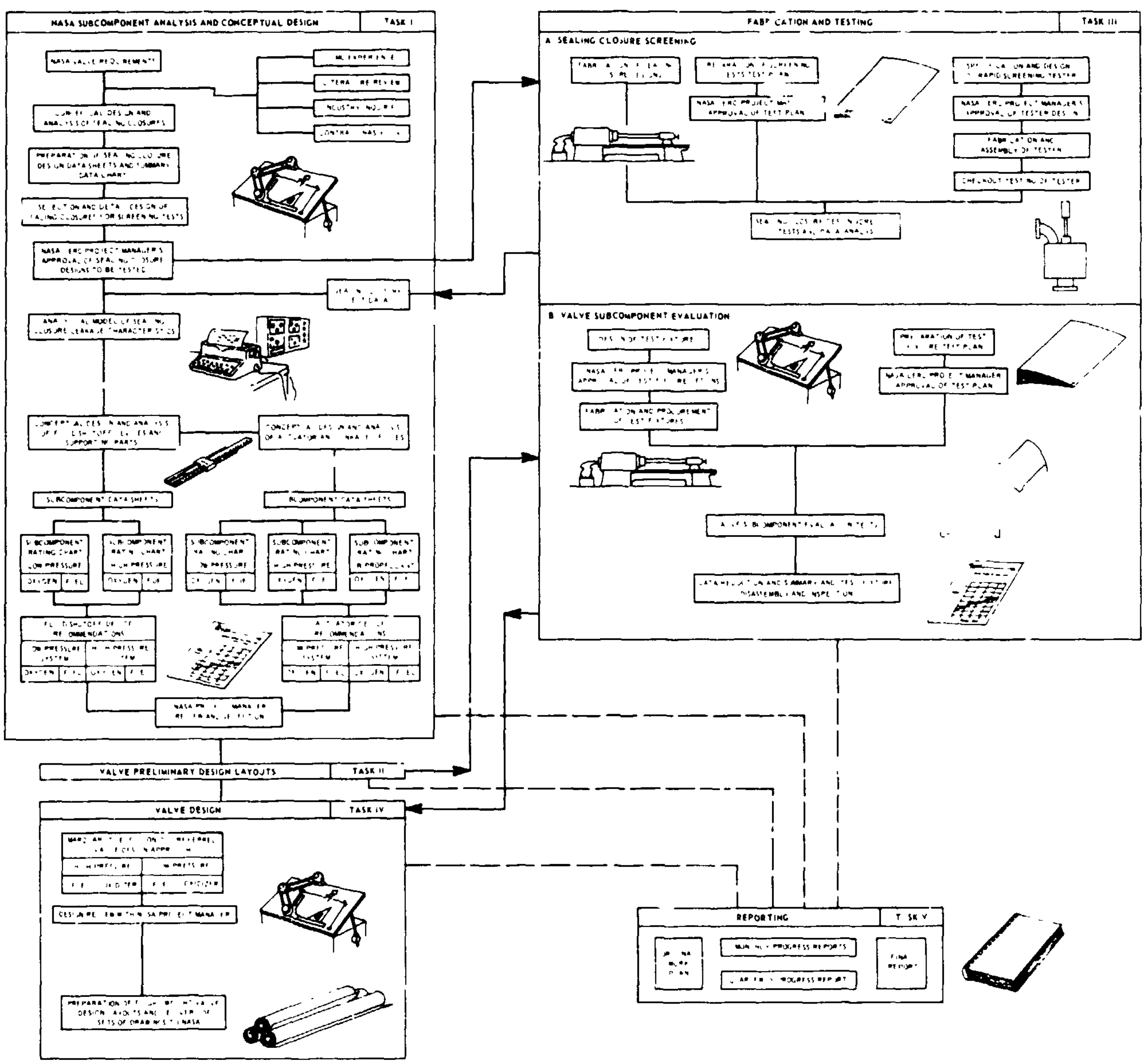


Figure 1 - APS Propellant Valve Technology Program Plan, Flow Diagram

Due to the severity of the leakage requirement over the wide temperature range and particularly at the elevated temperature of 850°R (472°K), and also to limit the number of tradeoffs which had to be accomplished, Task I of the program included the use of data from a series of sealing closure screening tests. To select the sealing closures to be subjected to these screening tests, a sealing closure tradeoff study was performed init ly which compared such characteristics as stroke, moving mass, seating force or torque, actuator force or torque required to move the closure within the required 15 to 20 millisecond response, predicted leakage rate, cycle capability, and filtration requirement. In addition, a qualitative point rating system utilizing a scale of 1 to 10 points was established to evaluate test parameters such as materials compatibility, pressure and venting hazards, reliability to operate at maximum and minimum temperature, degree of fabrication, inspection and assembly difficulty, contamination sensitivity, etc. Based on this criteria, the sealing closures subjected to the screening tests were selected. Subsequently, the data from the screening tests were utilized to update an analytical leakage characteristics model for each promising sealing closure and these sealing closures were then further considered during the Task I tradeoff study.

Once the most promising sealing closures were selected, the conceptual design and analysis tradeoff study considered various fluid shutoff devices and supporting parts, actuators and linkages which were applicable to the particular sealing closure design and a tradeoff of these devices was performed. At the conclusion of Task I, two fluid shutoff devices and two actuator concepts for each propellan ere selected.

Task II consisted of the preparation of hydrogen/oxygen valve preliminary design layouts incorporating the two selected configurations of fluid shutoff devices and associated actuators for the APS valves. These preliminary designs were scaled drawings which completely defined the valve concept and which were sufficient to permit the detailed design of critical valve subcomponents.

Task III consisted of two parts, namely: a sealing closure screening test phase and the valve subcomponent evaluation test phase. The sealing closure screening testing supported the conceptual design and analysis tradeoff performed during Task I. The valve subcomponent evaluation consisted of the testing of complete valving units of fluid shutoff and associated actuator concepts which featured the internal dimensions and mechanisms of the valve preliminary designs. Testing included cycling to 100,000 cycles and over the temperature range of 200° to 850°R (111 to 472°K), with numerous functional checks performed during the cycling. Subsequently, one prototype valve was cycled for one (1) million cycles at ambient temperature.

Task IV consisted of a review of the experimental valve subcomponent data d the selection of a preferred valve design approach for the hydrogen and oxygen valve based on both the experimental data and the analysis and conceptual design performed during the early part of the program. The experimental data and the preferred valve design approaches were then discussed at a design review meeting. At this design review, the optimum valve design was selected and a flightweight design layout of this concept was subsequently prepared.

The Space Shuttle Auxiliary Propellant Valves Program served to demonstrate new sealing closure concepts and guidance techniques, and that by properly combining these valve subcomponents, it is indeed possible to demonstrate shut-off valves featuring high cycle life and high response while reliably maintaining low leakage characteristics.

SEALING CLOSURE EVALUATION

The reliability of a sealing closure in any fluid shutoff valve depends on the severity of its allowable leakage requirement for a given number of closing cycles and the environment in which it must operate. Therefore, one of the first program tasks was to establish the design criteria for the sealing closure. This task consisted of a valve sealing closure which would best meet the stringent performance tradeoff study to select promising valve seal enclosures and a rapid screening test program to obtain actual performance data with these promising sealing enclosures.

To assure that the most advanced state-of-the-art of valve technology was being applied during the performance of this program, a special effort was made to review all available reports on valve technology and to contact various Government agencies and vendors for up-to-date information. A list of pertinent reports that were reviewed is presented in the section entitled, "References."

Sealing Closure Sizing

To facilitate the prediction of various performance and physical characteristics of the valves for the specific applications, certain design data are combined to yield the flow factor ($C_d A_t$) of each valve. This value is indicative of the valve's required flow area to meet the pressure drop requirement and represents the equivalent orifice of the valve. A substantial amount of data is available which relates various valve types to the equivalent orifice coefficient (C_d). Therefore, for each application, the flow factor can be calculated and for each valve type, the minimum flow area required can be determined by application of an appropriate equivalent orifice coefficient. Combining flow rate, pressure drop, inlet pressure and flowing media characteristics, the valve application flow factor can be expressed as:

$$C_d A_t = \frac{\dot{W}\sqrt{RT}}{27.82\, P_1 \sqrt{\frac{K}{K-1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{K}} - \left(\frac{P_2}{P_1}\right)^{\frac{K+1}{K}}\right]}}$$

where:

C_dA_t = flow factor -- in^2

$\dot{W}$ = flow rate -- lb/sec

R = gas constant of media - $\frac{in\text{-}lb}{lb\text{-}^oF}$

T = media temperature -- ^oR

P_1 = inlet pressure -- psia

K = specific heat ratio of media

P_2 = outlet pressure = $P_1 - \Delta P$ -- psia

Δp = allowed pressure drop -- psi

Marquardt previously conducted an extensive parametric study of gaseous propellant injector valves, under NASA-Houston Contract NAS9-10886. That effort utilized the valve flow factor (C_dA_t) as a key input to a program for predicting valve performance and physical characteristics. The calculation of the flow factor for this application permitted the prediction of a number of useful parameters by the methods developed during the parametric study.

For specific cases of this program, the calculated flow factors are:

Propellant		Oxygen		
Inlet Press. psia	20	($13.8 N/cm^2$)	400	($276 N/cm^2$)
ΔP psi	1	($0.69 N/cm^2$)	5	($3.5 N/cm^2$)
$\dot{W}$ pps	2.86	(1.3Kg/sec)	2.75	(1.25Kg/sec)
C_dA_t in^2	13.30	($85.81 cm^2$)	1.256	($8.103 cm^2$)

Propellant		Hydrogen		
Inlet Press. psia	20	($13.8 N/cm^2$)	400	($276 N/cm^2$)
ΔP psi	1	($0.69 N/cm^2$)	5	($3.5 N/cm^2$)
$\dot{W}$ pps	1.14	(0.52Kg/sec)	.69	(0.31Kg/sec)
C_dA_t in^2	21.20	($136.77 cm^2$)	1.256	($8.103 cm^2$)

Based on these calculations, it was evident that the valve orifice size range of interest was from 1-1/4 to 6-1/2 (3.175 to 16.51 cm) in diameter. Because of the greater difficulty of sealing at 400 psi ($276 N/cm^2$) as compared to 20 psi ($13.8 N/cm^2$) inlet pressure, a sealing closure evaluation diameter of 2 inches (5 cm) was chosen for the rapid screening tests.

The prediction of the closing motion impact magnitude requires knowledge of moving mass and dynamic characteristics of the closing motion. Employing the parametric analysis techniques developed under NASA Contract NAS9-10886, these characteristics can be estimated as a function of the calculated flow factors (C_dA_t) for each case.

Moving mass for the inlet pressure range from 0 to 400 psia ($276 N/cm^2$) has been found to correspond to the relationship

$$M_v = \frac{\left(\frac{C_dA_t}{C_d}\right)^{1.66}}{185}$$

M_v = moving mass - lbs

C_dA_t = flow factor -- in^2

C_d = orifice coefficient

This relationship is valid for poppet type closures wherein linear occurs. The moving mass (M_v) is defined as the mass of the closure moving element, excluding the actuator. Since the actuation mechanism may be any of a number of candidate devices, it is assumed that the motion of all moving elements takes place at constant acceleration. From the parametric analysis program, the valve stroke is related to the flow factor (C_cA_t) by

$$S = \frac{1}{4}\sqrt{\frac{C_d A_t}{.785\ C_d}}$$

where S = stroke -- in.

and if the allowable time for total motion (full open to full closed) is considered to be 15 milliseconds, the constant acceleration of the moving element is

$$a = \frac{2\ S}{t^2}$$

where a = acceleration -- $\frac{in}{sec^2}$

t = allowable motion time -- seconds

and the terminal velocity of the moving element becomes

$$v_t = \frac{2\ S}{t}$$

where v_t = terminal velocity -- $\frac{in}{sec}$

Assuming that the actuation device is a pneumatic cylinder, the parametric analysis program yielded data which estimates actuator moving mass (M_A). A summary of the resultant parameters which provided the basis for impact considerations is as follows:

Case	Tester Equiv. Size	400 psia ($276\ N/cm^2$) H_2 & O_2 Valves	20 psia ($13.8\ N/cm^2$) H_2 Valve	20 psia ($13.8\ N/cm^2$) O_2 Valve
Flow Factor (in^2)	2.04 ($13.16\ cm^2$)	1.25 ($8.06\ cm^2$)	21.2 $136.8\ cm^2$)	13.3 ($85.8\ cm^2$)
Stroke (in)	.5 (1.3 cm)	.393 (.998 cm)	1.62 (4.12 cm)	1.28 (3.25 cm)
Shutoff Device (lb) Moving Mass	.2 (6.09 Kg)	.18 (0.08 Kg)	1.70 (0.77 Kg)	.85 (0.39 Kg)
Actuator Moving (lb) Mass	.02 (0.009 Kg)	.012 (0.005 Kg)	.011 (0.005Kg)	.010 (0.004 Kg)
Acceleration (in/sec^2) (cm/sec^2)	4440 (11278)	3490 (8865)	8150 (20701)	6450 (16383)
Terminal (in/sec) Velocity (cm/sec)	66.6 (169.2)	52.3 (132.8)	163 (414)	129 (328)

Impact characteristics, based upon the total moving mass and dynamic characteristics can be determined from the total kinetic energy at impact. These values are tabulated below based upon the following relationships.

$$K.E. = \frac{1}{2} (M_v + M_A) V_T^2$$

$$M_t = (M_v + M_A) V_T$$

where K. E. = Kinetic Energy -- $\frac{in^2 \, lbf}{sec^2}$

M_v = moving mass (poppet) -- lb

M_A = moving mass (actuator) -- lb

V_T = terminal velocity

Case		Tester Equiv. Size	400 psia ($276 N/cm^2$) H_2 & O_2 Valves	20 psia ($13.8 N/cm^2$) H_2 Valve	20 psia ($13.8 N/cm^2$) O_2 Valve
$M_v + M_A$	lb	.22 (.485 Kg)	.192 (.423 Kg)	1.71 (3.77 Kg)	.86 (1.90 Kg)
V_T	in/sec	66.6 (169.2 cm/sec)	52.3 (132.8 cm/sec)	163 (414 cm/sec)	129 (328 cm/sec)
K. E.	$\frac{in^2 lbf}{sec^2}$	487 (.142 NM)	263 (.077 NM)	22,750 (6.66 NM)	7170 (2.10 NM)
M_t	$\frac{in \, lbf}{sec}$	14.6 ($1.65 \frac{NM}{sec}$)	10.05 ($1.14 \frac{NM}{sec}$)	279 ($31.5 \frac{NM}{sec}$)	111 ($12.54 \frac{NM}{sec}$)

Based on these kinetic energy determinations, it was decided to use 500 (.146) and 2500 lb (.782 NM) in^2/sec^2 as the test objective values for the high pressure and low pressure sealing closure tests, respectively. Use of the 2500 figure to simulate the low pressure valve kinetic energy is based on the square of the diameters ratio which applies approximately as a scaling factor.

Analytical Leakage Model

To assure the successful conception and design of seal enclosures for the Space Shuttle Auxiliary Propellant Valves, it was essential to develop an understanding of the factors affecting seal enclosure leakage characteristics. Furthermore, since the rapid screening testing of the seal enclosures were performed with the same size seal enclosure (2 inches nominal diameter), it was essential to develop the scaling capability from this nominal size to the actual valve sizes required as discussed in the section entitled, "Sealing Closure Sizing" of this report. For these reasons, substantial effort was expended in developing a preliminary valve leakage Math Model.

In general, the mechanism of leakage extends from permeation flow in solid walls to laminar flow in the interface between walls to gross flow in an interstice or gap. A preliminary objective in determining appropriate analytical techniques concerning leakage is to define the leakage path as a function of the forces involved. If this can be suitably achieved, certain theoretical equations can be employed to calculate the leakage values. The choice of the correct equation or combination of equations is dependent upon the particular flow regime. A tabular summary of the flow regimes and their identification is shown in the following table.

DEFINITION OF FLOW REGIMES

$N_{Re} > 2200$	**Turbulent Flow**
$1800 < N_{Re} < 2200$	**Mixed Flow**
$N_{Re} < 2000, \ \frac{\lambda}{h} < 1.00$	**Laminar Flow**
$0.01 < \frac{\lambda}{h} < 1.00$	**Transition Flow**
$\frac{\lambda}{h} > 1.00$	**Molecular Flow**
Not Defined	**Diffusion Flow**

Where:

N_{RE}	=	Reynolds number
λ	=	Molecular mean free path
h	=	Characteristic channel dimension

Examination of the auxiliary propellant valve leakage requirements ($Q \leq 100$ scc/hr of helium) identifies the flow region of interest as that of the transition flow regime. References 1 and 2 suggest the use of an empirical relationship combining the laminar flow and molecular flow equations in the form $W = W_{laminar} + \epsilon W_{molecular}$ to define the flow in the transition region. As discussed in Reference 1, the molecular flow factor ϵ is close to unity and was so assumed for this study.

The laminar and molecular flow equations using valve geometry notations are as follows:

$$W_L = K_1 \frac{H^3 D_s (P_1^2 - P_2^2)}{\mu L R T}$$

$$W_M = K_2 \frac{H^2 D_s (P_1 - P_2)}{L \sqrt{RT}}$$

In terms of volumetric flow at standard conditions, the above equations take the following form:

$$Q_L = \frac{4.64 \times 10^3 \ D_s (P_1^2 - P_2^2)}{\mu L T} \quad H^3$$

$$Q_M = \frac{4.14 \times 10^8 \ D_s (P_1 - P_2)}{L} \quad H^2$$

Q	=	Leakage rate of helium	cc/hr
D_s	=	Mean valve seat diameter	in.
P_1	=	Supply pressure	psia
P_2	=	Downstream pressure	psia
μ	=	Viscosity	$\frac{\text{lb-min}}{\text{in}^2}$
L	=	Valve land width	in.
T	=	Gas Temperature	^oR
H	=	Effective leak path height	in.

As repeatedly pointed out in the reference literature reviewed, the effective leak path dimension, H, is a most difficult parameter to define, particularly by analytical methods. References 1 and 2 both theorized solutions proposing H to be a function of the peak to valley roughness heights of the mated materials modified by a deformation factor which included the seating stress. From a direct analytical consideration, Reference 1 assumed various geometrically defined surfaces, including a sinusoidal topography and applied the Hertz stress equations to define surface finish deformation under load. This consideration took into account the elastic properties of the material and its yield strength. Reference 3 showed empirical results which related the cube of the leakage path height to a load factor composed of seating stress and material hardness. Considering the above results, path relationship composed of the following factors was used for this study.

$$H = M(h_1 + h_2)\left(1 - \frac{c}{\sigma y}\right)$$

where:

where:	**M**	**=**	**.68 for assumed sinusoidal surface, laminar flow**
	M	**=**	**.61 for assumed sinusoidal surface, molecular flow**
	h_1	**=**	**peak to valley roughness height for surface number 1, in.**
	h_2	**=**	**peak to valley roughness height for surface number 2, in.**
	σs	**=**	**seating stress, psi**
	σy	**=**	**yield strength of softer material, psi**
	N	**=**	**stress exponent.**

It was believed that the exponent N could be determined empirically from the upcoming test results. For preliminary design purposes, however, the work of Reference 1 has been used by Marquardt to approximate a value for N of 1.373. This value was determined using metal to metal surface properties characteristic of the Reference 1 work.

It was decided that this leak path height relationship could be extended to include plastic surfaces since analytical techniques regarding plastic are presently not available. For consistency, and until test data became available, the stress exponent value of 1.375 was assumed to be a "ballpark" indicator for both metals and plastics. Combining the stress factor with the leakage equation and recalculating the constants resulted in the following leakage model which was used for the program preliminary design calculations.

$$Q_T = Q_L + Q_M$$

$$Q_T = \frac{1.46 \times 10^3\, D_s (P_1^2 - P_2^2)}{\mu\, L\, T} \left[(h_1 + h_2)\left(1 - \frac{\sigma_s^{1.375}}{\sigma y}\right) \right]^3 +$$

$$\frac{1.55 \times 10^8\, D_s (P_1 - P_2)}{L} \left[(h_1 + h_2)\left(1 - \frac{\sigma_s^{1.375}}{y}\right) \right]^2$$

As previously stated, H_1 and H_2 are the peak to valley (PTV) surface roughness heights. These values, as discussed in Reference 1, are approximately equal to three times the surface finish for a sinusoidal topography.

The above equation was solved the the quantity $\left[(h_1 + h_2)\left(1 - \frac{\sigma_s^{1.375}}{\sigma_y}\right) \right]$ using a leakage rate of 100 cc/hr and, in turn, the seating stress, σ_s, was determined for various seating materials with given surface finishes. The materials evaluated were tungsten carbide, copper, teflon, and polyimide. The results of these calculations are shown in Table II.

TABLE II - SPACE SHUTTLE AUXILIARY PROPELLANT VALVE SEATING STRESS ESTIMATES

HIGH PRESSURE CONFIGURATION

Material Poppet/Seat	Surface Finish μIn. (cm)	PTV Height μIn. (cm)	Seat Diameter In. (cm)	Land Width In. (cm)	σ_y (Softer Mat.) psi (N/cm^2)	Seating Stress σ_s-psi (N/cm^2)
WC/WC	0.5(1.3) 0.5(1.3)	1.5(3.8) 1.5(3.8)	2.015 (5.118)	.015(.038)	690,000(475,738)	2910(2006)
WC/Soft Copper	0.5(1.3) 2.0(5.1)	1.5(3.8) 6.0(15.2)	2.015(5.118)	.015(.038)	10,000(6,895)	580(400)
WC/Hard Copper	0.5(1.3) 2.0(5.1)	1.5(3.8) 6.0(15.2)	2.015(5.118)	.015(.038)	45,000(31,026)	1740(1200)
WC/Teflon	0.5(1.3) 4.0(10.2)	1.5(3.8) 12.0(30.5)	2.030(5.156)	.030(.076)	1,250(862)	140(97)
WC/Teflon	0.5(1.3) 4.0(10.2)	1.5(3.8) 12.0(30.5)	2.060(5.232)	.060(.152)	1,250(862)	130(90)
WC/Polyimide	0.5(1.3) 2.0(5.1)	1.5(3.8) 6.0(15.2)	2.030(5.156)	.030(.076)	18,000(12,411)	760(524)
WC/Polyimide	0.5(1.3) 2.0(5.1)	1.5(3.8) 6.0(15.2)	2.060(5.232)	.060(.152)	18,000(12,411)	580(400)

NOTE: The calculations for teflon and polyimide were done for two land widths.

It should be recognized that the Table II results are only approximations of the true values particularly for the softer materials. They served, however, as a guide for the design and testing of the various valve closure configurations.

As can be seen from the equation, mathematically the leakage rate is reduced to zero when $\sigma_s^{1.375}$ is equal to σ_y. This is not proposed to be a true statement because of the limitations of the theory employed and the assumptions used. It did serve, however, to indicate limits for the use of the equation until test results became available.

Because of the complexities and the difficulties of measuring the minute dimensions associated with even large leakage changes, very little factually co-related experimental data on the mechanism of valve failures due to wear have been documented. When wear is due mainly to the shearing of junctions, which is referred to by Rubinowicz (Reference 4) as adhesive wear, an empirical relationship for the value of wear per unit distance of travel is:

$$Z = \frac{KW}{3P}$$

Where K represents the percent of friction junctions producing wear, W is the applied load, and P is the hardness of the wearing surface. Tabulated values for K, which is referred to by Rubinowicz as the wear coefficient, show that the wear rate between two surfaces is reduced by (1) the use of hard materials, and (2) the use of materials with low interaction, i.e., unlike materials of low solubility.

The above relationship does not, however, provide any information about the change in surface texture with wear. Rubinowicz (Reference 4) has shown that, in some cases, wear particle sizes can be predicted on the basis of elastic surface energy and hardness properties. Reference 5 used the empirical wear rate relationship, $Z = \frac{KW}{3P}$, in combination with the Rubinowicz theories to arrive at an equation for use in estimating surface finish change due to wear as a function of valve cycles, impact stress and poppet/seat scrubbing distance.

This equation has the form:

$$y_N = y^o \cdot \frac{k_{AD}}{11,900} N s \sigma_I \left[\frac{1.4}{P_A} - 17.6 \times 10^4 \frac{y^o}{G_{AB}} \right]$$

where: y_N = Surface finish after N cycles, in.

y^o = Original surface finish, in.

k_{AD} = Wear coefficient

N = Number of cycles

s = Lateral Sliding component [illegible] en poppet and seat (scrubbing distance), in.

σ_I = Poppet/seat impact stress,

P_A = Material hardness (softer surface), kg/mm^2

G_{AB} = Energy of adhesion, $ergs/cm^2$

The equation, as presented in Reference 5, was modified somewhat to more closely conform to the sinusoidal surface model as presented in Reference 1. This relates surface finish, y^o, to peak to valley height, h, by $y^o = 1/3$ h.

Values of the wear coefficient for a few materials determined from tests are presented in Reference 4, along with the following typical relationship for unlubricated (clean) surfaces.

MATING SURFACES	k_{AD}
Metal on Metal (like materials)	5×10^{-3}
Metal on Metal (unlike materials)	2×10^{-4}
Non-Metal on Metal	5×10^{-6}

The energy of adhesion between surfaces, G_{AB}, is a function of surface energy, γ, as discussed in Reference 4. The relationships given are:

$G_{AB} = 2\gamma_A$ identical materials ($\gamma_A = \gamma_B$)

$G_{AB} = 3/4\ (\gamma_A + \gamma_B)$ interacting materials

$G_{AB} = 1/2\ (\gamma_A + \gamma_B)$ Totally different materials.

A table of surface energy values is presented in Reference 4 for a wide range of materials.

Wear calculations for the materials previously considered in the leakage and seating stress analysis shown in Table II were performed to define scrubbing distance as a function of surface finish degradation. A surface finish change of 0.5×10^{-6} was arbitrarily assumed for calculation purposes. The impact stress was assumed to be 150% of the seating stress of Table II.

The results are presented in Table III for 10^5 valve cycles. As shown, the allowable scrubbing distance for the wear specified is approximately one to two orders of magnitude greater for the plastic compared to the metals. It is noted that the work presented in Tables II and III was done for the high pressure (450 psia) (310 N/cm^2) propellant valve configuration. The requirements are considerably less severe in terms of surface finish and/or seating stress for the low pressure (20 psia) (14 N/cm^2) configuration.

A comparison of the high and low pressure configurations using tungsten carbide as an example, indicates that the surface finish can be changed from the high pressure value of 0.5×10^{-6} at the same stress level with no change in leakage rate; or with no surface finish change that the seating stress can be reduced to a value just sufficient to keep the poppet in place. This results from an allowable effective leakage path height increase brought about by the lower pressure requirements when introduced into the leakage equation.

Sealing Closure Conception and Design

The valve leakage and wear analysis presented in the preceding section was used as the basis for conceiving the various sealing closures to be evaluated during the Space Shuttle Auxiliary Propellant Valves Program. Briefly reviewing the wear model, it is evident that materials featuring a low wear coefficient and a low energy of adhesion, but a high material hardness, are desirable. Furthermore, to minimize the effects of wear, it is desirable to minimize any sliding or scrubbing during the mating of the sealing interface and to minimize the impact stresses which occur during these sliding motions.

As far as leakage through a closed sealing interface is concerned (static condition), it is evident that the parameter defining the leakage path height is most important, since it occurs to a higher power in the leakage model. The leakage path height is directly related to the original surface finish except that it may be modified (reduced) due to surface loading; with respect to the latter, the sealing closure interface materials of course, play an important role. Finally, the leakage rate is also inversely proportional to the length of the leakage path.

In summary, it was concluded that the sealing closures for this application

TABLE III - SPACE SHUTTLE AUXILIARY PROPELLANT VALVE WEAR RELATIONSHIPS

[illegible]	Wear $y_N - y^o$ μ in. (cm)	Wear Coeff. k_{AD}	Cycles N	Impact Stress I psi(N/cm^2)	Material Hardness P_A Kg/mm^2	Adhesion Energy G_{AB} ergs/cm^2	Original Surf. Finish y^o μ in. (cm)	Allowable Scrubbing Distance s μ in. (cm)
[illegible]	0.5(1.3)	5×10^{-3}	10^5	4360(3006)	1800	1560	0.5(1.3)	3.78(9.60)
WC/Soft Copper	0.5(1.3)	2×10^{-4}	10^5	870(600)	20	1400	1.5(3.8)	4.63(11.76)
WC/Hard Copper	0.5(1.3)	2×10^{-4}	10^5	2610(1800)	80	1400	1.5(3.8)	6.60(17.76)
WC/Teflon	0.5(1.3)	5×10^{-6}	10^5	222(153)	1.88	400	3.5(8.9)	71.7(182.1)
WC/Polyimide	0.5(1.3)	5×10^{-6}	10^5	1140(786)	37	400	1.5(3.8)	281(714)

NOTE: Impact stress assumed equal to 150% of seating stress of Table should feature very fine surface finishes, low static interface loads, low impact loads during closure, minimum scrubbing distances during the mating of the sealing interface, and very hard materials either in combination with each other, or in combination with plastics. A total of ten (10) sealing closures were designed and submitted to the NASA-LeRC Project Manager for approval. The ten (10) sealing closures are identified as follows:

Drawing Number	
L4677	Flat Polyimide Seat
L4678	Spherical Copper Lip Seat, Bellows Loaded
L4679	Flat Metal Seat, Flexure Aligned Poppet
L4680	Flat Teflon Seat, Bellows Loaded
L4681	Spherical Teflon Seat, Bellows Loaded
L4682	Flat Metal Seat, Bellows Loaded
L4683	Flat Teflon Coated Lip Seat
L4684	Spherical Teflon Coated Lip Seat
L4685	Flat Polyimide Seat, Bellows Force Loaded
L4686	Spherical Polyimide Seat, Bellows Force Loaded

The sealing closures were designed so that they are applicable to a wide variety of fluid shutoff devices, such as ball, butterfly, poppet, blade, gate, etc. Table IV is an applicability matrix which shows what type of fluid shutoff devices utilize each of the designed sealing closures. The following paragraphs briefly describe these sealing closures which were designed:

P/N L4677

This sealing closure, shown in Figure 2, features a flat interface wherein a highly polished tungsten carbide poppet mates with a polyimide seat. The polyimide protrudes slightly above the metal of the seat, such that the static loading is sufficient to compress the polyimide to a point where the poppet contacts the polyimide and surrounding metal simultaneously. The metal surrounding the polyimide prevents cold flowing of the plastic and also serves as an impact force absorber.

P/N L4678

This sealing closure, shown in Figure 3, features a spherical interface employing a highly polished tungsten carbide poppet and a hard, well-polished copper seat. The copper seat is mounted on a single convolution bellows to achieve the design sealing closure interface load and to provide alignment between the seat and the poppet. The sealing closure design is such that some over-travel of the poppet is allowed in order to permit absorption of the major portion of the

TABLE IV - SEALING CLOSURE APPLICATION

	Drawing No.									
VALVE TYPE	L4677	L4678	L_679	L4680	L4681	L4682	L4683	L4684	L4685	L4686
Flat Poppet	X		X	X		X	X		X	
Spherical Poppet		X			X			X		X
Conical Poppet		X			X			X		X
Blade				X		X			X	
Gate				X		X			X	
Butterfly		X			X					X
Ball		X			X					X

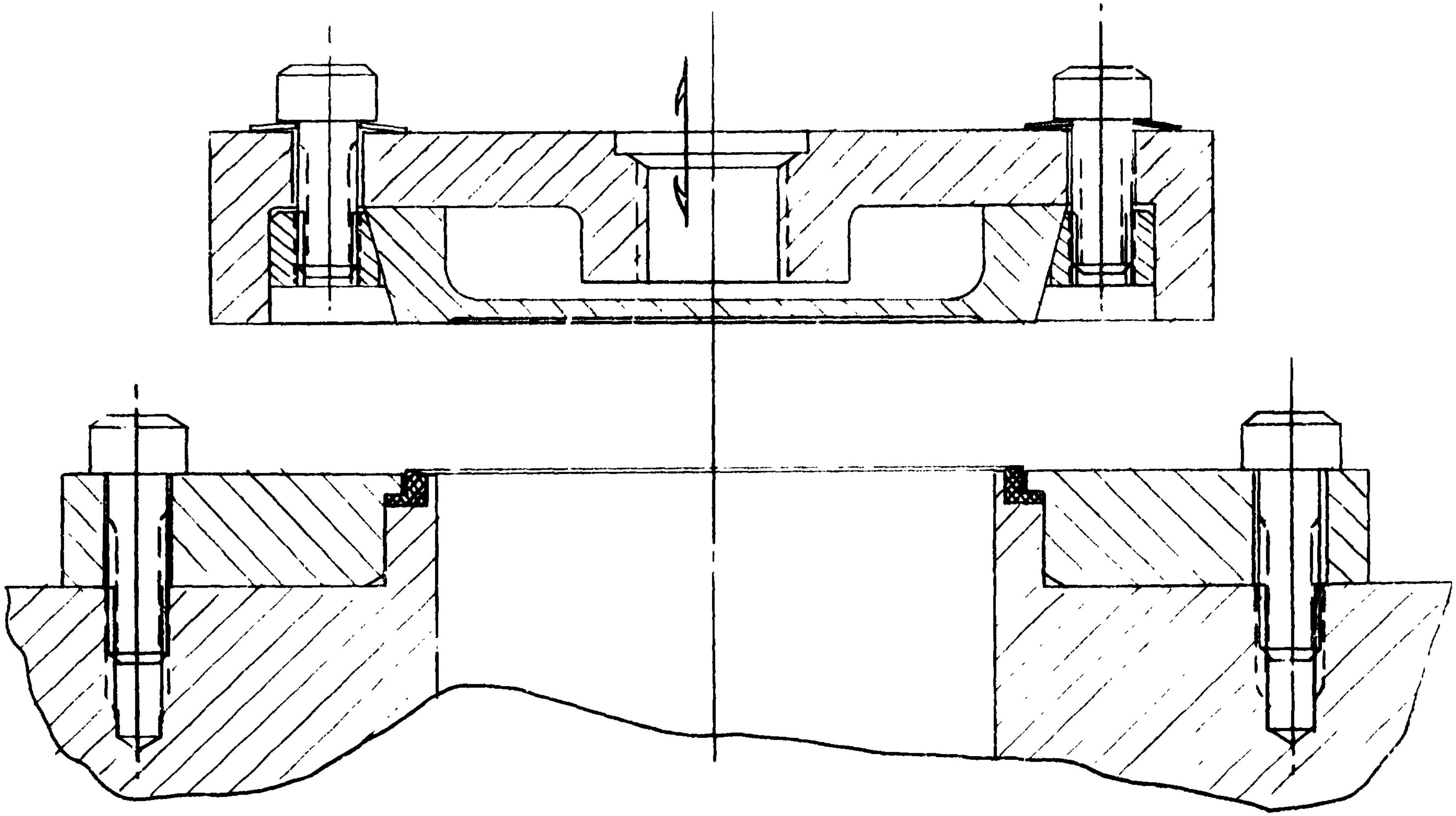


Figure 2 - Flat Polyimide Seat, Drawing No. L4677

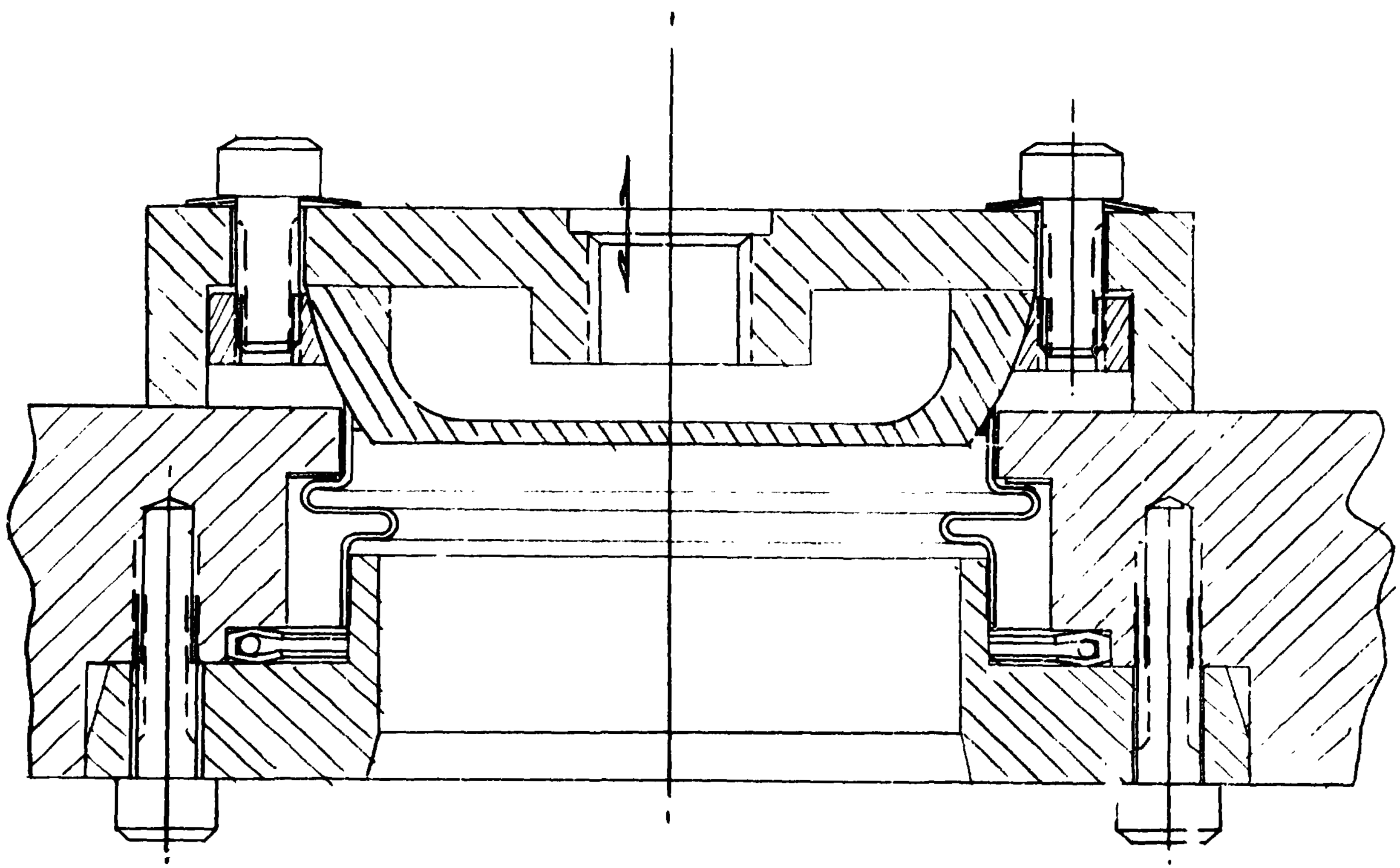


Figure 3 - Spherical Copper Lip Seat, Bellows Loaded, Drawing No. L4678

kinetic energy inherent in the poppet assembly closure movement by the bumper surrounding the poppet. This sealing closure is, of course, not only applicable to a poppet type valve, but also to a ball or butterfly type shutoff device.

P/N L4679

This design, shown in Figure 4, consists of a flat, highly polished tungsten carbide poppet mating with a flat, highly polished tungsten carbide seat. The seat features a narrow annulus to minimize poppet load requirements. Alignment of the poppet to the seat, as well as reduction of impact forces during closure, is achieved by incorporation of a flexure into the poppet assembly. This flexure allows the poppet assembly mass to over-travel past the poppet until the bumper surrounding the poppet strikes the seat outside of the sealing surface.

P/N L4680

The L4680 design, shown in Figure 5, features a flat sealing closure interface which employs a highly polished tungsten carbide poppet and a combination teflon/polyimide seat. The teflon serves as the primary sealing element in the seat, with the polyimide's function reduced to that of a stop and thermal compensator. The seat assembly is again bellows-mounted to achieve the correct sealing interface loads and to provide alignment between the poppet and seat. The major portion of the kinetic energy of the poppet assembly during closure is absorbed by the bumper surrounding the poppet.

P/N L4681

This design, shown in Figure 6, is very similar to the L4680 sealing closure except that a spherical interface is employed in place of the flat interface. The spherical interface design is, of course, equally applicable to a ball or butterfly fluid shutoff device as to a poppet fluid shutoff device.

P/N L4682

The sealing closure shown in Figure 7, features a highly polished flat tungsten carbide poppet mating with a highly polished flat tungsten carbide seat. The seat is bellows-mounted to assure achievement of the design interface load and alignment of the poppet to the seat. In addition, this design shows employment of two Belleville springs in parallel with the seat bellows springs to permit the evaluation of high sealing interface loads. Again, as in the other bellows-mounted seat configurations, most of the kinetic energy of the closing assembly is absorbed by the bumper surrounding the poppet.

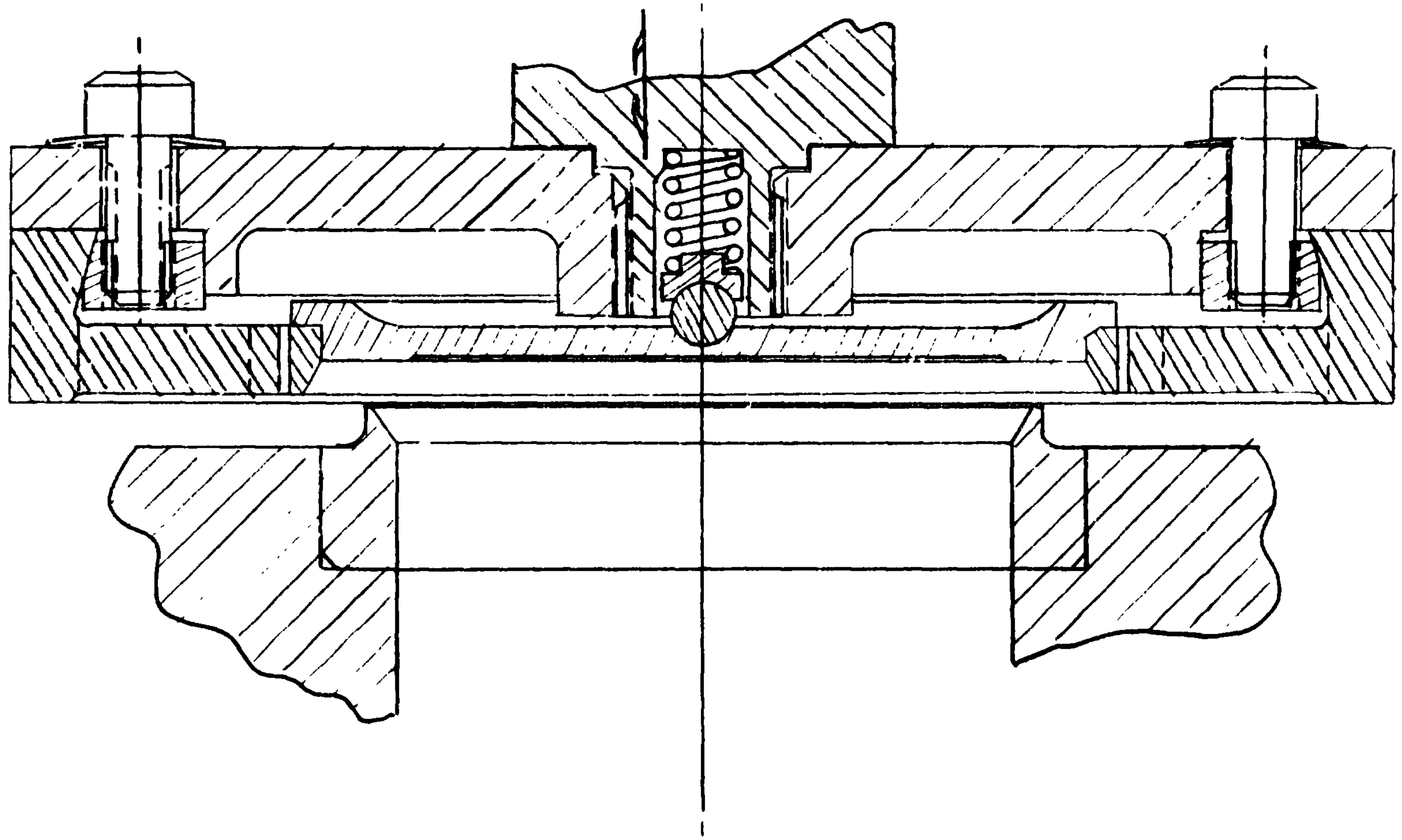


Figure 4 - Flat Metal Seat, Flexure Aligned Poppet,
Drawing No. L4679

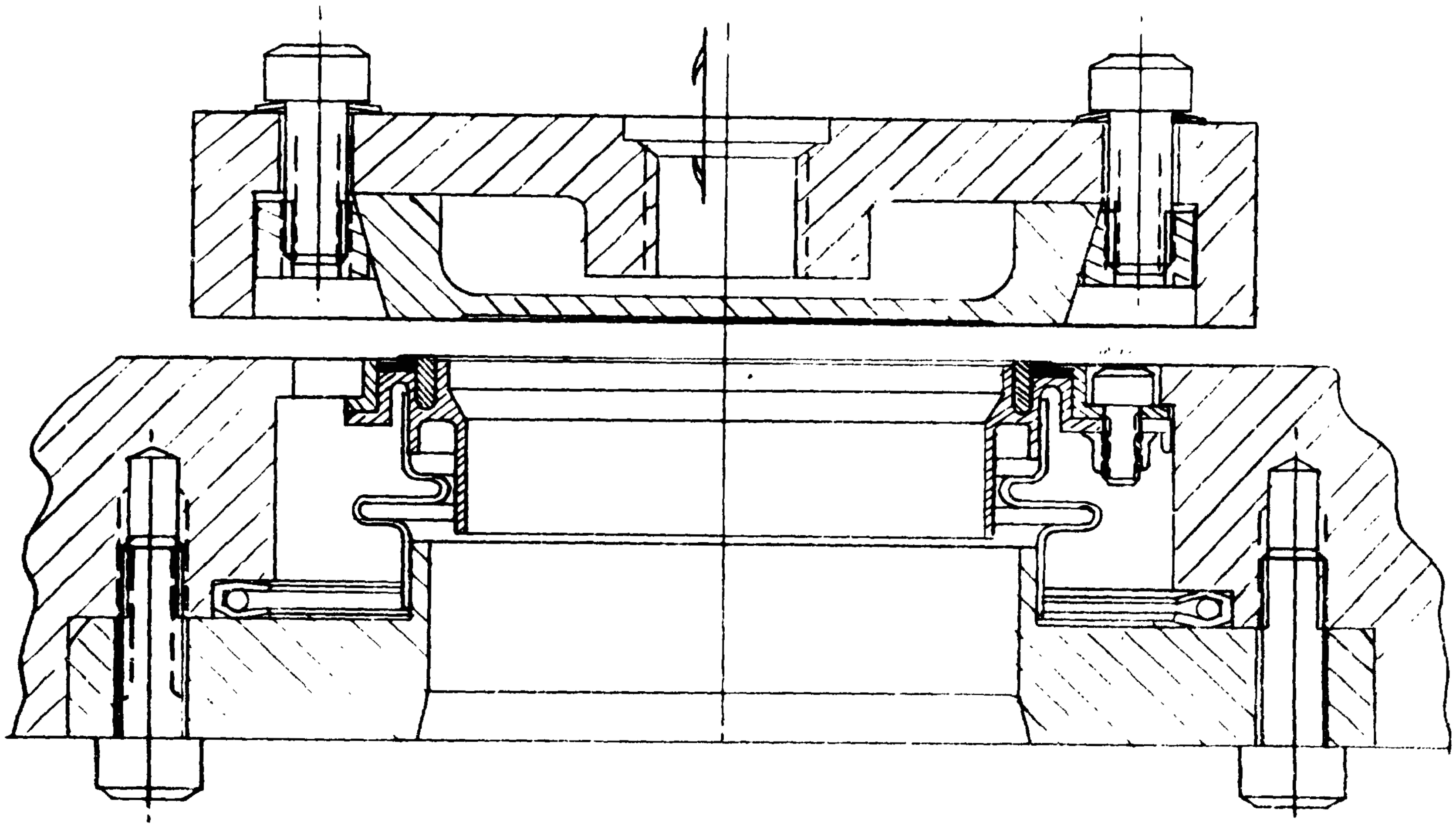


Figure 5 · Flat Teflon Seat, Bellows Loaded, Drawing No. L4680

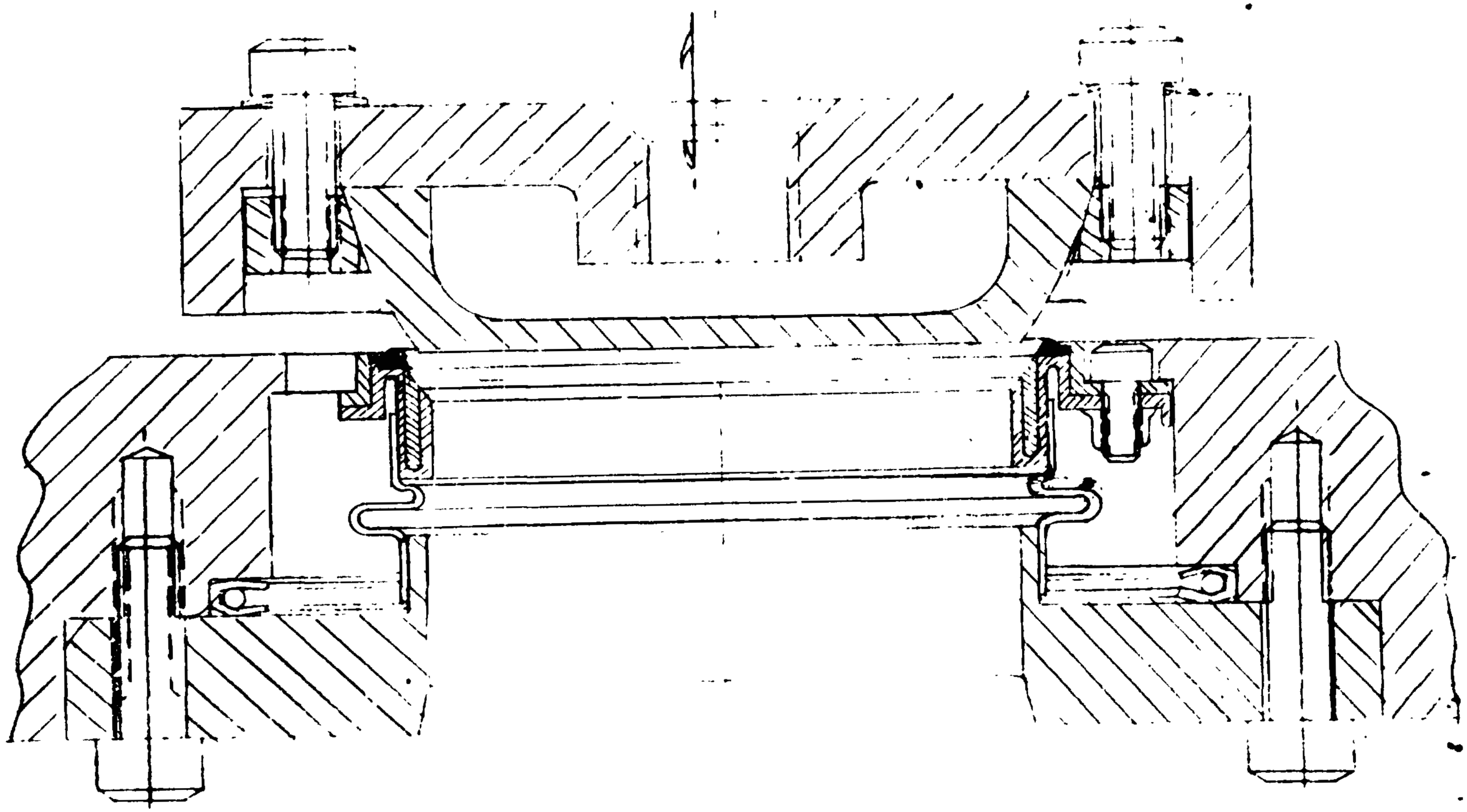


Figure 6 - Spherical Teflon Seat, Bellows Loaded, Drawing No. L4681

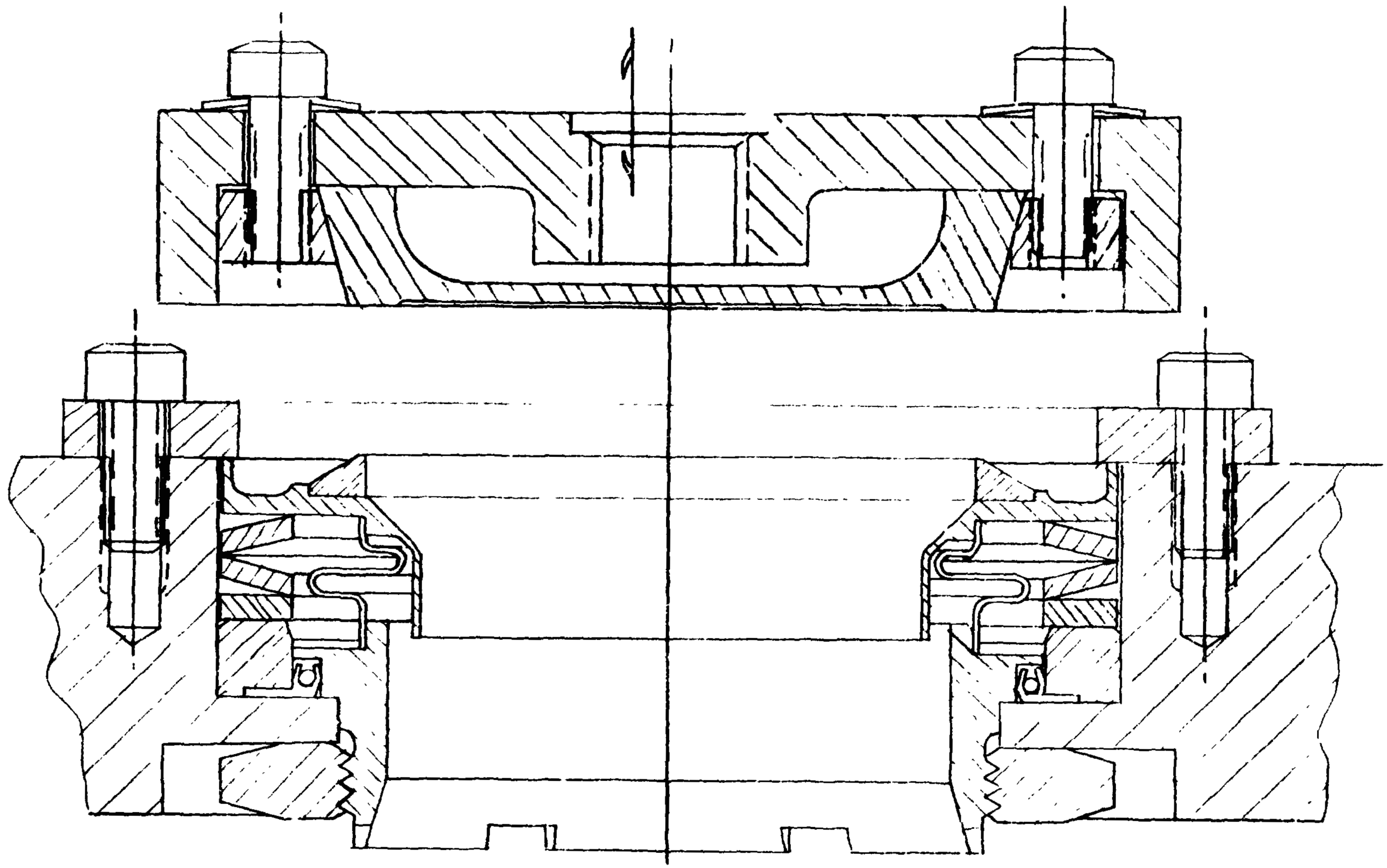


Figure 7 - Flat Metal Seat, Bellows Force Loaded, Drawing No. L4682

P/N L4683

This sealing closure, Figure 8, again features a flat interface this time employing a highly polished tungsten carbide poppet and a teflon coated metal lip seal seat. The spring rate of the metal lip seal is designed such as to permit over-travel of the poppet during closure and the resultant absorption of most of the kinetic energy of the poppet assembly by the bumper surrounding the poppet. The metal lip seal serves primarily as a structural support of the film on the seal which, in turn, serves as the actual sealing interface.

P/N L4684

The L4684 design, Figure 9, is similar to the L4683 design except that it employs a spherical interface in place of a flat interface. This approach is suitable for a ball or butterfly-type valve as well as a poppet valve.

P/N L4685

This sealing closure, Figure 10, utilizes a flat, highly polished tungsten carbide poppet mating with a polyimide seat. The seat is bellows-mounted to achieve the correct design sealing interface load and alignment between the poppet and the seat. This seat also features the Belleville springs in parallel with the bellows to permit the evaluation of high sealing interface loads. During closure, most of the impact force resulting from the kinetic energy of the poppet assembly is absorbed by the bumper surrounding the poppet.

P/N L4686

This sealing closure design, Figure 11, is similar to the L4685 design, except for the use of a spherical interface in place of the flat interface. This approach permits the application of this sealing closure to a butterfly or ball type valve in addition to its use as a poppet valve.

Upon completion of these designs, a stress analysis of all of the sealing closures was completed and the results of this analysis are discussed in the following section.

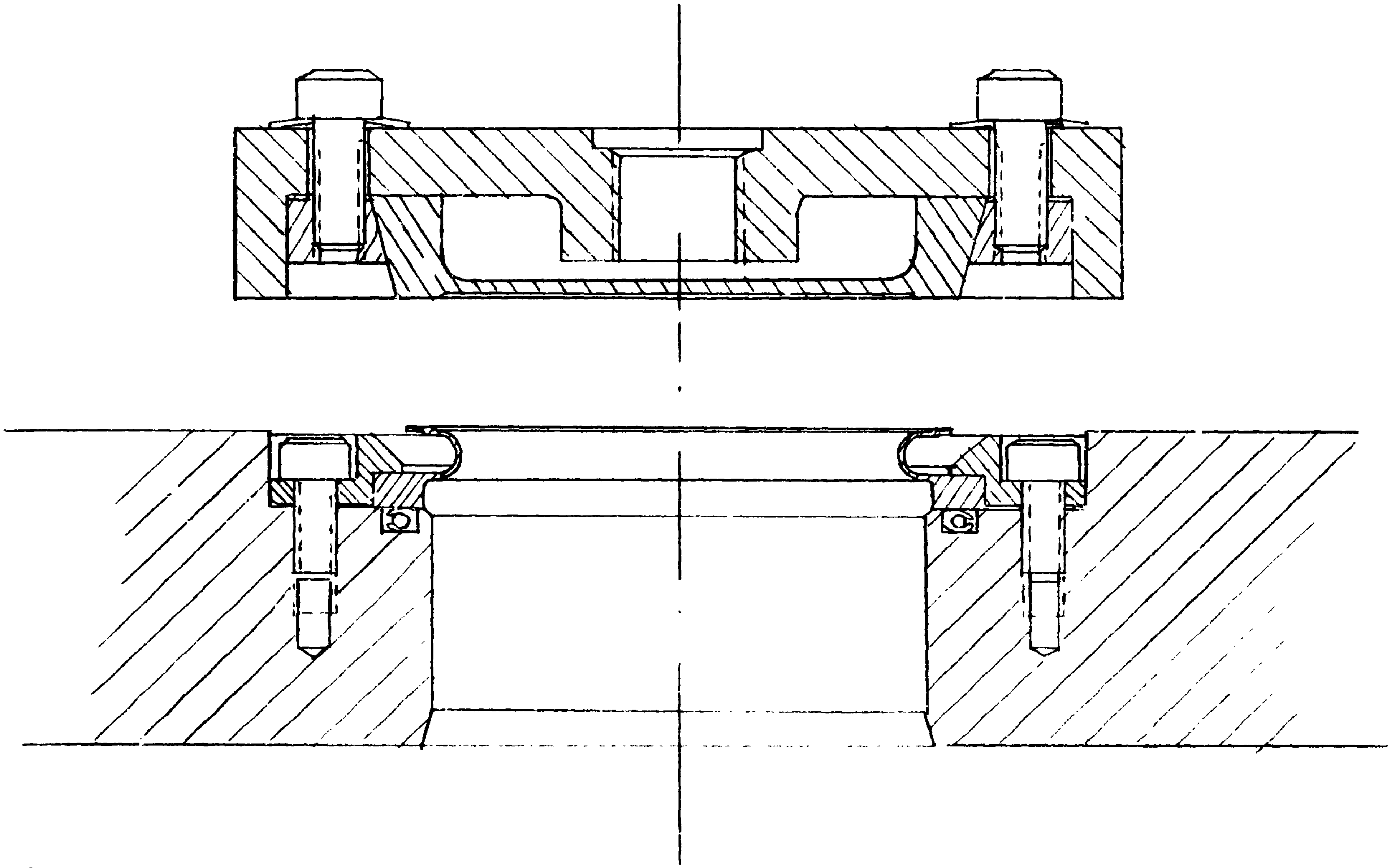


Figure 8 - Flat Teflon Coated Lip Seat,
Drawing No. L4683

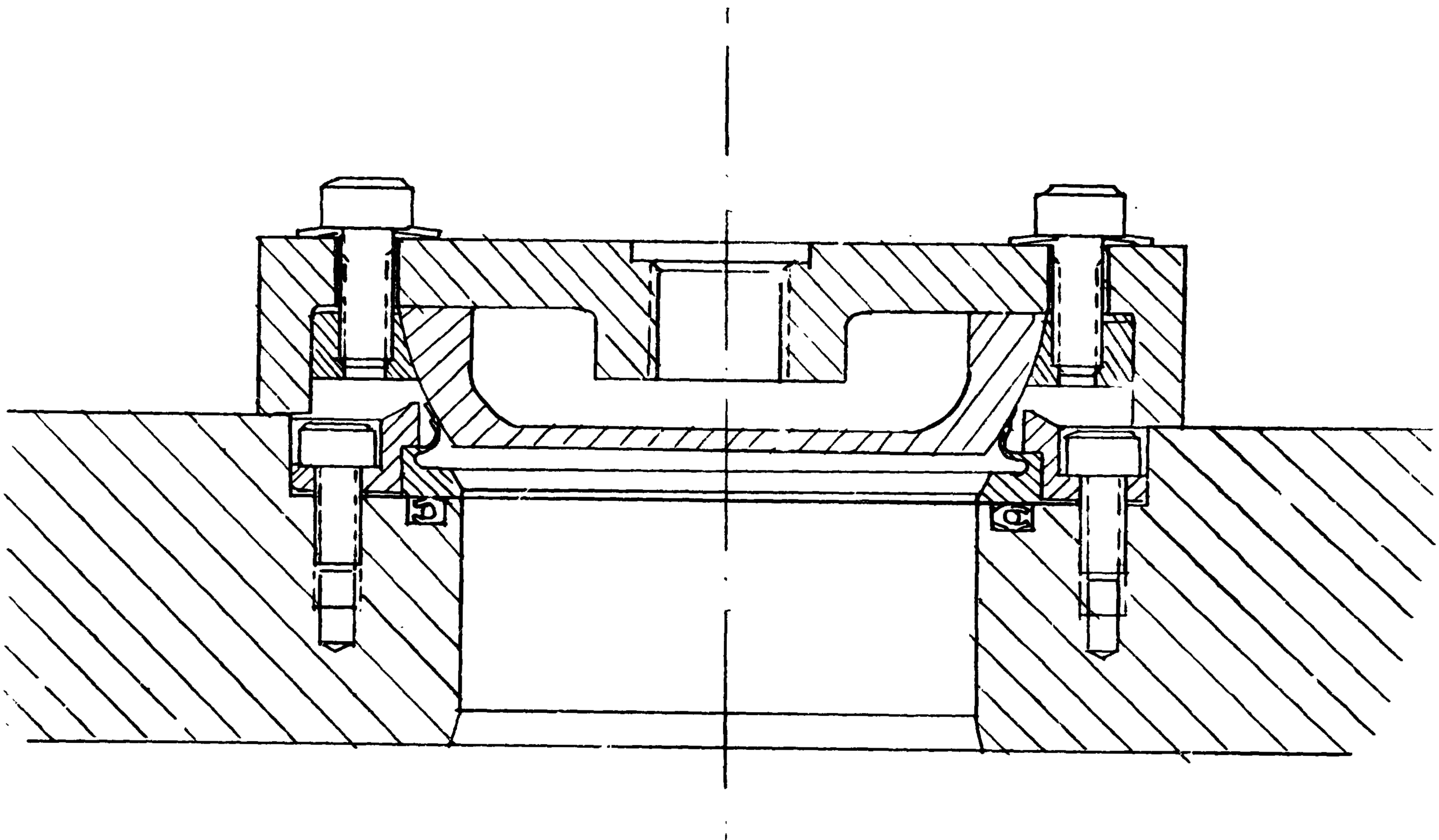


Figure 9 - Spherical Teflon Coated Lip Seat,
Drawing No. L4684

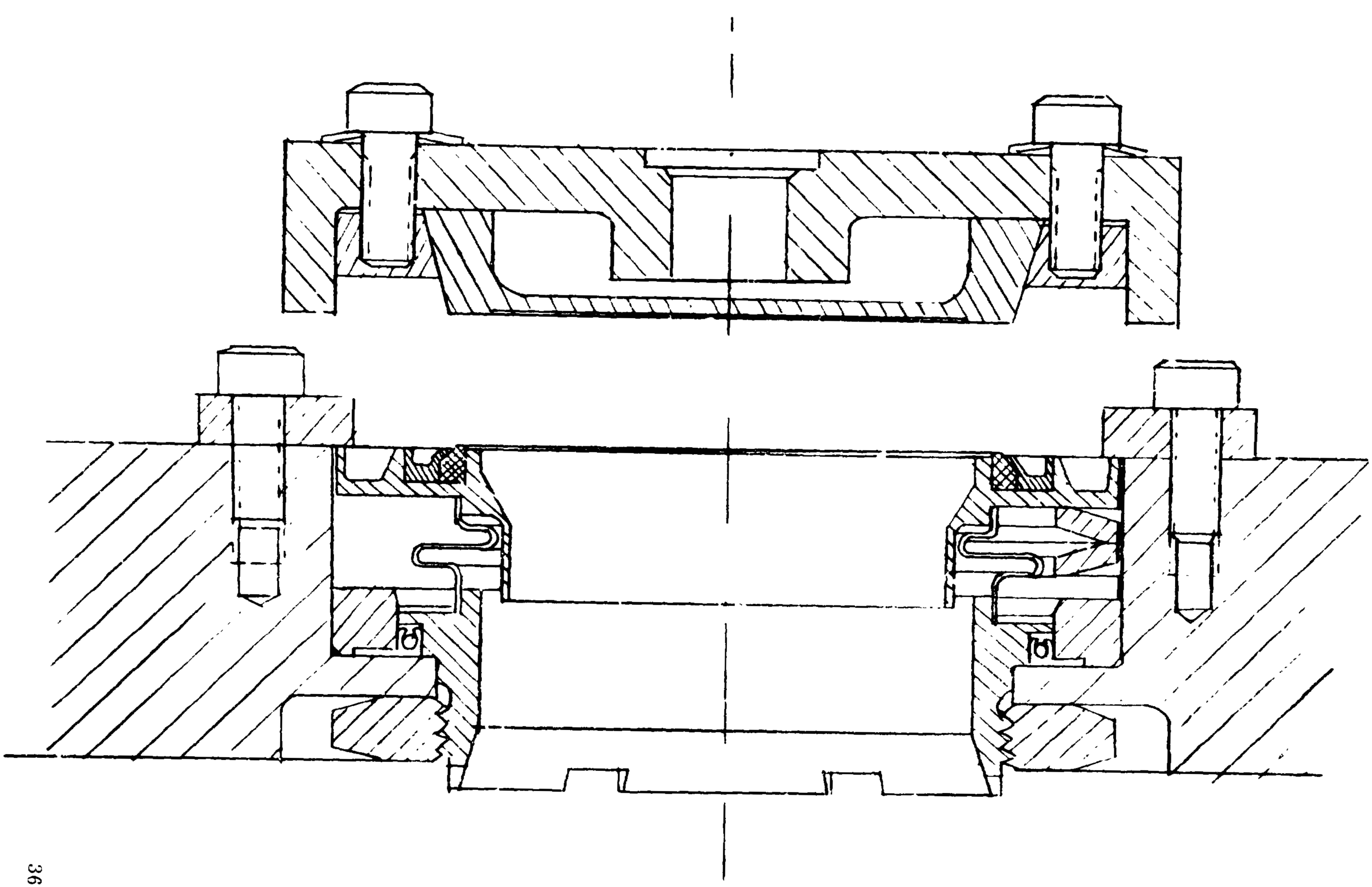


Figure 10 - Flat Polyimide Seat, Bellows Force Loaded, Drawing No. L4685

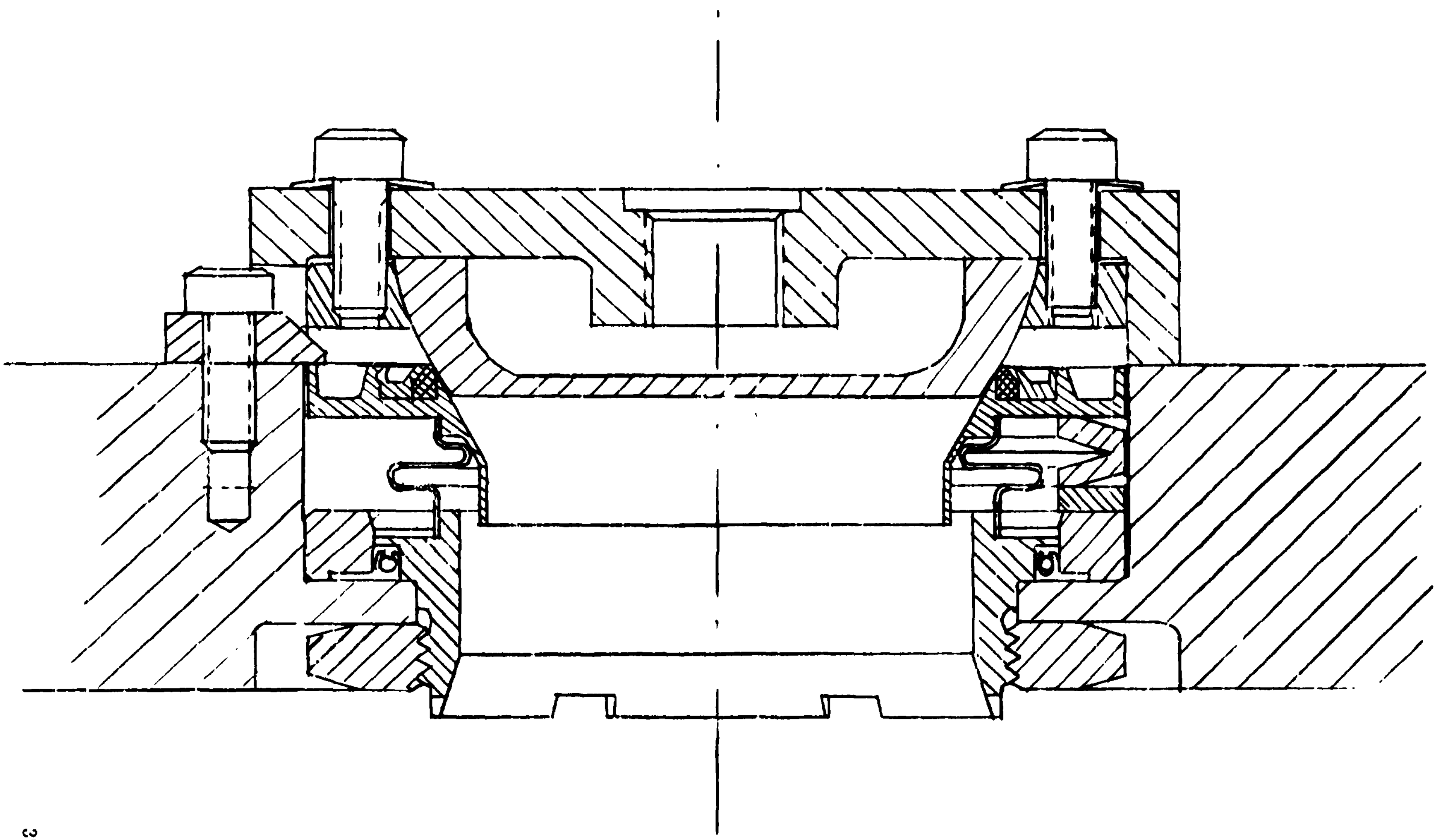


Figure 11 - Spherical Polyimide Seat, Bellows Force Loaded, Drawing No. L4686

Sealing Closure Stress Analysis

A structural analysis of the sealing closures was based on the following criteria:

1) The valve poppet was analyzed for the anticipated maximum kinetic energy developed during the rapid screening tester closing motion. This is 2500 lb, in.^{2}/sec^2 or 6.5 slugs-inches (.7315 N-M). An applied seat load of 1500 lbs (6672 N) also was considered. The poppet and seat must be capable of withstanding the impact resulting from the above energy input for at least one (1) million cycles.

2) The spring-supported seats were conservatively assumed to obtain the maximum poppet velocity at the moment of impact. This velocity was then used to calculate the kinetic energy of this seat-spring system. The kinetic energy of the seat was used to calculate the reaction force from the seat support spring. The seat and support springs must be capable of withstanding one (1) million impacts

3) The seat and poppet were also required to have the ability to withstand 450 psi (310 N/cm^2) operating pressure applied for one (1) million cycles.

4) The seats were required to be able to withstand axial deflections of .005 to .008 inch (.013 to .020 cm) at the sealing surface for one (1) million cycles without failure.

The details for the stress calculations of the sealing closures are presented in Appendix A. A summary of the stress levels and margins of safety designed into some of the more important closure components is presented in Table V.

Detail Design of Sealing Closures

Upon receipt of approval from the NASA-LeRC Project Manager for the sealing closure concepts, detail design drawings were prepared. During these detail design efforts, several minor modifications were incorporated into certain sealing closures for various reasons, such as improved alignment, improved assembly control, and ease of manufacture. The sealing concepts affected by the changes will be discussed briefly.

Spherical Poppet Alignment

To improve the concentricity alignment between the spherical poppets and

TABLE V - MINIMUM STRESS MARGINS - SEALING CLOSURES

Part Name	Condition	Remarks	Stress (ksi) of Loads (k) (K N/cm^2)	Margins of Safety
Poppet Support	KE = 2500 lbs in^2/Pw^2 (.7315 N-M)	.25 t Plate (.625 cm)	71.2 (49.1)	.96
Belleville Spring	Seat Load + 2500 lbs in^2/Pw^2 (.7315 N-M)	Bending on Spring	166 (114.4)	.20
Tung. Car. Poppet	600 psi (414 N/cm^2) Proof	Internal Press. Min. "t"	.051 in.	.88
Lip Seal	[illegible]00 psi (414 N/cm^2)	Bending on T p	200 (137.9)	.00
Spherical Lip Seal	Radial = .00433 in. (.0110 cm)	Bending on seal	1[illegible] (9[illegible]	.015
Flat Plate L = .100 (.254 cm)	y = .0065 in (.0165 cm)	Bending on Plate t = .013 in (.033 cm)	71.3 (49.2)	.96
Flat Plate L = .180 in (.457 cm)	y - .006 in (.0152 cm)	Bending on Plate t - .0125 in (.0318 cm)	140 (96.5)	.00
Flat Plate L = .55 in (1.397 cm)	y = .008 in (.0203 cm)	Bending on Plate t = .053 in (.1346 cm)	141 (97.2)	.00

seats an alignment post was added to the backside of the poppet as shown in Figure 12. This post aligns the center axis of the spherical poppet with a locating bore in the tester poppet holder within .0005 inch (.0013 cm). The locating hole in the poppet hole was machined within .0002 inch (.0005 cm) concentric to a locating bore at the bottom of the tester which positions the seat. Thus, an overall concentricity of .001 inch (.0025 cm) or better was achieved. This type of tolerance can be realistically achieved by a modern precision machine shop within reasonable costs.

The post shown was made of 300 series stainless steel and joined to the tungsten carbide by means of brazing and subsequently finish machined. The final operation on this part was the lapping of the sealing surface.

This modification applies to the poppets for sealing closures L4678, L4681, L4684 and L4686: Figures 2, 6, 9, and 11 respectively.

Belleville Spring Additions

...nal load capabilities of the seat designs featuring only a bellows but no Belleville spring to aid the bellows were determined to be marginal. This applies to sealing closures L4678, L4680 and L4681. Consequently, the areas immediately adjacent to the seals and bellows were modified on these sealing closures to include a Belleville spring. Figure 13 shows the typical modification of the L4680 seal, both before and after.

Addition to the Belleville springs raised the static sealing load capability of these sealing closures to 600 lbs (2669 N), which permitted an experimental tradeoff of sealing closure loads to that level during the rapid screening tests.

Polyimide Seal Retention

Further review of the "rolled over metal lip" technique previously shown for retaining the polyimide seal in sealing closure concepts L4685 and L4686 had disclosed that this technique was not easily controllable and that it could have resulted in subjecting the polyimide to vastly different stress levels from one part to another. Consequently, this design was modified to feature a 300 series stainless steel retaining ring held in place by a total of eight (8) screws. The before and after cross sections of the L4685 sealing closure concept shown in Figure 14. This modification has the additional advantage of readily permitting removal and replacement of the polyimide seal in the event of damage or other mishaps.

At a NASA-LeRC program review meeting held during April of 1971, re-direction was given to discontinue the evaluation of the P/N L4678 spherical

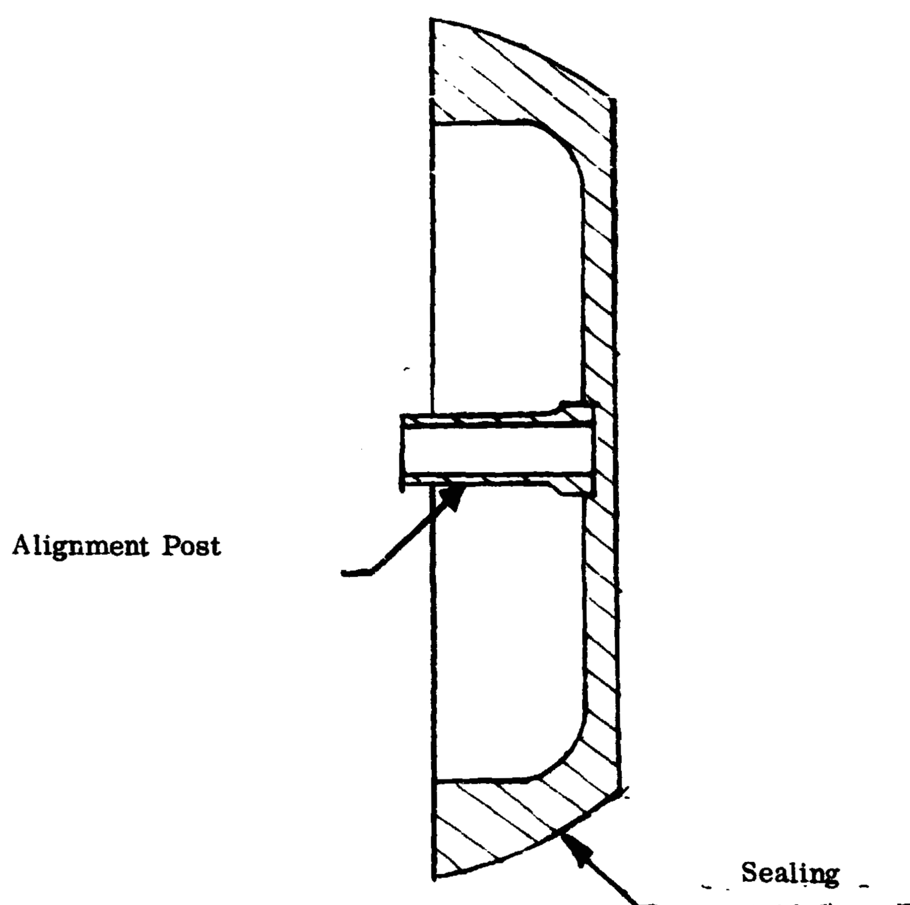


Figure 12 - Spherical Poppet Concentricity Alignment Post

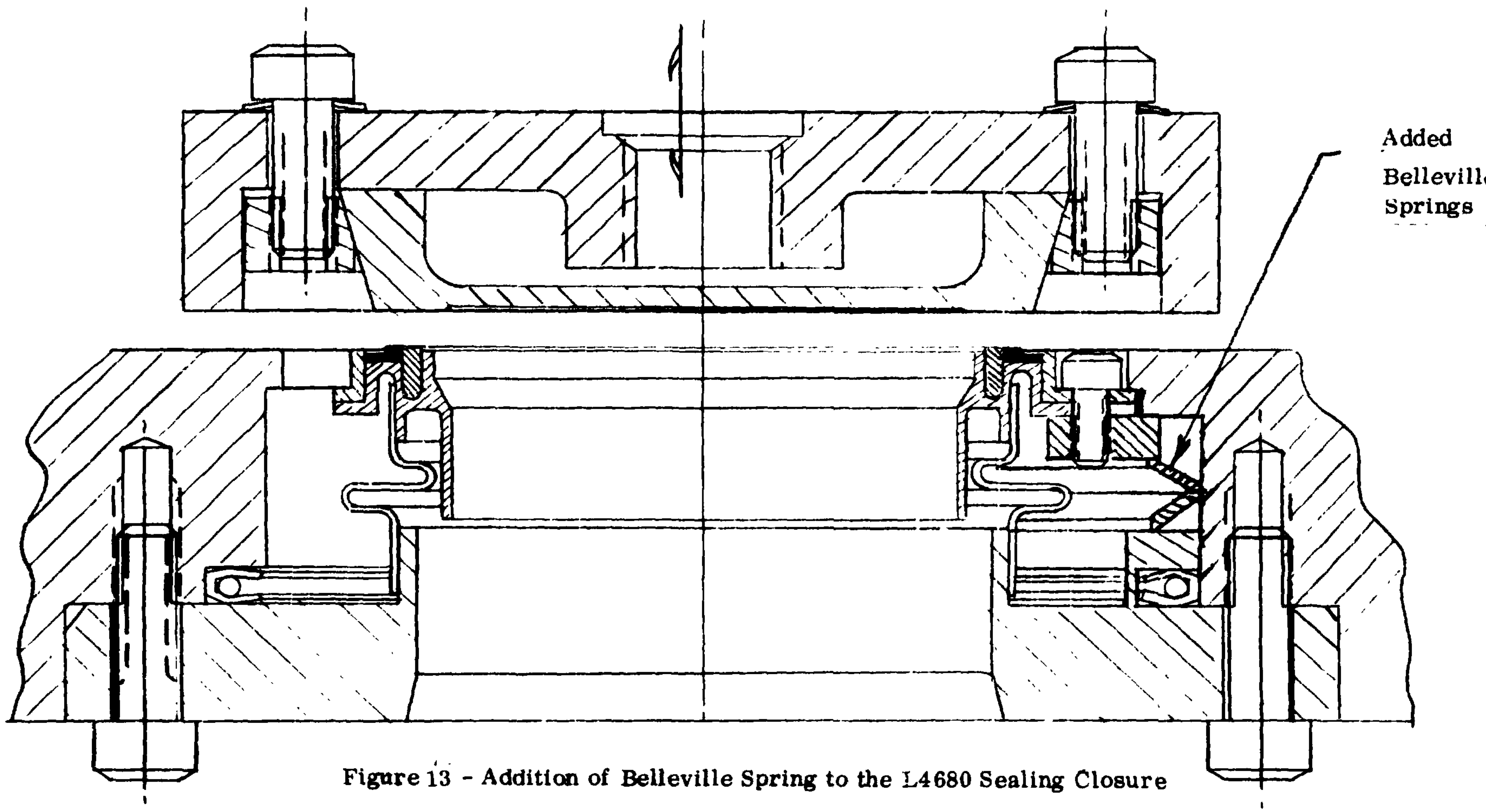


Figure 13 - Addition of Belleville Spring to the L4680 Sealing Closure

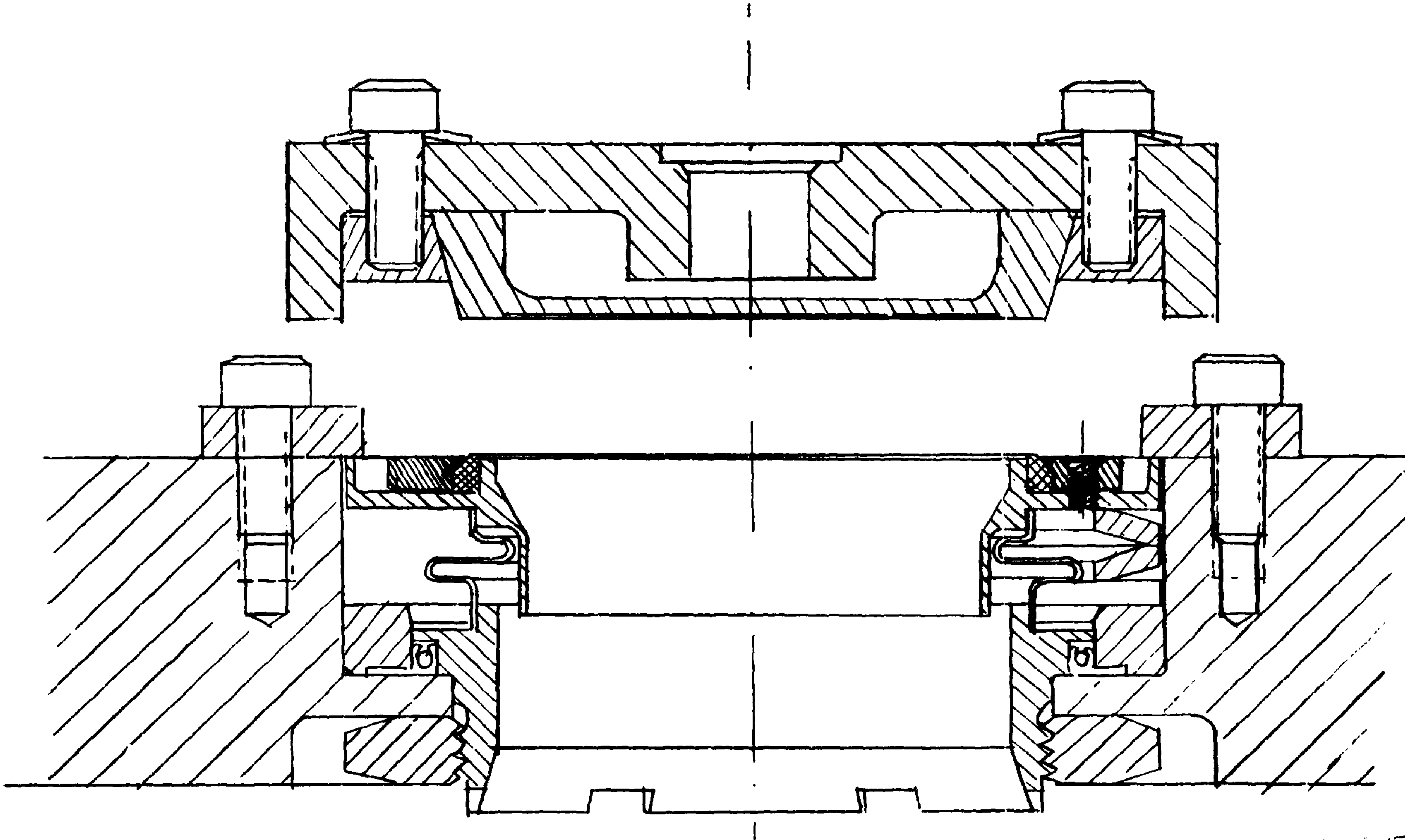


Figure 14 - Polyimide Seal Retention Modification to the L4685 Sealing Closure

copper lip sealing closure and to pursue instead analysis and design of a gold-plated flat sealing closure. This effort resulted in the seal configuration shown in Figure 15. The sealing closure consists of a flat bottom poppet utilizing either type 440C or tungsten carbide lapped to a 1AA finish. The seat consists of a lip seal with a thickness of approximately .020 inch (.076 cm). This lip seal includes a .020 inch to .025 inch (.151 N - .064 N) sealing land which has been slightly crowned and which is gold-plated.

Fabrication of the Sealing Closures

Initial release of sealing closure details for fabrication was made in December of 1970. At that time, also critical long lead time components were ordered. Final detail designs of the sealing closures included subcomponents which required considerably longer lead times than had originally been allowed for. In particular, the pacing item during the fabrication turned out to be the tungsten carbide poppets. Tungsten carbide blanks for these poppets were procured from Kennametal and were received late from the vendor on February 15, 1971. These blanks were subsequently machined and lapped.

During fabrication, several problems were encountered which required minor modifications in some of the designs. For example, the spherical teflon lip seal, identified as L4686 (Figure 9) had a spring rate that had a much greater stiffness than had been planned. The other sealing closures which were fabricated utilized bellows and Belleville springs to achieve precise seat load control. The bellows for these sealing closures were delivered by Gardner Bellows on February 15, 1971. Inspection disclosed that the vendor had decided to deliver bellows with 1-1/2 convolutions rather than the one convolution in order to meet the spring rate and effective diameter requirements established by Marquardt. The bellows as originally ordered and as delivered are shown in Figure 16 and Figure 17. This change in configuration made it necessary to trim the bellows lengths in order to fit them into the overall length of the sealing closures. In addition, the close spacing of the inside half convolution to the bellows end made it necessary to remove the flow contour shields on sealing closures L4682, L4685 and L4686 because of the resulting interference. Another problem with the delivered bellows was the fact that the bellows ends lacked concentricity requirements. Thus, considerable rework of the sealing closures was incurred in order to accommodate the bellows.

