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FINAL REPORT

THE DESIGN AND DEVELOPMENT OF A MINIATURE BI-STABLE LATCHING SOLENOID VALVE FOR LOW THRUST RESISTOJETS

BY

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Prepared Under Contract NAS 1-9601 by

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For

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
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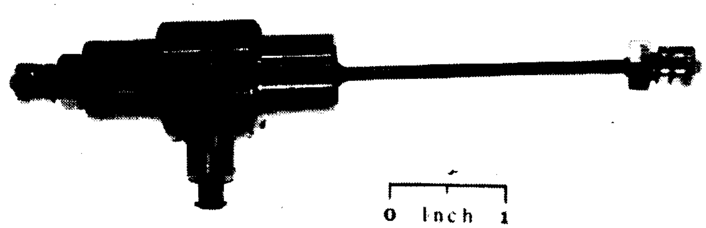
INTRODUCTION

Under NASA-Langley Contract NAS 1-9601, The Marquardt Company has designed, fabricated and development tested a miniature high performance magnetically latching solenoid valve, suitable for gaseous or liquid service with a wide variety of fluids over a wide temperature range. The valve features no sliding fits (flexure mounted moving element), an equivalent orifice of approximately .050 inches diameter, magnetic latching in both open and closed position, response time of 2.0 to 3.0 milliseconds over a pressure range of 0-400 psi, temperature range of -40 to +300°F and voltage range of 20 to 32 vdc, and a unit weight of 0.26 lb. A photograph, section view and tabulation of the valve's characteristics is presented in Figure 1.

The valve design and development testing described herein was performed to provide a component to control fluid flow to a Ruggedized Resistojet Thrustor under development by The Marquardt Company. The resultant valve was demonstrated as totally compatible with all requirements of this application. Demonstrated design margins indicate the valves suitability for application to many current and anticipated aerospace fluid control applications requiring long term compatibility, high reliability, high cycle life, fast repeatable response, low power consumption and low weight.

LATCHING SOLENOID VALVE
Two Way Miniature

P/N X28050



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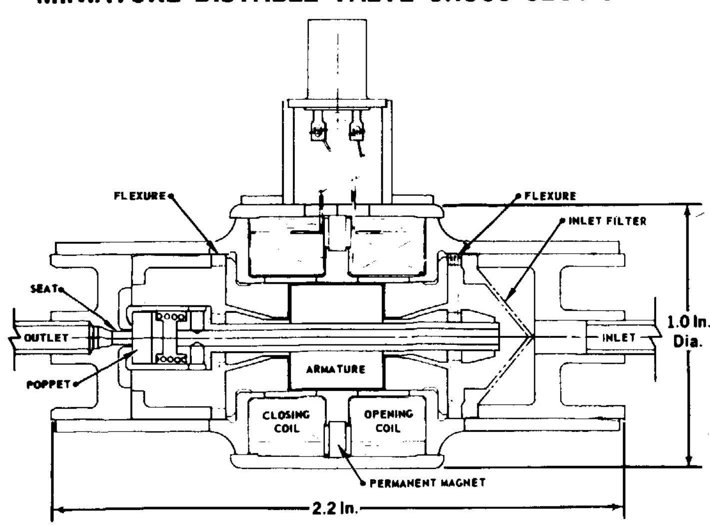
MINIATURE BISTABLE VALVE CROSS SECTION



- Valve Type . . . . Poppet, Coaxial Flow, Bistable
- Operating Fluids . . . . Hydrazine, GN_2, He, H_2O, NH_3, CO_2
- Weight . . . . 0.26 LB.
- Operating Voltage . . . . 15 - 32 VDC   −40° TO +300°F
- Power . . . . 25 Watts Steady State at 30 VDC 70°F
  0.25 Watt Sec. per Cycle
- Pressure Drop. . . . . < 1 PSI at 0.001 PPS H_2O
- Nominal Operating Pressure . . 0 - 400 PSIA
- Proof Pressure . . . . 800 PSIA - MIN.
- Leakage . . . . < 1 SCCH He 0 - 400 PSIA
- Opening Response . . . . 2.5 MS at 25 VDC, 70°F, 400 PSIA
  (Signal On to Full Open)
- Closing Response . . . . 2.5 MS at 25 VDC, 70°F, 400 PSIA
  (Signal On to Full Close)
- Cycle Life . . . . >100,000
- Program Application. . . . . . . Rugged Resistojet (NASA-Langley)
  0.1 Monopropellant Systems
- Special Features . . . . Integral 25 μ Abs. Filter
  No Electrical Power Required
  to Hold Open or Closed.
  Seat Seal of AF-E-102.
  All Welded Construction.
  Flexure Guided—No Sliding Fits.

A71-11-125-35A

Figure 1

SUMMARY

The program described herein documents the design approach and development demonstration of a flightweight, miniature, magnetic latching shut-off valve, suitable for use with the Ruggedized H_2 and NH_3 Resistojet and the Biowaste Resistojet. This effort was initiated on 22 January 1971 under NASA Contract NAS 1-9601 for the NASA-Langley Research Center. The contracting agency Project Manager was Mr. Earl Van Landingham.

The design criteria established provided for compatibility with specified ruggedized resistojet propellants as well as the biowaste and other propulsion system propellants. This approach assured meeting the primary goals of the program with margin and providing a valve that with little or no modification could be adapted to other applications such as the Biowaste Resistojet.

The success of this objective is demonstrated by the receipt of a contract from the Jet Propulsion Laboratory to build additional valves for their use and to supplement the resistojet program with hydrazine tests. The four valves built for the resistojet program are shown in Figure 2 along with the common design valves for the subsequent hydrazine program.

As a result of testing performed to date, the valve design has demonstrated compliance with all design requirements. The design margins demonstrated by the testing indicate the capabilities of the valve design are considerably greater than those initially anticipated. Development tests successfully demonstrated the valve's capability to perform in excess of 100,000 cycles of operation while maintaining a leak rate of less than 1.0 SCCH of helium. This cycle life capability was achieved over a temperature range of -40 to +300°F and at pressures from 0 to 400 psi. One valve accumulated 509,132 cycles of operation controlling hydrazine (N_2H_4) flow and exhibited 0.0 SCCH helium leakage at the conclusion of this testing. Dynamic environment testing was successfully completed, with no indication of valve failure, demonstrating compatibility with 11 ms saw tooth shock pulses to 30 g's, accelerations of 8 g's for 120 seconds, and random vibration spectra of up to 37 g rms wide band overall intensity. Including the results of design support testing of the valve seat configuration prior to valve fabrication, a total of 2,209,132 cycles of operation have been accumulated on the valve seat design without evidencing a failure. A summary of testing accomplished during this development effort is presented in Table I.

The wealth of performance data obtained over the total test range attests to the level of maturation achieved by the design. In view of the temperature capability demonstrated during valve testing and design support testing (to 390°F), the applicability of the valve to the biowaste resistojet (to 450°F) appears feasible without modifications. In the event this temperature (450°F) degrades valve life, alternate soft seal or a hard seat configuration can be readily incorporated without invalidating the test data accumulated during this effort.

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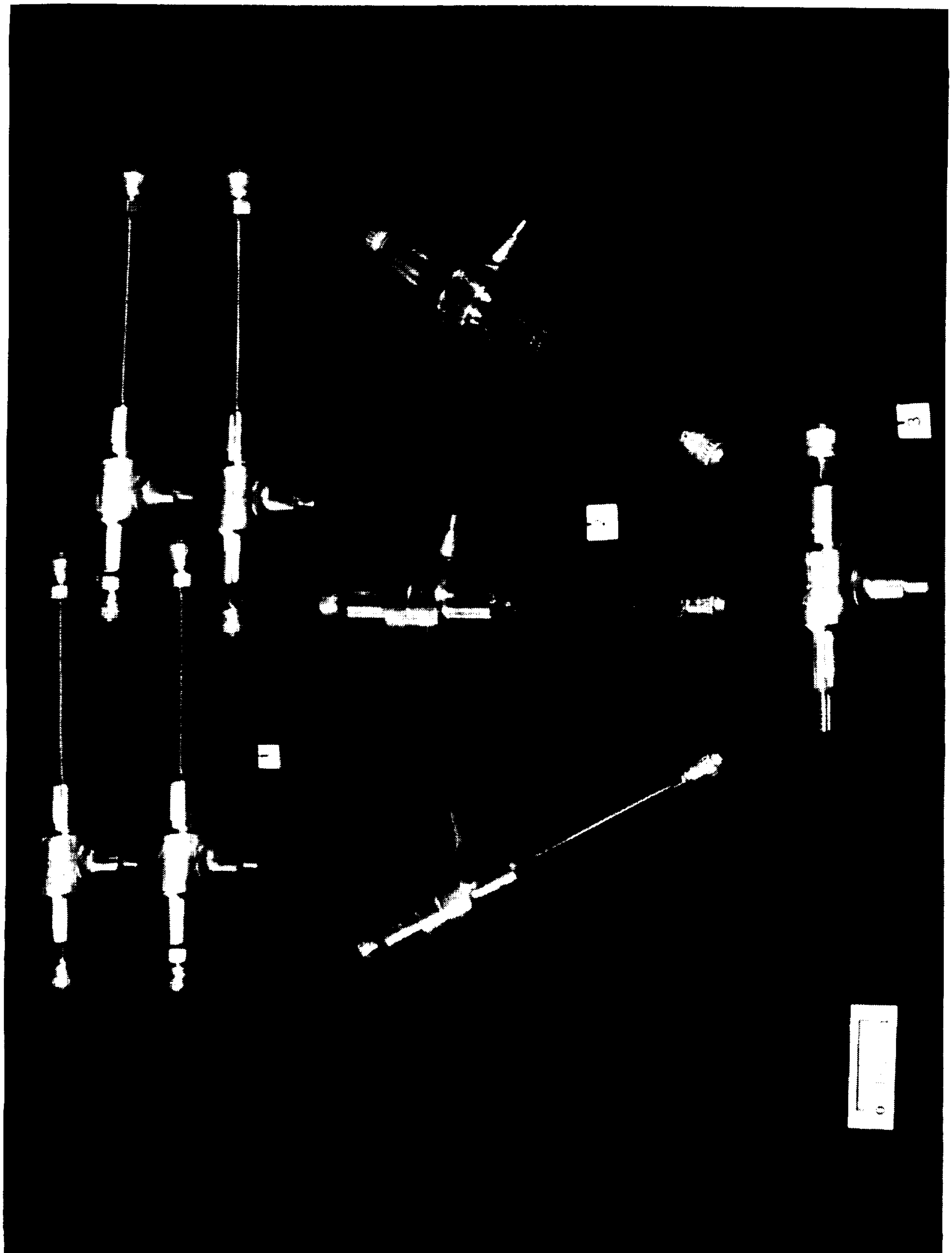
MINIATURE, BI-STABLE LATCHING SOLENOID VALVE



NEG. 71-151-3

Figure 2

TABLE I

DEVELOPMENT TEST SUMMARY

Test	Design Goal	Demonstrated Capability
Operating Pressure	0-400 psi	0-400 psi
Proof Pressure	800 psi min.	800 psi min.
External Leakage	$< 10^{-4}$ sccs He @ 60 psi	Zero bubbles 0-800 psi
Response Time	< 10 milliseconds	3.1 milliseconds max. @ 0-400 psi, 20-32 vdc, -40 to +300°F
Response Repeatability	< ±2 milliseconds	< ±0.2 milliseconds
Life	100,000 cycles min.	100,000 cycles (2 valves) 509,132 cycles (1 valve)
Pressure Drop	< 15 psi @ 0.01 pps N_2H_4	< 10 psi @ 0.01 pps H_2O
Internal Leakage	< 1.0 SCCH He 0-400 psi	< 1.0 SCCH He 0-400 psi
Vibration	1.0 g^2/Hz 40-800 Hz Decrease to 0.6 g^2/Hz @ 2000 cps (36 g rms)	.05 g^2/Hz @ 20 Hz Increase to 1 g^2/Hz @ 100 Hz 1.0 g^2/Hz 100 to 1000 Hz Decrease to .55 g^2/Hz @ 2000 Hz
Operating Temperature	-40 to +300°F	-40 to +300°F
Operating Voltage	15-32 vdc	8-32 vdc (-40 to +300°F, 0-400 psi)
Electrical Power	< 25 watts @ 30 vdc and 70°F	< 25 watts @ 30 vdc and 70°F
Shock	-	30 g 11 ms terminal peak saw tooth pulse
Acceleration	-	8 g for 120 sec/axis
Dielectric Strength	100 micro amps max. @ 600 vac rms	< 100 micro amps @ 600 vac rms
Insulation Resistance	100 megohms min. @ 500 vdc	> 100 megohms @ 500 vdc

DESIGN CRITERIA

Design Requirements

Design requirements, imposed by the contract work statement, are summarized in Table II. These requirements are responsive to current and anticipated characteristics necessary to produce a reliable resistojet thruster system for future spacecraft attitude control.

Additional Design Parameters

In addition to the design requirements of Table II, several other design parameters were self imposed in order to broaden the base of applicability of the valve design. Each additional design parameter was analyzed to evaluate its impact on the anticipated design and provide assurance that the contractual design requirements would be enhanced, not compromised. The additional design parameters are defined in Table III.

TABLE II

VALVE DESIGN REQUIREMENTS FOR RESISTOJET APPLICATION
(Ref. NAS 1-9601 Work Statement)

Characteristic	Requirement
Valve Type	Magnetically latching, minimum power, no sliding surfaces
Response Time	< 10 milliseconds
Response Repeatability	$< \pm 2$ milliseconds
Life	5 years operational, 3 years shelf 100,000 cycles minimum
Fluid Service	NH_3, H_2 and capability for superheated steam, CH_4 and/or CO_2 with minimum cost modification
Pressure Drop	< 1 psi @ 7×10^{-5} pps NH_3
Vibration Environment	(Random) 1.0 g^2/Hz 40-800 Hz decreasing to 0.6 g^2/Hz at 2000 Hz (36 g rms)
Leakage (External) (Internal)	10^{-4} SCCS max He @ 60 psia .005 mg/sec max. H_2 @ 50 psia
Weight and Volume	Minimum consistent with cost and performance
Growth Capability	Capable of controlling propellant to 10-50 millipound resistojet thrusters
Operating Temperature	+300°F to -40°F
Operating Voltage	15-32 vdc
Operating Pressure	0-60 psia

TABLE III

ADDITIONAL DESIGN PARAMETERS

Characteristic	Requirement
Operating Pressure	0-400 psia
Proof Pressure	800 psi min.
Burst Pressure	1600 psi min.
Operating Fluid	N_2H_4, H_2O, GN_2, isopropyl alcohol
Pressure Drop	< 15 psi @ 0.01 pps N_2H_4
Leakage (Internal)	< 1 SCCH He 0-400 psia
Electrical Power	< 25 watts @ 30 vdc 70°F
Dielectric Strength	100 microamps max. @ 600 vac rms
Insulation Resistance	100 megohms min. @ 500 vdc
Inlet Filter	25 micron absolute
General Packaging	Coaxial configuration

VALVE DESIGN AND ANALYSIS

Valve Sizing for Design Flow Rates

To establish the required flow passage cross sectional area, the valve is equated to an orifice and the orifice flow factor (C_dA_t) determined for the required flows and pressure drops (C_d = orifice coefficient, A_t = minimum flow cross sectional area). For the nominal design point of 7×10^{-5} pps of ammonia with less than 1 psi pressure drop, the flow factor (C_dA_t) is determined for the conditions of 40 psia inlet pressure and 200°F fluid temperature:

$$(C_dA_t)_{NH_3} = \frac{\dot{w}\sqrt{RT}}{8.02 \times P1\sqrt{\frac{K}{K-1}\left[\left(\frac{P2}{P1}\right)^{2/K} - \left(\frac{P2}{P1}\right)^{\frac{K+1}{K}}\right]}} = 31.15 \times 10^{-5} in^2$$

If an orifice coefficient of .65 is assumed, a minimum flow area of 81×10^{-5} in^2 is required, or a flow passage of .040 diameter.

For a design flow factor of 81×10^{-5} in^2 the "growth" potential of the valve can be determined by calculating the flow, hence thrust, that can be attained at various pressure drops for various propellants when a specific impulse is assumed for the respective propellants. This growth potential is stated in the following tabulation:

Propellant	NH_3	H_2	CO_2	H_2O	CH_4	N_2H_4
Assumed Isp (Seconds)	350	700	160	230	232	200
	Thrust Attainable at ΔP noted and assumed Isp - Lbs					
ΔP = 1 psi	.064	.045	.047	.052	.040	.51
= 2 psi	.082	.058	.060	.073	.052	.73
= 3 psi	.100	.071	.073	.088	.063	.89
= 5 psi	.128	.091	.094	.111	.081	1.15
= 10 psi	.189	.136	.140	.146	.121	1.63
= 15 psi	.216	.151	.157	.164	.135	2.00

Assuming a flow factor of 81×10^{-5} in^2 and a C_d = .65, which is characteristic of coaxial flow solenoid, the minimum flow passage cross sectional area is assumed to occur at the valve seat.

A plot of seat diameter (D_S) and poppet stroke (S) required to achieve a flow passage cross section area (A_t) of .001255 in^2, is presented in Figure 3. Since an unbalanced valve design is anticipated, the seat diameter (D_S) will establish actuator force requirements. With the valve closed, a nominal preload is required to assure a leak-tight seal. As inlet pressure is increased, closing forces increase due to the differential pressure acting over the valve seat area. The sum of these forces must be overcome by the solenoid actuator to effect valve opening.

For various inlet pressures (P_{inlet}) and seat diameters (D_S), Figure 4 presents a plot of total closing force (F_T) as a function of these variables. The characteristics of Figure 3 and 4 indicate that the smallest practical seat diameter (D_S) should be selected to minimize closing forces which must be overcome to effect valve opening. For this reason, an initial seat diameter of .050 inch is selected for purposes of design analysis. This selection requires a valve stroke of .008 inch maximum to assure the required flow passage cross sectional area. This also allows a practical stroke tolerance while minimizing valve-to-valve flow characteristic variations.

Poppet/Seat Interface

In the previous analysis to select a nominal seat diameter, a flat poppet/seat interface was assumed. In many valve seat designs, a conical interface is employed to effect a wedging action between the poppet and seat in an effort to achieve a better sealing action in the "valve closed" position. The conical configuration results in much higher loading at the sealing interface, but reduces the effective flow area per unit of valve stroke. For conical half angles from 30 to 90°, Figure 5 defines the flow area efficiency based upon a 90° cone half angle interface (flat seat) having the maximum attainable flow area per unit stroke for any constant seat diameter. Aside from reducing flow area efficiency as the cone half angle is decreased, conical interfaces in small valves require precise machining which must be performed with limited access to critical areas and are difficult to inspect.

For reasons of maximum flow area efficiency, ease of manufacture and inspection, and previous Marquardt development, a flat seat configuration is selected for this application.

Two candidate flat seat concepts were analytically evaluated and testing performed to substantiate the design parameters. These concepts, shown in Figure 6, were identified as "constrained" and "spring-loaded" to characterize the method of retention

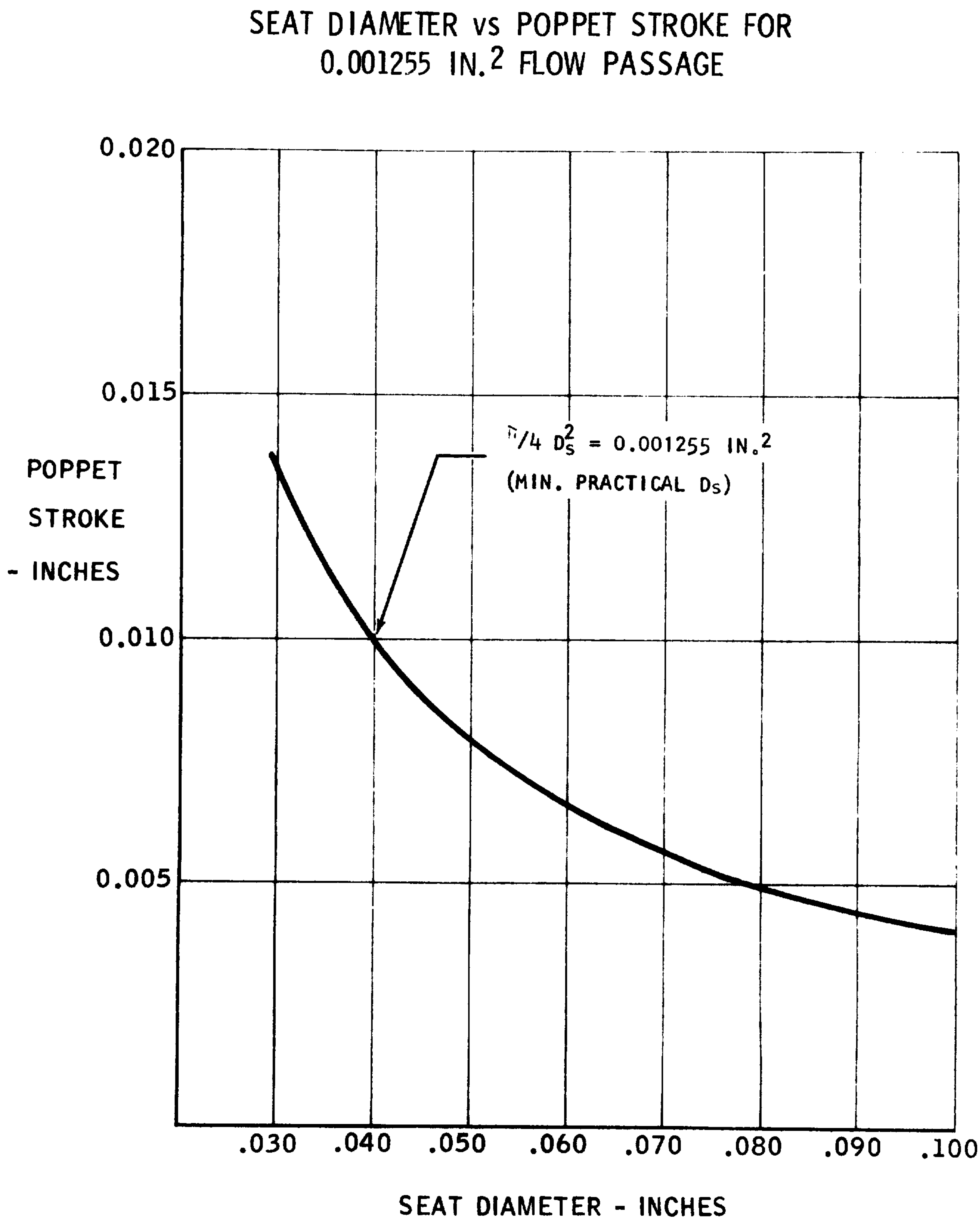

SEAT DIAMETER vs POPPET STROKE FOR
0.001255 IN.2 FLOW PASSAGE
0.020
0.015
0.010
0.005
POPPET
STROKE
- INCHES
π/4 Ds2 = 0.001255 IN.2
(MIN. PRACTICAL Ds)
.030
.040
.050
.060
.070
.080
.090
.100
SEAT DIAMETER - INCHES