During the fabrication of the tungsten carbide seat, P/N L4682, it was determined that the tungsten carbide ring which was brazed to a 300 series support, had cracked during the bracing operation. Subsequent failure investigation disclosed that the mating interface of the tungsten carbide and the 300 series stainless was not compatible with the expansion characteristics of the two materials, as they occur during the braze cycle. Consequently, the 300 series support was

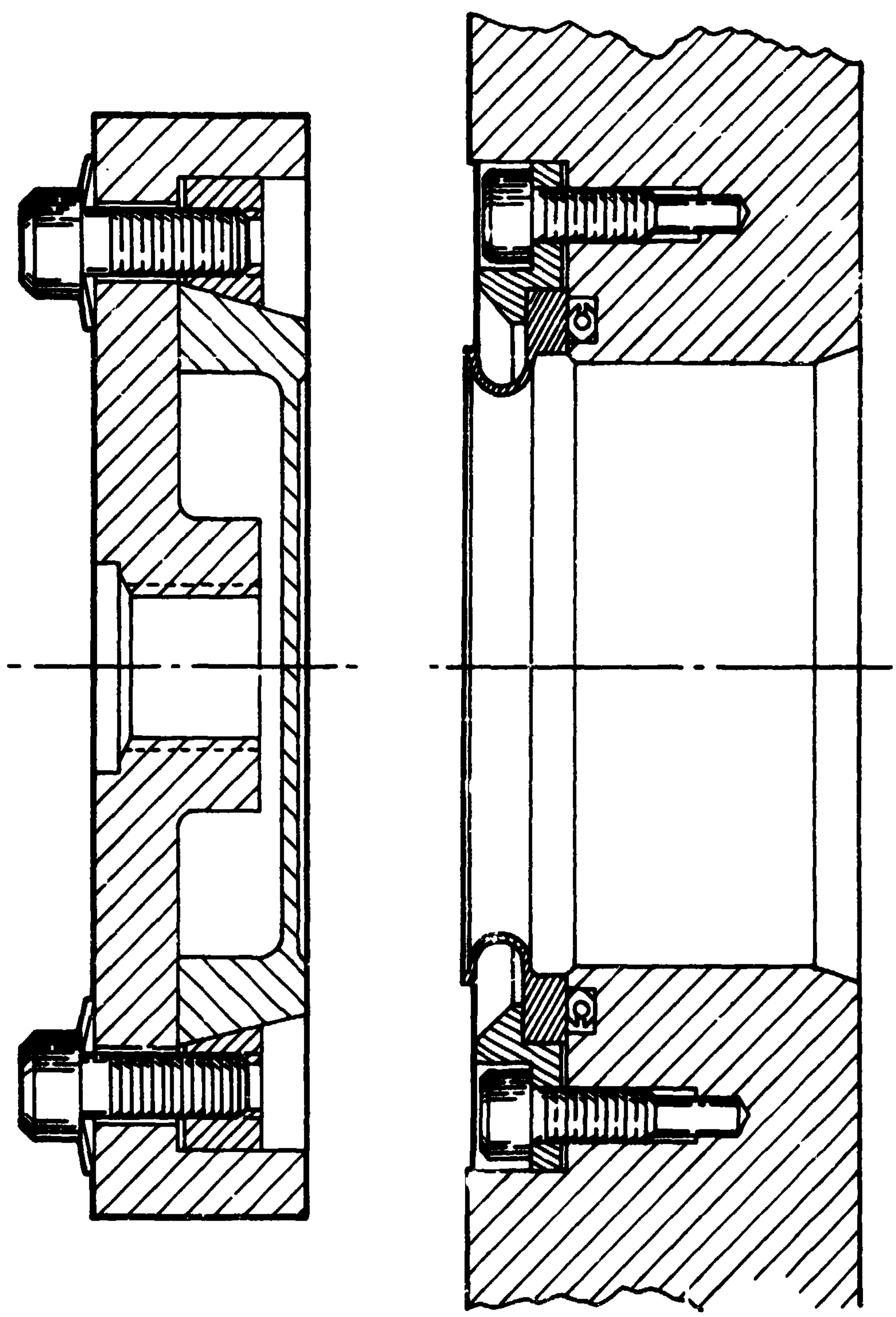


Figure 15 - Gold Plated Lip Seat, Drawing No. L4312

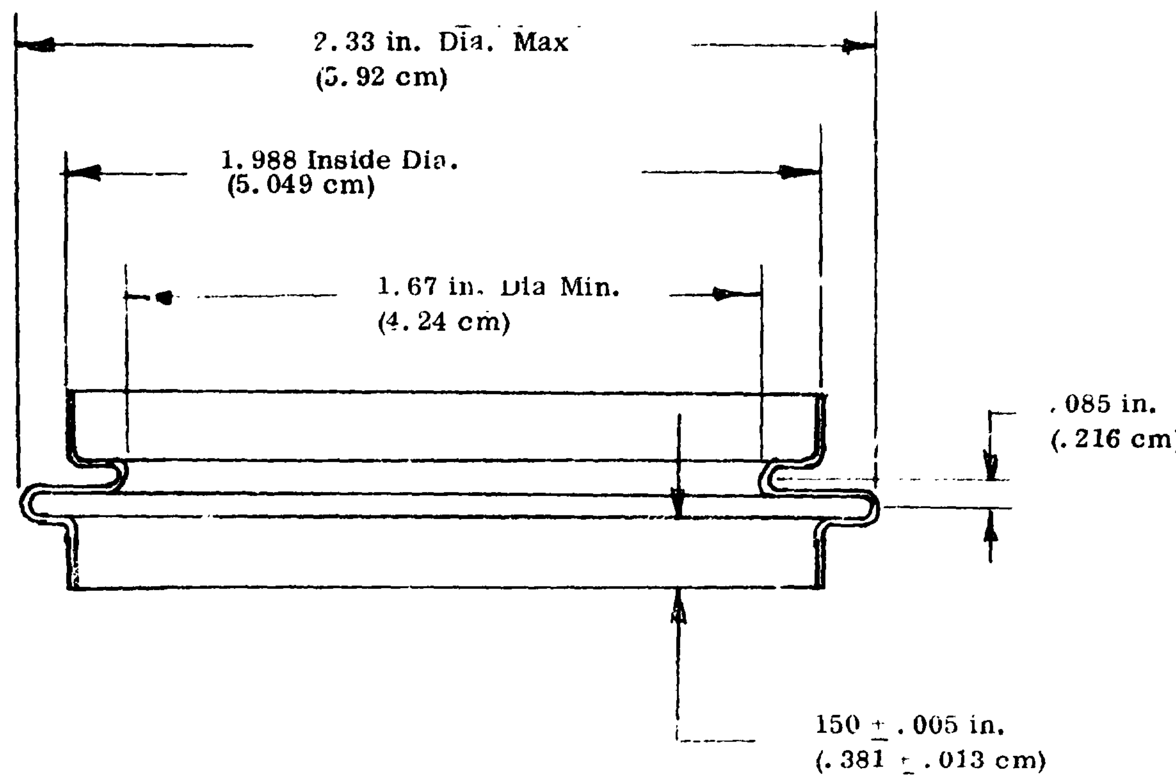


Figure 16 - Bellows as Originally Ordered by Marquardt

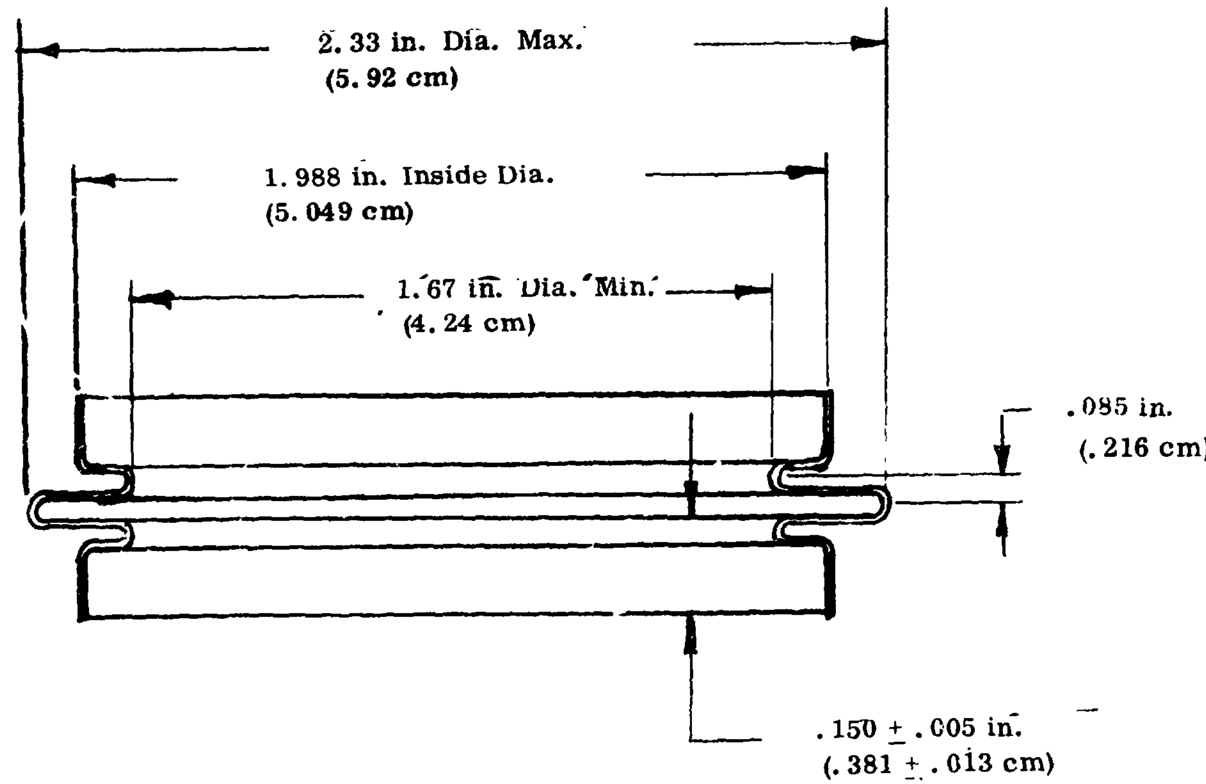


Figure 17 - Bellows as Delivered by Gardner Bellows

completely redesigned to feature a shear joint between the two materials and also to include a single convolution radial bellows. This configuration is shown in Figure 18. Following incorporation of these changes, and overcoming other minor problems that arose, the sealing closures were fabricated. Photographs of some of the sealing closures are presented in Figures 19 through 23. Following fabrication, the closures were subjected to the screening tests.

Rapid Screening Tester

The purpose of this phase of the program was to subject the selected sealing closure concepts to a series of rapid screening tests to evaluate their life-cycle capability. These tests were accomplished by means of a rapid screening tester. For this contract, the government furnished the rapid screening tester P/N IT13602-1, which was previously used under NASA Contract NAS 3-12029. After a careful review the tester design was considered inadequate to meet the needs of this program and was substantially redesigned. The only still-useful component of the tester was the thermal conditioning jacket. The following sections describe the rapid screening tester design criteria and the detail design which evolved from this criteria.

Design Criteria

The primary function of the rapid screening tester was to develop a capability that would allow the evaluation of a large number of sealing closures within a short period of time. Evaluation of the sealing closures was accomplished by rapidly cycling the tester at the design load with the proper impact, and inlet pressure conditions over the required temperature range. Periodically the sealing closures were leak checked during the cycling to verify satisfactory leakage characteristics. Based on these evaluation tests, the best performing sealing closures were pinpointed and the most suitable actuators, linkages, and supporting parts for these sealing closures were defined in order to subsequently achieve the most optimum valve design. The design of the rapid screening tester was completed, and submitted to the NASA-LeRC Project Manager for approval. Following approval, fabrication of the rapid screening tester was initiated.

The operating characteristics of the rapid screening tester were, of course, the same as those defined for the Space Shuttle Auxiliary Propellant Valves as presented in Table I. To limit the number of sealing closures to be evaluated, it was decided to choose one sealing closure size only and to scale up or down from this size analytically to the specific valve size requirements of the high pressure and low pressure valves. The final choice of the rapid screening tester sealing closure size was a 2 inch (5 cm) nominal diameter. This choice mentioned in section "Sealing Closure Sizing", was

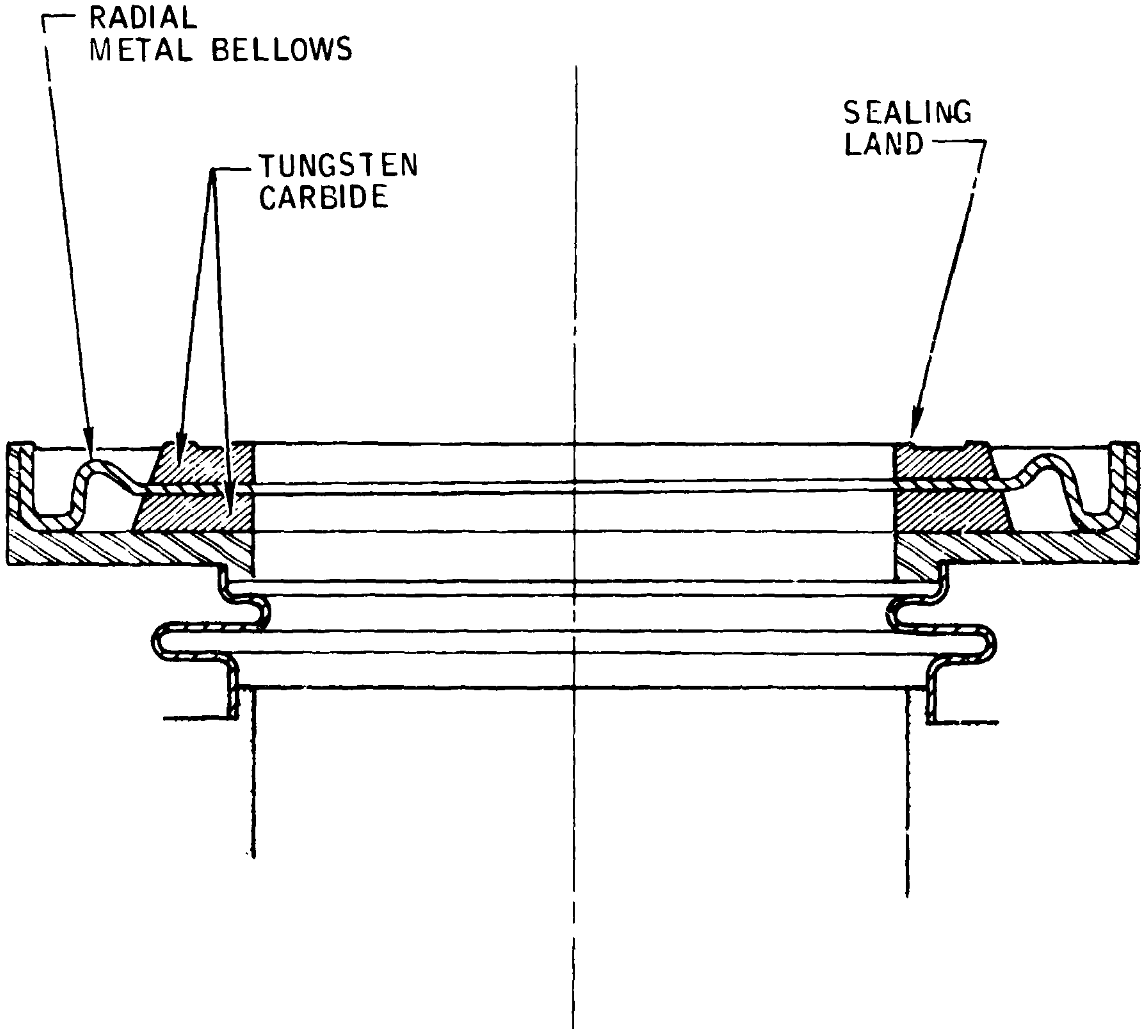


Figure 18 - Tungsten Carbide Seat (Cross Section)

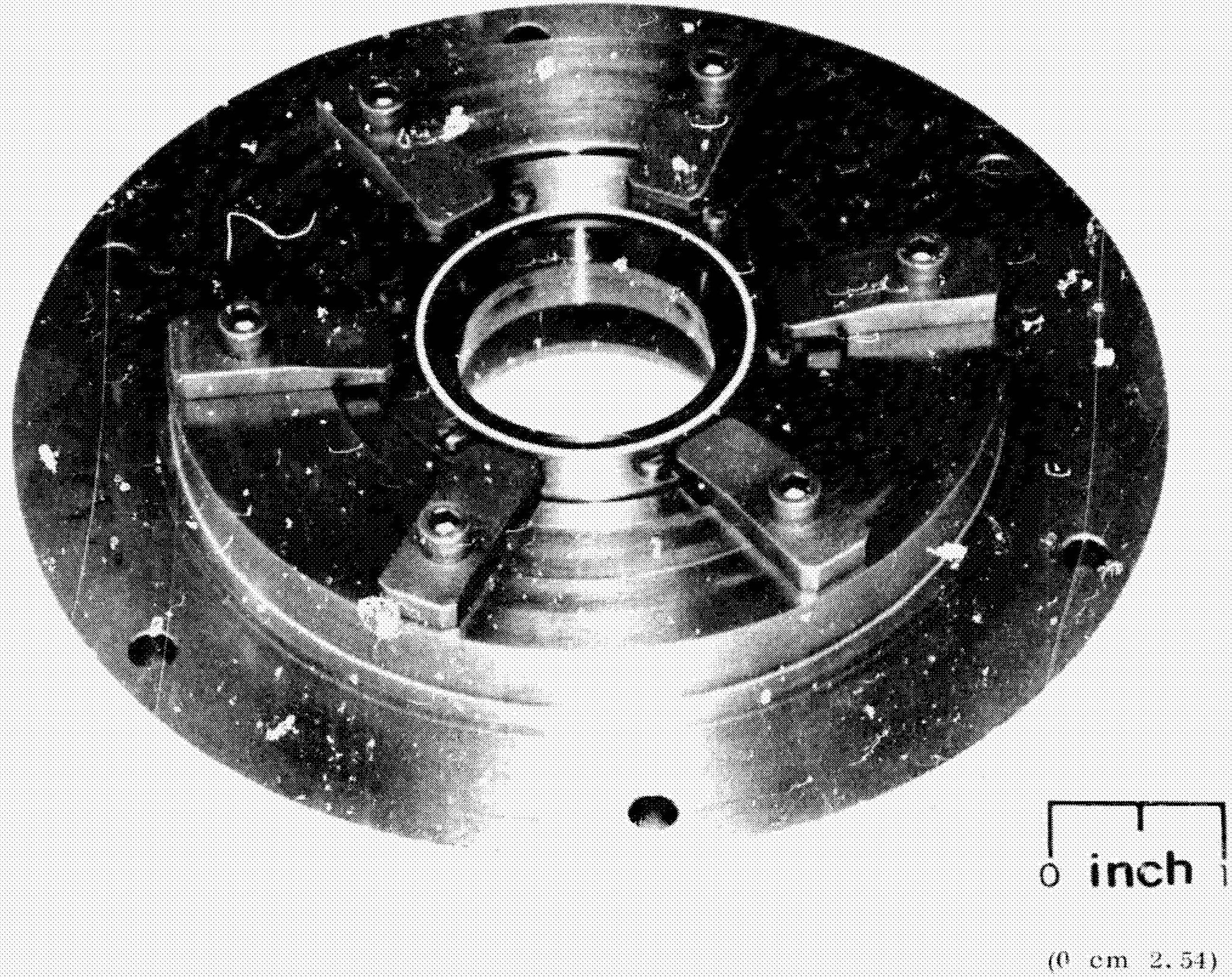


Figure 19 - Flat Teflon Seat - L4680

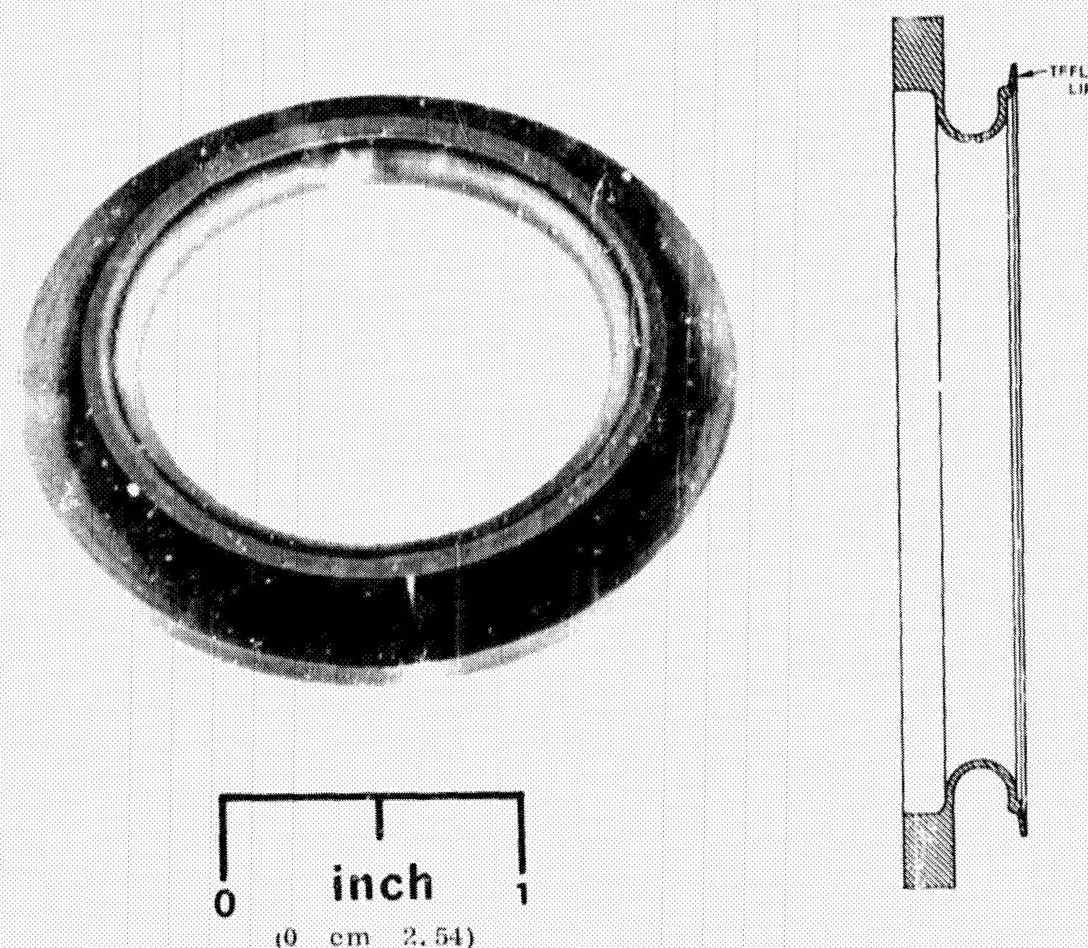


Figure 20 - Flat Teflon Lip Seat - L4683

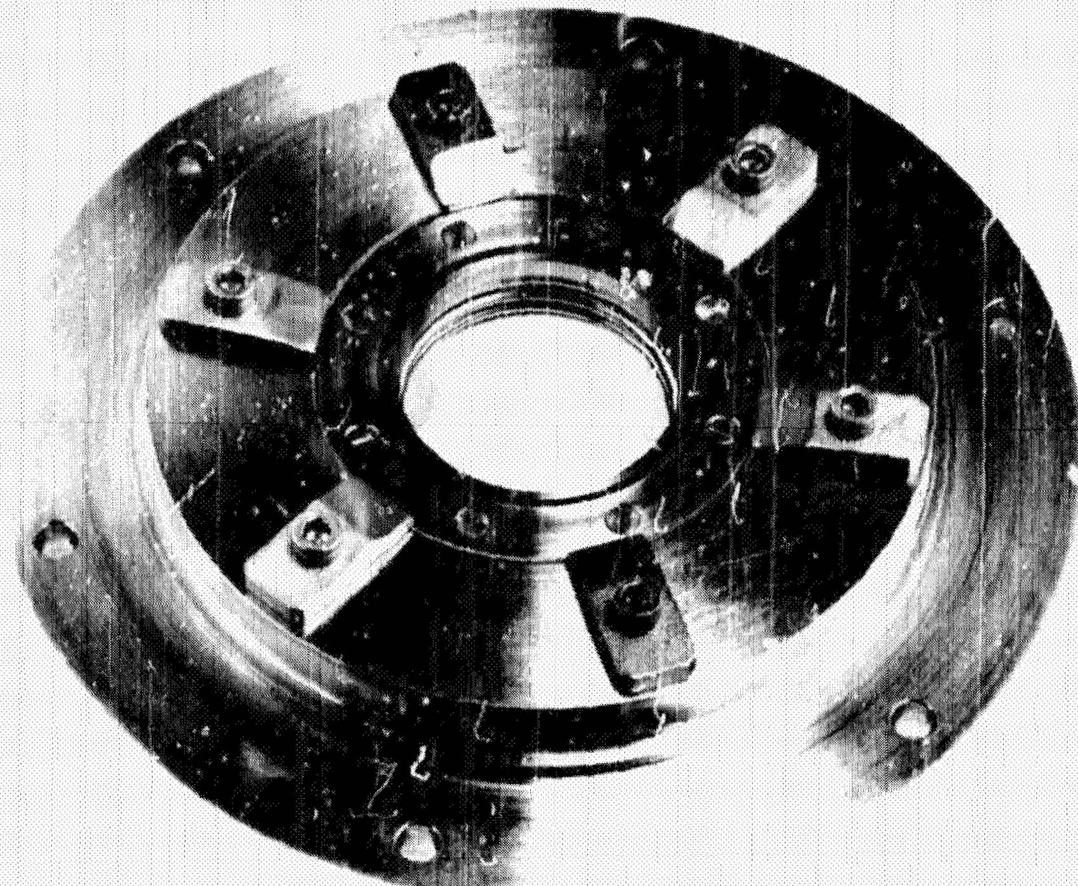


Figure 21 - Flat Polyimide Seat Assembly - L4685

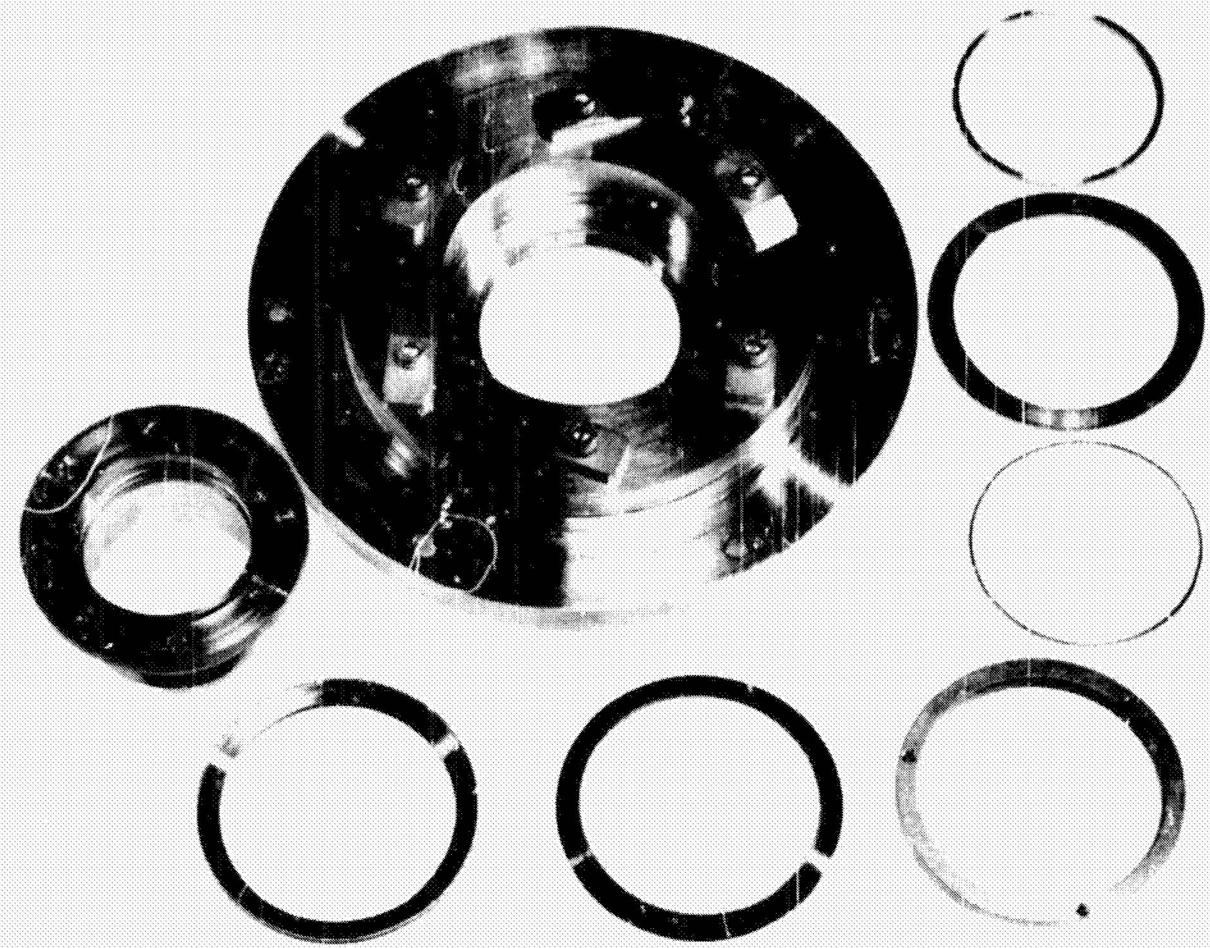


Figure 22 - Components of the Flat Polyimide Seat Assembly - L4685

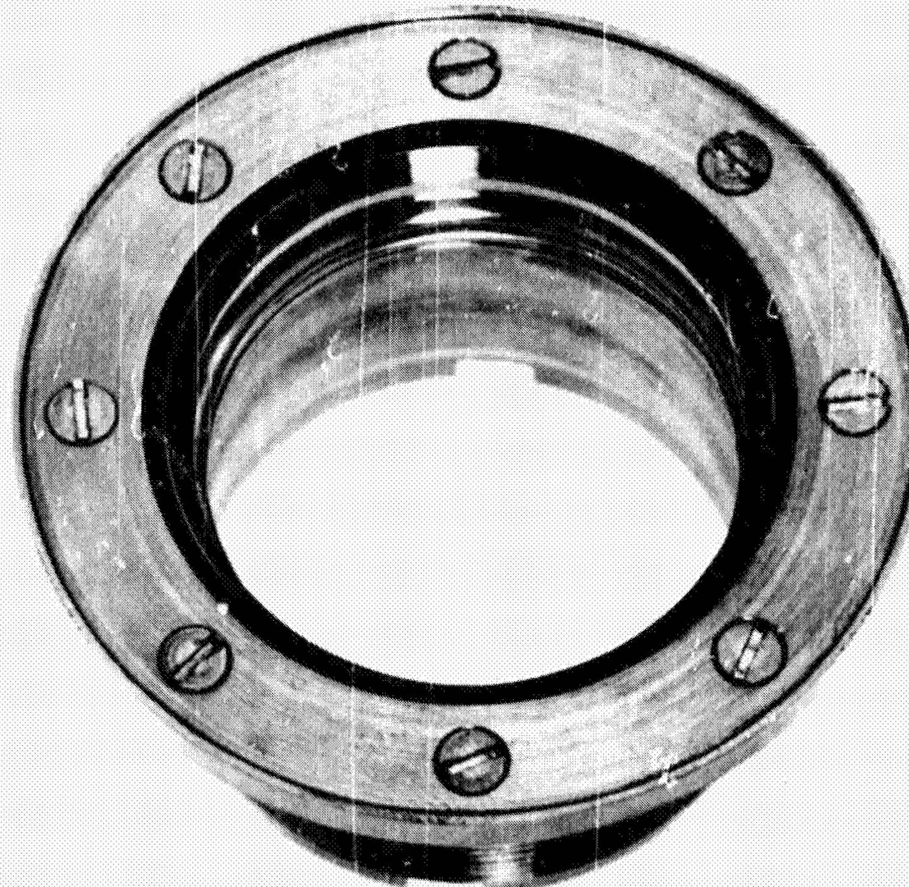


Figure 23 - Spherical Polyimide Seat - L4686

somewhat arbitrar but was based on the fact that a 2 inch (5 cm) diameter lies between the nominal high pressure valve sealing closure diameter of 1 1/2 inches (3.8 cm) and the nominal low pressure sealing closure diameter of approximately 6 inches (15 cm). Since the high pressure valves will be operated at an inlet pressure of 400 psi (276 N/cm^2), this condition was substantially more severe than the low pressure operating condition of 20 psi (14 N/cm^2) with respect to leakage characteristics, and therefore prompted a choice of the rapid screening tester sealing closure nominal diameter closer to the 1 1/2 inch (3.8 cm) high pressure size than the 6 inch (15 cm) low pressure size.

Because of the high response characteristics of these valves, the impact loads incurring during closure are quite substantial and it was essential that these impact loads be properly simulated. A discussion of these impact loads also was presented in section "Sealing Closure Sizing". It was then necessary to trade off the moving mass of the rapid screening tester with the stroke and the actuation force requirements in order to achieve the range of impact kinetic energies of interest. During the design, it became evident that the moving mass of the rapid screening tester was substantially greater than the moving mass of a flightweight valve. To minimize this mass, it was decided to incorporate a spring joint between the poppet assembly and the actuator yoke and cylinder, such that the mass of the yoke and actuating cylinder could be disregarded in the kinetic energy determination. Still, the moving mass of the poppet assembly was greater than that of a flightweight valve and this prompted the utilization of shorter strokes than would normally be experienced in the flightweight valve. Thus, the rapid screening tester did not have the capability of simulating kinetic energies during closure and resultant impact forces.

Another design criteria for the rapid screening tester was the desirability of providing repeatable closure motion while allowing some radial play as might be expected in the design of a sliding fit. As is evident from the discussions in sections "Analytical Leakage Model" and "Sealing Closure Conception and Design", any scrubbing motion which occurs during the mating of the halves of the sealing closure will substantially affect the wear capability of the sealing closure. In effect, preliminary analysis has shown that in order to achieve one million cycles without exceeding the leakage requirements of 100 scc's per hour of helium at the operating inlet pressure, it would be necessary to design into the valve extremely low scrubbing distances of the order of .001 inches (.0025 cm). To permit the evaluation of this most critical scrubbing distance, it was necessary to provide the tester with a feature which allowed the variation in radial play of the moving poppet.

The final design criteria for the rapid screening tester is listed in Table VI. As was mentioned, upon review of the rapid screening tester, P/N IT13602-1, which was furnished by the government for this program, it was determined that a major redesign had to be undertaken in order to make the performance characteristics compatible with these requirements. The final design configuration of the rapid screening tester is discussed in the next section.

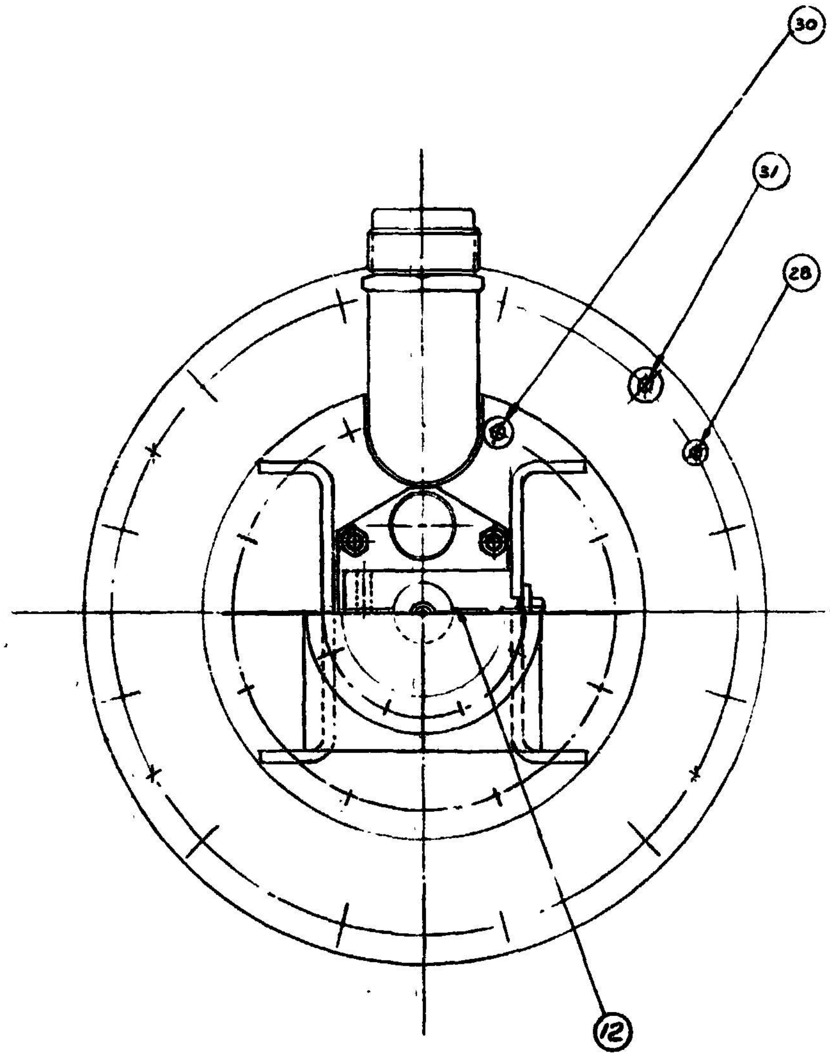
Detailed Design

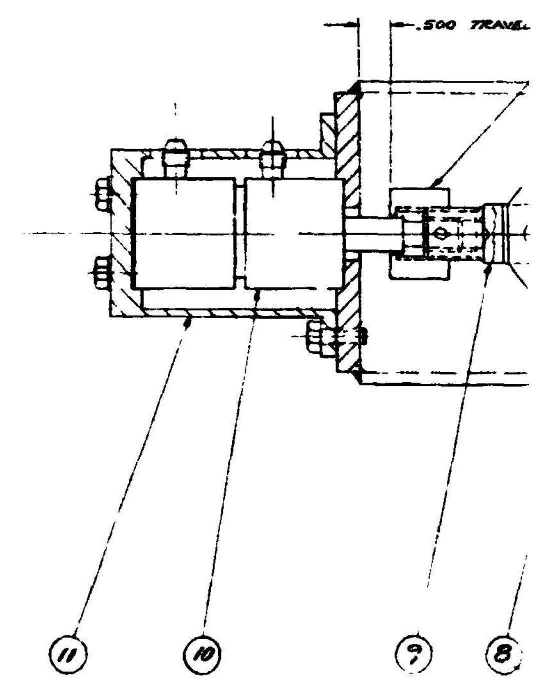
Redesign of the government furnished rapid screening tester resulted in the configuration as shown in Drawing L4688 (Figure 24). Of the components shown in this drawing, only the double-wall pressure vessel remains unchanged. As mentioned previously, one of the prime objectives of the tester design was to permit rapid installation and removal of sealing closures so that a large number of sealing closures could be evaluated within a short period of time. The design shown in Figure 24 readily accomplished this objective. The actuator, yoke, position transducer, and spring coupling can be removed from the poppet assembly by removing a total of 12 bolts. Subsequently, the lid of the tester, which includes the poppet assembly, flexure guidance and sealing closure, can be separated from the double wall chamber by removing the 12 lid bolts. Once the lid assembly is removed, both the seat and poppet become readily accessible. Two of these lid assemblies were fabricated so that one lid assembly could be built up in the clean room while the other lid assembly was being cycled in the rapid screening tester at the test facility.

To achieve perfect repeatable axial guidance of the poppet, two sets of axial guidance flexures were designed and spaced at a distance of 4 inches (10 cm). The Marquardt Company has had extensive experience in the design of axial guidance flexurea and has stressed this particular flexure so that a service life in excess of 10 million cycles is easily attain`e. The inner cylinder and the outer cylinders at the flexure are completely brazed assemblies so that no shift in alignment between the poppet and the seat can occur. Initial precise alignment of the poppet and seat were achieved by finish machining of the poppet back-stop surface, such that it was parallel within 0.0001 inches (.00025 cm) to the back-stop surface of the seat. Concentricities of the poppet to the seat were achieved in a similar manner by finishing the locating diameters of each of these parts within a very tight tolerance. Upon installation of the poppet and seat into the lid assembly, a final parallelism and concentricity check were made on the poppets and seat with an electronic indicator by scribing on surfaces provided specifically for this purpose.

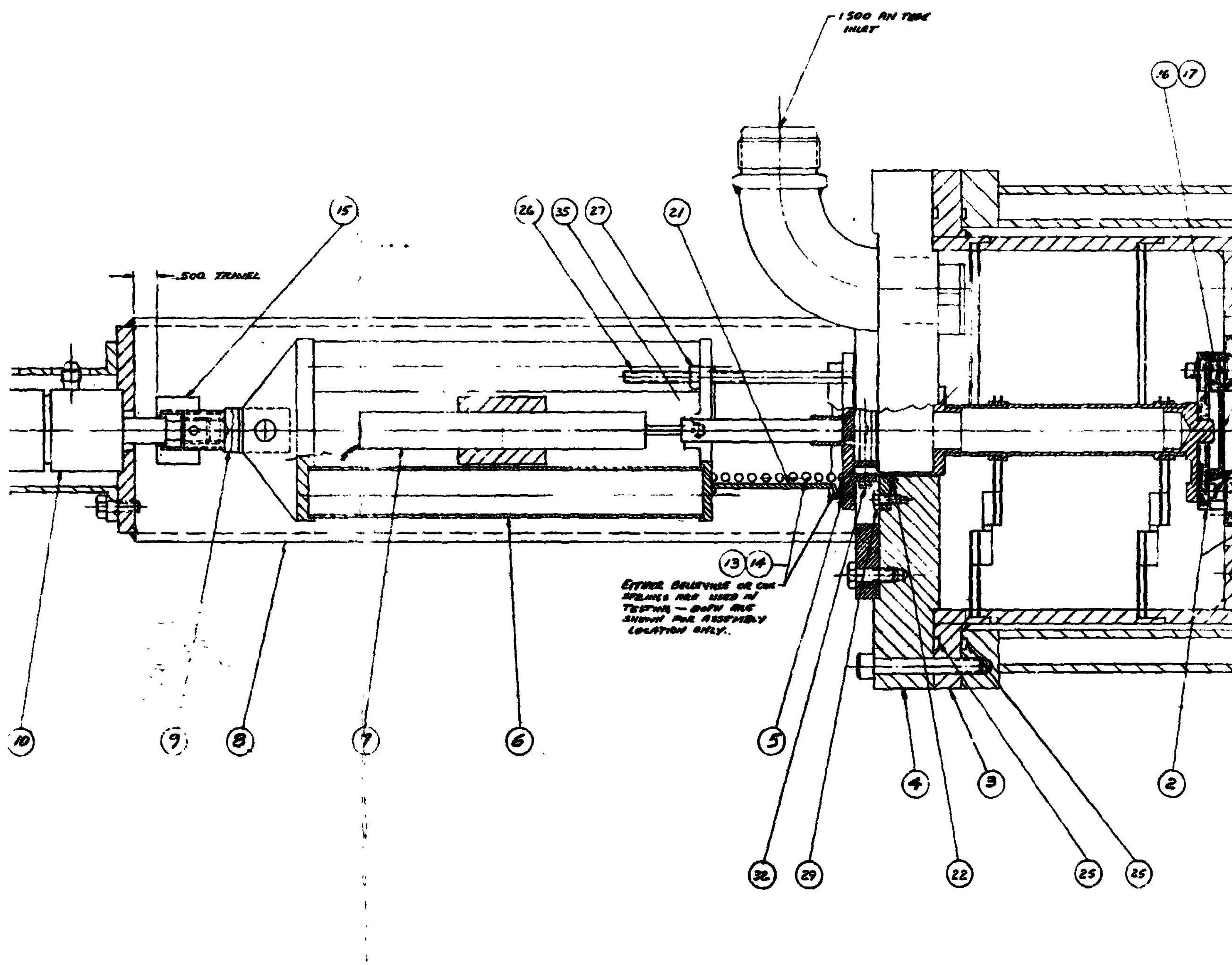
TABLE VI - RAPID SCREENING TESTER DESIGN REQUIREMENTS

Operating Pressure	20	-	400 psi
	(14	-	276 N/cm^2)
Operating Temperature	200	-	850^oR
	(111	-	472^oK)
Stroke Closed Load Capability	20-	-	1500 lbs
	(89	-	6672 N)
Closing Force Capability (dynamic)	10	-	200 lbs
	(44	-	890 N)
Opening Force Capability (dynamic)	10	-	200 lbs
	(44	-	890 N)
Stroke	0	-	1" Maximum
	(0	-	2.54 cm)
Operating Frequency	10 cps		
	(HZ)		
Linear Guidance Capability	0	-	.010" Radial Play
	(0	-	.025 cm)
Impact Kinetic Energy	500	-	2500# in^2/sec^2
	(.1463	-	.7315 N-M)
Nominal Sealing Closure Diameter	2 in		
	(5.1 cm)		





FOLDOUT FRAME



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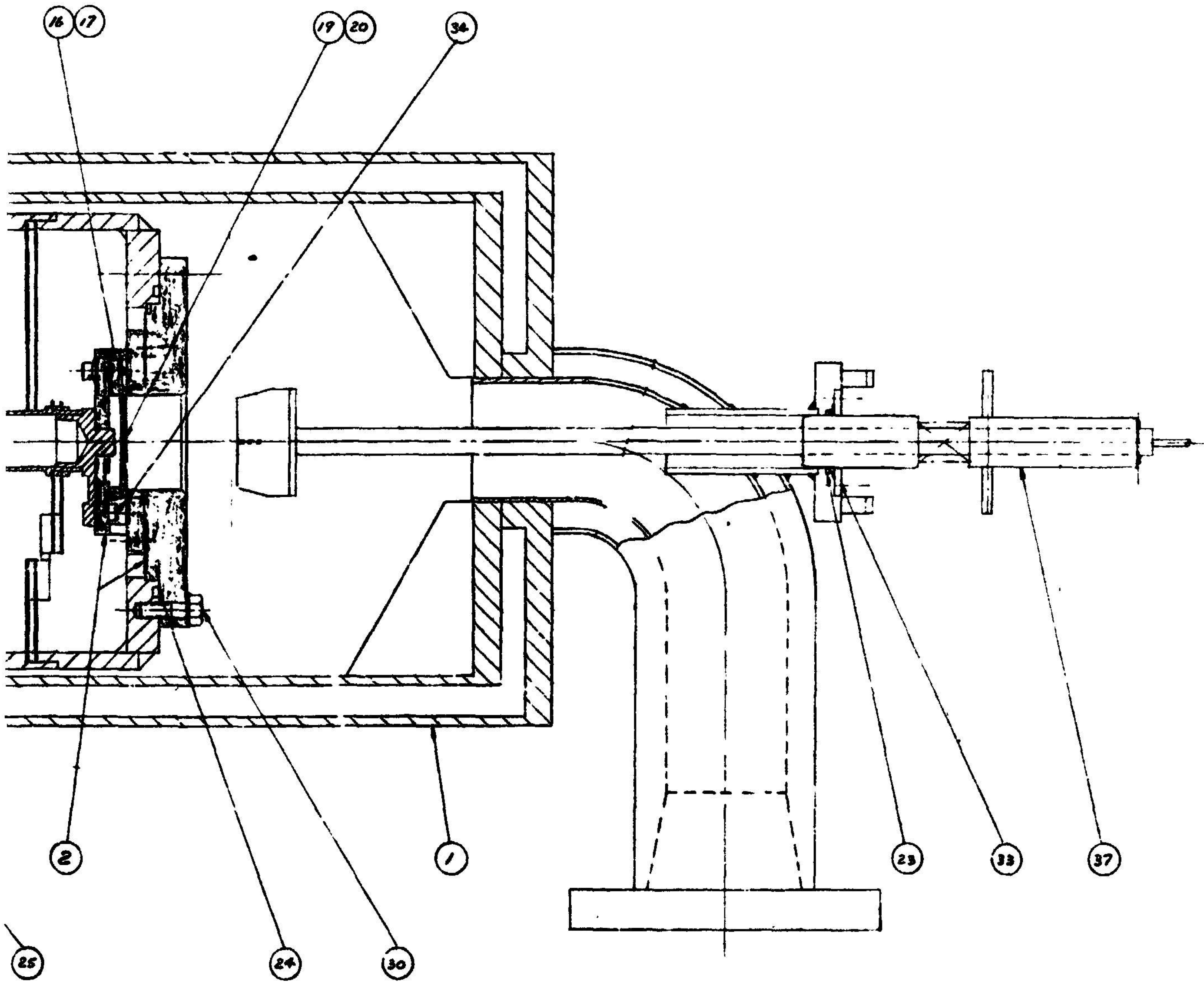
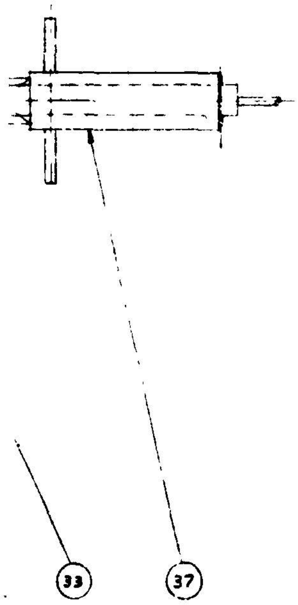


FOLDOUT FRAME 3

DESIGN SPECIFICATIONS

DESIGN PRESSURE OPERATING 20 PSIA. TO 45
DESIGN TEMPERATURE - 200°R TO 850°R
STROKE - AVAILABLE 0 TO 10 IN. DESIGN - 0 TO
SEALING FORCE WITH COIL SPRINGS - 30 TO 800
SEALING FORCE WITH BELLEVILLE SPRINGS - 1500
KINETIC ENERGY @ CLOSURE WITH COIL SPRINGS -
KINETIC ENERGY @ CLOSURE WITH BELLEVILLE SPRINGS

ZONE	LTR	REVISIONS DESCRIPTION	DATE	APPROVED



QTY REQD	CODE IDENT	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM NO.
							37
1		X27666	PLUG ASSY				36
2			SHOULDER SCREW	[illegible]	CRES		35
6			SCREW	8-32 X 2	CRES		34
6			SCREW	[illegible]	CRES		33
1			PIN	[illegible] X 2 5/8	CRES		32
12			SCREW	3/8-16 X 2 1/2	CRES		31
16			SCREW	[illegible]	CRES		30
6			SCREW	10-32 X 1/2	CRES		29
6			SCREW	[illegible]	CRES		28
4			NUT	1/4-28	CRES		27
4			STUD	1/4-28 X 5 1/4	CRES		26
1			OMNISEAL	AR10205 272			25
2			OMNISEAL	AR10205-255			24
1			OMNISEAL	AR10110-120			23
1			OMNISEAL	AR10110-136			22
1		X27661	TUBE		CRES 321		21
1		X27665	POPPET-SPHERICAL		K-96 TUNGSTEN CARBIDE		20
1		X27653	POPPET		K-96 TUNGSTEN CARBIDE		19
1		X27655	PLATE		CRES 321		18
6		X27652	WEDGE-CONICAL		CRES 347		17
6		X27664	WEDGE-SPHERICAL		CRES 347		16
1		X27650	STOP		A 286		15
2		X27472	BELLEVILLE SPRING		PERMET X15		14
1		X27501	SPRING				13
1		X27654	LVDT MOUNT ASSY		ALUM 6061 T6		12
1		X27656	ACTUATOR MOUNTING ASSY		[illegible] 4130 STEEL		11
1			ACTUATOR	RACINE MODEL A 1[illegible]-1-1-[illegible]			10
1		X27651	ACTUATOR COUPLING		CRES A-286		9
1		X27663	ACTUATOR MOUNT ASSY		CRES TYPE 300		8
1			LVDT	SCHAEVITZ TYPE AC-[illegible]			7
1		X27660	ACTUATOR LINKAGE ASSY		CRES TYPE 300		6
1		X27659	LVDT [illegible] ASSY		CRES TYPE 300		5
1		X27502	CLOSURE PLATE ASSY		CRES TYPE 321		4
1		X27662	FLEXURE ASSY		INCO 718 CRES TYPE 300		3
		X27658	POPPET SUPPORT		PYROMET X15		2
1		MDAC DWG 1713603	COOLING JACKET		EXISTING		1

LIST OF MATERIALS OR PARTS LIST

FOLDOUT FRAME 4

DESIGN SPECIFICATIONS

DESIGN PRESSURE OPERATING 20 PSIA. TO 435 PSIA.
DESIGN TEMPERATURE - 200°R TO 850°R
STROKE - AVAILABLE 0 TO [illegible] DESIGN 0 TO [illegible]
SEALING FORCE WITH COIL SPRINGS - 70 TO 800 POUNDS
SEALING FORCE WITH BELLEVILLE SPRINGS - 1500 POUNDS
KINETIC ENERGY @ CLOSURE WITH COIL SPRINGS - 500 LB-IN²/SEC²
KINETIC ENERGY @ CLOSURE WITH BELLEVILLE SPRINGS - 2500 LB-IN²/SEC²

DRAWN A.M. [illegible]

CCI AEROSPACE CORPORATION VAN NUYS, CALIFORNIA

RAPID SCREENING TEST FIXTURE

J 88845 L 4688

Figure 24

Analysis of the force requirements of Table VI disclosed that the pressure forces acting on a 2 inch (5.08 cm) diameter surface at 400 psia (275.8 N/cm^2) inlet pressure were much greater than the actuation force requirements. Consequently, to permit better control over the valve opening and closing times and, more importantly, over the kinetic energies during closure, it was considered absolutely esential to pressure balance the poppet. This pressure balance feature was accomplished by making the outer diameter of the moving center shaft of the poppet assembly where it came through the lid equal to the poppet effective sealing diameter. Since the pressures outside the rapid screening tester and downstream of the sealing closure were essentially equal at all times, this design approach effectively eliminated the pressure unbalancing forces. Any pressure unbalance which remained due to machining tolerances was determined precisely by matching the force output of the hydraulic actuator with the pressure force generated when the rapid screening tester was pressurized to the 400 psi (275.8 N/cm^2) operating condition.

The rapid screening tester is of an all stainless steel construction with omniseals used both as static seals where necessary, and as the dynamic seal where the poppet assembly shaft penetrates the lid.

A hydraulic actuator operating with pressures up to 3000 psi (2068 N/cm^2) was chosen to operate the sealing closure since the precise control of static as well as dynamic forces could be more easily accomplished with a hydraulic actuator than with a pneumatic actuator. Since the operating temperature of a hydraulic actuator is however, more limited than the 200-850° R (111-422° K) temperature range of the rapid screening tester, this actuator was thermally isolated from the tester by means of a support structure and actuating yoke. During checkout testing of the tester, temperatures at both the hydraulic actuator and the position transducer were monitored and additional cooling or heating could be provided to the actuator support structure as required. The static and dynamic force outputs of the hydraulic actuator were varied independently by providing check valves and variable area needle valves in parallel in each of the hydraulic supply lines to the actuator. Thus by setting the hydraulic supply pressure to a specific value, a specific load could be generated; and by adjusting the needle valve, a lower dynamic load could be produced.

The interface between the hydraulic actuator yoke and the poppet assembly was a spring coupling which included both a coil spring and Belleville washer. The range limit of the coil spring was approximately 200 lb (890 N), that of the Belleville washer - 1500 lb (6672 N). The Belleville washer was to be employed only if the closing dynamic force was in excess of 200 lb (890 N). The spring joint served two purposes. One was that it eliminated alignment problems between the actuator assembly and the poppet assembly. The other was that it decoupled the actuator and yoke mass from the poppet assembly mass when the poppet reached the closed position, thereby permitting more precise control of the closing kinetic energy and the resultant impact force.

Temperature conditioning of the rapid screening tester was accomplished by flowing cold gaseous nitrogen or heated air through the double-walled jacket. A leakage test fixture was also added to the rapid screening tester. The purpose of the leakage test fixture was to minimize the volume immediately downstream of the sealing closure interface and to minimize the effects of thermal transients on that volume and thereby on the true leakage rate. The test fixture consisted of a teflon plug which was positioned into the seat orifice and spring loaded against it with a coil spring which in turn was preloaded manually and locked in place by the twist of the wrist. A 1/8 inch (.3175 cm) diameter tube connected the volume upstream of the leak test fixture to a graduated cylinder which was used to measure the leakage rate. The tester also was equipped with other instrumentation such as sealing closure temperatures, jacket temperatures, actuator pressure drop, variable reluctance position transducer readout, etc.

The rapid screening tester was stress analyzed to assure satisfactory operation. Criteria employed in this stress analysis was as follows:

1) The flexures had to be capable of guiding the piston and poppet for one inch (2.54 cm) total travel, with the null point at 1/2 inch (1.27 cm) stroke, for at least 1 million cycles. (In practice, the strokes will be substantially shorter and the flexure life much greater than the 1 million cycles.)

2) The tester flexures were to be designed for an axial stiffness of 19.8 lb/in/plate (3.54 Kg/cm) and a radial stiffness of 2670 lb/in/plate (477 Kg/cm).

3) The tester jacket pressure requirement was 435 psi (300 N/cm^2) nominal and 670 psi (462 N/cm^2) proof. The jacket was to be capable of withstanding these pressures for a least 1 million cycles.

The detail stress analysis of various components of the tester is presented in Appendix B. A brief summary of the safety margins and operating stresses of various components of the rapid screening tester is shown in Table VII.

TABLE VII - MINIMUM STRESS MARGINS - TESTER

Part Name	Condition	Remarks	Stress (ksi) of Loads (k)	(K N/cm^2)	Margins of Safety
Tester Flexure	y = .500 in. (1.27 cm)	Bending t Torsion on Flex 10^6 cycles	78	(53.8)	.00
Tester Jacket	670 psi proof (462 N/cm^2)	Bending on Inner Wall	33.5	(23.1)	.07
Tester Jacket	Inner Plate	Bending on Plate + Web	20	(13.8)	.75
Attach Bolts	670 psi Proof (462 N/cm^2)	THD Shear in Plate Yield	18.3	(12.6)	.00
Tester End Cap	600 psi Proof (414 N/cm^2)	Bending on End Cap Min. "t"	.032	(.022)	Large
Tester Poppet Support	Max. Impact	Column Load on Support Tube	27.7	(19.1)	.08

Fabrication of the Rapid Screening Tester

Fabrication of tester details was initiated in December 1970 and completed in February 1971 except for the final assembly of the tester. Also in December 1970 procurement of all purchased parts and subcontracted parts was initiated. Except for the cooling jacket which was available from NASA Contract NAS3-12029 all parts shown in Figure 24 had to be fabricated or procured. Essentially one complete tester assembly was fabricated except that two of the flexure assemblies (P/N X 27662), two hydraulic actuators and from 2 to 6 seals for each application depending upon criticality of the seals were procurred or fabricated. The two flexure assemblies were required to permit build up of one assembly with a sealing closure in the clean room while the second assembly was being tested in the tester at the test site. Thus, two fixture assemblies permitted rapid and efficient testing.

Two hydraulic cylinders were procured to gain better actuation and static sealing load control. The smaller cylinder was employed for loads up to 450 lbs. (2002 N) and the larger could generate loads as high as 3000 lbs. (13345N). The most critical seal from the point of view of wear was the 2 inch (5.08 cm) diameter sliding seal utilized in the top lid of the tester where the poppet shaft connects to the actuator assembly. A commonly used performance factor for this type of seal is the product of operating pressure and average sliding velocity. This "PV" factor is over 100,000, a value which indicates severe wear and very short life for a seal operating at that level continuously. However, since the application here is intermittent rather than continuous, satisfactory performance of the custom seal selected was expected. Several spare seals were procured in the event rapid wear out was experienced.

Manufacture of the tester details did not involve any appreciable problems. Manufacturing costs experienced were somewhat higher than originally projected. This was in part due to the use of Inco 718 for several subcomponents. This material costs appreciably more ($6.25/lb.)($13.78/kg) than more common stainless steels but offers excellent spring characteristics for such components as flexures and bumpers. Inco 718 also is not very widely known as was determined by the fact that the first subcontractor to be awarded the task of chem milling the flexure gave up after two tries. The second subcontractor, although he had had prior experience with chem milling this material also had to prepare new etching solutions twice before finally completing the order. Thus the fabrication of the flexures delayed the completion of the tester by approximately one month. Another factor delaying assembly of the tester was the late delivery (by as much as 6 weeks) of Omni seals from Aeroquip Corporation.

Sealing Closure Tests

Prior to conducting the sealing closure tests, a test plan was prepared and submitted to the NASA LeRC Project Manager for approval. The test site was then built up and the rapid screening tests were conducted.

Test Plan

The test plan for the evaluation of the sealing closures in the rapid screening tester was completed and approved by the NASA LeRC Project Manager. This Test Plan is included as Appendix C of this report. This test plan defined all procedures, test sequences, test environment requirements, instrumentation requirements, data requirements and data handling procedures. The test plan also specified critical characteristics which were to be monitored to aid in the determination of the exact operating status of the screening tester throughout the program and established a procedure for dispositioning of discrepant performance or failure of any test item.

Initial testing as defined by the test plan consisted of a complete checkout of the rapid screening test fixture to assure that load/stroke capabilities were within the requirements established during the analysis and conceptual design phase. These tests consisted primarily of a calibration of the force/stroke characteristics and response characteristics. Proof pressure tests were also to be performed to verify structural integrity and leak tight performance of the tester.

Once the rapid screening tester had been checked out, each of the sealing closures was installed in succession and subjected to an evaluation program as defined in the test plan. A summary test matrix of the sealing closure testing is presented as Figure C-3 of Appendix C. The evaluation program consisted essentially of cycling the sealing closure at 400 psia (276 N/cm^2) inlet pressure and at hot (850^oR)(472^oK), ambient, and cold (200^oR) (111^oK) conditions to verify the ability of the sealing closure to seal satisfactorily. Other variables which were explored during the test program were static seat loading, impact loads during closure, and poppet alignment All of the test data were tabulated and utilized for the completion of the program effort.

Test Site Preparation and Test System Description

Upon completion of fabrication of the rapid screening tester it was installed in the southwest corner of Building 37 at the Marquardt, Van Nuys Test Facility. A better understanding of the type of support systems required for operation of the rapid screening tester may be gained from Figure 25. To permit operation of the tester hydraulic actuator a 3000 psi (2068 N/cm^2)

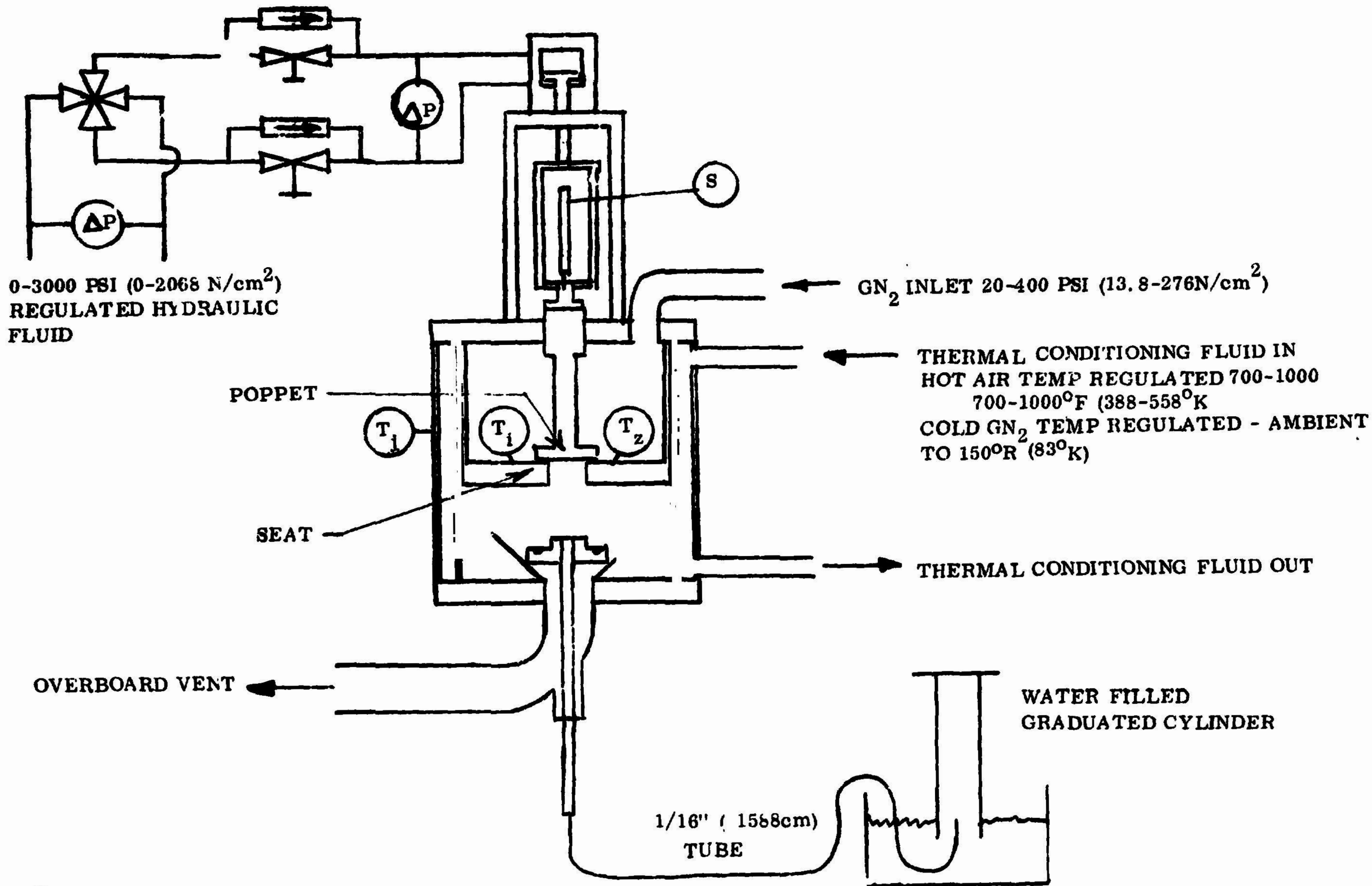


Figure 25 - Rapid Screening Tester Installation

hydraulic pump and hydraulic feed system including accumulator, pressure regulator, pressure transducers, throttling and check valves and servo valve were installed. A frequency generator and pulser driver for the servo valve also were provided.

The gaseous nitrogen supply from the central supply system to the tester was installed. This system utilizes a 2 in. (5.08 cm) diameter 2000 psi (1379 N/cm^2) line to within a short distance of the rapid screening tester and is regulated down to the required 20 to 450 psi (13.8 to 310 N/cm^2) run pressure at that location. A surge tank down stream of the pressure regulator was used to minimize gas pressure variations during the high response actuations of the tester. This system was sized such that the nominal gas flow rates required during later testing for valve pressure drop verifications could be readily supplied.

The Helium supply system also was located adjacent to the tester. This system utilizes portable Helium K bottles and was tied into the gaseous nitrogen supply system to permit leak checks with helium.

Cold conditioning of the rapid screening tester was accomplished by means of a GN_2 and LN_2 heat exchanger and mixing system. This system was available from another test program and was plumbed to the jacket of the rapid screening tester. Hot conditioning was provided by an electric heater which heated a gaseous nitrogen supply for conditioning the tester jacket to the 850°R (472°K) run temperature. An overall view of the sealing closure evaluation test set-up is shown in Figure 26. The rapid screening tester is located just to the right of the center of this photograph with the hydraulic feed system including an accumulator, servovalve, throttling valves, check valves, and differential pressure transducer located towards the upper left of the picture. In the lower left hand corner is a cold conditioning system which employed liquid nitrogen in a heat exchanger to cool gaseous nitrogen to the desired conditioning temperature. In the upper right hand corner of the picture is an electric heater used to heat gaseous nitrogen for conditioning of the rapid screening tester to the maximum 390°F (199°C) temperature.

Figure 27 shows the rapid screening tester actuation mechanism in greater detail. Evident in this picture are the hydraulic actuator at the top and proceeding down along the axis of the tester, the stroke adjustment nut, a yoke within which the LVDT position transducer is located, and the spring joint which isolates the mass of the yoke and the hydraulic actuator from the moving mass of the poppet assembly during impact of the poppet on the seat. Also shown in the right foreground, is the tester inlet filter which is removed with the tester to the Clean Room for removal and installation of the new

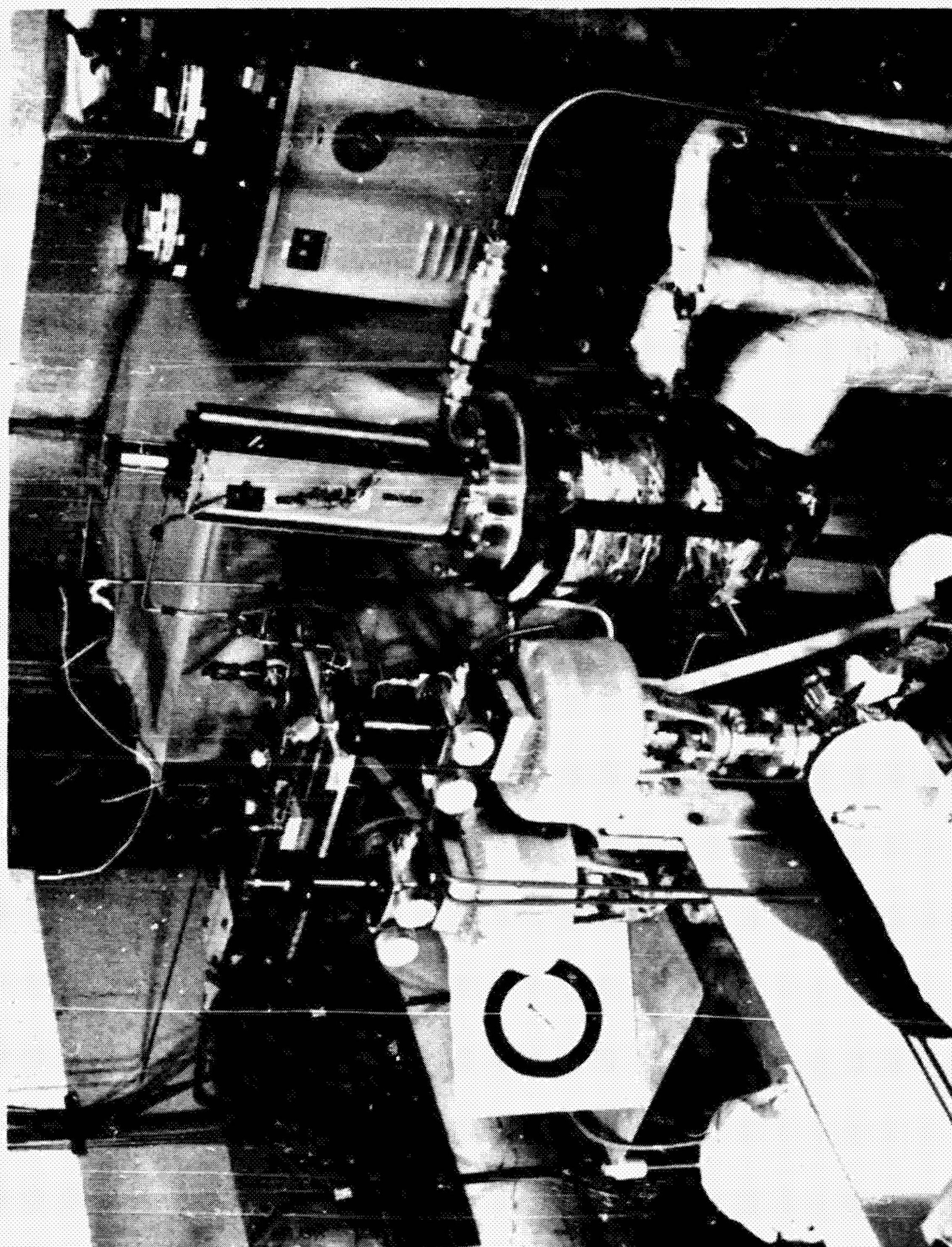


Figure 26 - Sealing Closure Evaluation Test Set Up

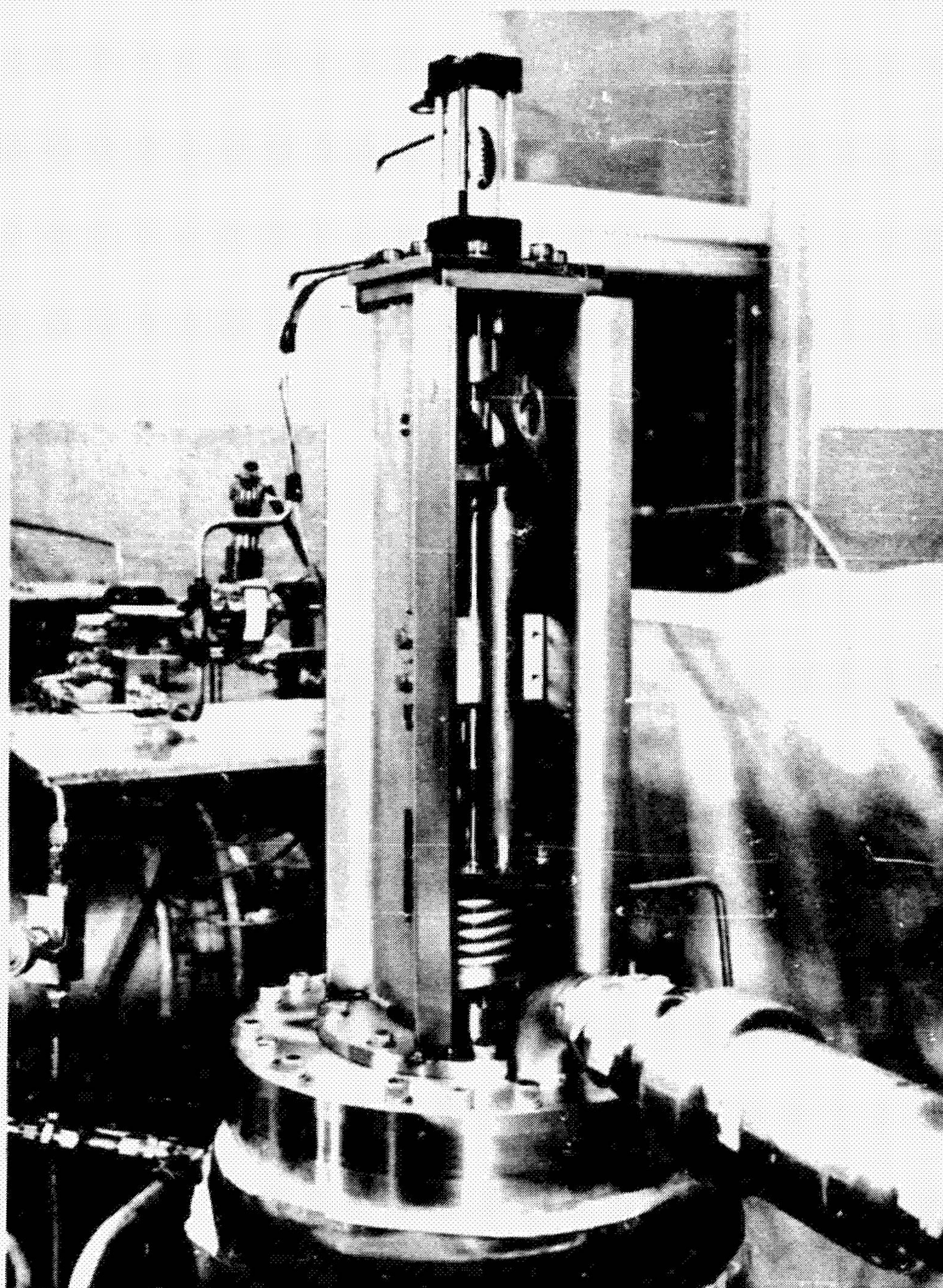


Figure 27 - Rapid Screening Tester Actuation Mechanism

sealing closures. High frequency response instrumentation was used and included a variable reluctance position transducer for the tester and a hydraulic differential pressure transducer for the tester actuator. These outputs were recorded on a dual beam oscilloscope. Sealing closure and tester jacket temperatures were monitored with thermocouples and were recorded on a strip chart recorder. Other instrumentation which was installed included various pressure gages which monitored such parameters as tester inlet pressure, hydraulic supply pressure, helium and GN_2 supply pressures, tester downstream pressure, tester jacket pressure and others. Figures 28 and 29 show some of the recording instrumentation.

Check-out Testing

Check-out testing of the rapid screening tester consisted essentially of three types of tests. These were checkout of the cold conditioning system, checkout of the hot conditioning system, and checkout of the tester actuation mechanism to verify its ability to generate poppet closing velocities of 16 inches per second (.4064 m/s) at a cycling frequency of 10 cps (HZ) and to control static and dynamic sealing closure interface forces.

During the temperature conditioning tests, it became evident that a substantial temperature differential (80^oR) (44^oK) existed between the tester conditioning jacket and the tester flexure assembly. This substantial temperature differential resulted from a 0.1 inch (.254 cm) air space between these two tester parts except at the top mounting flange where actual metal-to-metal was achieved. The substantial temperature differential coupled with the heat losses from the tester flexure unit and the tester actuation mechanism made it difficult to rapidly achieve the desired sealing closure temperatures. Consequently, to improve this condition, a copper liner consisting of two copper sheets with copper wool stuck between the two sheets was fabricated and inserted in the air space. This liner then provided a direct conduction path between the thermal conditioning jacket and the tester flexure units and appreciably improved the heat transfer characteristics.

Cold conditioning of the rapid screening tester to -260^oF (-162^oC) was eventually achieved in a period of approximately 1/2 to 1 hour by simply introducing liquid nitrogen directly to the conditioning jacket. This approach eliminated the liquid nitrogen to gaseous nitrogen heat exchanger equipment. Adequate control of the sealing closure temperature was achieved by simply throttling the liquid nitrogen to the tester jacket.

Hot conditioning of the rapid screen tester to 390^oF (199^oC) turned into quite a problem since the 9 kilowatt electrical gaseous nitrogen heater did not provide sufficient energy output to make up for all the heat losses of the

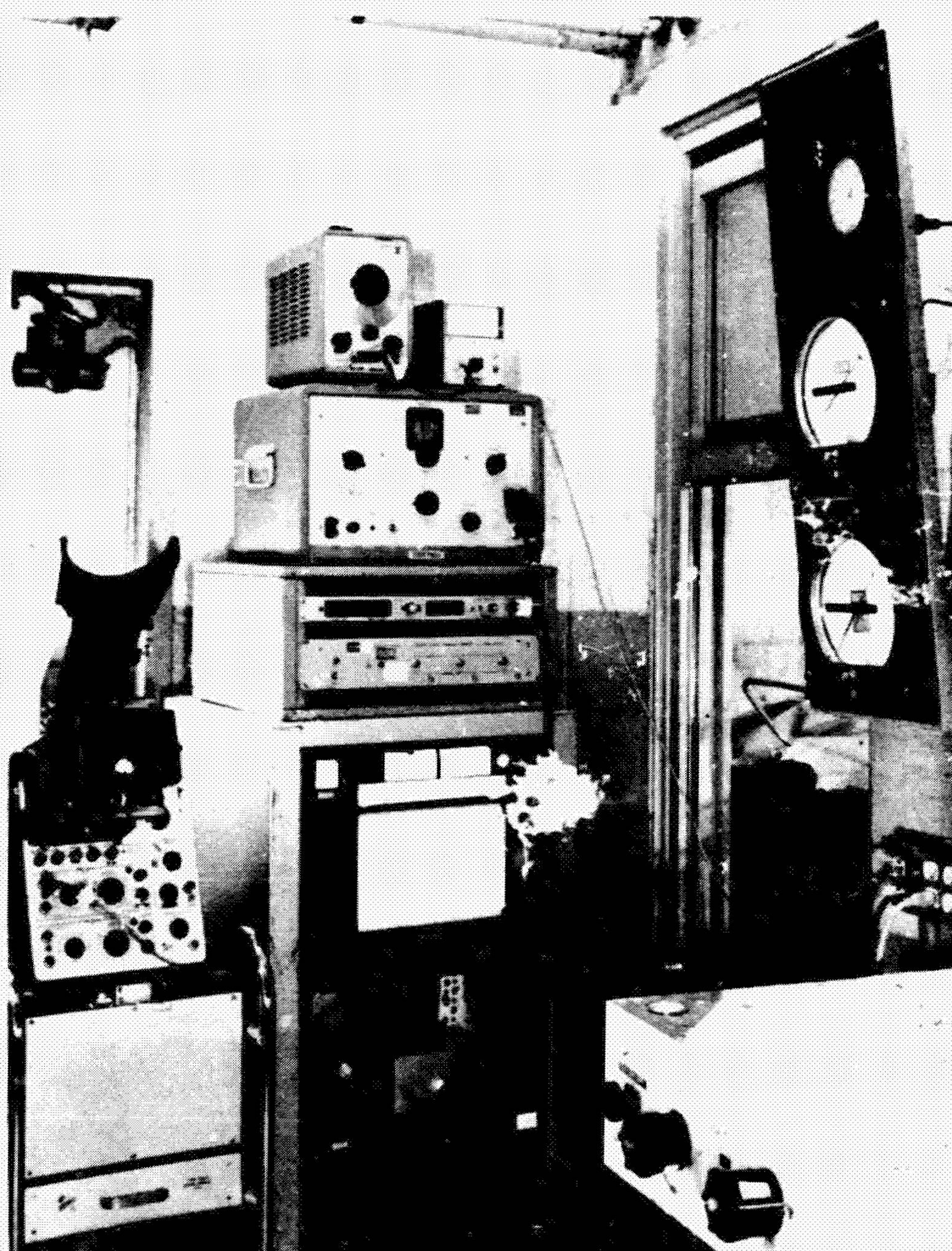


Figure 28 - Instrumentation Recorder and Control for Sealing Closure Tests

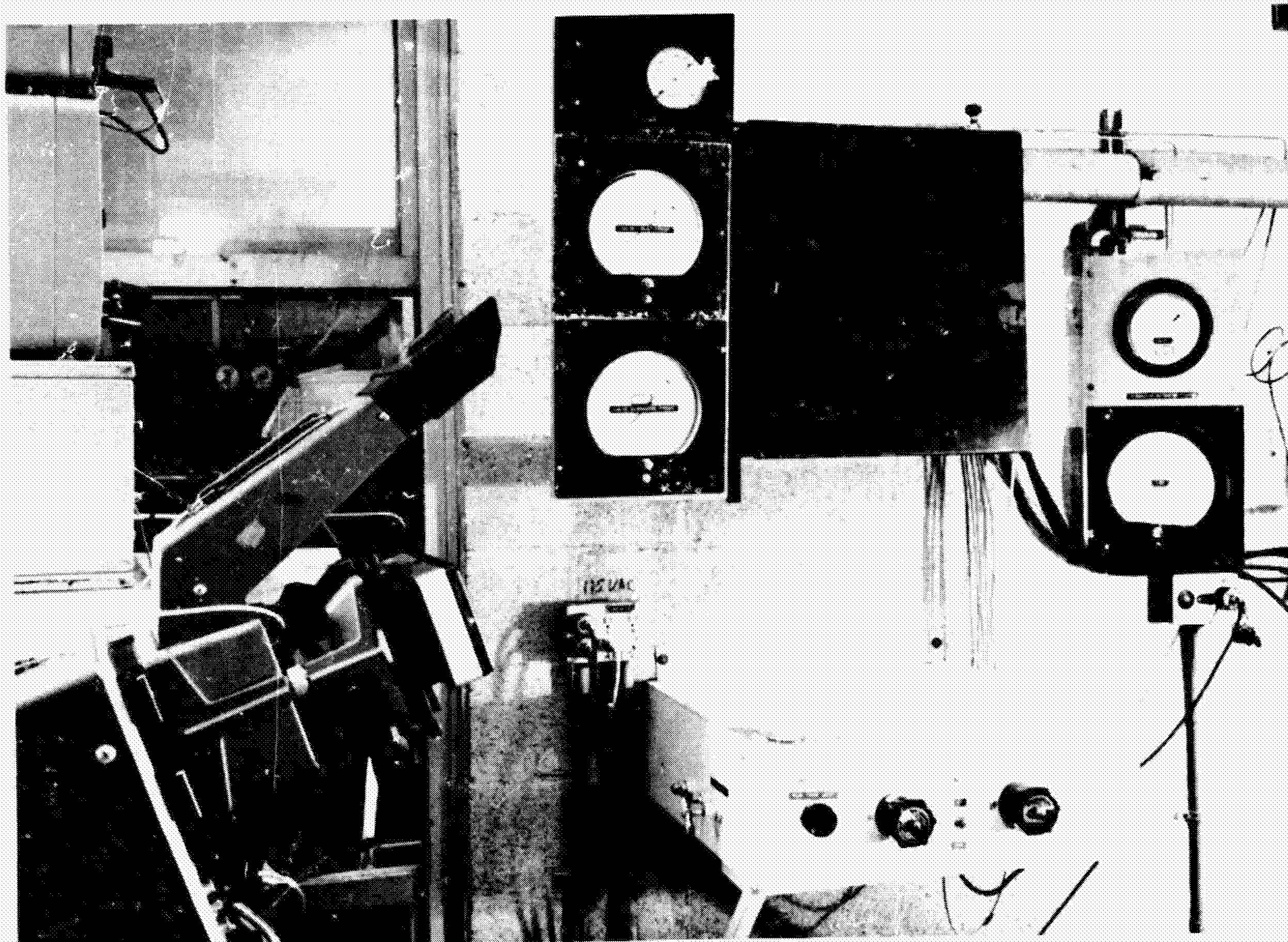


Figure 29 - Test Instrumentation

heat exchanger to tester transfer lines and the tester itself at a sealing closure temperature of +390°F (199°C). Several modifications were made to the hot conditioning system and these included the addition of insulation to the transfer lines, the addition of an external copper coil to the tester at the top of the tester, the addition of a hot gas bleed line into the outlet of the tester and finally the addition of a steam heat exchanger to pre-heat the gaseous nitrogen to approximately 200-300°F (93-149°C) before it entered the electric heater. During these attempts to increase the sealing closure operating temperature, several electric heater malfunctions also occurred, which resulted in a delay of the check-out testing of approximately 3 weeks. The final hot temperature conditioning system permitted attainment of the +390°F (199°C) high temperature in approximately 1 to 2 hours, and operated very satisfactorily during the evaluation of the sealing closures.

Check-out cycling tests of the rapid screening tester were performed by installing the P/N L4683 (Figure 8) sealing closure and by operating the tester at various hydraulic systempressures and throttle valve settings. This same valve sealing closure was also used for checkout of the leak test fixture. Initial cycling tests resulted in maximum achievalbe closing velo-ities of 8 inches (20.3 cm) per second. It was determined that the closing velocities were limited by the size of the hydraulic servo and consequently, a larger unit consisting of a servovalve and slave was installed to achieve the required 16 inches (40.6 cm) per second. Several thousand cycles, many of them at 10 cps (HZ), were accumulated during the checkout testing of the rapid screening tester. It was determined that the closing velocities were essentially independent of the stroke because the system was set up to limit the hydraulic flow to the actuator. Consequently, to minimize GN_2 consumption, it was decided to operate at relatively small tester strokes and all subsequent sealing closure testing was performed with strokes of approximately 0.020 to 0.040 inches (.0508-.1016 cm).

Sealing Closure Screening Test Results

During the screening test program, seven different sealing closure designs were evaluated in the rapid screening tester. In some cases, two units of the same design were tested. The evaluation program consisted essentially of cycling the sealing closures at 400 psia (276 N/cm^2) inlet pressure and at hot (850°F) (472°K), ambient, and cold (200°R) (111°K) conditions to verify the ability of the sealing closure to seal satisfactorily. Leakage tests at each condition included variations in helium pressure (20, 225, and 450 psia) (13.8, 155, 310 N/cm^2) and in static sealing closure interface loads. The goal was to accumulate 100,000 cycles with each sealing closure without exceeding the leakage requirements of 100 scc's per hour of helium at 450 psia (310 N/cm^2) inlet pressure.

Table VIII is a data summary of the pertinent results obtained with each closure design and the following paragraphs discuss some of the details.

Gold Plated Lip Seat - P/N L4312-1.

Testing of the gold plated lip seat (L4312) resulted in excellent results at hot, ambient, and cold temperature up to a total of 92,000 cycles. At that point, the lip seal broke along a groove which was caused during initial machining. Further investigation of the seal disclosed that the groove caused a stress concentration which in combination with the rather brittle characteristics of the material (Pyromet X-15) at cryogenic temperature resulted in the failure. The seal was subsequently redesigned slightly to reduce stress levels and to specify a better surface finish and Inconel 625 was substituted for the Pyromet X-15.

Fabrication of the new gold plated Inconel 625 lip seal (L4312-1) was completed and the seat was successfully tested under hot, ambient and cold temperature conditions through 200,000 cycles.

Flat Teflon Seat - P/N L4680

This sealing closure operated satisfactorily up to 5000 cycles at which time the sealing closure had to be removed from the assembly to permit a thermocouple repair. Subsequent reinstallation of the sealing closure revealed an initial leakage of 3000 scc's per hour, however, after conditioning the sealing closure to 390^{o}F (199^{o}C), the leakage was again zero. Subsequently, 3000 cycles were performed at hot conditions with leakage remaining well within specifications. These tests were followed by 12,000 cycles at ambient temperature for a total accumulated number of cycles of 19,000 with leakage measurements at sealing closure interface loads of 450 lbs (310 N/cm^2) and higher, well within specifications. The rapid screening tester was subsequently cooled to -260^{o}F (-162^{o}C) and leakage was again measured. Under these conditions, leakage in excess of 100 scc's per hour was encountered at sealing closure interface loads as high as 1600 lbs (1103 N/cm^2). An attempt to cycle the sealing closure for an additional 1000 cycles did not improve this condition. Consequently, further testing of this sealing closure was terminated.