Figure 3

CLOSING FORCE vs SEAT DIAMETER AT VARIOUS INLET PRESSURES

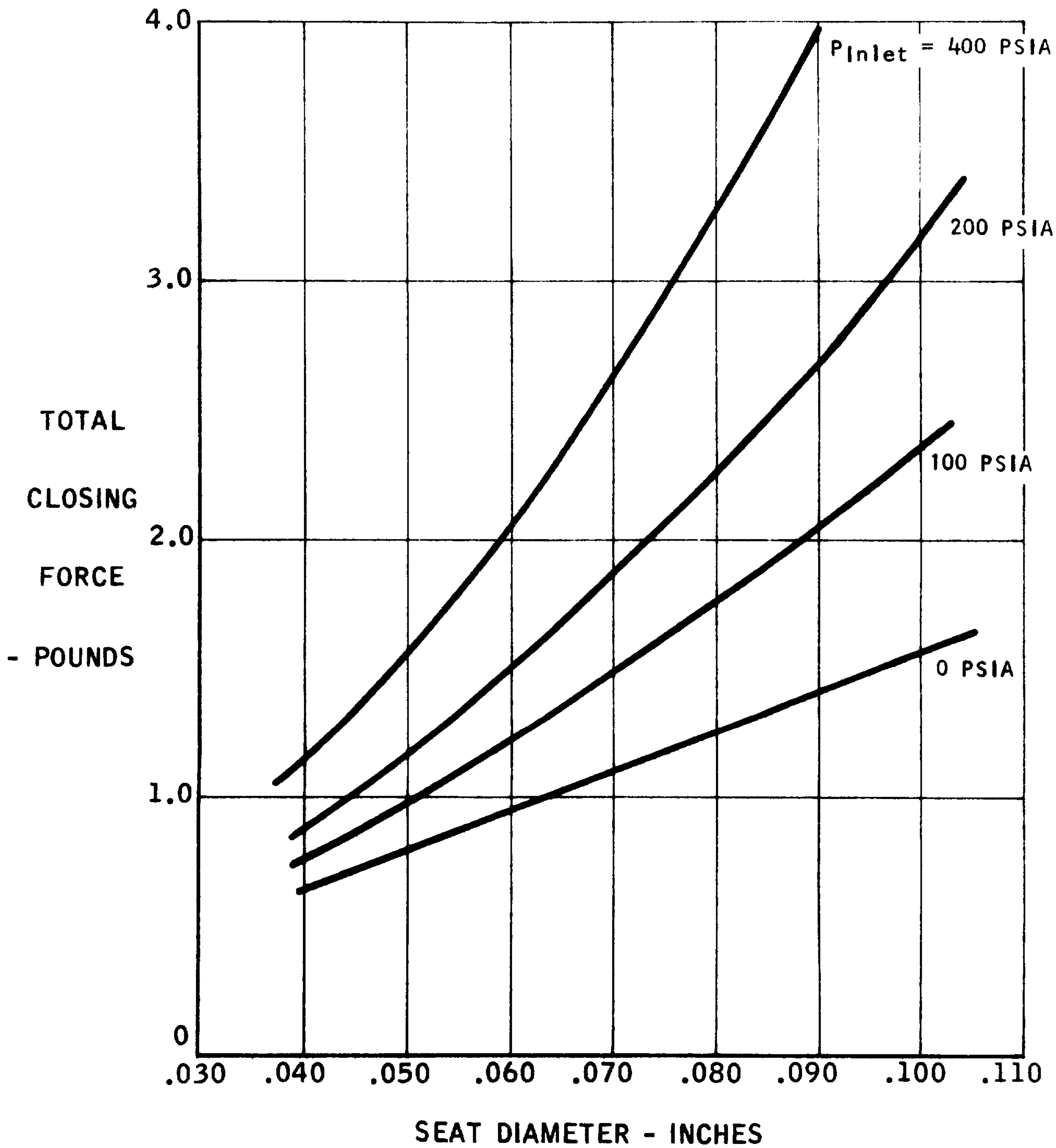


Figure 4

FLOW AREA EFFICIENCY vs CONE HALF-ANGLE

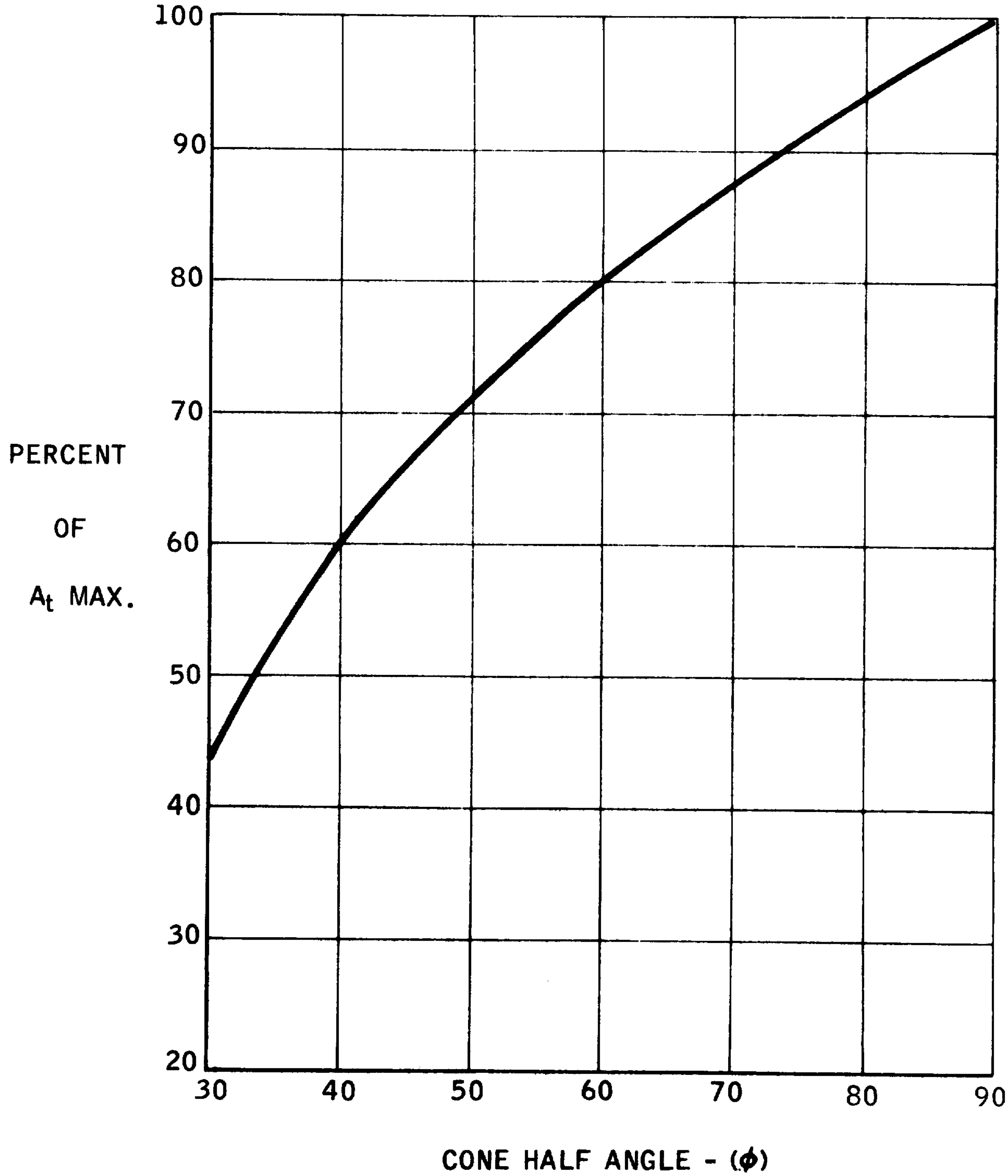


Figure 5

POPPET / SEAT CONCEPTS

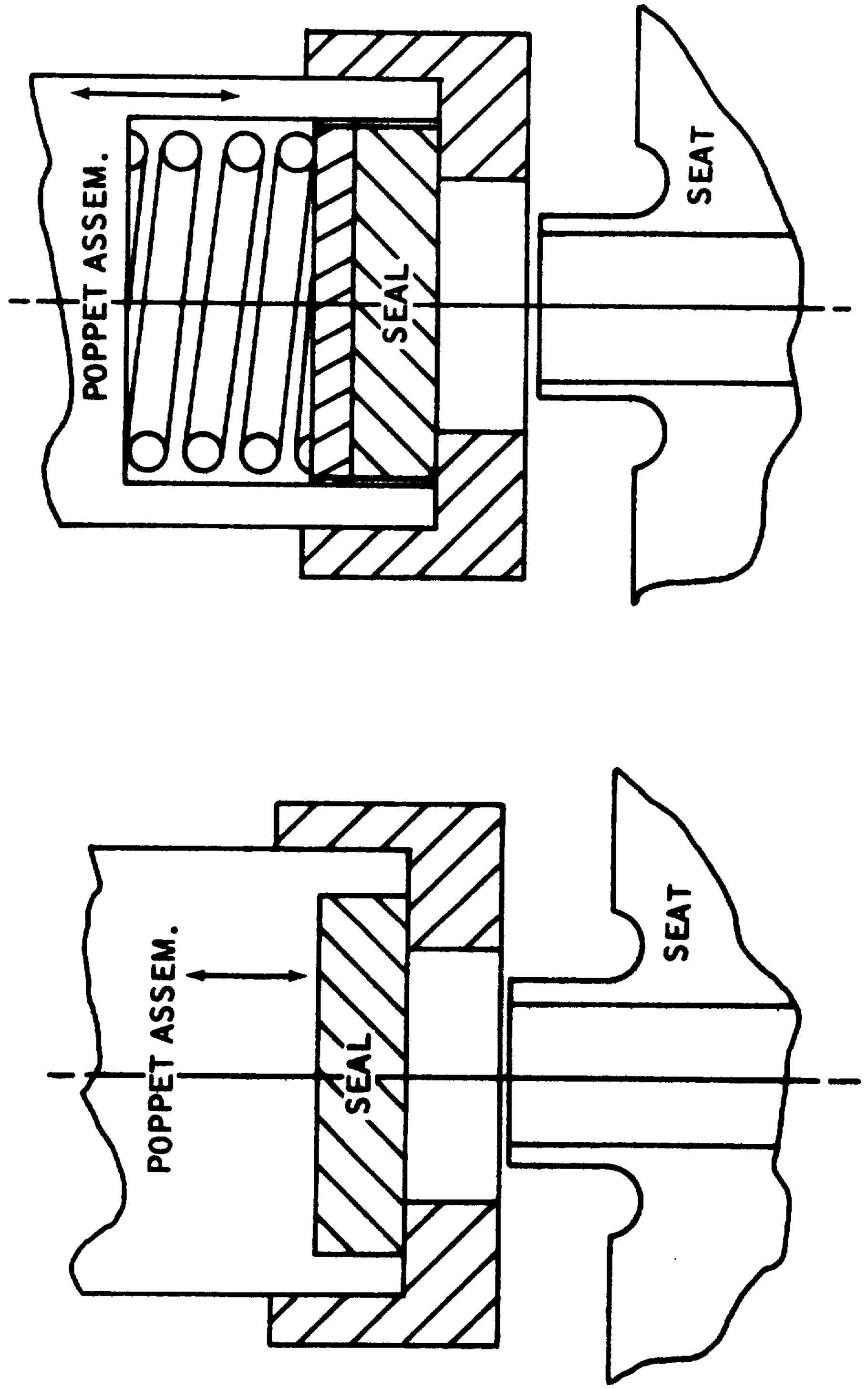


Figure 6

of the soft seal element. Several other considerations which influenced the general design of the concepts were:

- Ease of interchanging seal materials
- Adaptability to a hard seat interface
- Ease of controlling critical sealing parameters
- Ease of manufacture and inspection

From prior analysis the seat diameter (D_s) of .050 inch is taken as the inside diameter of the seat land. The width of the seat land will establish the maximum sealing diameter, hence pressure force, and the bearing load when the valve is closed. For a soft seat interface, operating in the design temperature range of -40 to +300°F, several candidate materials which are compatible with all anticipated usage fluids are presented in Table IV, along with published properties which are of interest for analyzing sealing characteristics. Potential hard seat materials are also presented.

A seat bearing stress of at least 500 psi is required to obtain a reliable seal under the operating conditions if mating surfaces are of good quality surface finish.

Figure 7 plots seat bearing stress as a function of seat land O.D. and inlet pressure. A nominal seat land O.D. of .0625 inches was selected and the preload reduced to minimize the bearing stress. The final criteria, including tolerances, for the seat/seal interface was established as:

$$\text{Seat land I.D.} = \begin{matrix} .048 \\ .050 \end{matrix} \ \text{inches}$$

$$\text{Seat land O.D.} = \begin{matrix} .0635 \\ .0615 \end{matrix} \ \text{inches}$$

$$\text{Seal Preload} = .70 \pm .05 \ \text{Lb}$$

The range of seat bearing stress, as a function of inlet pressure is plotted in Figure 8. This range of seat bearing stress is easily accomplished in the spring loaded concept by properly designing the coil spring. Using the constrained concepts requires setting the valve-closed latch force at 0.70 ± .05 Lb.

TABLE IV

CANDIDATE SEAL MATERIAL PROPERTIES

Type	Material	Compressive Strength (2% off-set) At Amb. Temp. Psi	Compressive Modulus Psi	Thermal Coefficient of Expansion in/in °F	Recommended Maximum Service Temp. °F
Soft	TFE	1200	$.8 \times 10^5$	5.5×10^{-5}	550
	Kel-F	2000	1.8×10^5	3.88×10^{-5}	380
	AF-E-102 Ethylene Propylene Terpolymer	≈1000 psi	$.05 \times 10^5$	$\approx 5.5 \times 10^{-5}$	375 min.
	25% Glass Filled TFE	1870	1.18×10^5	7×10^{-5}	500
	AF-E-124D (Proprietary Compound)	≈1400	$.008 \times 10^5$	$\approx 5.5 \times 10^{-5}$	375 min.
Hard	Pyromet x 15	90,000	30×10^6	$.82 \times 10^5$	1000
	Carbides	20,000	17×10^6	$.24 \times 10^5$	1000

SEAT BEARING STRESS AT VARIOUS INLET PRESSURES
vs SEAT LAND O.D.

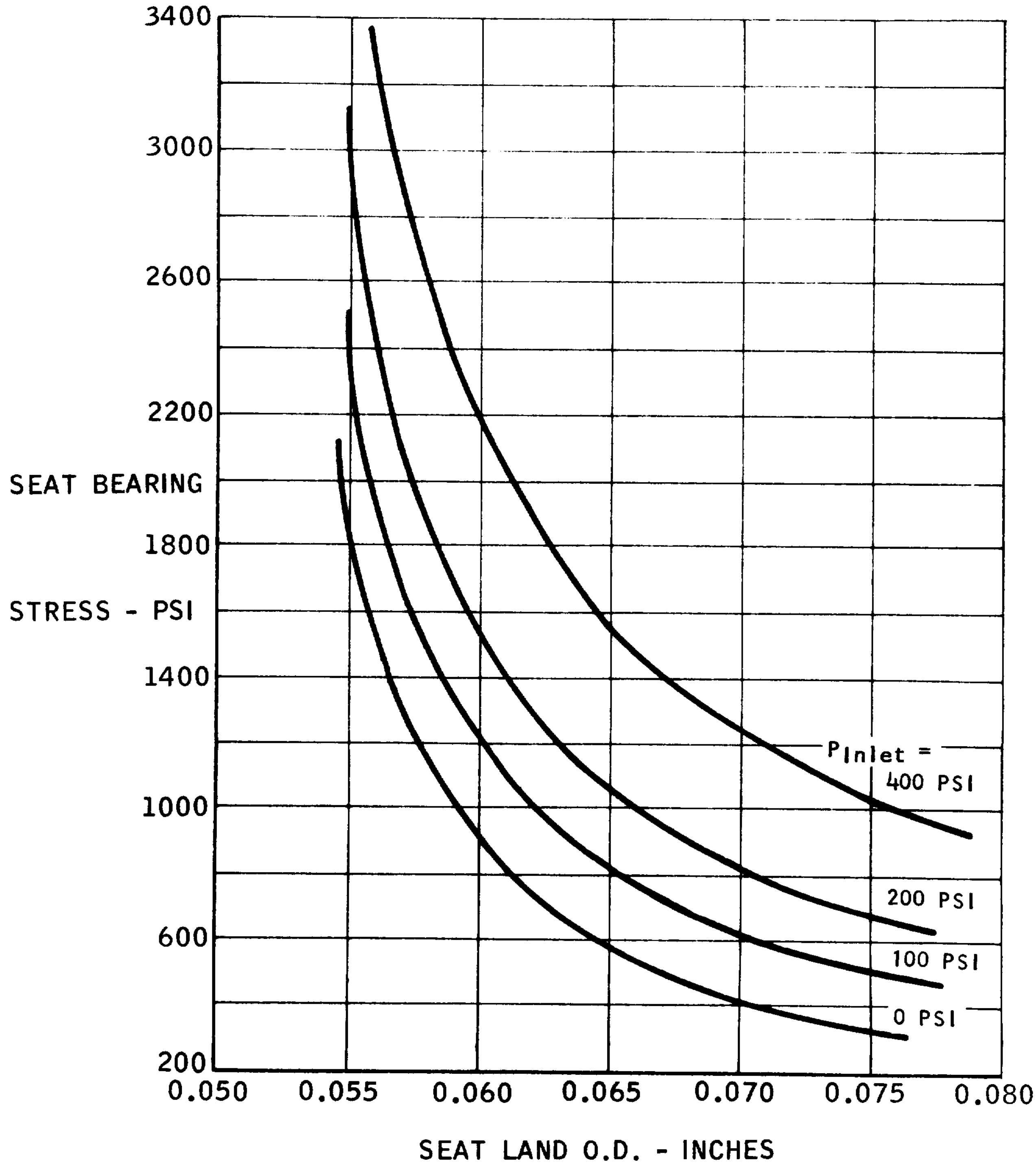


Figure 7

SEAT BEARING STRESS vs INLET PRESSURE

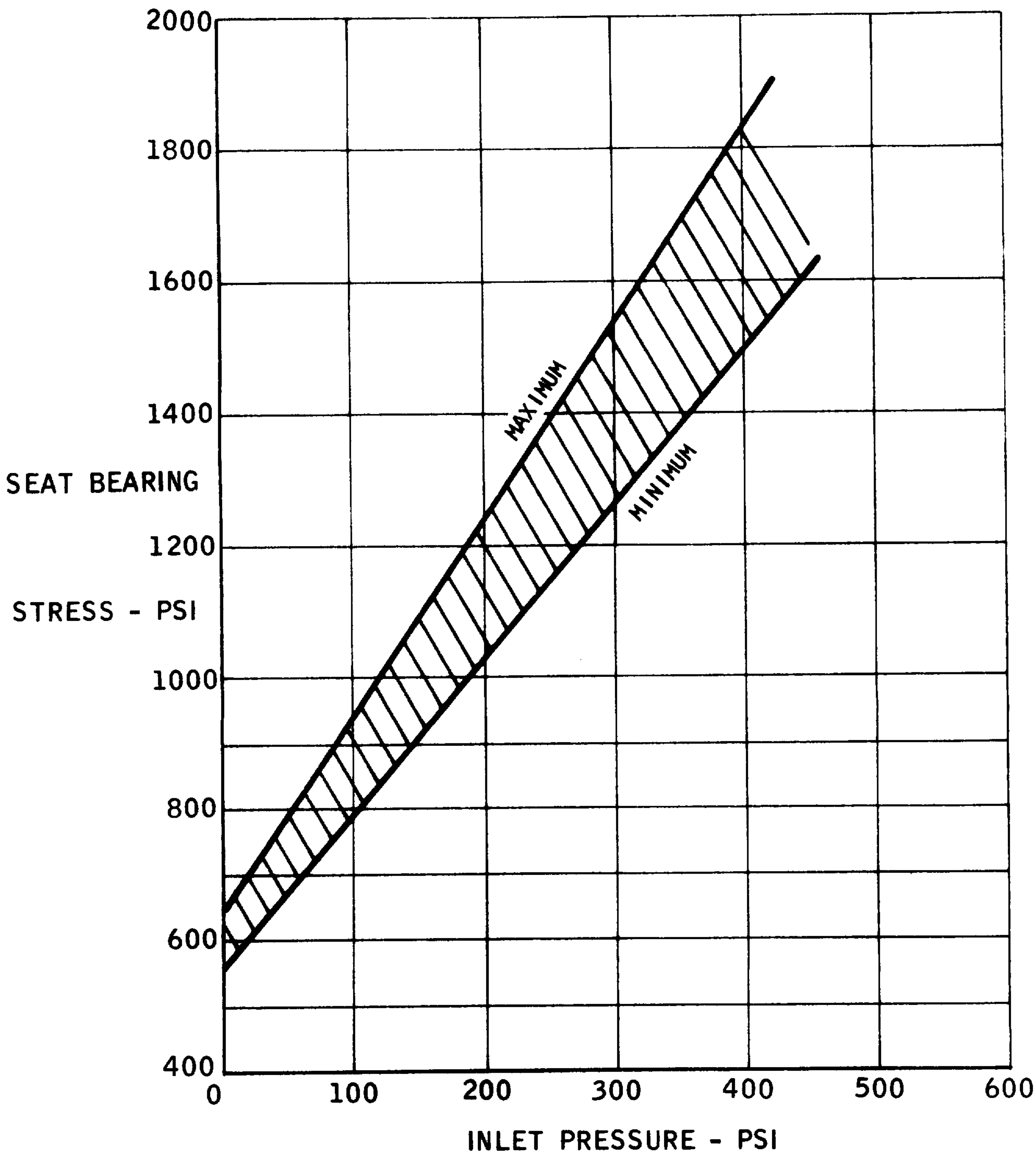


Figure 8

Using the compressive modulus of Table IV for AF-E-102, which appears to be the most promising seal material based on work reported in Reference (1), the seal material strain, resulting from a seat bearing stress of 550 to 1830 psi (Reference Figure 8) will be 11 to 36.6%. From past experience with soft seats and the data of Reference (1), this magnitude of strain will result in permanent set, particularly after high temperature exposure and/or time at this strain. The amount of "penetration" of the seat land into the seal material must therefore be dimensionally limited to a range which will assure a stable sealing interface. If penetration is controlled to .0025/.0030 inch with the valve closed, seal material strain will be limited to 3.6 to 4.3%. Resultant seat bearing stress will be 180 to 215 psi. Since the seal material is elastic, the differential pressure across the valve seat seal will generate additional seat bearing stress and a reliable seal will be effected if good quality surfaces are maintained at the interface.

Assuming a seal of .200 diameter and .075 thickness, required diametral clearance and axial growth can be determined to assure minimum seal constraint over the total usage temperature range. Using the thermal coefficient of expansion of Table IV, seal axial contraction (-) and expansion (+) based upon the .075 nominal thickness will be +.00095/-.00045 inch over the temperature range of -40 to + 300°F. Diametral growth, or required clearance to assure no diametral constraint, is +.00254/-.00120 inch.

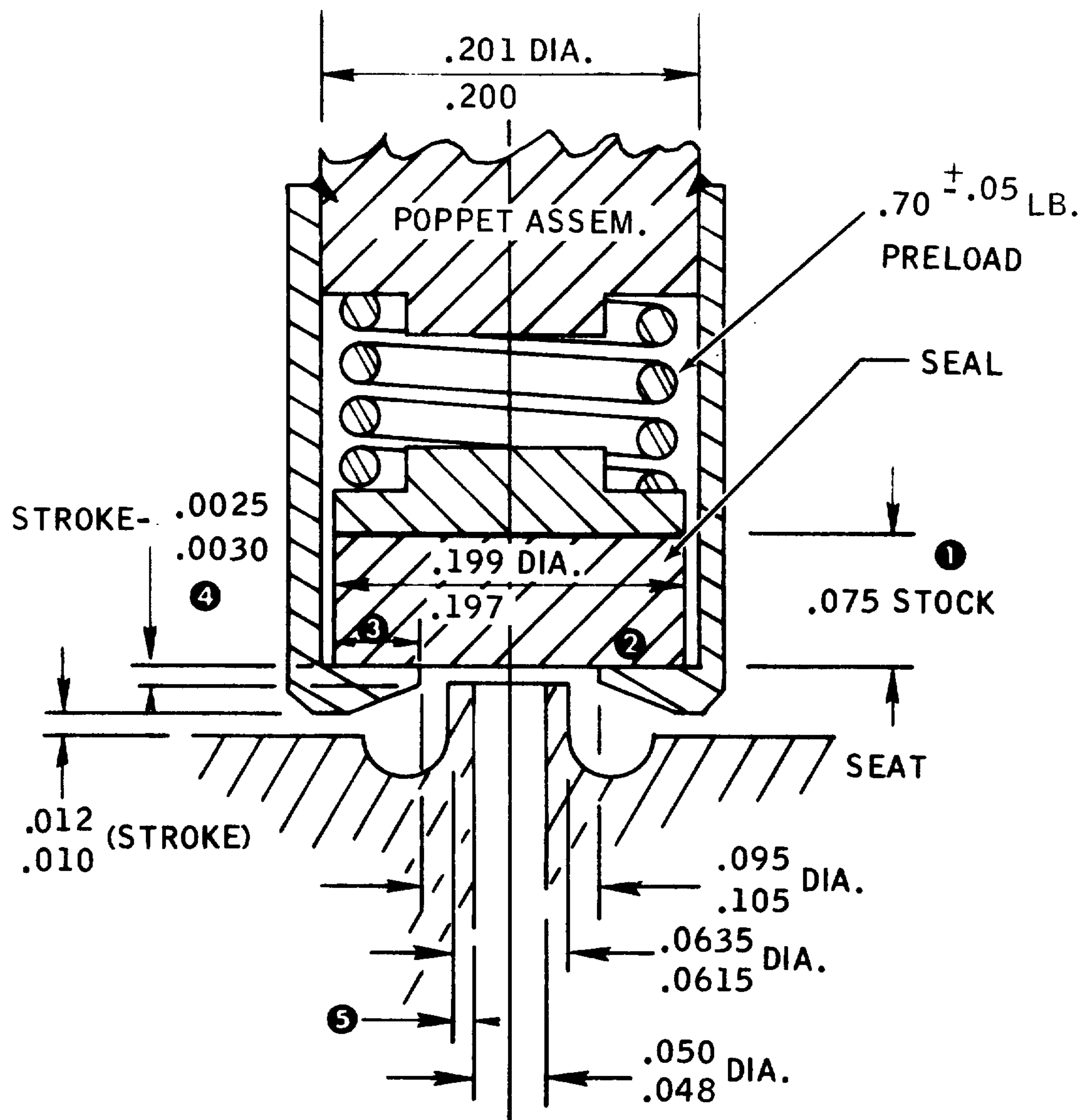
Figure 9 defines the final analytical seat/seal design criteria. Annotation is made to document calculated parameters of the design for the AF-E-102 seal material. Similar analyses for alternate seal materials, using the physical properties of Table IV were made, resulting in stress levels which would assure a long cycle life capability and thermal stability.

Design support testing was conducted to evaluate the seat design parameters and materials. This testing is described in detail later in this report. A total of 15 samples were cycle tested over a range of temperatures, with periodic leak checks. As a result of this testing, the spring loaded concept using AF-E-102 seal material was selected.

Latching Actuator Design

To accomplish valve operation and maintain the valve in the "last-commanded" position without requiring electrical power to the valve, several concepts of latching solenoid actuators were considered. Table V presents a summary of the various types considered and the attributes and characteristics of each type. On the basis of an

SEAT / SEAL CRITERIA - SPRING LOADED CONCEPT



❶ THERMAL EXPANSION (-40 TO + 300°F) = $^{+.00095}_{-.00045}$

❷ THERMAL EXPANSION (-40 TO +300°F) = $^{+.00254}_{-.00120}$

❸ BEARING STRESS (VALVE OPEN) = 27.1 TO 34.4 PSI

❹ COMPRESSIVE STRESS REQ'D TO DEFORM SEAL (VALVE CLOSED) = 167 TO 200 PSI

❺ BEARING STRESS (VALVE CLOSED) = (167 TO 200)+ P_{INLET}(2.34 TO 2.94)

Figure 9

TABLE V

LATCHING SOLENOID ACTUATOR CHARACTERISTICS

	TYPE	CHARACTERISTICS
I.	Magnetic Latch open Spring load closed	• Magnetic latch must overcome spring load • To close, must "buck" latching magnet • Large gap in valve closed position (high residual magnetic field potential)
II.	Magnetic Latch closed Spring loaded open	• Magnetic latch must overcome spring load • To open, must "buck" latching magnet • Fails open if magnet degrades • Large gap in valve open position (high residual magnetic field potential)
III.	Magnetic Latch open and closed	
	a. Permanent magnet 1. Stationary perm. Magnet (TMC "bi-stable")	• Permanent magnet residual flux acts thru min. gap - min. radiated flux • Energizing coils restrengthen permanent magnet effect motion • Permanent magnet isolated from fluid
	2. Moving permanent Magnet	• Same as III a-1, except permanent magnet may be exposed to fluid
	b. Residual flux latching	• Generally requires large cross section "iron circuit" to achieve latch forces at low residual flux density • High retentivity material desirable
IV.	Mechanical detent latch open and closed	• Latching mechanism degrades valve reliability • Requires spring to apply closing preload • Low residual magnetic field potential
V.	Mechanical detent open Spring loaded closed	• Same as IV except slight improvement in reliability due to single detent latching
VI.	Overcenter latching (Belleville spring)	• Maximum magnetic force required to translate through over-center position • "Chewing" by Belleville at I.D. and O.D. • Critical fabrication problems associated with Belleville spring production

evaluation of these characteristics and the requirements of the application the "Bi-stable Actuator", type (IIIa), was selected as optimum. Additional features, supporting this selection are:

- Extensive development of the actuator concept by TMC
- Minimum actuator size
- "Positive" latching in both positions
- "Maximum" response characteristics

From the analyses , the actuator stroke must effect a minimum valve open stroke, across the seal land of .008 inch. An additional .0025/.0030 inch of stroke is required to accomplish the deformation of the seat land into the valve seat, in the closed position. Actuator design stroke is, therefore, specified at .011/.012 inch.

In the valve closed position, sufficient latch force is required to apply the required seat preload and to constrain the moving mass of the valve under acceleration. To compress the seat land into the seal 0.003 inch, a load of .27 lb is required. Assuming a moving mass of .03 lb and an effective maximum acceleration field of $\pm$ 40 g's, a latch force of 1.2 + .27 = 1.47 lbs is required. To effect valve opening, the actuator must exert sufficient force to overcome pressure unbalance forces and acceleration force (1.2 lbs at 40 g). From the analysis, maximum pressure unbalance force at 400 psi inlet pressure is 1.26 lbs or a total force of 1.20 + 1.26 = 2.46 lbs is required to effect valve opening. Assuming the moving element guidance mechanism exerts 0.5 lb tending to unlatch the actuator, the actuator design point is defined as:

Stroke = .011 inch
Latch force = 1.47 + .5 = 2.0 lbs
Pull force at full stroke = 2.46 - .5 $\approx$ 2.0 lbs

From the data of Reference (3), the latch force to pull force ratio of 1.0 represents the optimum for achieving the fastest response.

Since a wetted solenoid design is anticipated type 446 cres is selected for the magnetic material on the basis of its magnetic characteristics, high resistance to corrosive attack by any of the anticipated usage fluids and prior usage by TMC in solenoid valve applications.

To determine optimum actuator size, Figure 10 plots air gap flux density as a function armature diameter required to produce 2.0 lbs force, and as a function of magnetomotive force (NI) required to drive a magnetic circuit of 1.5 inches iron length, .015 air gap length and iron and air gap cross sectional areas equal to the armature face area. An assumed operating point of 5.5 kilogauss is selected to: achieve weight economy, operate below "knee" of core material (446 cres) B-H curve, and require minimum magnetomotive force, hence minimum power and best response.

At the selected operating point, the radially polarized permanent magnet must be sized to produce the required flux density throughout the "closed gap circuit.". Alnico V cast material is selected for the permanent magnet material on the basis of availability, low cost, high energy product, high retentivity and resistance to thermal degradation. Based upon operation at the peak energy product, permanent magnet dimensions necessary to produce the required latching forces are:

Radial cross sectional area - .058 in^2

Radial length - .121 in.

Assuming a worst case operating condition of 15 vdc supply voltage and 350°F operating temperature, a copper coil is sized to provide the required magnetomotive force to create a flux density of 5.5 kilogauss at the open air gap. From Figure 10, the coil must provide 210 ampere turns at these conditions.

For a coaxial valve configuration with an armature diameter of .400 inch, a pressure vessel wall thickness of .030 inch is assumed and the coil window I.D. is .460. Assuming a coil window O.D. of .850 inch, a mean turn diameter can be calculated. Combining these relationships, the resistance (OHMS/FT) of the coil may be calculated. Referring to standard gage copper magnet wire tables, #34 AWG has a resistance coefficient of 0.261 OHMS/FT and has a diameter, with double insulation, of .0075 inch. For the assumed coil window, 26 layers of $34 AWG wire can be wound. The length of the coil window can now be determined for a coil that will not draw power in excess of 25 watts at 30 vdc and 70°F (Reference Table III). Since the coil window width permits 26 layers of #34 SWG wire, 31 turns must be in each layer to provide the required coil resistance and number of turns. The coil window length is thus .232 inch.

Preliminary sizing data for the latching actuator is shown in Figure 11. By the method of article 44 of Reference 5, a detailed analysis of the actuator design was made to confirm performance characteristics. Actuator latching force is plotted as a function of permanent magnet magnetomotive force (NI), on the basis of the dimensional relationships

2.0 LB. FORCE SOLENOID ACTUATOR DESIGN CURVE

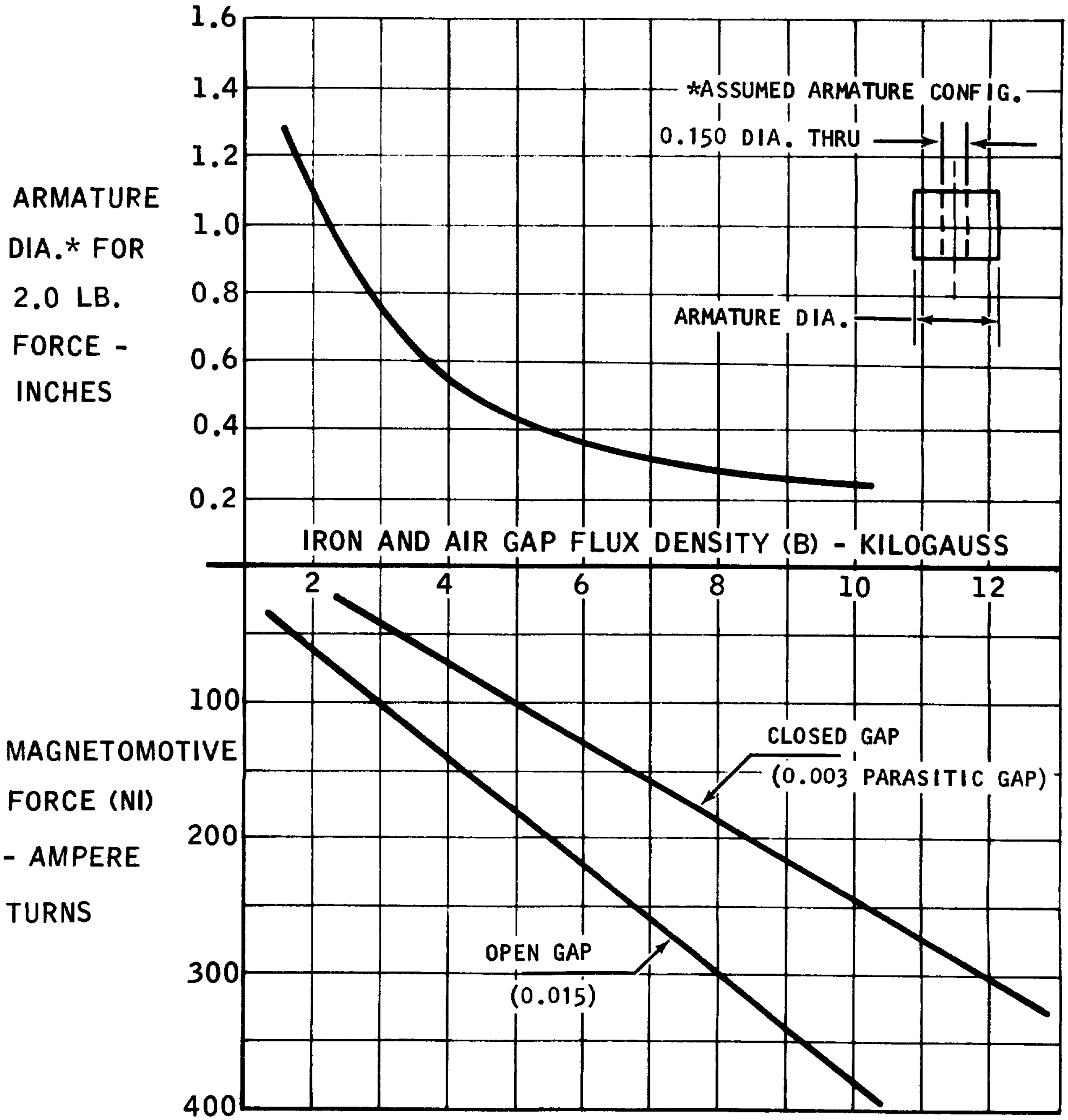


Figure 10

LATCHING ACTUATOR
PRELIMINARY DESIGN CRITERIA

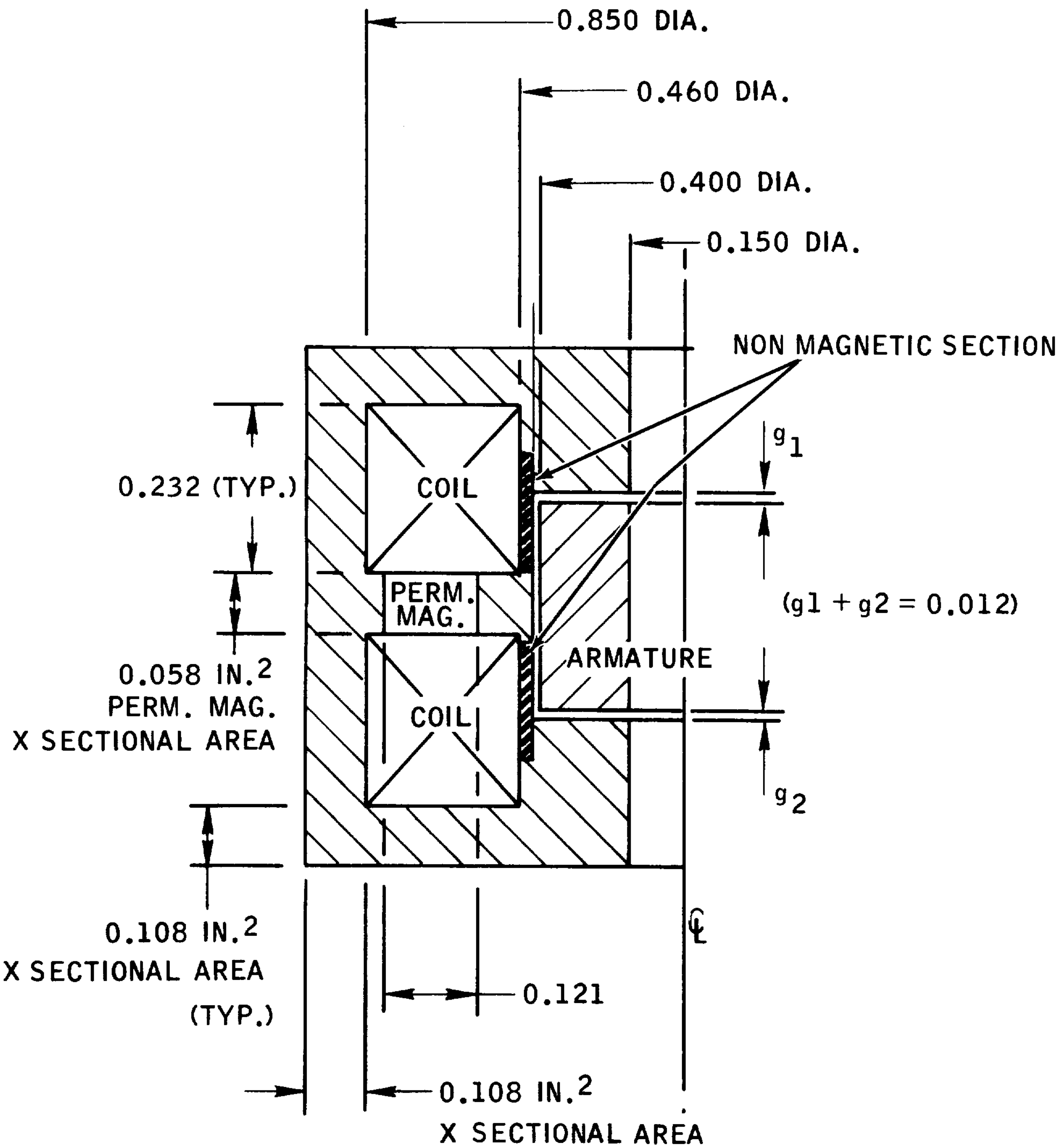


Figure 11

assumed for the design, in Figure 12. Force characteristics at the open gap, as a function of the open gap coil magnetomotive force are plotted in Figure 13. The predicted 210 ampere turn operating point (2.0 lb net force) is a conservative estimate in that the analytical technique does not consider a phenomenon of the bi-stable actuator concept wherein the flux, generated by the permanent magnet, is essentially shifted from the closed gap to the open gap once the energized coil has generated sufficient magnetomotive force to negate the latching force. Also, adequate margins have been provided in the analysis by assuming .001 inch parasitic gaps at each press fit joint of the actuator iron circuit, and all areas and lengths are taken for worst case tolerance conditions with respect to the magnetic performance.