Examination of the sealing closure disclosed no particular discrepancies. Based on these and other sealing closure tests, it is believed that the polyimide bumper which was used in parallel with the teflon seal in this design, did not deflect sufficiently to permit good contact between the teflon and poppet at the low temperature condition. It was recommended that a thinner polyimide bumper, and therefore one that would deflect more under the same load, be designed, fabricated, and evaluated during future testing. However, because of the promising results obtained with the other materials, no further work was done with teflon.

TABLE VIII - SEALING CLOSURE DATA SUMMARY

PART NO.	NAME	NO. CYCLES HOT	NO. CYCLES COLD	TOTAL NO. CYCLES	NO. LEAK CHECKS	AVG.* LEAK RATE	LEAK RATE* AT THE END OF CYCLING	COMMENTS
L4312-1	GOLD PLATED LIP SEAT	5,000	35,000	200,000	50	65	0	GOOD SEALING CHARACTERISTICS
L4680	FLAT TEFLON SEAT	3,000	2,000	20,000	36	10 EXCEPT COLD	EXCESSIVE	UNABLE TO MEET LEAKAGE COLD
L4681	SPHERICAL TEFLON SEAT	10,000	10,000	37,000	24	100 EXCEPT FAILURE	EXCESSIVE	TEFLON RETENTION INADEQUATE
L4682	FLAT TUNGSTEN CARBIDE SEAT	0	10,000	100,000	34	40	30	GOOD SEALING CHARACTERISTICS
L4683	TEFLON COATED LIP SEAT	11,000	10,000	62,000	34	80 EXCEPT FAILURE	EXCESSIVE	FAILED DUE TO OUT OF PRINT CONDITION
L4685	FLAT POLYIMIDE SEAT	20,000	15,000	100,000	30	6	0	EXCELLENT SEALING CHARACTERISTICS
L4686	SPHERICAL POLYIMIDE SEAT	10,000	10,000	100,000	29	0	0	EXCELLENT SEALING CHARACTERISTICS

*LEAKAGE IS MEASURED IN SCC/HR OF HELIUM AT 450 PSI (310 N/cm^2) INLET PRESSURE.

Spherical Teflon Seat - P/N L4681

The spherical teflon seat is identical in construction to the flat teflon seat, except that the poppet to seat interface is spherical rather than flat. A teflon land width of 0.045 inches (.1143cm) is employed in both designs. This sealing closure performed very satisfactorily at ambient, hot, and cold temperature, up to 37,000 cycles. At that point the test technician terminated the cycling tests due to the occurrence of unusual noise in the rapid screening tester. A leak check at that point indicated gross leakage and subsequent disassembly of the sealing closure disclosed that one of the clamps holding the spherical poppet to the poppet shaft had come off and another clamp had vibrated loose during the cycling tests and that the first of these clamps was lodged between the seat and the poppet. Examination of the seat also disclosed that the teflon was completely missing and it is believed that the teflon retaining clamp of the seat was inadequate, such that the teflon seal ring contracted during the cold test and pulled out of the retaining clamp. With the teflon missing, the spherical poppet made direct contact with the polyimide and was able to achieve an effective seal. However, since it was the purpose of this sealing closure to evaluate teflon, rather than the polyimide, the design in its present state was considered inadequate, and further testing of this sealing closure was discontinued. A sealing closure employing only polyimide was tested very successfully and is discussed in later paragraphs.

Flat Tungsten Carbide Seat - P/N L4682

The tungsten carbide seat leaked somewhat in excess of specification after initial installation and was, therefore, not cycled. Post test inspection disclosed the seat land to be out of flat by 3 light bands, apparently due to stresses induced by the Belleville springs which are used to set the sealing closure interface force.

This tungsten carbide seat was relapped and reinstalled in the test set-up. Initially the seat demonstrated excessive leakage, however during subsequent investigation, it was determined that the lugs which hold the poppet in position, could be tightened in such a manner as to cause distortion of the poppet face. This was corrected and subsequently the seat was successfully tested under hot, ambient and cold temperature conditions through 100,000 cycles.

Flat Teflon Lip Seat - P/N L4683

During the cold cycling tests with this design, the sealing closure leakage characteristics somewhat exceeded specification requirements with the maximum leakage reaching 230 scc's per hour after approximately 40,000 cycles. Leakage measurements at both ambient and +390°F (199°C) were at all times within specification requirements. Testing of this particular sealing closure was terminated after 62,000 cycles when leakage at hot conditions became excessive. Detailed examination of the sealing closure disclosed that during the final machining operation, the metallic bumper located immediately downstream of the teflon seal was machined down sufficiently such that the teflon coating thickness tapered to zero in one area of the seal. In this particular area, the extremely thin teflon coating had separated from the metal and the teflon was no longer effective as a seal. The fact that the teflon coating thickness tapered down to zero was an out-of-print condition, and it was for this reason that it was decided to manufacture another seat and to resubmit this sealing closure to further testing. The seat was recoated, however, the seat was very marginal because of repeated recoating in an effort to improve the quality of theTFE coating. Testing of this seat resulted in fracture of the metal lip which supports the teflon during the initial cycling in the rapid screening tester. Fabrication of a new teflon lip seal was initiated immediately.

Fabrication of the new teflon lip seal was completed and tests were initiated. Initial seat leakage was acceptable, however, after 1,000 cycles, leakage was excessive. Teardown inspection revealed that the teflon at the outer edge of the seal had separated in several areas from the base metal, apparently due to poor adhesion, folded inward over the sealing surface, causing a double thickness with resultant leakage. Additional effort with teflon coated lip seals was then discontinued.

Flat Polyimide Seat - P/N L 4685

Tests conducted with this particular seat assembly were very successful and leakage was zero at all temperatures up to 100,000 cycles with the exception of one leakage reading which was made at -260°F (-162°C) and after 20,000 cycles, and which showed a leakage at the specification limit of 100 scc's per hour. As with all other sealing closures, cycling was accomplished at a rate of 10 cps (HZ) and with a stroke of approximately .025 inches (.0635cm).

Spherical Polyimide Seat - P/N L4686

Test data obtained with this seat assembly was highly successful. Except for an initial leakage reading of 5 scc's per hour, all of the leakage tests disclosed zero leakage. Cycling again included both ambient, hot, and cold testing. A total of 100,000 cycles were accumulated.

From these results, summarized in Table VIII, it is evident that four of the sealing closures performed satisfactorily, whereas three were considered unsatisfactory on the basis of the specific specimens evaluated. It is also evident from this table that a wide variety of sealing closure materials were investigated. These ranged from such plastics as teflon and polyimide to metals and ceramics such as tungsten carbide. Since the discussion of the analytical leakage model data correlation requires a knowledge of certain sealing closure design parameters, Table IX is also presented. Table IX lists pertinent design data for each of the seven sealing closures tested by The Marquardt Company. As evident from this table, some of the sealing closures were evaluated at different impact load and static load levels whereas others were evaluated at only one impact load and one static load level. This aspect depended primarily upon the specific sealing closure in as much as these loads could be readily varied on some of the sealing closures due to inheren' .d varying provisions but would have required design modification on the . sealing closures.

The section entitled "Analytical Leakage Model" of this report presented a discussion of The Marquardt Company derived analytical relationships which expressed the helium gas leakage flow through sealing closure interfaces. Ad defined, the analytical leakage model consisted of two relationships. One examines the leakage through a static sealing closure interface, and the other attempts to predict the surface finish degradation of th ating sealing surfaces as a result of the wear incurred during the cycling of the sealing closure. Attempts have been made to correlate the experimental data with both parts of the analytical leakage model and the results of these correlations are described in the following sections.

Static Analytical Leakage Model Correlation

The section "Analytical Leakage Model" showed that volumetric leakage flow rate through a poppet seat interface is equal to the sum of the laminar flow and the molecular flow as follows:

$$Q_T = Q_L + Q_M$$

$$Q_T = \frac{1.46 \times 10^3 \; D_s (P_1^2 - P_2^2)}{\mu L T} H^3 + \frac{1.55 \times 10^9 \; D_s (P_1 - P_2)}{L} H^2$$

$$\text{Where } H = (h_1 + h_2) \left(1 - \frac{\sigma_s \psi}{\sigma_y}\right)$$

TABLE IX - SEALING CLOSURE INTERFACE DATA

Part No.	Name	Surface Finish Initial	Surface Finish Final	Land Width (Inch) (cm) Initial	Land Width (Inch) (cm) Final	Material Hardness at Ambient Temp.	Poppet Seat Parallelism (Inch) (cm)	Scrubbing Distance (Inch) (cm)	Impact Load (Lbs) (N)	Static Load (Lbs) (N)
L4312-1	Gold Plated Lip Seat	32	1	0.005 (.0127	0.018 (.0457)	75-160 Knoop	0.0015 (.0038)	0.00174 (.00442)	500 (2224)	300 & 450 (1334 & 2002)
L4680	Flat Teflon Seat	32	4	0.060 (.1524)	0.060 (.1524)	$R_R 115$	0.002 (.0051)	3×10^{-7} (7.62×10^{-7})	600-1600 (2669-7117)	300-1600 (1334-7117)
L4681	Spherical Teflon Seat	32	-	Sharp Edge	-	$R_R 115$	-	-	600 (2669)	450 (2002)
L4682	Tungsten Carbide Seat	0.5	0.5	0.015 (.0381)	0.015 (.0381)	1880 Knoop	0.0005	1×10^{-7} (2.54×10^{-7})	600 (2669)	300 & 450 (1334 & 2002)
L4683	Teflon Coated Lip Seat	32	2	0.010 (.0254)	0.030 (.0762)	$R_R 115$	2 1/2° (.0436 rad)	0.004 (.01016)	(500 psi) (345 N/cm^2)	(450 psi) (310 N/cm^2)
L4685-2	Flat Polyimide Seat	2	2	0.015 (.0381)	0.015 (.0381)	R_E 38	0.001 (.00254)	2×10^{-7} (5.08×10^{-7})	700 (3114)	300 & 600 (1334 & 2669)
L4686	Spherical Polyimide Seat	16	1	Sharp Edge	0.030 (.0762)	$R_E 38$	-	0.0013 (.0033)	600 (2669)	300 & 450 (1334 & 2002)

NOTE: All sealing diameters are 2 inches (5.08 cm). Poppet surface finish is always 1AA.

L	=	Land Width (In)	h	=	Peak to Valley Height (In.)
P	=	Pressure (PSIA)	H	=	Effective Leak Path Height (In.)
D_s	=	Seat Diameter (In.)	μ	=	Viscosity Lb - Min./In.^{2}
σ_s	=	Seating Stress (PSI)	σ_y	=	Yield Strength of Softer Material (PSI)
T	=	Gas Temperature	ψ	=	Coefficient

It is evident from these analytical relationships that the factor having the greatest influence on the volumetric leakage flow rate is the effective leak path height, since it occurs both as a cubed and squared function in the analytical leakage model. An understanding of this fact resulted in the design of sealing closures featuring very fine surface finishes as listed in Table [illegible]. Closer examination of the effective leak path height (H) shows that this factor is equal to the sum of the peak to valley heights of each of the mating sealing surfaces modified by a stress relationship. This stress modifier was derived to be the quantity $1 - \frac{\sigma_s \psi}{\sigma_y}$. Theoretical determination of the factor ψ is most complex since it depends upon a number of assumptions of the sealing surface configuration such as waviness and lay as well as other assumptions which relate whether or not individual surface peaks are simply compressed without a corresponding rise in the surface valley or whether there is indeed a corresponding rise in the surface valleys. Using metal-to-metal surface properties characteristics, the factor ψ was originally predicted to be 1.375.

It was the intent of the experimental program to generate data which would permit an updating of this factor ψ as well as the determination of ψ for a variety of sealing closure materials. To permit the determination of the factor ψ from experimental data, the sealing closure interface load data presented in Table IX was converted into seating stress values through the use of the curves presented in Figure 30. These stress values as well as the seat diameter, gas temperature, land width, gas viscosity, and actual measured leakage rate were inserted into the analytical leakage expression and this relationship was solved for the effective leak path height H. To facilitate the solution of this analytical expression, a series of curves showing the effective leak path height (H) vs. the leakage rate were prepared as shown in Figure 31. Figure 31 is a sample curve applicable to the flat tungsten carbide seat (L4682) at an inlet pressure of 450 psia (310 N/cm^2). Similar curves were prepared at inlet pressures of 225 and 20 psia (155 and 1[illegible] N/cm^2) for this as well as all of the six other sealing closure configurations. Figure 31 was then entered at a specific experimentally determined leakage rate and the corresponding effective leak path height H was read off. This effective leak path height was then used to enter Figure 32 and to determine the corresponding stress modifier $(1 - \frac{\sigma_s \psi}{\sigma_y})$ for any one particular sealing closure. Once the stress modifier was obtained, the coefficient ψ could be determined.

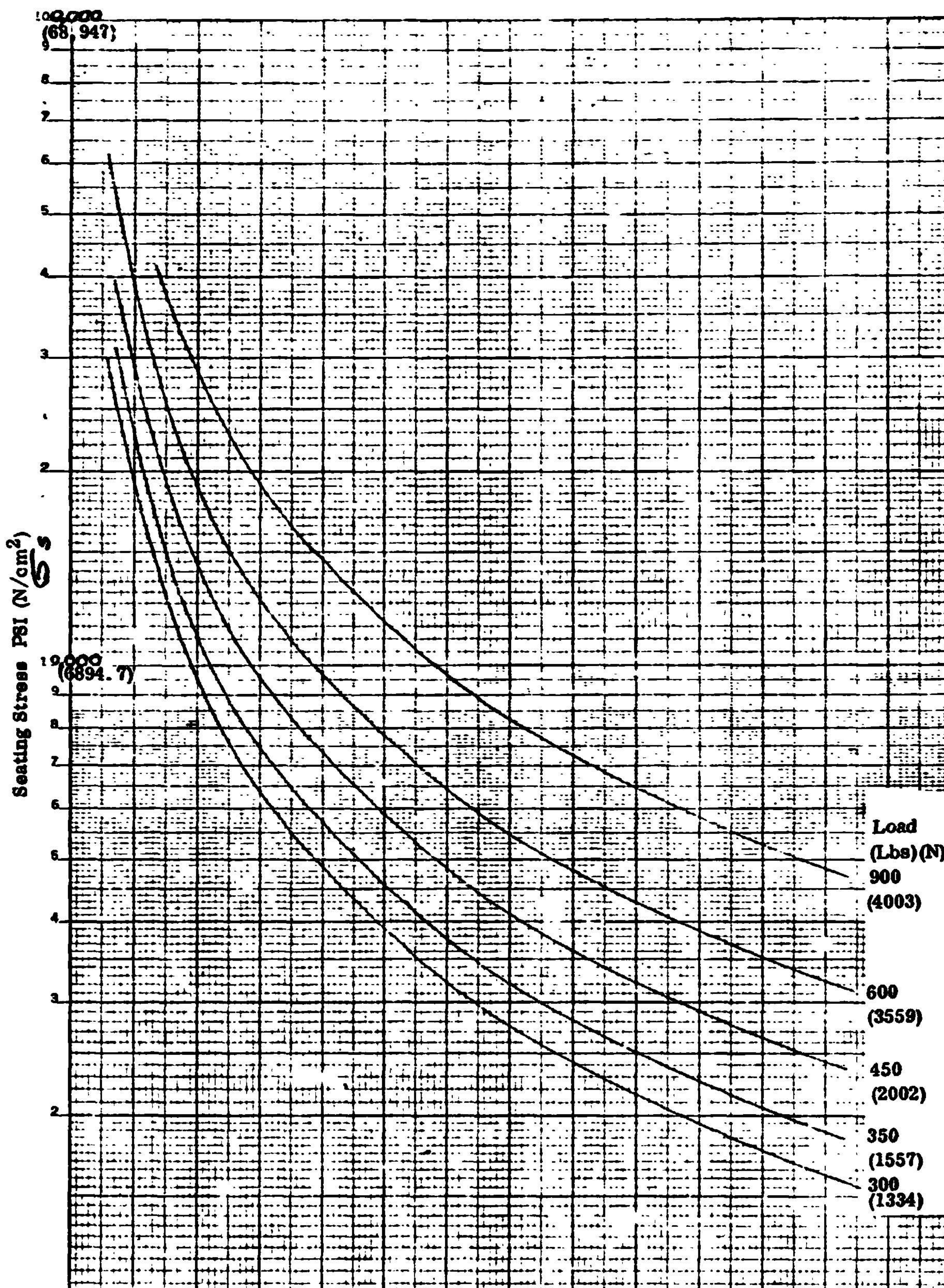


Figure 30 - Seating Stress Vs Land Width at Various Loads

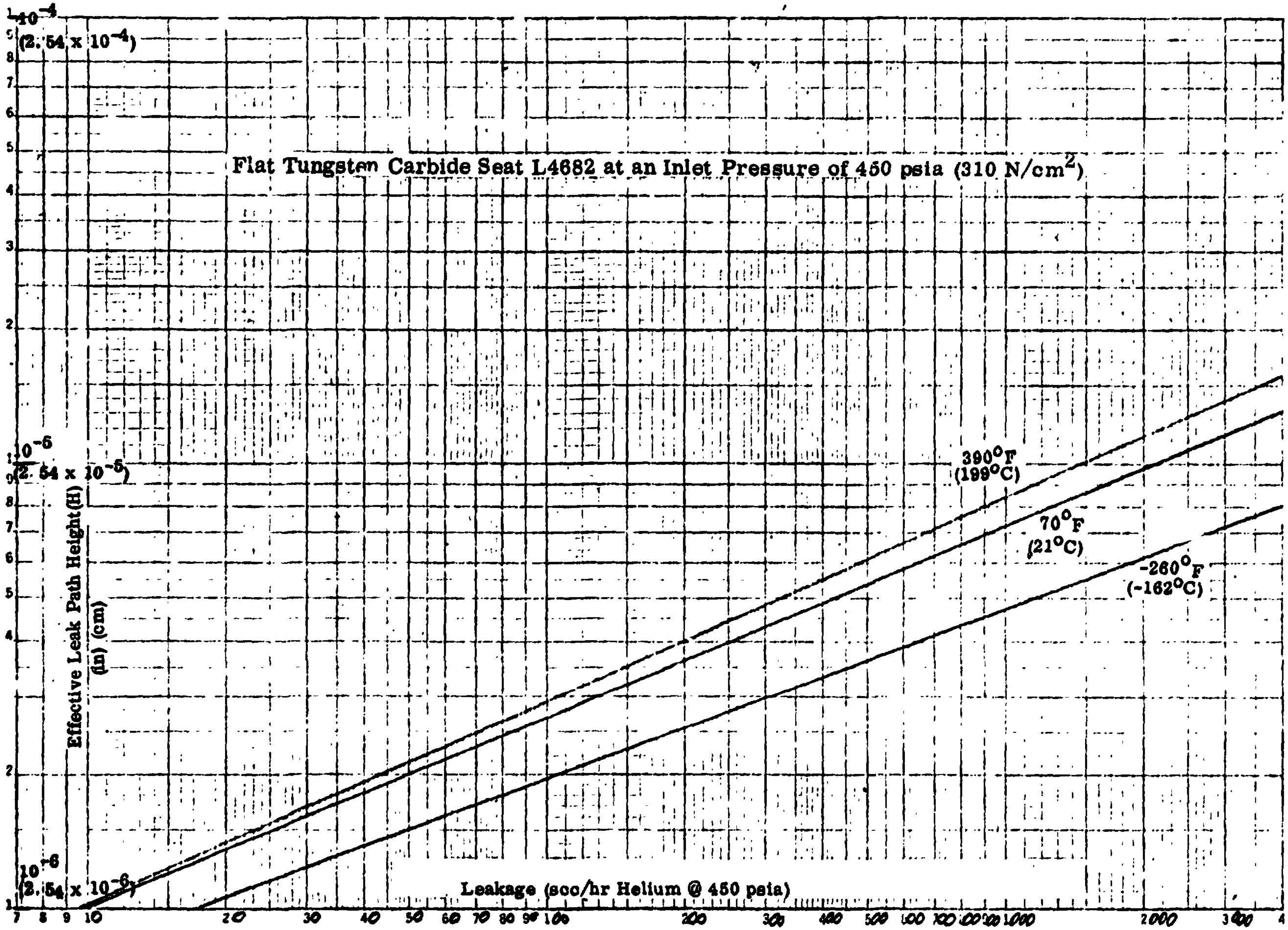


Figure 31 - Effective Leak Path Height (H) Vs Leakage

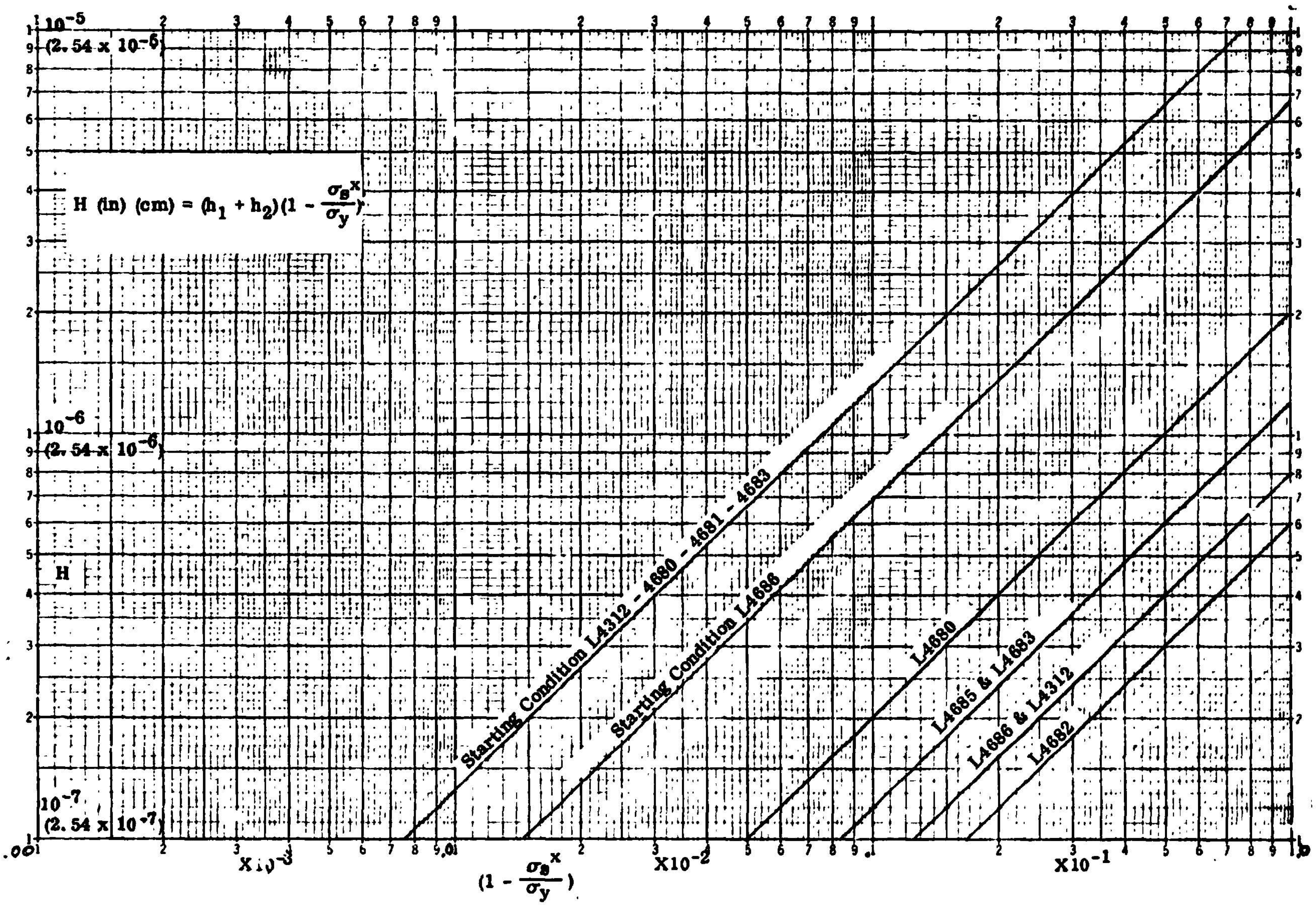


Figure 32 - Effective Leak Path Height Vs Stress Function

One minor problem with the model is the fact that with finite surface finishes it can only match experimentally obtained "zero" leakage by letting the stress term go to zero, that is $\sigma_s{}^{\psi}/\sigma_y = 1$. When that occurs, the effect of surface finish is eliminated, which of course is contrary to reality. To overcome this, an arbitrary, small leakage must be assumed to evaluate the H factor. In the evaluation conducted, 1 scc/hr was equated to "zero" leakage, which worked out quite well as indicated by subsequent correlation checks. At these low leakage rates the model becomes quite sensitive to the stress function exponent; however, this does not hold true when calculations are made for conditions resulting in higher leakage values, where the $\sigma_s{}^{\psi}/\sigma_y$ term becomes significantly smaller than 1.

The stress coefficient ψ and leakage parameter H which account for surface finish and seat stress effects are subject to significant variations for materials whose stress strain characteristics are a distinct function of temperature (e.g., polyimide and teflon). As shown in Table X, once the correct material yield strengths (i.e., the yield stess of polyimide used is 6200 psi (4275 N/cm^2) at ambient and 8000 psi (5516 N/cm^2) at -260°F (-162°C) is utilized the coefficient ψ can be determined to correlate quite well with the yield strength of the softer sealing closure material as shown in Figure 33. When using this experimentally determined coefficient ψ, reasonable prediction of leakage, certainly in the right order of magnitude, and in many cases within plus or minus 50 percent is possible. A typical example of the actual leakage data obtained with the tungsten carbide seat (P/N L4682) at a seat load of 450 pounds (2002 N) and 450 psia (310 N/cm^2) helium inlet pressure is shown in Fi ure 34. This figure also shows the ambient and cold leakage predictions based on the use of the experimentally determined exponent ψ = 1.55 from Figure 33.

Because of the great variety of influences and parameters affecting the leakage behavior of different seats, the available test data is statistically insufficient to fully evaluate and update the static analytical leakage model. Within the confines of the Space Shuttle Auxiliary Propellant Valves Program which did not allow for independent variation of such sealing closure characteristics as land width, surface finish, and seat diameter, it is nevertheless significant that a definite correlation between the stress exponent and the material yield strength was determined.

Dynamic Leakage Model

The analytical model which also was discussed under "Analytical Leakage Model", was prepared by The Marquardt Company to predict sealing closure cycle life consists of the determination of the surface finish degradation due to wear. This equation has the form:

TABLE X - LEAKAGE MODEL EVALUATION

SEAT	ORIG. MODEL VALUES ψ	ORIG. MODEL VALUES H in. (cm)	EXPERIMENTAL VALUES ψ	EXPERIMENTAL VALUES H in (cm) AT CONDITIONS SPECIFIED	CORRELATION & GENERAL COMMENTS
• Gold Lip L-4312	1.375	2.75×10^{-6} (6.985×10^{-6}) @F_s=350# (1557N)	1	$2.0 - 4.6 \times 10^{-6}$ $(5.08 - 11.68 \times 10^{-6})$ @ F_s = 350# (1557 N) 6.0×10^{-7} (15.24×10^{-7}) @ F_s = 450# (2002 N)	Good correlation with the two seat loads (2.27×10^{-6} calculated for F_s = 350# (1557 N) from 450# H (2002 N))
• Flat TFE Seat L-4380	1.375	4.85×10^{-6} (12.319×10^{-6})	1.042	5.6×10^{-6} (14.22×10^{-6}) @ 900 psi seat press. (620 N/cm²); 7.4×10^{-6} (18.80×10^{-6}) @ 450 psi seat press. (310 N/cm²) — after 1,000 cycles, ambient 7.45×10^{-7} (18.92×10^{-7}) @ 900 psi seat press (620 N/cm²) after 20,000 cycles, ambient 6.4×10^{-6} (16.26×10^{-6}) @ 900 psi seat press (620 N/cm²) after 20,000 cycles, ambient	Reasonably good correlation on surface finish (start, final) seat pressure and temperature
• Flat Tungsten Carbide Seat L-4682	1.375	275×10^{-6} (6.985×10^{-6})	1.48 to 1.54	$3.0 - 4.0 \times 10^{-6}$ $(7.62 - 10.16 \times 10^{-6})$, @ 0 cycles, F_s=450# (2002N) $3. \times 10^{-6}$ (7.62×10^{-6}) @ 30-70K cycles, F_s=300# (1334 N), Δp = 20 psi (13.8 N/cm²) $1.5 - 2.0 \times 10^{-6}$ $(3.81 - 5.08 \times 10^{-6})$ @ 30-70K cycles, F_s=300# (1334 N), Δp=450 psi (310 N/cm²) $1.1 - 1.4 \times 10^{-6}$ $(2.79 - 3.55 \times 10^{-6}$ @ 30-70K cycles, F_s=300# (1334 N), Δp=225 psi (155 N/cm²) 2.0×10^{-6} (5.08×10^{-6}) @ 100K cycles, F_s=300# (1334 N), Δp = 450 & 225 psi (310 & 155 N/cm²) 5.8×10^{-6} (14.73×10^{-6}) @ 100K cycles, F_s=300# (1334N), Δp = 20 psi (13.8 N/cm²)	Model correlates well with ψ = 1.54
• Flat TFE Lip L-4683	1.375	4.85×10^{-6} (12.319×10^{-3})	0.77 to 0.546	$1.5 - 6.2 \times 10^{-6}$ $(3.81 - 15.75 \times 10^{-6})$	Good correlation with temperature and seat load (F_s=450# (2002 N), -170°F (-112°C) predicted leakage of 300-400 scc/hr fits some of data well)
• Flat Polyimide L-4685	1.375	2.2×10^{-6} (5.588×10^{-6})	.99 to 1.08	4.7×10^{-7} (11.94×10^{-7}) Ambient & hot $1.5 - 3.7 \times 10^{-6}$ $(3.81 - 9.40 \times 10^{-6})$ Cold	Hot/cold correlation on Q_T quite good with exp. ψ
• Spherical Polyimide L-4686	1.375	2.2×10^{-6} (5.588×10^{-6})	1.07	6.3×10^{-7} (16.0×10^{-7}) Ambient & hot 2.3×10^{-6} (5.84×10^{-6}) Cold	Few "suspect" cold data points of high leakage do not fit model

NOTE:
1. "Zero" leakage → assumed as 1 scc/hr
2. Data from the spherical teflon seat (L4681) was not analyzed since it was not known whether seating was accomplished on the teflon or the polyimide
3. Seating of the flat teflon seat (L4683) was probably on the polyimide rather than on the teflon.

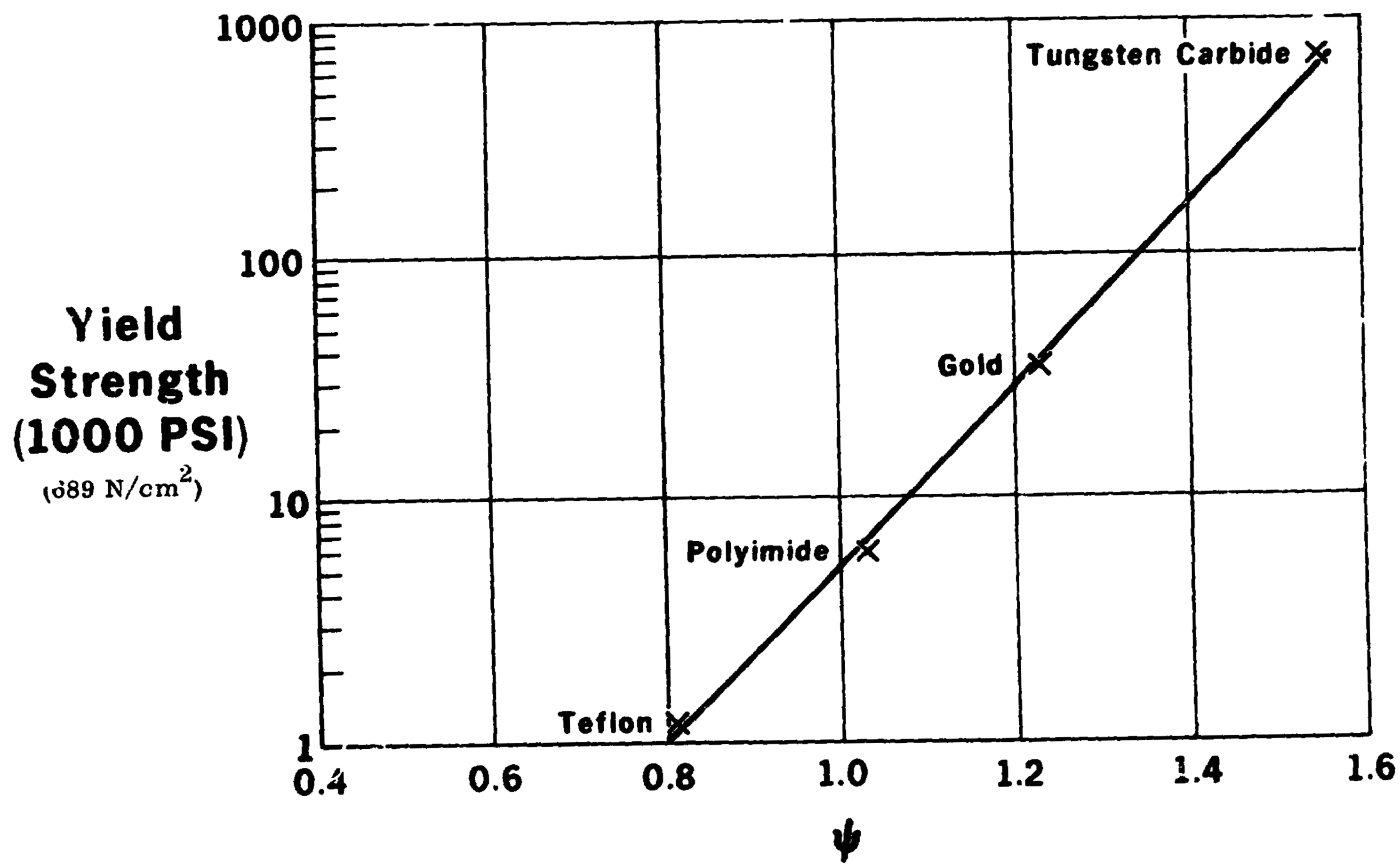


Figure 33 - Yield Strength Vs Exponent ψ

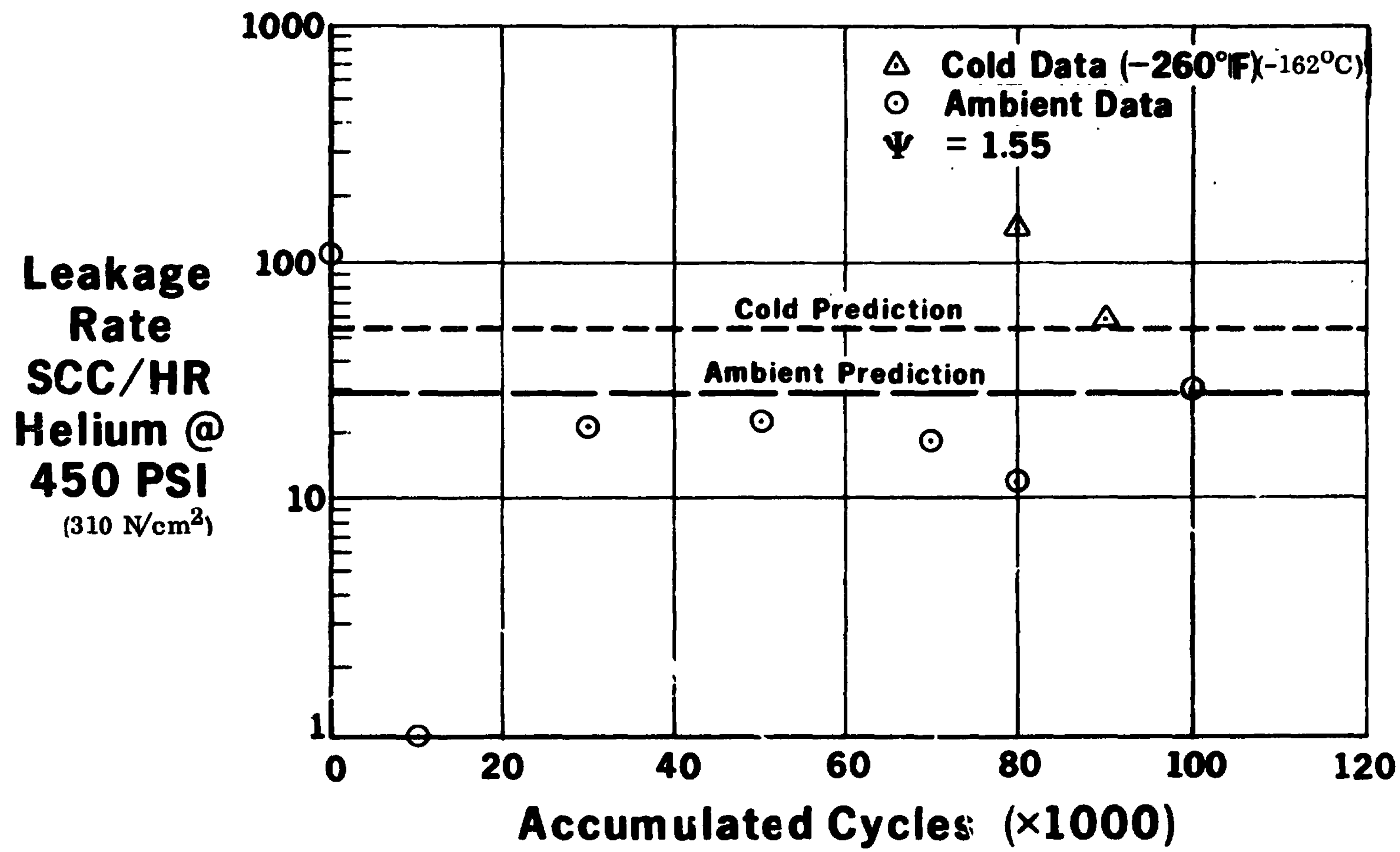


Figure 34 - Predicted to Actual Leakage Correlation - Tungsten Carbide Seat P/N L4682, 450 Lbs. (2002 N), 450 PSI (310 N/cm²) Helium

$$y_N = y^o + \frac{k_{AD}}{11,900} N \; s\sigma_I \left[\frac{1.4}{P_A} - 17.6 \times 10^4 \frac{y^o}{G_{AB}} \right]$$

where:

y_N	=	Surface finish after N cycles, in.
y^o	=	Original surface finish, in.
k_{AD}	=	Wear coefficient
N	=	Number of cycles
s	=	Lateral Sliding component between poppet and seat (scrubbing distance). in.
σ_I	=	Poppet/seat impact stress, psi
P_A	=	Material hardness (softer surface), kg/mm^2
G_{AB}	=	Energy of adhesion, ergs/cm^2

The surface finish, y^o, in this equation is related to the peak to valley height, h, of the static analytical leakage model by the relationship $y^o = 1/3h$. The expression for y_N was originally used by Marquardt in determining criteria for the sealing closure designs for this program. Thus, this analytical model made it apparent that minimum surface finish degradation and, therefore, high cycle life required sealing closure designs which featured minimum impact loads and minimum scrubbing distances. Typical sealing closure interface characteristics that were achieved are listed in Table IX.

The dynamic leakage model also shows that surface finish degradation is directly related to the number of cycles. Unfortunately, none of the configurations and materials tested followed the predicted gradual deterioration of the surface finish with cycling. On the contrary, most showed a reverse trend, at lease in the 0 - 60,000 cycle range, which, of course, is due to the improvement in surface finish of the softer material with cycling as shown by the surface finish measurements at the start and finish of the tests (Table IX). A typical leakage/cycle history is shown in Figure 35. Test results indicate that a more precise model would have to provide for a wear pattern giving a wear-in period, followed by a stable period, and finally followed by a wear-out period. Controlled tests would be necessary with many intermediate surface finish measurements to develop the characteristics associated with

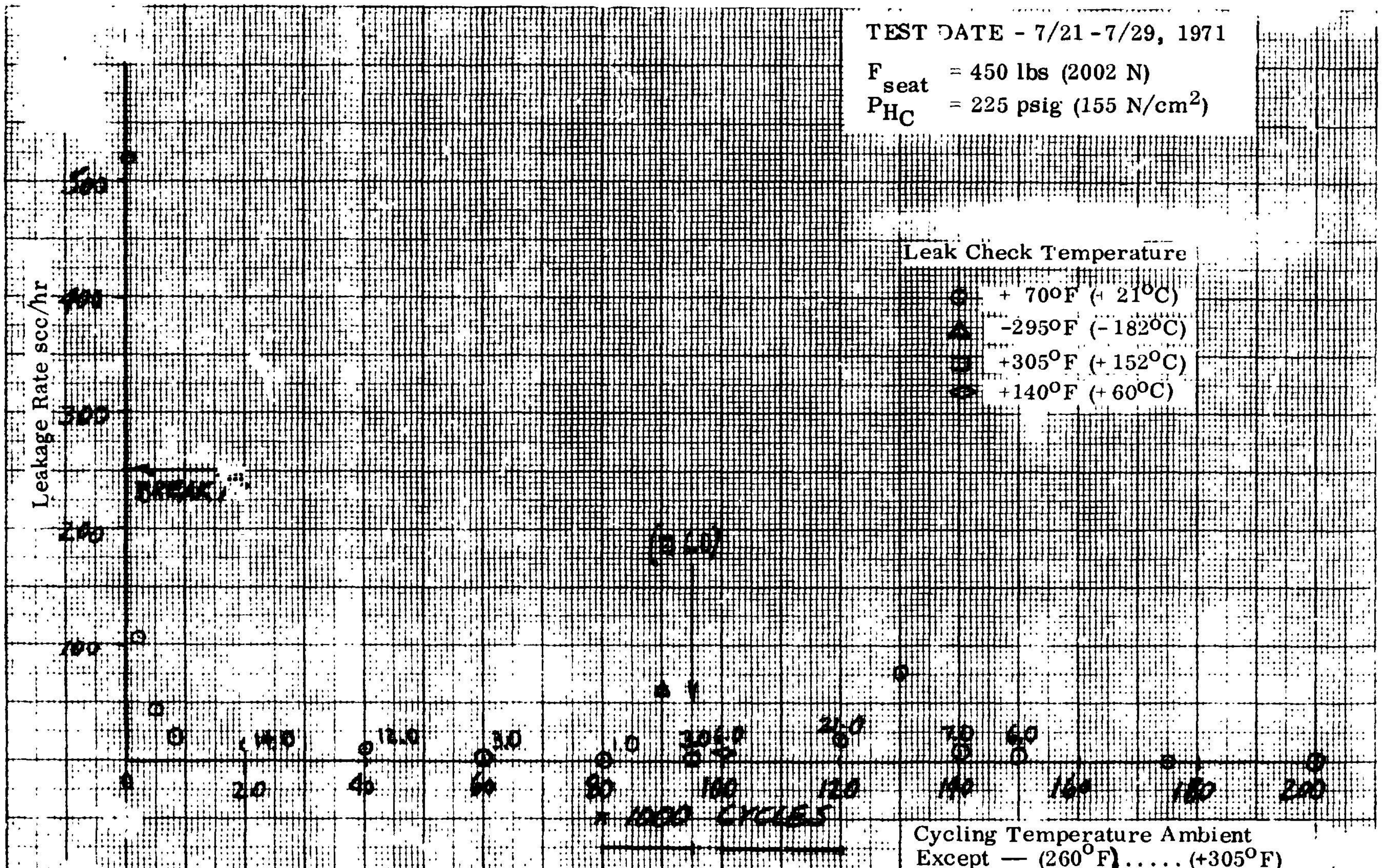


Figure 35 - Leakage Vs Accumulated Cycles,
Gold Plated Lip Seat, L4312, S/N 002

a given configuration and material combination. It also is evident that in many cases testing should be done far beyond the 100,000 cycles covered during these experiments, particularly since the original design goal was a cycle life of one million.

A comparison of the scrubbing distances that were predicted as being maximum allowables for 100,000 cycles life with the scrubbing distances actually experienced on three of the softer materials (gold plating, teflon coating, and spherical polyimide) indicates that these materials showed significantly less wear than was predicted by the analytical expression. Thus, the wear model appears to be definitely conservative for soft seat materials. The modeling of wear in typical poppet/seat sealing closures is a very complex one. It was not the intent of the Space Shuttle Auxiliary Propellant Valves Program to perform an in-depth investigation of this area, but rather the purpose of the preparation of a wear model was to point out those sealing closure characteristics that enhance cycle life and to thereby conservatively size sealing closure interfaces which can exceed the required cycle life. This objective was successfully accomplished since four different sealing closures were cycled over 100,000 times each without exceeding the required leakage specifications.

VALVE SUBCOMPONENT ANALYSIS AND CONCEPTUAL DESIGN

The purpose of this task was to perform a thorough conceptual design and analytical trade-off study of various types of shut-off devices, supporting parts, actuators, and linkages to select the best valve concept for application as the propellant control valves for gaseous hydrogen/gaseous oxygen rocket engines to be used on the Space Shuttle Auxiliary Propulsion System.

Conceptual Design and Analysis of Actuators, Linkages and Supporting Parts

In support of this effort, use was made of Marquardt's digital and analog computer programs for the study of dynamic valve behavior. The valve sizing computer program is organized such that each valve design is divided into two elements; the shut-off device component, which is the fluid control element and the actuator, which provides necessary force/stroke inputs to the moving components of the shut-off device. The interface between the shut-off device and the actuator can be characterized by force, stroke and time. Shut-off device performance is a function of the fluid control properties desired and actuator performance is established by available power source criteria. A total of valve performance predictions can be made when the shut-off device force-stroke-time input requirements equal the force-stroke-time output of the actuator. Both shut-off devices and actuators can be characterized by the nature of their motion, namely linear or rotary. This characterization is illustrated in Table XI. Having thus reduced the valves to basic elements, the analytical procedure was employed to establish compatible shut-off device/actuator interfaces for the various combinations of the elements.

In addition to the analytical computer studies, the designs of actuators, linkages and supporting parts involved in the valve design layouts made under Contract NAS 9-10886 were examined. The conceptual designs of Figures 36, 37 and 38 are representative of the initial design layout effort.

The double acting solenoid operated valve of Figure 36 was initially sized for a flow factor (C_dA_t) of 0.45 in^2 (2.90 cm^2) and an inlet pressure of 400 psia (276 N/cm^2). This concept incorporates several unique features. The double acting configuration (both seat and poppet move) wherein the sum of the motion of the seat and the poppet results in the appropriate flow area across the valve seat allows the use of flexures to provide friction free absolute guidance of the moving elements. The type of flexure anticipated has been developed and used in Marquardt's valve designs but is limited in stroke capability. The double action effectively doubles the range of applicability of flexure guides. Dividing the moving mass into two separate elements, magnetically linked, also proves to be advantageous considering

TABLE XI - VALVE ELEMENT - MOTION CHARACTERIZATION

MOTION TYPE	LINEAR	ROTARY
Shut Off Device	Poppet Plug Gate Sliding Spool Blade Diaphragm or Boot	Ball Butterfly Swing Flapper
Actuator	Cylinder Solenoid Piezoelectric Magnetostrictive Thermal Expansion Linear Motor	Torque Motor Electrical Motor Hydraulic or Pneumatic Motor Inertia Wheel

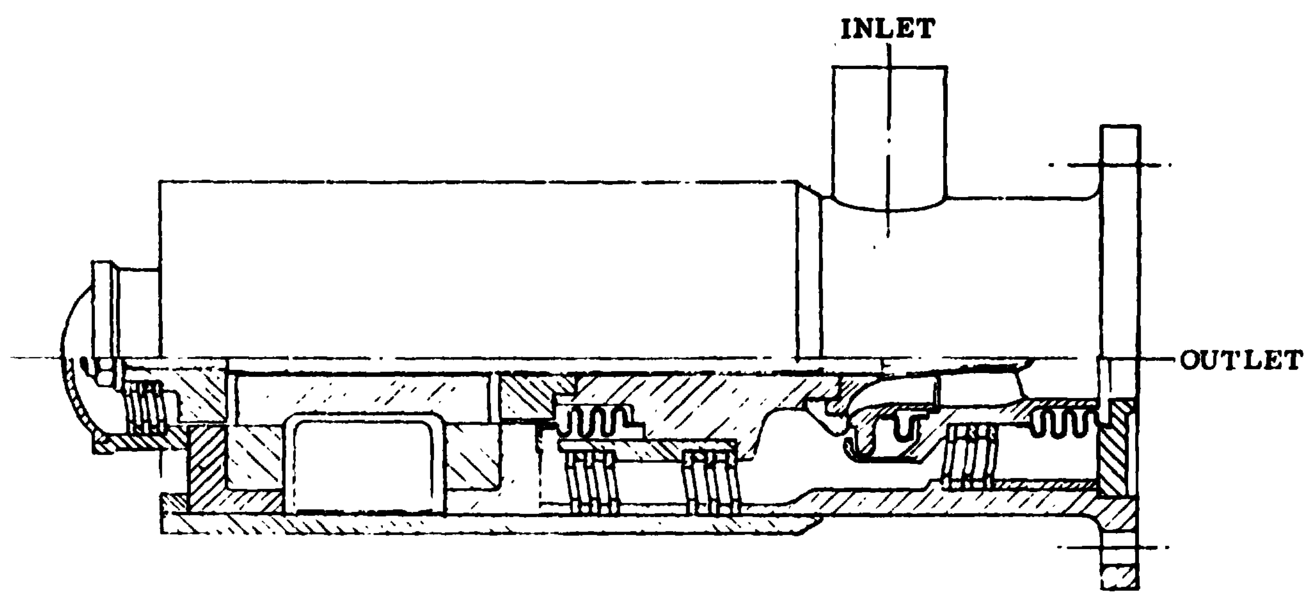


Figure 36 - Double Acting Poppet Valve, Hard Seat, High Press. Solenoid Operated

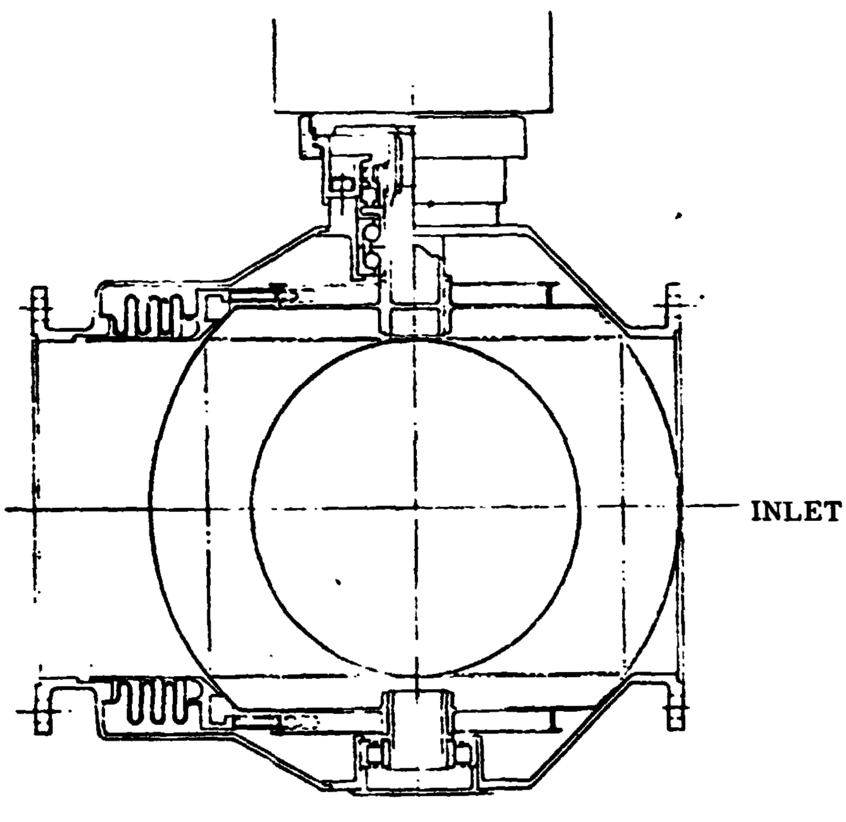


Figure 37 - Ball Valve, Soft Seat

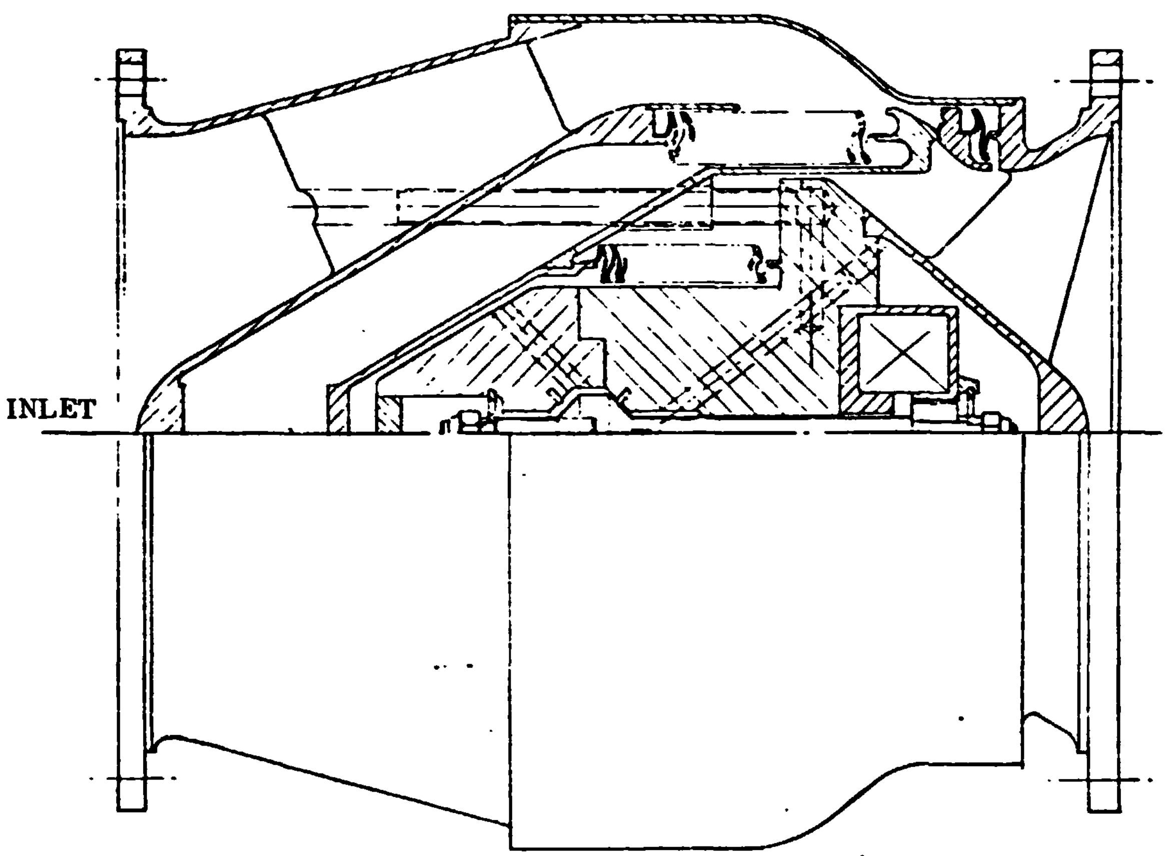


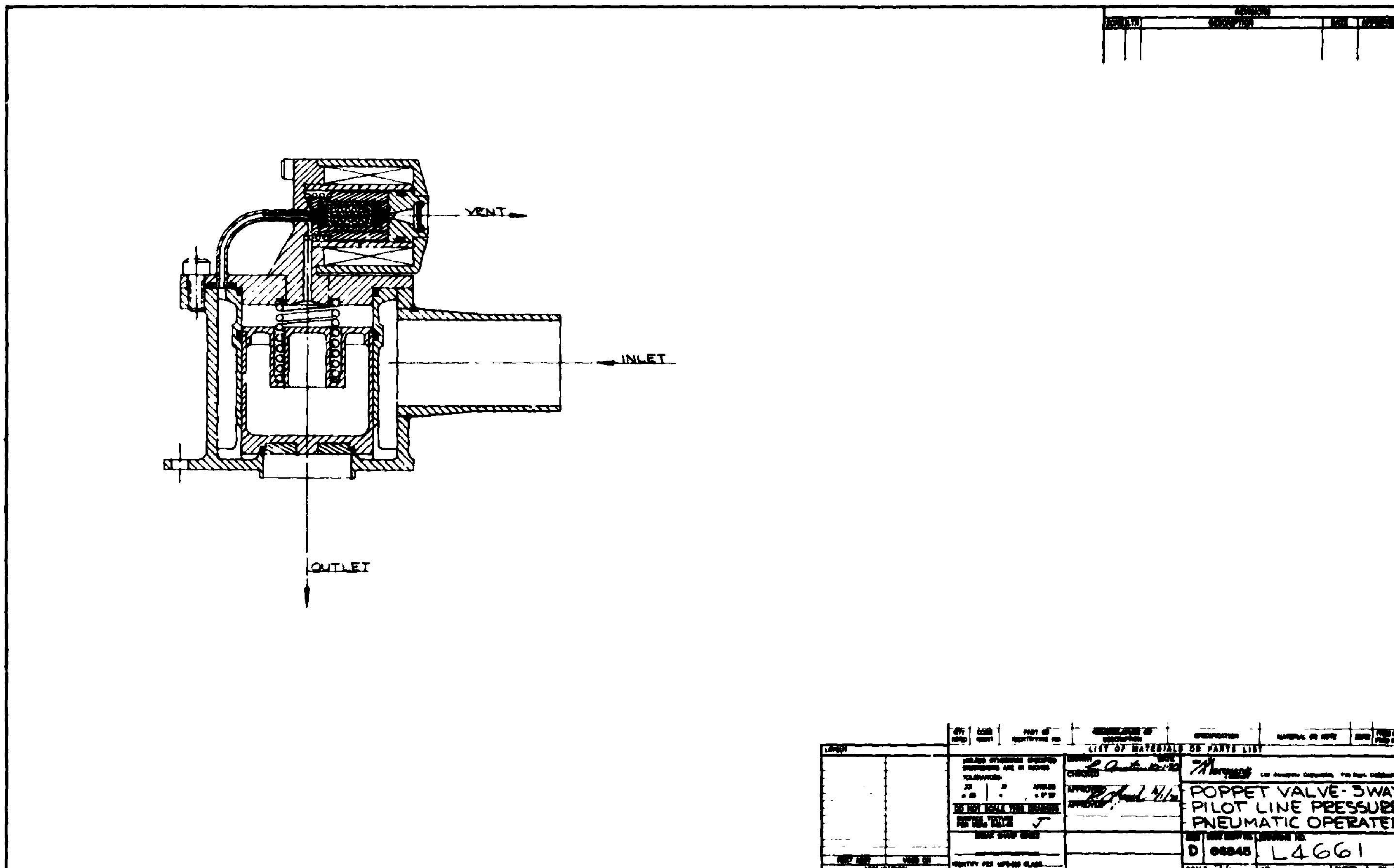
Figure 38 - Coaxial Poppet Valve, Pressure Actuated,
Low Pressure, Hard Seat

the force stroke characteristic of the solenoid. A reduction in solenoid force requirement is achieved since each moving mass moves only through 1/2 of the total stroke (1/2 the acceleration). The reduced stroke of each element also favors metal bellows as the method of accomplishing dynamic sealing at the balance area and required seat preloads. Though the analysis program ultimately indicated that a solenoid actuator for this application was not feasible, the configuration provided a basis for developing a relationship to predict shut-off device moving mass and documented several unique design concepts applicable to other configurations.

Figure 37 defines a rotary motion shut-off (ball) and torque motor actuator configuration. Original design criteria was for a flow factor (C_dA_t) of 12 in.^{2} (77.42 cm^2) and an inlet pressure of 30 psia (20.7 N/cm^2). Of interest is the cam actuation to retract the seal on opening, thereby reducing the required torque to effect opening and minimizing scrubbing of the seal. The seal is bellows mounted such that the seal is pressure energized against the ball, in the closed position, in addition to the preload resulting from initial compression of the bellows. "Wetted" ball bearing mounting of the rotating element was selected and a spring loaded shaft seal was incorporated. Performance analysis ultimately invalidated a torque motor actuator for this particular application. The concept documents the basic design of a rotary motion shut-off device assumed in the analysis program and substantiates envelope and weight estimates.

The coaxial, pressure operated valve of Figure 38 offers perhaps the highest density packaging concept. For the design criteria of 17 in.^{2} (109.7 cm^2) flow factor (C_dA_t) and 30 psia (20.7 N/cm^2) inlet pressure, the configuration incorporates a balanced poppet with metal bellows dynamic seal. The actuator section operates off line pressure and also incorporates a metal bellows dynamic seal. Pressure control of the actuator is accomplished by a 3-way solenoid operated valve. The pneumatic cylinder actuator is a single acting cylinder in that it is pressurized to effect opening, while closing forces are exerted by the inherent spring characteristic of the metal bellows when the cylinder is vented.

As the analysis program developed, more specific design layouts were made such as those shown by Figures 39 through 45. The line pressure pneumatic operated poppet valve of Figure 39 is a characteristic vented-pilot operated valve configuration. With the valve in the closed position, as shown, the seat is preloaded closed by the coil spring and inlet pressure acts to hold the poppet closed. Upon opening command (electrical signal) the solenoid causes the pilot valve poppet to move to a position where the bleed tube from the inlet manifold to the cavity within the main valve poppet is closed and the main poppet cavity is vented through the pilot valve. As cavity pressure decays, the differential pressure across the poppet creates opening forces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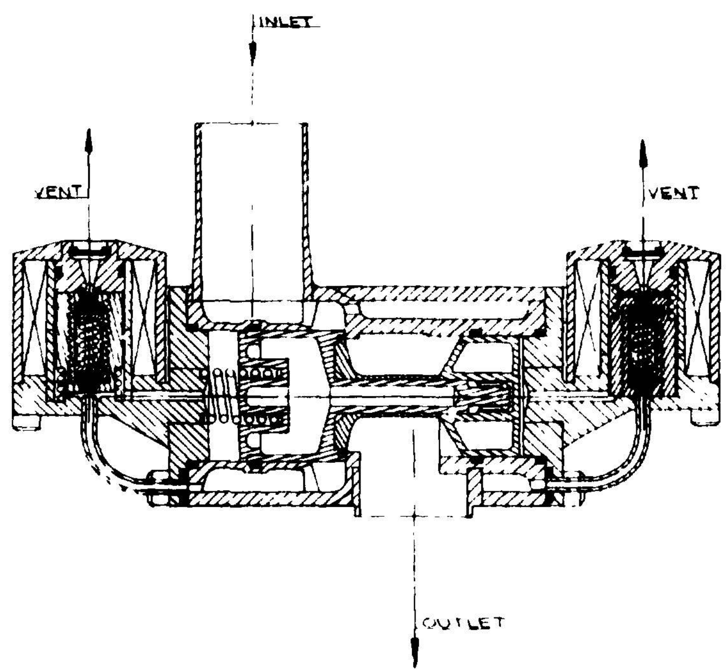
LIST OF MATERIALS OR PARTS LIST
APPLICATION
POPPET VALVE-3WAY-
PILOT LINE PRESSURE-
PNEUMATIC OPERATED
D
L4661
SCALE 2/1

Figure 39



REVISIONS

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BREAK SHARP EDGES

IDENTIFY PER MPS-800 CLASS

DRAWN DATE 10-8-70
CHECKED
APPROVED 10/12/70
APPROVED

Marquardt

POPPET VALVE-DUAL PILOTED LINE PRESSURE PNEUMATIC OPERATED

SIZE D | CODE IDENT NO 86845 | DRAWING NO L4662

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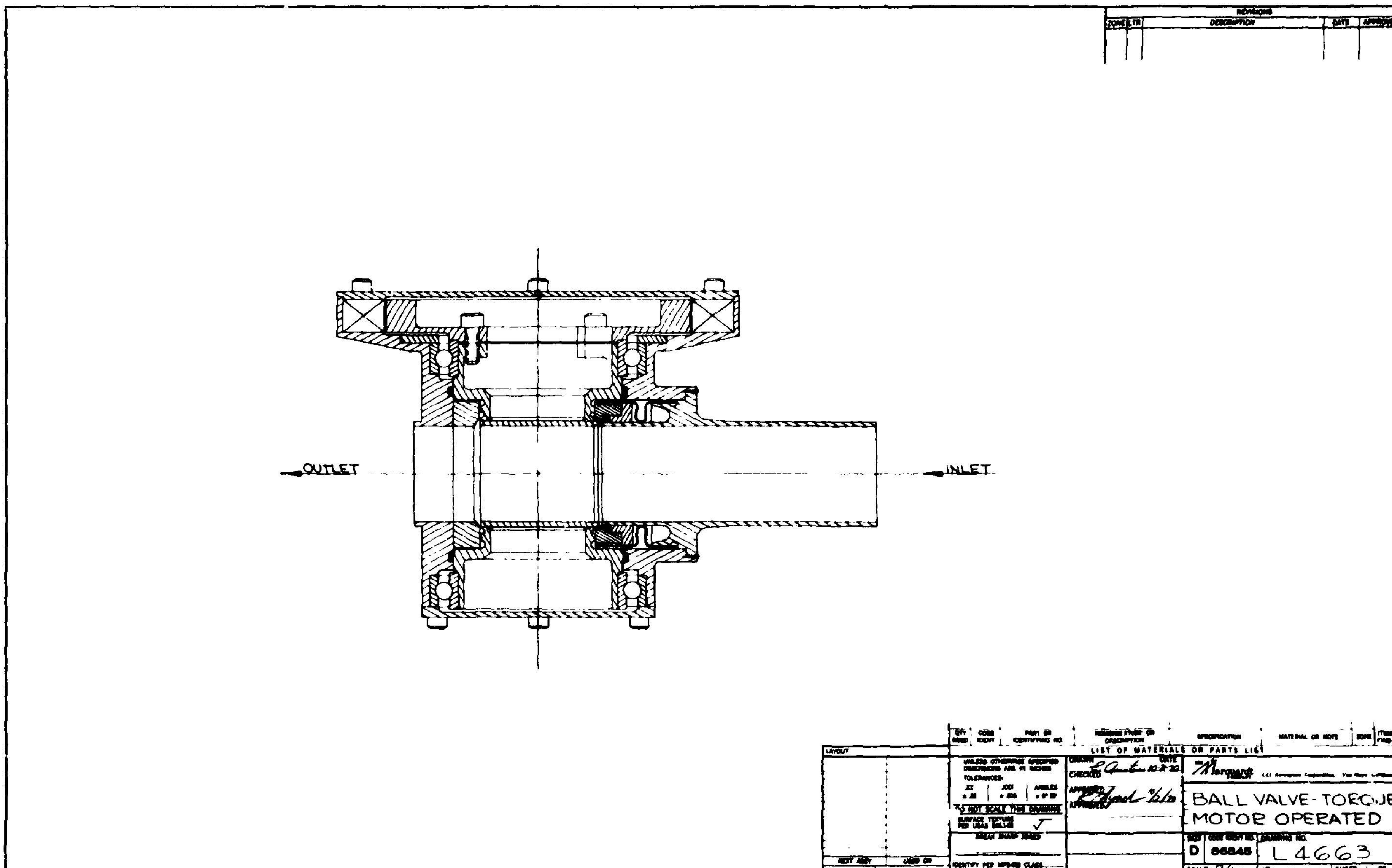

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Marquardt
BALL VALVE-TORQUE
MOTOR OPERATED
SIZE
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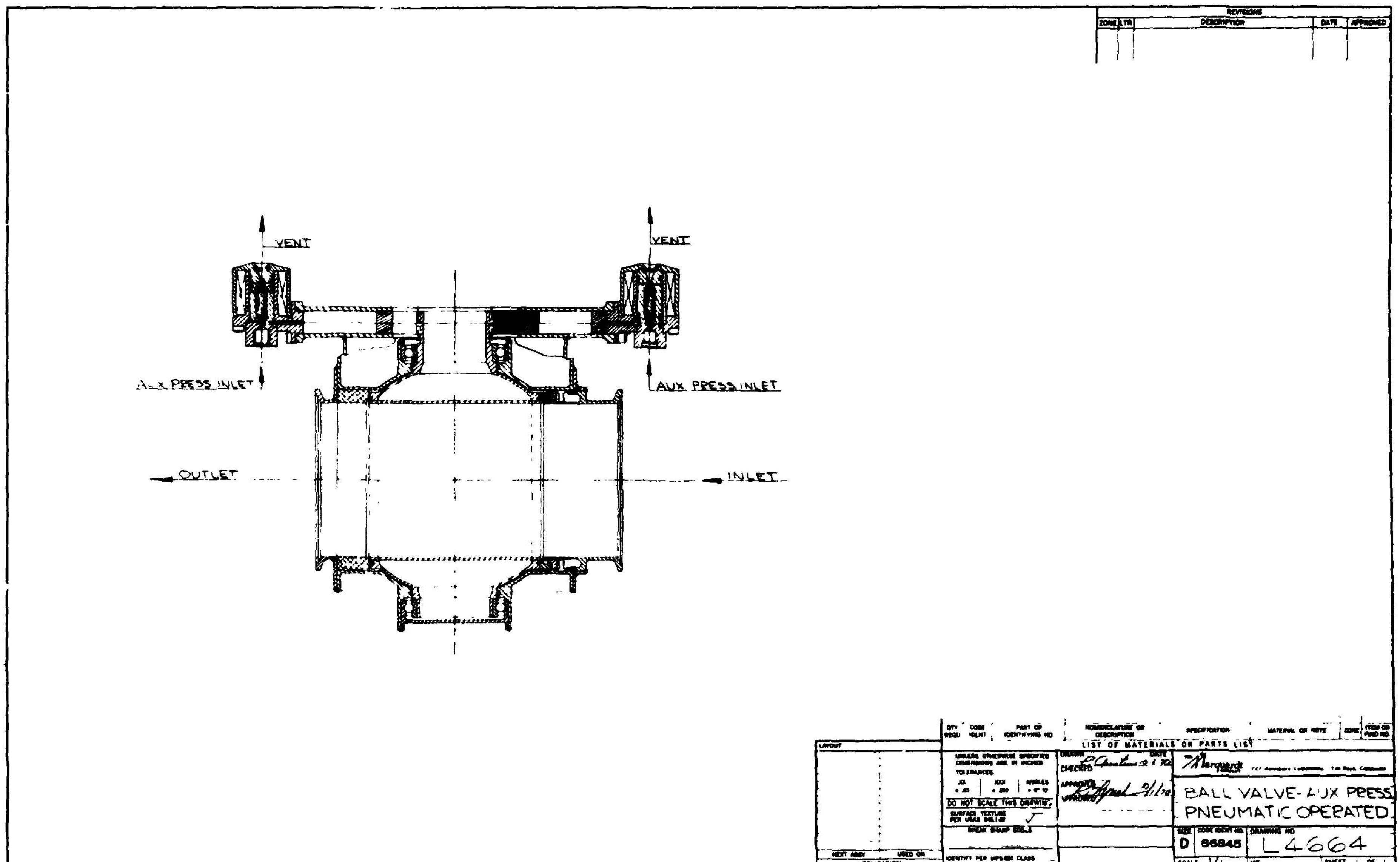
Figure 6



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Marquardt
BALL VALVE-AUX PRESS
PNEUMATIC OPERATED
SIZE
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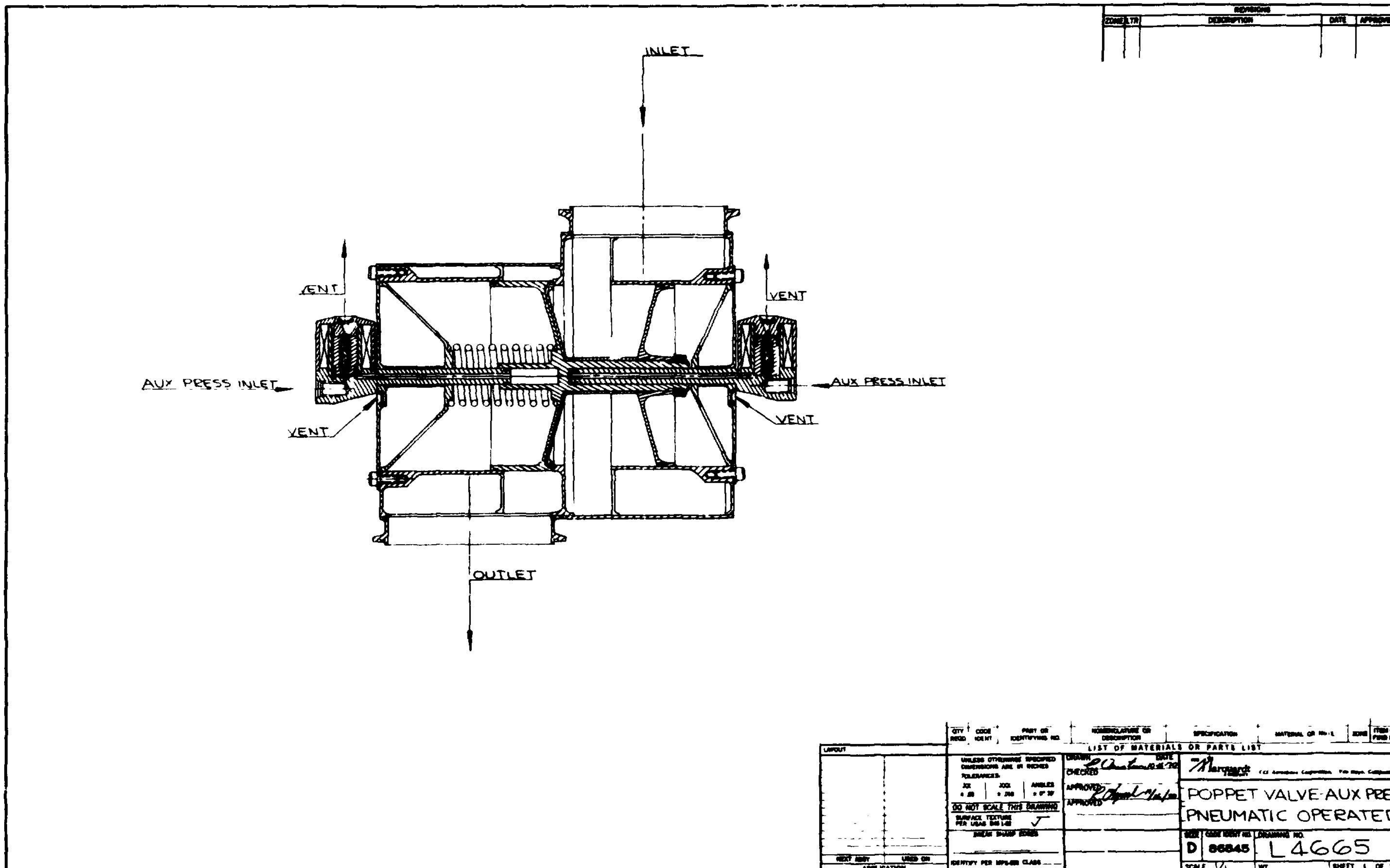
Figure 1.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POPPET VALVE-AUX PRESS
PNEUMATIC OPERATED
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APPLICATION

Figure 43

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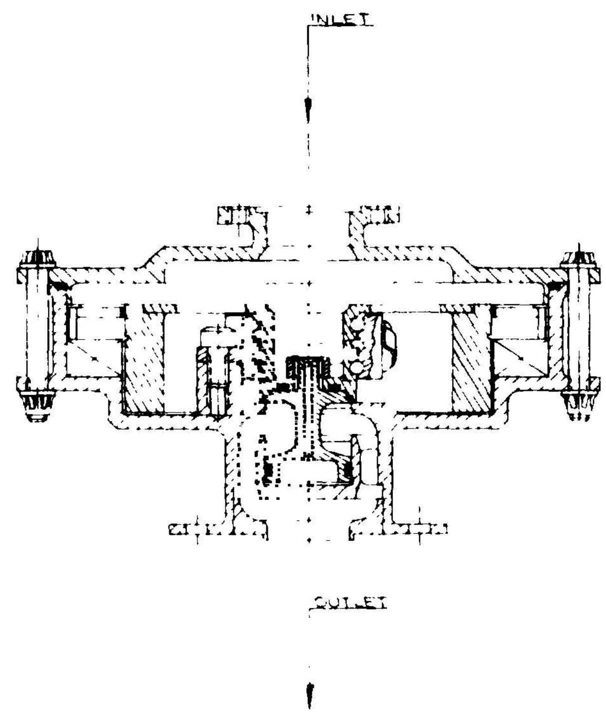

LIST OF MATERIAL OR PARTS

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APPROVED

Marquardt

POPPET VALVE-TORQUE MOTOR BALL SCREW OPERATED

SIZE D

DRAWING NO L4666

SCALE 2/1 | WT | SHEET 1 OF 1

Figure ... .00

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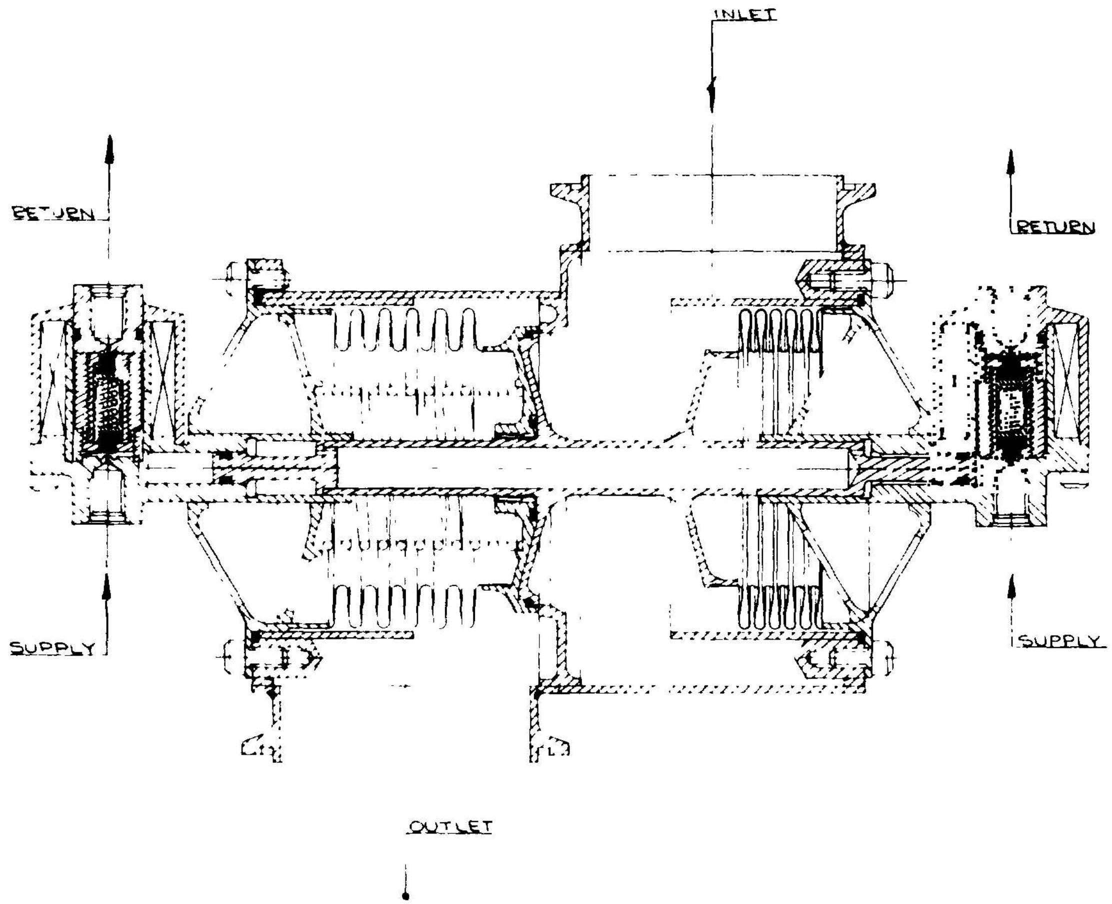

LIST OF MATERIALS OR PARTS LIST

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APPROVED

DATE

The Marquardt Company

POPPET VALVE, AUX PRESS HYDRAULIC OPERATED

SIZE	CODE IDENT NO	DRAWING NO
D	86845	L4667

SCALE 2/1 WT SHEET 1 OF 1

which cause the poppet to move to the full open position. Upon termination of the command signal the pilot poppet moves to effect closing of the vent passage and opens the bleed tube to permit the poppet cavity to be pressurized by inlet pressure, negating opening pressure forces until the spring moves the main poppet to the closed position. This configuration requires line pressure to effect operation and also exhibits a fail-safe characteristic in that the valve will close in the event of electrical failure or decrease in inlet pressure below a predetermined minimum pressure. Sizing of the valve was based upon a flow factor ($C_d A_t$) of 0.62 in.^{2} (4.0 cm^2) and an inlet pressure of 400 psia (276 N/cm^2).

Figure 40 defines a line pressure pneumatic operated poppet valve which embodies all characteristics of the assumed configuration of the analysis program. Sized for a flow factor ($C_d A_t$) of 0.62 in.^{2} (4.0 cm^2) and an inlet pressure (P_1) of 400 psia (276 N/cm^2), valve actuation is accomplished by a double acting cylinder. The packaging concept is unique in that minimum height and width of the package is achieved and the seat seal is the only potential internal leak path with the valve in the closed position. Three-way solenoid operated pilot valves are used to control the pneumatic cylinder actuator. With slight modification, the pilot valves could be combined into a single 5-way valve. Valve seat preloads are exerted by coil springs to assure a fail/safe design.

A rotary motion shut-off, torque motor actuated valve configuration is shown in Figure 41. Designed for a flow factor ($C_D A_t$) of 0.94 in.^{2} (6.06 cm^2) and 60 psi (41.4 N/cm^2), the configuration is totally consistent with the analysis program and embodies several unique features. Spherical surfaces of the rotating ball are minimal by virtue of the large diameter trunions. Minimum weight is achieved by the thin shell design and assembly is simplified by the design approach. As shown, the bearings are dry, being isolated from the flow media by shaft seals, however, the design is such that leakage of the shaft seals is tolerable since the total housing is a pressure vessel capable of being welded or bolted to contain the media. The valve seal is located at the inlet side of the rotating element, at some minor increase in dribble volume, but minimizing potential leak paths when the valve is in the closed position. The seat is a preloaded soft seal configuration employing metal bellows as a spring loading element and a positive dynamic seal to allow the seal to "float" to achieve the seat seal. As the valve is opened, a cam profile on the ball trunion mechanically retracts the seal to minimize scrubbing and reduce torque requirements to effect rapid opening.

For a flow factor ($C_d A_t$) of 10 in.^{2} (64.516 cm^2) and a valve inlet pressure of 30 psi (20.7 N/cm^2), a rotary motion auxiliary pressure pneumatic cylinder actuated valve design is shown in Figure 42. Actuator sizing is based upon the availability of a 1500 psia (1034 N/cm^2) auxiliary pressure source.

The rotating element is a trunion mounted thin wall spherical element. Shaft seals are eliminated by utilizing the cylinder piston seals to contain line pressure. The actuator is a double acting cylinder with actuator force output being converted to torque by a rack and pinion drive. Actuator pressure control is performed by three-way solenoid valves. As in the previous ball-valve design, cam action retraction of the seat seal is incorporated to minimize seat scrubbing. Metal bellows mounting of the seat seal is employed to achieve pressure energizing of the seal and allow "float" to enhance sealing capability with minimum preload.

An auxiliary pressure pneumatic cylinder actuator is employed to operate a linear motion shut-off in the valve design shown in Figure 43. The poppet is totally balanced in all positions with seat preload exerted by the coil spring, which also assures fail/safe operation. Actuator control, by the 3-way solenoid valves provides rapid response opening and closing.

The configuration of Figure 44 utilizes a torque motor actuator to drive a linear motion shut-off through a ball-screw linkage. By the unique employment of the ball-screw the poppet rotates 90^o ($\pi/2$ rad) as it is retracted. This type of motion imparts a self-cleaning capability or high dirt tolerance to the valve seat seal. The poppet is fully balanced to minimize operating force requirements. The ball screw drive mechanism provides greater than 90% efficiency power transmission as well as imparting the unusual poppet motion. Design criteria for the configurations were determined by the analysis program for a flow factor (C_dA_t) of 0.31 in^2 (2.0 cm^2) and an inlet pressure of 400 psia (276 N/cm^2).

Hydraulic cylinder actuation of a linear motion shut-off device is depicted by the design of Figure 45. For the design criteria of a flow factor (C_dA_t) of 3.3 $in.^2$ (21.3 cm^2) and an inlet pressure (P_1) of 30 psia (20.7 N/cm^2), metal bellows were selected for the dynamic seals in the propellant wetted volumes. Hydraulic cylinder pressure control is performed by 3-way solenoid operated valves. Valve seat preload is exerted by the coil spring to assure fail/safe operation, and moving element guidance is by sliding fits located outside the propellant cavities. Hydraulic fluid wetted volumes have been isolated from the propellant cavities and located such that appropriate thermal control of the hydraulic fluid may be incorporated, if required, without compromising the design.

A study of the components of these various valve designs provided insight to allow meaningful valve trade off studies to arrive at the best design for the intended application.

Valve Concept Trade-off Studies

In December 1970, The Marquardt Company completed its efforts in support of NASA Contract NAS 9-10866, the purpose of which was an analytical trade-off study of various types of injector valve concepts and their performance characteristics as applicable to gaseous oxygen/gaseous hydrogen rocket engines for Space Stations in the 500 to 2500 lbs. (2224 to 11,121N) thrust range. That study concluded with recommendations for valve designs for the 1000 lbs. (4448 N) thrust level operating at 20 and 400 psia (13.8 and 276 N/cm^2) inlet pressure.

The same analysis techniques which were developed under that contract were also applied to the specific requirements of the program reported herein. The principal differences in the two requirements are the engine thrust level (1500 lbs. vs. 1000 lbs.) (6672 N vs. 4448 N) valve response (30 ms vs 100-200 ms), maximum operating temperature (850°R vs. 660°R) (472°K vs 367°K) and valve life (1 million cycles vs. 100,000 cycles). The analysis employed were based on certain assumptions which permitted emphasis on comparing many different valves rather than optimizing a specific type of valve, without making the number of variables unmanageable. These assumptions are again reviewed in Table XII to put the results in proper perspective.

Low Pressure Concepts

By applying the developed analysis techniques to the valve requirements of this program, data for both low pressure and high pressure applications were generated. Table XIII presents the low pressure data that was computed. In reviewing these data, it is evident that a poppet type valve results in the lowest weight valve configuration. Actuation by a high pressure pneumatic or hydraulic source results in an attractive valve system. Data from Table XIII have been plotted in bar graph form and are presented as Figure 46 for the gaseous oxygen application. This figure presents a weight comparison for the most promising shut-off device and actuator combination. A chart with the same type of data, but for the low pressure hydrogen application is shown in Figure 47. Both of these figures indicate that a poppet valve featuring a high pressure hydraulic or pneumatic actuator is the most desirable approach for this application.

Early during May 1971, a meeting was held with the NASA LeRC Project Manager to review the program scope in light of recent systems contractors conclusions that the high pressure APS approach is favored over the low pressure APS approach. At that time, it was decided to terminate further effort in support of the low pressure system and instead to put forth a more concentrated effort in support of the high pressure system. Consequently, a final valve concept for the low pressure system was not completed.