Moving Element Guidance

To achieve the design requirement of no sliding fits while maintaining precise alignment of the poppet for sealing and minimum clearance of the actuator armature to minimize magnetic energy losses, the guidance method must achieve the following goals, assuming a coaxial valve design configuration:

- High radial stiffness
- Low axial stiffness with predictable spring rate
- Low stress levels over total deflection
- High cycle life capability
- Compatible with all propellants

Prior valve design and development, which included extensive tradeoff studies of the various no-sliding fit, sustentacular guidance techniques employed in valves, such as Belleville washers, slotted spring washers and special wire form assemblies, resulted in the design and development, by Marquardt, of a special form flexure which offers maximum design flexibility. The TMC flexure design shown in Figure 14 is a thin metal washer, through which a precise slot pattern is cut to produce the desired physical characteristics. Three flexures are assembled with spacer washers at the I.D. and O.D., between each flexure, to produce a flexure element. Each flexure is oriented such that the web interconnecting the outer annular ring to the middle annular ring is located 120° from the similar web of the other two flexures. With the inner and outer annular rings and spacers rigidly clamped, axial movement of the inner annular ring relative to the outer annular ring will result in bending of the webs and the middle annular ring. Accurate analysis to predict stresses and characteristics is facilitated by the simplicity of form of the design. Material thickness, web width and annular ring width can be selected to produce the optimum characteristics. Radial stiffness is extremely high, when assembled into an element, since the web orientation and web width present the axis of greatest stiffness to radial deflection.

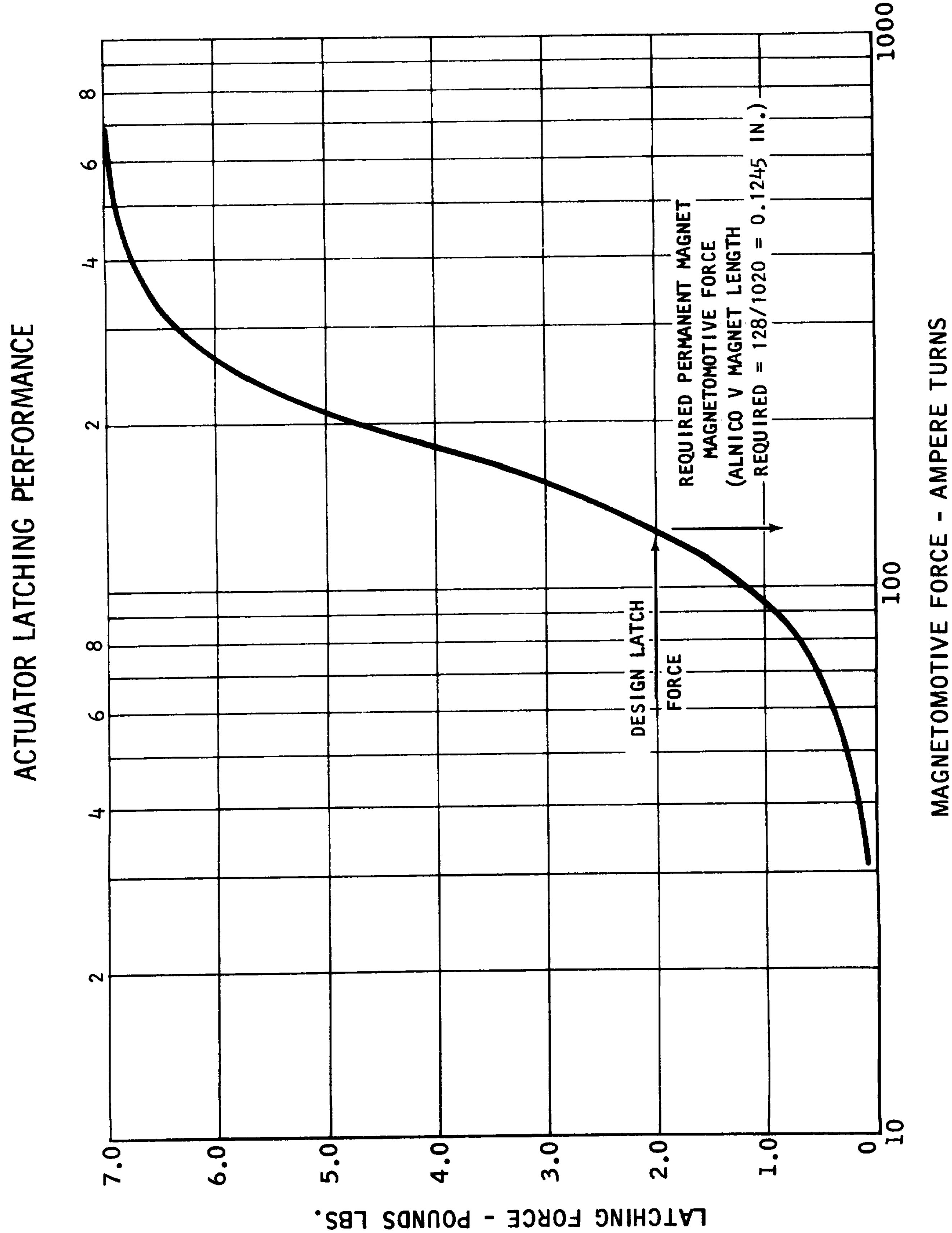

ACTUATOR LATCHING PERFORMANCE
LATCHING FORCE - POUNDS LBS.
MAGNETOMOTIVE FORCE - AMPERE TURNS
REQUIRED PERMANENT MAGNET
MAGNETOMOTIVE FORCE
(ALNICO V MAGNET LENGTH
REQUIRED = 128/1020 = 0.1245 IN.)
DESIGN LATCH
FORCE
7.0
6.0
5.0
4.0
3.0
2.0
1.0
0
10
100
1000

Figure 12

ACTUATOR ENERGIZING OPEN GAP COIL PERFORMANCE

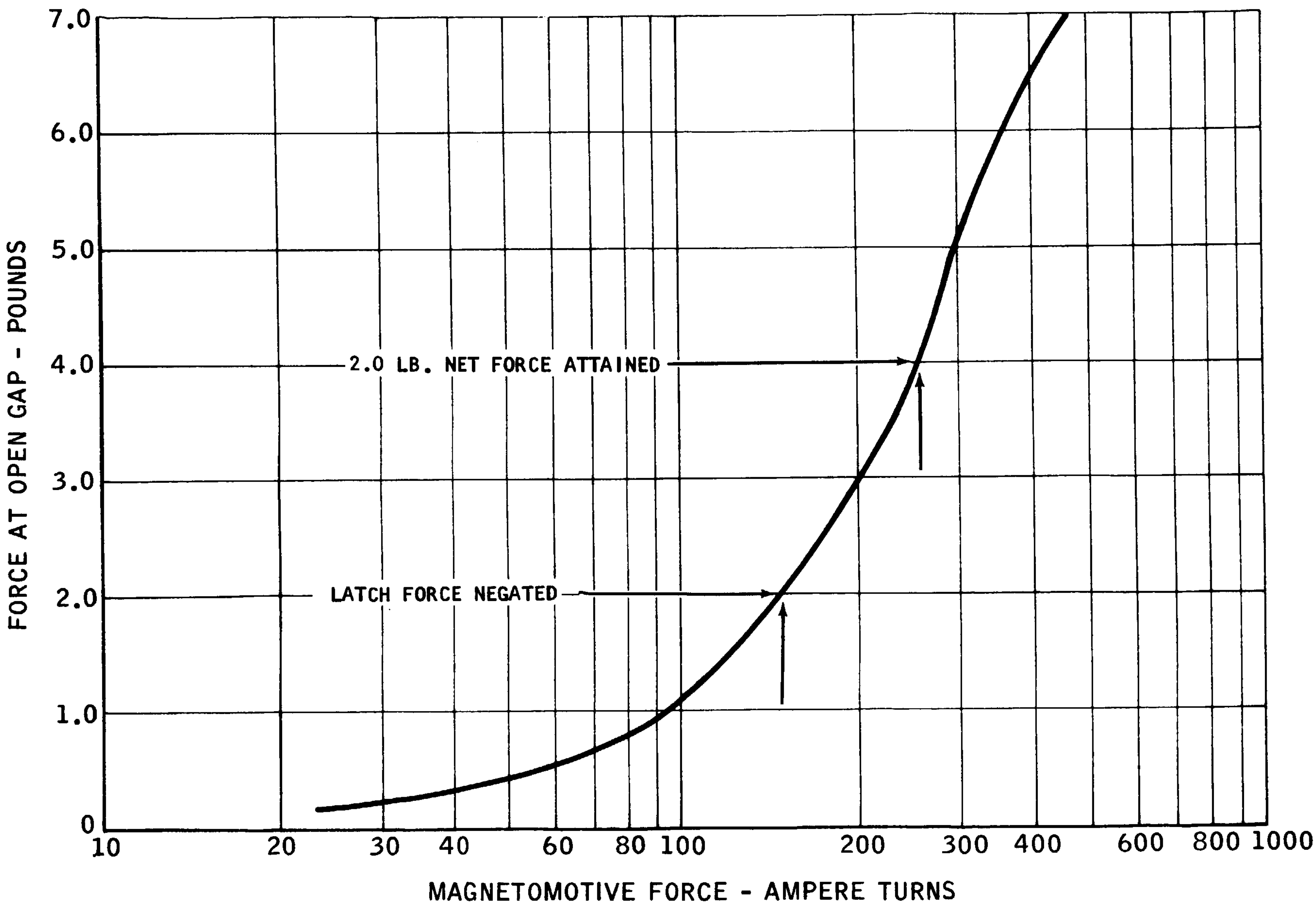


Figure 13

MOVING ELEMENT GUIDANCE FLEXURE

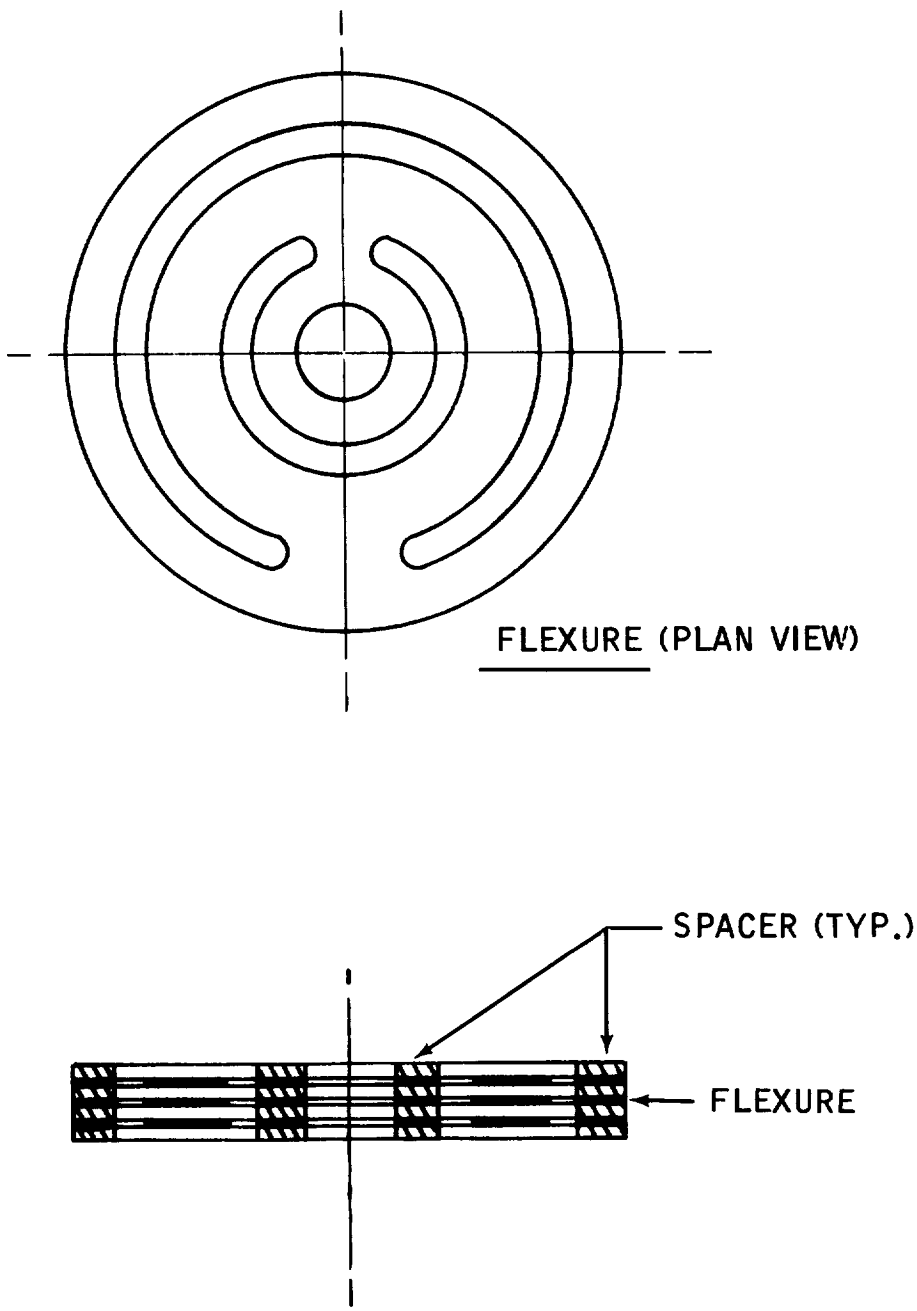


Figure 14

Stress and performance analyses were performed for several iterations of flexure design which would meet the anticipated design envelope. The selected configuration and its predicted characteristics are as follows:

O.D. = .625 inch

I.D. = .105 inch

Thickness = .005 inch 347 cres full annealed

Spring rate/flexure = 5.35 lb/in

Maximum stress at .005 axial deflection = 12,400 psi

Margin of safety based on 25,000 psi endurance limit for 10^5 cycles = 1.01

Spring rate of 2 flexure elements = 32.1 lb/in.

Materials Selection and General Configuration

Using the design criteria developed as described in the previous subsections, design layouts were made to study and resolve packaging and assembly problems. As a result of these studies, the configuration of Figure 15 was selected for detail design. This configuration allows completion of coil winding and potting before valve assembly, yet the coil and magnet can be reworked, if required, without disassembling the valve internals. Valve assembly is accomplished from both ends of the body, with all assembly welds readily accessible and provisions made for at least one disassembly and reweld of the inlet and outlet closures.

Materials of construction of valve components exposed to the operating fluids must be compatible for long term exposure. The actuator analysis was based upon the use of 446 CRES for the magnetic circuit material. Of the ferromagnetic materials available, this selection provides maximum resistance to attack by any of the operating fluids. 446 CRES parts which comprise portions of the pressure vessel valve body are machined of vacuum melted stock per MMS 2211 to assure highest quality material. Non-magnetic metallic structural elements of the valve are fabricated of type 347 CRES and Inconel 625. Type 347 CRES parts comprising the non-magnetic portions of the pressure vessel valve body are machined of consumable electrode melted stock per AMS 5654. The inlet filter is fabricated of type 347 CRES dutch twill double weave 200 x 1400 wire cloth, welded to a 347 CRES support ring. The poppet seal is AF-E-102 ethylene propylene terpolymer and the seal preload spring is wound of Inconel 600 cold drawn wire. Surfaces of the body, contacting the coils, are coated with a modified alkyd resin (Cico Corp. #C3500) to electrically isolate the coil. The Alnico V premanent magnets are held in position using Eastman 910

MINIATURE BISTABLE VALVE CROSS SECTION

LENGTH = 2.2 IN.
DIAMETER = 1.0 IN.

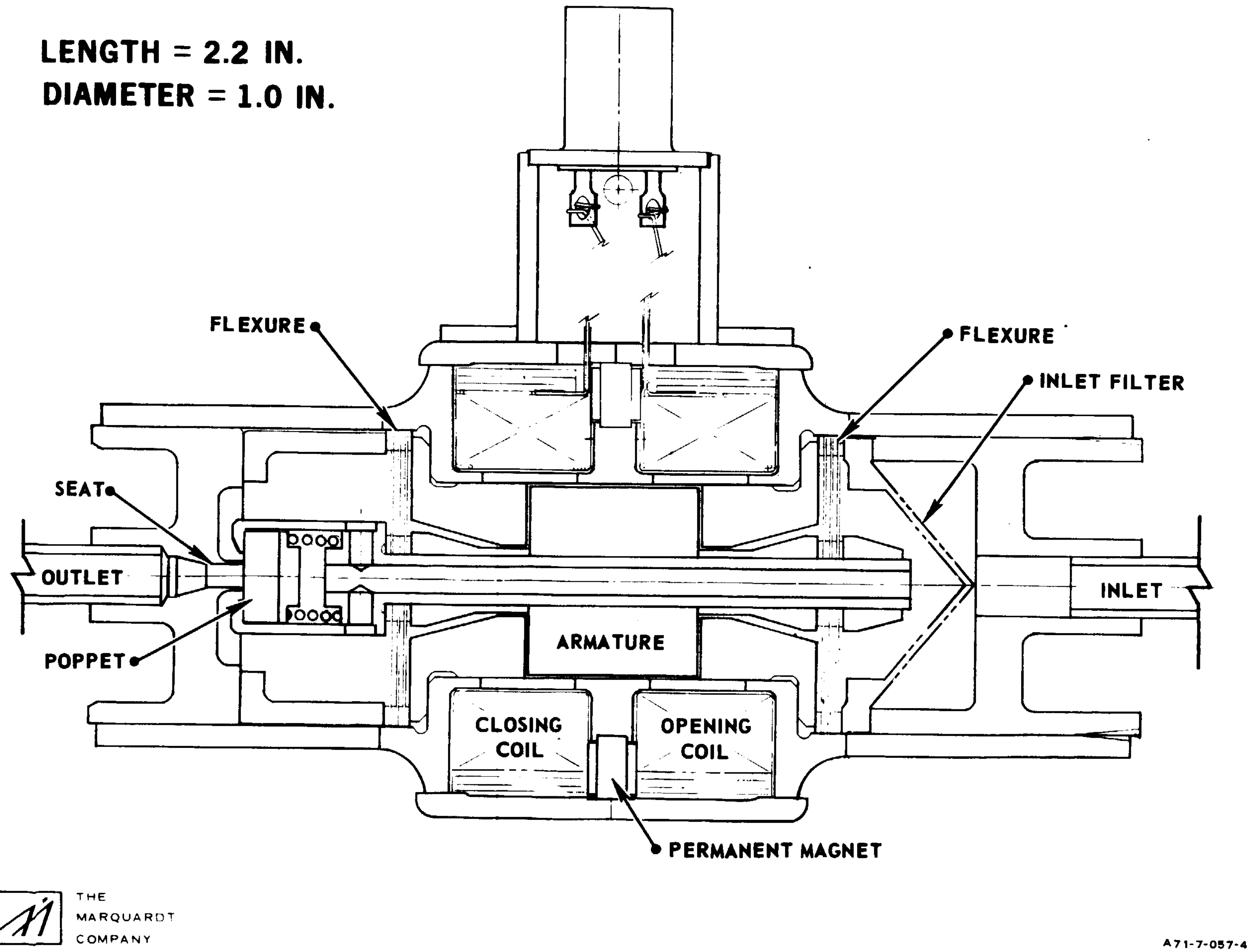


Figure 15

cement, epoxy/glass laminate split washers are bonded to the magnets with Eastman 910 cement, and the coils wound of Polythermaleze-F coated magnet wire. After installing the coil covers and connector assembly, using teflon sleeving over the coil leads and attaching the leads to the appropriate pins of the connector with solder (QQ-S-571D, S_b5 W-RA-P3), the entire coil cavity volume is vacuum impregnated with Scotchcast 250, epoxy potting resin.

All materials and processing required to produce the valve design were selected on the basis of documented compatibility with the intended usage fluids and environments, prior experience and availability.

Design Support Testing

Concurrent with the design analysis described previously, a design support test program was conducted to evaluate various seat seal candidate materials and the seat design parameters. Testing was conducted to evaluate the two poppet/seat concepts of Figure 6. The initial test vehicle was a Marquardt R-4D valve, modified such that the appropriate poppet/seat interface was achieved and various seal material samples could be interchanged. Moving element guidance in this vehicle was by a close tolerance, sliding fit of the poppet in a bore. Provisions were made for adjustment of the poppet preload, seal preload when a spring loaded concept was installed, and poppet stroke. Four test series were conducted using this test vehicle (series 1-4 of Table VI). Occasional poppet "hang-up" occurred during these tests and galling of the sliding surfaces soon became evident (no special preparation or treatment of these surfaces had been accomplished). Since this test vehicle did not simulate the anticipated valve design, a Marquardt X25475 magnetically linked bi-propellant valve was obtained and modified to serve as a test vehicle. This test vehicle would allow simultaneous evaluation of two samples and incorporates flexure guided moving elements similar to the anticipated design and provisions were made to allow poppet preload adjustment and seal preload adjustment. Prior testing with the R-4D valve test vehicle had provided evidence that the constrained seal concept was unsatisfactory for the temperature range of -40 to + 300°F, due to excessive extrusion and deformation of the soft seal material at the high temperature. A total of 10 seat seal samples were evaluation tested in the bi-propellant valve test vehicle.

Prior to the testing of each seat seal sample, the test vehicle and test parts were ultrasonically cleaned. Buildup of the test vehicle was performed in the clean room and 25 micron absolute filters were installed in all inlet lines. Testing was performed in Building 37 of the Marquardt Test Facility, using GN_2 as the test fluid for both cycling and leakage measurement. Thermal conditioning for temperatures greater than ambient was accomplished by placing the test vehicle, with inlet and outlet lines attached, into an electrically heated oven. Temperatures below ambient were achieved by submerging the test vehicle in an isopropyl alcohol bath chilled to the required temperature. Test condition temperature was monitored by a thermocouple attached to the test vehicle surface.

TABLE VI

VALVE SEAT DESIGN SUPPORT TEST SUMMARY

Test Series	Material	Configuration	Cycle/Temp. History	Maximum GN_2 Leakage @ 60 psig (Sample 1)	Maximum GN_2 Leakage @ 60 psig (Sample 2)	Observations and Remarks
1 -	Grade 7 TFE	Constrained	25,000/ambient 25,000/300°F	2.0 scch max. 1.3 scch max		Excess leakage after cool-down from 300°F. Permanent set from 300°F operation precludes seal.
2 -	Kel-F 81	Constrained	5,000/ambient 5,000/300°F	0.0 scch 0.0 scch		Began leaking at 250°F on cool-down. Leakage increased with decreasing temp. Permanent set at 300°F precludes seal.
3 -	Kel-F 81	Spring Loaded	25,000/ambient 25,000/300°F 1,000/ambient	10 scch 0.0 scch 15 scch		Imprint indicates out-of-parallel interface of seal with land. Depth of imprint not excessive.
4 -	AF-E-102 (ethylene propylene terpolymer)	Constrained	25,000/ambient 25,000/300°F 1,000/-40°F	0.0 scch 0.0 scch		Began leaking at 30°F on cool-down. Cycling did not recover seal. Excessive deformation load at 300°F.
5 -	AF-E-102 (2 Samples)	Spring Loaded	26,000/ambient 30,000/-40°F 1,000/+200°F 5,000/+250°F 20,000/+300°F 19,000/-50°F	1.0 scch 1.5 " 0.0 " 0.4 " 1.4 " 120* "	1.0 scch 1.5 " 0.0 " 0.4 " 0.0 " 1.7 "	
6 -	Grade 7 TFE	Spring Loaded	50,000/amb 25,000/+300°F 25,000/-40°F	0.5 scch 0.7 scch 5.5 scch		

* Reduced to 0.0 on warming to +10°F
Subsequent check at -50°F indicated 0.0 SCCH

TABLE VI (Cont'd)

VALVE SEAT DESIGN SUPPORT TEST SUMMARY (CONTINUED)

Test Series	Material	Configuration	Cycle/Temp. History	Maximum GN_2 Leakage @ 60 psig	Observations and Remarks
7 -	Kel F 81	Spring Loaded	50,000/ambient	0.0	
			25,000/+300°F	1.0 scch	
			25,000/-40°F	2.2 scch	Up to 65,000 total cycles. Excessive leakage occurred on all subsequent leak checks - contamination noted on seat.
8 -	Fluorogold (glass filled TFE)	Spring Loaded	25,000/ambient	0.4 scch	
			25,000/+300°F	0.0 scch	
			20,000/-40°F	43 scch	Recovered to 0.0 after warming to room temperature.
9 -	Graphite filled TFE	Spring Loaded	25,000/ambient	1.5 scch	
			25,000/+300°F	1.0 scch	
			20,000/-40°F	3.5 scch	(After cooling) became excessive after 10,000 cycles at -40°F. Recovered to 1800 scch after warming to ambient
10 -	AF-E-102	Spring Loaded	10^6/ambient and transition	1.2 scch	
			10,000/+300°F	0.0 scch	
			10,000/-70°F	1.3 scch	
			100/+340°F	0.0 scch	
			9900/-40°F	1.4 scch	
			10,000/390°F	0.0 scch	
11 -	EDC-006	Spring Loaded	50,000/ambient	1.2 scch	
			20,000/+300 to +390°F	2.0 scch	
			30,000/-70 to -50°F	Excessive	Lower temp. limit established at +20°F. Cycling at +20 to +390°F resulted in zero leakage.
12 -	AF-E-124D	Spring Loaded	950,000/ambient	.16 scch	
13 -	AF-E-102 (2 samples)	Spring Loaded	50,000/ambient	2.40 scch	
			25,000/+300°F	0.5 scch	
			25,000/-40°F	8.0 scch	

A summary of all testing accomplished is presented in Table VI and the temperature and leakage history of each test series is plotted in Figure 16. All cycling and leakage measurements were performed at an inlet pressure of 60 $\pm$ 5 psig. Leakage measurements were made by monitoring water displacement in a burette, with 0.05 cc increments, for periods of 6 to 12 minutes. Since the burette was located outside of the thermally conditioned test vehicle environment, unstable thermal gradients in the trapped volume between the valve outlet and water column can result in false leakage indications, particularly at off-ambient temperature tests. Prior to each leakage measurement, burette level was monitored, with zero psig valve inlet pressure, until a stable reading was achieved for at least three minutes. Valve leakage, at 60 psig, was then immediately measured following achievement of stablization.

After achieving the desired cycles or upon the occurrence of an excessive leakage rate, the test vehicle was disassembled to expose the seat seal sample. Photographs of the seat seal, at 12x magnification, were made to document seal condition. These photographs are shown in Figure 17.

As a result of this testing, the spring loaded seal concept, using AF-E-102 elastomer as a seal material, was selected as the prime design. As a backup material, both grade 7 TFE and AF-E-124D were acceptable alternates, though the test results indicated marginal capability of the TFE and insufficient thermal range demonstration of the AF-E-124D. Available data indicated all of the selected seal materials to be compatible with all of the intended usage fluids.

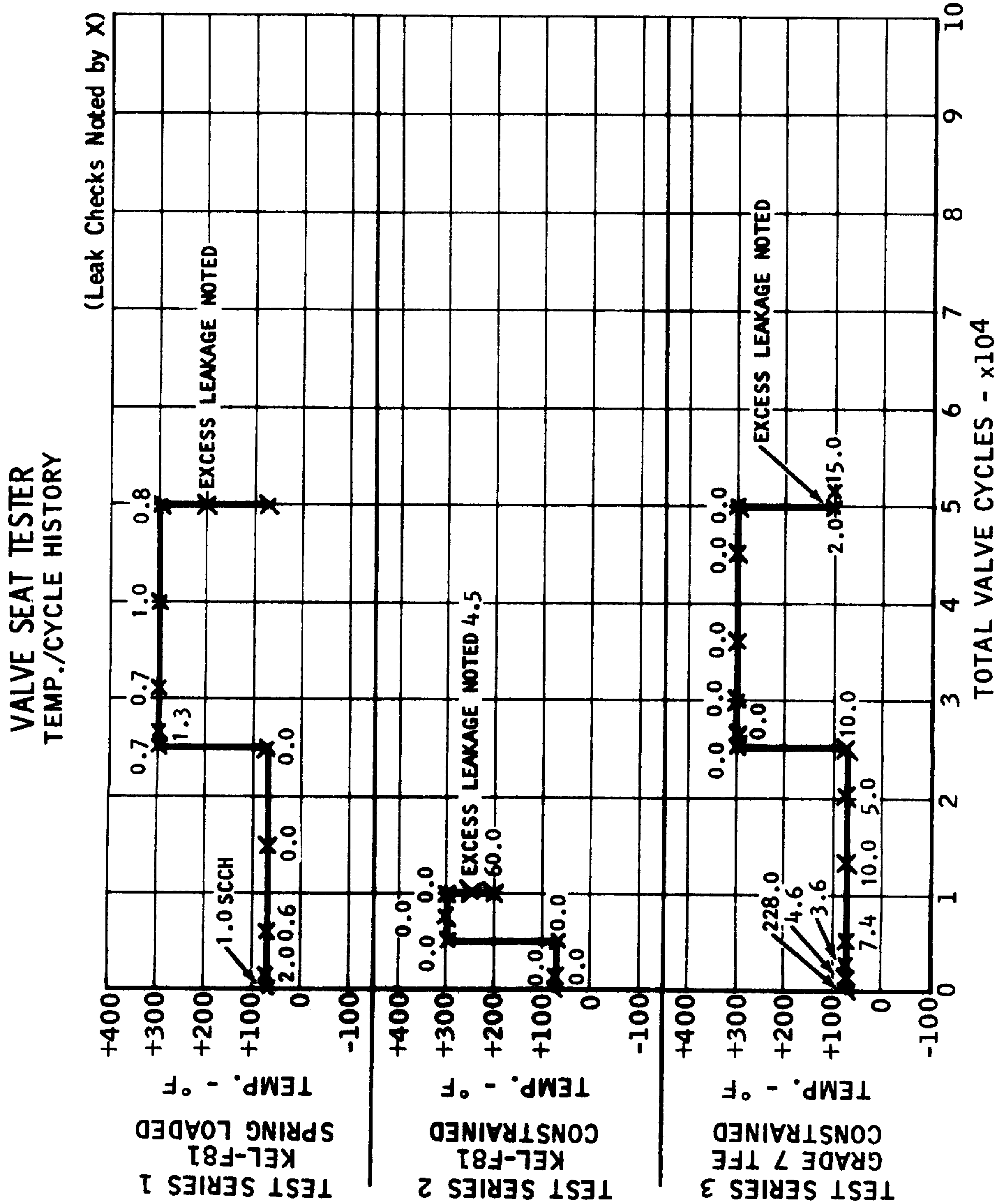

VALVE SEAT TESTER
TEMP./CYCLE HISTORY
(Leak Checks Noted by X)
TOTAL VALVE CYCLES - x104
TEST SERIES 1
KEL-F81
SPRING LOADED
TEST SERIES 2
KEL-F81
CONSTRAINED
TEST SERIES 3
GRADE 7 TFE
CONSTRAINED
TEMP. - °F
EXCESS LEAKAGE NOTED
1.0 SCCH