TABLE XII - TRADE-OFF STUDY ASSUMPTIONS

Item	Linear Motion Device	Rotary Motion Device
Flow Coefficient	.65	.90
Seat Nominal Diameter	$\sqrt{\frac{C_d A_t}{C_d \frac{\pi}{4}}}$	$\sqrt{\frac{C_d A_t}{C_d \frac{\pi}{4}}}$
Seat Land Width	.01 Inches (.0254 cm)	.03 Inches (.0762 cm)
Minimum Seat Preload	5 lbs/circumferential inch (8.76 N/circum. cm)	5 lbs/circumferential inch (8.76 N/circum. cm)
Seat Configuration	Flat on flat	Spherical on spherical
Stroking	At constant acceleration	90° at constant acceleration
Coefficient of Friction when Lifting Seal	0	0.1
Pressure Forces	Maximum pressure unbalance equal to P x 3% of seal area	Bearing and shaft seal drag negligible
Acceleration Environment	10 g	Unaffected
Material of Moving Mass	Stainless steel	Stainless steel
Maximum Electrical Power Defined at	32 VDC, 70°F (21°C)	32 VDC, 70°F (21°C)
Maximum Pull in Voltage @ Max. Temp. and Pressure	18 VDC	18 VDC
Linkage Efficiency	90%	90%

TABLE XIII - PREDICTED PERFORMANCE O_2 & H_2 VALVES FOR 1500 LB THRUST ENGINE @ O/F = 2.5, PINLET = 20 PSIA (13.8 N/cm^2)

C_dA_t (in^2) (cm^2)	P_1 (Psia) (N/cm^2)	Configuration Shut Off	Configuration Actuator	Act. Press. Paux (Psia) (N/cm^2)	Act. Media	Elect. Power (Watts)	t_{open} (MS)	t_{close} (MS)	Weight Cres. Construction (LB) (Kg)	Weight Est. Al. Construction (LB) (Kg)	Envelope (In.) (cm)	Act. Media Used Per Cycle Std. in^3 (cm^3)	Pilot Valve $(C_dA_t)_o$ (in^2) (cm^2)	Pilot Valve Arm. Dia. (in) (cm)	Pilot Valve Coil Turns N	Main Valve Travel Time t (MS)
13.5 (87.1) (O_2 Valve)	20 (13.8)	Linear	Cyl.	60 (41.4)	O_2	112	34.1	34.1	25.70 (11.66)	21.7 (9.84)	7.5 dia x 8.8 (19.05) (22.35)	50.4 (826)	.0865 (.5581)	.705 (1.791)	1213	20
				150 (103.4)	O_2	56	32.5	32.5	9.08 (4.12)	5.08 (2.30)	7.5 dia x 7.2 (19.05) (18.29)	45.4 (744)	.0289 (.1865)	.600 (1.524)	1445	20
						84	30.0	30.0	7.34 (3.33)	3.34 (1.51)					963	
						112	28.8	28.8	6.91 (3.13)	2.91 (1.32)					723	
						84	35.2	35.2	46.05 (20.89)	42.05 (19.07)	9.1 dia x 8.6 (23.1) (21.84)	65.5 (1073)	.083 (.5355)	.938 (2.383)	1617	10
						112	30.2	30.2	14.75 (6.69)	10.75 (4.88)	7.5 dia x 3.0 (19.05) (20.32)				1213	
				1500 (1034)	O_2	28	29.0	29.0	6.12 (2.78)	2.12 (.96)	7.4 dia x 7.2 (18.80) (18.29)	42.8 (701)	.0027 (.0174)	.506 (1.285)	970	20
						56	27.0	27.0	6.06 (2.75)	2.06 (.93)					485	
						84	26.4	26.4	6.04 (2.74)	2.04 (.92)					323	
						112	26.0	26.0	6.03 (2.73)	2.03 (.92)					243	
					O_2	28	30.4	30.4	7.03 (3.19)	3.03 (1.37)		61.1 (1001)	.0071 (.0458)	.794 (2.017)	1550	10
						56	22.8	22.8	6.55 (2.97)	2.55 (1.16)					775	
						84	20.2	20.2	6.43 (2.92)	2.43 (1.10)					517	
						112	19.0	19.0	6.40 (2.90)	2.40 (1.09)					388	
				3000 (2068)	O_2	28	29.3	29.3	6.04 (2.74)	2.04 (.92)		42.7 (700)	.0014 (.0090)	.507 (1.288)	747	20
						56	27.2	27.2	6.00 (2.72)	2.00 (.91)					374	
						84	26.5	26.5	5.99 (2.72)	1.99 (.90)					249	
						112	26.2	26.2	5.98 (2.71)	1.98 (.90)					187	
					O_2	28	25.9	25.9	6.44 (2.92)	[illegible] (1.11)		52.5 (860)	.0033 (.0213)	.758 (1.925)	1130	10
						56	20.5	20.5	6.28 ([illegible])	[illegible].28 (1.03)					565	
						84	18.7	18.7	6.23 (2.82)	[illegible].23 (1.01)					377	
						112	17.8	17.8	6.21 (2.82	2.21 (1.00)					283	
				1500 3000 (1034) (2068)	Hyd.	28	27.3	27.3	6.2 (2.81)	2.2 (1.00)						20
						56	25.8	25.8								
						84	24.8	24.8								
						112	23.6	23.6								
					Hyd.	28	16.9	16.9								10
						56	15.7	15.7								
						84	14.5	14.5								
						112	13.4	13.4								

TABLE XIII (Continued) - PREDICTED PERFORMANCE O_2 & H_2 VALVES FOR 1500 LB THRUST ENGINE @ O/F - 2.5, PINLET = 20 PSIA (13.8 N/cm^2)

C_dA_t (In^2) (cm^2)	P_1 Paux (N/cm^2)	Configuration Shut Off	Configuration Act-uator	Act. Pres. Paux (Psia) (N/cm^2)	Act. Media	Elect. Power (Watts)	t_{open} (MS)	t_{close} (MS)	Weight Cres. Construction (LB) (Kg)	Weight Est. Al. Construction (LB) (Kg)	Envelope (In.) (cm)	Act. Media Used Per Cycle Std. In^3 (cm^3)	Pilot Valve $(C_dA_t)_o$ (In^2) (cm^2)	Pilot Valve Arm. Dia. (In) (cm)	Pilot Valve Coil Turn N	Main Valve Travel Time t (MS)
13.5	20	Ball	Cyl.	1500	O_2	84	30.0	30.0	17.44 (7.91)	7.6 (3.45)	6.5x7.2x9.1	129.2 (2117)	.0414 (.2671)	.556 (1.412)	1133	20
(87.1)	(13.8)			(1034)							(16.51x18.29x					
(O_2						112	28.8	28.8	15.16 (6.88)	6.3 (2.86)	23.11)				850	
Valve)				3000	O_2	56	27.9	27.9	14.28 (6.48)	5.5 (2.49)	6.5x7.2x8.8	64.6 (1059)	.0155 (.1000)	.437 (1.110)	1060	20
				(2068)		84	27.0	27.0	13.97 (6.34)	5.1 (2.31)	(16.51x18.29x				707	
						112	26.5	26.5	13.88 (6.30)	5.0 (2.27)	22.35)				530	
				1500	Hyd.	28	18.5	18.5	17.5 (7.94)	8.7 (3.9)	6.5x7.2x9.0					10
				3000							(16.51x18.29x					
				(1034)							22.86)					
				(2068)		56	16.5	16.5	16.6 (7.53)	7.8 (3.5)	6.5x7.2x8.8					
											(16.51x18.29x					
						84	14.5	14.5	16.4 (7.44)	7.6 (3.4)	22.35)					
						112	12.5	12.5	16.3 (7.39)	7.5 (3.4)						
						28	24.6	24.6	14.7 (6.67)	5.9 (2.7)	6.5x7.2x8.7					20
											(16.51x18.29x					
						56	23.5	23.5	14.3 (6.49)	5.5 (2.5)	22.10)					
						84	22.4	22.4	14.2 (6.44)	5.4 (2.4)						
						112	21.9	21.9	14.2 (6.44)	5.4 (2.4)						
21.45	20	Linear	Cyl.	60	H_2	84	30.4	30.4	17.2 (7.8)	10.9 (4.94)	9.5 dia x 9	118.8 (1947)	.047 (.303)	.563 (1.430)	1200	20
(138.4)	(13.8)			(41.4)		112	29.1	29.1	13.2 (6.0)	6.9 (3.13)	(24.13 x 22.86)				900	
(Fuel				150	H_2	28	30.8	30.8	31.75 (14.40)	22.4 (10.16)	9.9 dia x 10.5	91.0 (1491)	.014 (.090)	.448 (1.138)	2040	
Valve)				(103.4)							(25.15 x 26.67)					
						56	28.0	28.0	10.25 (4.65)	3.9 (1.8)	9.5 dia x 9				1020	
						84	26.7	26.7	10.00 (4.54)	3.7 (1.7)	(24.13 x 22.86)				680	
						112	26.6	26.6	9.95 (4.51)	3.6 (1.6)					510	
					H_2	84	37.3	37.3	21.55 (9.78)	15.2 (6.9)		179 (2933)	.057 (.368)	.800 (2.032)	1350	10
						112	34.3	34.3	16.27 (7.38)	9.9 (4.5)					1013	
				1500	H_2	28	26.6	26.6	9.75 (4.42)	3.4 (1.5)		89.0 (1458)	.0014 (.0090)	.379 (.963)	705	20
				(1034)		56	25.9	25.9	9.74 (4.42)	3.4 (1.5)					353	
						84	25.6	25.6	9.73 (4.41)	3.4 (1.5)					235	
						112	25.5	25.5	9.73 (4.41)	3.4 (1.5)					177	
					H_2	28	33.1	33.1	10.23 (4.64)	3.9 (1.8)		138 (2261)	.0044 (.0284)	.642 (1.631)	1230	10
						56	29.1	29.1	10.05 (4.56)	3.7 (1.7)					615	
						84	27.8	27.8	10.01 (4.54)	3.7 (1.7)					410	
						112	27.1	27.1	9.99 (4.53)	3.6 (1.6)					308	

TABLE XIII (Continued) - PREDICTED PERFORMANCE O_2 & H_2 VALVES FOR 1500 LB THRUST ENGINE @ O/F = 2.5, PINLET = 20 PSIA (13.8 N/cm^2)

C_dA_t (In^2) (cm^2)	P_1 (Psia) (N/cm^2)	Configuration Shut Off	Configuration Act-uator	Act. Press. Paux (Psia) (N/cm^2)	Act. Media	Elect. Power (Watts)	t_{open} (MS)	t_{close} (MS)	Weight Cres. Construction (LB) (Kg)	Weight Est. Al. Construction (Lb) (Kg)	Envelope (In.) (cm)	Act. Media Used Per Cycle Std. In^3 (cm^3)	Pilot Valve $(C_dA_t)_o$ (In^2) (cm^2)	Pilot Valve Arm. Dia. (In.) (cm)	Pilot Valve Coil Turn N	Main Valve Travel Time (MS)
21.43	20	Linear	Cyl.	3000	H_2	28	26.3	26.3	9.66 (4.38)	3.3 (1.5)	9.5 dia x 9	89.0 (1458)	.0007 (.0045)	.363 (.922)	563	10
(138.4)	(13.8)			(2068)		56	25.7	25.7	9.65 (4.38)	3.3 (1.5)	(24.13 x 22.86)				282	20
(Fuel						84	25.5	25.5	9.65 (4.38)	3.3 (1.5)					188	
Valve)						112	25.4	25.4	9.65 (4.38)	3.3 (1.5)					141	
					H_2	28	30.5	30.5	9.91 (4.50)	3.6 (1.6)		133 (2179)	.0021 (.0135)	.640 (1.626)	907	10
						56	27.9	27.9	9.85 (4.47)	3.5 (1.6)					453	
						84	26.9	26.9	9.84 (4.46)	3.5 (1.6)					303	
						112	26.5	26.5	9.83 (4.46)	3.5 (1.6)					227	
				1500	Hyd.	28	28.4	28.4	9.60 (4.35)	3.3 (1.5)						20
				3000		56	27.5	27.5								
				(1034)		84	26.2	26.2								
				(2068)		112	25.1	25.1								
		Ball	Cyl.	1500	H_2	56	28.8	28.8	31.7 (14.4	16.7 (7.6)	8.3x8.8x10.4	193 (3163)	.0306 (.1974	.517 (1.313)	993	20
				(1034)		84	27.9	27.9	25.4 (11.5)	10.4 (4.7)	(21.08x22.35x				745	
						112	26.3	26.3	25.0 (11.3)	10.0 (4.5)	26.42)					
				3000	H_2	28	29.2	29.2	24.4 (11.1)	9.4 (4.3)	8.3x8.8x10.2	33.8 (554)	.0108 (.0697)	.400 (1.016)	1800	20
				(2068)		56	27.2	27.2	23.2 (10.5)	8.2 (3.7)	(21.08x22.35x				900	
						84	26.5	26.5	23.1 (10.5)	8.1 (3.7)	25.91)				600	
						112	26.2	26.2	23.1 (10.5)	8.1 (3.7)					450	
				3000	H_2	112	21.8	21.8	33.0 (15.0)	18.0 (8.2)	8.3x8.8x10.7	119 (1950)	.0760 (.4903)	.648 (1.646)	595	10
				(2068)							(21.08x22.35x					
											27.18)					
		Ball	Cyl.	1500	Hyd.	28	27.5	27.5	24.2 (11.0)	9.2 (4.2)	8.3x8.8x10.3					20
				3000							(21.08x22.35x					
				(1034)							26.16)					
				(2068)		56	25.8	25.8	23.9 (10.8)	8.9 (4.0)	8.3x8.8x10					
						84	23.5	23.5	23.8 (10.8)	8.8 (4.0)	(21.08x22.35x					
						112	22.6	22.6	23.8 (10.8)	8.8 (4.0)	25.40)					

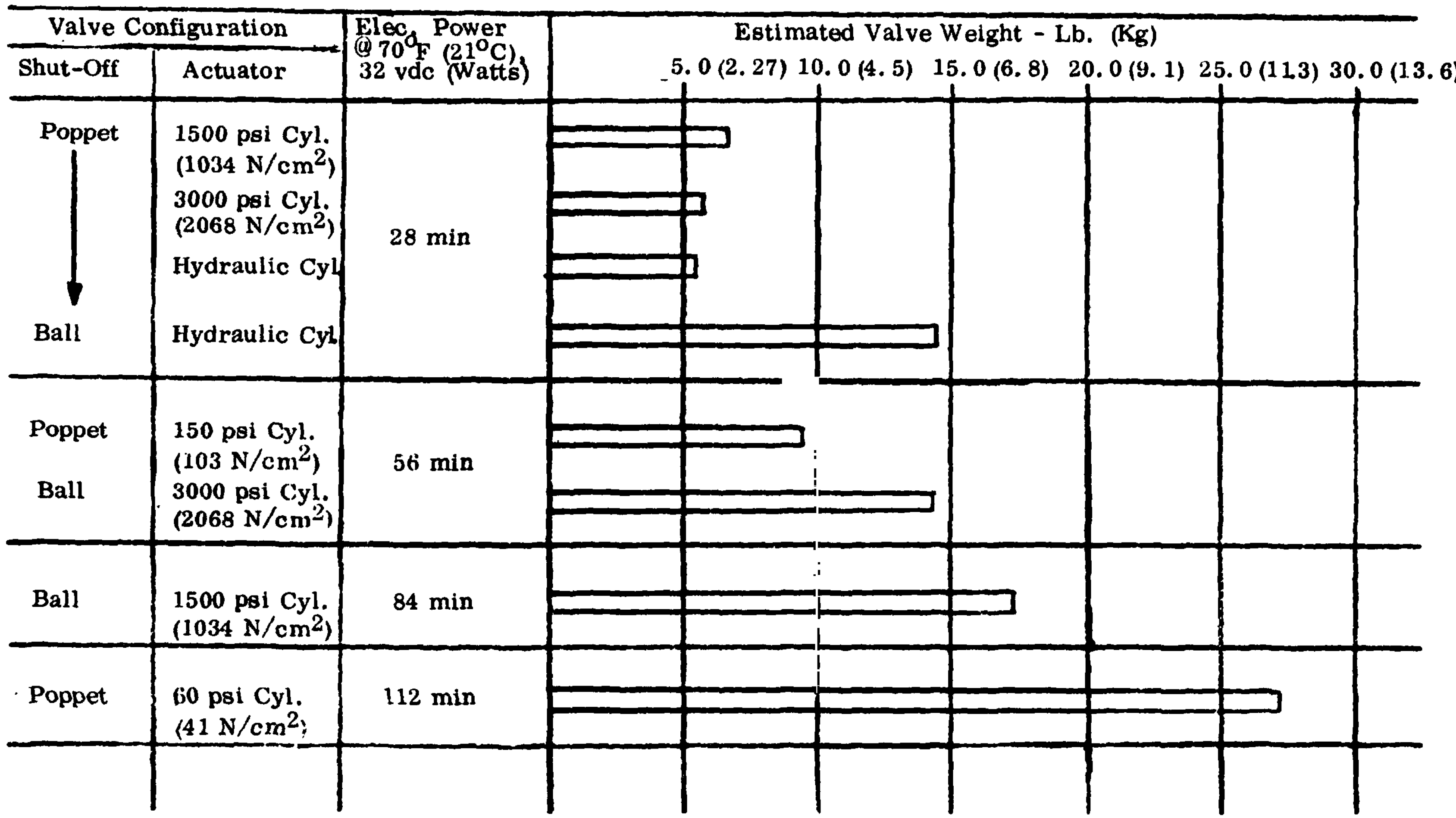


Figure 46 - Low Pressure (20 psia) (13.8 N/cm^2) O_2 Injector Valve Characteristics, Response Time $\leq$30 Milliseconds

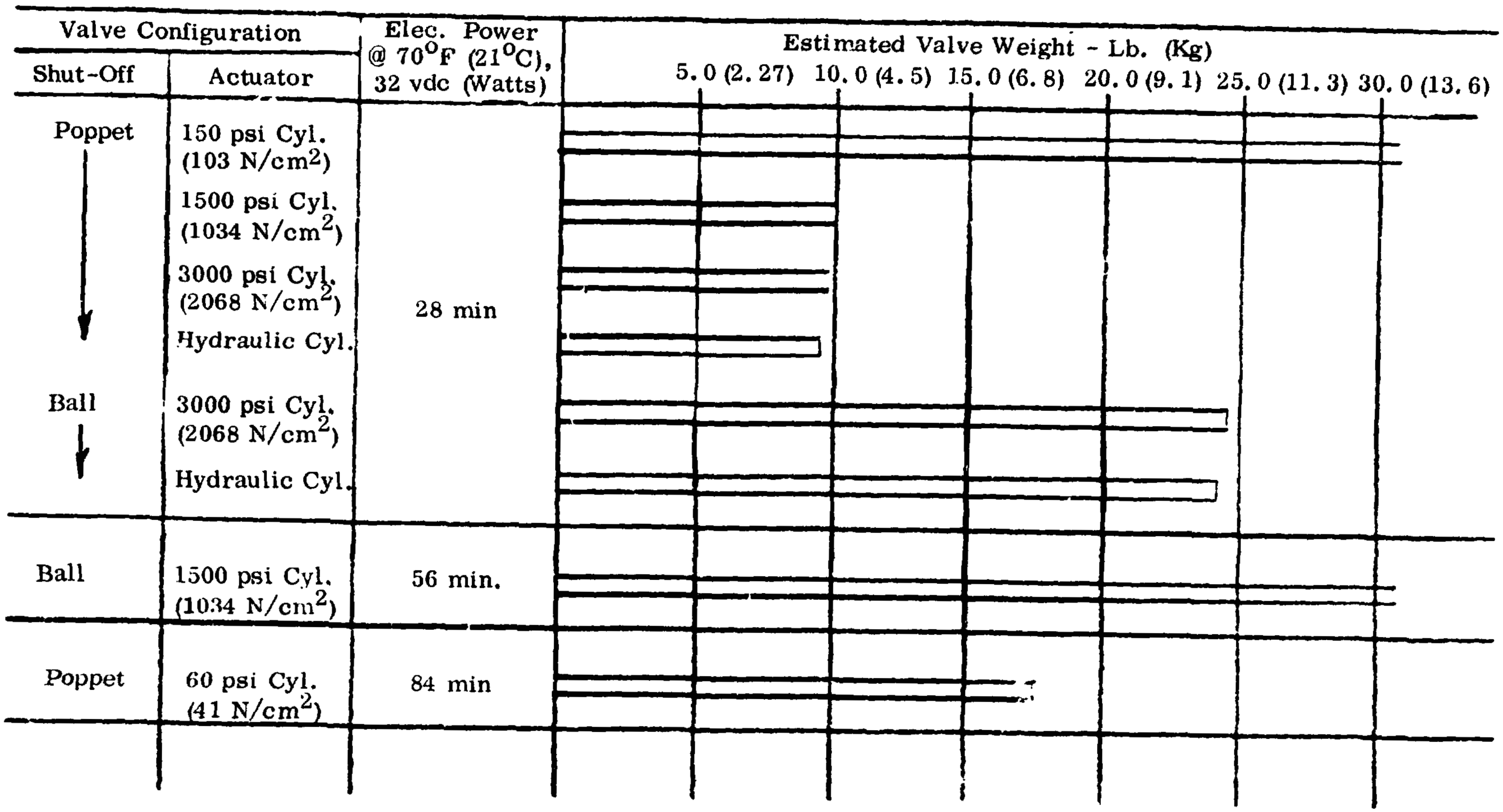


Figure 47 - Low Pressure (20 psia) (13.8 N/cm^2) H_2 Injector Valve Characteristics, Response Time ≤ 30 Milliseconds

High Pressure Concepts

The computer data generated for the high pressure concept is presented in Table XIV and appropriate weight data are presented in bar graph form in Figure 48. This study again concluded that the optimum valve configuration is a poppet valve featuring either propellant pressure operation or high pressure gas or hydraulic operation at an electrical power level of 56 watts per valve.

Further review of the power sources of these three types of actuators led to the preparation of the Table XV. This table is entitled, "Pressurant Supply Pressure Penalties" and shows that while all three types of energy sources require a vent-line or return line, only the propellant gas pressurization approach does not require a separate supply system. Both high pressure gas or high pressure hydraulic actuation would require feed lines, pressure regulation, over-pressure protection, filtration, instrumentation, and tankage as a minimum. In addition, the hydraulic system would require a pump drive and the required energy source for the pump drive. Finally, the operating temperature range of conventional hydraulic systems is not compatible with the minimum temperature requirements of the APS valves. Thus, it is evident that utilization of propellant gas for actuation is far and away the least complicated approach with hydraulic pressure actuation being the most complicated approach. Since the weight penalty incurred in choosing a propellant gas actuated poppet valve over a high pressure pneumatic or hydraulic actuated poppet valve is only 10 to 15 percent it appeared that the simplicity of the propellant gas actuation approach far outweighed this weight penalty and consequently this particular approach was selected as the most desirable approach for the high pressure system application.

The selection of propellant gas as the actuation medium still leaves several choices open. These include actuators which are pressurized to open and pressurized to close, actuators which are pressurized to open only, with a spring return, and actuators which are vented to open with a spring return to close. Furthermore, the valve subcomponents may include partial or full pressure balancing. To permit the selection of an optimum approach from among these choices, several analog computer programs were prepared which fully simulate the dynamic behavior of the valve and which define actuator areas, bleed orifices, and spring requirements. These analog programs were employed in conjunction with the data obtained from the sealing closure testing to permit final selection of the optimum valve concept.

The basic relationships which define the movement of a pneumatic piston actuator are shown in Figure 49. An analog computer flow diagram for the actuator in Figure 49, is shown in Figure 50. A list of various valve subcomponent performance characteristics and some representative values is shown

TABLE XIV - PREDICTED PERFORMANCE O_2 & H_2 VALVES FOR 1500 LB THRUST ENGINE @ O/F = 4.0, PINLET = 400 PSIA (276 N/cm^2)

C_dA_t In^2 (cm^2)	P_1 (Psia (N/cm^2)	Configuration Shut Off	Configuration Actuator	Act. Press. (Psia) (N/cm^2)	Act. Gas or Media	Elect. Power (Watts)	t_{open} (MS)	t_{close} (MS)	Weight (LB)	Weight (Kg)	Envelope In.	Envelope (cm)	Act. Media Used/ Cycle Std. In^3	Act. Media Used/ Cycle Std. (cm^3)	Pilot Valve Flow Factor $(C_dA_t)_o$ In^2	Pilot Valve Flow Factor $(C_dA_t)_o$ (cm^2)	Pilot Solenoid Arm. Dia. In.	Pilot Solenoid Arm. Dia. (cm)	Pilot Solenoid Coil Turns N	Main Valve Travel Time (MS)
1.325	400	Poppet	Cyl.	400	O_2	28	21.5	21.5	1.74	(.789)	2 dia x 3.75	(5.08 dia x 9.525)	6.42	(105.20)	.0085	(.0548)	.500	(1.27)	1600	10
(8.548)	(276)			(276)		56	18.3	18.3	1.17	(.531)	2 dia x 2.0	(5.08 dia x 5.08)							800	
						84	17.2	17.2	1.09	(.494)									534	
						112	16.7	16.7	1.07	(.485)									400	
					H_2	28	16.2	16.2	1.00	(.454					.00214	(.0138)	.286	(.726)	810	
						56	15.7	15.7	.98	(.445)									405	
						84	15.5	15.5	.96	(.435)									270	
						112	15.4	15.4	.93	(.422)									203	
		Ball	Cyl.	400	O_2	56	33.1	33.1	5.42	(2.459)	2.82x5.3x3	(7.163x13.462x7.62)	22.1	(362.15)	.0292	(.1884)	.875	(2.22)	1495	
				(276)		84	27.1	27.1	3.30	(1.497)	2.2x4.4x3	(5.588x11.176x7.62)							910	
						112	24.1	24.1	3.02	(1.370)	2.2x4.3x3	(5.588x10.922x7.62)							682	
					H_2	28	20.4	20.4	2.14	(.971)	2.2x4x3	(5.588x10.160x7.62)			.00735	(.04742)	.474	(1.204)	1500	
						56	17.8	17.8	1.74	(.789)	2.2x3.4x3	(5.588x8.636x7.62)							750	
						84	16.9	16.9	1.68	(.762)	2.2x3.2x3	(5.588x8.128x7.62)							500	
						112	16.4	16.4	1.66	(.753)	2.2x3.2x3	(5.588x8.128x7.62)							375	
		Poppet	Cyl.	1500	O_2	28	21.0	21.0	1.69	(.767)	2 dia x 3.5	(5.08 dia x 8.89)	6.40	(104.88)	.00226	(.01458)				
				(1034)		56	18.1	18.1	1.12	(.508)	2 dia x 2.0	(5.08 dia x 5.08)								
						84	17.0	17.0	1.04	(.472)										
						112	16.5	16.5	1.02	(.463)										
				3600	O_2	28	21.0	21.0	1.69	(.767)	2 dia x 3.3	(5.08 dia x 8.38)	6.35	(104.06)	.00057	(.00368)				
				(2068)		56	18.1	18.1	1.12	(.508)	2 dia x 2.0	(5.08 dia x 5.08)								
						84	17.0	17.0	1.04	(.472)										
						112	16.5	16.5	1.02	(.463)										
		Ball	Cyl.	1500	O_2	28	21.5	21.5	2.22	(1.007)	2.2x4.25x3	(5.588x10.795x7.62)	22.6	(370.35)	.00795	(.05129)				
				(1034)		56	18.3	18.3	1.65	(.748)	2.2x3.5x3	(5.588x8.89x7.62)								
						84	17.2	17.2	1.57	(.712)	2.2x3.4x3	(5.588x8.636x7.62)								
						112	16.7	16.7	1.55	(.703)	2.2x3.3x3	(5.588x8.382x7.62)								
				3000	O_2	28	21.5	21.5	1.69	(.767)	2.2x4.25x3	(5.588x10.795x7.62)	22.1	(362.15)	.00201	(.01297)				
				(2068)		56	18.3	18.3	1.12	(.508)	2.2x3.5x3	(5.588x8.89x7.62)								
						84	17.2	17.2	1.04	(.472)	2.2x3.4x3	(5.588x8.636x7.62)								
						112	16.7	16.7	1.02	(.463)	2.2x3.3x3	(5.588x8.382x7.62)								
		Poppet	Cyl.	1500	Hyd.	28	20.0	20.0	1.69	(.767)	2 dia x 3.5	(5.08 dia x 8.89)	.0407/.0203		.00112/.00028					
				3000		56	18.0	18.0	1.12	(.508)	2 dia x 2.0	(5.08 dia x 5.08)	(.6670/.3327)		(.00723/.001806)					
				(1034)		84	16.6	16.6	1.04	(.472)										
				(2068)		112	16.0	16.0	1.02	(.463)										
		Ball	Cyl.	1500	Hyd.	28	20.9	20.9	2.22	(1.007)	2.2x4.25x3	(5.588x10.795x7.62)	.144/.0725		.0039/.00098					
				3000		56	17.8	17.8	1.65	(.748)	2.2x3.5x3	(5.588x8.89x7.62)	(2.360/1.1880)		(.02516/.00632)					
				(1034)		84	16.7	16.7	1.57	(.712)	2.2x3.4x3	(5.588x8.636x7.62)								
				(2068)		112	16.1	16.1	1.55	(.703)	2.2x3.3x3	(5.588x8.382x7.62)								

Valve Configuration: Shut-Off	Valve Configuration: Actuator	Elec. Power @ $70^{o}F$ ($21^{o}C$), 32 vdc (Watts)
Poppet	400 psi Cyl. (276 N/cm^2)	
↓	1500 psi Cyl. (1034 N/cm^2)	
↓	3000 psi Cyl. (2068 N/cm^2)	
↓	Hydraulic Cyl.	2
Ball	1500 psi Cyl. (1034 N/cm^2)	
↓	3000 psi Cyl. (2068 N/cm^2)	
↓	Hydraulic Cyl.	
Poppet	400 psi Cyl. (276 N/cm^2)	
↓	1500 psi Cyl. (1034 N/cm^2)	
↓	3000 psi Cyl. (2068 N/cm^2)	56
↓	Hydraulic Cyl.	
Ball	1500 psi Cyl. (1034 N/cm^2)	
↓	3000 psi Cyl. (2068 N/cm^2)	
↓	Hydraulic Cyl.	
Ball	400 psi Cyl. (276 N/cm^2)	84

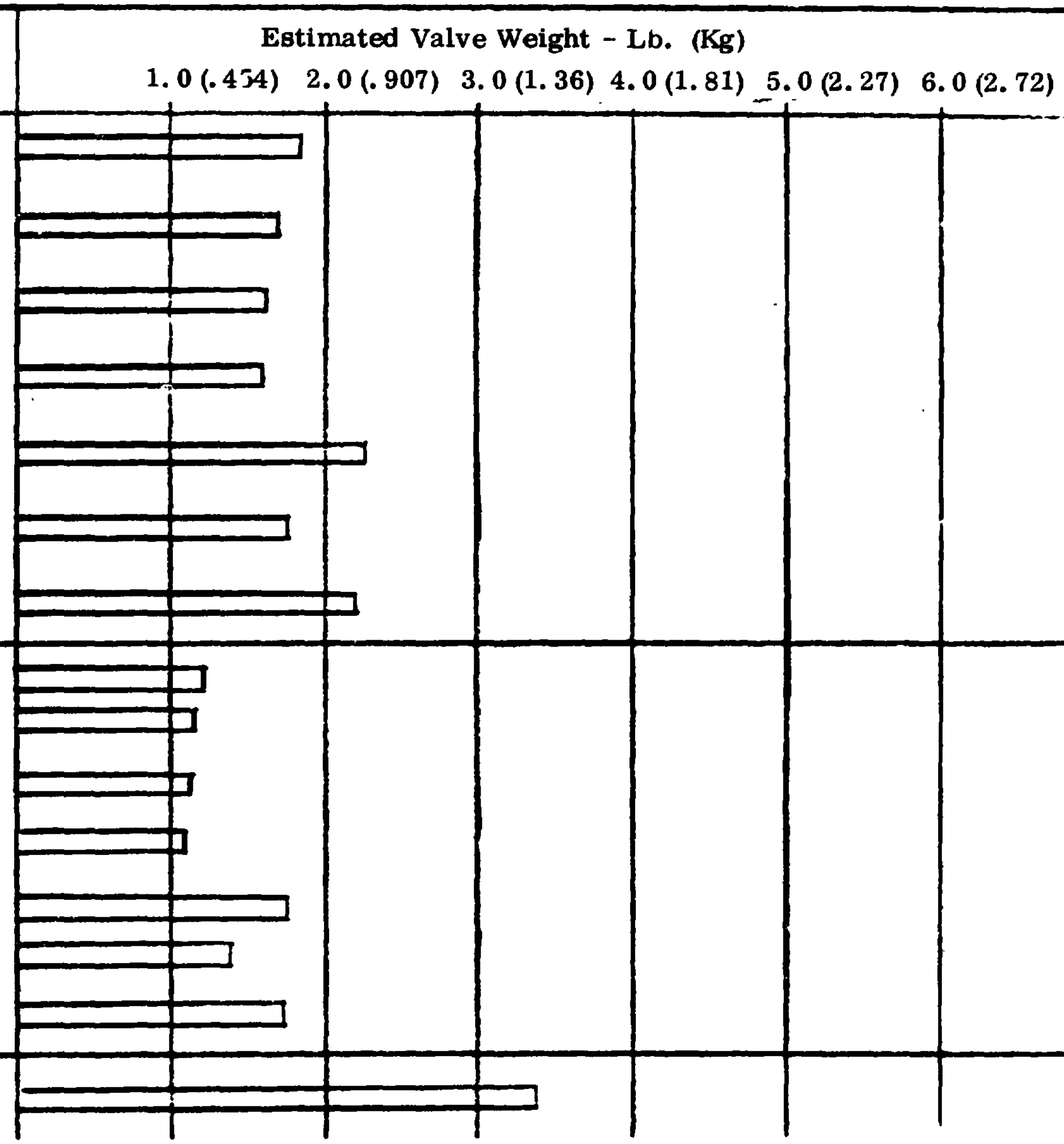


Figure 48 - High Pressure (400 psia) (276 N/cm^2) O_2/H_2 Injector Valve Characteristics, Response Time $\leq$ 30 Milliseconds

TABLE XV - PRESSURANT SUPPLY SYSTEM PENALTIES

PROPELLANT	HIGH PRESSURE GAS	HIGH PRESSURE HYDRAULIC
VENT LINE	VENT LINE	RETURN LINE
	PRESSURIZING LINE	PRESSURIZING LINE
	PRESSURE REGULATION	PRESSURE REGULATION
	OVER PRESSURE PROTECTION	OVER PRESSURE PROTECTION
	FILTRATION	FILTRATION
	INSTRUMENTATION	INSTRUMENTATION
	REDUNDANCY CONSIDERATION	REDUNDANCY CONSIDERATION
	TANKAGE	RESERVOIR
	BOOSTER PUMPS	PUMP
	PUMP DRIVES	PUMP DRIVE
	ENERGY SOURCES	ENERGY SOURCES

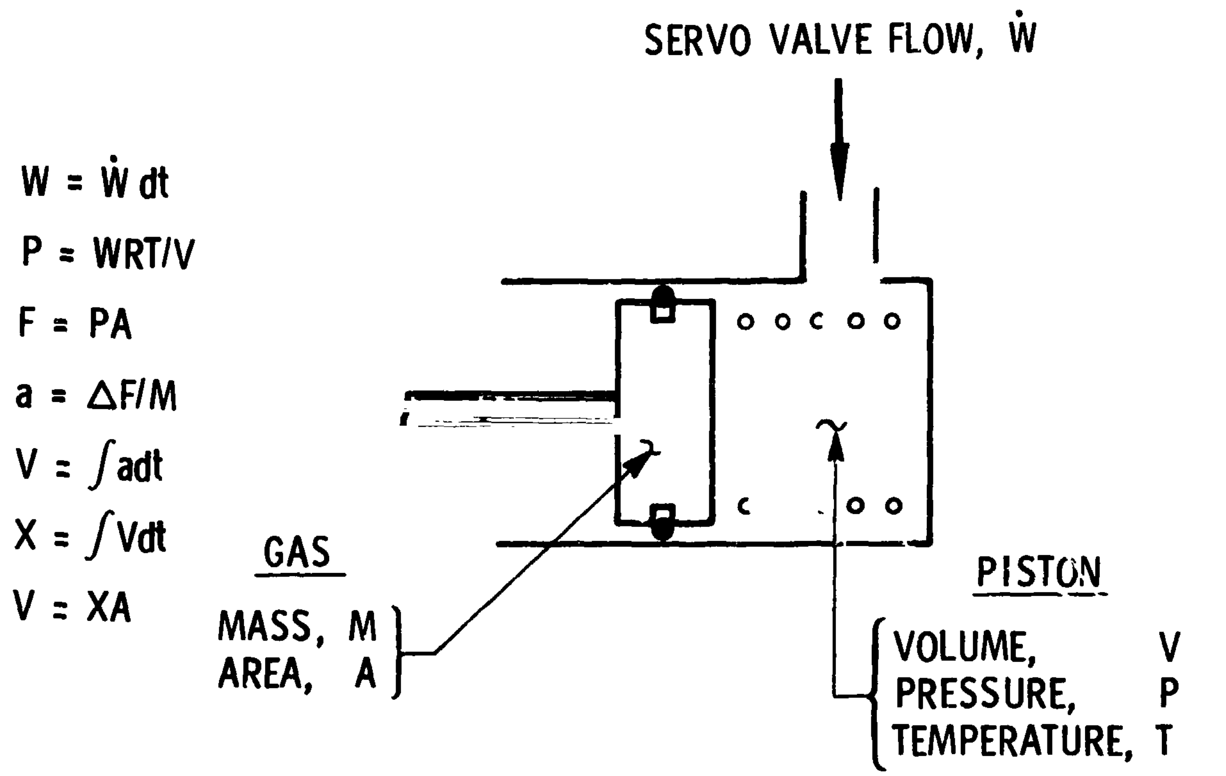


Figure 49 - Pneumatic Actuator Schematic

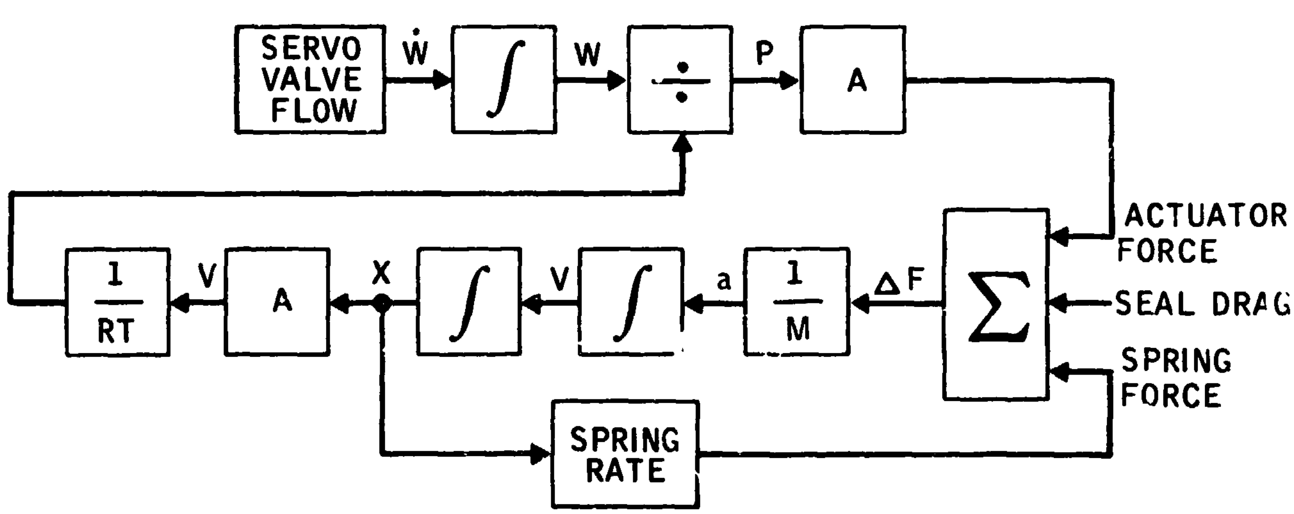


Figure 50 - Pneumatic Actuator Analog Computer Flow Diagram

in Table XVI. The primary variables were the seat load, operating gas, mass, initial volume, pilot delay, spring pre-load, piston area, and pilot orifice diameter. Well over 100 data runs of both the opening motion and the closing motion were performed.

In order to select the best combination of components, rating criteria were established for the various components. Table XVII is an example of the rating criteria for two types of actuators. Figures 51 and 52 show schematic drawings of a number of actuators that were studied and show their relative ratings. From Figure 52, it is evident that the pneumatic piston and solenoid received the highest rating.

Some of the sealing closures that were studied are presented schematically in Figures 53 and 54. Of the sealing closures studied, the flat poppet received the highest rating followed by the spherical poppet.

Relative weights for pneumatic operated poppet valves are presented in Figure 55 in bar graph form for convenience. These data are taken from Table XIV and show that electrical power available to the pilot valves is more significant than the pressure source up to a power level of 56 watts. Above 56 watts, no furhter improvements in value performance characteristics are obtained.

To further optimize the performance characteristics of specific valve concepts, the analog computer programs were run on those valve concepts which appeared most promising based on the concept trade-off studies and which are compatible with the sealing closures being evaluated. The outputs from these analog computer programs were recorded by X-Y plotter and Figures 56 and 57 are representative of the type of data attained. Figure 56 shows various parameters of interest during the opening motion of a pneumatically operated, double pressurized piston actuator operating a poppet type sealing closure, as shown schematically in Figure 58. Figure 57 shows the same parameters during closing motion.

The initial waviness in the pilot valve position is due to limitation of the analog programming; however, all other trace oscillations are real. The traces give an excellent picture of valve motion and the factors affecting valve motion and have revealed several interesting phenomena. Among these is the fact that the lightest moving mass does not necessarily result in the fastest response but that the valve may be "tuned" for better response by actually increasing the moving mass. Similarly the oscillations occurring in valve velocity in Figures 56 and 57 may be damped out by proper selection of the parameters shown in these figures.

TABLE XVI - VALVE SUBCOMPONENT PERFORMANCE CHARACTERISTICS

Seat Load	-	80 lbf	(356 N)
Spring Rate	-	85 lbf/in	(150.9 N/cm)
Pressure	-	450 lbf/in^2	(310 N/cm^2)
Stroke	-	0.32 in	(.8128 cm)
Seal Drag	-	4 lbf	(17.8 N)
Flexure	-	± 5 lbf	(22.2 N
Bellows	-	± 2 lbf	(8.9 N)
Gas	-	O_2	
Temperature	-	520°R	(289°K)
Mass	-	0.3 lbm	(.1361 Kg)
Initial Volume	-	0.1132 in^3	(1.8550 cm^3)
Pilot Delay	-	5 ms	
Pilot Time Constant	-	2 ms	
Spring Pre-load	-	28 lbf	(124.5 N)
Piston Area	-	0.354 in^2	(2.1639 cm^2)
Pilot Orifice Diameter	-	0.035 in	(.0889 cm)
Opening Time	-	24 ms	

TABLE XVII - SAMPLE SUBCOMPONENT RATING CHART
CHART II - ACTUATORS

SUBCOMPONENT CHARACTERISTIC	ELECTRIC TORQUE MOTOR	ELECTRIC MOTOR
POWER REQUIRED	1	U (UNABLE TO MEET RESPONSE)
EXPECTED CYCLE LIFE	8	9
WEIGHT	2	2
FORCE	4	2
STROKE	10	10
CONTAMINATION SENSITIVITY	9	9
RELIABILITY	9	9
PRESSURANT VENTING HAZARD	10	10
DEGREE OF FABRICATION DIFFICULTY	9	9
DEGREE OF INSPECTION DIFFICULTY	8	8
DEGREE OF ASSEMBLY DIFFICULTY	9	9
CLEANING & HANDLING REQUIREMENTS	9	9
MATERIALS COMPATIBILITY	2	2
ABILITY TO OPERATE @ 850°R	7	7
ABILITY TO OPERATE @ 200°R	1	1
TOTAL POINTS	98	95U

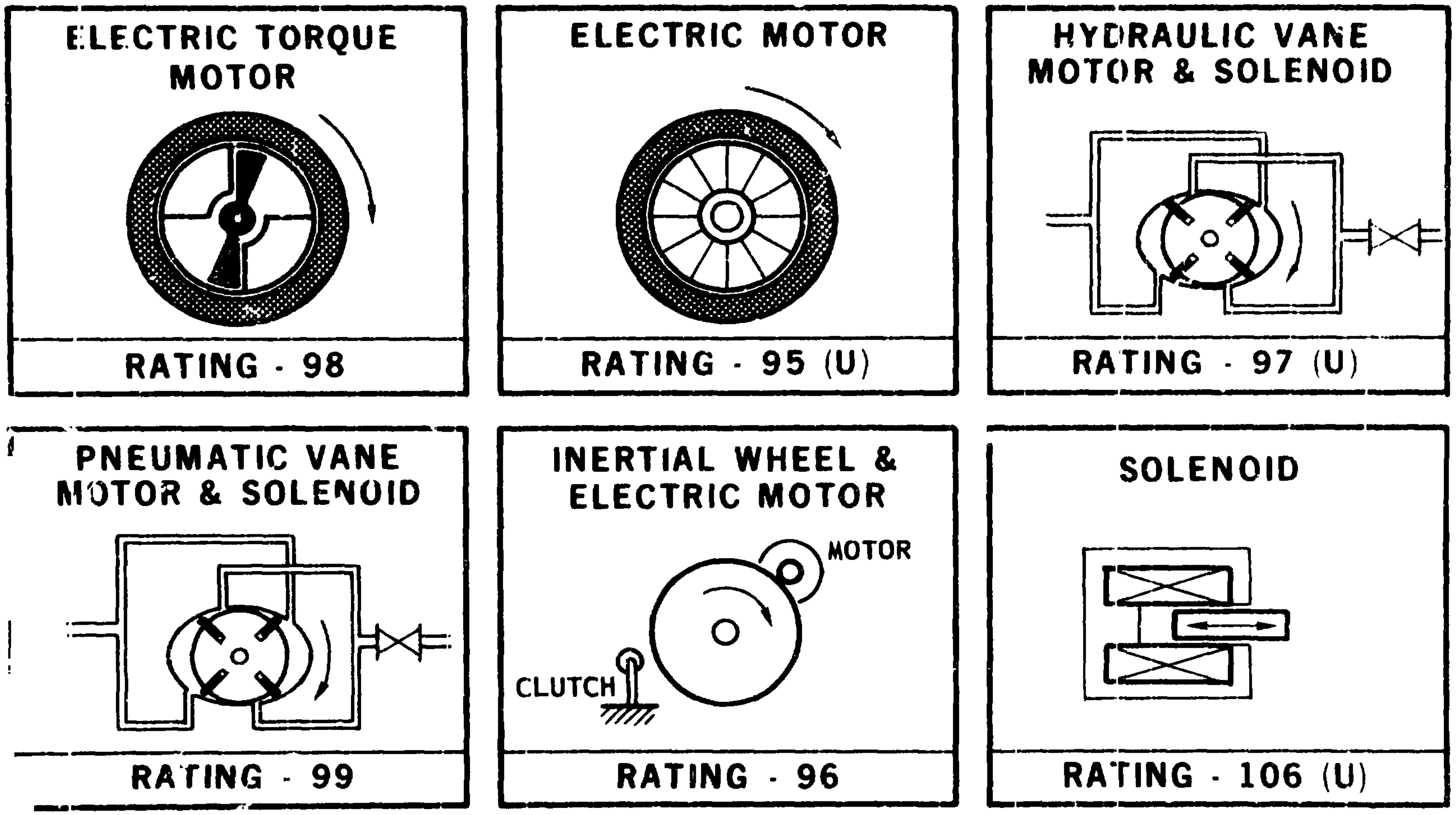


Figure 51 - Valve Subcomponent Rating Chart II - Summary
Actuators

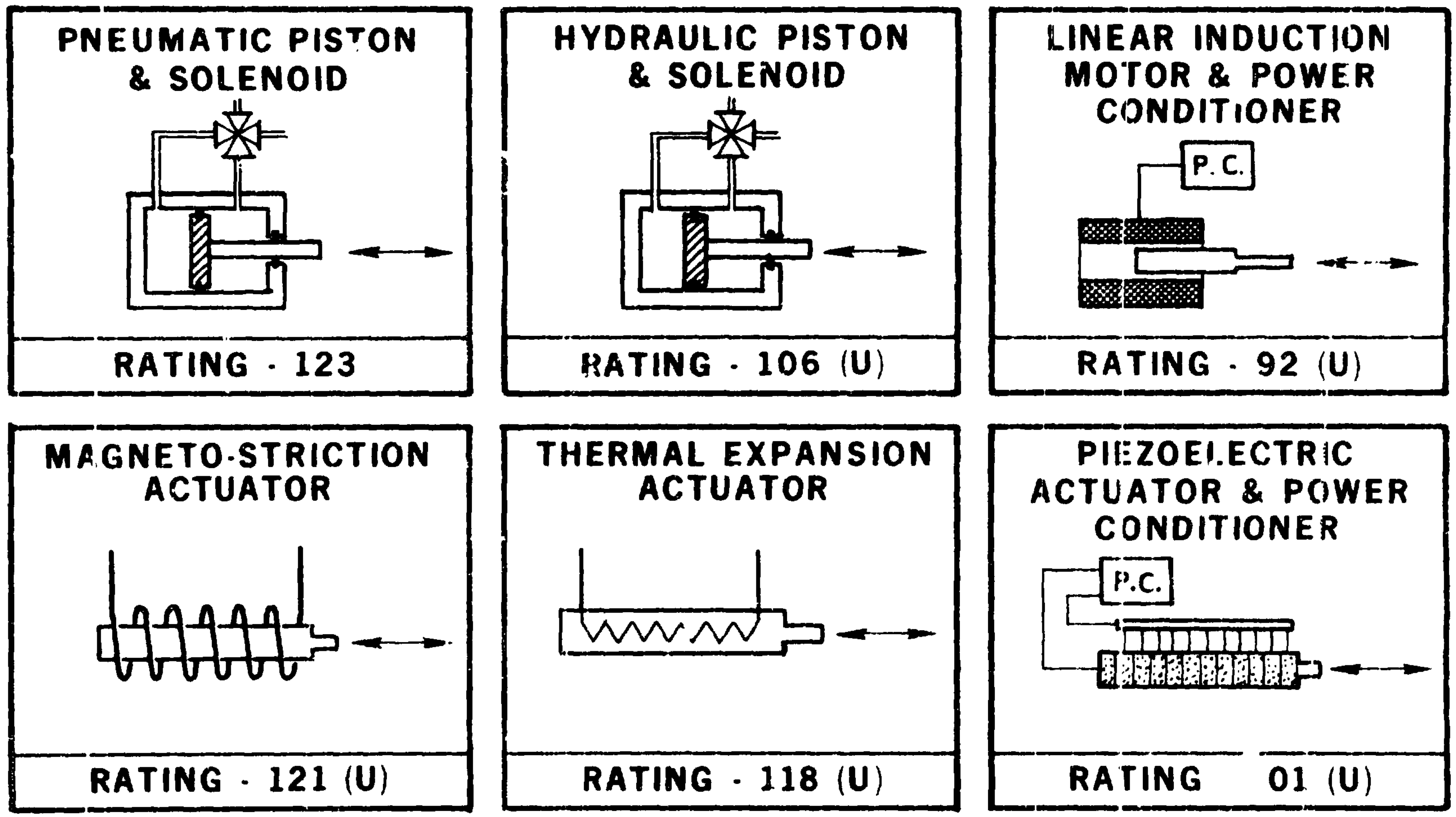


Figure 52 - Valve Subcomponent Rating Chart I - Summary
Sealing Closures

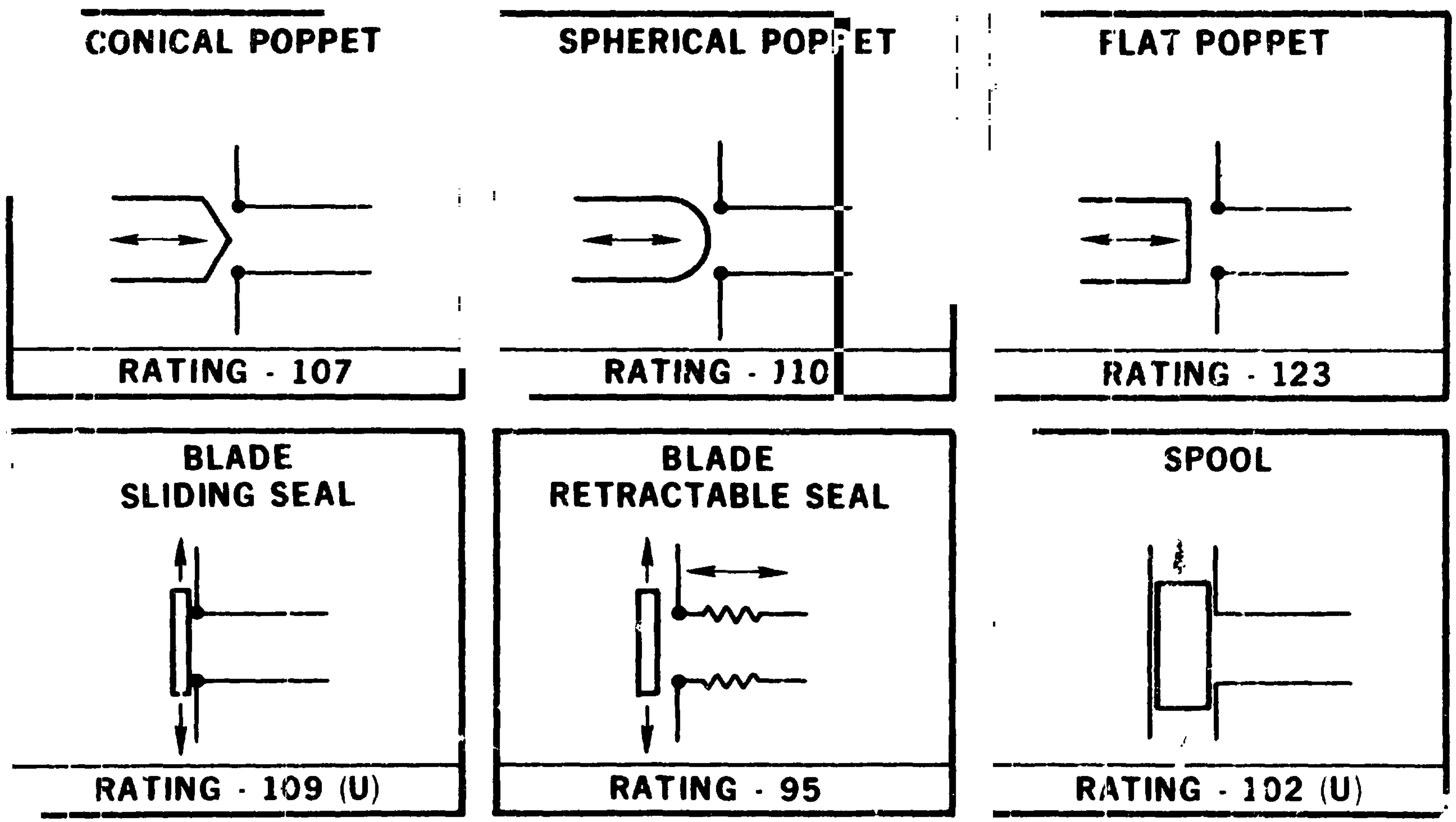


Figure 53 - Valve Subcomponent Rating Chart I - Summary
Sealing Closures

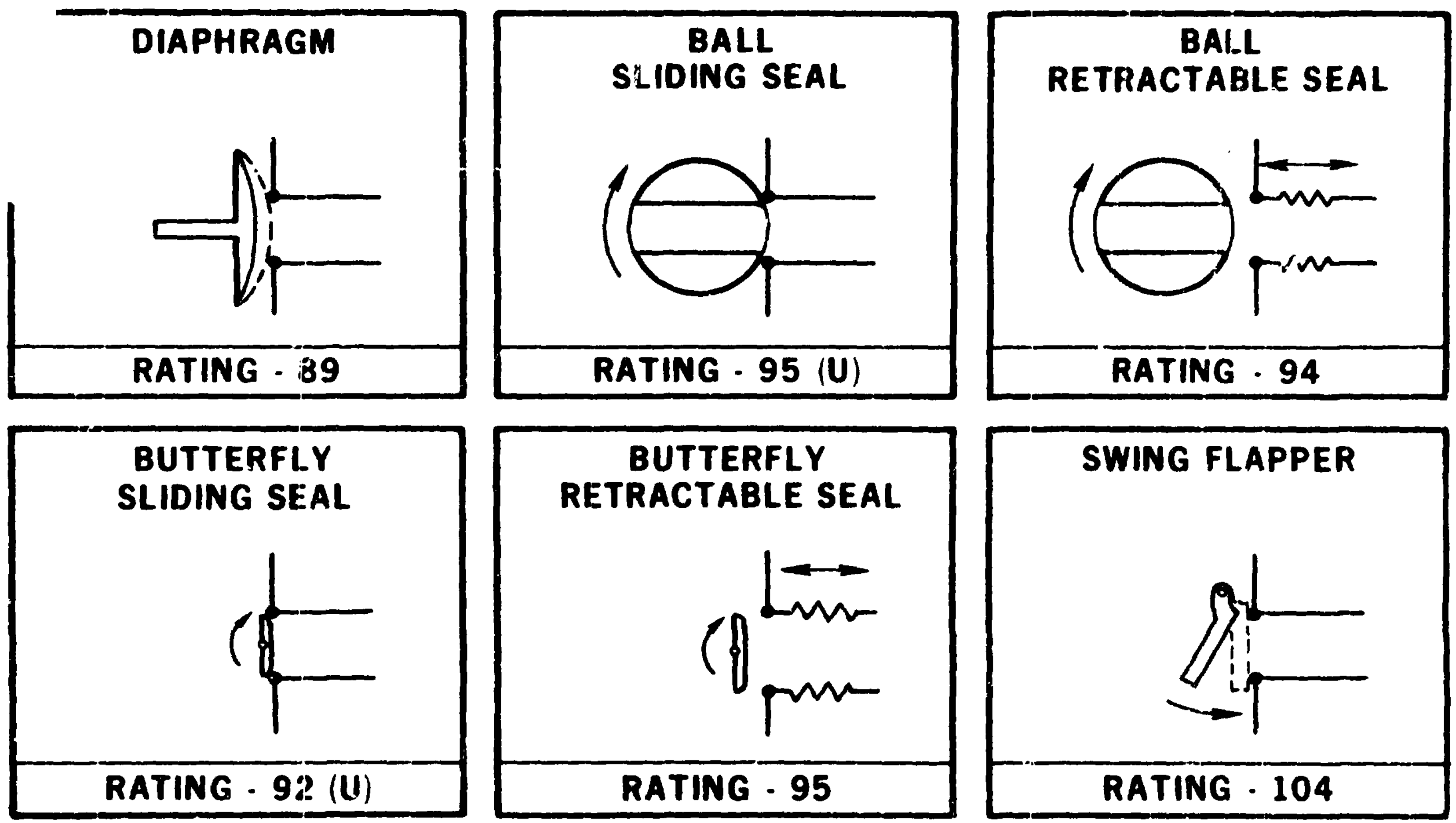


Figure 54 - Valve Subcomponent Rating Chart I - Summary
Sealing Closures

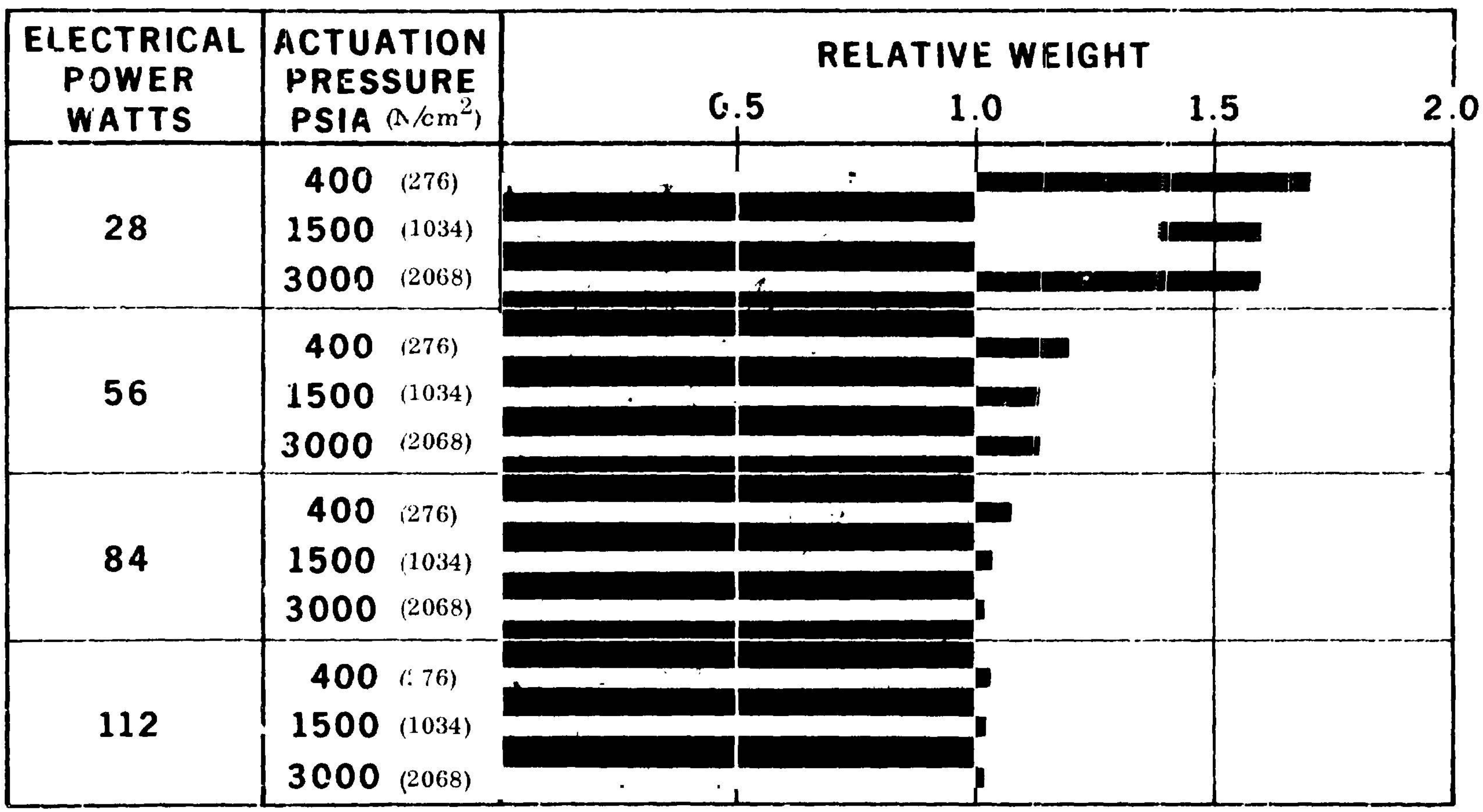


Figure 55 - Relative Weights of Pneumatic Piston Operated Poppet Valves

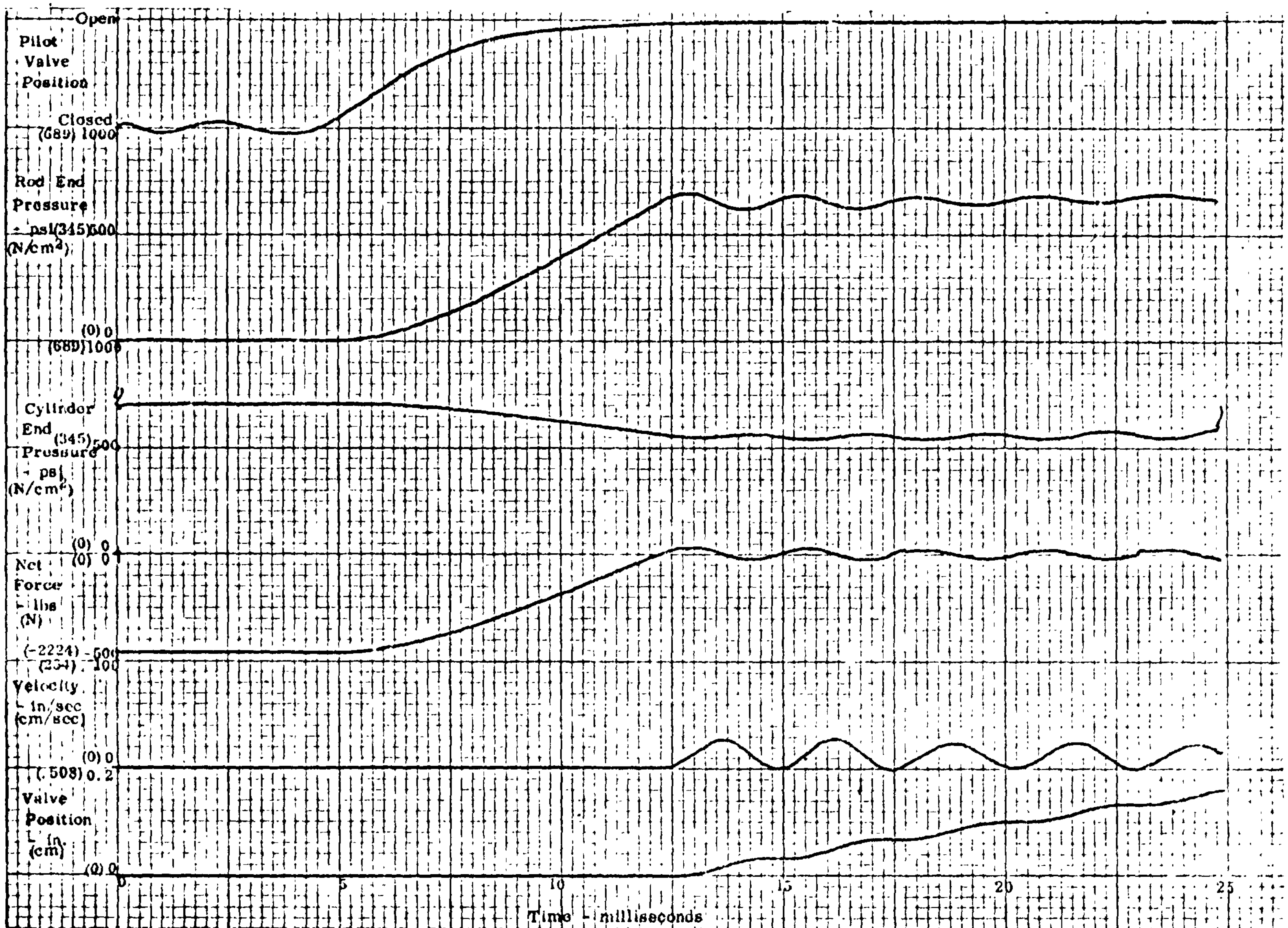


Figure 56 - Analog Computer Program Print Out - Valve Opening Motion

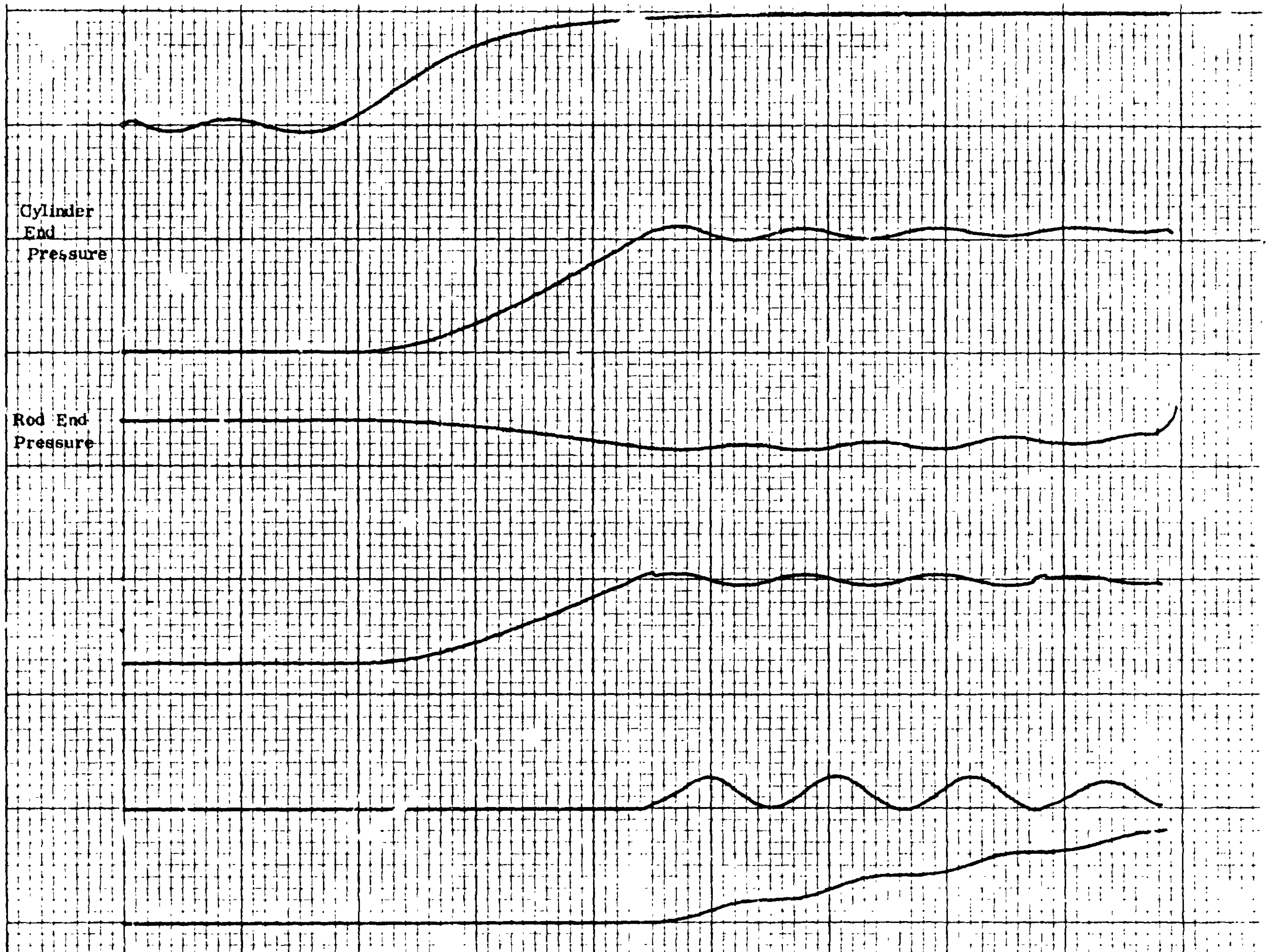


Figure 57 - Analog Computer Program Print Out - Valve Closing Motion

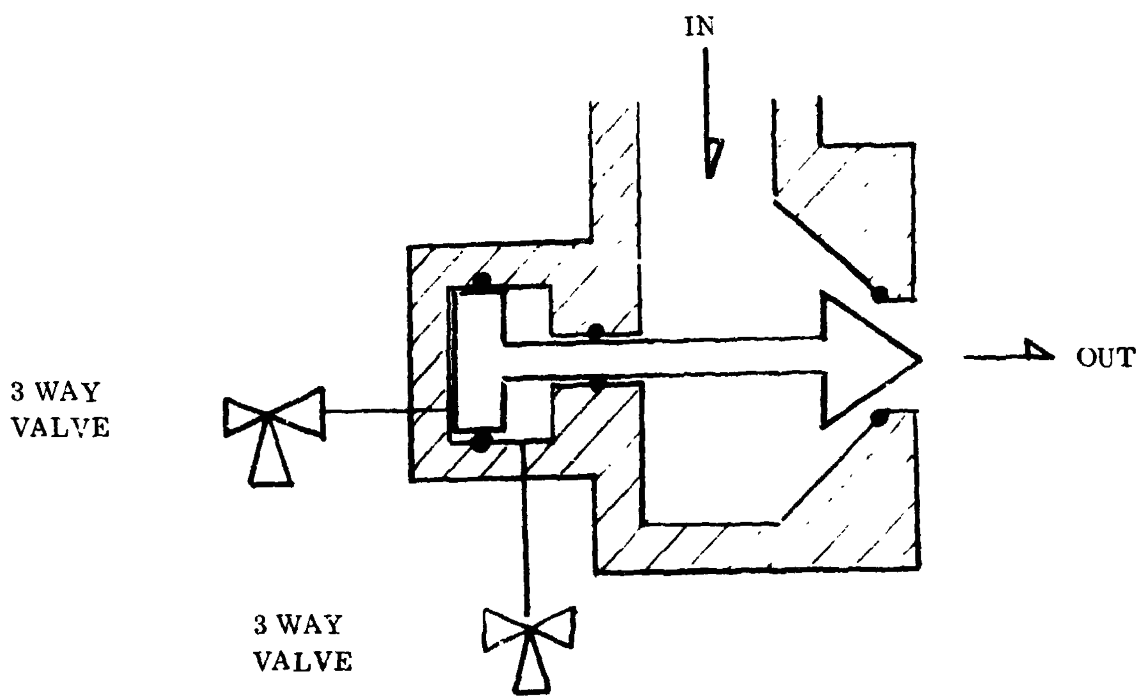


Figure 58 - Schematic of a Pneumatically Operated Poppet Valve

A summary chart showing possible pneumatic piston operated poppet valve concepts is presented in Figure 59. From this chart it can be concluded that the fifth concept shown which is the vent to open, spring return, pressure unbalanced poppet concept, is considered the most desirable concept. This concept was subsequently employed for the test fixture identified as P/N X28400. The second concept of Figure 59, which is the double acting piston, pressure balanced poppet valve, was subsequently developed as the second test fixture and is identified as P/N X27449. This concept was of interest primarily because of its very fast response characteristics; however, it was recognized that the concept was inherently less reliable than the P/N X28400 concept.

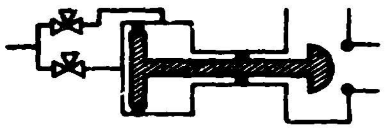
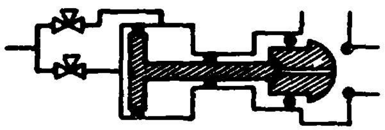
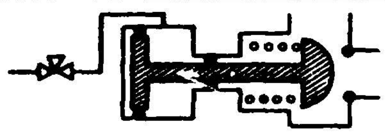
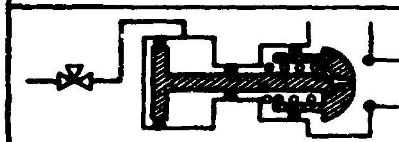
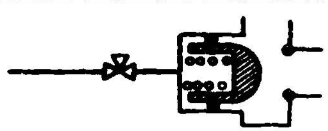
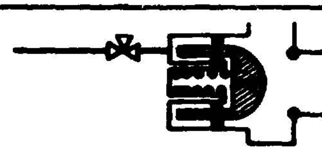
	RELATIVE CHARACTERISTICS RATING			COMMENTS
	WEIGHT	RELIABILITY	SEAT LOAD	
 DOUBLE ACTING PISTON. PRESSURE UNBALANCED POPPET.	5	5	8	REQUIRES TWO 3-WAY VALVES EASILY ADAPTS TO BIPROPELLANT VALVE SUBSTANTIAL RESPONSE FLEXIBILITY.
 DOUBLE ACTING PISTON. PRESSURE BALANCED POPPET.	7	4	7	REQUIRES TWO 3-WAY VALVES EASILY ADAPTS TO BIPROPELLANT VALVE GREAT RESPONSE FLEXIBILITY
 PRESSURIZE TO OPEN SPRING RETURN. PRESSURE UNBALANCED POPPET.	4	6	8	REQUIRES ONLY ONE 3-WAY VALVE EASILY ADAPTS TO BIPROPELLANT VALVE
 PRESSURIZE TO OPEN SPRING RETURN. PRESSURE BALANCED POPPET.	5	5	7	REQUIRES ONLY ONE 3-WAY VALVE EASILY ADAPTS TO BIPROPELLANT VALVE.
 VENT TO OPEN SPRING RETURN. PRESSURE UNBALANCED POPPET	8	8	8	VERY SIMPLE APPROACH REQUIRES ONLY 1 DYNAMIC SEAL, WHICH IS ACTIVE ONLY WHILE THE VALVE IS OPEN
 VENT TO OPEN SPRING RETURN. PRESSURE BALANCED POPPET.	9	7	7	QUITE SIMPLE. REQUIRES ONLY ONE 3-WAY VALVE

Figure 59 - Pneumatic Piston Operated Poppet Valve Concepts

VALVE PRELIMINARY DESIGNS

As a result of the valve tradeoff studies discussed in the previous section of this report, three pressure-actuated poppet valve concepts were further investigated and preliminary design layouts of these concepts were prepared. All three of these concepts utilized upstream propellant pressure for actuation and the concepts have been identified by how this pressure is utilized during operation. Thus, we have a double pressure-actuated valve, a pressurized-to-open/spring-to-close valve, and a vent-to-open/spring-to-close valve. These valve concepts are described in the next three sections.

Vent-to-Open Concept

This concept is shown in Figure 60. The valve is shown in the closed position and flow occurs from the left to the right. The upstream propellant cavity has been shaded coarsely and the downstream propellant cavity has been shaded finely. As shown in this figure, the seat consists of a gold-plated lip seal which mates with a flat poppet. A stop in parallel with the seat permits an initial deflection of up to 0.006 inch (.01524 cm) of the lip seat as the poppet makes the closure and then absorbs most of the impact energy of the closing poppet. By including the stop in parallel with the seat, impact forces and therefore wear of the sealing surface of the seat are minimized. The poppet is made of Inco 718 and has been lapped to one-half AA finish. Repeated cycling of the valve causes the poppet to deform the gold plating of the seat so that it also approaches the one-half AA finish. This sealing closure design was previously successfully demonstrated during the sealing closure evaluation test with the Rapid Screening Tester.

The poppet is in the shape of a cylinder with the downstream end of the cylinder open to the outlet side of the valve. This same downstream end also constitutes the sealing surface annulus. The poppet is guided by means of a metallic axial guidance flexure located within the poppet cylinder which is fixed to the valve housing through a centrally located shaft which, in turn, is fastened to a spider arrangement located in the outlet of the valve. The outer diameter of the poppet cylinder is sealed to the center flange body by means of a bellows. By venting the pressure trapped in the volume inclosed by the poppet cylinder, the bellows, and the center flange, a differential pressure force acts upon the poppet cylinder to move it in the open direction. This differential pressure force is equal to the difference in areas between the effective seating diameter and the effective bellows diameter times the pressure differential across the valve. Venting of this cavity is accomplished by means of a three-way solenoid valve which is located externally to the main shutoff valve and which is supplied with upstream propellant pressure. To close this valve, the same cavity is simply repressurized and the pressure force thus generated in combination with the spring forces of the bellows and the axial guidance flexures returned the poppet to the closed position.

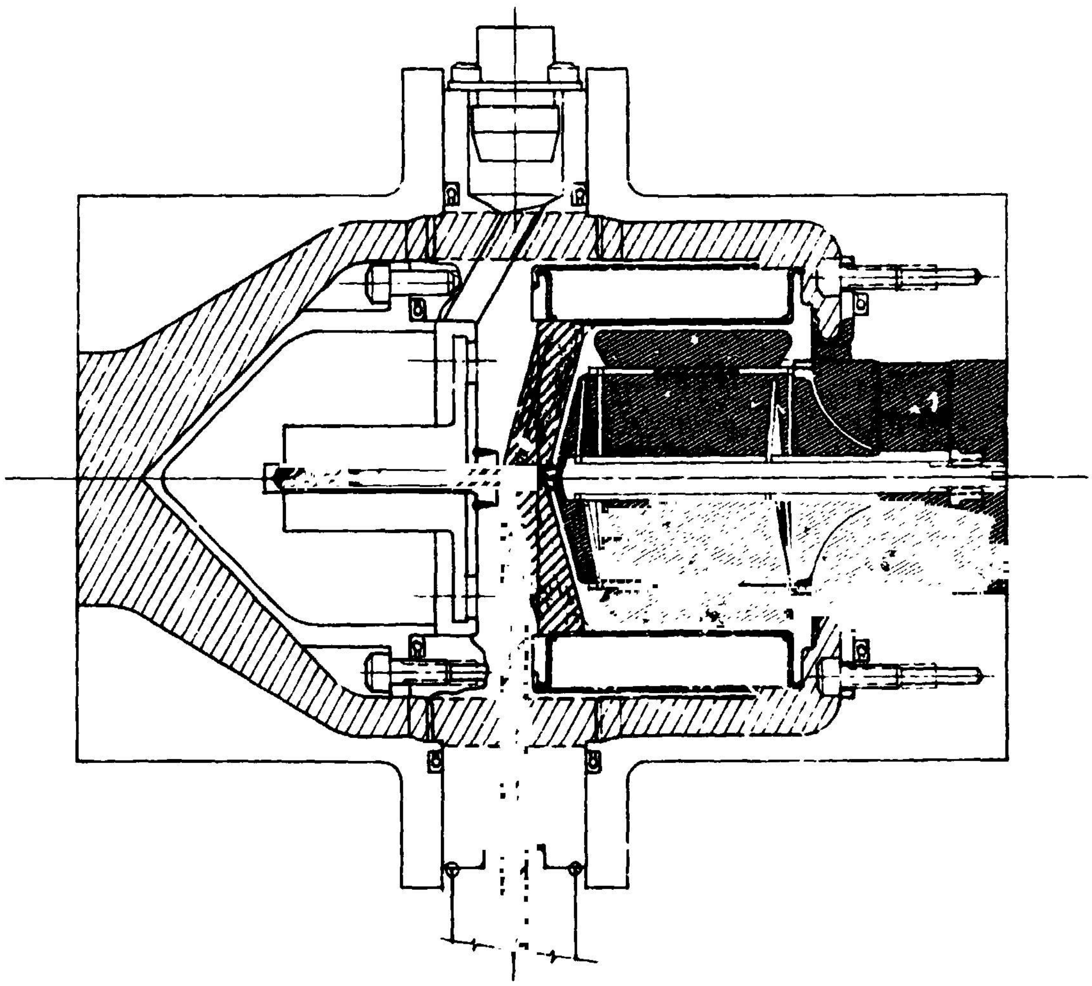


Figure 60 - Coaxial Poppet Valve - Vent Open/Pressure Close

This valve concept, as well as the double pressure-actuated valve concept discussed in the following section entitled, "Double Pressure Actuated Concept," features an LVDT transducer for the monitoring of valve position. The transducer armature is located inside the actuation volume and is attached to the moving poppet. This concept, as well as the other two valve concepts, also features a coaxial design -- meaning inlet and outlet are in line -- as evident from Figures 60, 61, and 62. Right angle configurations of essentially the same actuation and sealing concept were also reviewed, but since no particular advantage weight-wise or otherwise could be found, were not pursued any further. The particular valve concept shown in Figure 60 was subsequently modified to include a polyimide seat and was successfully cycled over one million cycles during the Advanced Technology for Space Shuttle Auxiliary Propellant Valves Program.

Double Pressure Actuated Concept

As the name implies, the double pressure actuated valve concept features a piston actuator which is alternately pressurized and vented on both sides to obtain high pneumatic actuation forces. These high pneumatic actuation forces, coupled with the pressure balanced poppet, result in an exceptionally fast response characteristic. Pressure actuation is accomplished by means of two externally located three-way pilot valves which control each side of the pneumatic piston actuator.

As shown in Figure 61, the valve is in the open position with flow from the left to the right. This valve is again shown with the gold-plated lip seal which was discussed in the preceding section. Pressure balancing of the poppet is accomplished by means of redundant sliding teflon-jacketed seals located at the same effective diameter as the effective seating diameter. Thus, there are never any pressure forces acting on the poppet. Sealing closure interface forces are generated by means of a coil spring which is located concentrically with the poppet and which also aids the closing motion. The poppet is connected to the piston actuator through a shaft arrangement and a small shaft seal prevents the piston open pressure from leaking into the valve downstream cavity. The shaft seal as well as the piston actuator seals is also of the teflon-jacketed configuration.

The redundant teflon-jacketed seals used for pressure balancing constituted a new seal design concept which was intended to minimize potential low temperature leakage problems encountered with radial teflon-jacketed seals of this type. The principle of operation of these redundant seals was to locate both the stationary and the sliding interfaces of the seal on the inner diameter. Thus, there is a tendency of the seal to shrink down to exert greater forces at the sealing surfaces as the temperature is reduced. Since the actuation velocity of this particular valve was very high, all teflon-jacketed seals utilized bronze impregnated teflon rather than pure teflon. The bronze impregnated

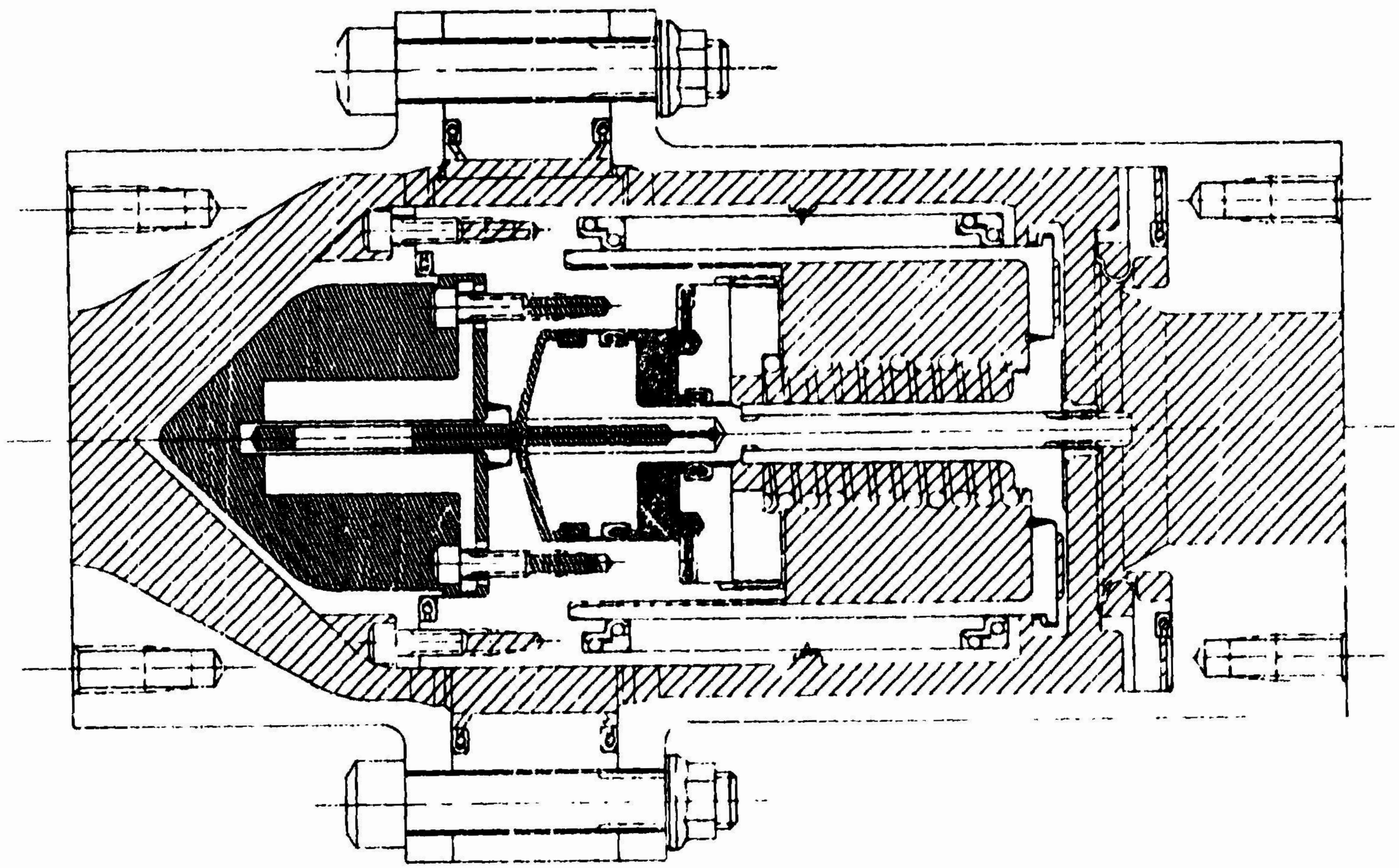


Figure 61 - X27449 Coax Valve Design Details

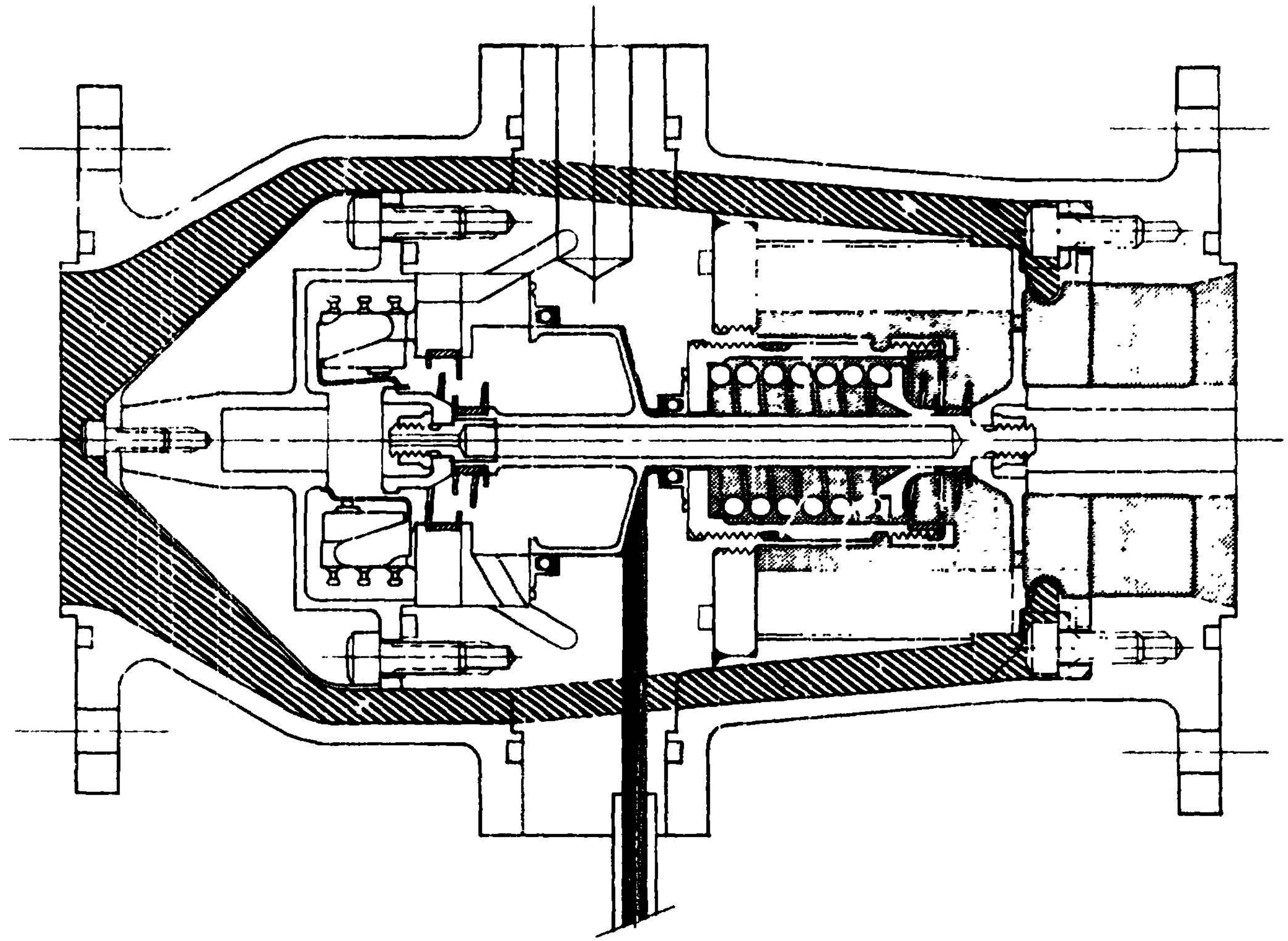


Figure 62 - Coaxial Poppet Valve - Pressure Ope :/Spring Close

teflon was previously demonstrated to feature the best wear characteristics. Several static seals were also employed in this valve. All static seals were of the face-seal type and utilized pure teflon jackets.

To monitor the position of the poppet, the valve includes a LVDT transducer. The armature of this transducer is located in the closing actuation volume of the piston actuator and is attached directly to the piston actuator. This particular valve design also included one other unique design feature. This was a poppet self-alignment flexure. A need to include such a flexure was determined during a tolerance stack-up when it became apparent that the guidance of the poppet cylinder had to allow for 0.008 inch (.02032 cm) radial clearance to assure that the poppet would operate freely over the wide temperature range. This clearance also resulted in the potential misalignment between the poppet sealing surface and the seat of approximately 0.008 inch (.02032 cm). Since the deflection capability of the seat was only 0.006 inch (.01524 cm), it was considered desirable to include a flexural element into the poppet such that the poppet surface could self-align with the seat. This flexural element is evident in Fig 6 where it appears as a thin, diaphragm-like section of the poppet surf

Pressurize-to-Open Concept

This particular concept is shown in Figure 62 . As the name implies, a single three-way solenoid valve located externally to the main valve provides valve upstream propellant supply pressure to one side of a pneumatic piston to open the valve and vents this pressure overboard to permit spring forces to return the valve to the closed position. This valve is also shown in the closed position and flow occurs from left to right.

To minimize actuation force requirements, the poppet has been pressure balanced as in the case of the valve presented in Figure 61. However, in this concept the pressure balancing dynamic seal is a bellows assembly rather than a sliding seal. The seat shown is again the gold-plated lip seal in combination with a stop.

Poppet/actuator piston shaft guidance is accomplished by means of metallic axial guidance flexures located near the left and right extreme ends of the shaft assembly. Sealing of the actuation cavity is performed by means of two dynamic sliding teflon-jacketed seals. In place of the LVDT position transducer shown in the other valve concepts, this valve concept employs two mechanical micro switches, one for the open position and one for the closed position. Materials of construction intended for this concept were similar to those of the other concepts.

In comparing the pressurize-to-open concept with the other two valve concepts under consideration, it was determined that this particular concept was somewhat of an in-between arrangement featuring neither the simplicity and high reliability of the concept shown in Figure 60 nor the high response characteristics of the concept shown in Figure 61. Since the program scope included detail design, fabrication and test evaluation of two valve concepts, it was decided not to pursue this particular valve concept past the preliminary design stage as presented in Figure 62.

VALVE SUBCOMPONENT EVALUATION

Activities performed in support of this section included the completion of fabrication of the P/N X27449 and X28400 Test Fixtures and the subsequently performed valve cycling program. Valve cycling was accomplished at hot, ambient, and cold conditions and included periodic measurements of responses, leakage, and pressure drop. Upon completion of the valve cycling program, test fixtures were disassembled, closely examined and photographed to evaluate the effects of the test program on the various valve components. The test data was then summarized and presented to the NASA-LeRC Technical Project Manager at a Design Review at which time the final flightweight valve configuration was selected.

Test Fixture Fabrication

P/N X27449 - Test Fixture

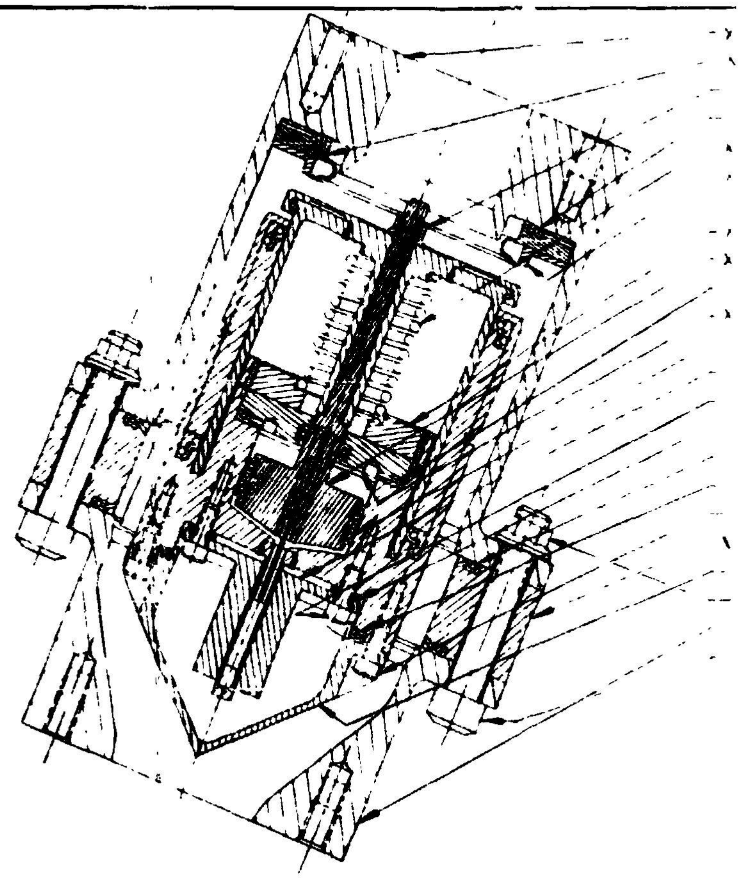
Fabrication of various elements of the P/N X27449 Test Fixture, shown in Figure 63, was primarily accomplished in the Marquardt-Van Nuys Model Shop, employing the development fabrication system. Inspection was performed and inspection records of critical characteristics were maintained in accordance with engineering instructions in the log book of the Test Fixture. In the case of critical dimensions, actual dimensions measured were recorded rather than simply the fact that the particular dimension was within the tolerances specified. Liaison engineering performed all expediting through the shop as well as the necessary coordination with outside vendors. Disposition of discrepant hardware was the joint responsibility of the Liaison Engineer and the Project Engineer.

A photograph of the completed P/N X27449 Valve is shown in Figure 64 and an exploded view showing the various valve components is shown in Figure 65. Purchased parts included in this assembly were the electrical connector, LVDT transducer, normally open and normally closed pilot valves, seven static teflon seals and five sliding dynamic teflon seals, the poppet return spring, and various nuts and bolts required to make the assembly. In addition, such processing as lapping, gold-plating, and electrolyzing was subcontracted. Because of the criticality of the gold-plating and the lapping of the poppet, special care was taken by Marquardt in making certain that the vendor fully understood the function and the criticality of the particular component. This attention to detail paid off in the case of the poppet, since this component functioned well throughout the cycle program; however, considerable difficulties were incurred with the gold-plating as explained later in the section describing the testing of this valve assembly.

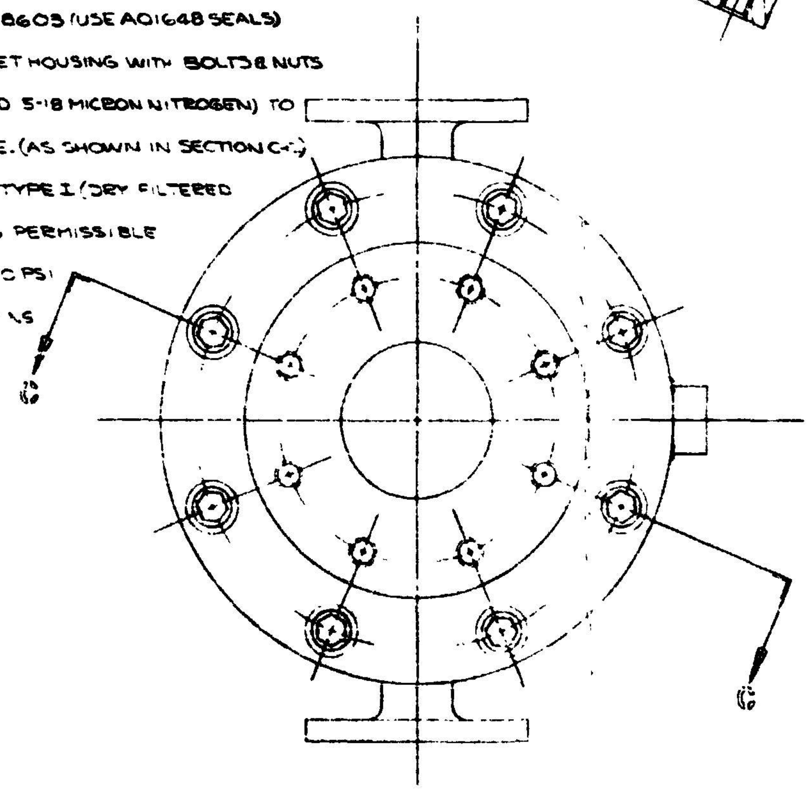
Close-up photographs of some of the valve details which were subject

NOTES: UNLESS OTHERWISE SPECIFIED

1. ALL TORQUE VALUES ARE IN ADDITION TO RUNNING TORQUE
2. TORQUE SCREWS TO 20-25 IN LB
3. TORQUE SCREWS TO 3-4 IN LB.
4. TORQUE SCREW/NUT TO 10-12 IN LB.
5. TORQUE NUT TO BOTTOM DISC ON INNER-BODY. TACK WELD APPROX .06 DIA ON THREADS TO LOCK NUT TO INNER-BODY PER MPS 1601 CLASS B. (USE TOOL T18601)
6. TORQUE BOLT/NUT TO 250-270 IN LB
7. LUBRICATE THREADS AND BEARING SURFACES WITH KRYTOX 143AC LUBRICATE PER MPS 1103
8. CLEANLINESS REQUIRED PER MPS 210
9. TRIM WIRES FROM TRANSDUCER LEAVING THE MAX LENGTH POSSIBLE. SOLDER WIRES IN RECEPTACLE PER WIRING DIAGRAM WITH Pb5/Sn95 SOLDER PER QQ-S-571 TYPE SMA. POT SOLDERED WIRE CONNECTIONS IN RECEPTACLE WITH EPOXY EPX 400
10. TORQUE NUT TO 40-50 IN LB
11. FLOW FITTINGS T18602 AND T18603 (USE A01648 SEALS)
12. TO INSTALL INLET HOUSING & OUTLET HOUSING WITH BOLTS & NUTS APPLY 100±20 PSI (DRY FILTERED 5-18 MICRON NITROGEN) TO INLET PORT OF N.C PILOT VALVE. (AS SHOWN IN SECTION C-C)
13. PRESSURE CHECK PER MPS 1300 TYPE I (DRY FILTERED 5-18 MICRON NITROGEN, NO LEAKS PERMISSIBLE. CHECK LEAKS WITH SNOOP 6500 PSI FOR 10 MINUTES. USE END FITTINGS T18602 WITH T18604 CAP

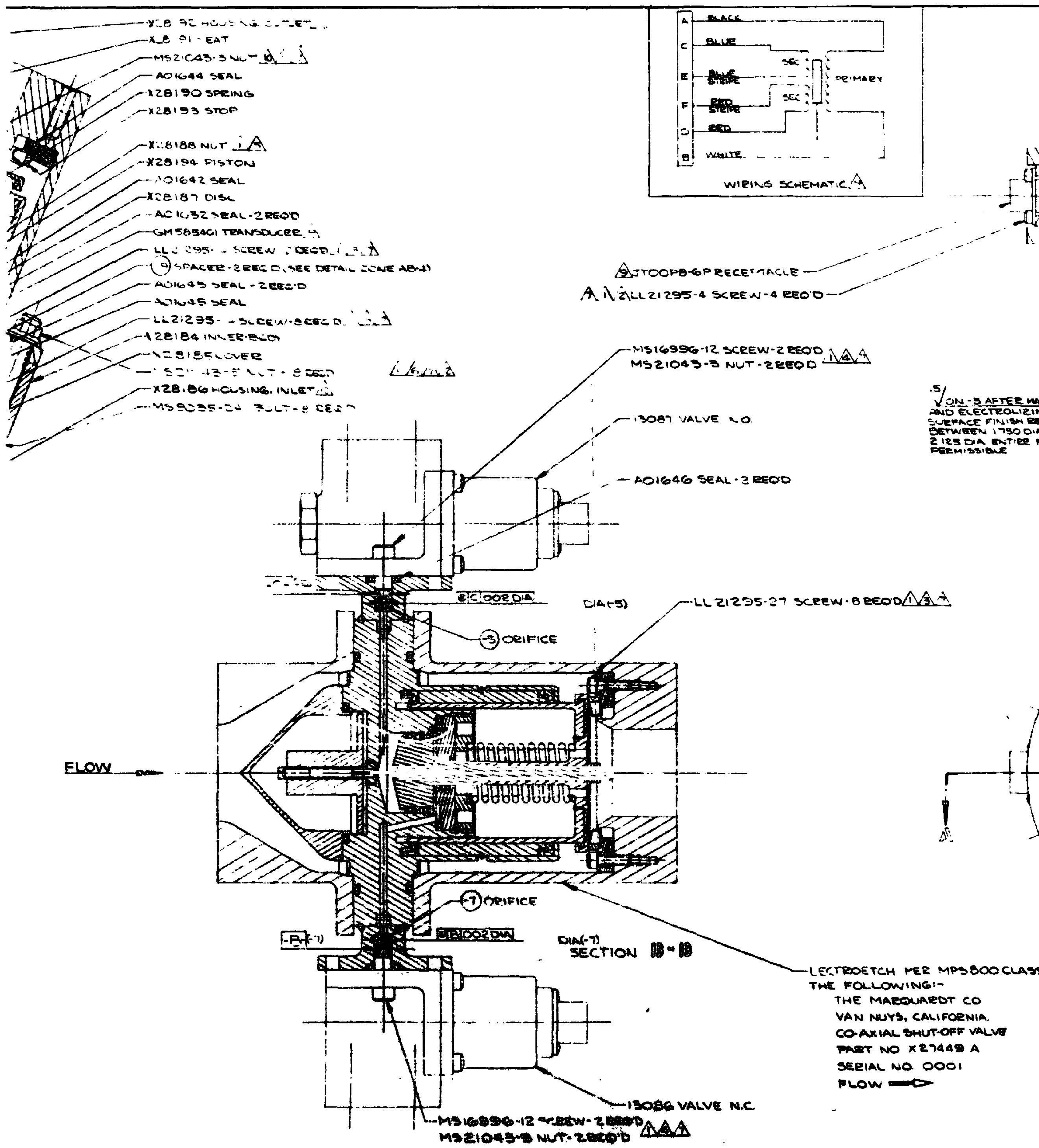


SECTION C-C
SHOWN OPEN



FLOW

FOLDOUT FRAME



FOLDOUT FRAME 2

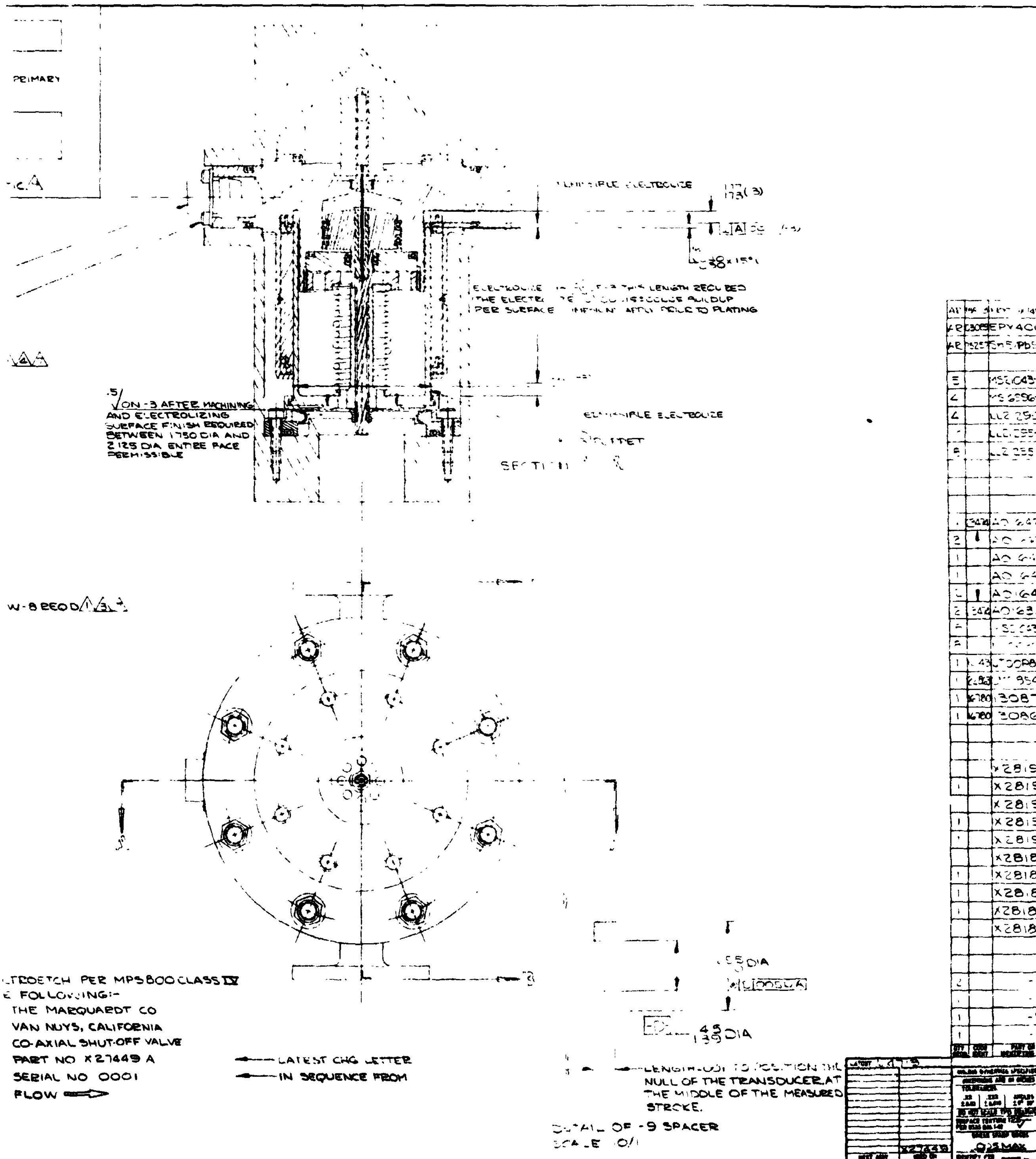
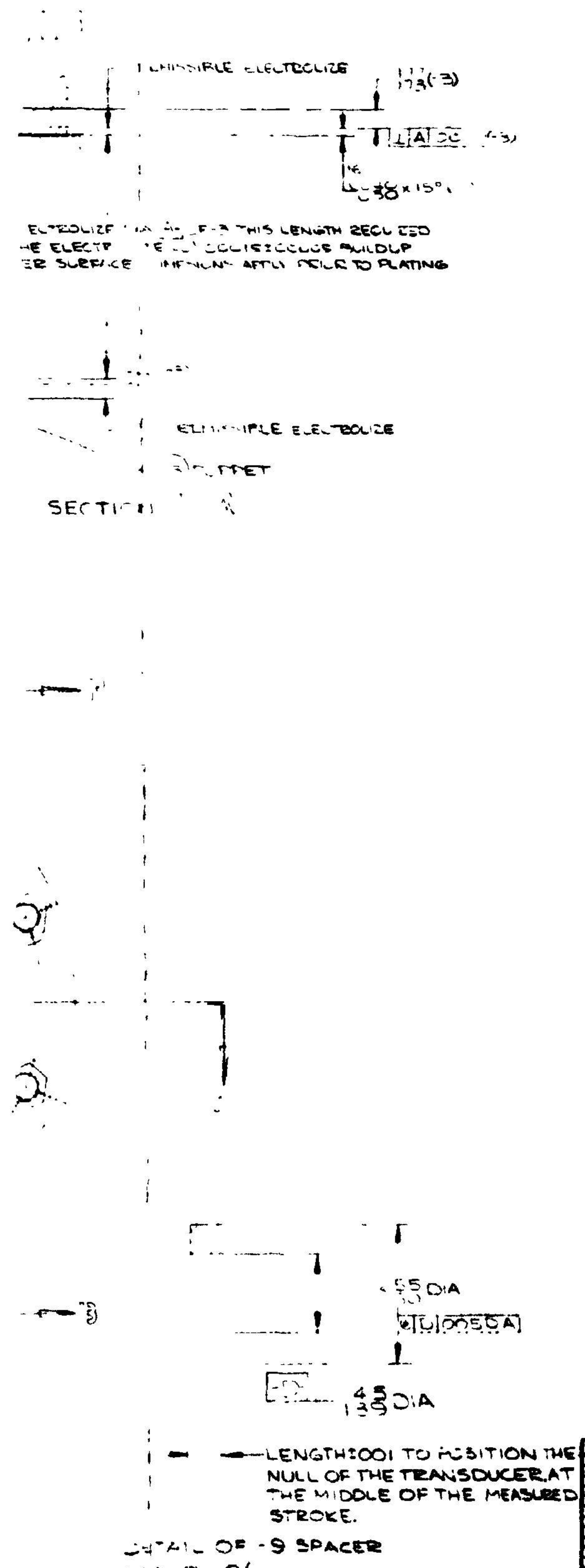


Figure 63

FOLDOUT FRAME 3

ZONE	LTR	DESCRIPTION	DATE / APPROVED
	A	INITIAL DRAWING ISSUE	



AR	19873	KEYTON 143AC	LUBRICANT				42
AR	03085	EPY400	EPOXY			E6	41
AR	75257	Sn5/Pb95	SOLDER			F5	40
							39
5		MS21043-3	NUT			H11 E8	38
4		MS 6356-12	SCREW			F5 A9	37
4		LL212954	SCREW, KEL-F			F8	36
[illegible]		LL21295[illegible]	SCREW, KEL-F			G11 E	35
8		LL2 2957	SCREW, KEL-F			D8	34
							33
							32
							31
1	33474	AO 642	SEAL, FACE			G	30
2	↓	AO1643	SEAL, FACE			F11	29
1		AO1644	SEAL, FACE			H11	28
1		AO1645	SEAL, FACE			F11	27
2	↓	AO1646	SEAL, FACE			E8	26
2	33474	AO1632	SEAL			G1	25
5		MS21043-5	NUT			F11	24
8		[illegible]3524	BOLT			F11	23
1	02143	UTOOP86P	RECEPTACLE			G8	22
1	24863	UMF85401	TRANSDUCER			G1	21
1	16780	13087	VALVE NO			F8	20
1	16780	13086	VALVE NC			A9	19
							18
							17
1		X28194	PISTON			G1	16
1		X28193	STOP			H11	15
1		X28192	HOUSING, OUTLET			H11	14
1		X28191	SEAT			H11	13
1		X28190	SPRING			H11	12
1		X28188	NUT			G11	11
1		X28187	DISC			G11	10
1		X28186	HOUSING, INLET			F11	9
1		X28185	COVER			F11	8
1		X28184	INNER-BODY			F11	7
							6
							5
2		-9	SPACER	AMS5654	CRES 347	A4 B11	4
1		-7	ORIFICE NC	MAKE FROM	X28196	B9	3
1		-5	ORIFICE NO	MAKE FROM	X28196	D9	2
1		-3	POPPET	MAKE FROM	X28189	E4	1
QTY REQD	CODE IDENT	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL	ZONE	ITEM NO.

LIST OF MATERIALS OR PARTS LIST

The Marquardt Company — CCI Aerospace Corporation, Van Nuys, California

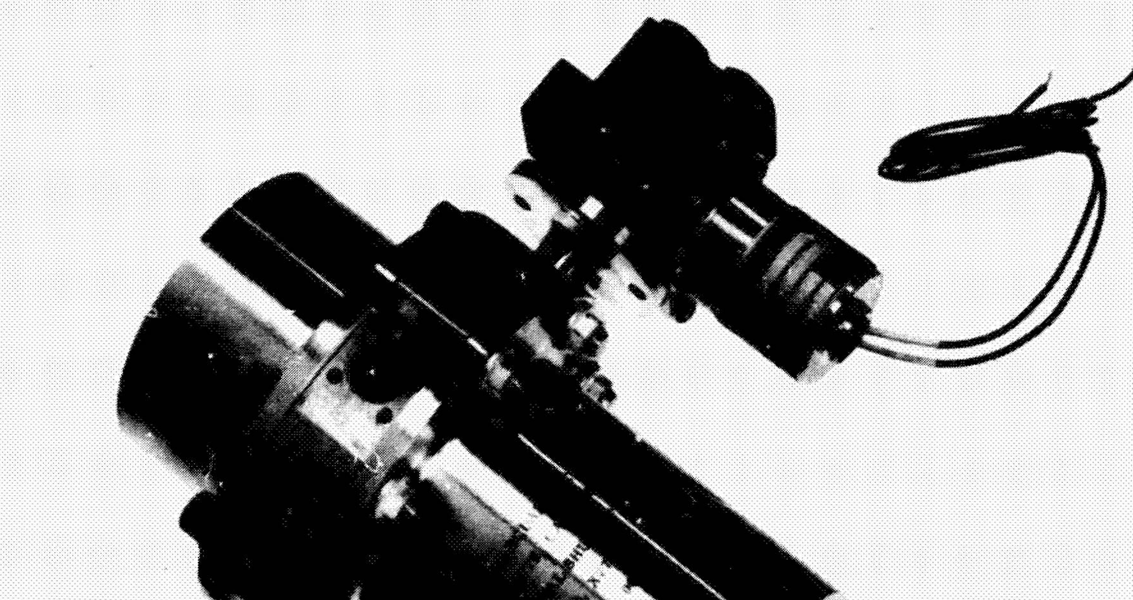
VALVE ASSEMBLY, CO-AXIAL SHUT-OFF

J | 88945 | X27449

SCALE 2/1 | SHEET 1 OF 1

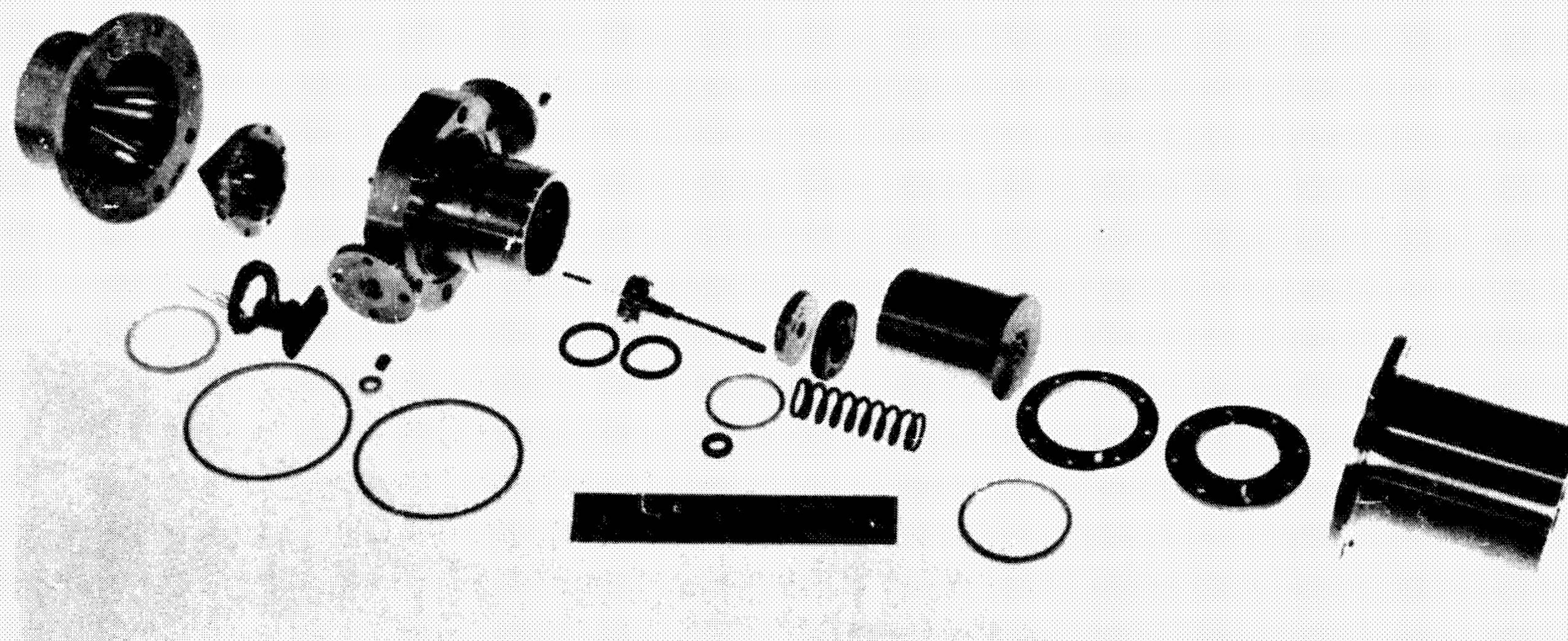
Figure 63

FOLDOUT FRAME 4



(0 2.54 5.08 7.62) cm

Figure 64 - P/N X27449 Valve Assembly



(0 2.54 5.08 7.62 10.16 12.7 15.24)
cm

Figure 65 - Coax Valve Components, P/N X27449

to wear during the test cycle program are shown in Figure 66 and Figure 67. These photographs show the poppet/seat interface, the poppet pressure balancing cylinder which is used to guide the poppet, the redundant seals, piston seals, and rod seal as well as the piston rod which is subjected to sliding friction. In general, the fabrication of P/N 27449 Valve Assemoly proceeded quite smoothly. Examples of some minor problems which were encountered during the manufacture are as follows: As shown in Figure 66 the poppet face features a granular area extending from approximately 30% to 80% of the diameter. This granular area is a slight recess of the poppet surface and this recess was added by electric discharge machining after it was found extremely difficult to lap the entire poppet surface to a one helium light band flatness. The difficulty in obtaining this flatness was due to the fact that the poppet surface is relatively thin as specified by the design drawings to compensate for out-of-parallel conditions between the poppet surface and the seat. Another example of a minor manufacturing problem is the fact that after the poppet rod had been welded to the main poppet by means of electron beam welding, it was determined that the rod was significantly more out-of-perpendicular with respect to the poppet face than had been allowed for with a clean-up cut. Consequently, the rod had to be turned down farther and a spacer sleeve installed over the rod to reattain the specified rod dimensions. Some difficulties were also encountered in the installation of the dynamic rod and piston seals. A tapered tool was fabricated to gradually cause expansion of the piston seals as they were forced over the piston ends and into the piston grooves. Initial attempts to force the rod seal into the groove in the seal retainer resulted in damage to the rod seal on two successive tries. It was then decided to make the rod seal retainer a two-part assembly such that the rod seal could be easily dropped into the groove prior to accomplishing the press fit of the two halves of the rod seal retainer.

The final assembly of the test fixture was accomplished in the Clean Room of Building 32 at the Marquardt-Van Nuys facility. Individual valve parts were cleaned per Marquardt Process Specification No. 210 prior to final assembly. Upon completion of the assembly, the various static and dynamic seals in the test fixture were leak checked with 450 psi (310 N/cm^2) helium to verify that the test fixture was ready for acceptance testing. A 25 μ absolute inlet filter was then installed upstream of the test fixture and the unit was transferred to Building 37 for acceptance testing.

P/N X28400 - Test Fixture

The P/N X28400 Test Fixture, shown in Figure 68, also was fabricated. Fabrication approach and controls were identical to those exercised during the fabrication of the P/N X27449 Test Fixture. A photograph of the completed P/N X28400 Valve is shown in Figure 69 and an exploded view

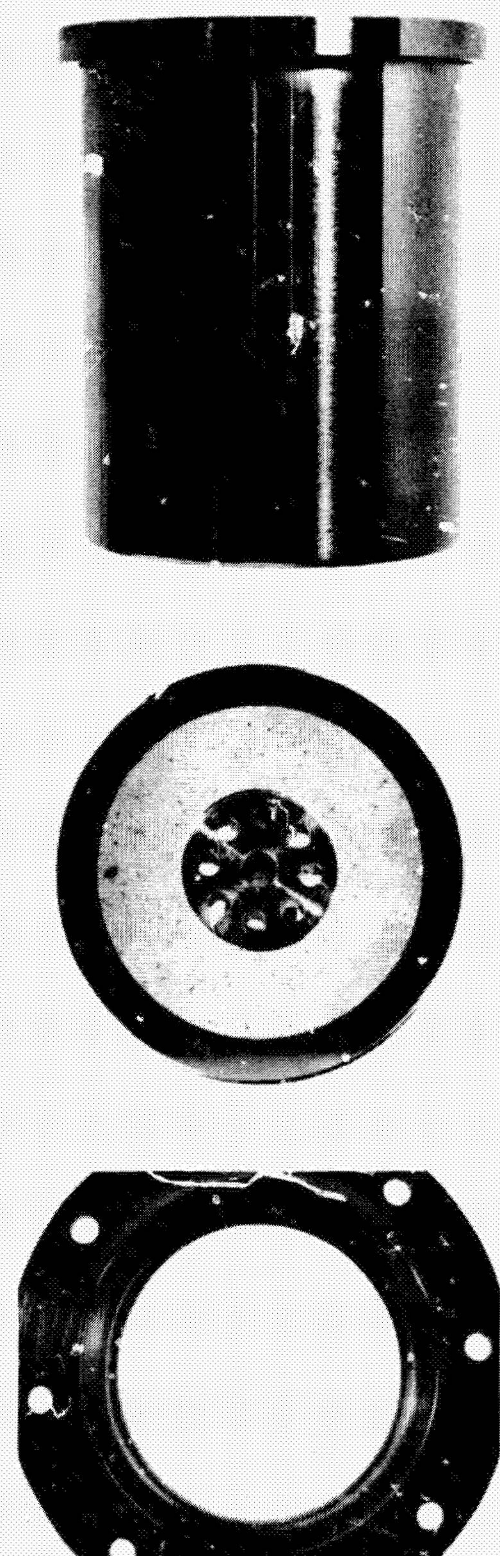
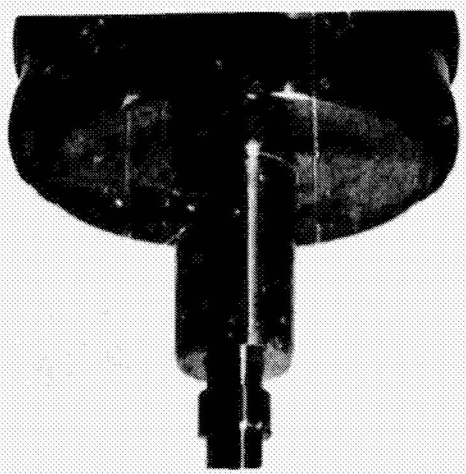
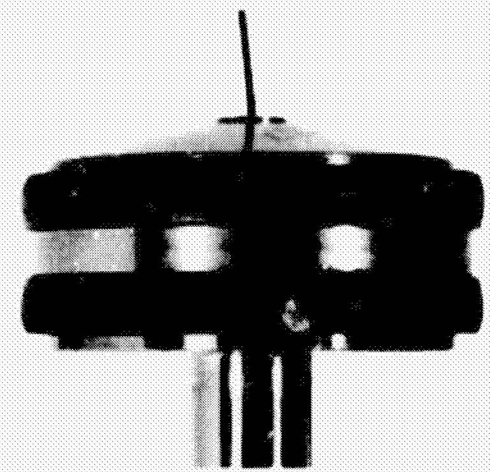


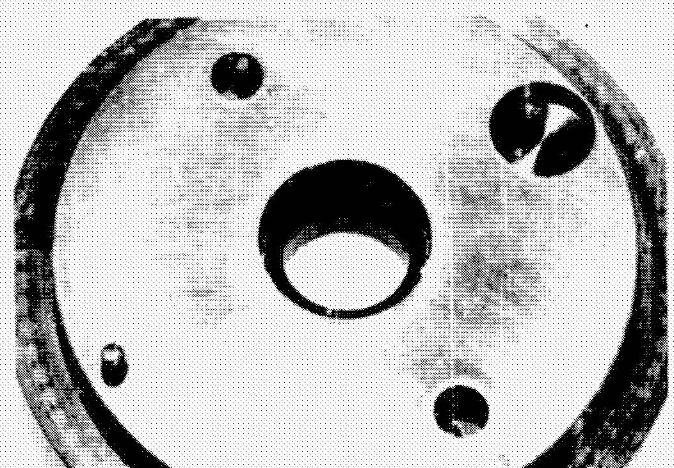
Figure 66 - Poppet and Seat Details, P/N X27449



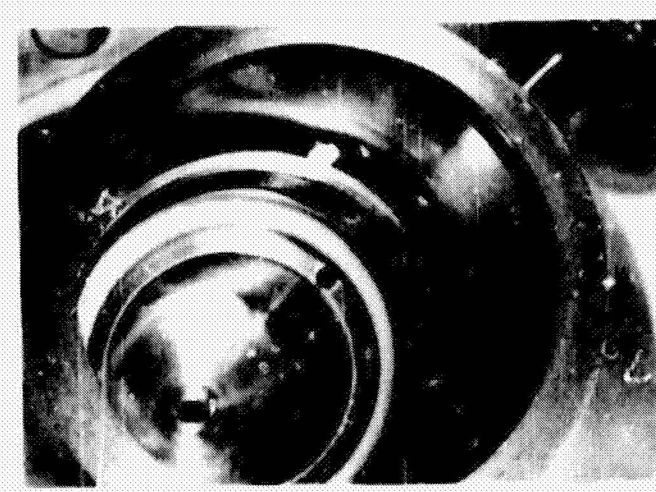
VALVE ASSEMBLY P/N 27449
PISTON SHAFT AND SLIDING SEALS



VALVE ASSEMBLY P/N 27449
PISTON SLIDING SEALS



VALVE ASSEMBLY P/N 27449
PISTON SHAFT SEAL

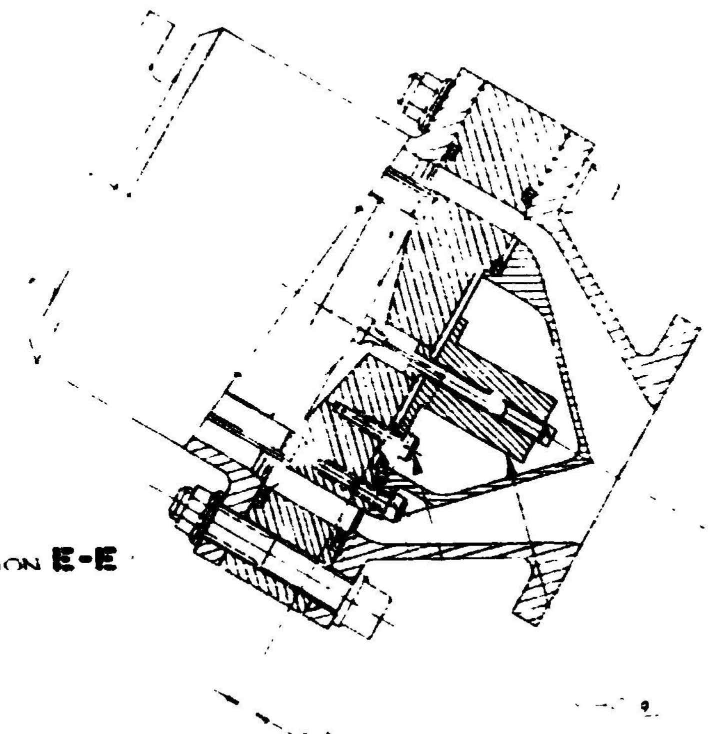


VALVE ASSEMBLY P/N 27449
INNER BODY POPPET SEAL

F[illegible]e 6 [illegible]Actu[illegible] De[illegible]

NOTES: UNLESS OTHERWISE SPECIFIED

1. ALL TORQUE VALUES ARE IN ADDITION TO RUNNING TORQUE
2. TORQUE SCREWS TO 2.0 2.5 IN LBS
3. TORQUE SCREWS TO 3-4 IN LBS
4. TORQUE SCREWS TO [illegible]-12 IN LBS
5. TORQUE BOLT/NUT TO 250-270 IN LBS
6. TORQUE NUT TO 34-38 IN LBS
7. CLEANLINESS REQUIRED PER MPS 210
8. LUBRICATE THREADS AND BEARING SURFACES WITH ITEM 28 PER MPS 1103
9. TRIM WIRES FROM TRANSDUCER LEAVING THE MAX LENGTH POSSIBLE. SOLDER WIRES IN RECEPTACLE (ITEM 21) PER WIRING DIAGRAM WITH ITEM 26 PER QQ-S-571 [illegible] TYPE RMA. POT SOLDERED WIRE CONNECTIONS IN RECEPTACLE (ITEM 21) WITH EPOXY ITEM 27
10. PRESSURE CHECK PER MPS 1500 TYPE I (DRY FILTERED 5-8 MICRON NITROGEN, NO LEAKS PERMISSIBLE. CHECK LEAKS WITH SNOOP GSC 2 10 PSI FOR [illegible] MINUTES. USE [illegible] FITTING [illegible] TYP BOTH ENDS.

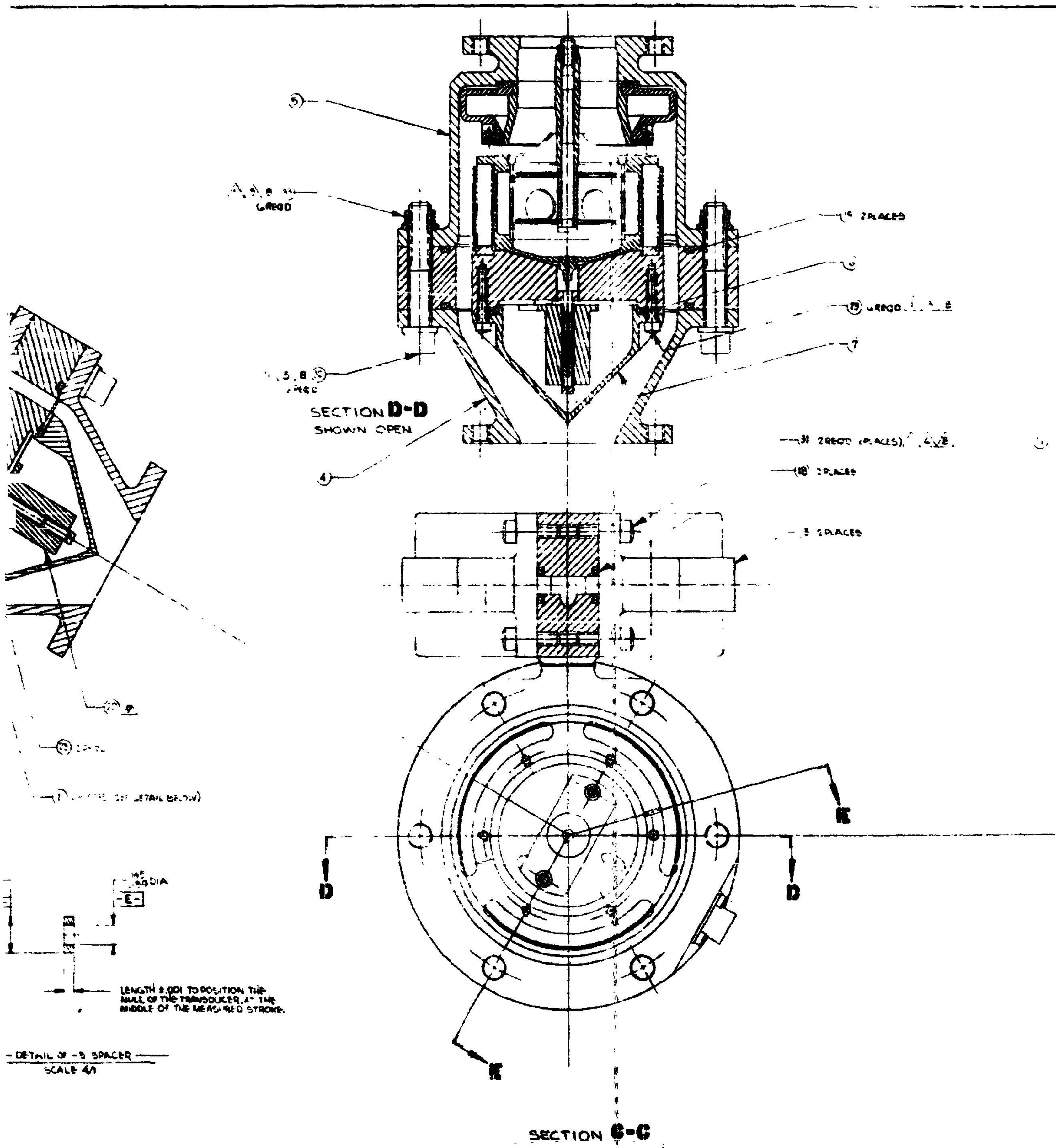


SECTION E-E

DETAIL [illegible] SPACER

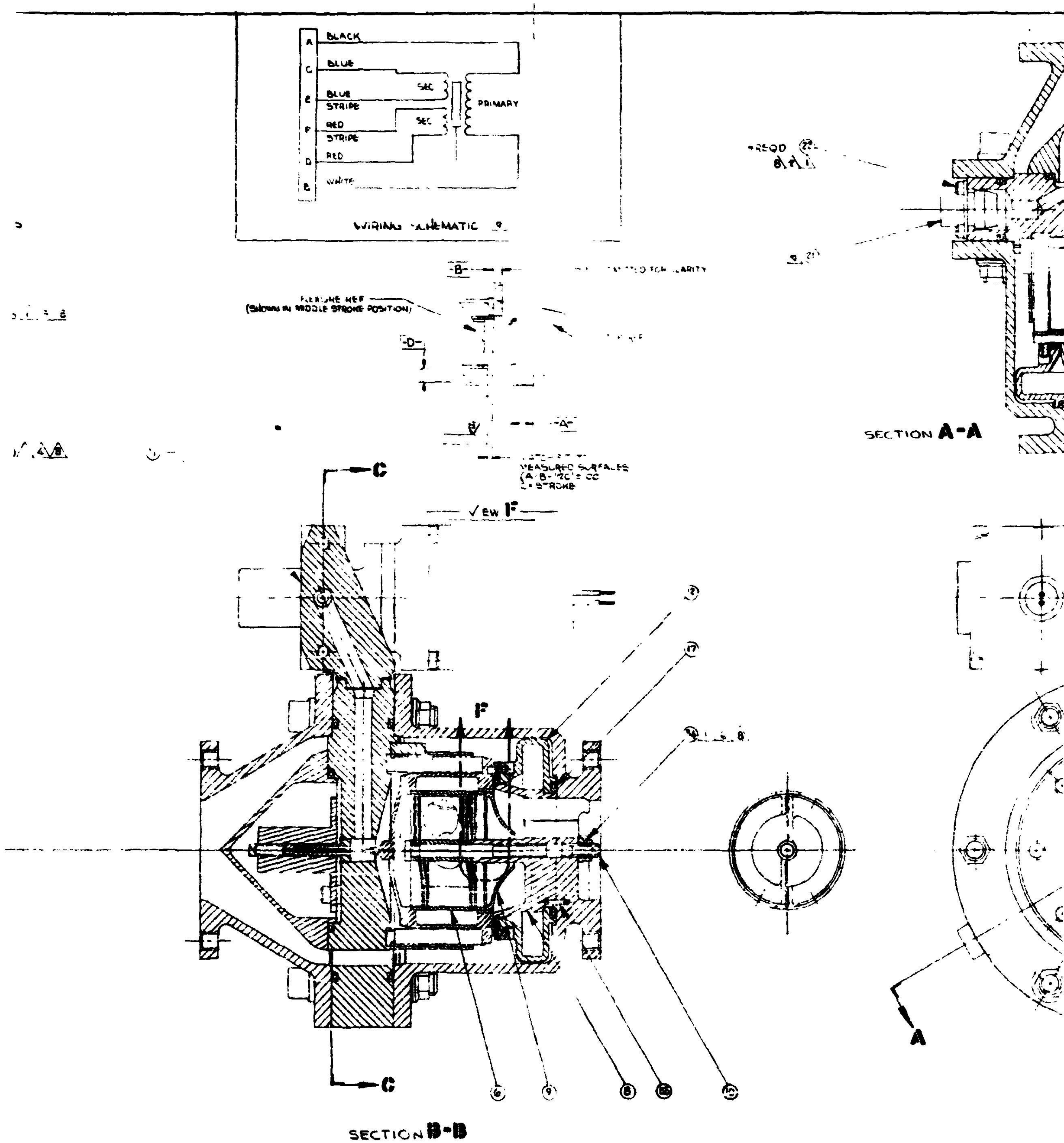
SCALE 4/1

FOLDOUT FRAME



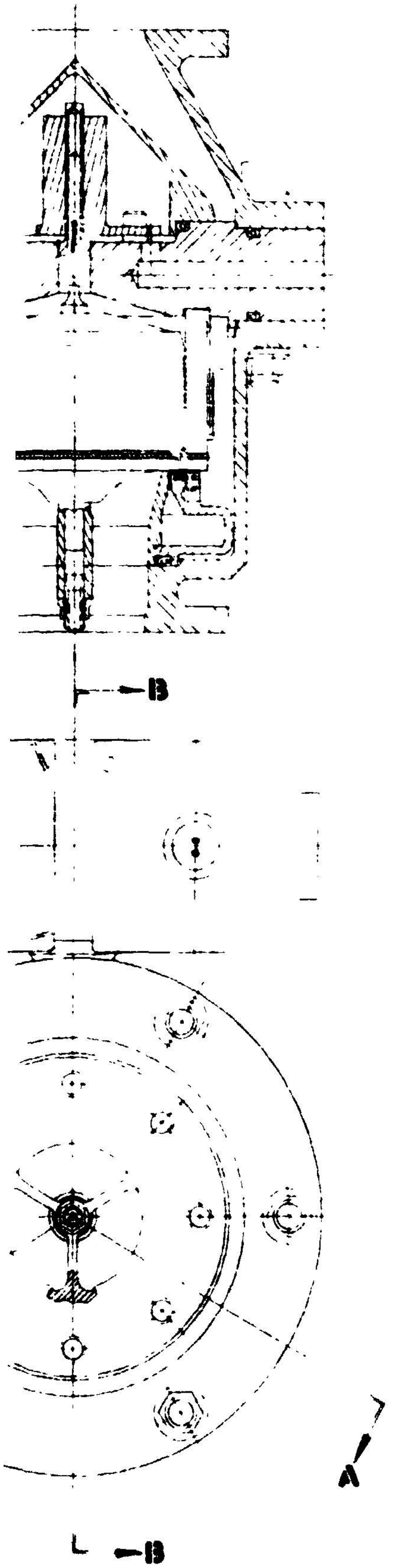
FOLDOUT FRAME

REPRODU



FOLDOUT FRAME

LTR	DESCRIPTION	DATE	APPROVED
A	INITIAL DWG RELEASE		



QTY REQD	CODE IDENT	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	SPECIFICATION	MATERIAL OR NOTE	ZONE	ITEM OR FIND NO.
		MS 3555-602	PIN				
		MS2 043-3	NUT				
6		MS2 043-3	NUT				
6		MS9035-24	BOLT				
4		MS 9996-2	SCREW				
AS REQD		KRYTOX 43AC	LUBRICANT				
AS REQD	09090	EPY400	EPOXY			E20	
AS REQD	75297	Sn5/Pb95	SOLDER			E20	
8		LL2 295-4	SCREW, SELF			C6 F12	23
4		LL2 295-4	SCREW, SELF			G8	22
	2 49	JT00P8-6P	RECEPTACLE			G8	2
1		GM985401	TRANSDUCER			D6	20
2	28474	AO 684	SEAL			G12	19
2		AO1646	SEAL			E 2	18
1		AO1645	SEAL			D8	
	28474	AO1644	SEAL			F 2	
2	730	1D090	VALVE PILOT	(NORM OPEN)		E 2	13
1		X28397	SEAT ASSY, POPPET			F 8	12
1		X28396	BODY ASSY, INNER			E 11	11
1		X28390	BOLT			A8	10
1		X28389	NUT			A9	9
1		X28388	STOP, POPPET			A8	8
1		X28385	COVER			F 2	7
1		X28380	FLEXURE ASSY			A9	6
1		X28372	HOUSING, OUTLET			H15	5
1		X28371	HOUSING, INLET			E14	4
2		-3	SPACER	AMS 5654	CRES 347		1

LIST OF MATERIALS OR PARTS LIST

LAYOUT L 4734

APPLICATION

.030 MAX

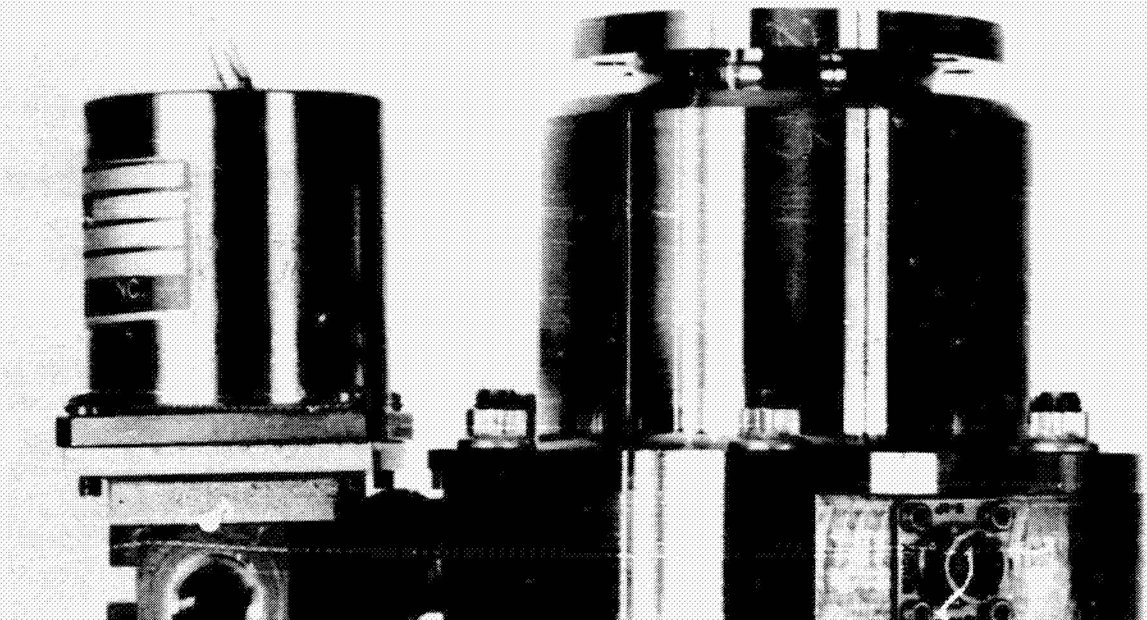
VALVE ASSEMBLY,
COAXIAL SHUT-OFF

J 88846 X28400

SHEET 1 OF 1

FOLDOUT FRAME 4

Figure 68



(0 2.54 5.08 7.62)

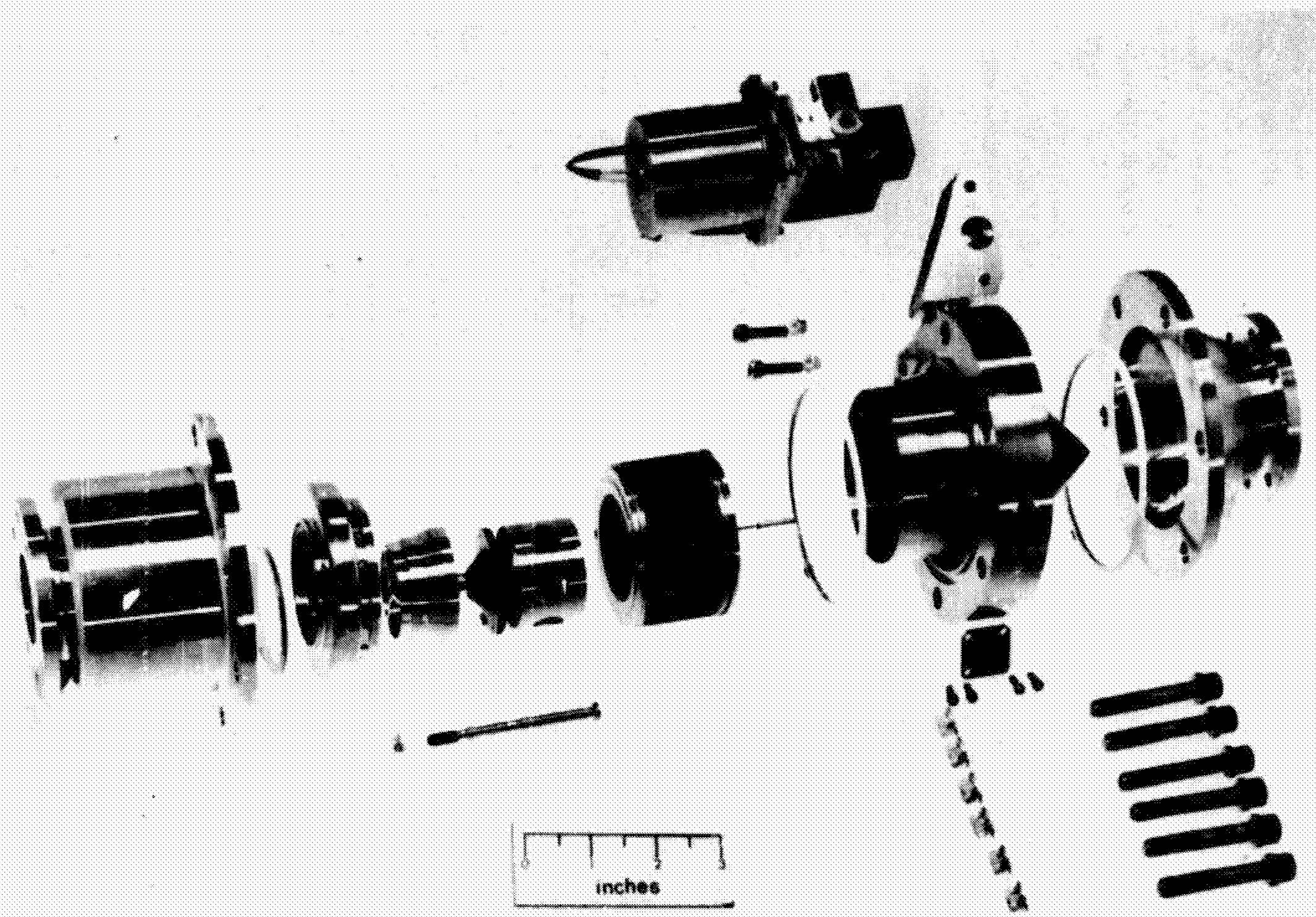
cm

Figure 69 - P/N X28400 Valve Assembly

showing the various valve components is shown in Figure 70. Purchased parts included in this assembly were the welded bellows, LVDT transducer, normally open pilot valve, five static teflon seals, and various nuts and bolts required to make the assembly. In addition, such processing as lapping of the poppet and seat, and the chem milling of the axial guidance flexure plates were contracted. Particular care was taken in coordinating the design and fabrication of the poppet/bellows assembly with the bellows vendor, since this component constituted the most critical component in the entire valve assembly. Bellows operating stresses were intentionally designed to low levels to enhance bellows life for the one million cycles. A photograph of the poppet/bellows assembly prior to final lapping of the poppet sealing surface is shown in Figure 71. The raised surface immediately adjacent to the internal threads constitutes the poppet sealing surface. A close-up of the seat and the poppet stop may be had in the photograph of the valve outlet housing shown in Figure 72. Examination of this photograph shows two narrow black rings at a diameter of approximately two inches (5.08 cm). The inner of these rings is a space between the poppet stop and the seat. The outer ring is the 0.015 inch (.0281 cm) wide polyimide sealing surface which has been lapped to a 2-AA finish. This view also shows the spider arrangement located in the outlet port which is used for holding the central valve shaft which supports the axial guidance flexure.

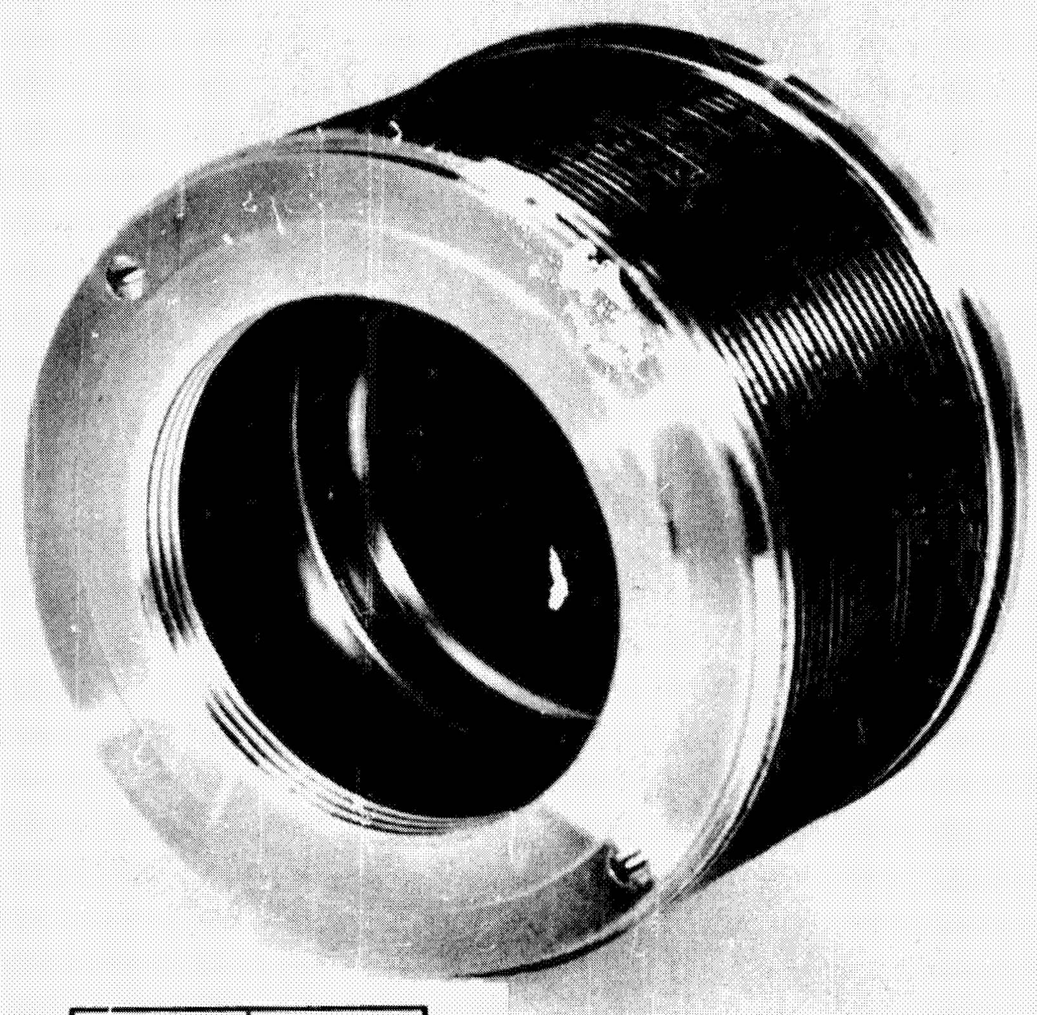
The fabrication of the P/N X28400 Valve Assembly proceeded very well. The only problem area noted was an inability on the part of the vendor who performed the chemical milling of the axial guidance plates to properly etch the patterns. However, this was traced to the use by the vendor of an old solution and by using a new etching solution, the vendor was able to produce excellent parts. Load versus deflection tests of the various spring elements in the P/N X28400 Valve were made and these are presented in Figure 73, 74, and 75. These data are for the bellows assembly, the axial guidance flexure assembly, and the seat, respectively. The data obtained agreed quite well with analytical predictions. The final deflection design points selected were a precompression of 0.051 inch (.127 cm) with a maximum compression of 0.10 inch (.4826 cm) for the bellows, over center operation of the axial guidance flexure to a maximum deflection of 0.07 inch (.1778 cm) and a compression of between 0.005 and 0.006 inch (.0127 and .01524 cm) of the valve seat.

The final assembly of the test fixture was accomplished in the Clean Room of Building 32 at the Marquardt-Van Nuys Facility. As with the P/N X27449 Valve, individual valve parts were cleaned per Marquardt Process Specification No. 210 prior to final assembly. Upon completion of the assembly, the various static seals and the main seat of the Test Fixture were leak checked with 450 psi (310 N/cm^2) helium to verify that the Test Fixture was ready for



(0 2.54 5.08 7.62)
cm

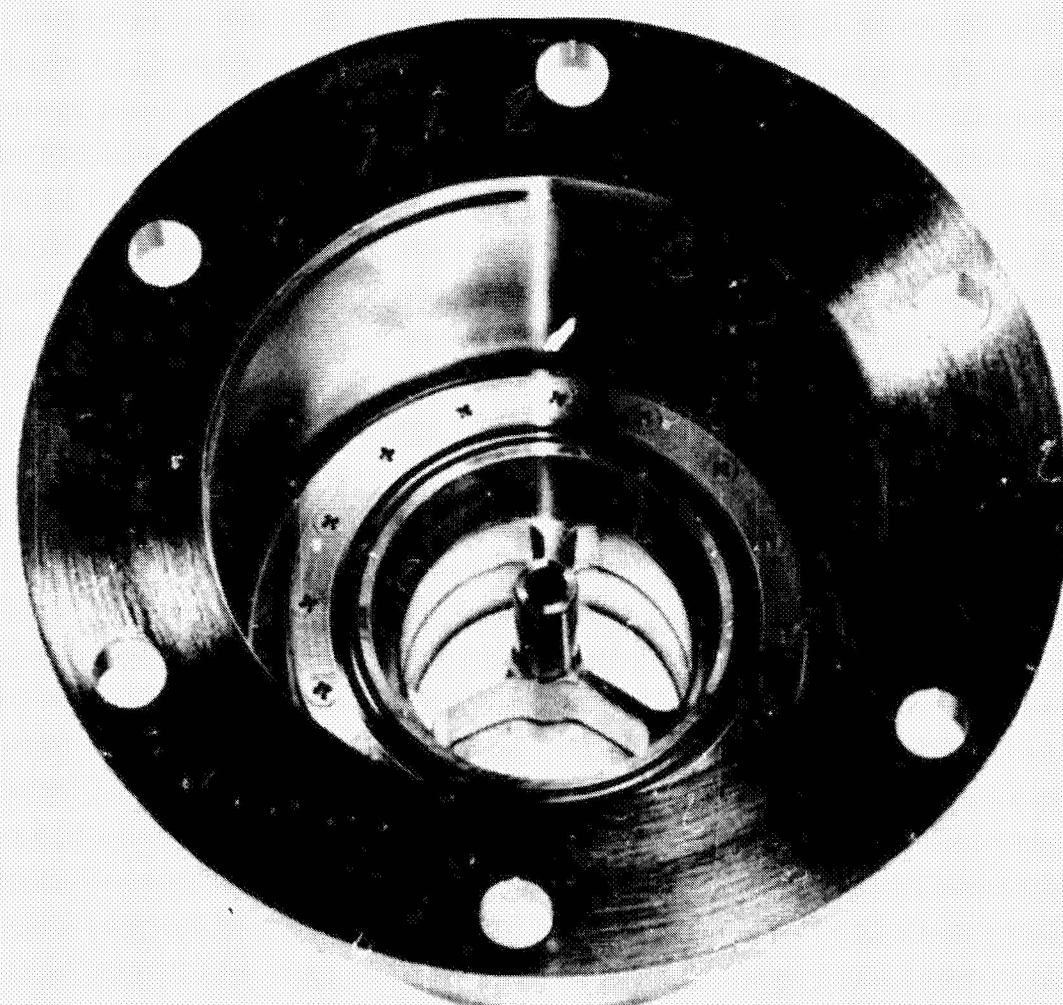
Figure 70 - P/N X28400 Valve, Exploded View



0 Inch 1

(0 cm 2.54)

Figure 71 - P/N X28400 Valve, Bellows Assembly, Downstream Side



0 Inch 1

(0 cm 2.54)

Fig 72 N X 00 O Ho In ing an ppe p

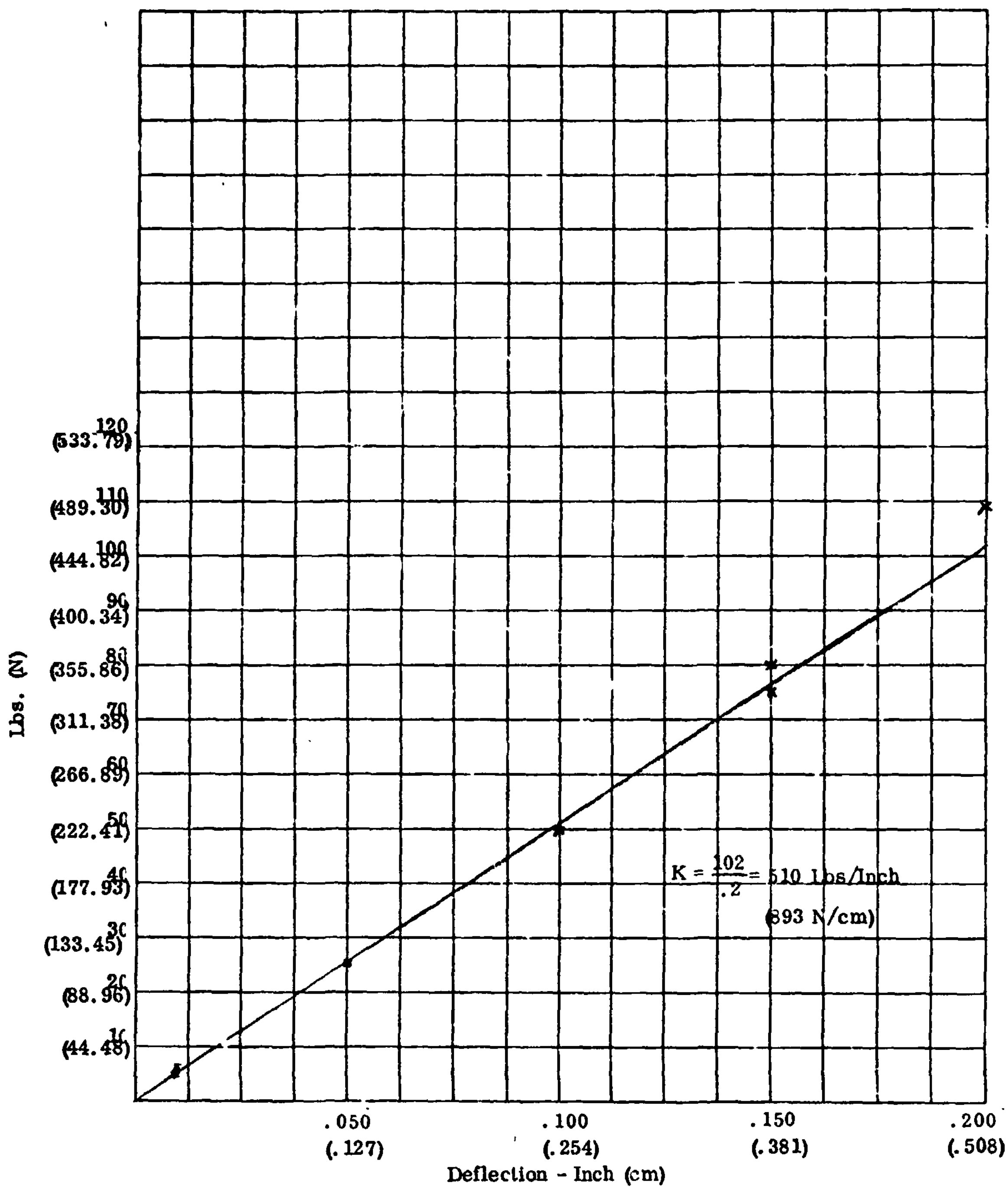


Figure 73 - Load Vs. Deflection - Bellows Assembly P/N X28400 Valve

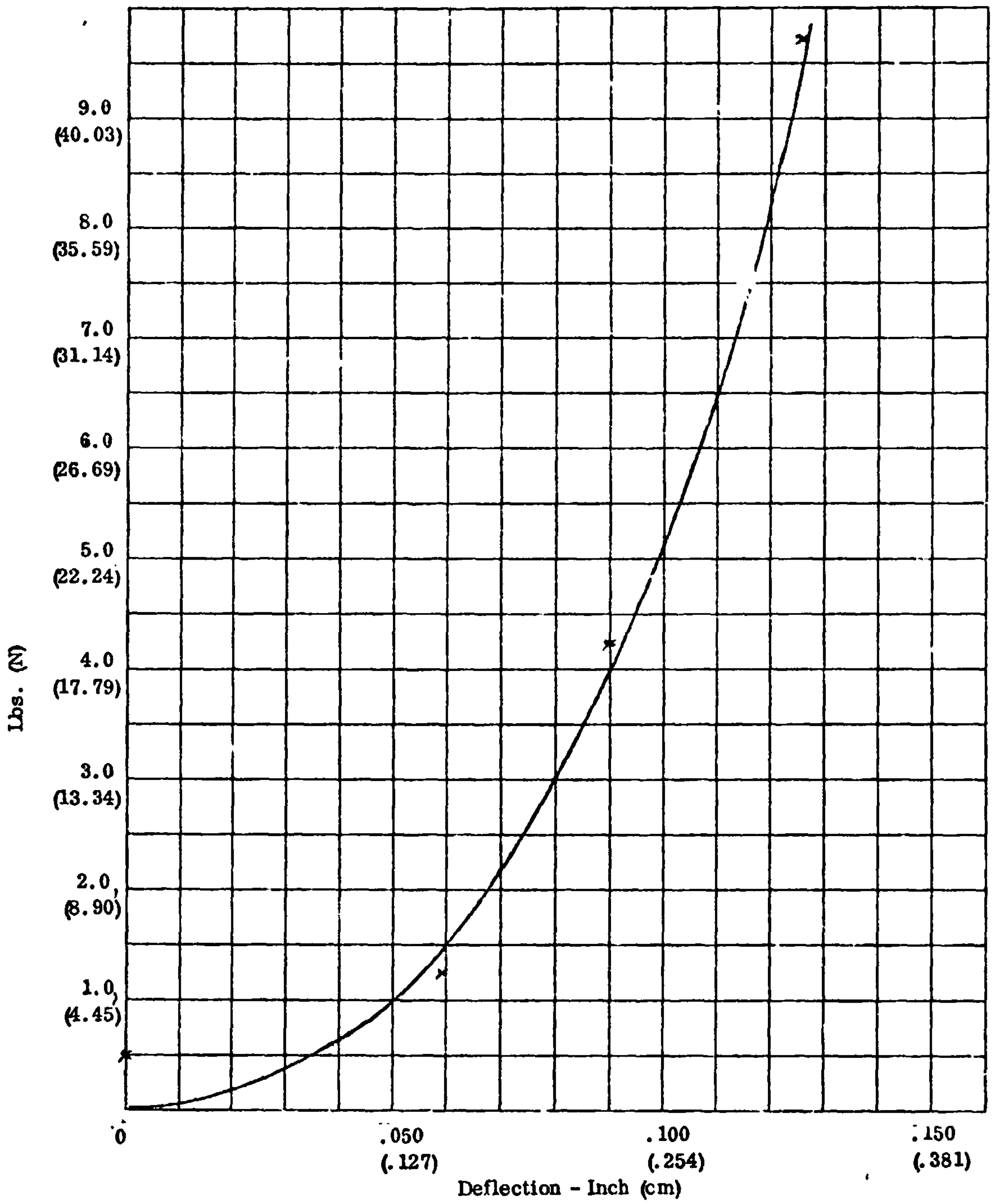


Figure 74 - Load Vs. Deflection - Flexure Assembly P/N X28400 Valve

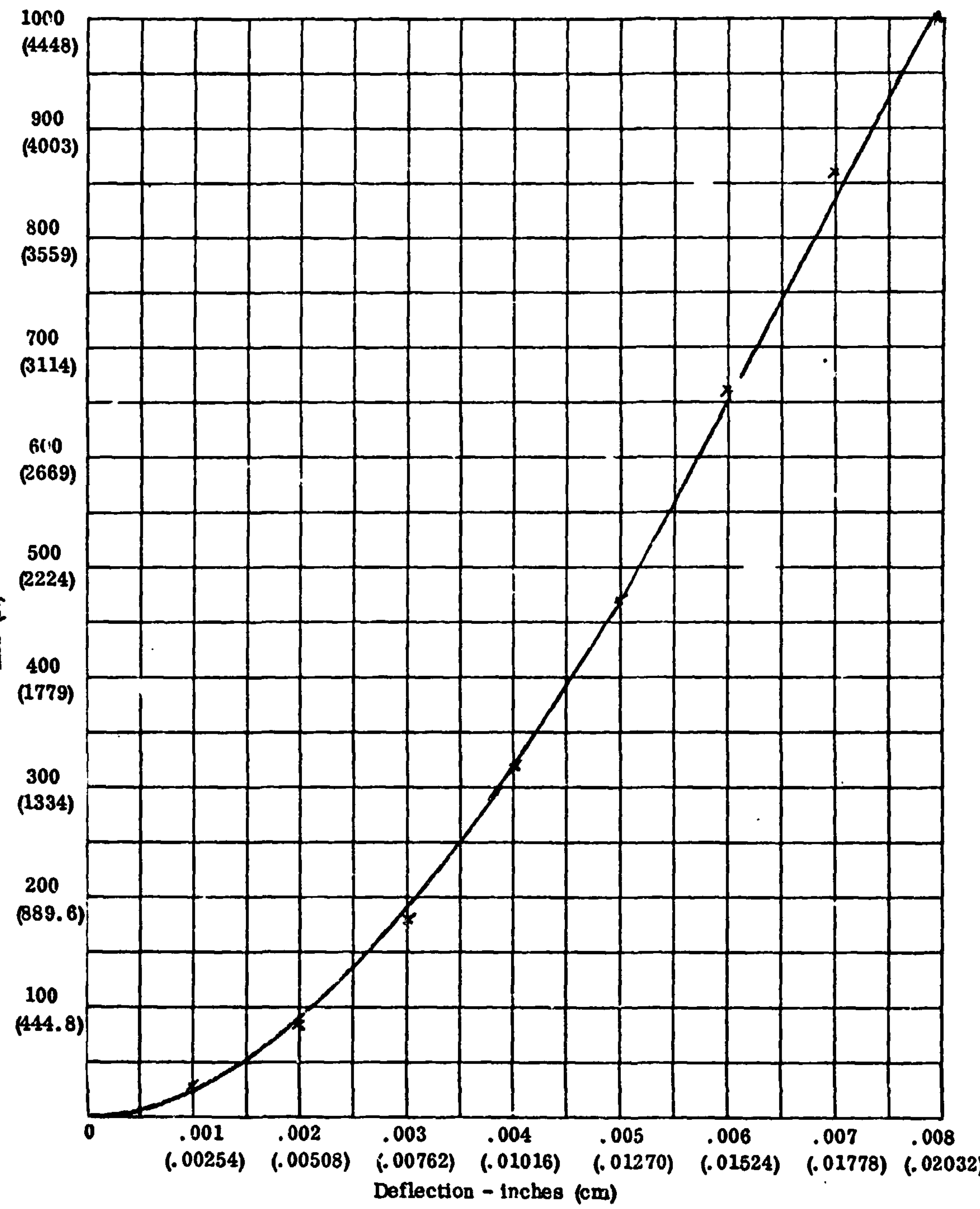


Figure 75 - Load Vs. Deflection - P/N X28400 Valve Seat

acceptance testing. A 25 micron absolute inlet filter was then installed upstream of the Test Fixture and the unit was transferred to Building 37 for performance of the cycling tests.

Test Plan

Upon installation of the P/N X27449 Test Fixture in Building 37, acceptance tests were conducted in accordance with Marquardt Test Plan No. 0192 which is included as Appendix D. Acceptance tests were also conducted on the P/N X28400 Test Fixture. Following successful acceptance tests, both units were evaluated (performance tested) in accordance with Marquardt Test Plan No. 0193 which is included as Appendix E.

Test System Preparation and Description

The purpose of the test program was to accumulate 100,000 cycles of operation on the various valve subcomponents at an operating pressure of 450 psia (310 N/cm^2) and operating temperatures of 200 °R (111°K), ambient, and 850°R (472°K). Table XVIII presents the Test Matrix that was planned for each of the two Test Fixtures, namely the P/N X27449 Valve and the P/N X28400 Valve. As evident from this table, baseline tests consisting of open and closing responses, leakage and pressure drop were performed periodically throughout the cycling program. Testing was performed in accordance with Marquardt Test Plan No. 0193 entitled, "Valve Test Fixture Tests." This Test Plan is included in this report as Appendix E. Also included as Appendix E is a list of the instrumentation used in support of this program showing the range, manufacturer, Marquardt identification number, and the latest calibration and due dates.

Testing was also accomplished in the southwest corner of Building 37 at the Marquardt-Van Nuys Test Facility. This is the same location that was previously used to perform the sealing closure evaluation. The required pressurant supplies as well as hot and cold conditioning equipment were already available at that site from the sealing closure evaluation tests. A schematic of the test fixture test setup is shown in Figure 76. The test system consisted essentially of provisions for mounting the test fixture inside an environmental box and supplying the test fixture through a 25 micron absolute rated filter with regulated gaseous nitrogen pressure at the required 2.57 lbs (1.166 Kg) per second of GN_2 flowrate at 450 psia (310 N/cm^2) inlet pressure and ambient temperature. In addition, a separate regulated pressure supply system capable of delivering either gaseous helium or gaseous nitrogen to the pilot valves of the test fixture was available. Thermal conditioning of the test fixture was accomplished by bleeding gaseous nitrogen, heated by means of an electric heater, into the environmental chamber, or by introducing a mixture of gaseous notrogen and liquid nitrogen into the environmental chamber to cool the test fixture. To obtain realistic valve opening and valve closing responses, a

TABLE XVIII - TEST FIXTURE EVALUATION TEST PROGRAM

CUMULATIVE CYCLES		Initial	500	1000	5,000	10,000	50,000	100,000
CYCLING TEMPERATURE		Hot			Cold		Ambient	
OPENING RESPONSE	Hot	X		X				
	Ambient	X				X	X	X
	Cold			X		X		
CLOSING RESPONSE	Hot	X		X				
	Ambient	X				X	X	X
	Cold			X		X		
LEAKAGE	Hot	X		X				
	Ambient	X				X	X	X
	Cold			X		X		
PRESSURE DROP	Ambient	X				X	X	X

Operating Pressure 450 PSI (310 N/cm^2)

Operating Temperature
Hot = +390°F (+ 199°C)
Ambient = +70°F (+ 21°C)
Cold = −260°F (− 162°C)

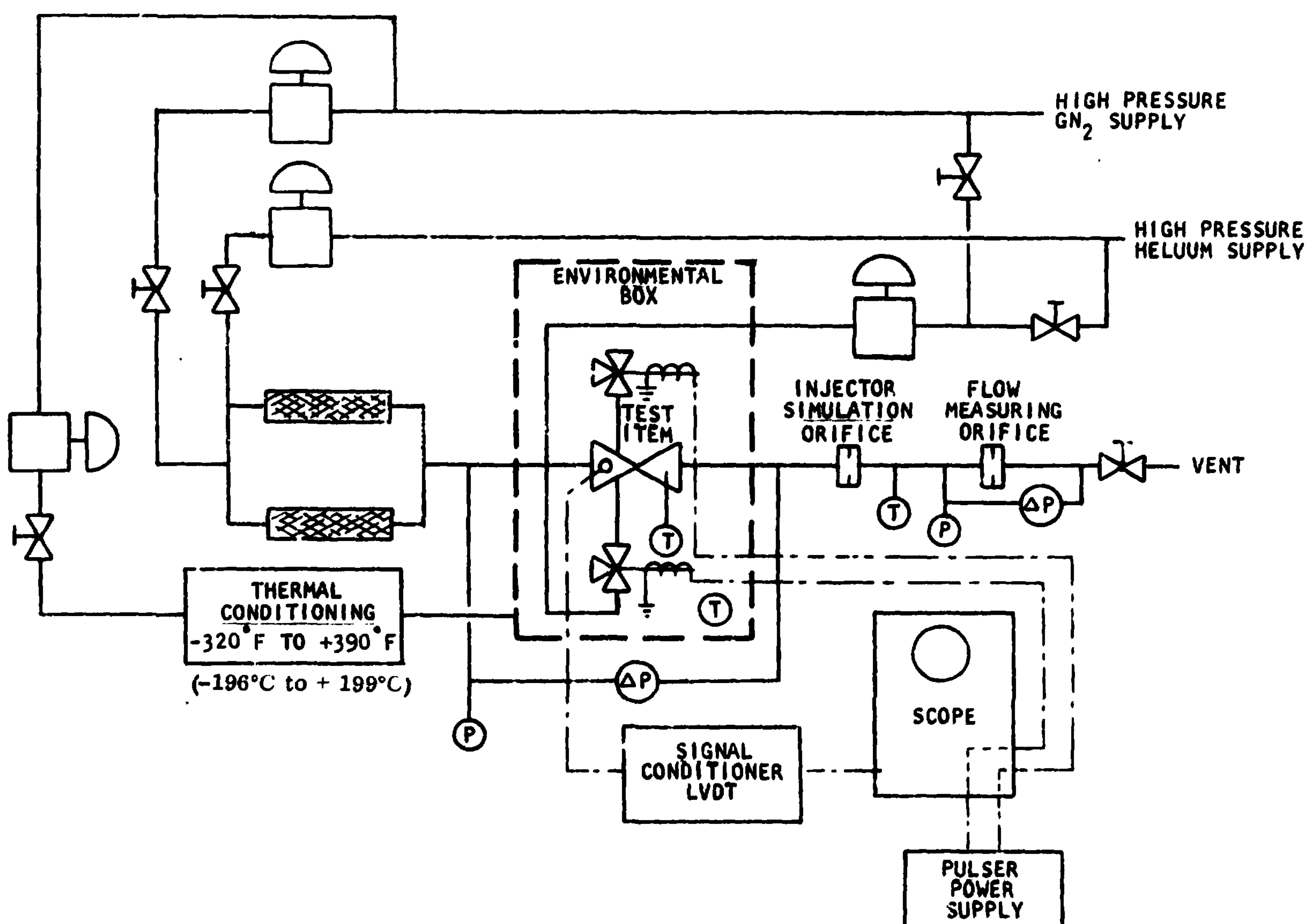


Figure 76 - Test Fixture Test Setup Schematic

presized orifice simulating the overall injector orifice size was installed in a pre-selected location downstream of the test fixture so as to simulate an injector dribble volume of 16 cubic inches (262 cm^3). The system was equipped with the necessary instrumentation to measure temperature, pressure, pressure drop, and GN_2 flowrate. In addition, the LVDT transducer incorporated in the test fixture was connected to an oscilloscope in parallel with pilot valve current and voltage traces to permit the measurement of the test fixture response characteristics.

A photograph of the test fixture evaluation system as used for the ambient testing is shown in Figure 77. Figures 78 and 79 show the environmental box used during the hot and cold temperature testing as well as a closeup of the test fixture installed inside the environmental box. The test fixtures were operated by means of a pulser/driver at voltages from 24 to 28 volts dc. The nominal operating frequency used during the cycling program was 5 cps (H_z), although operating frequencies as high as 10 cps (H_z) were demonstrated on occasion. As mentioned previously, both the P/N X27449 and X28400 Test Fixtures were evaluated in this system and the following two sections describe the test results obtained with these test fixtures in chronological order.

Test Results

P/N X27449 - Test Fixture

Initial baseline tests of the P/N X27449 Valve disclosed that the pressure drop through the valve was only 7 psi (4.8 N/cm^2) at the nominal GN_2 flowrate as compared to the originally predicted 15 psi (10.3 N/cm^2) pressure drop. From this data it was concluded that the valve discharge coefficient was considerably better than had been originally assumed and that indeed a significant amount of pressure recovery was obtained. The initial tests also disclosed that the valve did not fail safe closed when the supply pressure was reduced to zero. Investigation of this matter disclosed that the static friction of the dynamic seals in the test fixture was approximately 30 to 40 pounds (133 to 178 N) whereas the force output of the coil spring was only 24 pounds (107 N). A decision has therefore been made to replace the coil spring with a stronger spring which will provide approximately 48 pounds (213 N) force in the open position.