Figure 16

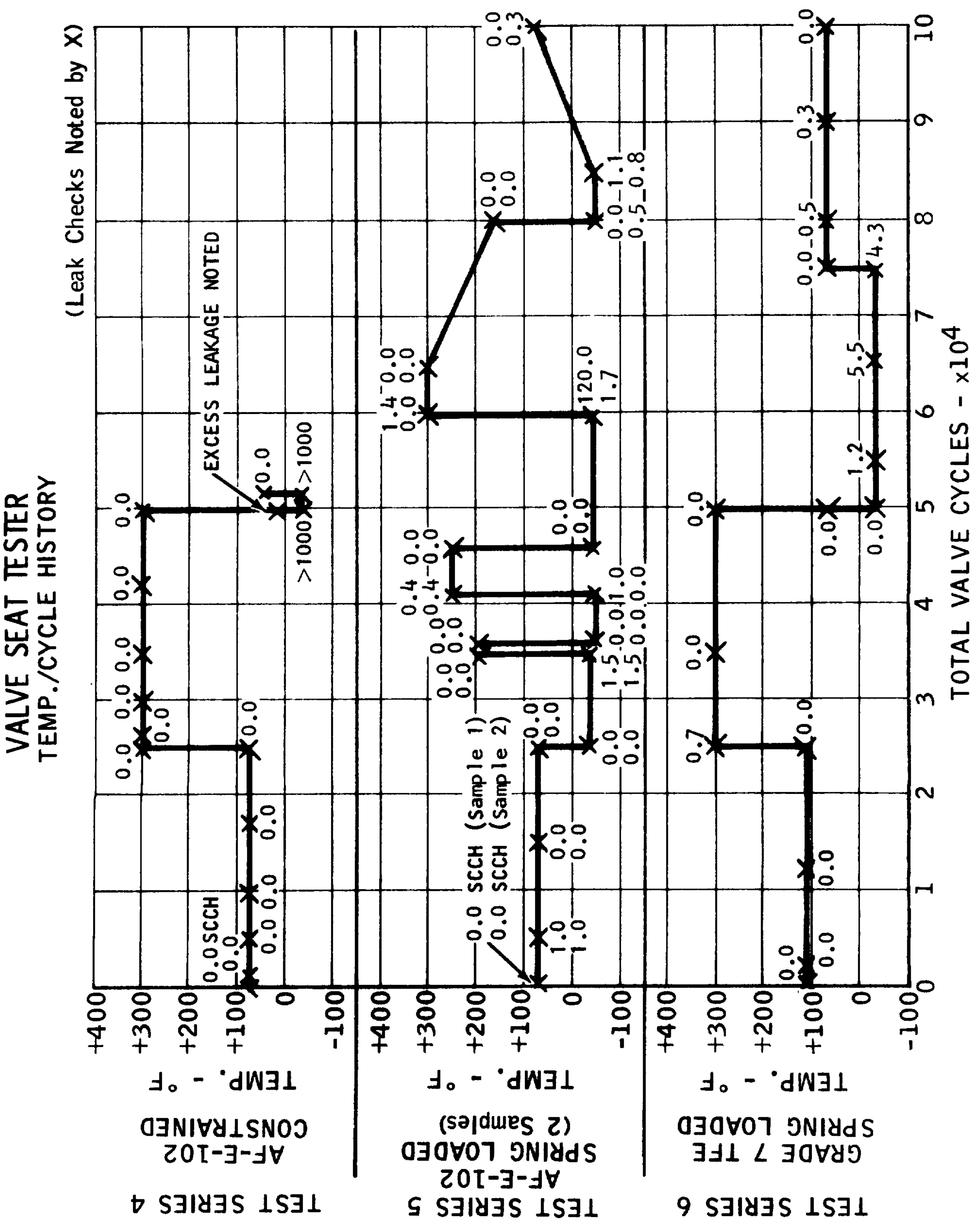


Figure 16 (Cont.)

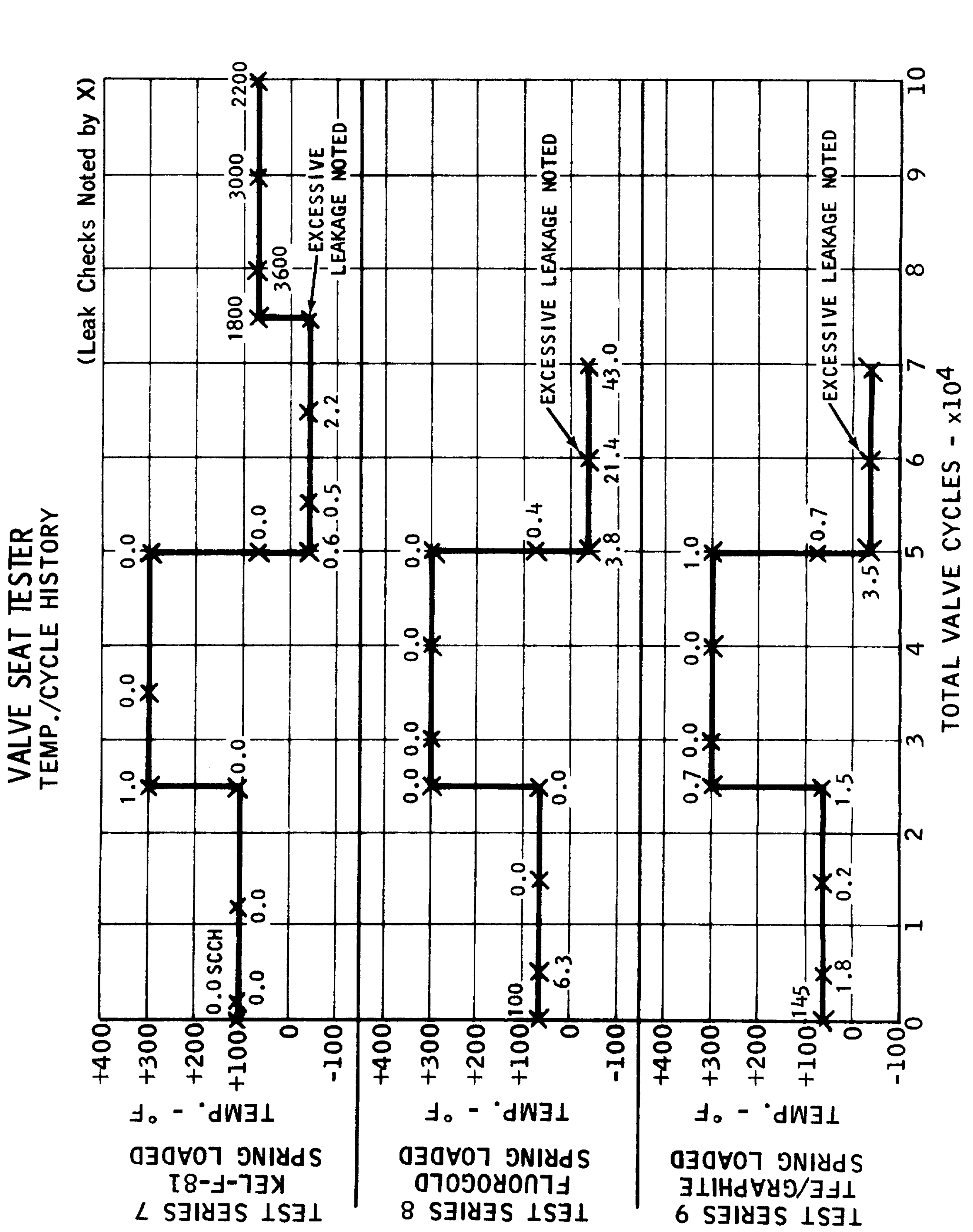


Figure 16 (Cont.)

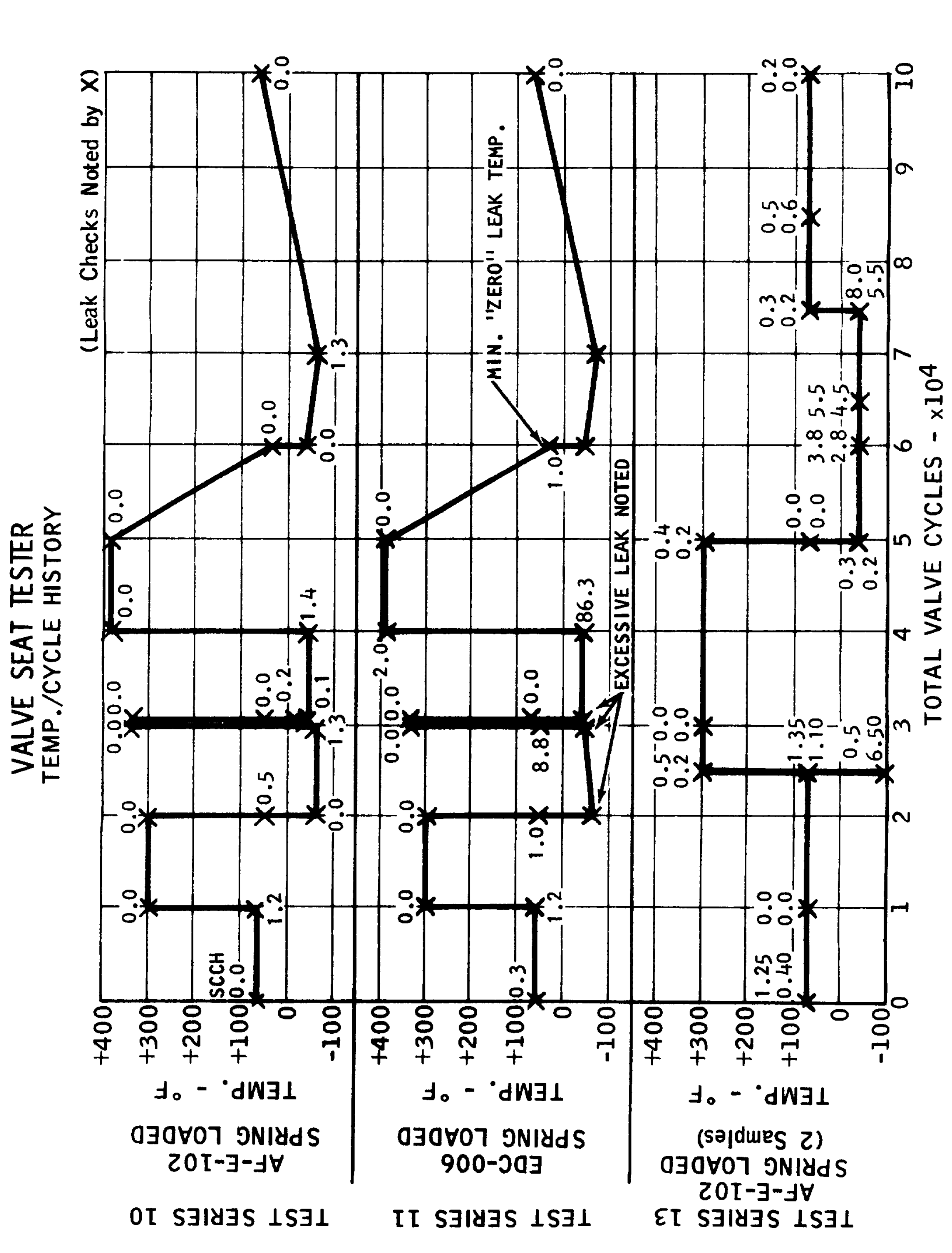
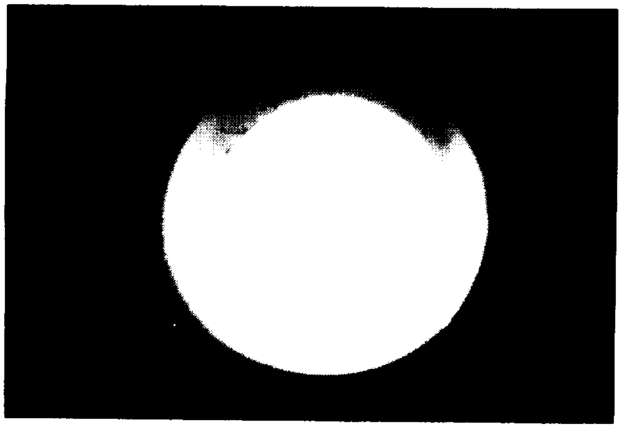


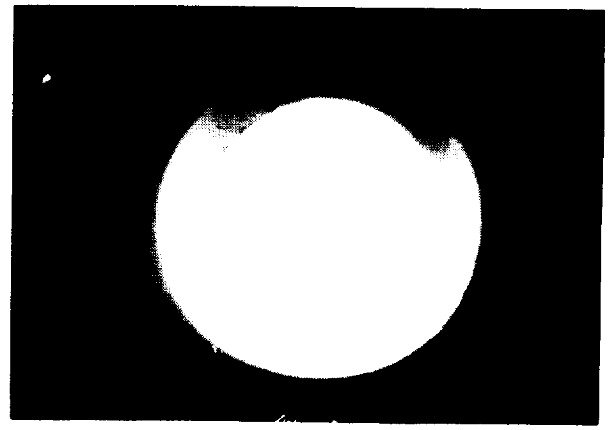
Figure 16 (Cont.)

LATCH VALVE SEAT SEAL DESIGN SUPPORT TEST

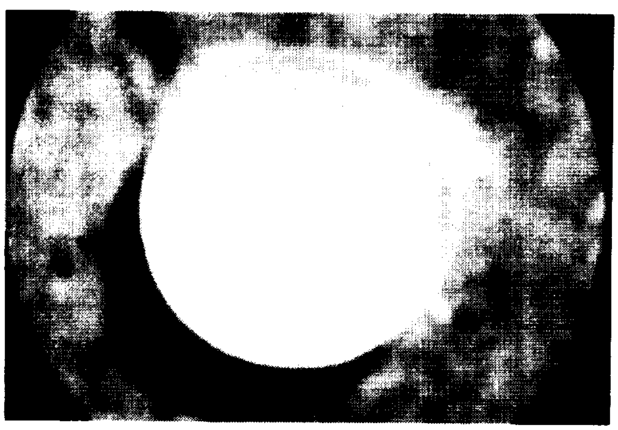
POST TEST APPEARANCE



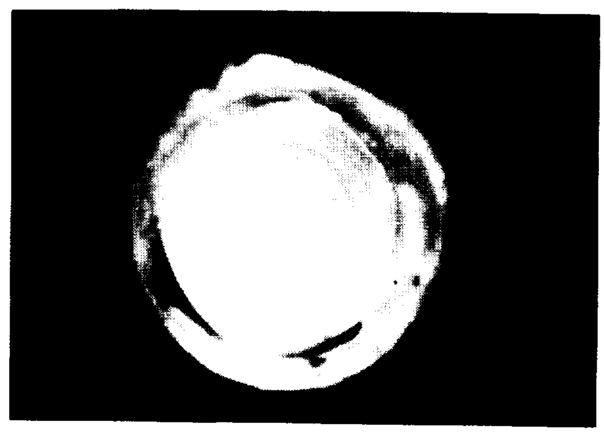
TFE SOLID MOUNT R4D TEST UNIT
25K AMB., 25K 300°F, LEAK



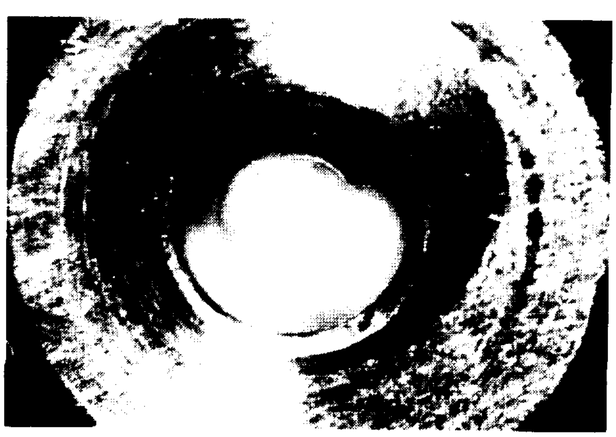
TFE SOLID MOUNT R4D TEST UNIT
25K AMB., 25K 300°F, LEAK