The initial leak checks of the various dynamic valve seals also indicated the following leakage rates in scc's per hour at 450 psia (310 N/cm^2) helium pressure:

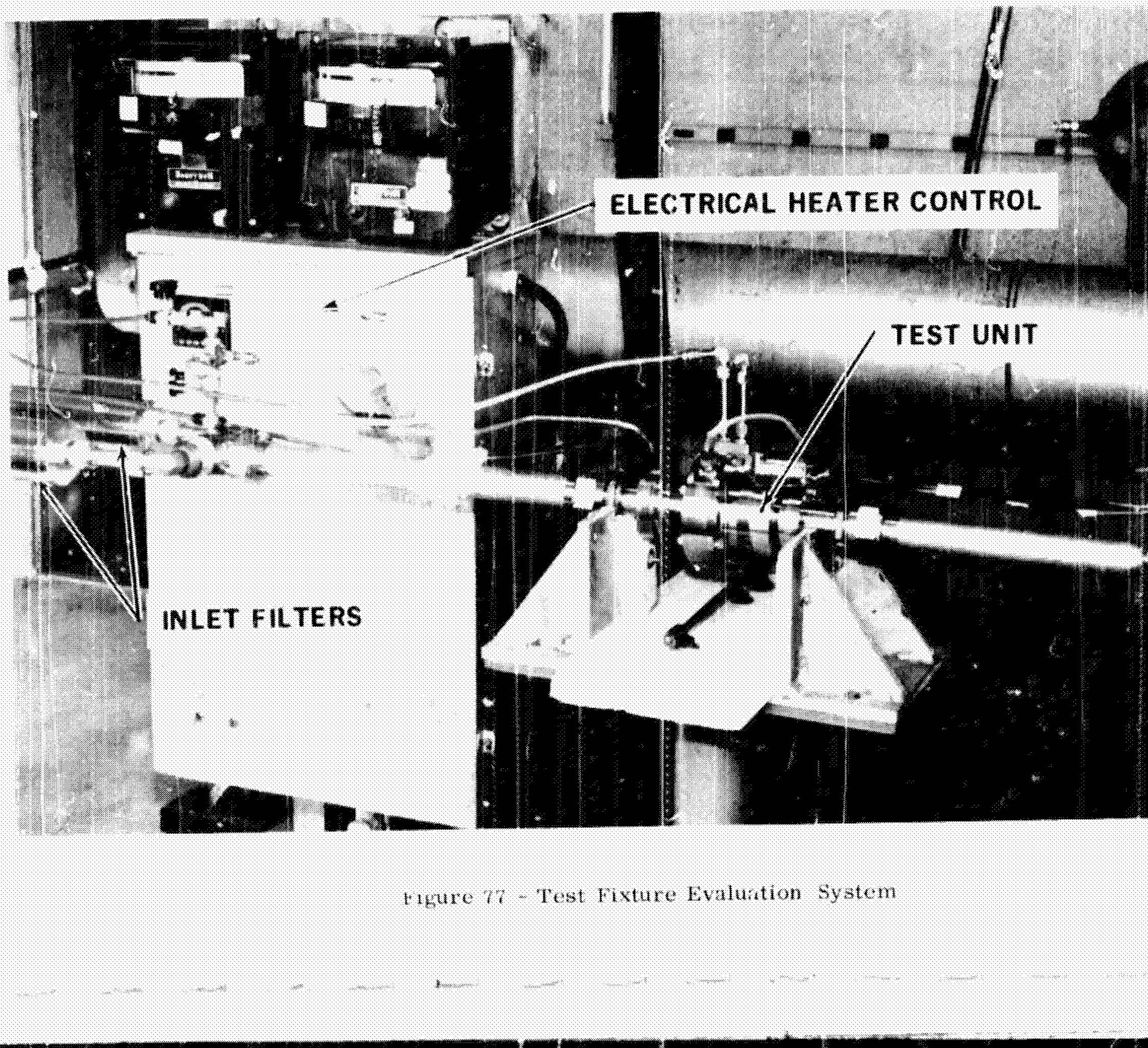


Figure 77 - Test Fixture Evaluation System

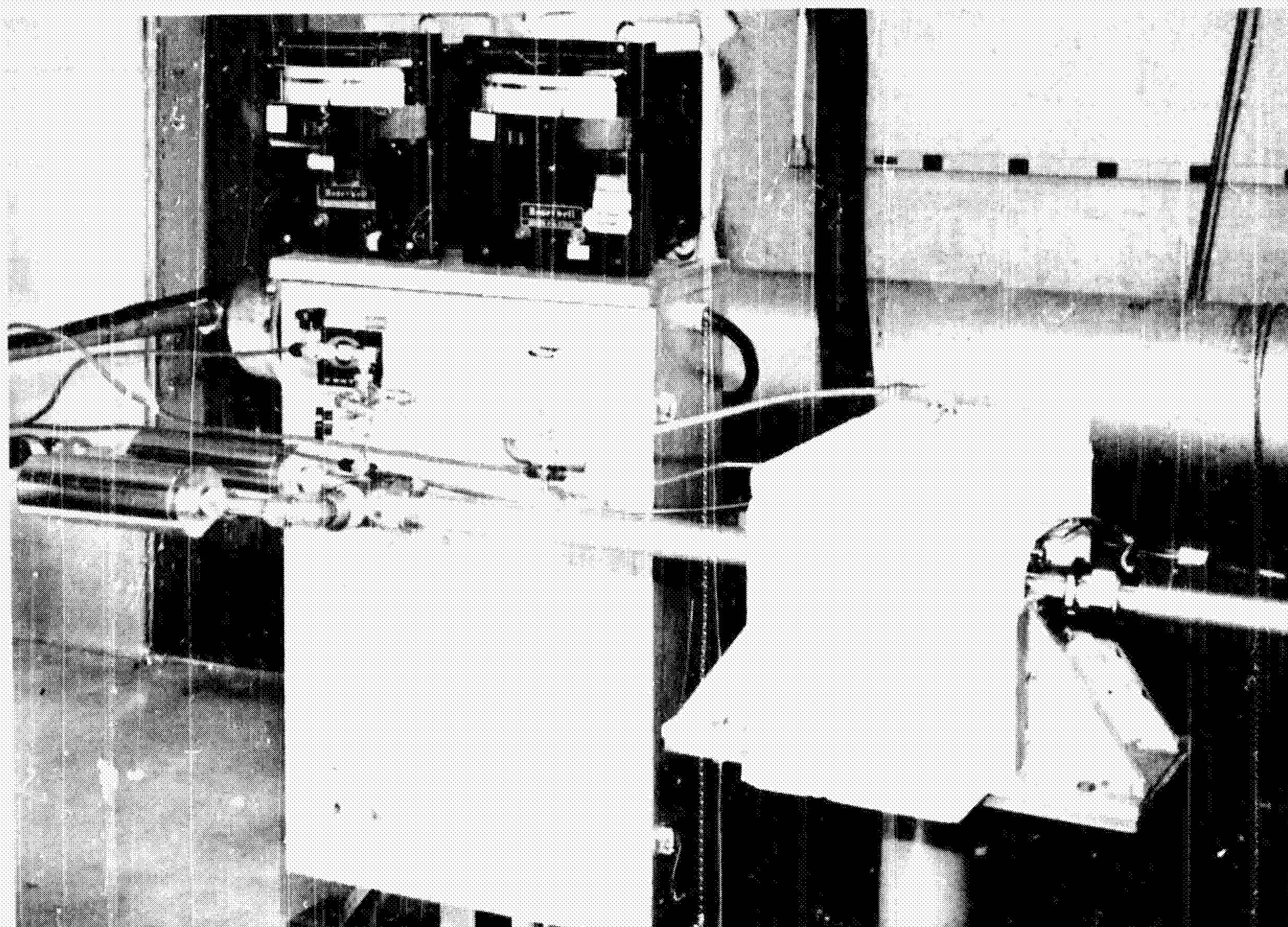


Figure 78 - Test Fixture Evaluation System Including Environmental Box

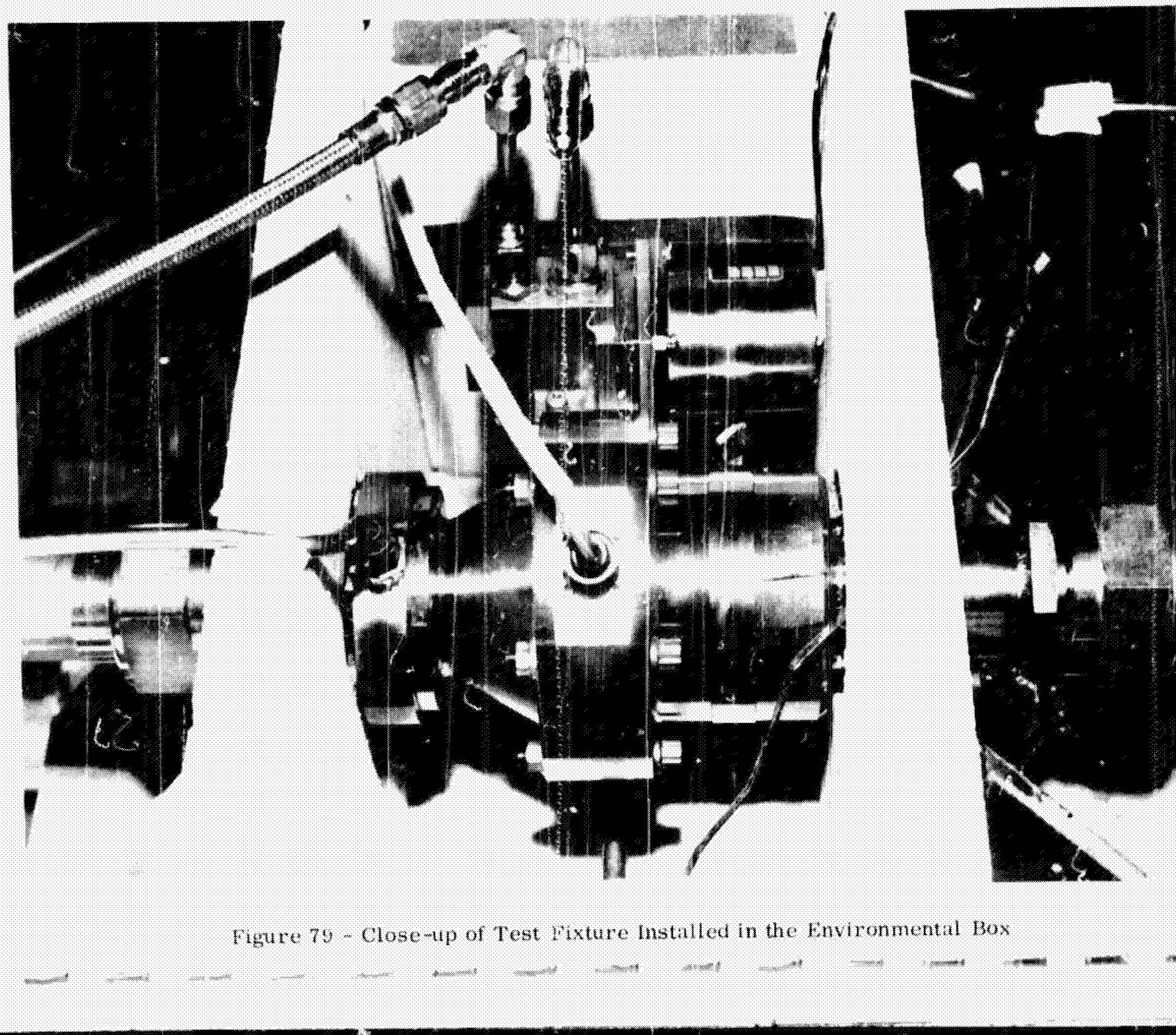


Figure 79 - Close-up of Test Fixture Installed in the Environmental Box

Main Seat	240
Downstream Piston Seal	180
Upstream Piston Seal	140
Rod Seal	52
Redundant Seals	0

While some of these leakages were slightly in excess of the 100 scc's per hour allowable, they were believed to be close enough to permit initiation of the cycling program in order to determine whether wear-in and the type of leakage improvement previously observed on other sealing closures would also occur with these seals.

Initial response testing of the P/N X27449 Valve demonstrated the very fast response characteristics previously predicted with Marquardt's analog computer program. Figure 80 shows the oscilloscope traces obtained during initial baseline tests with 375 psig (258 N/cm^2) GN_2 in the main valve and 375 psig (258 N/cm^2) helium to the pilot valves. As evident from this oscilloscope picture, the valve opening response from electrical ON to valve fully open is 12 to 13 milliseconds and the valve closing response from electrical OFF to valve fully closed is 11 milliseconds. These response characteristics compare very well with the analog computer printouts shown in Figures 81 and 82, particularly when it is taken into account that the actual pilot valve opening responses experienced are approximately one millisecond faster than had been assumed for the computer program and that the actual average pilot valve closing responses experienced are approximately four milliseconds faster than had been assumed for the analog computer program.

In view of the slightly above specification leakage data obtained during initial baseline tests, it was decided to start cycle testing at ambient temperature. Ten thousand ambient cycles were accumulated and the leakage data measured during subsequent baseline test is shown in Figure 83. As evident from Figure 83, observed leakage rates did not change appreciably in comparison to those originally measured. The test fixture assembly was subsequently heated to +390°F (+199°C) and leakage measurements were repeated. Without cycling the test fixture, the maximum leakage rate observed at elevated temperature was 40 scc's per hour. Subsequently, 1000 cycles were accumulated at elevated temperature. Repetition of the leakage measurements at elevated temperature after a total accumulated number of cycles of 11,000 disclosed that the rod seal and main seat leakage increased dramatically. The valve was then allowed to cool to ambient temperature and another leak check was made. Again, main seat leakage and rod seal leakage were excessive. It was then decided to cycle the valve an additional 1000 cycles at ambient temperature to determine if additional cycling had an effect on the main seat and rod seal leakage rates. After 12,000 accumulated cycles, both the main

Main Valve Supply: 375 PSIG GN_2 (258 N/cm^2)
Pilot Valve Supply: 375 PSIG He (258 N/cm^2)
Voltage: 28 VDC

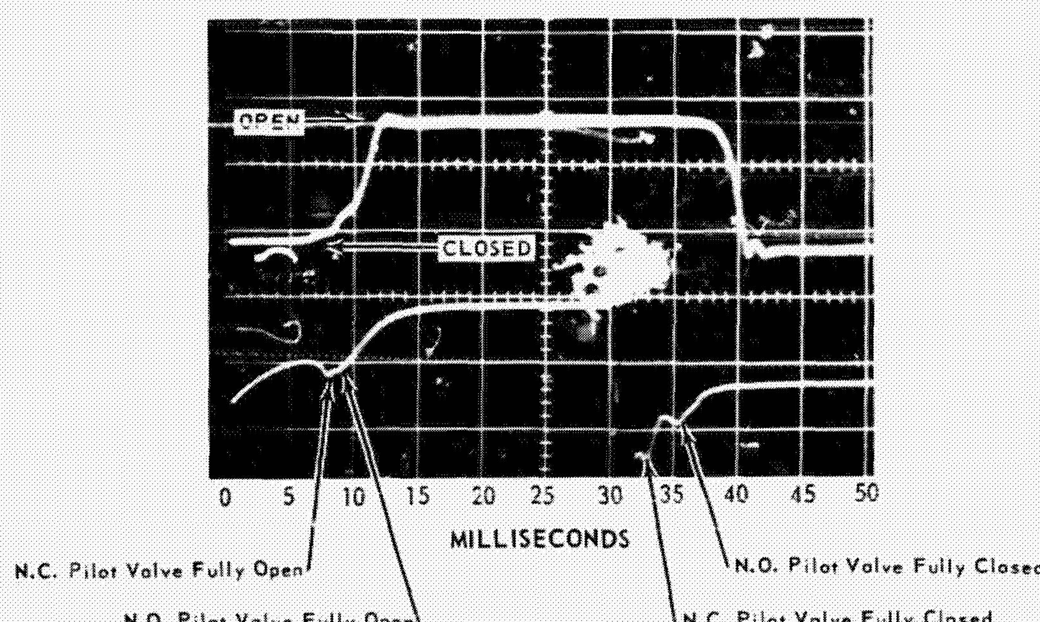


Figure 80 - Response Data, P/N X27449 Valve, S/N 001

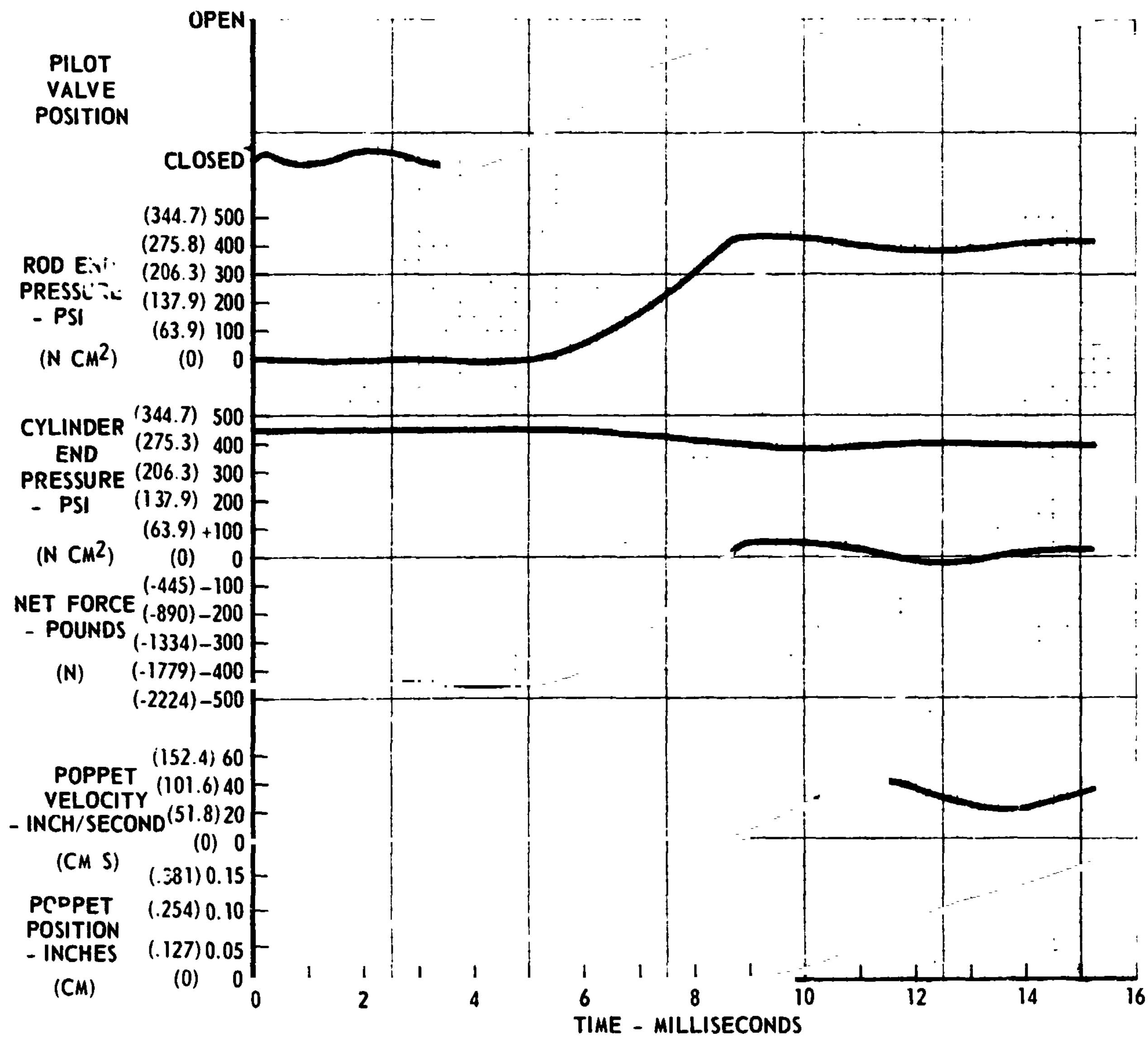


Figure 81 - Analog Computer Program Printout, Valve Opening Motion, P/N X27449 Valve, Helium Actuation

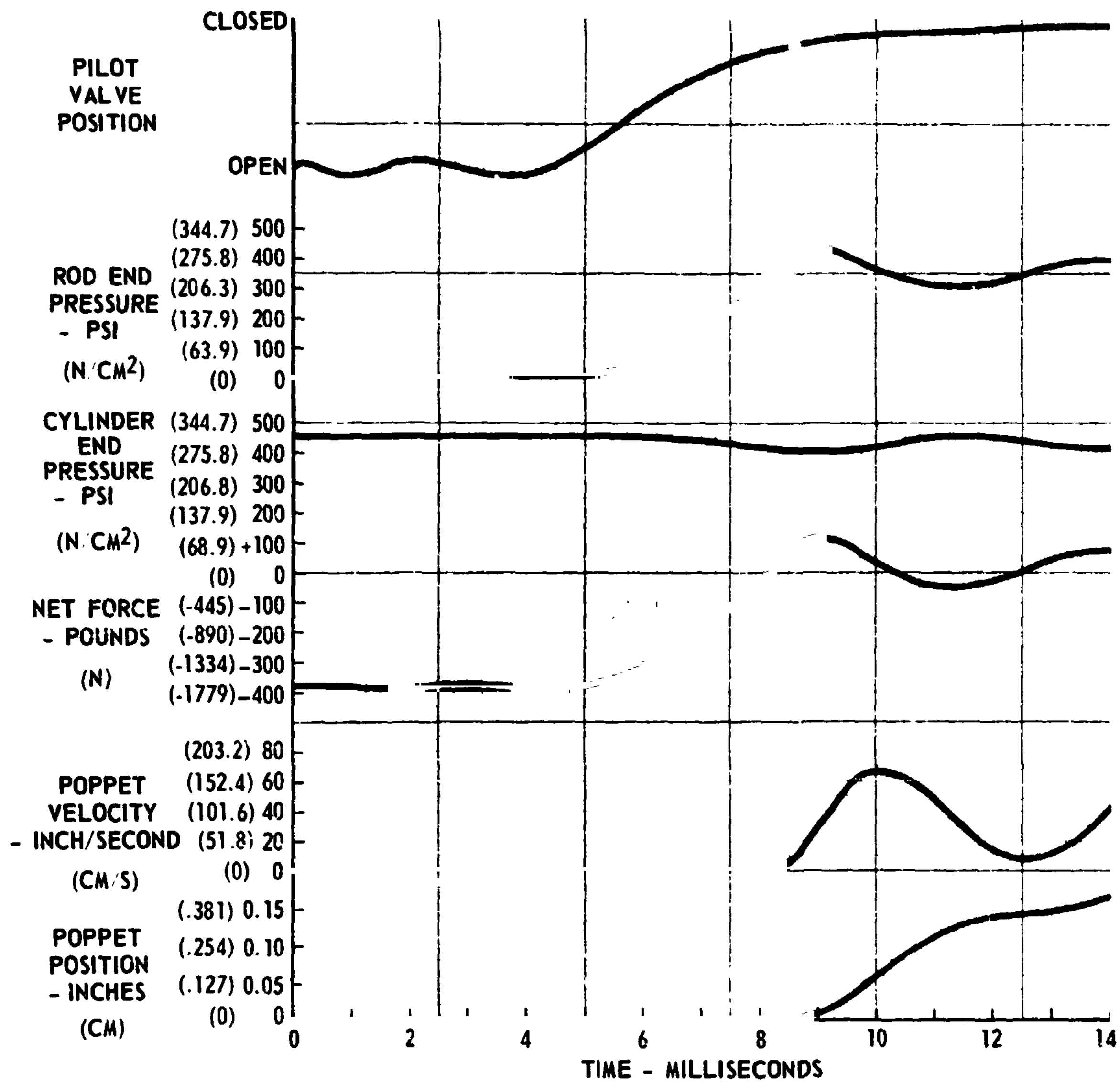


Figure 82 - Analog Computer Program Printout, Valve Closing Motion, P/N X27449 Valve, Helium Actuation

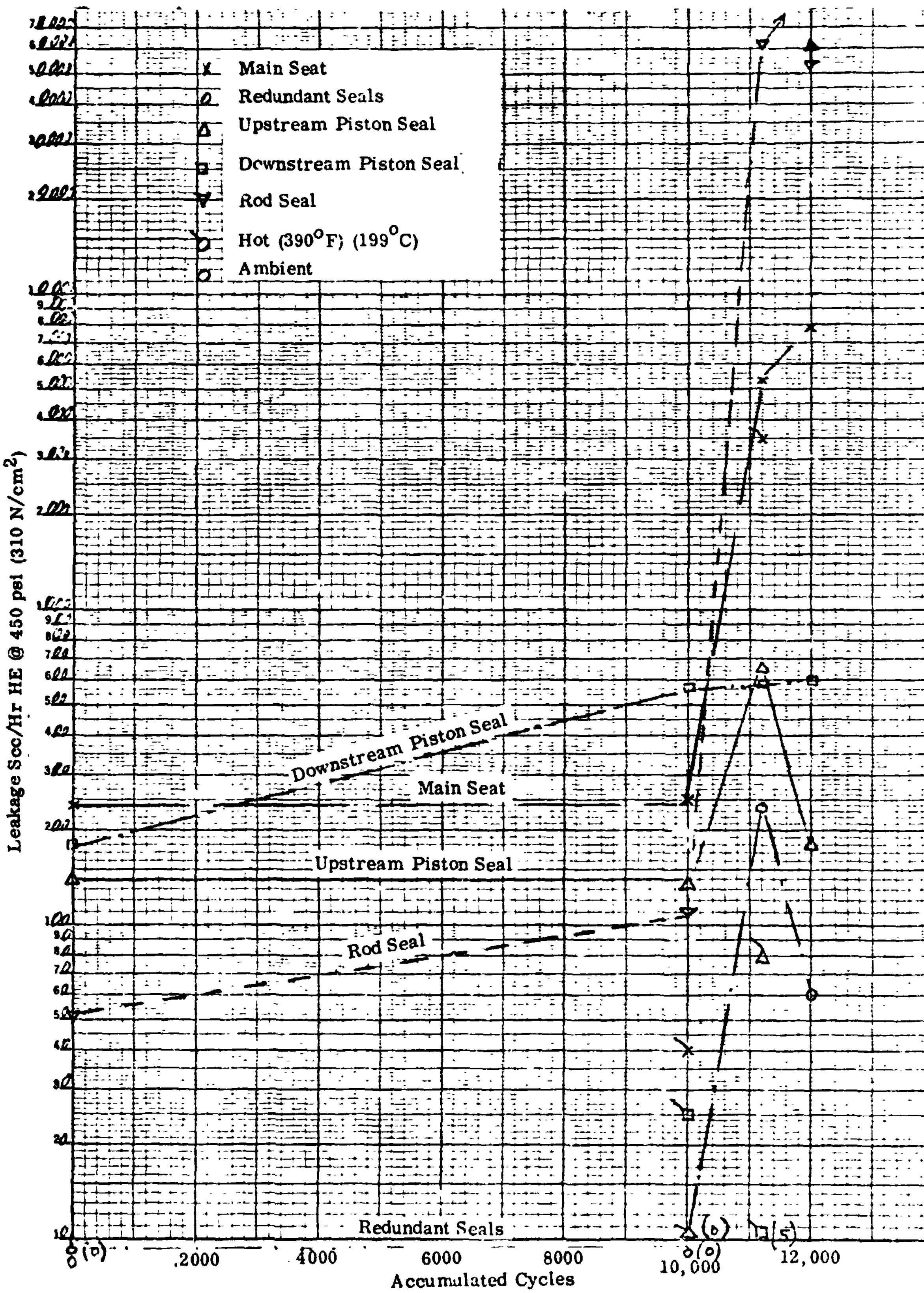


Figure 83 - Leakage Vs. Accumulated Cycles, P/N X27449 Valve, S/N 003

seat and rod seal leakage increased further. The other valve dynamic seals did not change appreciably. Since the main seat and rod seals had obviously failed, it was decided to terminate testing of the X27449 Valve at that point and to disassemble the valve for inspection.

Disassembly of the P/N X27449 Test Fixture disclosed the rod seal condition shown in Figure 84 . This photograph shows that the inner diameter of the rod seal has been extruded forward along the shaft, completely out of the rod seal cavity. This is best seen by comparing this photograph with Figure 67 . It is evident in Figure 67 that the forward wall of the seal cavity leaves considerable space between its inner diameter and the rod inner diameter. This design configuration was originally recommended by the seal vendor to facilitate installation of the rod seal into the groove in the original seal holder design which was a single piece of construction. However, since Marquardt had gone to a two-piece construction anyway, this excessive clearance was no longer required and the seal holder was therefore modified as shown in the photograph in Figure 85 . This modification proved to be effective during subsequent testing in as much as no further seal extrusion was encountered.

Examination of the main seat disclosed that the gold plating had flaked off the base metal in numerous places. A photograph of this condition through a 105 power microscope is shown in Figure 86. In this photograph, flaking of the left side of the land and a folding over of the flaked gold on the main sealing area can be noted. A meeting was subsequently held with the gold-plating vendor in an effort to determine the cause of this flaking condition. The vendor was unable to explain this condition except to note that perhaps initial cleaning of the base metal was inadequate, consequently adherence of the gold plating was insufficient. Methods to improve adherence of the gold plating were discussed with the vendor and his suggestion of initially flashing the base metal with copper was accepted. The seat was subsequently stripped and replated. Prior to reinstallation of the replated seat into the P/N X27449 Valve, the sealing surface was lapped slightly. This lapping operation disclosed that the gold surface appeared to scratch much more easily than previous gold platings and also again resulted in some flaking action. A photograph of this gold plating in the lapped sealing area is shown in Figure 87.

The valve seat was then again returned to Vendor A and Marquardt requested to see the processing records to verify that the last gold-plating indeed agreed with the earlier gold-plating. Although the vendor had previously assured Marquardt that he maintained a record for each plating job, he was unable to produce such a record. Further discussions with the vendor indicated that he had a number of gold-plating solutions, all of which were 24 karat gold, in the processing area and that any one of the solutions could have been used for either of the plating jobs. Marquardt there upon decided to seek another gold-plating vendor.

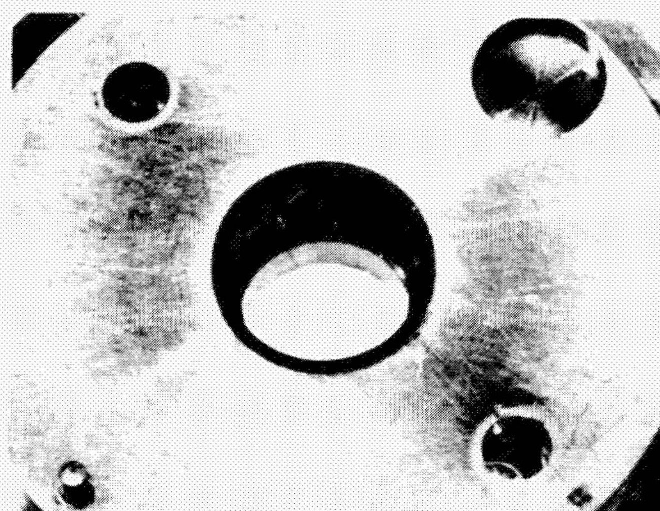


Figure 84 - Valve Piston Shaft Seal After 12,000 Cycles, P/N X27449, S/N A Valve

Figure 85 - Modified Valve Piston Shaft Seal, P/N X27449, S/N B Valve

Figure 86 - S/N A - After 12,000 Cycles (X105)

Figure 87 - S/N B - After Replating by Vendor A (X105)

Figure 88 - S/N B - After Replating by Vendor B (X105)

To gain a better understanding of the various gold-platings available and their properties, The Marquardt Company arranged several meetings with the Sel-Rex Corporation. In addition, Englehard Minerals and Chemical Corporation was contacted to determine what gold-plating solutions they had available. Based on data from References 39, 40, and 41, The Marquardt Company prepared a list of the most promising 24 karat gold-plating solutions. This list is presented in Table XIX. As evident from this Table, although all of the golds listed are 99.99% pure, the properties of the platings do vary significantly.

As a result of discussions with gold plating Vendor A, Marquardt decided that it was most likely that the vendor had previously used the Sel-Rex process identified at BTD-200. Consequently, Marquardt funded a second gold-plating vendor to replate the seat with this solution. The vendor was also thoroughly indoctrinated into Marquardt's previous problems with Vendor A. Upon receipt of the seat plated by Vendor B, Marquardt again lapped the sealing land light. During this lapping operation a total of three small flaked areas were again uncovered. One of these areas is shown in Figure 88. The part was subsequently returned to Vendor B for examination and again Vendor B could conclude only that the part may not have been adequately cleaned prior to plating. The part was therefore re-plated by Vendor B and this time lapping operations at Marquardt did not result in any evidence of flaking. Consequently, the seat was re-installed in the P/N X27449 Test Fixture and the valve was returned to Building 37 for further cycle testing.

The first baseline tests of the P/N X27449 Valve Searl No. B. disclosed leakage rates as follows:

Main Seat	<10 scc's per hour
Downstream Piston Seal	53 scc's per hour
Redundant Seals	270 scc's per hour
Upstream Piston Seal	320 scc's per hour
Rod Seal	1900 scc's per hour

The valve was subsequently cycled at hot, cold, and ambient temperature for 100,000 cycles. These cycling tests disclosed excessive leakage of all dynamic seals at cold temperature and subsequently at ambient temperature. The excessive dynamic seal leakage also made it difficult to measure main seat leakage.

TABLE XIX - PROPERTIES OF SEVERAL 24K GOLD DEPOSITS

SEL-REX PROCESS	PURITY %	HARDNESS kg/mm^2	DENSITY g/cc	WEAR $mg/hr/10\ cm^2$	APPROXIMATE ENGLEHARD EQUIVALENT
BTD-200	99.995	131-185	19.2	0.42	ECF-60
Pur-A-Gold	99.999+	44-78	19.1	0.50	E-50
Temperex HD	99.999+	52-78	19.2	0.07	E-56
Temperex SP	99.999+	78-129	19.1	0.06	
Autronex NI	99.998	135-167	17.9	1.01	E-90

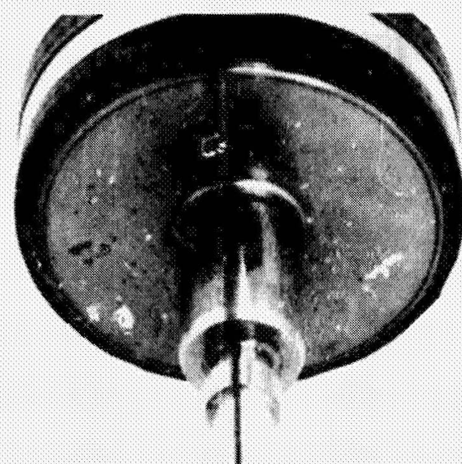
* SEL-REX Supplies 9 More Gold Solutions Which are 99.9+ Pure

Leakage measurements after 100,000 cycles at ambient temperature resulted in the following data:

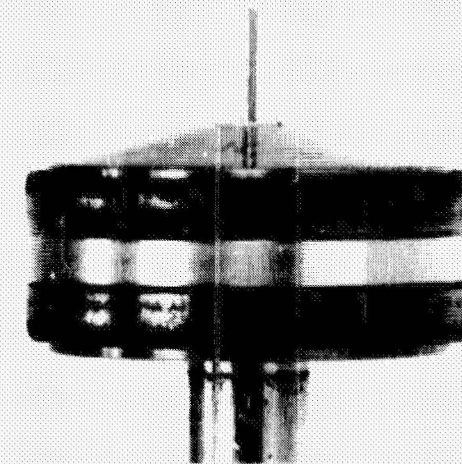
Main Seat	**700 scc's per hour**
Upstream Piston Seal	**1200 scc's per hour**
Redundant Seals	**4500 scc's per hour**
Downstream Piston Seal	**5000 scc's per hour**
Rod Seal	**7000 scc's per hour**

A photograph showing the condition of the various dynamic seals after 100,000 cycles of operation is shown in Figure 89. This photograph should be compared with Figure 66, which shows the same seals in the original condition. As evident from Figure 89, all of the dynamic seals feature a worn-out condition. In particular, close examination of the rod seal shows that the bronze impregnated teflon is actually completely worn through down to the Eligiloy spring. Both the piston seals and the redundant seals show considerable loss of material which has been transferred over to the mating parts. This is particularly evident on the outer diameter of the poppet as shown in Figure 90. By examining the downstream piston seal in Figure 89, it may be also seen that the Eligiloy spring has been fairly well flattened. This condition is apparently due to the fact that the bronze impregnated teflon shrinks considerably more at cryogenic conditions than the metal and has sufficient force under those conditions to collapse the spring. The Marquardt Company met subsequently with the seal vendor and the conclusion of this meeting was that the conditions the weals were being subjected to were simply too severe and the seals were worn out. Examination of the poppet/seat sealing interface as shown in Figure 90 disclosed that a small amount of gold had transferred from the seat to the poppet. Photographs of the mating surface of the poppet and the seat taken through a microscope are shown in Figure 91. The trnasfer of this gold resulted in the degradation of the original 1-AA finish and explained the 700 scc's per hour of helium leakage at 450 psia (310 N/cm^2) inlet pressure measured. It was, however, surprising to find that this surface finish degradation had taken place in light of the fact that the same type of gold lip seat had been previously cycled successfully in the rapid screening tester f for 200,000 cycles. The flatness of the gold plated lip was subsequently checked on a Bendix Proficorder. Figure 92 is a reproduction of the Proficorder chart and shows a maximum out-of-flatness of 0.003" (0076 cm). This is well within the deflection capability of the lip (0.006")(.0152cm).

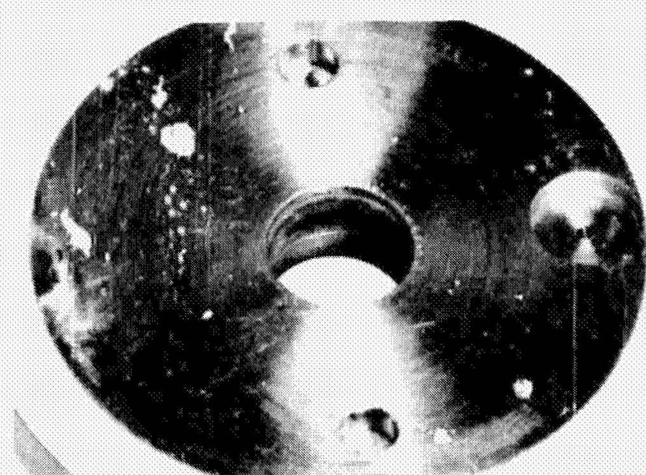
In an effort to determine whether the gold-plated seat surface finish degradation was a result of the change in poppet material or change in poppet guidance from the rapid screening tester to the P/N X27449 Test Fixture or



VALVE ASSEMBLY P/N X27449
PISTON SHAFT AND SLIDING SEALS



VALVE ASSEMBLY P/N X27449
PISTON SLIDING SEALS



VALVE ASSEMBLY P/N X27449
PISTON SHAFT SEAL

VALVE ASSEMBLY P/N X27449
INNER BODY POPPET SEAL

...re ... Act...or L...ls A... 10...0 C...s

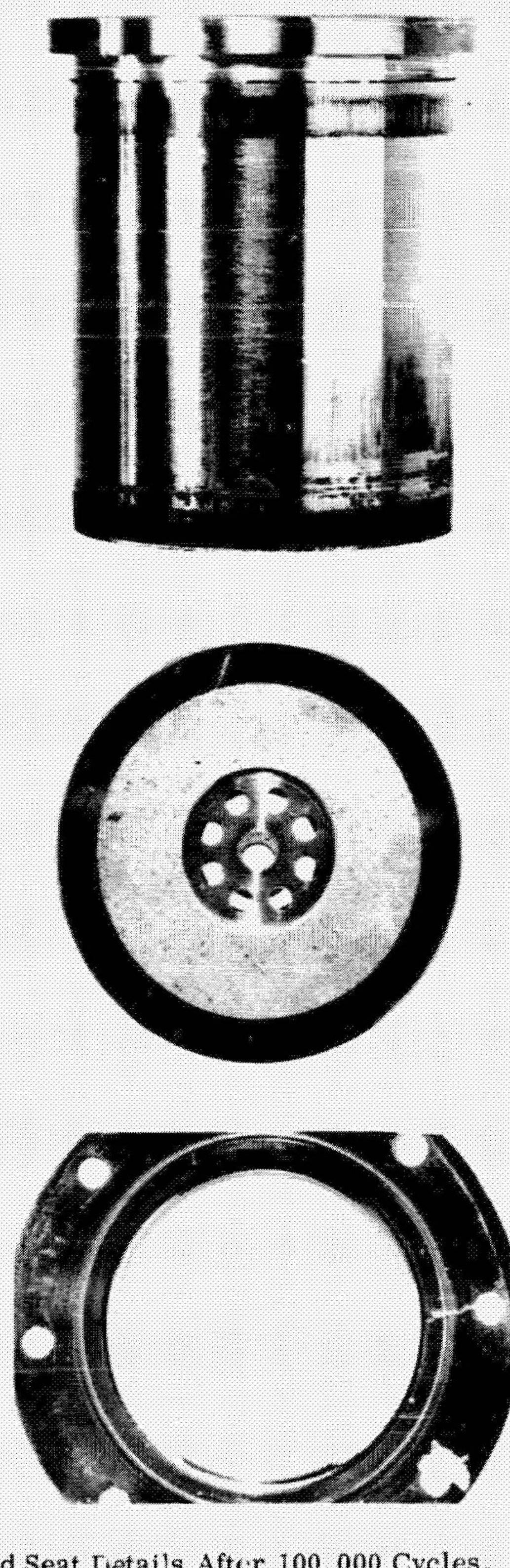
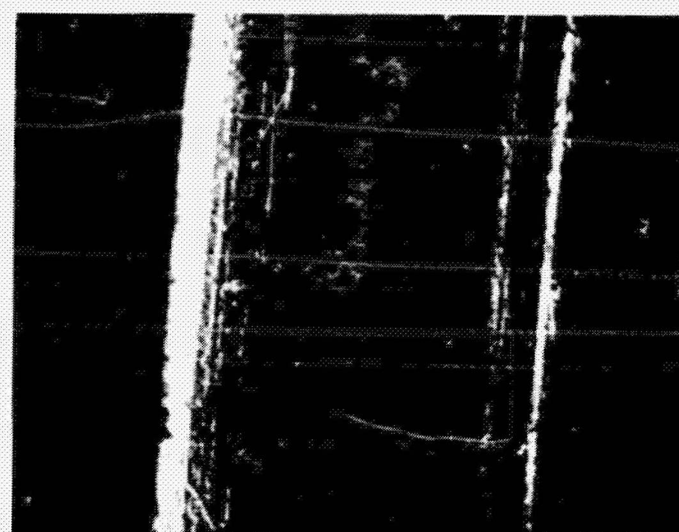


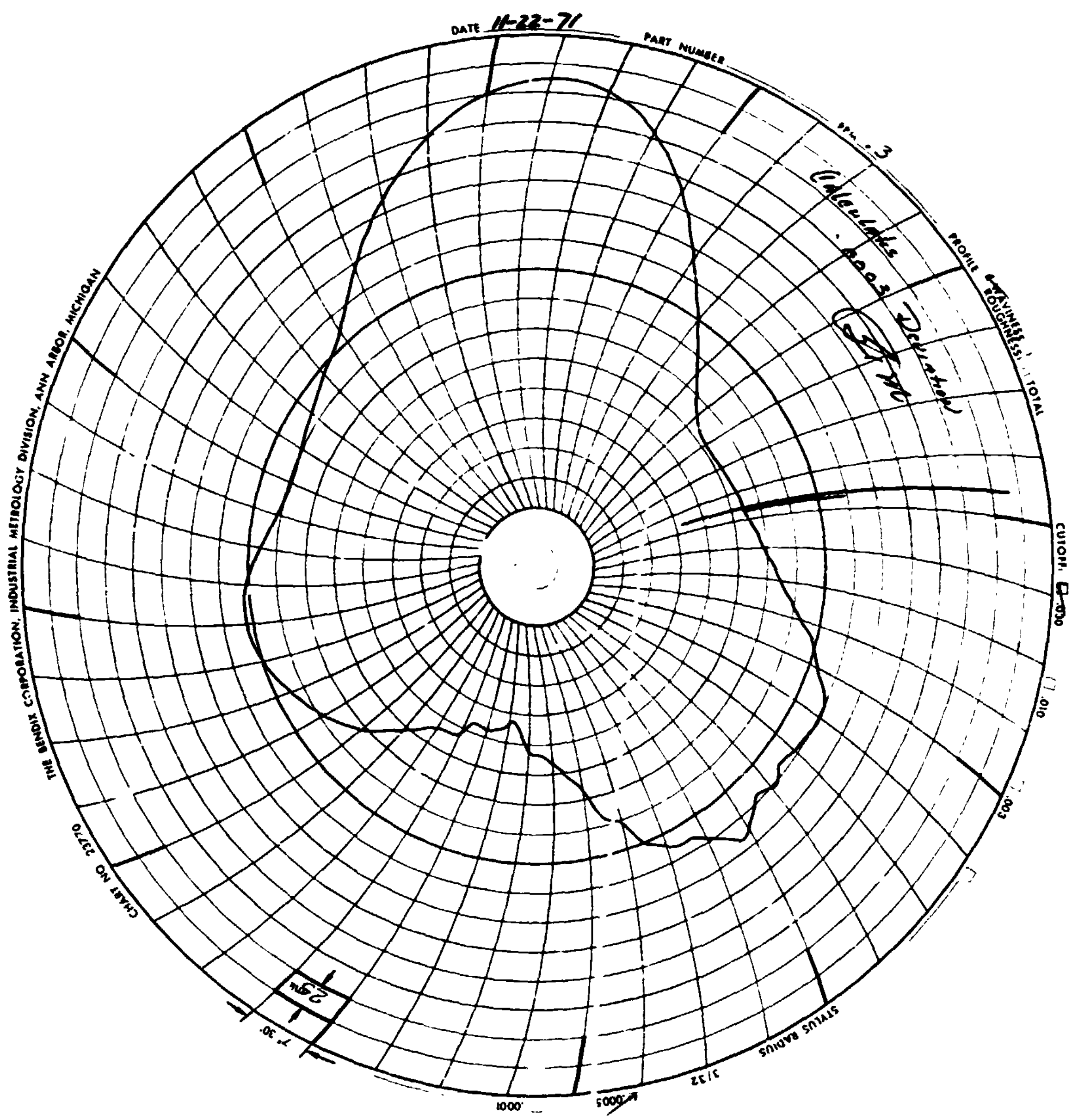
Figure 90 - Poppet and Seat Details After 100,000 Cycles, P/N X27449 Valve



GOLD LIP SEAT SEALING LAND X 105

POPPET SEALING SURFACE X 50

Figure 91 - Gold Lip Sealing Closure After 100,000 Cycles



. re ?? - Gold Plated Lip Seat Sealing Land Flatness After 100,000 Cycles of Operation, Valve P/N X27449 was related to the gold-plating used on this seat, the seat from the P/N X27449, Serial No. B Test Fixture was installed in the rapid screening tester and cycled 100,000 times with the tungsten carbide poppet previously successfully employed in the cycling of another gold-plated seat. This sealing closure evaluation effort resulted in the attainment of excellent sealing characteristics up to approximately 10,000 cycles, but a subsequent leakage characteristic deterioration which reached 450 scc's per hour after 30,000 cycles and ranged between 300 and 450 scc's per thereafter until 100,000 cycles. Post test examination of the gold-plated lip seat and the tungsten carbide poppet disclosed the same type of gold transfer and surface finish degradation observed upon completion of the 100,000 cycles in the P/N X27449 Test Fixture. Consequently, it was concluded that this gold-plating was not the same as the gold-plating originally successfully demonstrated in the rapid screening tester. Unfortunately, the correct call-out for the successful gold-plating is not known at this time because the gold-plating vendor kept insufficient records.

A summary of the P/N X27449 Valve test data and a comparison to the contract requirements is presented in Table XX. As noted previously, the valve test fixture was cycled 100,000 times at hot, ambient, and cold conditions. These tests disclosed that the dynamic bronze impregnated teflon seals are not suitable in their present configuration for this application. The tests also disclosed that the gold-plating employed on this particular seat is not quite adequate, but there are apparently gold platings available which perform satisfactorily under the particular load conditions encountered in this valve. The valve demonstrated exceptionally fast response characteristics and no response degradations were noted at ambient temperature. Low pressure drop characteristics of 7 psi (4.8 N/cm^2) at nominal flow rates were verified during all baseline tests. Other valve components, such as static seals, the LVDT transducer, the poppet self-alignment flexure, and the sliding fits employed in this valve design functioned well.

P/N X28400 - Test Results

Initial baseline tests of the P/N X28400 Valve disclosed that the pressure drop through the valve was 16 psi (11.0 N/cm^2). This compares to a nominal design point of 15 psi (10.3 N/cm^2). Response testing of this valve at 375 psig (258 N/cm^2) gaseous nitrogen, 24 volt dc, and ambient temperature showed an opening response of 29 milliseconds and a closing response of 43 milliseconds. A photograph of the LVDT trace and the current trace on the oscilloscope is shown in Figure 93 . From this photograph, it can also be seen that the pilot valve opening response at this condition is 11 milliseconds and the pilot valve closing response is 10 milliseconds. The opening response design goal was 30 milliseconds at 28 volts, so that the actual response achieved is slightly better than this goal. Similarly, the design goal for the closing response was 30 milliseconds. This goal was not achieved and the reason for this lack of

TABLE XX - X27449 VALVE TEST DATA SUMMARY

Performance Characteristic	Test Data	Contract Req't
PRESSURE DROP @ 2.76 LB/SEC GO_2 @ 540°R (1.25 Kg SEC) (300°/K)	7 PSI (4.8 N CM^2)	15 PSI Nominal (10.3 N CM^2) 25 PSI Maximum (17.2 N CM^2)
• Leakage @ 450 PSI Helium (310 N CM^2)		
•• Main Seat	700	100
•• Dynamic Seals	up to 60,000	
OPENING RESPONSE @ 375 PSI (258 N CM^2), 28 VDC, 530°R (294°K)		
• Nitrogen, Signal to Full Open	17	30 MS Total
• Helium, Signal to full Open	12	30 MS Total
• Nitrogen, Travel Time Only	8	15 MS Travel Time
• Helium, Travel Time Only	5	15 MS Travel Time
CLOSING RESPONSE @ 375 PSI (258 N CM^2), 28 VDC, 530°R (294°K)		
• Nitrogen, Signal Off To Valve closed	12	30 MS Total
• Helium, Signal Off to Valve closed	11	30 MS Total
• Nitrogen, Travel Time Only	4	15 MS Travel Time
• Helium, Travel Time Only	3	15 MS Travel Time
LIFE DEMONSTRATION CYCLES	100,000	100,000
OPERATING TEMPERATURE RANGE	140 - 850°R (78 - 472°K)	200 - 850°R (111 - 472°K)

375 PSIG GN_2, 70°F, 24 VDC
(258 N/CM^2) (21°C)

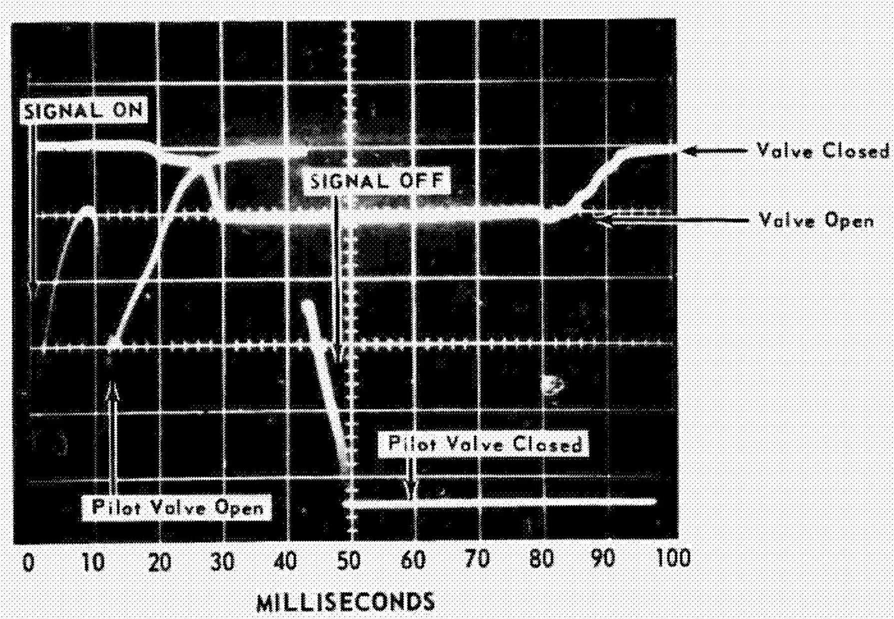


Figure 93 - Response Data, P/N X28400 Valve, S/N 001

achievement was not known at the time that these data were taken. However, upon completion of the P/N X28400 test program and disassembly of the valve it was discovered that an inlet filter had been installed in the supply pressure line to the pilot valve that featured only a 0.075 inch (.1905 cm) orifice. The purpose of this inlet filter was to prevent contamination from being introduced into the pilot valve during the disconnecting and reconnecting of the pressure supply line for leak-check purposes. Analog computer simulation of the dynamics of this valve had previously predicted that a pressurizing orifice of 0.100" (.254 cm) was required in order to achieve the 30 milliseconds response time. Consequently, it is apparent that the smaller filter orifice was controlling the flow to the pilot valve and resulted in the slower closing response performance. (It should be noted that the opening response is not affected by this orifice since the pressure behind the poppet is bled overboard through the vent port of the pilot valve to open the main valve, and thus is not affected by an orifice installed in the pressure supply port of the pilot valve.)

Initial leak checks of the main valve and the pilot valve disclosed leakage rates of 0 and 25 scc's per hour of helium at 450 psi (310 N/cm^2) inlet pressure, respectively. These data and leakage data obtained subsequently during the cycling of the P/N X28400 Test Fixture are plotted in Figure 94 . After ambient leakage tests, the test fixture was heated to 390°F (199°C and the leak checks were repeated. At this point pilot valve leakage increased to 35 scc's per hour and main valve leakage to 85 scc's per hour. The valve was then cycled at 390°F (199°C) for 1,000 cycles. Another leak check at elevated temperature disclosed main seat leakage of 80 scc's per hour and pilot valve seat leakage of 185 scc's per hour. The valve was then cooled to -260°F (-162°C and leak checks were repeated prior to any actuation of the valve. At this point, the main valve leakage decreased to zero and the pilot valve leakage increased to 1,000 scc's per hour. The valve was then cycled cold for 9,000 cycles followed by additional cold and ambient leak checks, and this was followed by 90,000 cycles at ambient temperature. The various leakage data measured are evident from Figure 94.

In summary, leakage testing showed a gradual degradation of the pilot valve leakage characteristics from an initial 25 scc's per hour to approximately 7,000 scc's per hour after 100,000 cycles. The sealing closure interface featured in the pilot valve was not based on the sealing closure development accomplished during this program, but rather was an interface which was available from the pilot valve vendor. The emphasis on the leakage testing was, of course, on the main valve. As indicated from Figure 94, leakage measurements on the main valve proved to be somewhat erratic. This erratic behavior led to the suspicion that the static seal which is in parallel with the main seat in the valve might have been leaking during the cycling program. Consequently, at the conclusion of the test program when the valve featured a leakage of approximately 95 scc's per hour, the valve outlet adapters were removed and

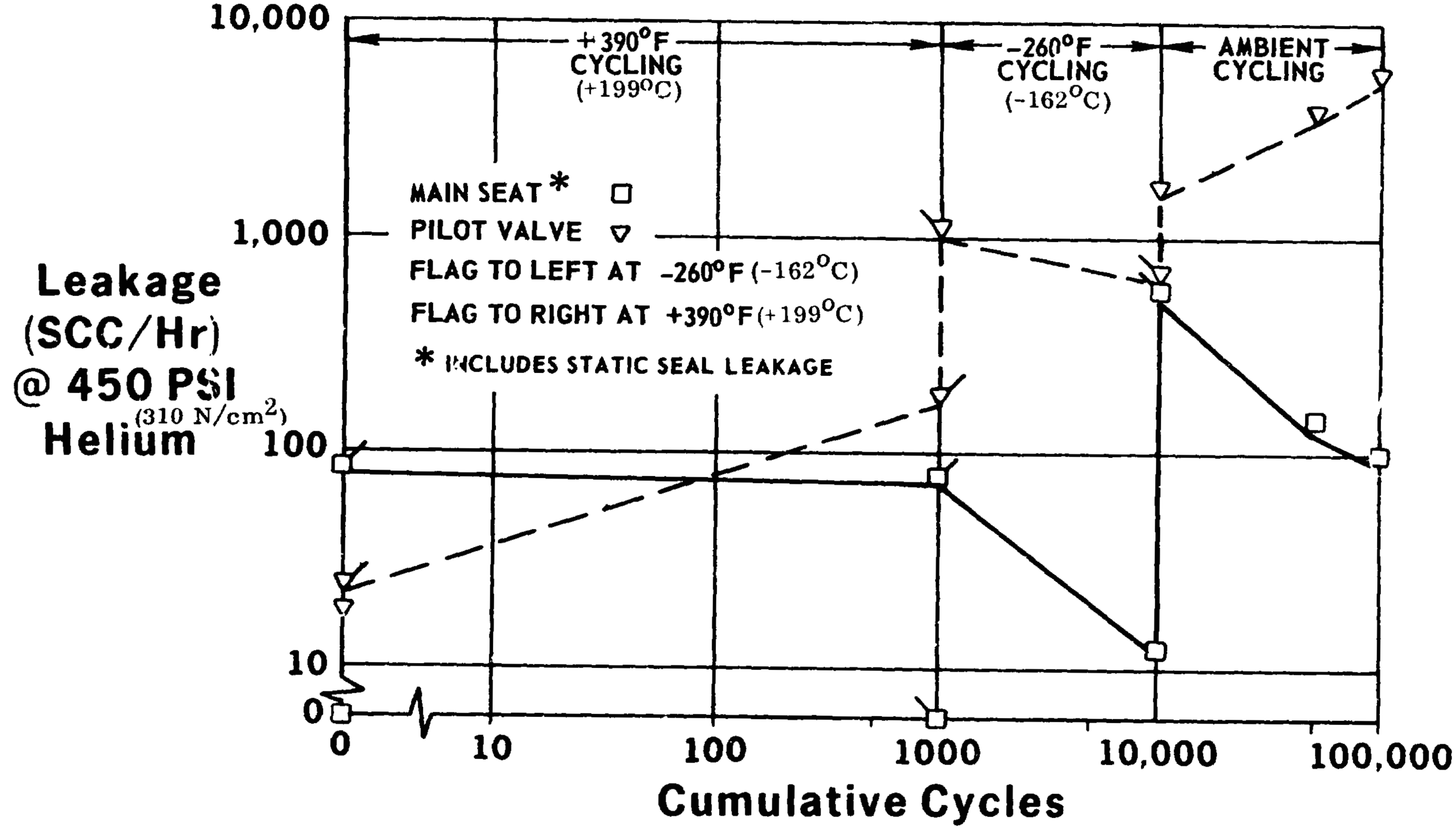
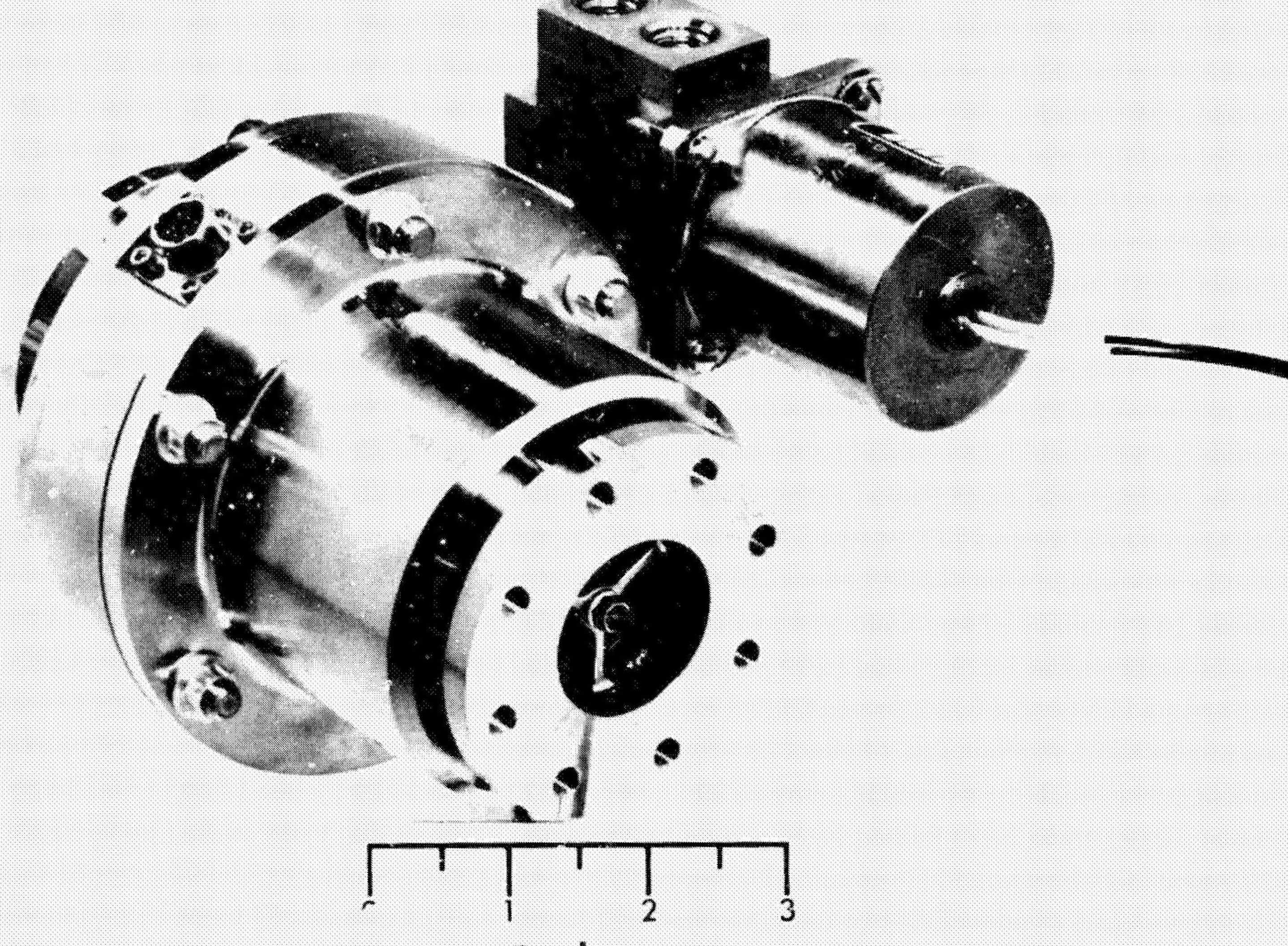


Figure 94 - Leakage Vs Cumulative Cycles

a leak check solution was applied, first at the location of the static seal and then at the poppet/seat interface. There was noticeable bubbling from the static seal crevice, but absolutely no evidence of bubbles from the poppet/seat interface. Consequently, it was concluded that most of the leakage, if not all of the leakage, observed on the main valve was due to the static seal rather than to the poppet/seat interface. This conclusion was further reinforced by the discovery of another problem during the disassembly of the P/N X28400 Valve.

During disassembly of this test fixture it was discovered that the lock nut holding the axial guidance flexure in position had backed off approximately 3/4ths of a turn. This corresponds to an axial play of 0.020 inches (0508 cm). The particular lock nut may be seen in Figure 95 in the outlet of the valve at the center of the three-legged spider. The looseness of the lock nut allowed considerable radial displacement of the axial guidance flexure and the poppet/bellows assembly. This radial displacement resulted in rubbing of the bellows against the flow shield around the bellows and also of the area scheduler against the valve inner body. A photograph of the damaged bellows assembly is shown in Figure 96 . Close examination of this photograph reveals that the top two convolutions have lost considerable material on their outer diameter. The particles generated during this rubbing inadvertantly resulted in a seat contamination tolerance test. Figure 97 is a photograph of the polyimide sealing land at a magnification of 105. This photograph shows one 35 micron particle, one 20 micron particle and numerous smaller particles imbedded in the polyimide. Furthermore, examination of the static teflon seal revealed that it, too, had imbedded numerous metal particles which must have been the result of some gas flow past the static seal. It is very probable that gas flow past the static seal did occur during each opening cycle since the loose lock nut allowed the seal cavity to expand by the 0.020 (.0508 cm) dimension every time the valve opened.

A final summary of the P/N X 28400 valve test data is presented in Table XXI. Except for the changes in leakage rates and the damage resulting from the loose lock nut, there was no apparent degradation in valve performance characteristics throughout the valve cycling program. It was believed that by providing the lock nut with a positive mechanical lock and by incorporating into the pilot valve a sealing closure which is based on the sealing technology developed during this program, the P/N X28400 Valve would prove to be a very reliable, high performance concept. Consequently, additional funding was obtained to refurbish the P/N 28400 valve fixture and conduct extended cycle tests.



(0 2.54 5.08 7.62) (cm)

Figure 95 - Valve Assembly, Valve P/N X28400, S/N 001

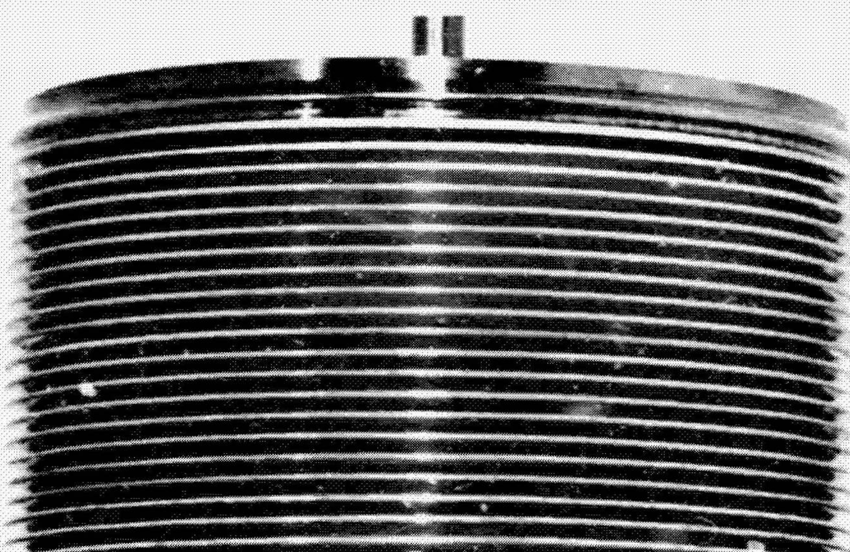


Figure 96 - Valve Bellows After 100,000 Cycles of Operation, P/N X28400, S/N 001

Figure 97 - Polyimide Sealing Land with Particle Contamination after 100,000 Cycles of Operation

TABLE XXI - X28400 VALVE TEST DATA SUMMARY

Performance Charactistic	Test Data	Contract Requirement
PRESSURE DROP @ 2.76 LBS/SEC GO_2 @ 540°R (1.25 Kg SEC) (300° K)	16 PSI (11.0 N CM²)	15 PSI Nominal (10.3 N CM²) 25 PSI Maximum (17.2 N CM²)
Leakage @ 450 PSI Helium (310 N CM²)		
Main Seat	< 100	100
Pilot Valve	5800	
OPENING RESPONSE @ 375 PSI GN_2, 24 VDC, 530°R (258 N CM²) (294°K)		
Signal to Full Open	29	30 MS Total
Travel Time Only	3	15 MS Max.
CLOSING RESPONSE @ 375 PSI GN , 24 VDC, 530°R (258 N CM²) (294°K)		
Signal Off, to Valve Closed	43*	30 MS Total
Travel Time Only	9	15 MS Max.
LIFE DEMONSTRATION CYCLES	100,000	100,000
OPERATING TEMPERATURE RANGE	140 - 850°R (78 - 472°K)	200 - 850°R (111 - 472° K)

*Slow response is due to an Orificing Error

Extended Cycle Testing of P/N 28400 Test Fixture

During May 1972, a contract amendment was received which provided for refurbishment of the subject test fixture and the requirement to extend testing to cover 1,000,000 cycles.

Test Fixture Refurbishment - A cross section of the P/N X28400 valve is shown in Figure 98. Previous test results and an examination of the valve revealed that one design change was mandatory and two others were desirable before additional testing was initiated. The mandatory design change consisted of providing a positive mechanical lock at the nut which is used to assemble the moving parts of the valve to prevent its loosening during cycle testing. The two desirable design changes which were incorporated included the addition of a high spring rate bumper in the valve body to reduce the poppet impact loads during the opening motion and the addition of a guiding sleeve to the area scheduler to assure engagement of the area sch duler into the bleed hole at all times. The valve drawings were updated in accordance with the design changes.

The P/N X28400 valve was modified to incorporate these design changes. In addition those valve components that were damaged during the previous cycle tests were repaired. These included the axial guidance flexure, bellows assembly, the poppet and seat, and the LVDT armature. The three-way pilot valves were returned to the vendor for replacement of the seats with new seats made from polyimide. New static seals for the various pressure joints also were procured.

Following the above modifications, the valve (test fixture) was reassembled in the building 32 clean room. The unit was then leak checked and mated with the inlet filter assembly prior to its transfer to the building 37 GN_2 flow facility.

Test Plan - In preparation for the extended cycling tests a test plan (MTP 0201) was prepared and submitted to NASA for approval. This test plan was a revision of MTP 0193. These revisions primarily extended the required cycling tests from 100,000 to a total of 1,000,000 cycles.

Test Results - Initial acceptance tests of the P/N X28400 valve assembly in the clean room demonstrated a main seat leakage rate of 3 scc's per hour of helium at 450 psia (310 N/cm^2) inlet pressure. This was well within the apecification requirements. Cycle testing was initiated with baseline tests performed initially and after 50,000 cycles had been accumulated. These baseline tests consisted of leakage, opening and closing response and pressure drop tests. During the cycling program, valve operation was verified every

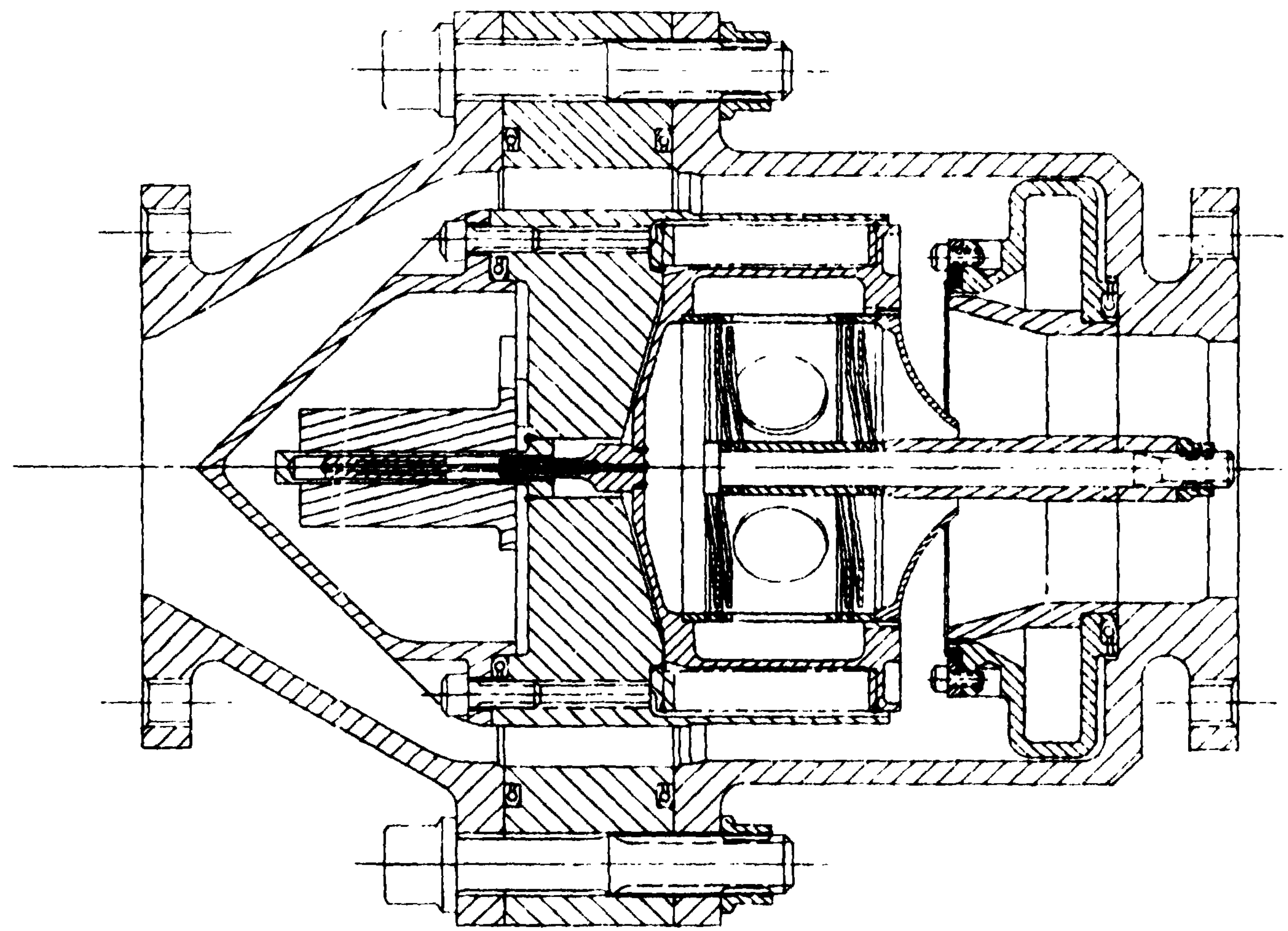


Figure 98 - Coaxial Poppet Valve, Vent to Open/Pressure to Close

10,000 cycles by observation of the LVDT trace on the oscilloscope. After 90,000 cycles of operation it was noted that the main valve poppet was no longer operating, although the three-way pilot valve continued to operate. Subsequent leak checks disclosed that the welded bellows assembly which is used as a dynamic seal on the main poppet was leaking excessively. The valve was subsequently disassembled to determine the nature of the bellows failure.

Examination of the failed bellows revealed that the fifth convolution from the stationary end of the bellows had cracked in the heat affected zone immediately adjacent to the inside diameter weld. Subsequent review of bellows processing and bellows sizing analysis resulted in the conclusion that the bellows vendor had stressed this bellows for a mean operating stress of 84,500 psi (58,260 N/cm^2) which greatly exceeded the allowable mean operating stress of 43,000 psi (29,647 N/cm^2) for one million cycles of operation. While the maximum operating stress of 141,000 psi (97,215 N/cm^2) was within the allowable stress of 150,000 psi (103,420 N/cm^2) for the Inco 718 material, it was incorrectly stressed for fatigue life and simply failed after approximately 85,000 cycles. A review of possible alternatives to obtain one-million-cycle bellows life within the Part No. X28400 valve configuration determined that this cycle life could be attained by increasing the bellows thickness from 0.008 inches (.0203 cm) to 0.010 inches (.0254 cm) and by reducing the valve operating stroke by a factor of 2.

To permit a satisfactory solution to the bellows problem and to permit completion of the extended cycling tests, a new bellows, correctly stressed, was obtained and installed in the prototype valve. Cycle testing was resumed and all components operated satisfactorily until 180,000 cycles were accumulated on the valve with the new bellows. At that point, the pilot valve ceased to operate. Upon disassembly of the pilot valve, it was evident that the valve was stuck in the pressurizing mode. It was also obvious that the failure was the result of self-generated contamination which had caused the stem to cease.

The nonoperating pilot valve was removed and replaced by a second pilot valve and testing was resumed. After an additional 220,000 cycles (total of 400,000) the second pilot valve failed. This valve failed in the vent mode and upon disassembly of the pilot valve, it was apparent that failure resulted from an undersized spacer used at the end of the stem. This spacer had shifted over to one side which resulted in very high localized stresses and resulting failure. Both pilot valves were then reworked, using the correct size spacer, cleaned and lubricated with Krytox. One of the valves was then reinstalled in the system and it operated satisfactorily throughout the remainder of the program. Leakage from the pilot valve was higher than desired throughout the extended cycling tests but that did not affect the operation of the prototype valve being evaluated.

Upon installation of the reworked pilot valve, testing was resumed. At about 500,000 cycles, the leakage rate of the P/N X28400 valve became excessively high. An examination of the valve revealed that the weld holding the stem which operates the LVDT armature had cracked. A close examination of the weld indicated that the weld was not concentric with the stem and that on the side where the crack appeared, there was very little fusion between the two parts. At that point, the part was rewelded and testing was resumed. No further difficulties were encountered and 1,000,000 cycles were achieved.

The leakage history of the prototype valve during the extended cycle tests is shown in Figure 99. As is evident from this figure except for the time when the stem weld of the LVDT armature cracked at around 500,000 cycles, the leak rate was much less than the allowable 100 scc/hour. During the leak checks, it was noted that the temperature of the gas affected the measured leak rate. The leak rate measurement was basically a volume measurement and if the leak check was made immediately after stopping the valve, when the gas was quite cold, a higher leak rate would be recorded because the gas warmed up and gave a higher reading. If the valve and gas were allowed to reach ambient temperature before the check was made, a lower and more nearly correct reading would be made. After the 1,000,000 cycles had been achieved, the valve was allowed to warm up before the leak check was made. This accounts for the apparent reduction in leak rate at the end of the test. Because of this temperature affect, the leak data shown in Figure 99 are believed to be conservative.

Valve response data are presented in Figure 100. The upper curves show the opening response and the lower ones the closing response. The opening response was slightly higher than the desired response of 30 m.s. This response was higher because the pilot valve flow orifice was smaller than specified. A slightly larger orifice would have resulted in a faster response of the prototype valve. Also shown in Figure 100 is the opening response of the pilot valve. As was mentioned previously, the pilot valve required replacing and rework. The increase in opening response of the X28400 valve at 400,000 cycles is the result of the pilot valve response. The opening travel time was consistently about 5 m.s. As is evident from Figure 100 , the closing response of the X28400 valve varied considerably. The decrease between 300,000 and 400,000 cycles resulted from the decrease in the pilot valve response. The reason for the increase in both closing response and travel time after 500,000 cycles is not known but evidently the rewelding and reassembly of the valve at that point resulted in a slower valve. The closing response was still below the design goal of 30 m.s. and the travel time was well below the design goal of 15 m.s. however.

The valve was still operating well when testing was stopped after 1,000,000 cycles were accumulated. Photographs of some of the parts are

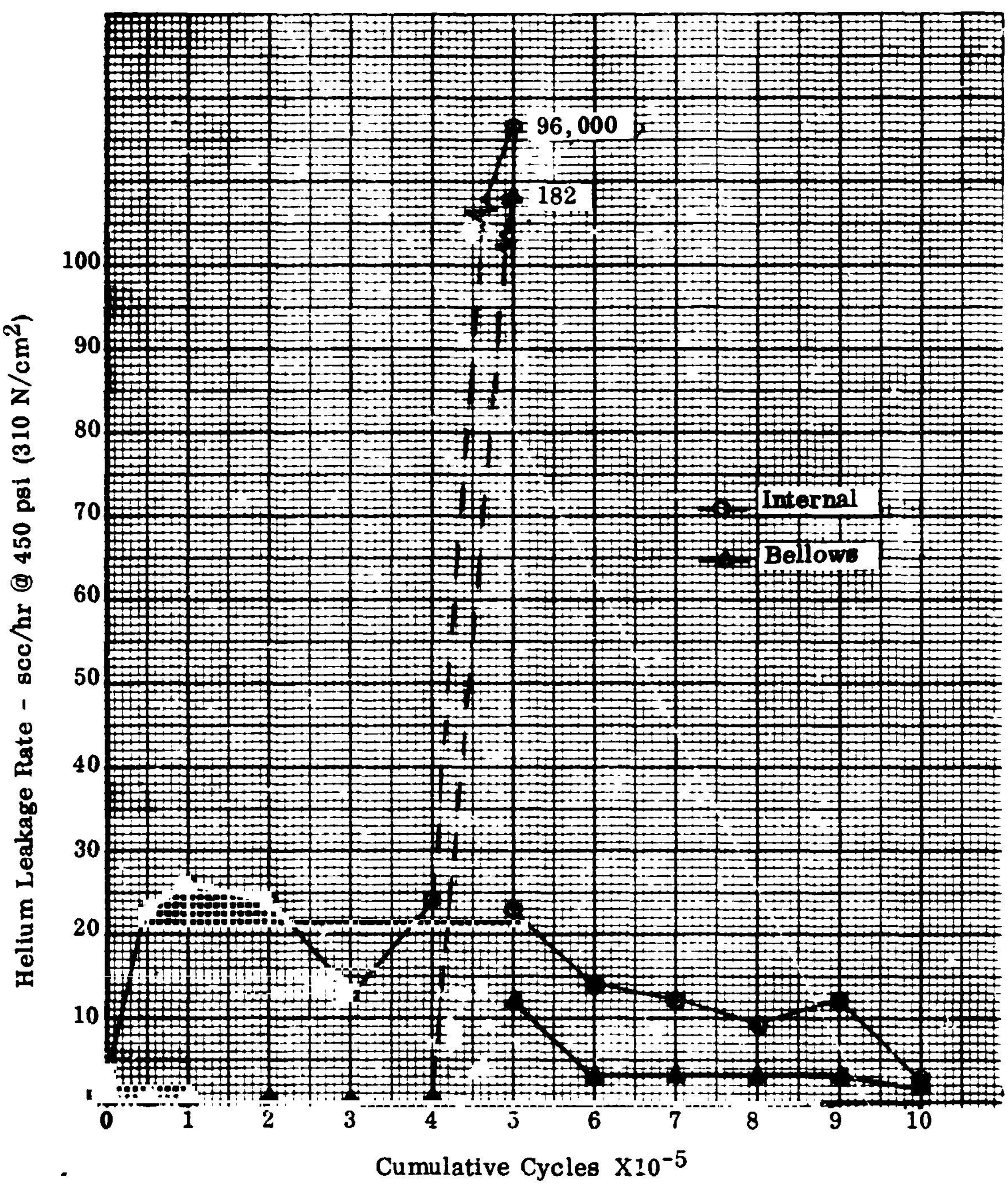


Figure 99 - Leakage Cycle History, P/N X28400 Extended Cycle Tests

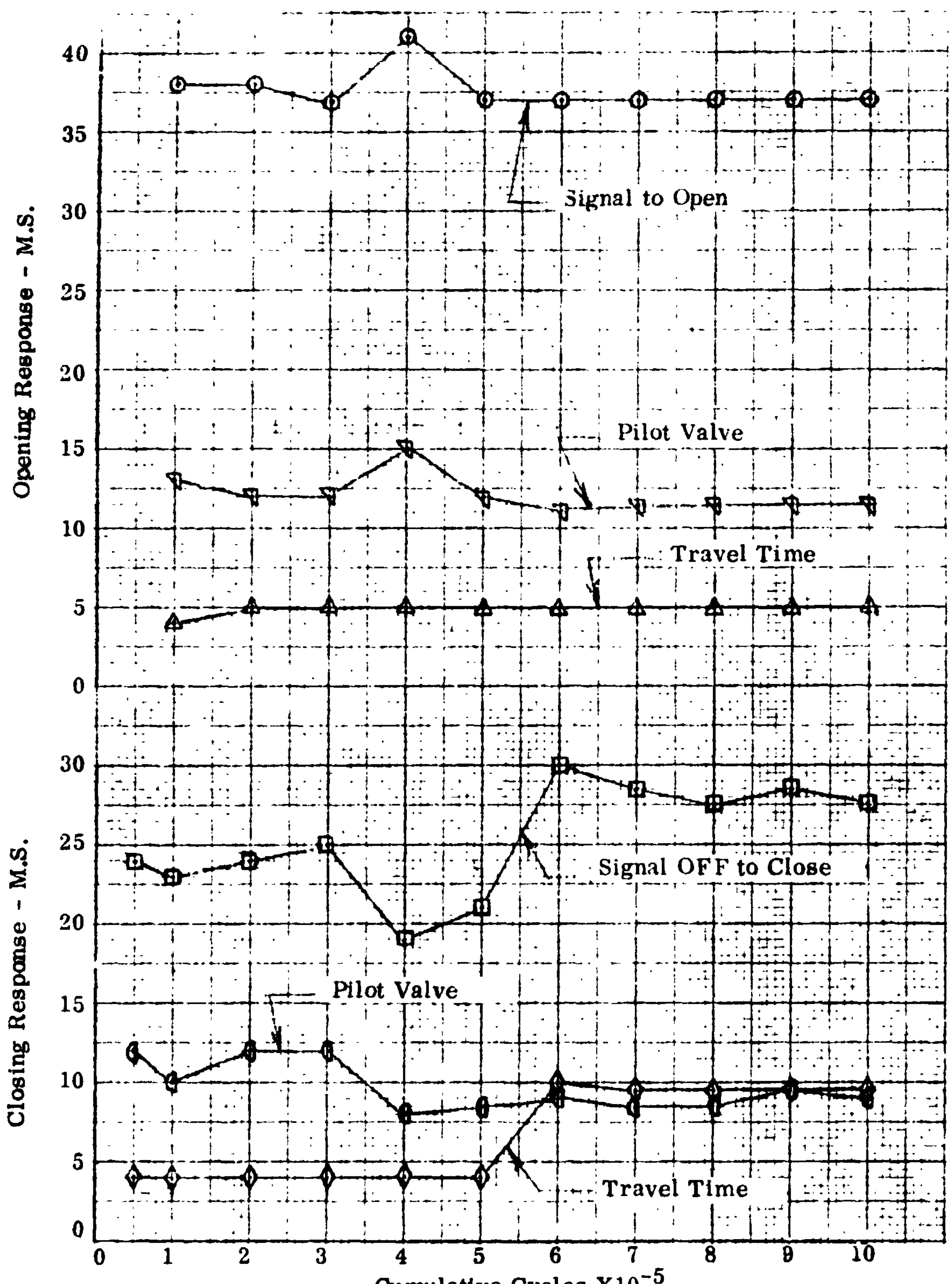


Figure 100 - Valve Response Results, P/N X28400 Extended Cycle Tests, 450 psi GN_2 (310 N/cm^2) and 28 V.D.C.

shown in Figures 101 through 103. Figure 101 shows the poppet seat (the outer black ring just inside the screw heads) which was polyimide. The inner black ring is a space between the poppet stop and the seat. The polyimide was in very good condition after the tests. Evidence of wear on the poppet stop is visible in Figure 101.

Figure 102 shows the poppet sealing surface. This surface was still in excellent condition. A fine deposit of polyimide is evident on the mirror-like finish of the sealing surface but no degradation in the surface occurred. The flexure, located inboard from the sealing surface also was in excellent condition following the test. Figure 103 shows the end of the bellows containing the poppet sealing surface. The bellows also was in good condition.

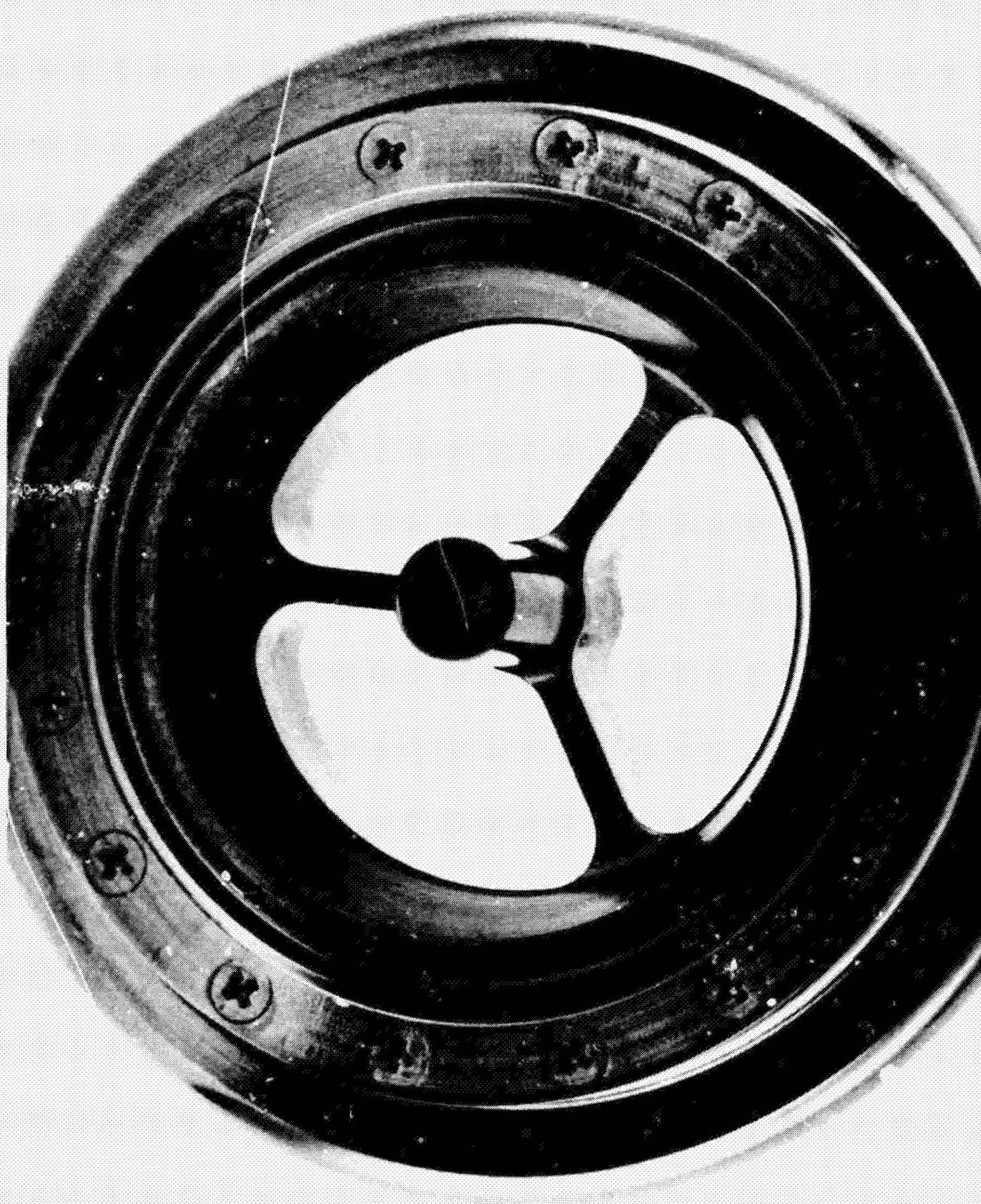
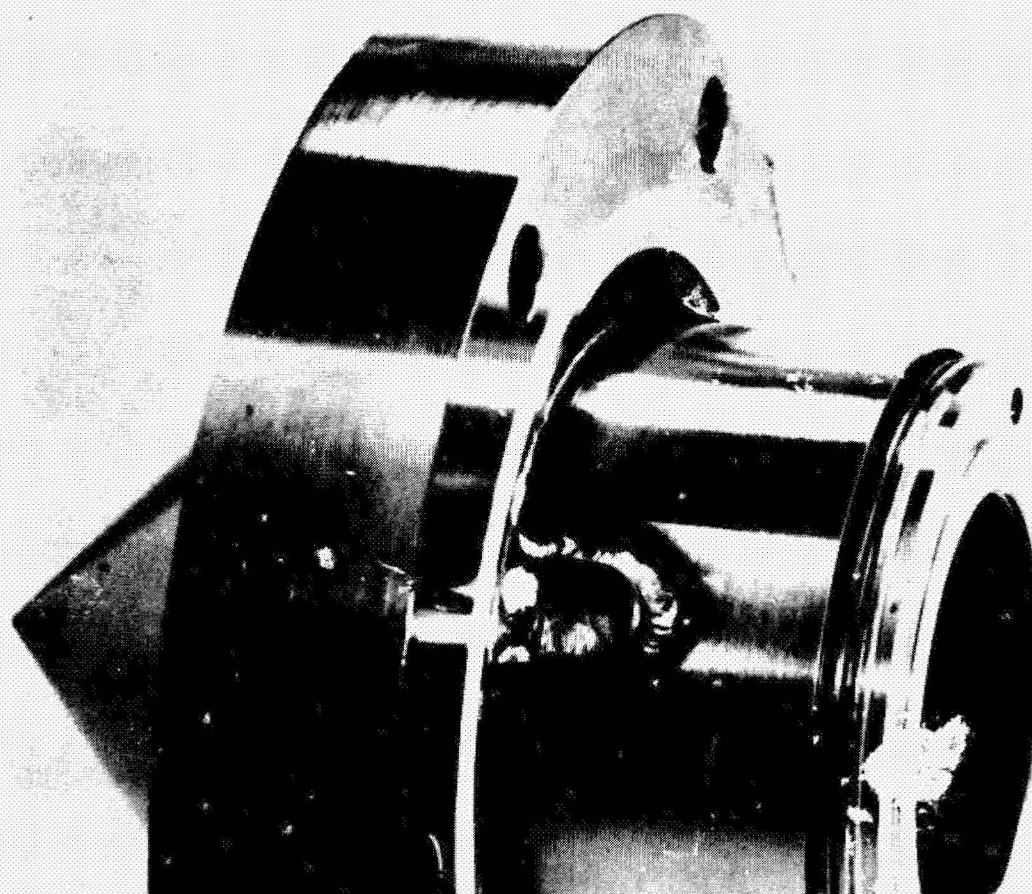


Figure 101 - Seat After 1,000,000 Cycles

Figure 102 - Poppet After 1,000,000 Cycles



(0 cm 2.54)

Figure 103 - Inner Body After 1,000,000 Cycles

VALVE DESIGN

Selection of Flightweight Valve Design

Upon completion of the test fixture test programs with both the P/N X27449 and X28400 Valves, but before the extended cycle tests with the P/N X28400 Valve, a data and design review was held with the NASA-LeRC Project Manager at The Marquardt Company. At that time, the data presented in the section entitled, "Valve Subcomponent Evaluation" and the section entitled, "P/N X28400 Test Fixture," were reviewed. The original purpose of the test fixture tests was to evaluate various valve subcomponents. For this reason, the two valves tested featured substantial differences in their components. Table XXII is a listing of these differences. The components listed on the left side of Table XXII were included in the P/N X27449 Valve and the components listed on the right side of Table XXII were included in the P/N X28400 Valve.

When a final comparison of these valve subcomponents was made, it was concluded that essentially all of the components featured in the P/N X28400 Valve were preferred to those of the P/N X27449 Valve. The reasons for this preference are briefly discussed in the following paragraphs.

While both static seals and welded seals functioned well during the test fixture cycling program, it is believed that static seals may be more susceptible to adverse effects from extensive thermal cycling, an environment that was beyond the scope of this contract, and that property changes in the teflon after ten years may be more significant than in metallic joints and may adversely affect static seal performance. Consequently, for the Space Shuttle application, a welded joint configuration is preferred. The sliding seals evaluated during this program proved to be unsatisfactory, whereas the bellows functioned well. While improvements to the sliding seal configurations could no doubt be made, there is no assurance at this time that the 100 scc's per hour of helium at 450 psi (310 N/cm^2) inlet pressure leakage rate can ever be met reliably for one million cycles at the high sliding velocities and over the temperature range of interest for the Space Shuttle Auxiliary Propellant Valves Program. Consequently, the bellows is preferred. Flexure poppet guidance is preferred to sliding poppet guidance since it assures repeatable and precise guidance of the poppet assembly. The repeatability of sliding poppet guidance will always be limited to the clearances between the sliding parts. Furthermore, sliding surfaces do generate contamination, whereas this undesirable characteristic is eliminated with the flexure guidance. In comparing a pressure balanced poppet with a pressure unbalanced poppet design, it is apparent that the pressure balanced poppet is a more complicated configuration since it requires at least one additional dynamic seal. Thus, the pressure balanced poppet would appear to be a less reliable configuration. Although it is recognized that pressure balancing does result in a faster response configuration, the response characteristics of the pressure un-

TABLE XXII - VALVE SUBCOMPONENT SUPPORTING PARTS TRADEOFF

P/N X27449 FEATURES	P/N X28400 FEATURES
• STATIC SEALS	• WELDED JOINTS
• SLIDING SEALS	• BELLOWS
• SLIDING POPPET GUIDANCE	• FLEXURE POPPET GUIDANCE
• PRESSURE BALANCED POPPET	• PRESSURE UNBALANCED POPPET
• SEALING CLOSURE NO.1	• SEALING CLOSURE NO. 2
• DOUBLE ACTION CYLINDER	• SINGLE ACTION CYLINDER SPRING RETURN
• POPPET STOP UPSTREAM OF SEAT	• POPPET STOP DOWN STREAM OF SEAT

balanced poppet are satisfactory for this application and consequently, this latter configuration is preferred.

Sealing Closure No. 2, the flat polyimide seat, is considered more desirable than the gold-plated lip seat since the polyimide appears to be inherently less contamination sensitive and since Marquardt was unable to conclusively demonstrate the optimum gold-plating during this program. In comparing a double-action cylinder with a single-action cylinder and spring return, it is again noted that the double-action cylinder is more complicated and, therefore, less reliable than the single-action cylinder and spring return. This is due primarily to the requirement of at least one additional dynamic seal and one additional pilot valve for the double-action cylinder. While the double-action cylinder does result in faster response characteristics, the response characteristics of the single-action cylinder and spring return are considered satisfactory and this configuration was, therefore, chosen for the Space Shuttle Auxiliary Propellant Valves application.

Finally, location of the poppet stop downstream of the seat appears more desirable than upstream of the seat, since the impact of the poppet on the stop will result in some contamination generation regardless of how minute this might be and since it is not desirable to pass this contamination through the poppet/seat interface. In summary, the valve subcomponents generally featured in the P/N X28400 Test Fixture were considered the preferred components and approval was received from the NASA-LeRC Project Manager for the preparation of a final flightweight valve design layout generally conforming with this configuration.

Design Layout

A detailed valve design layout of the final flightweight valve configuration is shown in Figure 104. This valve is of the coaxial configuration and features an integral three-way pilot valve. Examination of Figure 104 shows that the effective bellows diameter is greater than the effective sealing diameter at the poppet/seat interface. By venting the left side of the poppet through the three-way pilot valve overboard, the area between the effective bellows diameter and the sealing diameter is acted upon by the valve inlet pressure to cause an opening force upon the poppet/bellows assembly. This force accelerates the poppet until the poppet strikes the impact absorber. Once the pressure downstream of the valve builds up, an increased pressure force is available to lock the poppet in the open position. To close the valve, the left side of the poppet is simply repressurized with the valve inlet pressure by means of the three-way pilot valve and this pressure force, in combination with the spring forces from the axial guidance flexure and the bellows, return the poppet to the closed position. The poppet is stopped in the closed position by means of a poppet stop to minimize the impact energy transferred to the

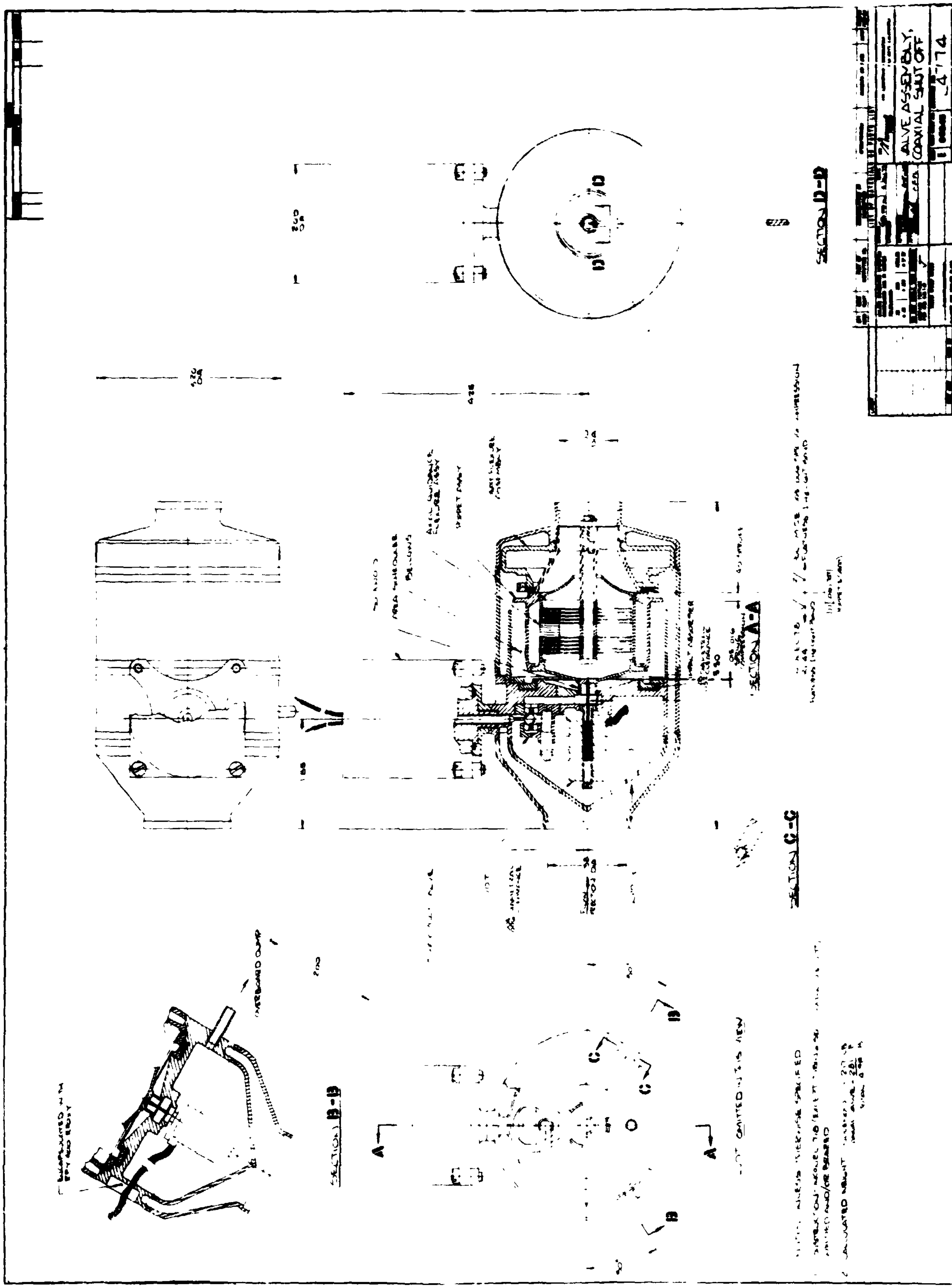

SECTION D-D
SECTION A-A
SECTION C-C
SECTION B-B
OVERBOARD DUMP
ALVE ASSEMBLY, COAXIAL SHUT OFF

Figure 101

valve seat. The valve seat, in effect, is mounted on a spring support structure which controls the poppet/seat interface load in the closed position. Guidance of the poppet/bellows assembly during the opening and closing motions is provided by means of an axial guidance flexure assembly which, in turn, is supported on a centrally located shaft rigidly attached to the outlet of the valve. A mechanical tang has been incorporated in the nut which makes this shaft assembly rigid to prevent the nut from loosening during valve cycling.

The valve configuration shown in Figure 104 also features an integral LVDT transducer to monitor valve position. Although the valve mechanism of the three-way pilot valve has been incorporated integrally with the main valve, the solenoid actuator required to operate the pilot valve is still located external to the main valve and can be readily removed. Except for a single static seal employed between this solenoid actuator and the main valve body, the flightweight valve design is a completely welded assembly using a series of electron beam (EB) welding operations to provide the required strength, cleanliness, and welding quality needed for such hardware. The principal material of construction is Inco 718, heat-treated and welded in the heat-treated conditions. This material was selected because of its demonstrated structural properties in hydrogen and oxygen environments at the temperatures and pressures specified for operation. Other materials used .. the construction of this valve are Type 321 stainless steel for the pilot valve inlet tubing, polyimide for the valve seats, Type 430 stainless stee' or magnetic parts of the solenoid, 303/304 stainless steel for the non-magnetic parts of the solenoid, a Type 440-C stainless steel ball for the pilot valve poppet, copper and nichrome coil wiring in the solenoid, Carpenter HP 49 material electrolized in the LVDT armature, Type 316 stainless steel in the LVDT pressure vessel, Type 304 stainless steel in LVDT cover, and copper windings in the LVDT.

In general, the valve design is not particularly subject to critical fits and dimensions. The only critical specification relates to the sealing closure interface as defined by the surface finish, flatness and parallelism of the sealing surfaces. These are called out in Figure 104. The valve consists of three subassembly components:

- Centerbody assembly
- Poppet assembly
- Seat assembly - Outlet housing

A pilot valve solenoid, a position indicating LVDT, and assorted shell elements complete the valve. The total valve weight is calculated to be 4.98 lbs (2.26 Kg). Interface definition lacking, the layout shows sockets for the connections of supply and discharge tubing. Alternate arrangements for mounting the valve on rocket injector pads can be made when the engines are selected.

The sequence of assembly and the important processes are:

1. Centerbody Assembly (Inlet Side)

   a. The LVDT armature diaphragm is pressed into its pilot diameter socket and EB welded.

   b. The pilot valve subassembly with a straight pressure supply tube is then formed into the flexible shape shown and aligned with the hole in the conical-inlet shell.

   c. The LVDT coil assembly is installed, its electrical responses verified and then EB welded into three tank standoffs.

   d. The conical inlet shell is EB welded to the centerbody and then the pilot pressure supply tube is welded to the cone shell using the inert gas technique.

   e. The outer inlet housing is EB welded to the centerbody.

2. Centerbody Assembly (Discharge Side)

   a. The poppet impact absorber is EB welded.

   b. The impact absorber and the centerbody stop surfaces are machined to provide the 0.015 inch (.0381 cm) offset deflection dimension needed for energy absorption.

   c. The poppet assembly is installed and EB welded to the centerbody.

   d. The bellows shroud is installed and EB welded at its three strut locations.

   e. The seat assembly-outlet housing structure is EB welded to the outer diameter of the centerbody.

   f. The pilot valve solenoid adapter is EB welded to the centerbody boss projection.

3. Poppet Assembly

   a. The axial guidance flexures are assembled with foil braze alloy, vacuum furnace brazed and heat treated after brazing.

   b. The axial guidance flexure cartridge inside and outside diameters are machined after the end diaphragm and flow guide are EB welded to the flexure ring extensions.

   c. The bellows assembly is inert gas welded to the poppet and the poppet adapter flange.

   d. After helium leak check, the poppet diameters are machined to mate with the flexure assembly.

e. The poppet is lapped and polished to the required surface finish. Surface protection is maintained from this operation on.

f. The poppet is EB welded to the flexure assembly at the diametral joint remote from the poppet sealing surface.

g. Installation of the draw bolt into the flexure assembly is followed by EB welding the LVDT armature assembly into the poppet diaphragm.

h. The poppet assembly is assembled under 2-c above.

4. <u>Seat Assembly - Outlet Housing</u>

a. The outlet housing, seat adapter and spring support, and poppet stop are assembled and vacuum furnace brazed and heat treated.

b. The brazed assembly is machined to provide true pilot diameters needed for final assembly.

c. The polyimide seat is installed, and lapped and polished to the required surface finish. Seat protection is maintained.

d. The outerbody shell extension is EB welded to the outlet housing.

e. The shell extension pilot diameter is machined in a final turning operation.

f. The seat assembly is matched to the centerbody assembly under 2-c above.

g. The draw bolt antirotation key is installed and the draw bolt torqued to its required preload. The key tank is inert gas welded to the retaining hex nut.

Installation of the pilot valve solenoid completes the valve assembly.

Refurbishment of either the poppet or seat can be accomplished by removing the shell extension and remachining its fit-up diameters. The poppet and the seat are then exposed for further rework, refurbishing or replacement. Rebuilding the valve follows the above outlined procedure.

Performance characteristics of this flightweight valve conform to the contract requirements as listed in Table XXI.

CONCLUSIONS AND RECOMMENDATIONS

The Advanced Technology for Space Shuttle Auxiliary Propellant Valves Program successfully demonstrated a prototype valve test fixture for one million cycles of operation. Furthermore, the inherent design features of this test fixture have the capability of operating for ten years with zero maintenance as required by the Space Shuttle mission. In generating the valve technology to permit this demonstration the subject program utilized analytical models for the static leakage through a sealing closure interface as well as for the sealing closure interface degradation due to wear. Furthermore a substantial number of valve subcomponents showing promise for long life with zero maintenance capability were evaluated. In addition to the excellent leakage results obtained during this program, very fast response characteristics were also demonstrated. Total opening or closing responses of 11 milliseconds at operating pressures of 450 psia (310 N/cm^2) for this size of valve had not been demonstrated previously.

The program again demonstrated that a thorough initial analytical effort to define the valve subcomponent configurations is essential to the successful development of high capability valves. This was particularly evident in the case of the critical bellows assembly utilized for the one million cycle life demonstration during this program. Sufficient experimental data was obtained during this program to substantiate the analytical model for static leakage through a sealing closure. However, insufficient data was obtained to confirm the wear model for the sealing closure interface. Indications are that the wear model as it was defined during this program is too conservative, at least as far as plastic sealing closures are concerned. A significant contribution to valve technology could be made by planning a program around Marquardt's analytical wear model to gain a better understanding of the significance of such parameters as scrubbing distance, impact loads, surface finishes, and material characteristics for very high life cycle valves.

Although sealing closure contamination sensitivity was a topic of major concern during this program, the program scope did not permit a significant experimental evaluation of this matter. Some contamination sensitivity data was obtained with the polyimide sealing closure as a result of accidentally introduced contamination and this data indicated the sealing closure to be quite contamination resistant. Specifically, several particles in excess of 75 microns in size were observed imbedded in the polyimide without any noticeable rise in leakage rates. The program, however, in effect, did not yield any significant quantitative data regarding contamination tolerance of the sealing closures. It would be desirable to conduct a contamination tolerance program in the future which would permit a comparison between various sealing closure materials such as polyimides, plastics, metallics, ceramics, and such newer materials as AFE 124-D and AFE 102 to obtain quantitative contamination tolerance data defining both the quantity and the size of particles that can be tolerated without significant leakage performance degradation.

The subject program concentrated on optimizing the main shutoff valve. The pilot valve required for operation of the shutoff valve was procured from a subcontractor. The pilot valve design features included several sliding fits which resulted in the generation of large amounts of contaminant particles during the life cycle program. This contamination in turn resulted in leakage performance degradation and the jamming of one of the pilot valves. It would be desirable to optimize the pilot valve design and to eliminate the sliding fits in a manner similar to that employed on the main valve so that reliable performance of the pilot valve for one million cycles can also be demonstrated. Along with this effort the possibility of building an integral pilot valve/main valve configuration such as is shown for the flightweight valve design layout should be pursued. Finally, the complete flightweight valve design should be fabricated and test evaluated to demonstrate its performance capability and to substantiate the calculated weight.

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APPENDIX A

SEALING CLOSURE STRESS ANALYSIS

AND

APPENDIX B

RAPID SCREENING TESTER STRESS ANALYSIS

SUMMARY

Appendix A - Sealing Closure Stress Analysis

Appendix B - Tester Stress Analysis

STRESS MEMO #326

INTRODUCTION

The attached stress analysis was made in support of the 5085 Valve Seal Program. The analysis was based on the following criteria:

1.	Nominal Test Conditions:	Stroke	=	.125 in.
		Accel. (2# wgt)	=	5.2 g's
		Travel Time	=	11 ms
		K.E. at Closure	=	500# $in^2/sec^2/g$
		Seat Force (Static):		20 - 200#
2.	Maximum Test Conditions:	Stroke	=	.500 in.
		Accel. (2# wgt)	=	6.5 g's
		Travel Time	=	20 ms
		K.E. at Closure	=	2500# $in^2/sec^2/g$
		Seat Force (Static)		200 - 1500#

3. Life 10^6 cycles per configuration.

4. Seat and Poppet capable of withstanding 450 psi operating pressure for 10^6 cycle.

5. Each Seal Lip was analyzed for a nominal deflection of .005 in. at the bearing surface, with over travel deflections to .008 in. as applicable for the design.

6. Proof Pressure = 1.5 Times Operating Pressure

7. Burst Pressure = 1.33 Times Proof Pressure

8. Temperature Range = $200^{\circ}R$ to $850^{\circ}R$

SUMMARY OF RESULTS

The table of margins is shown on Page 3. The margins are considered satisfactory for the purpose of meeting the structural criteria.

METHOD OF ANALYSIS

The structural analysis of the poppet and seat was based on the forces resulting from the absorbtion of 2500 lbs in^2/sec^2 kinetic energy at impact, for 10^6 cycl. The spring-supported seats were conservatively assumed to obtain the maximum poppet velocity at the moment of impact. This velocity was then used to calcul the kinetic energy of the seat-spring system. The kinetic energy of the seat was then used to calculate the reaction force from the seat support springs based on the relationship:

$$K.E. = \frac{1}{2} P\delta$$

where P = Impact (Reaction) Force

δ = Deflection of Seat Spring System for a Unit Load

The Reaction Force thus calculated was then used as a static load to determine the stresses in the various components, by means of conventional structural analysis methods.

The tester jacket was analyzed for the critical proof pressure of 670 psi because of the low allowable yield stress of the 304-L CRES material. Ultimate (Burst) stresses were not critical.

THE Marquardt COMPANY VAN NUYS, CALIFORNIA

TABLE 1

MINIMUM STRESS MARGINS - SEALING CLOSURES

Part Name	Condition	Remarks	Stress (ksi) of Loads (k)	Margins of Safety
Poppet Support	KE = 2500 lbs/in.	.25 t Plate	71.2	.96
Belleville Spring	Seat Load + 2500 lbs/in.	Bending on Spring	166	.20
Tung. Car. Poppet	600 psi Proof	Internal Press. Min. "t"	.051 in.	.88
Lip Seal	600 psi	Bending on Tip	200	.09
Spherical Lip Seal	Radial Δ = .00433 in.	Bending on Seal	138	.015
Flat Plate L = .100	y = .0065	Bending on Plate t = .013	71.5	.96
Flat Plate L = .180	y ≃ .006	Bending on Plate t = .0125	140	.00
Flat Plate L = .55	y = .008	Bending on Plate t = .053	141	.00

TABLE 2

MINIMUM STRESS MARGINS - TESTER

Part Name	Condition	Remarks	Stress (ksi) of Loads (k)	Margins of Safety
Tester Flexure	y = . .500	Bending + Torsion on Flex 10^6 cycles	78	.00
Tester Jacket	650 psi proof	Bending on Inner Wall	33.5	.07
Tester Jacket	Inner Plate	Bending on Plate + Web	20	.75
Attach Bolts	670 psi proof	THD Shear in Plate Yield	18.3	.30
Tester End Cap	600 psi proof	Bending on End Cap Min. "t"	.032	Large
Tester Poppet Support	Max. Impact	Column Load on Support tube	27.7	.08
Coil Spring	Solid Height	Torsion Stress	78	.005

APPENDIX A

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APPENDIX B

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THE MARQUARDT CORPORATION

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REPORT

DATE 8 Dec 58

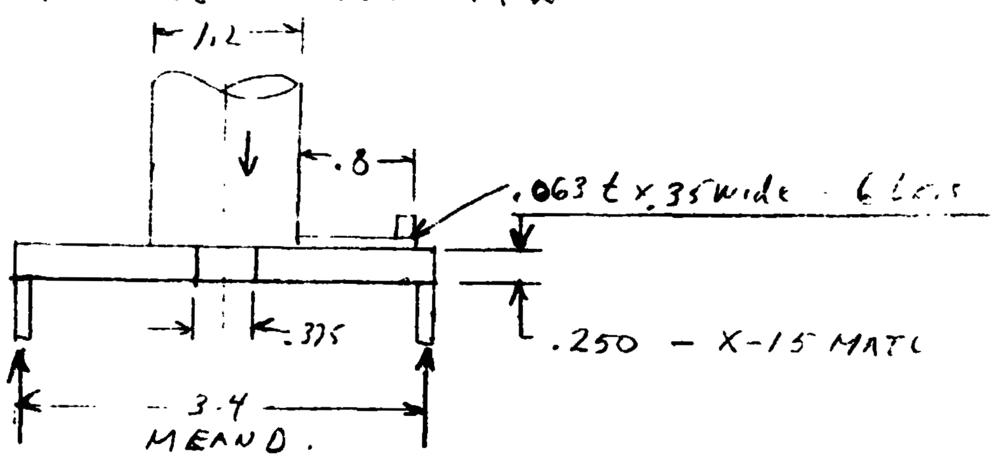
VALVE SEAL PROGRAM.

POPPET IMPACT AND SEAT FORCE ANALYSIS

REVISED MAXIMUM ENERGY CONDITION

KE MAX $2500/386$ = 6.5 LBS IN. MASS

SEAT FORCE = 1500# MAX



ANALYZE PER CASE 59, Roark pg 232, AND A BEAM

$a = 1.7$, $a^2 = 2.9$ $\qquad m = \frac{1}{\nu} = 3.33$, $m^2 = 11.1$

$b = .188$, $b^2 = .0352$ $\qquad t^3 = (.25)^3 = .01563$, $t^2 = .0625$

$r_0 = .6$, $r_0^2 = .36$ $\qquad E = 30 \times 10^6$

For PLATE

$$y = \frac{3W(m^2-1)}{2\pi E m^2 t^3}\left[\frac{(a^2-b^2)(3m+1)}{2(m+1)} - (b^2+r_0^2)\log_e\frac{a}{r_0} + (b^2-r_0^2) - \frac{r_0^2(a^2-b^2)(m-1)}{2a^2(m+1)}\right]$$

$$- \frac{CM(m^2-1)}{Emt^3}\left[\frac{b^2}{m+1} + \frac{2a^2b^2}{(a^2-b^2)(m-1)}\log\frac{a}{b}\right]$$

$$M = \frac{W}{8\pi m}\left[(m+1)\left(2\log_e\frac{a}{r_0} + \frac{r_0^2}{a^2} - 1\right)\right]$$

TMC A668

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PAGE 1.2

REPORT

DATE 8 DEC 1970

VALVE SEAL PROGRAM

POPPET IMPACT ANALYSIS – MAX ENERGY COND.

PLATE DEFLECTION

$$M = \frac{W}{8\pi\, 3.33}\left[(4.33)\left(2\log_e\frac{1.7}{.6} + \frac{.36}{2.9} - 1\right)\right]$$

$$= \frac{W}{83.6}(5.21)$$

$$M = .0623\,W$$

$$y = \frac{3W(10.1)10^{-6}}{2\pi(11.1)(.0563)}\left[\frac{(2.9-.035)(11)}{2(4.33)} - (.035+.36)\log_e\frac{1.7}{.6} + (.035-.36) - \frac{.36(2.9-.035)(2.33)}{2(2.9)(4.33)}\right]$$

$$- \frac{6\times.0623W(10.1)10^{-6}}{30(3.33)(.0563)}\left[\frac{.0352}{4.33} + \frac{2(2.9)(.0352)}{(2.9-.0352)(2.33)}\log_e\frac{1.7}{.6}\right]$$

$$y = \frac{W\,10.1\times10^{-6}}{30\times.0563}\left[.043\left[3.63 - .41 - .325 - .0955\right] - .112\left[.00813 + .0306\times2.205\right]\right]$$

$$y = 21.5\,W\times10^{-6}(.121 - .00548)$$

$$y_{PLATE} = 2.42\,W\times10^{-6}$$

C A669

VALVE SEAL PROGRAM

POPPET IMPACT ANALYSIS - MAX. ENERGY COND

LEG DEFLECTION

$y = \frac{WL^3}{3EI}$ PER LEG OR $\frac{WL^3}{18EI}$ /LEG FOR 6 LEGS

$l = .8$, $l^3 = .51$

$E = 30 \times 10^6$

$I = \frac{.35}{12}(.063)^3 = 7.3 \times 10^{-6}$

$y = \frac{W(.51)}{18 \times 30 \times 7.3} = \underline{129 \times 10^{-6} W}$

TOTAL STIFFNESS = $129 + 2.42 = 131.42$

y TAKEN BY PLATE = $\frac{129}{131.42} \times 2.42 W \times 10^{-6} = \underline{2.38 W \times 10^{-6}}$

y TAKEN BY BEAMS = $\frac{2.42}{131.42} \times 129 W \times 10^{-6} = \underline{2.38 W \times 10^{-6}}$

$KE = \frac{1}{2} P \delta \qquad \delta = 2.38 P \times 10^{-6}$

$6.5 = \frac{1}{2}(2.38 \times 10^{-6}) P^2$

$P^2 = \frac{13.0 \times 10^6}{2.38} = 5.45 \times 10^6$

$\underline{P = 2330\,\#}$ IMPACT

TOTAL FORCE = $2330 + 1500 = \underline{3830\,\#}$

VALVE SEAL PROGRAM

POPPET IMPACT ANALYSIS – MAX ENERGY CONDITION

STRESS ON PLATE FOR 3830# TOTAL FORCE

$$S_t = -\frac{3W}{2\pi m t^2}\left[\frac{1}{2}(m-1) + (m+1)\log\frac{a}{r_0} - (m-1)\frac{r_0^2}{2a^2}\right] - \frac{6M(a^2+b^2)}{(a^2-b^2)t^2}$$

$$S_t = \frac{-3(3830)\times\frac{129}{131.4}}{2\pi\, 3.33(.0625)}\left[\frac{1}{2}(2.33) + 4.33(1.04) - (2.33)(.0442)\right]$$

$$-\; 6(.0623)(3830\times\frac{129}{131.4})\frac{(2.4 + .052)}{(2.4 - .0352)(.0625)}$$

$$= \frac{3760}{.0625}\left[-.1442(5.575) - .374(1.023)\right]$$

$= 71200$ psi

Fend = 140,000 psi FOR X-15

$$M.S. = \frac{140}{71.2} - 1 = .96$$

FOR BEAMS – INCO 718 MATL

$W = \frac{2.42}{131.4}\times 3830 = 70.5\,\#/6 = 11.8\,\#/\text{BEAM}$

$M = 11.8 \times .8 = 9.44$ in#

$f_b = 9.44(.031)/7.3\times10^{-6} = 40000$ psi

Fend = .60 × 185 = 110,000 psi UNI-DIRECTIONAL

M.S. = LARGE

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VALVE SEAL PROGRAM

IMPACT FORCES ON FLAT ON FLAT IVC SEAT AND SUPPORT

NORMAL OPERATING CONDITION

ASSUME SEAT ATTAINS MAXIMUM VELOCITY OF POPPET AT IMPACT. STROKE .125 IN. t = .011

$V_{max} = 2 \times .125 / .011 = 22.3$ in/sec

MASS OF SEAT AND SUPPORT = .16 #/g

= 415×10^{-6} SLUGS

$K.E. = \frac{1}{2} MV^2 = \frac{1}{2}(415)10^{-6}(22.3)^2 = .104$ LBS-IN.

FROM DWG, SPRING RATE BELLOWS = 4600 #/in

$\delta/\# = \frac{P}{4600} = .000217 P$ in/#

$KE = \frac{1}{2} P\delta = \frac{1}{2}(.000217)P^2 = .104$

$P^2 = .208 / (.000217) = 960$

$P = 31\#$

$\delta = 31 \times .000217 = .00675''$

ABOVE NEGLECTS SMALL PRELOAD ON BELLOWS

FORM TMC 885 REV. 3-70

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VALVE SEAL PROGRAM

IMPACT FORCES ON FLAT ON FLAT WC SEAT AND SUPPORT -

MAXIMUM OPERATING CONDITION

FOR TOTAL ARMATURE $KE = 2500/g$ ($W_{ARM} = 2$#)

$V_{MAX} = 50$ in/sec by calculation

FOR SEAL MASS + $\frac{1}{3}$ SPRING MASS $= .20$#$/g$

$V_e = \frac{.157}{.20}(50) = 39.3$ in/sec , $KE = \frac{1}{2}\frac{(.20)}{386}(39.3)^2 = .400$ in-lb

SPRING RATE TOTAL $= 88000 + 4600 = 92600$ #/in.
(Belleville + ...)
(M1-11)

PRELOAD ≈ 1500 # SEAT FORCE

$KE = \frac{1}{2}P\delta + 1500\delta$

$\delta = \frac{1}{K} = \frac{1}{92600} = 10.8 \times 10^{-6}$

$.400 = \frac{1}{2}(10.8\times10^{-6})P^2 + 1500(10.8\times10^{-6}P)$

$.400 = 5.4\times10^{-6}P^2 + 16200\times10^{-6}P$

$5.4P^2 + 16200P - 400{,}000 = 0$

SOLUTION

$P = \frac{-16200 \pm \sqrt{(262)10^6 + (8.64)10^6}}{2\times5.4}$ $\frac{(-3000)}{2\times}$

$P \approx 24.0$ # (TOTAL LOAD = 1524#)

$\delta_{IMPACT} = 24 \times 10.8\times10^{-6} = .000260''$

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VALVE SEAL PROGRAM

IMPACT FORCES ON FLAT ON FLAT WC SEAT AND SUPPORT

MAXIMUM OPERATING CONDITION — 200 # PRELOAD SEAT FORCE

$K = 4600$ #/IN.

LET $K.E = \frac{1}{2} M V^2$

Wgt SEAT = .16#

v max = 50 in/sec

$$KE = \frac{1}{2}\frac{(.16)(50)^2}{386} = .518 \text{ LBS-IN}$$

$$KE = \frac{1}{2} P\delta + 200\,\delta$$

$$\delta = \frac{1}{K} = \frac{1}{4600} = .217\times10^{-3} P$$

$$.518 = \frac{1}{2}(.217\times10^{-3})P_1^2 + 200(.217\times10^{-3})P_1$$

$$.518 = .1085\times10^{-3} P_1^2 + 43.4\times10^{-3} P_1$$

$$.1085 P_1^2 + 43.4 P_1 - 518 = 0$$

SOLVE

$$P_1 = \frac{-43.4 \pm \sqrt{(43.4)^2 + 4(518\times.1085)}}{2\times.1085}$$

$$= \frac{+2.5}{.217}, \quad -\frac{89.3}{.217} \text{ NOT REAL}$$

$$P_1 = 11.5^{\#}$$

$$\delta_1 = 11.5\times.217\times10^{-3} = .00250 \text{ IN}$$

CHECK $E = \frac{(200+211.5)}{2}(.00250) = .514$ LBS-INS ✓

$P_{TOTAL\ MAX} = 211.5^{\#}$

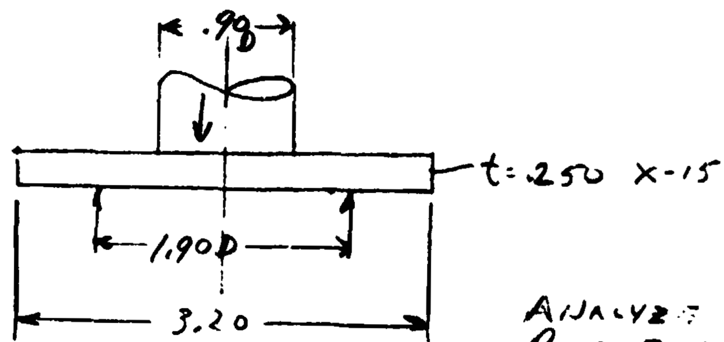
VALVE SEAL PROGRAM

POPPET IMPACT AND SEAT FORCE ANALYSIS

FLAT POLYIMIDE SEAT CONFIGURATION

K.E. MAX = $2500/386$ = 6.5 LBS-INS.

SEAT FORCE ≅ 1000#



ANALYZE AS CASES I, II
Prod. Eng'g JUNE 11, 1962

DEFLECTION AT IMPACT DIAM.

$$y \cong \frac{1.90}{3.20}\left[K\frac{WR^2}{Et^3}\right]$$

Where $W_I = P = W_{II}$

$R^2 = (1.6)^2 = 2.56$

$Et^3 = .47 \times 10^{-6}$

$a/r_{0_I} = \frac{3.2}{1.9} = 1.73$

$a/b_{II} = \frac{3.2}{.9} = 3.5$

$K_I = .38$

$K_{II} = .25$

$$y = .6\left[\frac{P(2.56)}{.47\times10^{-6}}\right](.38 - .25)$$

$$\underline{y = .425\times10^{6}P} \qquad ①$$

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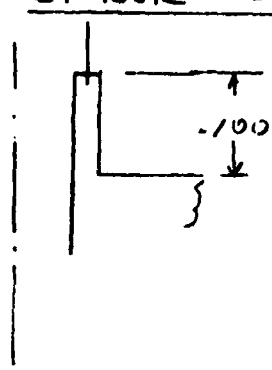
VALVE SEAL PROGRAM

POPPET IMPACT AND SEAT FORCE ANALYSIS

FLAT POLYIMIDE SEAT CONFIGURATION

DEFLECTION OF SEAT AREA

INNER LIP

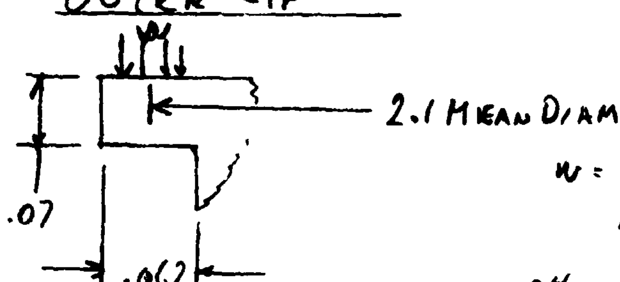


MEAN DIAM = 1.845 , t = .035 AVER

$$y = \frac{PL}{AE} = \frac{P(.100)}{\pi(1.845)(.035)\,30\times10^{6}}$$

$$= .0164\times10^{-6}\,P$$

OUTER LIP



$$w = \frac{P}{\pi(2.1)(.062)} = 2.44P \text{ psi}$$

$$y = \frac{wl^4}{8EI}\times\frac{1}{2}$$ FOR EFFECTIVE δ OF POPPET CONTACT POINT

$I = \dfrac{1\times(.07)^3}{12}$

$I = 28.4\times10^{-6}$

NEGLECT SHEAR DEFLECTION SINCE LOAD POINT WILL VERY CLOSE TO FIXED END.

$$y = \frac{2.44(.062)^4 P}{16\times30\times28.4} = .00264\times10^{-6}\,P$$

P DISTRIBUTES INVERSELY TO "y"

VALVE SEAL PROGRAM.

POPPET IMPACT AND SEAT FORCE ANALYSIS

DEFLECTIONS OF SEAT AREA

DISTRIBUTION OF IMPACT FORCE

TO INNER LIP $$P' = \frac{.00264\ P}{(.00264 + .0164)} = .139\ P$$

TO OUTER LIP $$P' = \frac{.0164}{(.00264 + .0164)} = .861\ P$$

NET y INNER LIP = $.0164 \times .139 \times 10^{-6} P = .00228 \times 10^{-6} P$

NET y OUTER LIP = $.00264 \times .861 \times 10^{-6} P = .00228 \times 10^{-6} P$

TOTAL DEFLECTION POPPET AND LIPS.

$y_{TOT} = (.425 + .00228) 10^{-6} P = .427 \times 10^{-6} P$

$KE = \frac{1}{2} P \delta$ $\quad \delta = y_{TOT}$, $KE = 6.5$ LB. IN.

$6.5 = \frac{1}{2}(.427) 10^{-6} P^2$

$\frac{13}{.427 \times 10^{-6}} = P^2$

$P = 5500\#$ IMPACT

TOTAL FORCE = $5500 + 1000 = 6500\#$

VALVE SEAL PROGRAM

POPPET IMPACT AND SEAT FORCE ANALYSIS

FLAT POLYIMIDE SEAT CONFIGURATION

BEARING ON OUTER LIP - X-15 MATL

LET $P = 6500^{\#}$, $A_{br} = \pi(2.1)(.05) = .33$

$f_{br} = 6500/.33 = \underline{19700 \text{ psi}}$

Shear: $A_s = \pi(2.1)(.07) = .46$

$f_s = \frac{6500 \times 1.5}{.46} = \underline{21200 \text{ psi}}$

BENDING

$M = 6500 \frac{(.062)^2}{2} = 12.5 \text{ in}^{\#}/\text{in}$

$f_b = 6(12.5)/(.07)^2 = 15300 \text{ psi}$

STRESS CONCENTRATION FACTOR = 2.0

MAX $f_b = 2 \times 15300 = \underline{30600 \text{ psi}}$

X-15, $F_{ty} = 200,000$ psi

M.S. = LARGE

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VALVE SEAL PROGRAM.

BELLEVILLE SPRING ANALYSIS

FOR MAXIMUM ENERGY CONDITION

SEAT LOAD = 1000# ALLOW FOR IMPACT = 1500# TOTAL

USE MAXIMUM IMPACT + SEAL LOAD = 2000#

$t^2 = \frac{.96P}{S_n}$ $S_y = 200{,}000$ psi FOR Custom PH 15-7 Mo X-15

$t^2 = \frac{.96(2000)}{200{,}000} = .0096$

$t = .098$ — USE $t = .090$

FOR $R/r = \frac{2.88 \text{ OD}}{2.40} = 1.2$ $C = 3.5$ $\left[\begin{matrix} OD = 2.88 \\ ID = 2.40 \end{matrix}\right]$

$P = C_1 C E t^4 / R^2$ $\frac{t = .090}{E = 30 \times 10^6}$, $t^4 = 6560 \times 10^{-8}$

$R^2 = (1.44)^2 = 2.07$

$h/t = 1.25$ $h = .1125$

$1500 = \frac{C_1\, 3.5 \times 30 \times 10^6 (6560) 10^{-8}}{2.07}$

$C_1 = \frac{3100}{3.5 \times 30 \times 6560 \times 10^{-2}} = .45$

$\delta/t = .19$, $\delta = .0171$/spring $R = 1500/.0171 = 88000$ #/in

$S_T = 1.1 E \delta C \frac{[+.9(.1125 - .0085) + .875 \times .09]}{R^2}$

$= [1.1(30)(.0171) 3.5 (.1731)/2.07] 10^6$

$= 163{,}000$ psi ok

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VALVE SEAL PROPOSAL

BELLEVILLE SPRING ANALYSIS

FOR MAXIMUM ENERGY CONDITION – IMPACT ON SPRING

$P = 2000\# = C_1 C E \frac{t^4}{R^4}$ FOR PRELOAD = 1000#

$C_1 = \frac{2000}{1500} \times .45 = .60$,

$\delta/t = .25$ FOR $h/t = 1.25$

$\delta = .25 \times .090 = .0225$

$S_{T2} = 1.1 E C \delta \left[.9(.1125 - .01125) + .875 \times .090\right] / R^2$

$= 1.1 \times 30 \times 10^6 (3.5)(.0225)\left[.1696\right] / 2.07$

$= 213000$ psi

CLOSE ENOUGH FOR NOW!

SINCE MAX LOAD = 1530# TOTAL (+ IMPACT)

(REF P 1.5.)

MAX S_{T2}: $1530/1500 \times 163000 = 166000$ psi

$F_{TY} = 200{,}000$ psi

$M.S. = \frac{200}{166} - 1 = .20$

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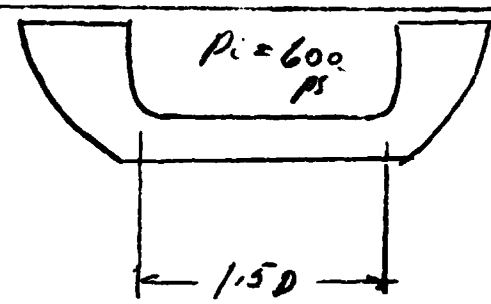
VALVE SEAL PROGRAM

TUNGSTEN CARBIDE POPPET

$F_{tu} \simeq 100K$

$F_{end} = 70K - 65K$

UNIDIRECTIONAL 10^6 cycle



p = 600 psi Proof

p = 400 psi OPERATING

$$\text{Max Stress} = .75\, w\, a^2/t^2$$

$$65000 = .75(400)(.75)^2/t^2$$

$$t^2 = \frac{169.0}{65000} = .0026$$

$$t = .051 \approx .062$$

From Dwg $t = .070$ Min

$$M.S. = \left(\frac{.070}{.051}\right)^2 - 1 = .88$$

FORM TMC 889 REV. 3-70

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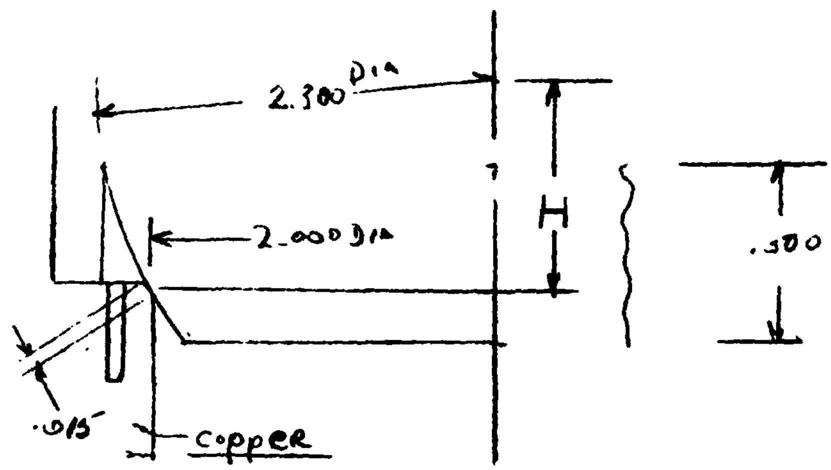
VALVE SEAT LOAD PROGRAM — COPPER LIP

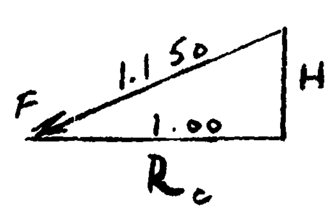


1. DESIGN COPPER LIP FOR 600 psi SEALING STRESS FOR AXIAL POPPET MOTION OF .002, .005, .010 INCHES

   F_{TY} SOFT COPPER = 10,000 psi

2. SAME AS ABOVE, BUT FOR 1740 psi SEALING STRESS FOR HARD COPPER F_{TY} = 45000 psi

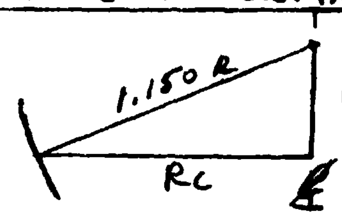
FOR INITIAL CONTACT — "F" CONSTANT FORCE



H = .565

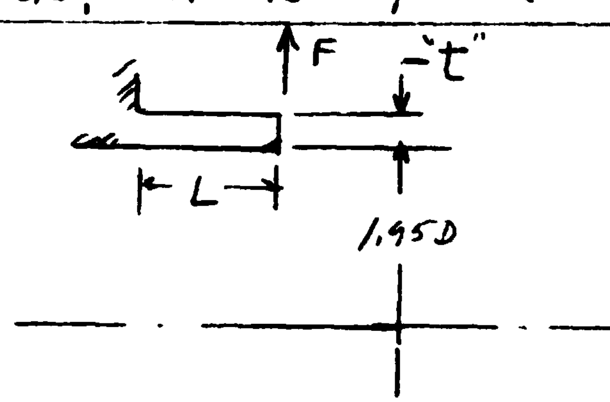
VALVE SEAT LOAD PROGRAM

RADIAL DEFORMATION OF COPPER LIP



$E = 17\times10^6$

ΔH			
.000, FOR H_0 = .565,	R_c = 1.000	,	ΔR = 0.0 in
.002,	H = .563,	R_c = 1.0015 ",	ΔR = .0015 in.
.005,	H = .560,	R_c = 1.003 ",	ΔR = .003 in
.010,	H = .555,	R_c = 1.006 ",	ΔR = .006 in.



$F = 600\times.015\left(\frac{1.003}{1.150}\right) = 7.85\#$ RADIAL

PRELIMINARY SIZING (ASSUME CANTILEVER)

$M = FL$

$y = \frac{FL^3}{3EI}$

$EI = 17\times10^6 \times \frac{t^3}{12} = 1.415\times10^6 t^3$

$I/y = \frac{t^2}{6}$

$S = M/I/y$

$S = FL6/t^2$

$FL = \frac{St^2}{6}$

$y = \frac{St^2(L)^2\,10^{-6}}{6\times3\times1.415\,t^3} = \frac{S_{TY}\,L^2\times10^{-6}}{25.4\,t} = \frac{FL^3\,10^{-6}}{4.23\,t^3}$

, BUT SEE page 1.116

VALVE SEAT LOAD PROGRAM

COPPER LIP SEAL – $E = 17 \times 10^6$

NOTE: MEAN DIAM = ± 2.00, R = 1.00

FOR $\Delta R = .0015$, $\epsilon = \frac{.0015}{1}$, $S = .0015 \times 17 \times 10^6$

$S = 25400$ psi HOOP

~~Fty SOFT COPPER = 10,000 psi~~

USE HARD COPPER, Fty ≅ 45,000 psi

SEALING STRESS = 1740 psi

SEALING LOAD = 1740 × .015 = 26.1 #/in

RADIAL LOAD = $\frac{1.00}{1.15} \times 26.1 = 22.7$ #/in

FOR LONG CYL $\Delta R = \frac{V_0}{2D\lambda^3}$, $D = \frac{Et^3}{12(1-\nu^2)} = 1.56 t^3 10^6$

$\lambda = \frac{1.285}{\sqrt{Rt}} = 1.285 t^{-\frac{1}{2}}$ $(R \cong 1)$

$$\Delta R = \frac{V_0}{2(1.56)10^6 t^3 [1.285 t^{-\frac{1}{2}}]^3}$$

$$\Delta R = \frac{V_0}{6.6 \times 10^6 t^{3/2}}$$

FOR $V_0 = 22.7$ #/in

$$\Delta R = \frac{22.7}{6.6 \times 10^6 t^{3/2}} = 3.44 \times 10^{-6} / t^{3/2}$$

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VALVE SEAT LOAD PROGRAM

COPPER LIP SEAL - HARD COPPER $F_{TY} = 45000$ psi
$E = 17 \times 10^6$

MAX ALLOW ΔR = $\frac{45000}{17 \times 10^6} \times 1'' R = .00264''$

$\Delta H \cong .0045''$

't' REQUIRED

$\Delta R = 3.44 \times 10^{-6} / t^{3/2}$ FOR $V_0 = 22.7$ #/in

$.00264 = 3.44 \times 10^{-6} / t^{3/2}$

$t^{3/2} = 3.44 \times 10^{-6} / .00264$

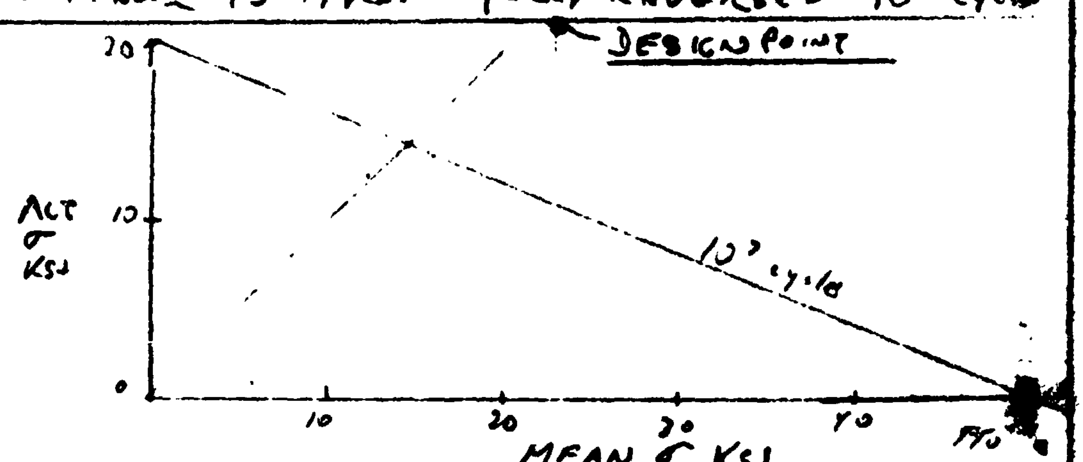
$t^{3/2} = 1.3 \times 10^{-3}$

$t = 1.19 \times 10^{-2} = .012''$ (EST. LIFE = 10^4 cycles)

$\lambda = 1.285 / \sqrt{tR} = 11.8$

FOR $\lambda L = 5$, $L = 5/11.8 = .425''$ SLOT MINIMUM.

NOTE $F_{end} \cong 15-17$ KSI - FULLY REVERSED 10^7 cycles



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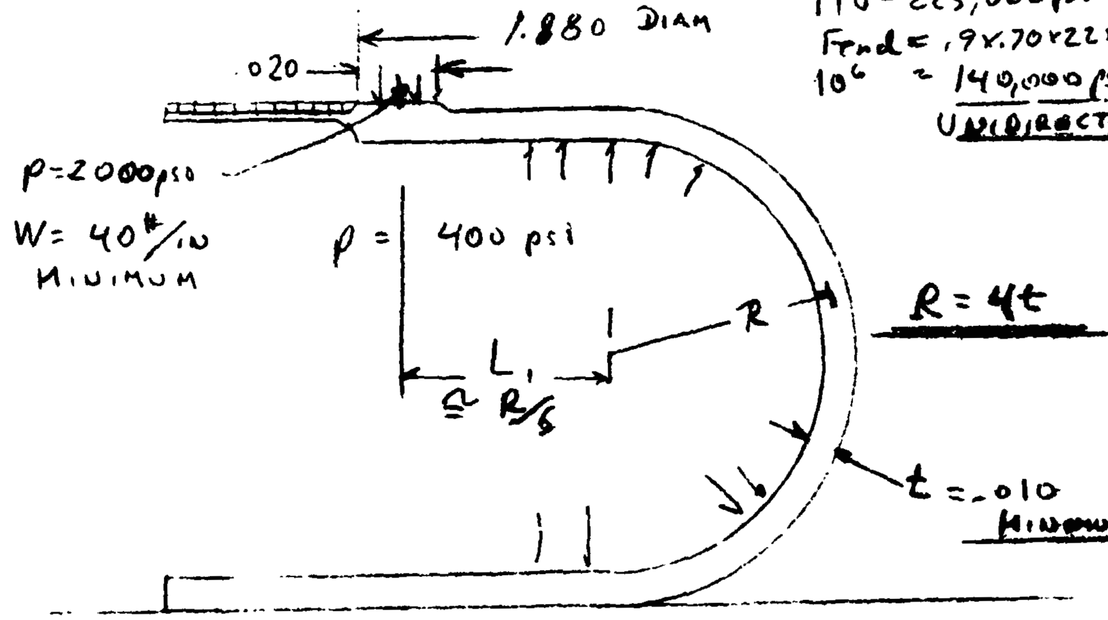
VALVE SEAL PROGRAM

LIP SEAL - TEFLON TIPPED

MATL - X-15
F_{tu} = 225,000 psi
F_{end} = .9×.70×225K
10^6 = 140,000 psi
UNIDIRECTION.



FOR PRELOAD USE 40#/in, Δ AT PAD = .005

FOR CURVED RING WITH L = R/6

$\Delta \cong 3PR^3/EI$ REF MACHINE DESIGN, JULY 23, 1959 pg 152.

$E = 30\times10^6$

$P = 40\#/in$

$I = t^3/12$

$.005 = \frac{3(40)}{30\times10^6}\,\frac{R^3}{I}$

$\frac{R^3}{I} = \frac{50\times10^3}{40} = 1250$, $\frac{R^3}{t^3} = 1250/12 = 104$, $R = 4.7t$

MAX STRESS (ENDURANCE) $\pm\frac{40\times1.2R\times6}{t^2} + \frac{40}{t} \pm \frac{400R}{t} = 140,000$

$288\frac{(3.97)}{t} + \frac{40}{t} \pm 400(3.97) = 140,000$, (LET R = 4t)

$\frac{1}{t}(1190) = 141600$, $t = .012$, $R = .048$
(MINIMUM)

VALVE SEAL PROGRAM

LIP SEAL TEFLON TIPPED — MATL - X-15, $F_{end} = 140K$

FOR $t = .020$, $R = .080$

$$\text{MAX ST} = \frac{360(4)}{.020} + \frac{40}{.020} - 1600 = 72{,}400 \text{ psi} \quad \text{ok}$$

$$M.S. = \frac{140}{72.4} - 1 = .93$$

MAX HOOP STRESS ($t = .012$, $R = .048$)

$$\text{PRESSURE LOAD} = 400 \times .080 = 32\ \#/\text{in}$$

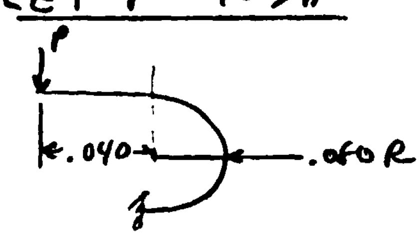
$$\text{Area} \cong 2\pi r t = \pi(.916)(.012) = .00350\ \text{in}^2$$

$$R_{MAJOR} = \frac{1.88 - 2 \times .024}{2} = .916$$

$$\text{HOOP STRESS} = WR/A = \frac{32(.916)}{.0036} = 8150 \text{ psi}$$

ANGULAR DEFLECTION OF LOAD PAD

LET $P = 40\ \#/\text{in}$



$t = .020$

$$EI = 30 \times 10^6 \times \frac{(.020)^3}{12} = 20$$

$$\theta = \frac{P}{2}\frac{(.040)^2}{EI} + \frac{2PR^2}{EI} + \frac{3.14PR(.040)}{EI}$$

$$\theta = \frac{P}{EI}(.0008 + .0128 + .01005)$$

$$\theta = \frac{40}{20}(.02365)$$

$$\theta = .0473 \text{ rad.} = 2.71^\circ$$

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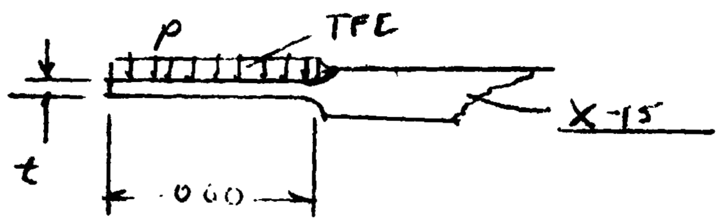
VALVE SEAT LOAD PROGRAM

SPRING - TEFLON COATED



F_{TU} = 225 KSI MIN
F_{TY} = 200 KSI "
AT 72°F
E = 30×10⁶

LET p = 500 - 600 psi

$M_{MAX} = \frac{wl^2}{2} = 600(.060)^2/2 = 1.08$ in#

$I/y = \frac{t^2}{6}$, $f_b = M/\frac{I}{y} = 6 \times 1.08/t^2$

FOR F_{TY} = 200,000 psi

$t^2 = \frac{6 \times 1.08}{200,000} = 32.4 \times 10^{-6}$

t = .0057 in FOR L = .060 * M.S = .00

FOR L = .030

$M_{MAX} = 600 \times (.030)^2/2 = .27$ in#

$f_b = 6 \times .27/t^2$

$t^2 = \frac{6 \times .27}{200,000} = 8.1 \times 10^{-6}$

t = .003 in FOR L = .030

$* \Delta y = \frac{wl^4}{8EI} = \frac{600(1300)10^{-8}}{8 \times 30 \times .018} = .0018$ "

$I = \frac{(.006)^3}{12}$

$= .018 \times 10^{-6}$

FORM TMC 888 REV. 2-70

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VALVE SEAL PROGRAM

LIP SEAL - TEFLON TIPPED - X27512 - MATL X-15

RATE OF SEAL

$R = \frac{P \pi D}{\Delta} = \frac{E I \pi D}{10 r^3}$

where R = RATE #/"

I = $t^3/12$

D = 1.88 DIAM

r = mean radius of circular cross section.

coeff for Design As Defined ...

For t = .020

$I = .667 \times 10^{-6}$

$r = .080$, $r^3 = 510 \times 10^{-6}$

$R = \frac{30 \times .667 \pi (1.88)}{10(510) 10^{-6}} = 23100$ #/"

For P = 300 #, Δ = .013"

For t = .030

$I = 2.25 \times 10^{-6}$

$r = .0825$, $r^3 = 560 \times 10^{-6}$

$R = \frac{30 \times 2.25 \pi (1.88)}{10(560) 10^{-6}} = 71,000$ #/"

For P = 300 #, Δ = .0042" ok

For P = 300 # @ .005"

$t = \left(\frac{.0042}{.005}\right)^{1/3} \times .030 = \underline{.028}$

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VALVE SEAL PROGRAM

LIP SEAL TEFLON TIPPED — MATL X-15 DWG X27512

FOR INTERNAL PRESSURE - 400 psi -

ASSUME EFFECTIVE RADIUS: $\frac{(2.00 + 1.76)}{2 \times 2} = .940$

AREA $\frac{\pi}{4}\left[(2.00)^2 - (1.76)^2\right] = .70 \text{ in}^2$

$P = 400 \times .70 = \underline{280^{\#}}$

FOR EFFECTIVE RADIUS = 1.80 MIN

AREA $\frac{\pi}{4}\left[(2.00)^2 - (1.80)^2\right] = .59 \text{ in}^2$

$P = 400 \times .59 = \underline{236^{\#}}$

LET NET NET PRESSURE FORCE = 240#

FOR 300# TOTAL NEED 300-240 = 60# SPRING FORCE

FOR t = .030 R = 71,000 #/in

$\Delta = \frac{60}{71000} = \underline{\underline{.00085''}}$

FOR 60# FORCE = .005" DEFLECTION

$\underline{t} = \left(\frac{60}{300}\right)^{\frac{1}{3}} \times .028 = \underline{\underline{.0160 \text{ in.}}}$

$R = \left(\frac{.016}{.020}\right)^3 \times 23100 = \underline{11800 \text{ \#/in}}$

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VALUE SEAL PROGRAM

LIP SEAL - TEFLON TIPPED - MATL X-15 DWG X27512

FOR $t = .031$ $R = \left(\frac{.031}{.030}\right)^3 \times 71{,}000 = 78{,}000$ #/in

FOR $\Delta = .007"$, $P = 546$#

$\text{LOAD/INCH} = \frac{546}{\pi(1.80)} = 96.6$ #/in

$\text{MAX STRESS} = \frac{P \times 1.5 R \times 6}{t^2} + \frac{P}{t}$

$\sigma = 96.6 \times \frac{1.5(.063)6}{(.031)^2} + \frac{96.6}{.031}$

$\sigma = 75{,}000 + 3120 = 78120$ psi

SINCE Fend = 140 KSI UNI-DIRECTIONAL 10^6 cycles

M.S. = LARGE

FOR $t = .016$, $R = .0671 + .008 = .0755 + .005 = .080$

$P = 60$# FOR .005", 84# FOR .007" OVERTRAVEL

$\text{LOAD/IN} = \frac{84}{\pi(1.80)} = 14.9$ #/in

$\sigma = 14.9 \times \frac{1.5(.080)6}{(.016)^2} + \frac{14.9}{.016}$

$\sigma = 41{,}700 + 930 = 42{,}600$ psi

M.S. = LARGE

FORM TMC 88F REV. 8-70

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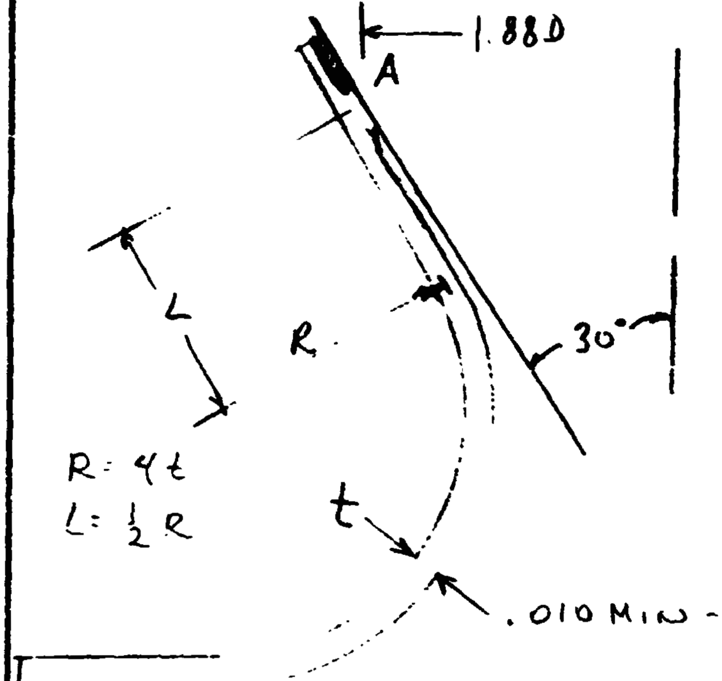
VALVE SEAL PROGRAM

LIP SEAL – TEFLON TIPPED – No2.



CONDITION ①

AXIAL Δ AT A = .005 in

CONDITION ②

NORMAL Δ AT A = .005 in AT 30° FROM HORIZONTAL

CONDITION ① AXIAL Δ AT A = .005"

RADIAL Δ = $.005 / \tan 30° = .00866$

strain = $.00866 / .94 = .00922$

FOR X-15, $E = 30 \times 10^6$

stress = $30 \times 10^6 \times .00922 = 277{,}000$ psi — N.G. TOO HIGH

✓ CONDITION ② NORMAL Δ AT A = .005, AXIAL = .0025

RADIAL Δ = $.005 \cos 30° = .00433$

STRESS = $\frac{.00433}{.94} \times 30 \times 10^6 = 138{,}000$ psi OK

Δθ = .50° AT PAD.

$1.50 \frac{140}{138} \times = .015$

FORM TMC 600 REV. 8-70

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VALUE SEAL PROGRAM

LIP SEAL - TEFLON TIPPED - No. 2

DEFLECTION STIFFNESS OF SEAL

FOR CYLINDER AT A

V_A = RADIAL NORMAL LOAD

$$\Delta R = \frac{V_A}{2D\lambda^3}$$

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{30\times10^6(.020)^3}{10.9} = 22.0$$

$$\lambda = 1.285/\sqrt{Rt} = \frac{1.285}{\sqrt{.020}} = 9.1$$

$$\lambda^3 = 750$$

$$\Delta R = \frac{V_A}{2\cdot22\cdot750} = 30.3\times10^{-6}\,V_A$$

FOR CURVED SEAL AS CANTILEVER

$$\Delta N\cos30^\circ = \Delta R = 5\,V_0 R^3/EI \cos 30^\circ \qquad R^3/I = 750 \text{ (pg 1.30)}$$

$$E = 30\times10^6$$

$$\Delta R = \frac{5\,V_A\,750}{30\times10^6}\cos30^\circ$$

$$\Delta R = 108\times10^{-6}\,V_A$$

CYLINDER $\frac{108}{30.3}$ = 3.56 TIMES STIFFER THAN CANTIL.

LOAD TO CYLINDER = $\frac{108}{(108+30.3)} P_{TOTAL_A}$ = .782 P_{TOTAL}

LOAD TO CANTILEVER = $\frac{30.3}{(108+30.3)} P_{TOTAL_A}$ = .218 P_{TOTAL}

VALVE SEAL PROGRAM

LIP SEAL TEFLON TIPPED - No. 2

DEFLECTION STIFFNESS OF SEAL

TOTAL RADIAL DEFLECTION = .00433"

$.00433 = .782\,P_T(30.3\times10^{-6}) + .218\,P_T(108\times10^{-6})$

$.00433 = 23.7\times2\,P_T\times10^{-6}$

$P_T = \frac{4330}{47.4} = 91.5\ \#/in$ RADIAL LOAD

AXIAL LOAD = $91.5/\tan 30° = 158\ \#/in$

TOTAL LOAD = $158\times\pi(1.88) = \underline{935\ \#}$

FOR $t = .010$ STIFFNESS = EQUAL CYLINDER AND CANTILEVER

$.00433 = .535\,P_T(94\times10^{-6}) + .465\,P_T(108\times10^{-6})$

$.00433 = 2\times50\times10^{-6}\,P_T$

$P_T = 43.3\ \#/in$

AXIAL LOAD = $43.3/\tan 30° = 75\ \#/in$

TOTAL LOAD = $75\times\pi(1.88) = 442\ \#$

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REPORT VALVE SEAL

DATE 4 JAN 197[illegible]

VALVE SEAL PROGRAM

LIP SEAL – CONICAL

DEFLECTION OF LIP UNDER PRESSURE

(1) AS CONE $p = 600$ psi

$$\Delta = \frac{p(R')^2}{tE} \qquad R' = 1.88/\cos 30^\circ = 2.17''$$

$(R')^2 = 4.72''$

$t = .0057$

$E = 30 \times 10^6$

$$\Delta R = \frac{600(4.72)}{(.0057)(30)10^6} = .0165''$$

(2) AS CANTILEVER $L = .060$

$\Delta R = .0018''$ REF PG 1.20

THEREFORE, SINCE STIFFNESS UNDER PRESSURE IS 9X GREATER AS CANTILEVER THAN CONE

Δy OF TIP WILL BE $.0018 \times .9 = .00163''$ FOR 600 psi

RADIAL $= .00122''$ FOR 450 psi

SPHERE RELATIVE DISPLACEMENT = .0011" RADIAL AT MAXIMUM AXIAL MOVEMENT

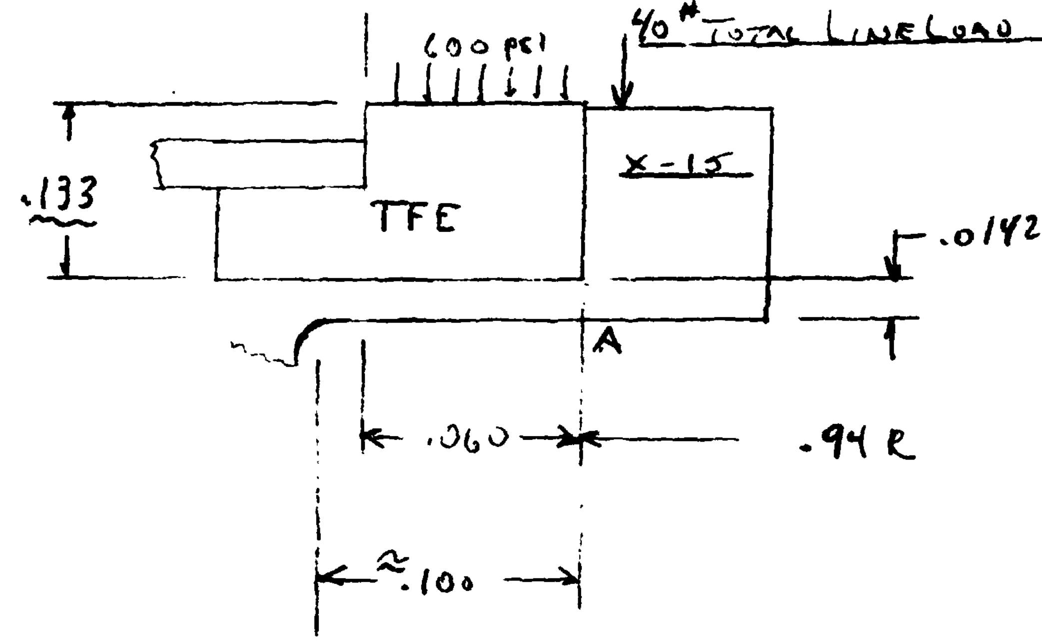
THEREFORE, TIP SHOULD SEAL

A605

VALVE SEAT LOADS PROGRAM

TEFLON CONCEPT No 2. – L=.100 PLATE

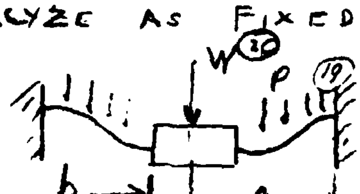
40# TOTAL LINE LOAD



DESIRED DEFLECTION AT A = .002" = y

ANALYZE AS FIXED-FIXED PLATE, ROARK, pg 222

Cases 19, 20



a/b $\quad 1.04/.94 = 1.11$

$p = 600$ psi

$W = 40$ #

$a = 1.04, \; a^2 = 1.08$

$a^4 = 1.165$

$$y = \frac{\alpha p a^4}{E t^3} + \frac{\alpha W a^2}{E t^3}$$

$\alpha_{19} = .00022$

$\alpha_{20} = .0004$

$$y = \frac{1}{Et^3}\left[.00022(600)1.165 + .0004(40)(1.08)\right]$$

$$y = \frac{1}{Et^3}(.154 + .0173) = .0057 \times 10^{-6}/t^3$$

FOR X-15

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VALUE SEAT PROGRAM

TEFLON CONCEPT No. 2 - L=.100 PLATE

$y = .0057 \times 10^{-6} / t^3$

FOR $y = .002 = .0057 \times 10^{-6} / t^3$

$t^3 = 2.85 \times 10^{-6}$

$t = 1.42 \times 10^{-2}$ (See pg 1.27)

MAX STRESS

$\sigma_{MAX} = \beta w a^2 / t^2 + \beta \frac{w}{t^2}$

$\beta = .01$

$t^2 = 2.02 \times 10^{-4}$

$\sigma_{MAX} = \frac{.01 [(600 \times .08) + 40]}{2.02 \times 10^{-4}}$

$\sigma_{MAX} = 34000$ psi ok FOR 10^7 cycles

LENGTH (HEIGHT) TFE TO COMPRESS .002

E TFE $\cong$ 40,000 psi

$\Delta = \frac{SL}{E}$ $S = 600$ psi

$.002 = \frac{600 L}{40 \times 10^3}$

$L = \frac{80}{600} = .133$

VALVE SEAT PROGRAM

TEFLON CONCEPT No 2. — p = 450 psi L = .109 PLATE

$$y = \frac{1}{Et^3}\left[.00022(450)1.165 + .0004(40)(1.08)\right]$$

$$= \frac{1}{Et^3}(.115 + .0173) = .0044 \times 10^{-6}/t^3$$

For $y = .003 = .0044 \times 10^{-6}/t^3$

$t^3 = 1.47 \times 10^{-6}$

$t = 1.14 \times 10^{-2}$ [For $y = .002$, $t = .013$]

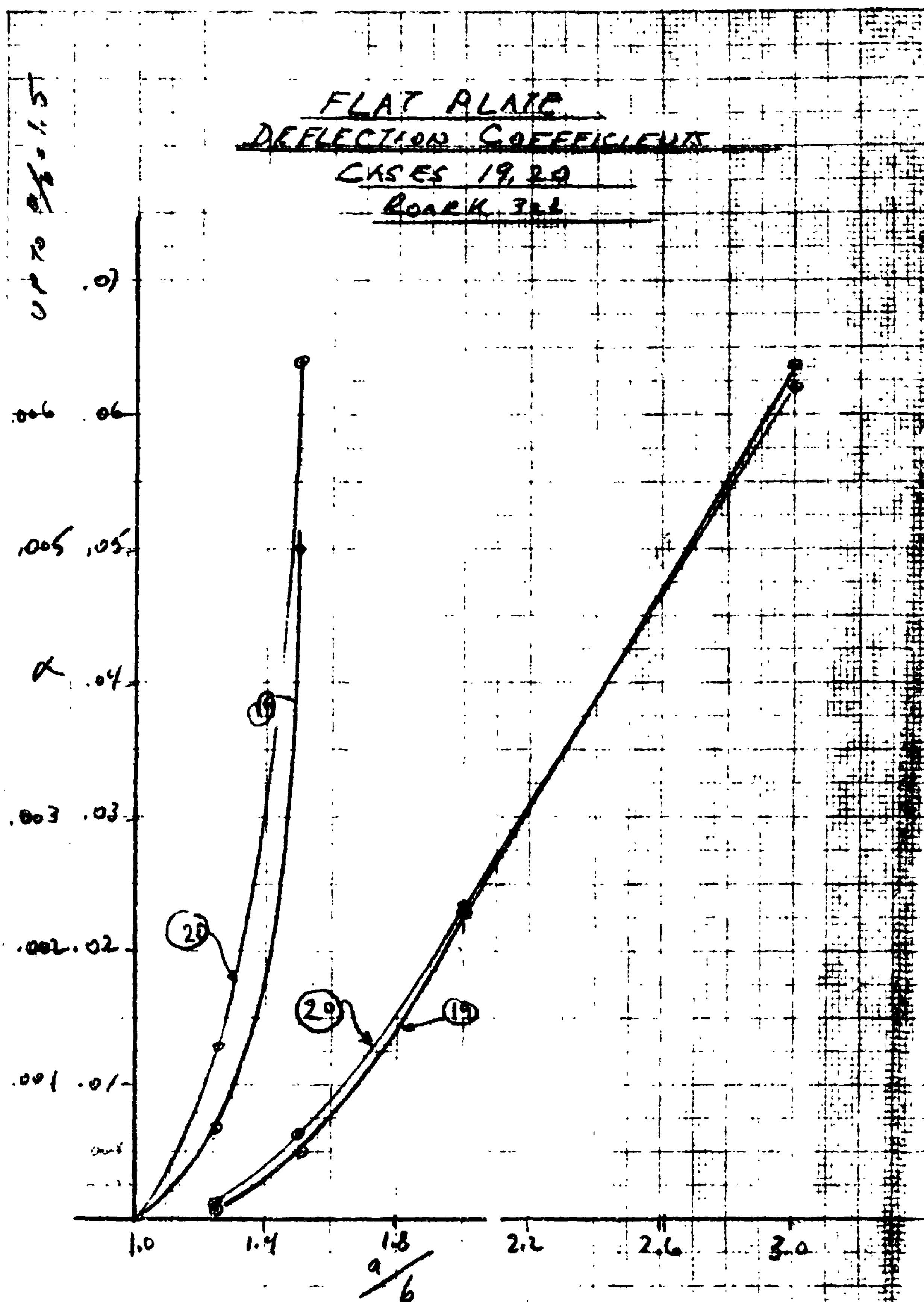
MAX STRESS

$$\sigma_{MAX} \quad \beta w a^2/t^2 + \beta \frac{W}{t^2}$$

$$= \frac{.01\left[(450)(1.08) + 40\right]}{1.3 \times 10^{-4}} \qquad t^2 = (.0114)^2$$

$= 40,000$ psi ok

$$M.S. = \frac{140}{40} - 1 = \text{LARGE}$$


FLAT PLATE
DEFLECTION COEFFICIENTS
CASES 19, 20
ROARK 321
.07
.006 .06
.005 .05
α .04
.003 .03
.002 .02
.001 .01
19
20
20
19
1.0
1.4
1.8
2.2
2.6
3.0
a/b

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VALUE SEAT PROGRAM

TEFLON CONCEPT No. 2 $p = 450$ psi + $\Delta = .006$

PLATE L = .100 $t = .013''$ X-15

$$y = \frac{1}{Et^3}\left[.00022(450)1.165 + .0004(P)1.08\right]$$

$a/b = 1.11$, $\alpha = .00022$, $\alpha = .0004$

$y = .0065$

$Et^3 = 30 \times 10^6 (.013)^3$

$= 65.7$

($a = 1.04$ ($l_{eq} = .100''$))

$.0065 = .00176 + 6.56 \times 10^{-6} P$

$P = 720^{\#}$

MAX Stress: $\dfrac{.01(450 \times 1.08 + 720)}{(.013)^2} = 71300$ psi

$M.S. = \dfrac{140}{71.3} - 1 = .96$

FOR $t = .010$ MIN., $y = .0060$

$$y = \frac{1}{30}\left[.1150 + .000432\,P\right] = .006$$

$.1150 + .000432\,P = .18$

$P = .065 / .000432 = 150^{\#}$

MAX Stress = $\dfrac{.01(450 \times 1.08 + 150)}{(.01)^2} = 63600$ psi ok

$F_{End} = 140{,}000$ psi for X-15

ok FOR .011 t ALSO

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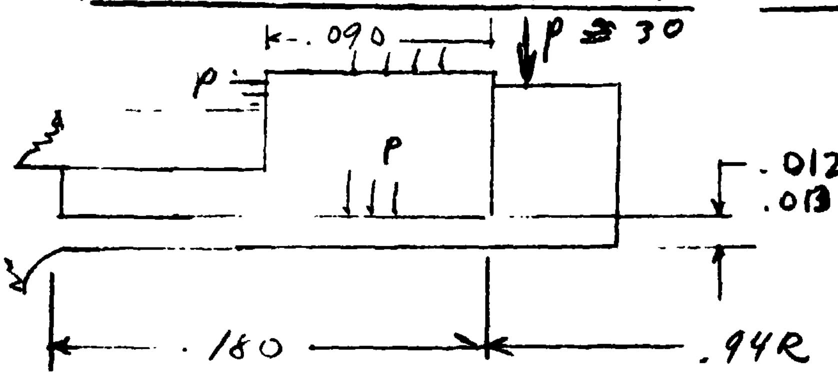
VALVE SEAT PROGRAM

TEFLON SEAL CONCEPT   L = .180 PLATE



X-15 MATL.

FOR DEFLECTION AT LOAD = .0060 in, p = 450 psi

ANALYZE PER ROARK pg 222, cases 19 + 20

$a = 1.12$, $a/b = 1.12/.94 = 1.19$

$\alpha(19) = .00045$ $a^2 = 1.26$, $a^4 = 1.58$

$\alpha(20) = .0008$ ($t = .014$ TRIAL)

$$y = \frac{\alpha p a^4}{E t^3} + \frac{\alpha P a^2}{E t^3}$$

$y = .006$
$p = 450$ psi
$E t^3 = 82$ $\boxed{t = .014}$

$$.006 = \frac{1}{82}\left[.00045(450)(1.58) + .0008(1.26)P\right]$$

$$.492 = .32 + .001008P$$

$$P = .172/.001008 = 171^{\#}$$

$$\text{MAX Stress} = \frac{1}{t^2}\left[.035(450)(1.26) + .060(171)\right]$$

$\beta(19) = .035$
$\beta(20) = .060$

$= 5100(30.15)$

$\sigma = 154000$ psi   N G. — Fend = 140 KSI

FORM TMC 888 REV. 3-70

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VALVE SEAT PROGRAM

TEFLON SEAL CONCEPT - L=.180 PLATE

LET t=.016

$Et^3 = 30(1.6)^3 = 123$

FOR DEFLECTION $.006 = \frac{1}{123}\left[(.32 + .001008P\right]$

$.738 = .32 + .001008P$

$P = .418 / .001008 = \underline{415}$

MAX STRESS $= \frac{1}{t^2}\left[19.9 + .060(415)\right]$

$= 44.8 / (.016)^2 = \underline{175000}$ psi Too High

LET t=.0125

$Et^3 = 30(1.25)^3 = 58.6$

$.006 = \frac{1}{58.6}(.32 + .001008P)$

$.352 = .32 + .001008P$

$.032 / .001008 = P = \underline{32^{*}}$

Max Stress $= \frac{1}{(.0125)^2}\left[19.9 + .060(32)\right] = 6400(21.82)$

$= \underline{140,000 \text{ psi}}$

$F_{END.} = 140,000$ psi

$\underline{M.S.} = \frac{140}{140} - 1 = \underline{.00}$

FORM TMC 688 REV. 3-70

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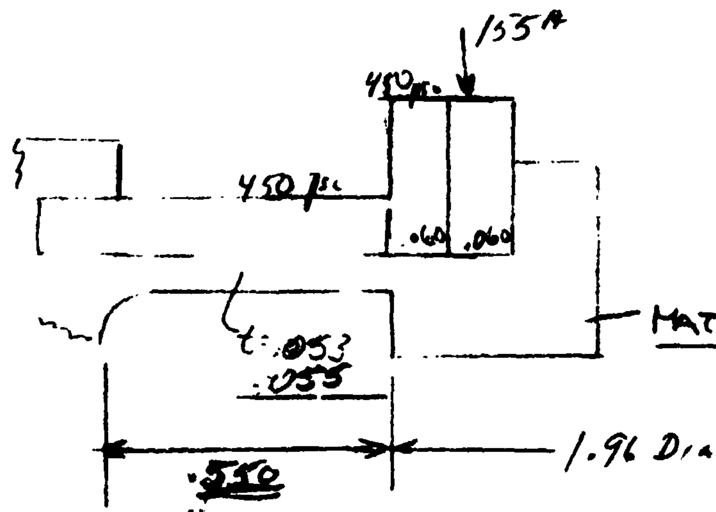
VALVE SEAT PROGRAM

TEFLON SEAL CONCEPT   L = .550 PLATE

NOMINAL DEFLEC = .005

MAX DEFLECT = .008



ANALYZE PER ROARK, pg 222, cases 19, 20

$a = .98 + .245 = 1.225 \qquad a/b = 1.225/.98 = 1.25$

$a^2 = 1.5 \ , \ a^4 = 2.25$

$\alpha_{19} = .00068 \ , \ \beta_{19} = .070$

$\alpha_{20} = .0013 \ , \ \beta_{20} = .0984$

$$y = \frac{\alpha p a^4}{Et^3} + \frac{\alpha P a^2}{Et^3}$$

LET $P_1 = \pi(1.90)(.060)(450)$
$= 161^{\#}$
$P_2 = 155^{\#}$
TOTAL $P = 316^{\#}$

$$.005 = \frac{1}{Et^3}\left[.00068(2.25)(450) + .0013(316)(1.5)\right]$$

$$Et^3 = \frac{1}{.005}\left[.688 + .616\right] = 260.8$$

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VALVE SEAT PROGRAM

TEFLON SEAL CONCEPT L = .550 PLATE

FOR STEEL X-15, $E = 30 \times 10^6$

$Et^3 = 260.8$

$t^3 = 6.7 \times 10^{-6}$

$t = .0203'$

MAX STRESS OCCURS AT .008 DEFLECTION

$$.008 = \frac{1}{260.8}\left(.688 + .0013\, P_{MAX}(1.5)\right)$$

$$2.085 = .688 + .00195\, P_{MAX}$$

$$P_{MAX} = \frac{1.397}{.00195} = 716^{\#}$$

MAX STRESS

$$S = \beta \frac{wa^2}{t^2} + \beta \frac{W}{t^2}$$

$t^2 = .0004$

$W = 716^{\#}$

$$= \frac{1}{.0004}\left(.070 \times 450 \times 1.5 + .0984\,(716)\right)$$

$S = 243000$ psi too high

FORM TMC 688 REV. 5-70

VALVE SEAT PROGRAM

TEFLON SEAL CONCEPT $L = .55$ PLATE

Let $a = 1.47$, $b = .98$, $a/b = 1.5$, $y = .005$

$a^2 = 2.16$, $a^4 = 4.66$

$\alpha_{(19)} = .0050$, $\beta_{(19)} = .183$, $.114$

$\alpha_{(20)} = .0064$, $\beta_{(20)} = .168$, $.220$

$$.005 = \frac{1}{Et^3}\left[.0050(450)4.66 + .0064(316)(2.16)\right]$$

$$Et^3 = \frac{1}{.005}(10.5 + 4.36) = 2972$$

For $E = 30 \times 10^6$, $t^3 = 99 \times 10^{-6}$

$t = .0463$

MAX LOAD AT DEFLECTION $= .008 = y$

$$.008 = \frac{1}{2972}\left(10.5 + .0138 P_{MAX}\right)$$

$23.75 - 10.5 = .0138\,P$

$P = 960^{\#}$ MAX

MAX Stress $= \dfrac{10^4}{21.5}\left[.183(450)(2.16) + .168 \times 960\right]$ $\quad t^2 = .00215$

$= 157,000$ psi

$= \dfrac{10^4}{21.5}\left[.114(450)(2.16) + .220 \times 960\right]$

$= 150,000$ psi

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VALVE SEAT PROGRAM

TEFLON SEAL CONCEPT L = .550 PLATE

LET $a = 1.53$; $b = .98$, $a/b = 1.56$ $y = .005$

$a^2 = 2.34$, $a^4 = 5.48$

$\alpha(19) = .0067$ $\beta(19) = .125$, $.195$

$\alpha(20) = .0073$ $\beta(20) = .233$, $.175$

$$.005 = \frac{1}{Et^3}\left[.0067(450)5.48 + .0073(316)(2.34)\right]$$

$$Et^3 = 200\left[16.5 + 5.4\right] = 4380$$

$$t^3 = 4380/30\times10^6 = 146\times10^{-6}$$

$t = .0527$ MIN $= .053$ $t^2 = .00279$

MAX LOAD AT .008" DEFLECTION = y

$$.008 = \frac{1}{4380}\left[16.5 + .0171 P\right]$$

$$35 - 16.5 = .0171 P$$

$P = 1080$ # MAX

MAX STRESS $S = \frac{10^4}{27.9}\left[.125(450)2.34 + .233(1080)\right]$

$= 138000$ psi

$S = \frac{10^4}{27.9}\left[.195(450)2.34 + .175(1080)\right]$

$= 141,000$ psi ok

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SEAL TESTER – FLEXURE ANALYSIS

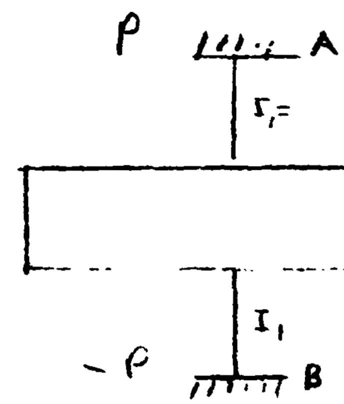


FIG 1.

FOR VERTICAL DISPLACEMENT OF A REL B – = 1" TOTAL

BUT IF WE LIMIT DISPL– OF A TO B TO ±.500, THEN A TO B = 1.00 – DISP TOTAL

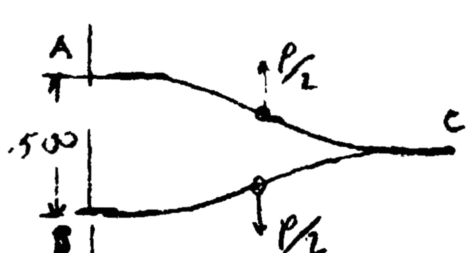
$I_1 = \frac{.9(t)^3}{12} = .075\,t^3$, $L_1 = 1.0$ in

$I_2 = \frac{.25\,t^3}{12} = .0208\,t^3$, $L_2 = 2.0$ in (Average)

$I_3 = I_2 = .0208\,t^3$ $L_3 = .75$ in (.50 FOR TORSION)

FOR UNIT LOAD P

$\delta_1 = 2\left[PL^3/3EI\right] = 2 \times P(1)^3/3E(.075)t^3 = 8.9\,P/Et^3$



$\delta_{12} = 2\left[2\,\frac{P}{2} \times \frac{.75}{2}\,\frac{L_1^2}{2EI}\right] 2 = 36.0\,P/Et^3$

TOTAL $\delta_1 = 13.90\,P/Et^3$

FIG. 2.

$\delta_2 = 4 \times \frac{P}{2} \times \left(\frac{2}{2}\right)^3 / 3EI = 2P/3E(.0208)t^3 = 32\,P/Et^3$

$\theta_C = \frac{P}{2} \cdot \frac{L_2}{2} \times \frac{L_3}{2} / KG$

AT END

$K = a\,t^3/3.33 = \frac{.25\,t^3}{3.33}$

$K = .075\,t^3$

$G = .38\,E$

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SEAL TESTER - FLEXURE ANALYSIS

DEFLECTION OF FLEXURE - UNIT LOAD P

$\theta_c = \frac{P}{2} \frac{L_2}{2} \cdot \frac{L_3}{2} \Big/ KG$

$K = .075t^3$
$G = .38E$
$L_3 = .50$ For Torsion
$L_2 = 2$

$\theta_c = \frac{P}{2} \times \frac{2}{2} \times \frac{.50}{2} \Big/ .075 \times .38 E t^3$

$\theta_c = 4.38 \, P \big/ Et^3$

$\theta_c = M_c L_2 \big/ EI_2$

$M_c = \frac{\theta_c E I_2}{L_2} = \frac{4.38 P}{Et^3} \times \frac{E(.0208)t^3}{2} = .0455P$

$\delta_{1\theta} = \left[\frac{M_c L_2^2}{2EI}\right] 2 = \frac{.0455 P (2)^2}{E(.0208)t^3} = 8.73 P \big/ Et^3$

TOTAL $\delta_2 = (32 + 8.73) P \big/ Et^3 = 40.7 P \big/ Et^3$ TOTAL

$\delta_3 = 2\left[\frac{P}{2}\left(\frac{L_3}{2}\right)^3 \Big/ 3EI\right] = 2 \frac{P}{2} (.375)^3 \big/ 3E(.0208)t^3$

$\delta_3 = .843 \, P \big/ Et^3$ TOTAL

TOTAL $\delta_1 + \delta_2 + \delta_3 = (13.90 + 40.7 + .843) \frac{P}{Et^3} = \frac{55.40 P}{Et^3}$

RATE $= \#/_{11} = \frac{Et^3}{55.40}$ PER LOOP

FOR 3 LOOPS PLATE, RATE $= .0542 \, Et^3$ / PLATE

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SEAL TESTER - FLEXURE ANALYSIS

RATE = $.0542\,Et^3$ / PLATE

FOR E = 30×10^6

RATE = $1.626 \times 10^6\, t^3$ / PLATE

t	RATE (#/")	P FOR .500" Δ TOTAL/PLATE	M CONN	M C = Tq
.010	1.626	.813 #	.373 in#	.136 in#
.020	13.0	6.50 #	2.98 "	1.09 "
.030	44.0	22.0 #	10.10 "	3.67 "
.040	104.0	52.0 #	23.8 "	8.70 "
.025	25.4	12.7 #	5.95 "	2.11

AT CONNECTOR $M = 1.375 \times P_{TOT}/3 = .458\, P_{TOT}$

AT C $M_{BENDING} = M_{Tq} = \frac{1 \times P_{TOT}}{6} = .1667\, P_{TOT}$

$(T_q = .375 \times P_{TOT}) = .0626\, P_{TOT} = T_q$

<u>AT CONNECTOR</u>

$$f_b = My/I = \frac{M \times 6}{bt^2} = \frac{M \times 6}{.9\,t^2} = 6.67\, M/t^2$$

<u>AT C</u>

$$f_b = \frac{M \times 6}{bt^2} = \frac{M \times 6}{.25\,t^2} = 24\, M/t^2,$$

$$f_s = \frac{3.3M}{bt^2} = \frac{3.3M}{.25\,t^2} = 13.2\, M/t^2$$

NOTE: AT C f_b not at same plane as f_s (max)

f_s max not at same plane as f_b max

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SEAL TESTER FLEXURE ANALYSIS

BENDING AND TORSION ON FLEXURE

FOR .500 DEFLECTION

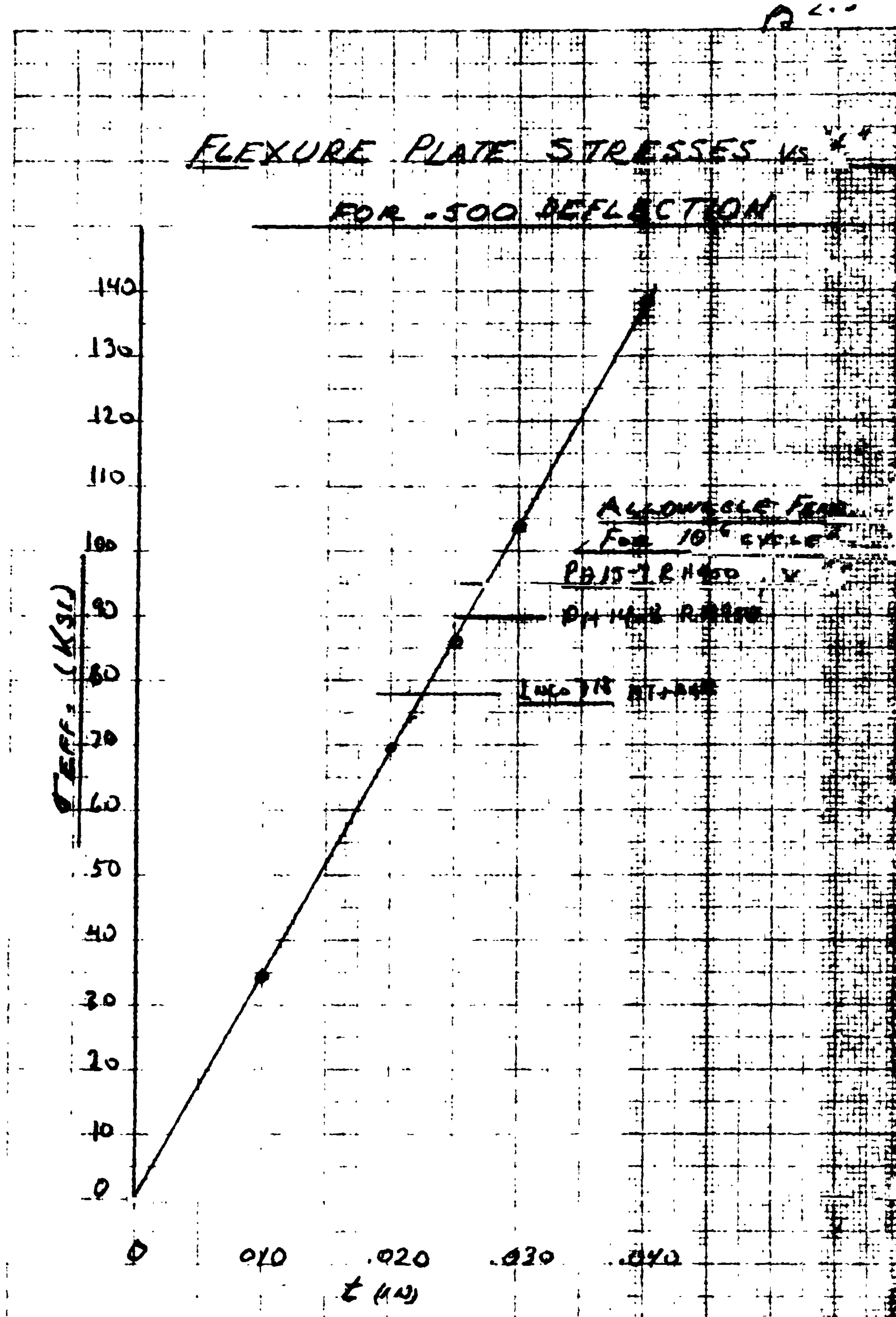
t	M_{COUN}	$f_{b_{COUN}}$	Mc	Tq	f_b	f_s	f_s max
.010	.373	24100	.136	.051	32700	6740	17900
.020	2.98	49600	1.09	.406	65600	13400	35800
.030	10.10	75000	3.67	1.38	98000	20200	54000
.040	23.6	99000	8.10	3.26	130,600	27000	72000
.025	5.95	63500	2.11	.795	81000	16800	44600

AT C Max $\sigma_{eff} = (\sigma_b^2 + 3\tau^2)^{1/2}$

t	$\sigma_b^2 \times 10^6$	$\tau^2 \times 10^6$	$3\tau^2 \times 10^6$	$\sigma_b^2+3\tau^2 \times 10^6$	$(\sigma_b^2+3\tau^2)^{1/2} = \sigma_{eff}$
.01	1070	45.4	136	1206	± 34800
.02	4310	179.0	537	4847	± 69600
.03	9600	408	1224	10824	± 104000
.04	17000	730	2190	19190	± 138600
.025	6560	282	846	7406	± 86100

FOR INCO 718, $F_{END} = .42 \times 185$ KSI = 78 KSI $\simeq 10^6$ cycles

$t_{MAX} = .0225''$


FLEXURE PLATE STRESSES vs t
FOR .500 DEFLECTION
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
σ EFF. (KSI)
ALLOWABLE FOR
FOR 10⁶ CYCLES
PH 15-7 RH 950
0
.010
.020
.030
.040
t (IN)

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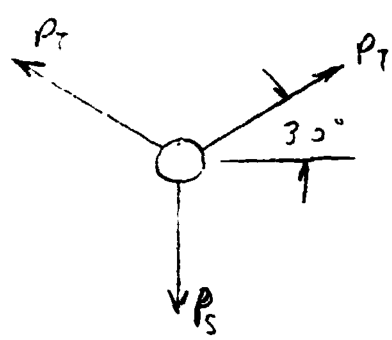
SEAL TESTER – FLEXURE ANALYSIS

SIDE LOAD STIFFNESS AND LOAD CAPABILITY



$$2 P_T \sin 30° = P_S$$

$$P_T = \frac{P_S}{2 \sin 30°} = P_S$$

DEFLECTIONS FOR UNIT SIDE LOAD

SEE FIG 1, FIG 2

CONNECTORS $\delta_1 = 2P_s L / AE = 2P_s(1)/.9tE = 2.22 P_s/Et$

SIDE FLEXURES $\delta_2 = \frac{P_s}{2} \times 4 \times \left(\frac{2}{2}\right)^3 / 3EI$

$$I = t\,\frac{(.25)^3}{12} = .0013t$$

$$\delta_2 = 2 P_s / 3(.0013t)E$$

$$\delta_2 = 513\, P_s/Et$$

TOTAL DEFLECTION (IN LINE P_T) $= (2.22 + 513)\, P_s/Et = (515\, P_s/Et) \sin 30°$

RATE $Et/515 \sin 30° = Et/258$ PER PLATE

SEAL TESTER - FLEXURE ANALYSIS

SIDE STIFFNESS

RATE = $Et/258$

FOR $E = 30 \times 10^6$ $R = .116 \times 10^6\, t$ / PLATE (#/in)

t	RATE	FLEXURE ALLOWABLE LOAD (P_S or P_{CR})	ALLOW Δ_S
.010	1160 #/in	4.9 # BUCKL *	.0042"
.020	2320	33.4 #	.0144
.025	2900	41.6 #	↕
.030	3480	50.0 #	↕
.040	4640	66.8 #	.0144

* Pg 28

MOMENT ON FLEXURE = $\frac{P_S}{2} \times 1 = .5 P_S$

$f_b = \frac{My}{I} = \frac{6M}{t(.25)^2} = \frac{6 \times .5 P_S}{t(.25)^2} = 48 \frac{P_S}{t}$

If we use F_{END} = 80,000 psi INCO 718 FOR 10^6 CYCLES

$80K = 48 \frac{P_S}{t}$

$P_S = \frac{80K\, t}{48} = 1.67 K t$

$P_{BUCKLING} = \frac{.669\, t^3 h \sqrt{EG(1 - .63 \frac{t}{h})}}{L^2} = \frac{.669 \times .25\, t^3 \sqrt{30(11.5)(1 - .63 \frac{t}{.25})}}{(1)^2\, 10^{-6}}$

$P_{BUCKLING} = 3.1\, t^3 \times 10^6 \sqrt{(1 - 2.52 t)}$, $P_b = P_S/2$

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SEAL TESTER FLEXURE ANALYSIS

SIDE STIFFNESS – ALLOWABLE LOAD

BUCKLING ON LEGS –

REF ROARK pg 344, case 16

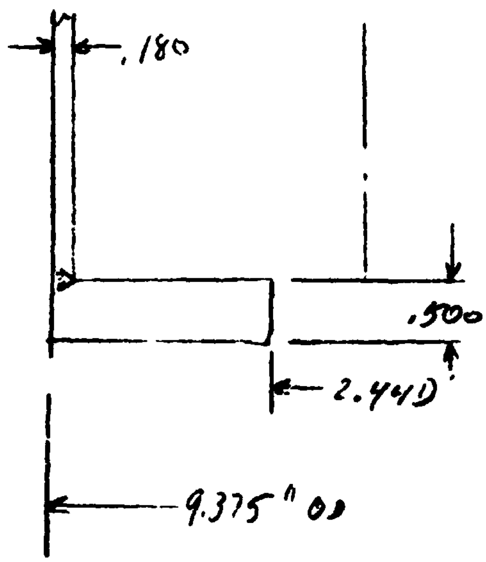
$P_{b_{cr}} = 3.1 t^3 \times 10^6 \sqrt{(1-2.52t)}$ FOR $h = .25$, $L = 1$ CANTILEVER

t	$2.52t$	$1-2.52t$	$(1-2.52t)^{\frac{1}{2}}$	t^3	P_{cr}	P_{ALLOW} $2P_{cr}/1.25$
.01	.0252	.9748	.987	1×10^{-6}	3.06	4.9 #
.02	.0504	.9496	.975	8×10^{-6}	24.2	38.7
.025	.063	.937	.967	15.6×10^{-6}	46.8	75.0
.030	.0756	.9244	.961	27×10^{-6}	80.5	129.0
.040	.1008	.8992	.946	64×10^{-6}	187.0	300.

NOTE: $P_{b_{cr}} = P_S/2$

SEAT TESTER – JACKET – TEST FIXTURE

304L MATERIAL



P_1 : 435 NOMINAL
PROOF = 670 psi

AT CYLINDER WALL $\sigma = 2 \times 670 \times \frac{(9.375 - .18)}{2 \times .180} = 33500$ psi

$F_{ty} = 30000 \times 1.2 = 36000$ psi

$M.S. = \frac{36000}{33500} - 1 = .07$

PLATE ANALYZE PER ROARK, case 13

$a/b = 9.375/2.44 = 3.84$, $\beta = 2.05$

Max $\sigma = \beta w a^2/t^2 = 2.05(670)(4.59)^2/(.5)^2$

$\sigma = 116{,}000$ psi

For CASE 37 $\beta = 1.4$

$\sigma = \frac{1.4}{2.05} \times 116{,}000 = 79000$ psi Too High!

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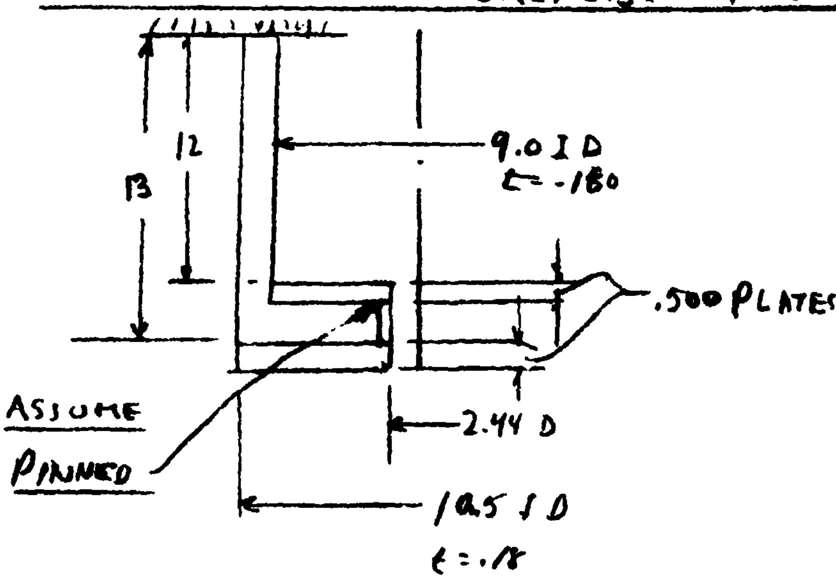
SEAL TESTER – JACKET TEST FIXTURE

304L MATERIAL

COMPATIBILITY ANALYSIS OF END PLATES



DEFLECTION OF INNER PLATE FOR 670 psi

ROARK, case 13

$y = \alpha w a^4 / E t^3$, $\alpha = .829$, $a^4 = (4.59)^4 = 445$

$t^3 = .125$

$E = 28 \times 10^6$

$$y = \frac{.829(670)(445)}{28 \times .125 \times 10^6} = .070500$$

SHELL δ $\frac{PL}{AE}$

$P = 670\left(\frac{\pi}{4}\right)\left[(9)^2 - (2.44)^2\right]$

$= 40000$ #

$$\delta = \frac{40000 \times 12}{5.48 \times 28 \times 10^6} = .00313$$

$L = 12$

$A = \pi(9.18)(.18) = 5.48$

TOTAL $\delta_{11} = .0705 + .00313 = .0736$ "

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SEAL TESTER – JACKET TEST FIXTURE

COMPATIBILITY ANALYSIS

UNIT LOAD ON PLATES

INNER PLATE $y = \alpha \frac{Wa^2}{Et^3}$ $\alpha = .726$ CIRC 14, ROARK $a^2 = (4.59)^2 = 21.1$

$$y = \frac{.726(1)(21.1)}{28 \times .85 \times 10^6}$$

$$y = 4.38 \times 10^{-6}$$

INNER SHELL $\delta = .00313/40000 = .078 \times 10^{-6}$

TOTAL $\delta = (4.38 + .078)10^{-6} = 4.458 \times 10^{-6}$ /#

OUTER PLATE $y = \frac{\alpha W a^2}{E t^3}$ $a/b = 10.68/2.49 = 4.36$ $\alpha = .717$ $(a)^2 = (5.34)^2 = 28.5$

$$y = \frac{.717(1)(28.5)}{28 \times .125 \times 10^6}$$

$$y = 5.85 \times 10^{-6}$$

OUTER SHELL $\delta = \frac{PL}{AE}$ $L = 13''$ $A = \pi(10.68)(.19) = 6.35$ $E = 28 \times 10^6$

$$\delta = \frac{13}{6.35 \times 28 \times 10^6} = .073 \times 10^{-6}$$

TOTAL $\delta = (5.85 + .073)10^{-6} = 5.923 \times 10^{-6}$ #

TOTAL STIFFNESS $= (4.458 + 5.923)10^{-6} = 10.38 \times 10^{-6}$

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SEAL TESTER - JACKET TEST FIXTURE

COMPATIBILITY ANALYSIS

DEFLECTION OF INNER PLATE ALONE

$\delta_L = .0736''$ AT INNER EDGE

RELIEVING LOAD $= \frac{4.458}{10.38} \times \frac{.0736}{4.458 \times 10^{-6}} = -7100^{\#}$

LOAD FROM 2.44 DIAM HOLE $= 670\ \frac{\pi}{4}(2.44)^2 = 3140^{\#}$

DISTRIBUTE TO INNER PLATE $= 3140 \times \frac{5.923}{10.38} = +1790^{\#}$

NET EDGE LOAD TO INNER PLATE $= -7100 + 1790 = -5310^{\#}$

FROM Pg 8, STRESS ON INNER PLATE (NO CENTER LOAD)
$= 116{,}000$ psi

USE RELIEVING LOAD AT CASE 14,

$\sigma = \beta W/t^2 \quad , \beta = 2.12$

$\sigma = 2.12\,(5310)/.25$

$\sigma = -45100$ psi

NET STRESS $= 116000 - 45100 = 70900$ psi TOO HIGH!

USE REINFORCEMENT WEBS

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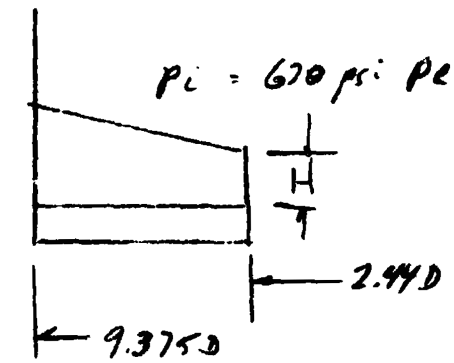
INNER PLATE - REINFORCED WITH WEBS - 304L



HEIGHT H = K(DIAM) AT ANY DIAM.

ANALYZE PER ROARK, case 13

$\sigma = \beta w a^2/t^2$

where $\sigma_{max} = 30{,}000$ psi
$w = 670$ psi Proof
$a^2 = (4.59)^2 = 21.1$
$\beta = 2.05$ for $a/b = 3.84$

$t^2 = \dfrac{2.05(670)(21.1)}{30000}$

$t^2 = .967$

$t = .983$ in equivalent

Note: Fend = 35000 psi for psi 485 psi

$M.S. = \frac{35}{20} - 1 = .75$

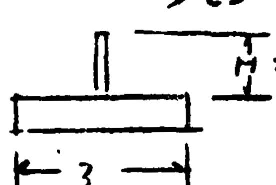
CIRCUM AT ID = $\pi(2.44) = 7.65$

CIRCUM AT OD = $\pi(9.0) = 28$

USE 6 WEBS

AT MID RADIUS $\pi(11.44/2) = 18''$ CIRCUM

SECTION =



H = .279 DIAM.

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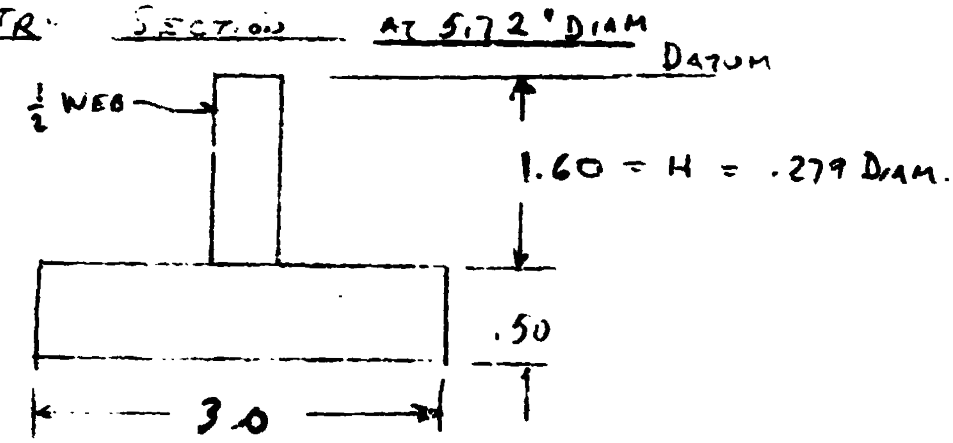
SEAL TESTER – JACKET TEST FIXTURE

INNER PLATE REINFORCEMENT WEBS

REQUIRED $t = .983''$

I/y for 3" width = $\frac{3}{6}(.983)^2 = .454$

TR. SECTION AT 5.72" DIAM



PART	Area	y	Ay	Ay^2	I_o
Web	.80	.80	.64	.512	.1710
Plate	1.50	1.85	2.77	5.125	.0313
	2.30		3.41	5.637	.2023

$\bar{y} = 3.41/2.30 = 1.48$

$I_{NA} = .2023 + 5.637 - 3.41(1.48) = .7893$

$I/y = .7893/1.48 = .531$

For above: Max $f_b = 30,000$ psi

MAKE H PROPORTIONAL TO CIRCUM. AT DIAM.

H = .279 DIAM

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SEAL TESTER – JACKET TEST FIXTURE.

ATTACH BOLTS AND FLANGE THREADS

O-RING SEAL DIAM = 9.88 - .260 = 9.62 D

TOTAL LOAD AT 670 psi = $\frac{\pi}{4}(9.62)^2(670)$ = 48600 # proof

LOAD/BOLT = 48600/12 = 4050 # Proof

BOLTS ARE 3/8 × 24 THREAD

MAX TENSILE = 4050/.081 = 50,000 psi LIMIT

BOLTS ARE A-286 H.T

F_{TU} = 135000 psi
F_{TY} = 85,000 psi

M.S = $\frac{85}{50}$ - 1 = .70

SHEAR ON THDS IN PLATE (304 L MAT'L)

ASSUME ENGAGED LENGTH = .375 in MIN.

$A_s = \pi(.375)(\frac{.375}{2}) = .221\ in^2$

f_s = 4050/.221 = 18300 psi

F_{sy} = 30,000 × .6 = 18000 psi

M.S. = .00

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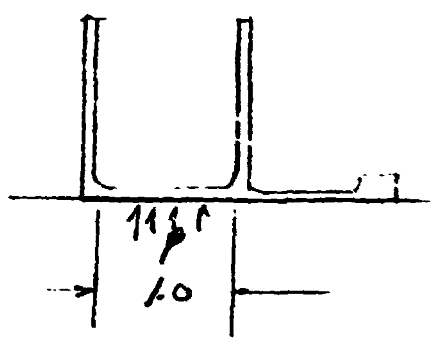
VALVE SEAL PROGRAM - SEAL TESTER

END CAP OF TESTER ADAPTER   INCO 718

$F_{TU} = 185$ KSI



$p_{PROOF} = 600$ psi

$MAX\ S = .75\ p \frac{a^2}{t^2}$

Let $S \cong 110$ KSI FOR FEND

$a^2 = .25$

$$t^2 = \frac{.75(600)(.25)}{110 \cdot 10^3}$$

$t^2 = .001020$

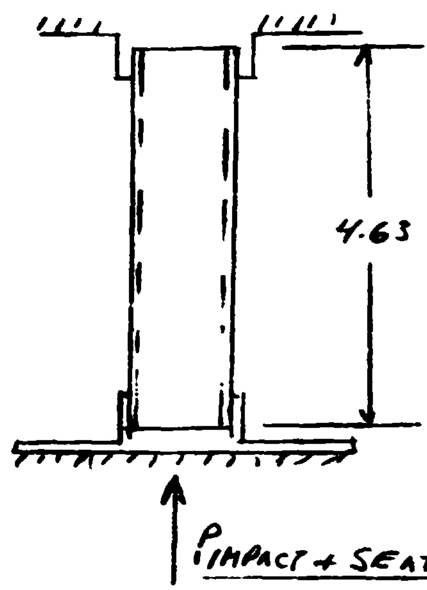
$t = .032$ MIN

For $t = .063$

$$M.S = \left(\frac{.063}{.032}\right)^2 - 1 = \text{LARGE}$$

VALVE SEAL PROGRAM - SEAL TESTER

TEST RIG POPPET SUPPORT TUBE - 304L CRES



TUBE IS 1.125 OD x 1.00 ID

$P_{IMPACT} = 2330^{\#}$

$P_{SEATLOAD} = 1500^{\#}$

COMPRESSION AREA $= \pi(1.062)(.062)$

$= .207 \text{ in}^2$

TREAT AS SEMI-FIXED COLUMN

$\rho = .7R = .7 \times .53 = .372$

LET $C = 2.0$, $L' = \dfrac{L}{\sqrt{2}} = \dfrac{4.63}{1.41} = 3.28$

$L'/\rho = 3.28/.372 = 8.82$

$\beta = \dfrac{1}{\pi}\sqrt{\dfrac{F_{0.7}}{E_c}}\left(\dfrac{L'}{\rho}\right)$ $F_{0.7} = 24$ KSI , $n \approx 5$

$E_c = 28 \times 10^6$

$\beta = \frac{1}{\pi}(2.93 \times 10^{-2})(3.28) = .0306$

NOT CRITICAL AS COLUMN

$f_c = (2330 + 1500)/.207 = 18500$ psi

$F_{cy} = 30{,}000$ psi

$F_{end} = 35{,}000$ psi ULT M.S. $= \dfrac{30}{18.5 \times 1.5} - 1 = .08$

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SEAL TESTER R11

COIL SPRING — MUSIC WIRE

PRELOAD = 200#

RATE ≈ 200-300 #/IN.

O D = 1.50, d = .218 WIRE, R = 1541 #/in/COIL - STL.

MEAN DIAM = 1.50 - .218 = 1.282

N = 8 ACTIVE COILS, 10 TOTAL FOR 200 #/in TOTAL R

MAX WORKING LOAD = 220#

FREE LENGTH = $10 \times .218 + .20 + \frac{220}{200} = 3.48$

LOAD AT SOLID = (3.48 - 2.18)200 = 260#

STRESS AT SOLID = 81000 K × 1.25 = 101000 psi

Fsy = .45 × 240,000 psi UTS = 108000 psi

$$MS = \frac{108000}{101000} - 1 = .07$$

BUCKLING CRITERIA

$$R = F.L./M.D. = \frac{3.48}{1.282} = 2.7$$

$$R = DEFL./F.L. = \frac{1.1}{3.48} = .317$$

OK PER SPRING HANDBOOK

A688

SEAL TEST RIG

COIL SPRING - STAINLESS W.F. -302

PRELOAD: 200# RATE = 200#/in

OD = 1.500" d = .244, R = 2220 #/in/coil

MEAN DIAM = 1.500 - .244 = 1.256

N = 11 ACTIVE, 13 TOTAL COILS FOR 200#/in R.

MAX WORKING LOAD = 220#

FREE LENGTH = 13 × .244 + .25 + 220/200 = 4.52

LOAD AT SOLID = (4.52 - 3.17)(200) = 270#

STRESS AT SOLID = 60000 K

= 60000 × 1.30 = 78000

FSY AT SOLID = 78000 PSI

M.S. = 78/78 - 1 = .00

INSTALLED H = 4.52 - 1 = 3.52"

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SEAL TEST RIG

COIL SPRING - STAINLESS WIRE - 302 CRES

PRELOAD = 200#, RATE = 200#/.

OD = 2.300, d = .283 M.D = 2300 - .283 = 2017

N = 5.0 ACTIVE, N = 7.00 TOTAL D/d = 7.11

SOLID HEIGHT = 7.0 × .283 = 1.98

FREE HEIGHT = 1.98 + .20 + 220/200 = 3.28

LOAD AT SOLID = (3.28 - 1.98) 200 = 260#

STRESS AT SOLID = 60000 K. K = 1.2

= 72000 psi

ALLOWABLE STRESS AT SOLID = 74000 psi

M.S = 74/72 - 1 = .03

APPENDIX C

SEALING CLOSURE TEST PLAN

1.0 OBJECT AND SCOPE

The purpose of this test plan is to define the checkout testing to be performed with the rapid screening tester as well as to evaluate the performance of the 10 sealing closures in the rapid screening tester. These test efforts are in support of the Space Shuttle Auxiliary Propellant Valves contract with the NASA-Lewis Research Center, as defined in Tasks I and III-A of that contract. The overall objective of this contract is to develop valve technology for gaseous oxygen and gaseous hydrogen propellant valves for the auxiliary propulsion rocket engines operating at a thrust level 1500 lbs and at inlet pressures of 20 or 400 psia. The specific objective of this test plan is to demonstrate the ability of various valve sealing closures to reliably seal for up to one million cycles while operating at the previously mentioned inlet pressures and over a temperature range of 200 to 850°R.

Upon completion of the test program defined herein, sufficient confidence in several valve sealing closures will be established to commit them to the follow-on tradeoff study which results in the conceptual design and analysis of the actuators and supporting parts required to operate these sealing closures.

2.0 DESCRIPTION OF TEST HARDWARE

The rapid screening tester to be employed during this test program is defined on drawing No. L4688. This tester consists of a flexure guided poppet assembly which is connected to a hydraulic actuator. The actuator mass and that of the attached position indicator are decoupled from the poppet mass during the closure motion by means of a spring joint. The hydraulic servo system has the capability of individually varying valve seating load and valve actuation force during the closing motion. The poppet assembly is housed in a double wall pressure vessel which permits thermal conditioning over the required temperature range. Both the poppet and seat are readily removable to permit quick changeover of one sealing closure to another.

The rapid screening tester will be utilized to evaluate a total of 10 valve sealing closures. These sealing closures are defined on drawings No. L4677 through L4686. These sealing closures feature a variety of materials including hard metal, soft metal, teflon, and polyimide, as well as two geometrical interfaces, namely spherical and flat. These sealing closures are applicable to a wide variety of valves including poppets, ball, butterfly, gate, etc.

3.0 REFERENCES

The following documents are applicable as specified in this test plan.

3.1.1 Government

Mil-P-27401 pressurizing agent, nitrogen

3.1.2 The Marquardt Company

MPS 210 - Cleanliness Requirements for Reaction Control System Engines

4.0 GENERAL CONSIDERATIONS

The specific objectives, methods, and required data retrieval for each test of the sealing closure screening test program are detailed in Section 6 of this test plan. In the event additional supplementary tests are considered necessary, an amendment specifying the test(s) to be conducted will be added. The intent herein is to outline the general test requirements and controls to be employed during the test program.

4.1 Cleanliness and Handling

- All test fluids are to be passed through multiple filters between the storage tanks and test valves. Filters of at least 25μ absolute rating are to be installed upstream as close to the point of test valve hook-up as is possible.

Assembly of all poppet and seats into the rapid screening test fixture prior to testing shall be accomplished in the Building 32 Clean Room. During this assembly a filter of at least 25μ absolute rating shall be installed at the inlet to the rapid screening tester and this filter shall remain there during the entire test cycle to prevent the introduction of contaminants into the upstream side of the sealing closure interface. Proper alignment of the poppet to the seat and satisfactory leakage characteristics shall also be verified in the Clean Room prior to transfer of the assembly to the test site.

4.2 Instrumentation

The accuracy of all measuring and recording devices used during the program shall be verified prior to their use. Standard instrument inspection/calibration periods shall not be permitted to lapse during the subject test program. Test equipment description shall include the following minimum information:

- Descriptive Name
- Manufacturer's Name
- Manufacturer's Model Number
- Serial Number
- Range and Accuracy
- Frequency of Calibration
- Date of Last Calibration

4.3 Facility

- Final decision as to the adequacy of the test setup and conduct of the test, with the exception of the operation of the test facility, shall be at the discretion of the cognizant development engineer.
- All liaison concerning the test program shall be coordinated through the cognizant development engineer.
- The facility plumbing, seals and test setup constituents shall be of materials which are compatible with the test fluids being employed.

4.4 Test Discrepancy Procedure

4.4.1 Classification

4.4.1.1 Procedural Deviation

Procedural deviation is defined as any change to this test plan test procedure, test setup or instrumentation which affects valve operation or data reduction.

4.4.1.2 Performance Deviation

The objective of the screening test is to define sealing closure performance. Performance will be documented and evaluated in terms of the design requirements.

4.4.1.3 Malfunction

A malfunction is defined as any operation of the test facility equipment or human error which causes a discrepancy in the testing.

4.4.1.4 Failure

A failure is defined as the inability of the sealing closure to provide the function for which it was designed while operating within the specified environment and operating time and/or cycles as defined in this test plan.

4.4.1.5 Others

This classification includes all changes or deviations to the test plan or test procedure which are not defined in paragraphs 4.4.1.1, 4.4.1.2, 4.4.1.3 or 4.4.1.4. Included in this classification would be typographical and/or obvious errors which occurred during the preparation of the test procedures and/or test plan.

4.4.2 Discrepancy Evaluation

4.4.2.1 In the event that any problem occurs during testing, the Development Engineer Delegate and the Test Operation Engineer shall make a preliminary investigation. No action shall be taken that will destroy evidence.

4.4.2.2 Procedural deviations per paragraph 4.4.1.1 and others per paragraph 4.4.1.5 shall be dispositioned per paragraph 4.4.3.

4.4.2.3 If the problem is suspected to be malfunction per paragraph 4.4.1.3 [illegible] ure per paragraph 4.4.1.4, an investigation will be conducted by the Test Committee to determine the classification of the problem. The Test Committee shall be composed of the Development Engineer, the Test Operations Engineer, and the Project Engineer. When the classification is determined, the dispositon shall be made per paragraph 4.4.3. The step-by-step trouble shooting procedure shall be documented in the inspection sealing closure log bo[illegible] along with the reason for, and the results of each particular step.

4.4.3 Discrepancy Disposition

4.4.3.1 If the problem is defined per paragraph 4.4.1.5, a deviation sheet, Figure 1, shall be completed. The deviation sheet shall be initiated by the Development Engineering Delegate and be effective immediately. Written approval shall be obtained during the next regularly scheduled day shift if the discrepancy occurs during "off" shift operation.

4.4.3.2 In the event that the problem is defined as a procedural deviation or malfunction as described in paragraphs 4.4.1.1 or 4.4.1.3, a deviation sheet, Figure 1, shall be completed. The deviation sheet shall be originated by the Development Engineer and approved by the Project Engineer.

4.4.3.3 In the event the problem is defined as a failure under the terms of 4.4.1.4, a Failure/Malfunction Investigation will be conducted according to Quality Assurance Procedure (QAP) 12.2 except an Inspection Rejection Report (IRR) may not be written. The Failure/Malfunction Investigation may be completed during the Discrepancy Evaluation per paragraph 4.4.2.

4.4.4 Deviation Sheet Distribution

Copies of the approved deviation sheet shall be distributed as follows:

Development Test Log Book
Sealing Closure Log Book
Test Operations Engineer
Project Engineer

5.0 DOCUMENTATION

5.1 Witnesses

The cognizant development engineer shall be informed prior to the start of all tests and at any time when an unanticipated situation affecting the test setup, method of test item occurs. The development engineer or his designated representative shall be present during the conduct of tests.

5.2 Test Data and Identification

A history of tests completed shall be maintained in a test log book as well as in the sealing closure log book at the completion of the test program with the particular sealing closure. The data recorded shall be marked with the information necessary to completely identify it. The following items are considered as minimum required test identification and will be the responsibility of the cognizant test engineer.

- Sealing closure part number and serial number being tested.
- Type of test to be conducted, MTP Number, and applicable section identification.
- Type, range, and identification number of each measuring instrument used during the tests.
- Identification of test operator, facility, time, date, and test witness.

5.3 Data Storage and Processing

Components Engineering Group shall maintain a record of all data retrieved from the tests as supplied by test operations personnel. To facilitate intended test data processing and analysis, data reduction will be accomplished by the development engineer.

6.0 DETAILED TEST PROCEDURES

6.1 Rapid Screening Tester Checkout

6.1.1 Objective

The objective of this test series is to initially check out the rapid screening tester to verify its suitability for evaluation of the valve sealing closures. Checkout testing consists of the setting of the required strokes, seating force, opening force, and closing force in order to achieve the required sealing closure static loading and impact loading. In addition, the checkout will serve to demonstrate the tester cycling capability and to obtain data on the thermal conditioning performance, in particular, the time required to chill or cool the sealing closures, and the temperature changes incurred during cycling due to heat transfer into the operating fluid.

6.1.2 Tester Installation

The part number L4688, Rapid Screening Tester, shall be installed in the southeast section of Building 37, as shown in Figure 2, and shall be provided with the following test fluids.

1. Hydraulic pressure, regulated 0-3000 psi.
2. Filtered gaseous nitrogen, regulated 20-400 psi.
3. Hot air temperature regulated from 700-1000°R for a thermal conditioning.
4. Gaseous nitrogen temperature regulated from ambient to -150°R for thermal conditioning.

6.1.3 Instrumentation

a. The following high response (1000 cps) instrumentation shall be provided and shall be recorded on a dual-beam oscilloscope.

Hydraulic piston pressure differential, -3000 psi to + 3000 psi.

Poppet position, 0 to 1/2 inch.

b. The following steady state (1/2 cps) instrumentation shall be provided.

Seat temperature number 1 - 200 - 850°R

Seat temperature number 2 - 200 - 850°R

Tester jacket temperature - 200 - 850°R

Tester gaseous nitrogen inlet pressure - 20 - 400 psi

Hydraulic supply pressure, 0 - 3000 psi

Number of actuations, 0 - 100,000 cycles

Water displacement cylinder, 0 - 20 cubic centimeters

6.1.4 Test Sequence

To permit the checkout testing of the rapid screening tester, one of the sealing closures shall be installed in the tester. Prior to assembly of the sealing closure into the tester, all parts upstream of the sealing closure, which will come in contact with the operating gaseous nitrogen, shall be cleaned per MPS 210. Actual assembly of the tester and sealing closure shall occur in the clean room of Building 32, to assure a contaminant free system. Subsequently, the tester shall be moved to the test site in Building 37 and connected to various supplies and instrumentation as shown in Figure 2.

The test sequence is divided into a series of ambient tests and a series of tests under temperature conditioning.

a. Ambient Testing

The purpose of this test phase is to determine the actuator forces required to achieve the desired terminal velocity during closure at each of the three stroke conditons of interest. To achieve this objective, testing will be performed according to the matrix shown in Table I. In addition, the sealing closure leakage measurement technique will be evaluated by measuring leakage at 3 static load conditions, namely 20, 150, and 1500 lbs.

b. <u>Thermal Conditioning System Evaluation</u>

The purpose of the thermal conditioning checkout testing of the rapid screening tester is to evaluate thermal response characteristics of the tester as well as any adjustments which may be required to the actuator forces in order to achieve the desired response times and closing kinetic energy levels.

1. The first test in the series will consist of thermally conditioning the tester from ambient to 850°R and monitoring the time required to reach equilibrium sealing closure temperatures. Once the steady state sealing closure temperatures of 850°R has been achieved, this temperature shall be monitored for a period of 1/2 hour to determine temperature oscillations in the tester system. Subsequent to this period, the tester shall be operated at various frequencies and valve open times for each of the 3 stroke conditions of interest, namely, 0.010, 0.125, and 0.5 inches. Valve responses and valve open times to be evaluated shall be those which will result in the desired 500 and 2500 lb in.^{2}/sec^2 kinetic energies, which were determined during the phase A testing. During these tests, the sealing closure temperatures shall be monitored to observe temperature decay rates due to the cooling effect of the gaseous nitrogen flowing through the tester.

Upon completion of the hot thermal conditioning system evaluation, the same procedure will be utilized for evaluating this system at cold conditions (200°R).

Subsequent to the cold test series, leak checks of the sealing closure will be made at both the cold and hot conditions to verify the capability of the leak measuring equipment to operate at these temperatures.

6.1.4. Sealing Closure Screening Tests

The purpose of the sealing closure screening tests is to subject each of the sealing closures to cycling tests at hot, ambient, and cold temperatures and to make periodic leak checks to verify the sealing closure capability to seal effectively. In preparing the test matrix, primary attention was paid to the two inlet pressure operating levels to make certain that the test data is directly applicable to the valves that will be developed during the later tasks of this program. Verification of the leakage math model was considered a secondary objective.

It is expected that the high pressure valve configuration (400 psi inlet pressure) will employ a flexure guidance and will feature an impact load of approximately 500 lb in.^{2}/sec^2 for the non-retractable seal configurations. The retractable seal configurations will, of course, feature a lower impact load. Consequently, initial testing with all sealing closures will consist of operation at this particular pressure and impact load level. If the sealing closure passes 50,000 cycles successfully under these operating conditions, the impact load will be increased to 2500 lb in.^{2}/sec^2 and the valve will be cycled at an inlet pressure of 20 psia and with a poppet which will allow radial movement up to 0.004 inches. This configuration is considered representative of a sliding fit type low pressure valve. In the event that the sealing closure does not pass the 50,000 cycle test series at the 400 psia inlet pressure, the impact load will be reduced to a level compatible with the retractable seal configuration and the valve will be recycled up to 50,000 cycles under that condition. A failure of the valve to pass leakage requirements will be considered to have occurred when the leakage through the valve exceeds approximately 10 scc's per hour of helium at the particular operating pressure.

A total of 10 sealing closures will be evaluated. These sealing closures are presented in drawings L4677 through L4686 and are listed in Table II. The sealing closures include both flat and spherical interfaces and such material combinations as hard metal on hard metal, soft metal on hard metal, polyimide on hard metal, and teflon on hard metal. These sealing closures are applicable to a wide variety of valves as indicated in Table III.

Initial assembly of all sealing closures into the rapid screening tester will be accomplished in the clean room in Building 32. At that time also, alignment of each of the sealing closures will be verified. The movable portion of the rapid screening tester will then be taken from Building 32 and installed into the fixed portion of the rapid screening tester in Building 37. Thereafter, testing in accordance with Figure 3 will be performed. Data obtained during the testing will be recorded in Table IV for later application to the valve leakage math model. Upon completion of the test matrix of Figure 3, the movable portion of the rapid screening tester will be returned to the clean room for disassembly, inspection, and photographic coverage of the sealing closure condition.

TEST PROGRAM DEVIATION SHEET

Sealing Closure P/N__________ S/N__________ Date__________

Facility Building #37 Test No.__________ Originator__________

The following change (shall be/was) made to the Approved__________

Project__________

Test Plan Title Sealing Closure Screening

Instrumentation__________ Hardware__________, Facility (Mech.)__________

Other (State)__________

Change to:

____

____

____

____

____

This change affects runs__________ through __________.

Was:

____

____

____

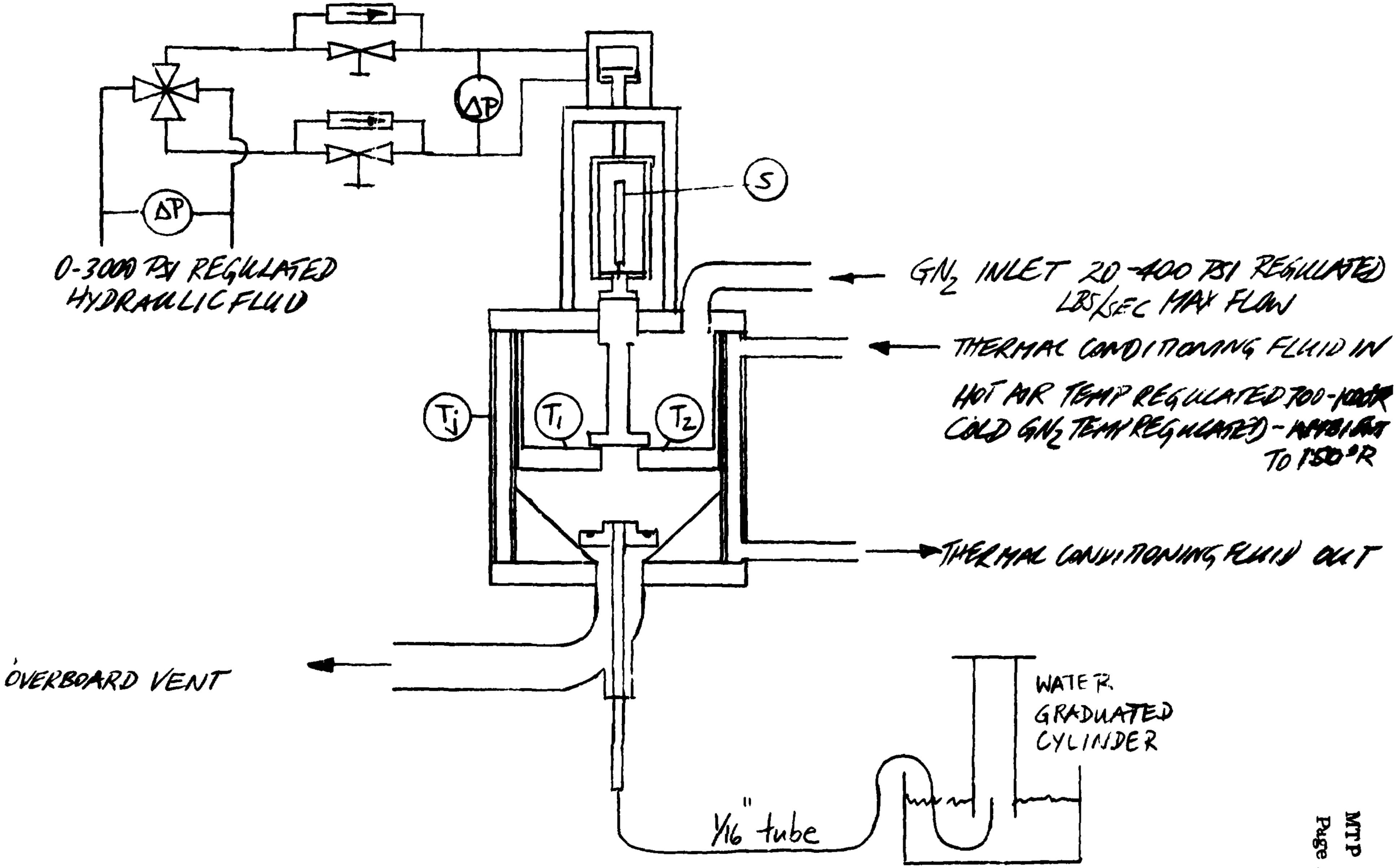
Reason:

____

____

Distribution: Development Test Log Book
Sealing Closure Log Book
Test Operations Engineer
Project Engineer

Figure 1



RAPID SCREENING TESTER INSTALLATION FIG. 2

TEST MATRIX Fig 3

BASE LINE TEST

Leak check @ 20 & 400 PSI & @ 3 STATIC LOADS
Stroke Verification
Response Verification

HIGH TEMPERATURE TEST - 850°R

Leak check @ 20 & 400 PSI @ 3 static loads
Cycle 600 times @ 400 PSI & 500 #in²/sec² IMPACT
Leak check @ 20 & 400 PSI @ 3 static loads
Cycle 5 times @ 400 PSI & 500 #in²/sec² IMPACT
Leak check @ 20 & 400 PSI @ 3 static loads
Cycle 10000 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI @ 3 static loads
Cycle 5 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI @ 3 static loads

IF FAILURE OCCURS

AMBIENT TEMPERATURE TEST - 530°R

Leak check @ 20 & 400 PSI @ 3 static loads
CYCLE 10000 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI @ 3 static loads
CYCLE 5 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI @ nominal static load
Cycle 20000 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI @ nominal static load
Cycle 5 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI @ nominal static load

IF FAILURE OCCURS

COLD TEMPERATURE TEST - 200°R

leak check @ 20 & 400 PSI @ 3 static loads
Cycle 10000 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI & Nominal static load
cycle 5 times @ 400 PSI & 500 #in²/sec² IMPACT
leak check @ 20 & 400 PSI & nominal static load

IF FAILURE OCCURS
IF NO DEGRADATION OCCURS

POST TEST INSPECTION

Disassemble
Verify alignment
record seal condition
photograph seal

2ND SEALING CLOSURE TEST

Repeat entire test series except:
minimum impact
20 psi cycling only

REPEAT POST TEST INSPECTION

HIGH IMPACT TEST

Repeat entire test series except:
2500 #in²/sec² IMPACT
.004" Radial poppet play
20 PSI cycling only

REPEAT POST TEST INSPECTION

TABLE I

TEST MATRIX – TESTER CHECKOUT

Stroke in.	Inlet Press PSI	Seat load lbs	Closing Force* lbs	Terminal Velocity in/s	Movement Time sec	Closing K.E. lbs in²/sec²
.125	400	20	10			
			15			
			20			
		50	20			
			50			
		100	20			
			50			
			100			
		150	10			
			20			
			50			
			100			
			150			
		200	150			
			200			
		1500	10			
			100			
			200			
.5	20	20	10			
			20			
		50	20			
			50			
		150	10			
			20			
			50			
		1500	10			
			20			
			50			
.010		20	2			
			5			
			10			
			20			

* LIMIT NOT TO EXCEED 750 #in²/sec² @ 400
" " " " 3000 " " 2

(TABLE I cont.)

Stroke in	Inlet Press PSI	Seat load lbs	Closing Force* lbs	Terminal Velocity in/s	Movement Time Sec	Closing K.E. lbs in²/sec²
.000	20	50	2			
			10			
		150	2			
			10			
		1500	2			
			10			
			50			
	400	20	2			
			5			
			10			
			20			
		50	2			
			10			
		150	2			
			10			
		1500	2			
			10			
			50			

* LIMIT NOT TO EXCEED 750 #in²/sec² @ 400 PSI
" " " " 3000 " " 20 "

TABLE II

LIST OF SEALING CLOSURES

DRAWING NO.	
L4677	Flat Polyimide Seat
L4678	Spnerical Copper Lip Seat, Bellows Loaded
L4679	Flat Metal Seat, Flexure Aligned Poppet
L4680	Flat Teflon Seat, Bellows Loaded
L4681	Spherical Teflon Seat, Bellows Loaded
L4682	Flat Metal Seat, Bellows Loaded
L4683	Flat Teflon Coated Lip Seat
L4684	Spherical Teflon Coated Lip Seat
L4685	Flat Polyimide Seat, Bellows Force Loaded
L4686	Spherical Polyimide Seat, Bellows Force Loaded

SEALING CLOSURE APPLICATION - TABLE II

	DRAWING NO.									
VALVE TYPE	L4677	L4678	L4679	L4680	L4681	L4682	L4683	L4684	L4685	L4686
Flat Poppet	X		X	X		X	X		X	
Spherical Poppet		X			X			X		X
Conical Poppet		X			X			X		X
Blade				X		X			X	
Gate				X		X			X	
Butterfly		X			X					X
Ball		X			X					X

TABLE IV
SEALING CLOSURE SCREENING TEST DATA SHEET

TEST DATE: ________ SEALING CLOSURE P/N ________ S/N ________

#	TEST PERFORMED	TEMP (1) °F	TEMP (2) °F	PRESS PSI	IMPACT #IN²/SEC	ALIGNM. "rad.	Static F lbs	leakage scc/Hr	Remarks

APPENDIX D

TEST PLAN 0192

1.0 **OBJECTIVE**

The objective of this test plan is to specify the tests and procedures to be used in acceptance testing the P/N X27449 cryogenic, coaxial shutoff valve.

2.0 **TEST PROCEDURES AND CONTROLS**

a. The valve shall be cleaned per MPS 210 and assembled in the clean room in Bldg. 32 and provided with an 18 micron absolute inlet filter at that time. This filter shall not be removed during acceptance testing in Bldg. 37. Upon completion of the acceptance tests, the valve and filter shall be returned to the clean room for removal of the filter and final packaging for shipment to the vendor.

b. During acceptance tests the valve shall not be opened unless a minimum of 5 psig gaseous nitrogen is present in the valve.

c. To facilitate installation of the upstream filter and downstream connections, the valve shall be provided with the P/N's T18602 and T18603 inlet and outlet fittings in the clean room. The installation drawing of the valve is P/N X28195.

2.1 **Instrumentation**

Instrumentation is required to measure the following parameters over the ranges and with the accuracies listed:

Parameter	Range	Accuracy
Coil Resistance (Valves)	0 - 40 ohms	± 0.2 ohms
Coil Resistance (LVDT)	0 - 200 ohms	± 1 Ω
Inlet Pressure	0 - 650 psig	± 5 psi
Valve Pressure Differential	0 - 40 psi	± 0.5 psi
Flow Rate	2.0 - 3.0 lbs/sec GN_2 @ 375 psia, ambient temperature	± .05 lbs/sec
Internal Leakage	50 scc per 3 minutes	± 1 scc
Pull In Current	0 - 1.5 amps	± .01 amps
Response	0 - 50 ms	± 0.5 ms

2.1 Instrumentation - Continued

Parameter	Range	Accuracy
Position Indication*	0 - 0.3"	± .015"
Voltage	0 - 36 volts	± 0.2 volts
Number of Actuations	0 - 10,000	± 1

*Utilizing signal conditioning equipment compatible with KAVLICO P/N GM 5854.

3.0 TEST PROCEDURE

The acceptance test data sheet identified as Table I shall be completed during the acceptance test sequence.

3.1 Coil Resistance

Measure and record the coil resistance between the two wires of each of the pilot valves at ambient temperature.

Measure and record the coil resistances of the position transducer primary (Pins A to B, black and white wires); left secondary (Pins C to E, blue and blue stripe wires); and right secondary (Pins F to D, red stripe and red wires) at ambient temperature.

3.2 Proof Pressure

Slowly pressurize the valve with gaseous nitrogen or gaseous helium to a pressure of 1,000 psi for 1 minute. No permanent distortion or damage shall result.

3.3 External Leakage

With the valve closed and the outlet port capped, slowly pressurize the main valve with 650 psi GN_2 and the pilot valves with 650 psi helium. Look for gas leakage with snoop at all flange joints and at the vent port to the LVDT cavity. Open the valve and repeat the leak check at both pilot valve joints and mounting flanges.

3.4 Internal Leakage

Pressurize the main valve and the pilot valves with 400 psig GN_2. Measure internal leakage through the main valve as well as at the vent ports of both pilot valves. Cap the outlet of the main valve and open the valve. Do not supply GN_2 pressure to the main valve until the pilot valves are energized. Again measure the leakage at the vent port of each pilot valve.

3.5 Pull In Current

Pressurize the pilot valves with GN_2 or helium to 650 psi. Slowly increase the actuation voltage and monitor the coil current at the time each valve operates.

3.6 Opening Response

Pressurize the main valve with 375 psig GN_2 and the pilot valves with 375 psig helium. Set the oscilloscope triggering to positive, sensitivity to approximately 20 v/cm and time base to 2 ms/cm. Monitor the LVDT output on the upper trace and the current to the pilot valves on the lower trace. Set the pulser to 28 VDC (at the pilot valves) and the pulse width to 30 ms. Verify that the plumbing to the valve is as in Figure 1 and includes the simulated downstream injector volume. Open the valve and obtain polaroid photographs of the two traces. Identify the photo with the date of test, valve P/N and S/N, voltage, main valve pressure and medium, pilot valves pressure medium, sensitivity and time base. Record the opening response from the start of the current rise (current trace) to completion of the valve poppet travel (LVDT trace).

3.7 Closing Response

All settings shall be the same as in Paragraph 3.6 except that the voltage shall be recorded instead of the current and the oscilloscope triggering shall be set to negative. Verify that the throttling valve in Figure 1 is adjusted so that the flow rate through the main valve is 2.57 lbs/sec GN_2. Closing response is measured from the time of electrical dropout (voltage trace) to the completion of valve poppet travel (LVDT trace).

3.8 Pressure Drop

With the setup as shown in Figure 1, close the throttling valve, pressurize the main valve and pilot valves with 375 psig GN_2 and open the main valve. Gradually open the throttling valve. Record the pressure drop across the main valve at a GN_2 flow rate of 2.57 lbs/sec.

3.9 Position Indication

This test can be verified from the response photos.

3.10 Failsafe Close

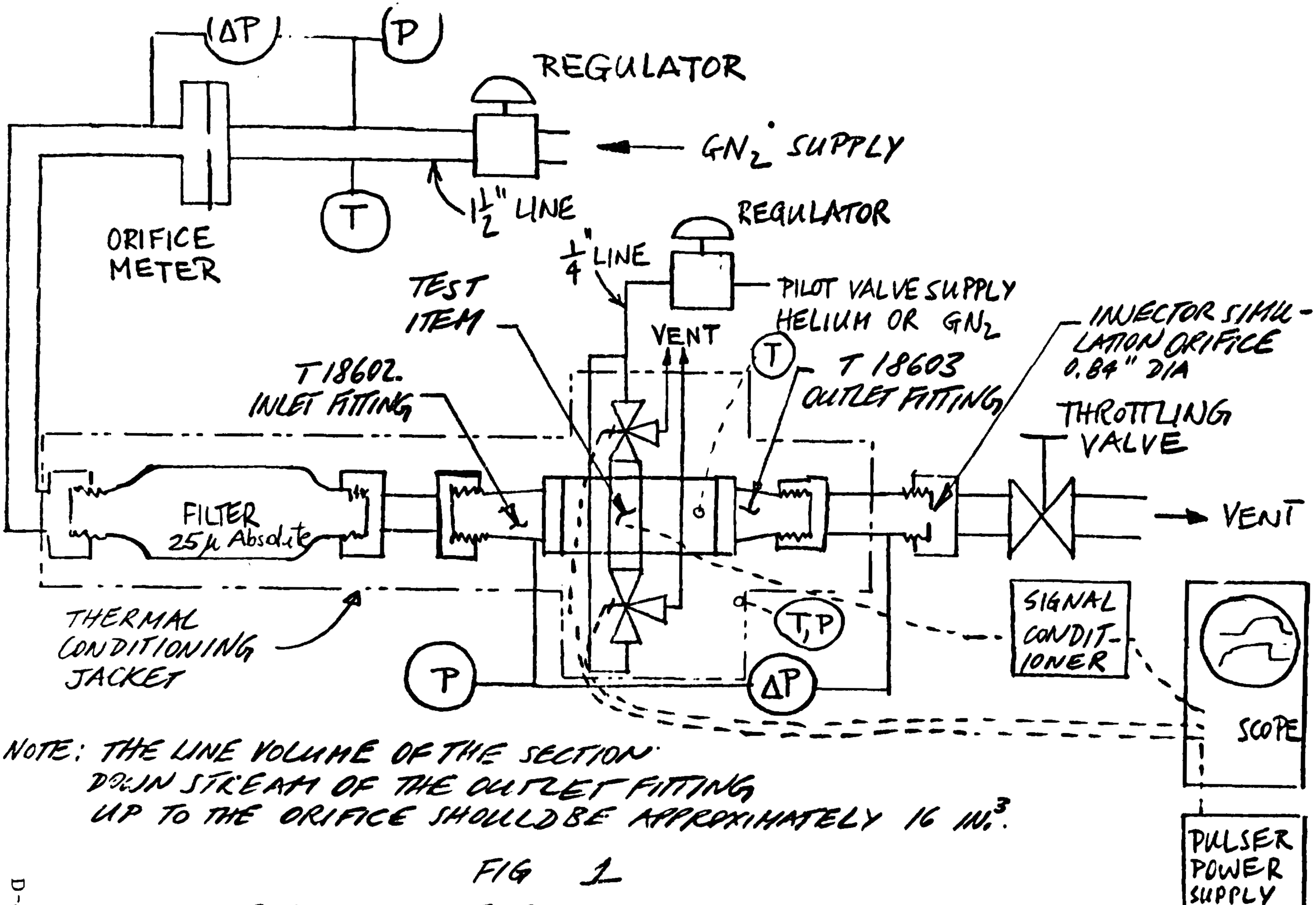
With the valve open as in Paragraph 3.8, bleed the GN_2 supply pressure to zero and verify (from the LVDT) that the valve closes.

ACCEPTANCE TEST DATA SHEET

VALVE P/N X27449, S/N

TEST NO.	PARAMETER	TEST CONDITION	DESIGN GOAL	ACTUAL
3.1	Coil Resistances	At Ambient Temperature		
	P/N 13087 NO Valve	Across the Two Wires	28 Ω	
	P/N 13086 NC Valve	Across the Two Wires	28 Ω	
	LVDT Primary	Pins A to B	200 Ω	
	LVDT Left Secondary	Pins C to E	50 Ω	
	LVDT Right Secondary	Pins F to D	50 Ω	
3.2	Proof Pressure	1000 PSI	1000 PSI	
3.3	External Leakage	650 PSIG GN_2	Zero Bubbles	
3.4	Internal Leakage	400 PSIG GN_2	1000 scc/hr	
	Main Valve	Valves De-energized	Total Maximum	
	NO Pilot Vent Port	Valves De-energized	Either Position	
	NC Pilot Vent Port	Valves De-energized		
	NO Pilot Vent Port	Valves Energized		
	NC Pilot Vent Port	Valves Energized		
3.5	Pull In Current	650 PSIG, Ambient Temp.	0.6 Am.	
	NO Pilot Valve		Per Pilot Valve	
	NC Pilot Valve			
3.6	Opening Response	375 PSIG GN_2, 28 VDC, Ambient Temp., Injector Vol. Simulation, 375 PSIG He to Pilot Valves.	16 ms	
3.7	Closing Response	375 PSIG GN_2, 28 VDC, Ambient Temp., Injector Vol. Simulation, 375 PSIG He to Pilot Valves, 2.57 lbs/sec GN_2 flowing.	15 ms	
3.8	Pressure Drop	375 PSIG GN_2 Inlet Pressure, Back Pressure Throttled to Obtain Desired Flow Rate, Ambient Temperature.	2.57 lbs/sec GN_2 @ ΔP = 25 PSI Max	
3.9	Position Indication	Verify Proper Operation During Response Tests.	Minimum 1/2" Deflection for Stroke	
.10	Failsafe Close	Bleed Off All Pressures to Zero.	Valve Should Close	

TMC A1833-1



NOTE: THE LINE VOLUME OF THE SECTION DOWN STREAM OF THE OUTLET FITTING UP TO THE ORIFICE SHOULD BE APPROXIMATELY 16 IN.^{3}

FIG 1

TEST FIXTURE TEST SET UP

APPENDIX E

TEST PLAN 0193

1.0 OBJECT AND SCOPE

The purpose of this test plan is to define the testing to be performed with the two valve test fixtures. These test efforts are in support of the Space Shuttle Auxiliary Propellant Valves, contract with the NASA-Lewis Research Center, as defined in Tasks I and III-B of that contract. The overall objective of this contract is to develop valve technology for gaseous oxygen and gaseous hydrogen propellant valves for the auxiliary propulsion rocket engines operating at a thrust level 1500 lbs and an inlet pressure of 400 psia. The specific objective of this test plan is to demonstrate the ability of various valve subcomponents to reliably perform for up to one million cycles while operating at the previously mentioned inlet pressures and over a temperature range of 200 to 850°R.

Upon completion of the test program defined herein, sufficient confidence in various valve subcomponents will be established to permit the selection of subcomponents for the final valve design layout.

2.0 DESCRIPTION OF TEST HARDWARE

The valve test fixtures to be evaluated during this test program are defined by drawing P/N's X27449 and X28400. Both of these test fixtures are basically coaxially, pneumatically operated poppet valves. They differ primarily in the type of subcomponents incorporated in each configuration and in the pneumatic actuation mode. The P/N X27449 valve is pressure actuated to open and to close; the P/N X28400 valve employs an actuator which vents a single cavity to open and which repressurizes this cavity to close.

To facilitate testing of these test fixtures, inlet and outlet adapter fittings have been fabricated which are identified as P/N's T18602 and T18603, respectively. In addition, the Test Department will provide an inlet filter of approximately 25 microns absolute rating which will be connected directly to the test fixture inlet fitting during initial buildup in the Bldg. 32 Clean Room. Furthermore, the Test Department will provide a section of 1-1/2 inch tubing which mates with the outlet fitting and which features a line volume of approximately 16 cubic inches and an orifice of 0.84 inches in diameter to simulate oxidizer injector dribble volume and oxidizer injector orifice characteristics.

A schematic of the setup to be employed during this test program is shown in Figure 1. An installation drawing of the P/N X27449 test fixture is also available and is identified as Drawing X28195.

3.0 REFERENCES

The following documents are applicable as specified in this test plan.

3.1.1 Government

Mil-P-27401 pressurizing agent, nitrogen.

3.1.2 The Marquardt Company

MPS 210 - Cleanliness Requirements for Reaction Control System Engines.

4.0 GENERAL CONSIDERATIONS

The specific objectives, methods, and required data retrieval for each test of the valve test fixture test program are detailed in Section 6.0 of this test plan. In the event additional supplementary tests are considered necessary, an amendment specifying the test(s) to be conducted will be added. The intent herein is to outline the general test requirements and controls to be employed during the test program.

4.1 Cleanliness and Handling

- All test fluids are to be passed through multiple filters between the storage tanks and test valves. A filter of at least 25 μ absolute rating is to be installed upstream as close to the point of test fixture hook-up as is possible.

Assembly of all valve test fixtures prior to testing shall be accomplished in the Building 32 Clean Room. During this assembly, a filter of at least 25 μ absolute rating shall be installed at the inlet to the test fixture and this filter shall remain there during the entire test cycle to prevent the introduction of contaminants into the upstream side of the test fixture.

4.2 **Instrumentation**

The accuracy of all measuring and recording devices used during the program shall be verified prior to their use. Standard instrument inspection/calibration periods shall not be permitted to lapse during the subject test program. Test equipment description shall include the following minimum information:

- **Descriptive Name**
- **Manufacturer's Name**
- **Manufacturer's Model Number**
- **Serial Number**
- **Range and Accuracy**
- **Frequency of Calibration**
- **Date of Last Calibration**

4.3 **Facility**

- **Final decision as to the adequacy of the test setup and conduct of the test, with the exception of the operation of the test facility, shall be at the discretion of the cognizant development engineer.**
- **All liaison concerning the test program shall be coordinated through the cognizant development engineer.**
- **The facility plumbing, seals and test setup constituents shall be of materials which are compatible with the test fluids being employed.**

4.4 Test Discrepancy Procedure

4.4.1 Classification

4.4.1.1 Procedural Deviation

Procedural deviation is defined as any change to this test plan, test procedure, test setup or instrumentation which affects valve operation or data reduction.

4.4.1.2 Performance Deviation

The objective of the screening test is to define sealing closure performance. Performance will be documented and evaluated in terms of the design requirements.

4.4.1.3 Malfunction

A malfunction is defined as any operation of the test facility equipment or human error which causes a discrepancy in the testing.

4.4.1.4 Failure

A failure is defined as the inability of the sealing closure to provide the function for which it was designed while operating within the specified environment and operating time and/or cycles as defined in this test plan.

4.4.1.5 Others

This classification includes all changes or deviations to the test plan or test procedure which are not defined in paragraphs 4.4.1.1, 4.4.1.2, 4.4.1.3 or 4.4.1.4. Included in this classification would be typographical and/or obvious errors which occurred during the preparation of the test procedures and/or test plan.

4.4.2 Discrepancy Evaluation

4.4.2.1 In the event that any problem occurs during testing, the Development Engineer Delegate and the Test Operation Engineer shall make a preliminary investigation. No action shall be taken that will destroy evidence.

4.4.2.2 Procedural deviations per paragraph 4.4.1.1 and others per paragraph 4.4.1.5 shall be dispositioned per paragraph 4.4.3.

4.4.2.3 If the problem is suspected to be malfunction per paragraph 4.4.1.3 or failure per paragraph 4.4.1.4, an investigation will be conducted by the Test Committee to determine the classification of the problem. The Test Committee shall be composed of the Development Engineer, the Test Operations Engineer, and the Project Engineer. When the classification is determined, the dispositon shall be made per paragraph 4.4.3. The step-by-step trouble shooting procedure shall be documented in the inspection sealing closure log book along with the reason for, and the results of each particular step.

4.4.3 <u>Discrepancy Disposition</u>

4.4.3.1 If the problem is defined per paragraph 4.4.1.5, a deviation sheet, Figure 2, shall be completed. The deviation sheet shall be initiated by the Development Engineering Delegate and be effective immediately. Written approval shall be obtained during the next regularly scheduled day shift if the discrepancy occurs during "off" shift operation.

4.4.3.2 In the event that the problem is defined as a procedural deviation or malfunction as described in paragraphs 4.4.1.1 or 4.4.1.3, a deviation sheet, Figure 2, shall be completed. The deviation sheet shall be originated by the Development Engineer and approved by the Project Engineer.

4.4.3.3 In the event the problem is defined as a failure under the terms of 4.4.1.4, a Failure/Malfunction Investigation will be conducted according to Quality Assurance Procedure (QAP) 12.2 except an Inspection Rejection Report (IRR) may not be written. The Failure/Malfunction Investigation may be completed during the Discrepancy Evaluation per paragraph 4.4.2.

4.4.4 <u>Deviation Sheet Distribution</u>

Copies of the approved deviation sheet shall be distributed as follows:

Development Test Log Book

Sealing Closure Log Book

Test Operations Engineer

Project Engineer

5.0 **DOCUMENTATION**

5.1 **Witnesses**

The cognizant development engineer shall be informed prior to the start of all tests and at any time when an unanticipated situation affecting the test setup, method of test item occurs. The development engineer or his designated representative shall be present during the conduct of tests.

5.2 **Test Data and Identification**

A history of tests completed shall be maintained in a test log book as well as in the sealing closure log book at the completion of the test program with the particular sealing closure. The data recorded shall be marked with the information necessary to completely identify it. The following items are considered as minimum required test identification and will be the responsibility of the cognizant test engineer.

- Test fixture part number and serial number being tested.
- Type of test to be conducted, MTP Number, and applicable section identification.
- Type, range, and identification number of each measuring instrument used during the tests.
- Identification of test operator, facility, time, date, and test witness.

5.3 **Data Storage and Processing**

Components Engineering Group shall maintain a record of all data retrieved from the tests as supplied by test operations personnel. To facilitate intended test data processing and analysis, data reduction will be accomplished by the development engineer.

6.0 DETAILED TEST PROCEDURES

6.1 Objective

The objective of this test phase is to evaluate the two valve test fixtures and their subcomponents for compatibility the requirements of the Space Shuttle Auxiliary Propellant Valve. The test series consists of the measurement of various valve performance characteristics such as pressure drop response, leakage, etc. while the valve test fixtures are being cycled 100,000 times at ambient, +390°F, and -260°F.

6.2 Valve Test Fixture Installation

The valve test fixtures shall be installed in the test setup shown schematically in Figure 1 and located in the southwest corner of Bldg. 37 at the TMC, Van Nuys Test Facility. During the test series, the test item will be located in an environmental chamber which will be supplied with electrically heated gaseous nitrogen to achieve the +390°F condition and with liquid nitrogen cooled gaseous nitrogen to achieve the -260°F condition. The main gaseous nitrogen supply to the test item must be capable of supplying 3 pounds per second of gaseous nitrogen at a nominal inlet pressure of 375 psia and at a maximum inlet pressure of 450 psia. A separate supply system capable of furnishing either gaseous nitrogen or gaseous helium through a 1/4 inch line at pressures up to 750 psia must be provided for operation of the test item pilot valves.

6.3 Instrumentation

Instrumentation is required to measure the following parameters over the ranges and with the accuracies listed:

Parameter	Range	Accuracy
Coil Resistance (Valves)	0 - 40 ohms	± 0.2 ohms
Coil Resistance (LVDT)	0 - 250ohms	± 5 ohms

Parameter	Range	Accuracy
Inlet Pressure	0 - 650 psig	± 5 psi
Test Fixture Pressure Differential	0 - 40 psi	± 0.5 psi
Flow Rate	2.0 - 3.0 lbs/GN_2 @ 375 psia, ambient temperature	± .05 lbs/sec
Internal Leakage	50 scc per 3 minutes	± 1 scc
Pull In Current	0 - 1.5 amps	± .01 amps
Response	0 - 50 ms	± 0.5 ms
Test Fixture Seat Temperature	-320 to +450°F	± 5°F
Environmental Chamber Wall Temperature	-320 to +450°F	± 5°F
Environmental Chamber Pressure	0 - 100 psig	± 2 psi
Position Indication*	0 - 0.3 "	± .015 "
Voltage	0 - 36 volts	± 0.2 volts
No. of Actuations	0 - 100,000	± 1

*Utilizing signal conditioning equipment compatible with KAVLICO P/N GM 5854.

6.4 Test Sequence

Each of the valve test fixtures to be evaluated shall be thoroughly cleaned per MPS 210 and subsequently assembled in the Clean Room in Bldg. 32. During this assembly an upstream filter shall be installed to assure a contaminant free system when the test fixture is removed to the test area in Bldg. 37. The test item shall then be installed in the test setup according to the schematic shown in Figure 1.

The test sequence consists of a set of baseline tests which are repeated periodically while the test fixture accumulates 100,000 cycles while at + 390°F, ambient, and -260°F temperatures.

a. Baseline Tests

The baseline test data sheet identified as Table 1 shall be completed during the performance of the baseline tests. These tests are scheduled after initial installation both at ambient and at hot temperature, after 1,000 cycles at hot and at cold temperature, after 10,000 cycles at cold and at ambient temperature, after 50,000 cycles at ambient temperature, and again after 100,000 cycles at ambient temperature. Each of the baseline tests is briefly discussed in the following paragraphs.

1. Internal Leakage

All leak checks shall be performed with the main valve and the pilot valves pressurized with Helium to 400 psig.

In the closed position, measure internal leakage through the main valve as well as at the vent ports of both pilot valves. To determine leakage through the upstream piston seal of the P/N X27449 valve, remove the pressurant supply to the normally closed pilot valve and cap the supply port, then measure leakage through the vent port. Subsequently, reconnect the pressurant supply to the normally closed pilot valve. To measure the redundant pressure balancing seal leakage of the P/N X27449 valve, install the P/N T12125 test fixture into the outlet of the valve being careful not to exert a force greater than 10 lbs. Measure leakage through this fixture. To measure piston rod seal leakage of the P/N X27449 valve, energize the normally closed pilot valve only and again measure leakage through the test fixture. The difference in the leakage rates observed during the two measurements constitutes the piston rod seal leakage. Deenergize the N.C. pilot valve and remove the test fixture.

Cap the outlet of the main valve and open the valve. Again measure the leakage at the vent ports of each pilot valve. To measure the downstream piston seal leakage of [illegible] valve, remove the pressurant supply of the normally open pil[illegible] the valve supply port. Measure the leakage through the vent port of [illegible] pilot valve.

To measure bellows leakage of the P/N X28400 valve, disconnect the pressurant supply to the pilot valves and cap the valve supply port. Energize the pilot valves and measure leakage through the vent port.

2. Opening Response

Pressurize the main valve with 375 psig GN_2 and the pilot valves with 375 psig helium. Set the oscilloscope triggering to positive, sensitivity to approximately 20 v/cm and time base to 2 ms/cm. Monitor the LVDT output on the upper trace and the current to the pilot valves on the lower trace. Set the pulser to 28 VDC (at the pilot valves) and the pulse width to 30 ms. Verify that the plumbing to the valve is as in Figure 1 and includes the simulated downstream injector volume. Open the valve and obtain poloroid photographs of the two traces. Identify the photo with the date of test, valve P/N and S/N, voltage, main valve pressure and medium, pilot valves pressure and medium, sensitivity and time base. Record the opening response from the start of the current rise (current trace) to completion of the valve poppet travel (LVDT trace).

3. Closing Response

All settings shall be the same as in Paragraph 2 except that the voltage shall be recorded instead of the current and the oscilloscope triggering shall be set to negative. Verify that the throttling valve in Figure 1 is adjusted so that the pressure drop across the main valve is 25 psi. Closing response is measured from the time of electrical dropout (voltage trace) to the completion of valve poppet travel (LVDT trace).

4. Pressure Drop

With the setup as shown in Figure 1, close the throttling valve, pressurize the main valve and pilot valves with 375 psig GN_2 and open the main valve. Gradually open the throttling valve. Record the pressure drop across the main valve at a GN_2 flow rate of 2.57 lbs/sec.

b. Cycling Tests

The purpose of the cycling tests is to demonstrate the ability of the various test fixture subcomponents to reliabiy perform 100,000 cycles of operation and to further project the ability of these components to reach the 1,000,000 cycle life goal. The test program has been divided into four cycling periods at various temperatures as follows: from initial installation to 1,000 at +390°F; from 1,000 to 10,000 cycles at -260°F; from 10,000 cycles to 50,000 cycles at ambient temperature; and again from 50,000 to 100,000 cycles at ambient temperature. Baseline tests will be performed at the beginning and at the end of each of the cycling periods as discussed in the preceding section.

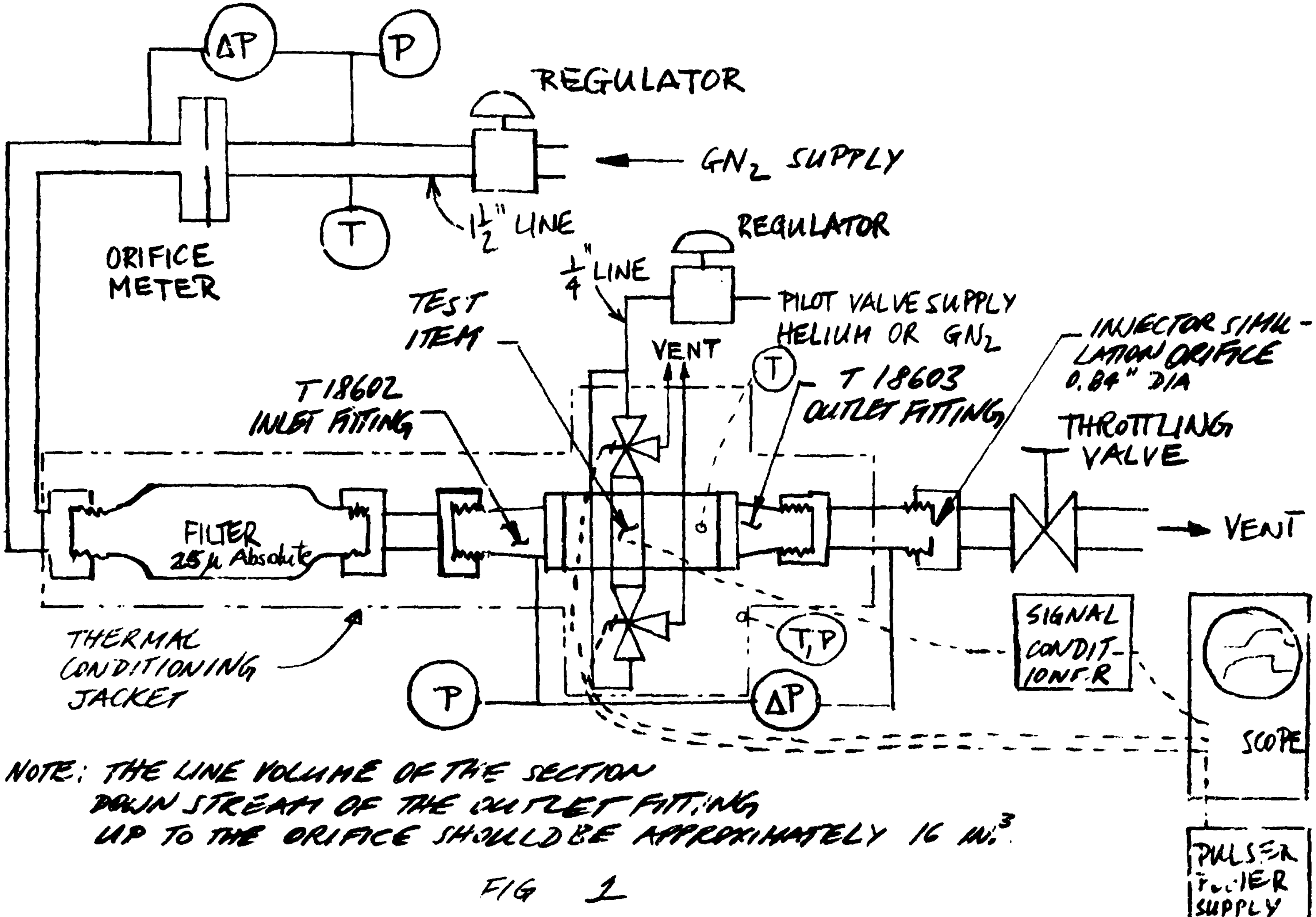
During the initial buildup of the valve subcomponents into the valve test fixture, photographic coverage as well as dimensional inspection of all valve subcomponents will be obtained. These same data will again be obtained at the conclusion of the cycling program and a comparison of the original data with the final data will be made to determine what, if any, where, and/or degradation has occurred to the valve subcomponents.

Actual cycling will be performed at a frequency of 10 cps and with an electrical pulse width of 30 ms. Gaseous nitrogen at 450 psia will be supplied to the main valve.

c. Contamination Sensitivity Test

To determine the sensitivity of the polyimide seal in the P/N X28400 valve to contamination, remove the valve to the clean room in Building 32. Disassemble the downstream body from the valve. Place two nickel particles each of 30 and 60 micron size on the center of the sealing land 90° apart. Reassemble the valve making certain that the particles do not fall off the sealing land. Pressurize the valve with 400 psi Helium and without actuating it and leak check the main seat. Cycle the valve three times and again leak check. Disassemble valve and inspect locations where the particles were placed.

Repeat the above procedure except place two .001" thick stainless steel wires across the sealing land 180° apart instead of the nickel particles.



NOTE: THE LINE VOLUME OF THE SECTION DOWN STREAM OF THE OUTLET FITTING UP TO THE ORIFICE SHOULD BE APPROXIMATELY 16 in.^{3}

FIG 1

TEST FIXTURE TEST SET UP

TEST PROGRAM DEVIATION SHEET

Sealing Closure P/N ________________ S/N ________________ Date ______________

Facility Building #3. ________________ Test No. ______________ Originator ______________

The following change (shall be/was) made to the Approved ______________

Project ______________

Test Plan Title Valve Test Fixture Tests

Instrumentation ________________ Hardware ______________, Facility (Mech.) __________

Other (State) ________________

Change to:

____

____

____

____

____

This change affects runs __________ through __________.

Was:

____

____

____

Reason:

____

____

Distribution: Development Test Log Book
Sealing Closure Log Book
Test Operations Engineer
Project Engineer

Figure 2

FACILITY CERTIFICATION CONTROL INSTRUMENTATION EQUIPMENT LIST

BUILDING 37

PARAMETER/ COMPONENT	TYPE/RANGE	MANUFACTURE/ MODEL	IDENT. MT NUMBER	CALIB. DATE	DUE DATE
Coil Resistance	Wheatstone Bridge 0-10,000,000 ohms	Rubicon 1052	110 80-253	11-30-71	6-5-72
Test Fixture Inlet Press.	8-1/2" Dia. 0 - 750 psig	Heise	360 25-500	7-29-71	3-18-72
Test Fixture Δp Press.	0 - 50 psi	Barton	360 30-416	7-29-71	1-28-72
Pilot Valve Press.	0 - 800 psi	Ashcroft	360 46-225	8-27-71	4-7-72
Venturi Press.	8-1/2" Dia. 0 - 400 psi	Heise	360 25-352	8-25-71	4-6-72
Pull in Current	Amperes D.C. 0 - 1.5	Weston 931	530 10-183	6-3-71	12-23-71
Response	Oscilloscope	Tektronix 502A	620 00-190	9-17-71	5-1-72
Power Supply	0 - 10 amps 0 - 40 volts	Harrison Labs	650 50-258	10-13-71	10-11-72
Test Fixture Seat Temp.	Strip Recorder -350°F +600°F	Bristol	720 14-127	8-3-71	1-26-72
Body Temp. Box Temp. Box Input Temp.	"	"	"	"	"
Venturi GN_2 Temp.	"	"	"	"	"
Response Photos	Oscilloscope Camera	Tektronix	150 30-104	8-9-71	2-23-72
Actuations	Scaler Timer 0-99,999	Anadex CF 604R		8-2-71	1-26-72
Voltage	Oscillator	Hewlett Packard 200 CD	610 10-169	2-22-71	2-21-72
Voltage	Decibles 1 MW	Hewlett Packard 400 EAC	530 65-253	8-2-71	1-26-72
Pulser		TMC Model 14	685 00-127	9-15-71	3-20-72
Thermocouple Ref Jct		Pace	730 00-153	8-3-71	1-26-72

(Continued)

FACILITY CERTIFICATION CONTROL INSTRUMENTATION EQUIPMENT LIST
BUILDING 37

PARAMETER/ COMPONENT	TYPE/RANGE	MANUFACTURE/ MODEL	IDENT. MT NUMBER	CALIF. DATE	DUE DATE
Strain Gage Pwr. Supply	Mod. PSG-3 0 - 8 V	Kintel PSG-3	650 60 - 137	1-11-71	1-31-72
Stop Watch	0-30 Sec. Dial 0-15 Min. Dial	Minerva	N/A	9-4-71	7-2-72
Heater Control	Pyrovane 0-1000 °F	Minneapolis Honeywell	200 00-485	9-23-71	5-4-72
Cal. Rvd Control	Pyrovane 0-1000°F	Minneapolis Honeywell	200 00-480	9-23-71	1-4-72
GN_2 Flow Rate	Venturi .655 Dia	TMC	350 00-655	N/A	N/A
Internal Leakage	Burette 0-10 ML			N/A	N/A
Internal Leakage	Graduate 0-200 ML	Corning Glass		N/A	N/A
Heater Press.	Dial 0-100 psig	Acragage	Indicator only	N/A	N/A
GN_2 Main Line Press.	Dial 0-2000 psig	U.S. Gage	Indicator only	N/A	N/A