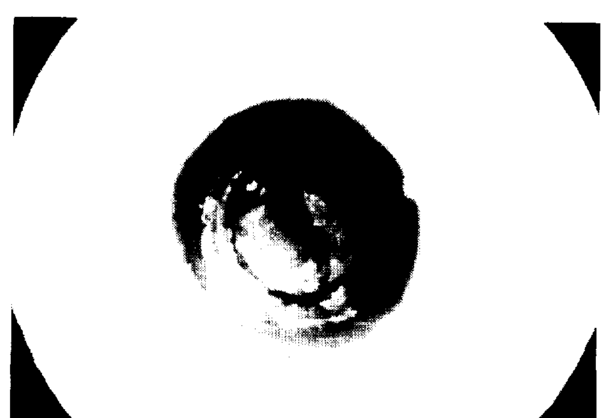
KEL-F SOLID MOUNT R4D TEST UNIT
12.5K TOTAL CYCLES, EXCESS LEAK



KEL-F SPRING MOUNT R4D TEST UNIT
25K CYCLES, LEAK



KEL-F SPRING MOUNT R4D TEST UNIT
25K AMB., 25K 300°F, LEAK

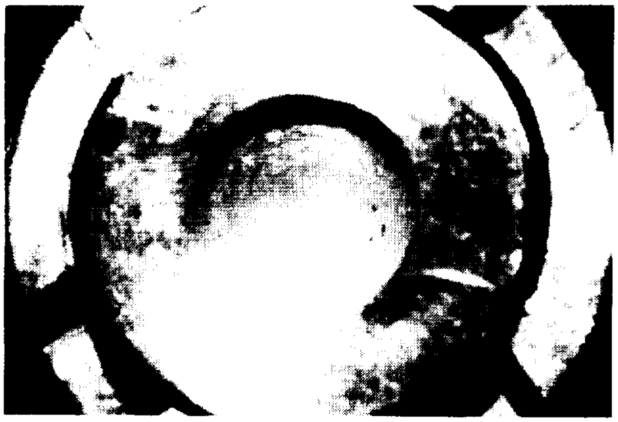


AF-E-102 SOLID R4D TEST UNIT
25K AMB., 25K 300°F, LEAK AT < 30°F

Figure 17

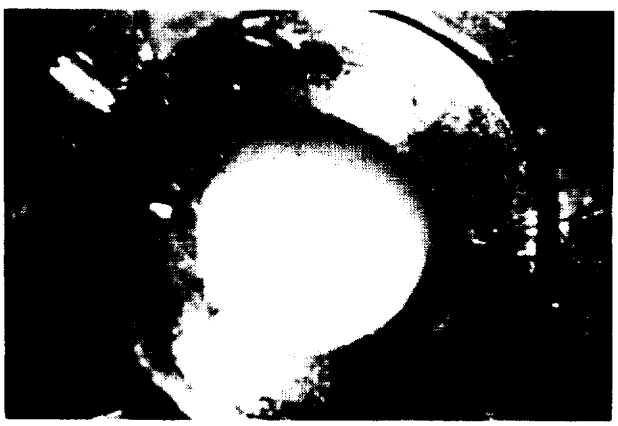
LATCH VALVE SEAT SEAL DESIGN SUPPORT TEST

POST TEST APPEARANCE

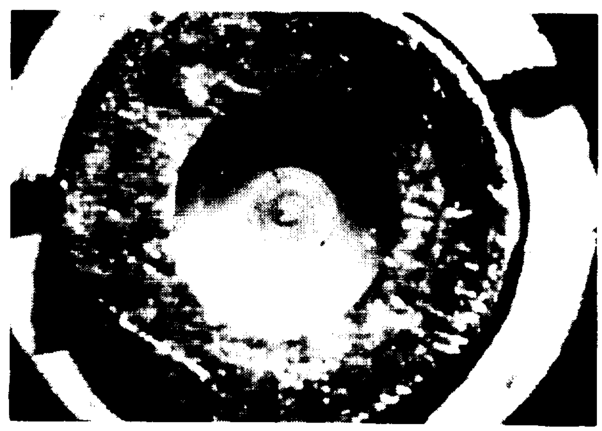


AF-E-102 SPRING MOUNT BIPROP TESTER (FUEL) 100K, -40 TO +300°F

AF-E-102 SPRING MOUNT BIPROP TESTER (OX) 100K, -40 TO +300°F

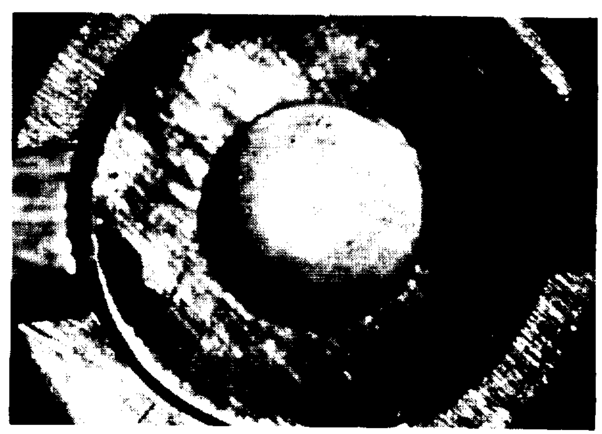


TFE SPRING MOUNT BIPROP TESTER (FUEL) 100K, -40 TO +300°F



KEL-F SPRING MOUNT BIPROP TESTER 100K, -40 TO +300°F

FLUOROGOLD SPRING MOUNT BIPROP TESTER 70K, -40 TO +300°F

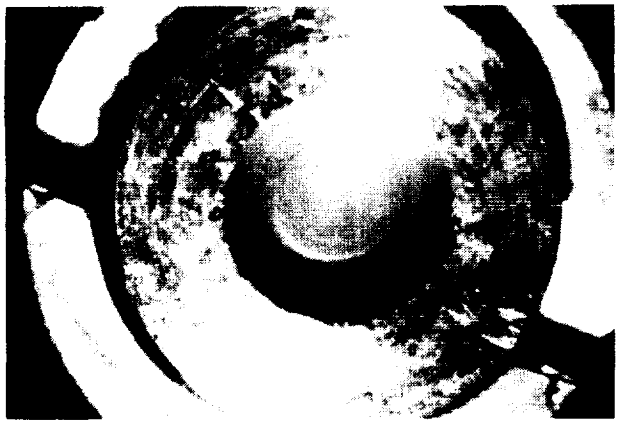


TFE/GRAPHITE SPRING MOUNT BIPROP TESTER (OX) 70K, -40 TO 300°F

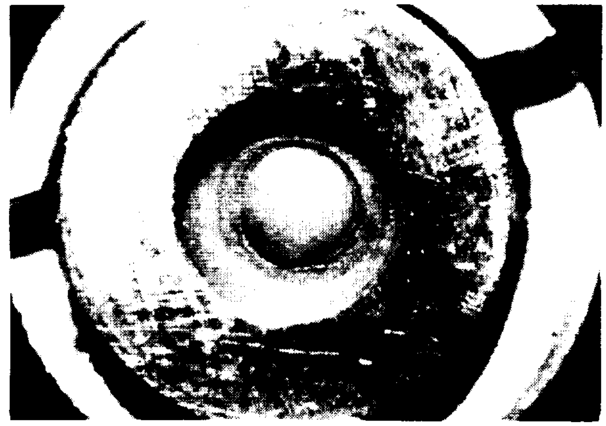
Figure 17 (Cont.)

LATCH VALVE SEAT SEAL DESIGN SUPPORT TEST

POST TEST APPEARANCE

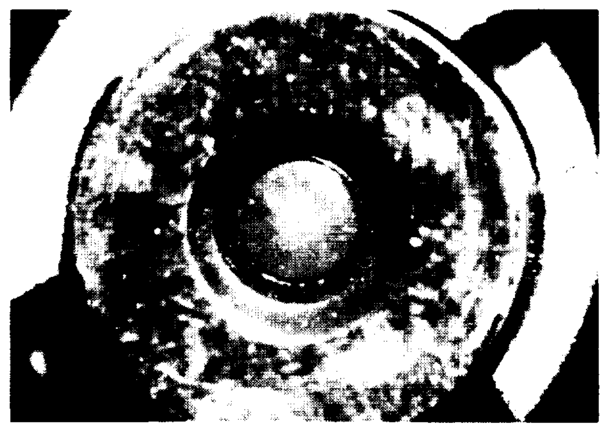


AF-E-102 SPRING MOUNT BIPROP TESTER (FUEL), 100K, -70 TO +390°F

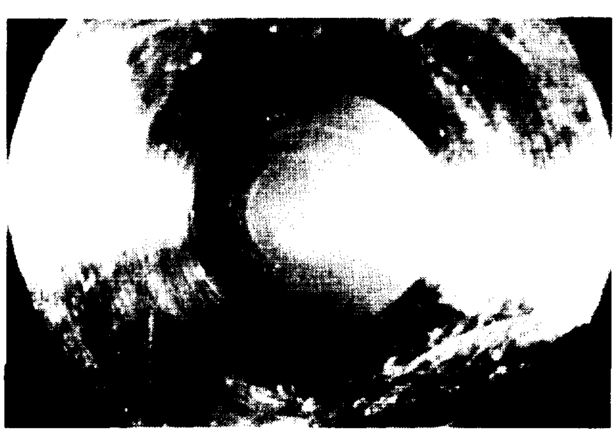


EDC-006 SPRING MOUNT BIPROP TESTER (OX), 100K, -70 TO +390°F

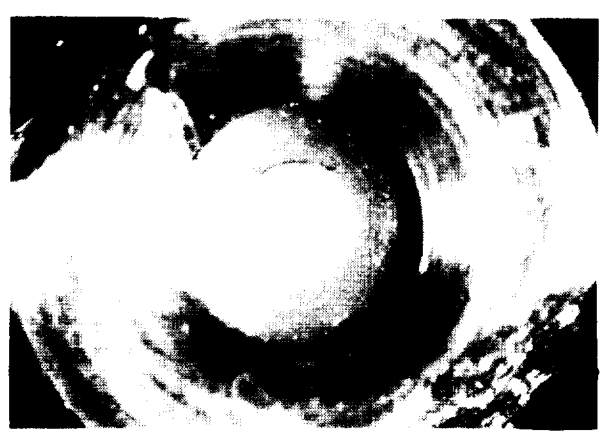
AF-E-102 SPRING MOUNT BIPROP TESTER (FUEL), ABOVE $+.95 \times 10^6$ CYCLES AT 60 PSIG, 70°F



AF-E-124D SPRING MOUNT BIPROP TESTER 0.95×10^6 CYCLES AT 60 PSIG, 70°F



AF-E-102 SPRING LOADED FINAL CONFIG. (OX), 100K CYCLES, -50 TO +300°F



AF-E-102 SPRING LOADED FINAL CONFIG. (OX), 100K CYCLES, -50 TO +300°F

Figure 17 (Cont.)

VALVE FABRICATION

Fabrication and Assembly Plan

Following detail design, an in-house design review and formal design review with the NASA-Langley Project Manager were conducted prior to release of the valve for fabrication. Upon approval of the design, detail part fabrication was initiated and purchased parts were procured. Figure 18 presents a flow diagram for valve build-up and assembly.

Detail Parts Fabrication

Developmental planning was prepared for each detail part, to provide sufficient documentation of manufacturing processes required, to control the flow of the detail parts through the shop, and to establish a control document for recording actual manufacturing techniques. Detail parts fabrication required, primarily, lathe turning operations. No unique special tooling or fixturing are required.

The flexures and spacers were fabricated by chemical milling the required form from 347 CRES sheet stock. The lot of flexure elements was sample tested to verify load-deflection characteristics. Figure 19 plots the envelope of load-deflection characteristics for a single flexure element. The increase in flexure element spring rate, over that analytically predicted, is the result of two considerations not included in the analysis performed:

1. The web width connecting the respective annular rings was taken as the minimum dimension. The corner radii of the milled slots result in a greater effective web width which increases stiffness in bending.

2. The wicking of the braze alloy creates a fillet where the web blends into the inner and/or outer annular ring, stiffening this member in bending.

The flexure elements, as fabricated, will result in a net force of 0.5 lb. tending to unlatch the armature. This net unlatch force is consistent with the assumed value used in determining actuator design parameters.

RESISTOJET LATCH VALVE BUILD-UP FLOW DIAGRAM

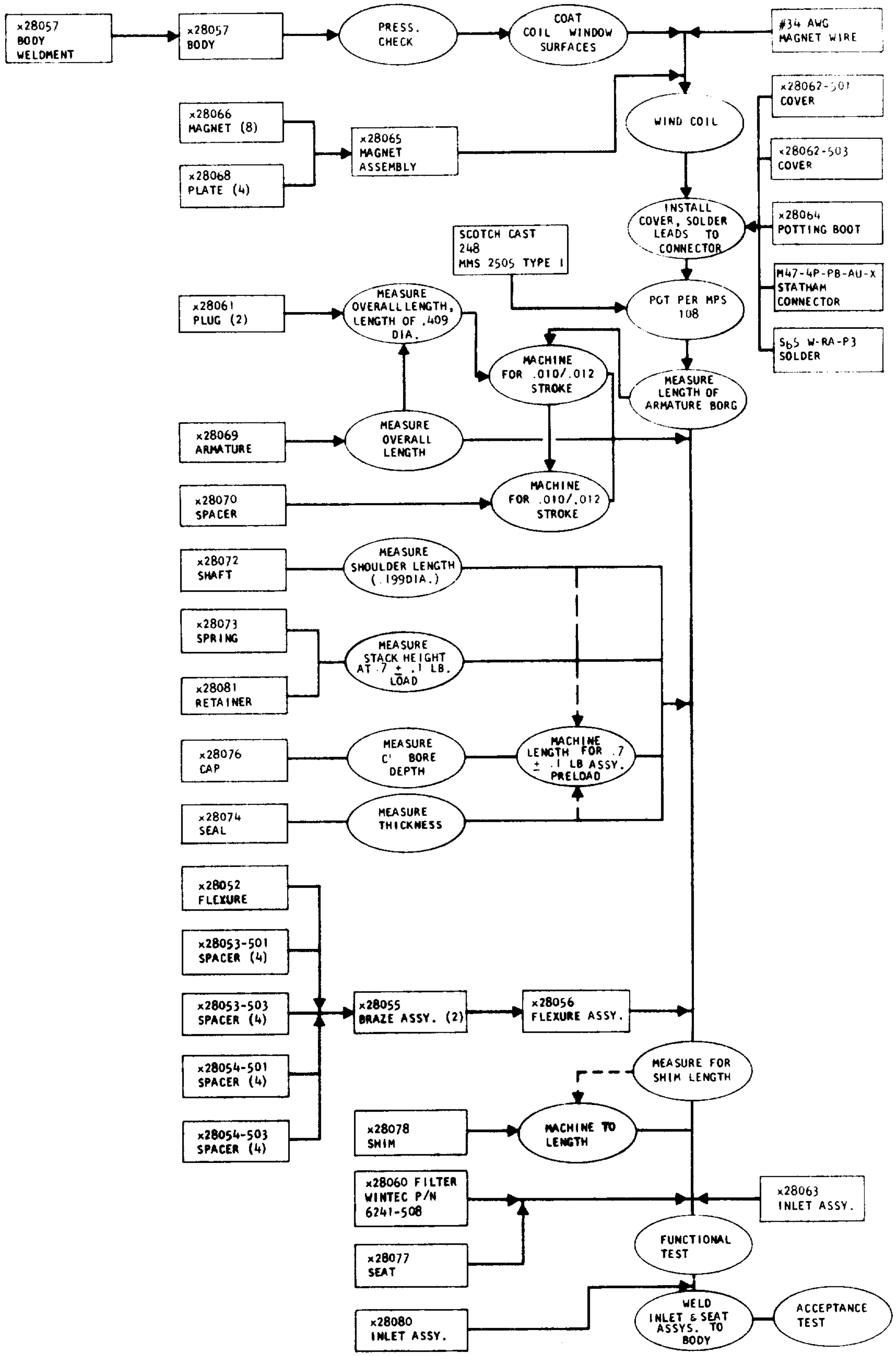


Figure 18

FLEXURE ELEMENT LOAD

DEFLECTION CHARACTERISTICS

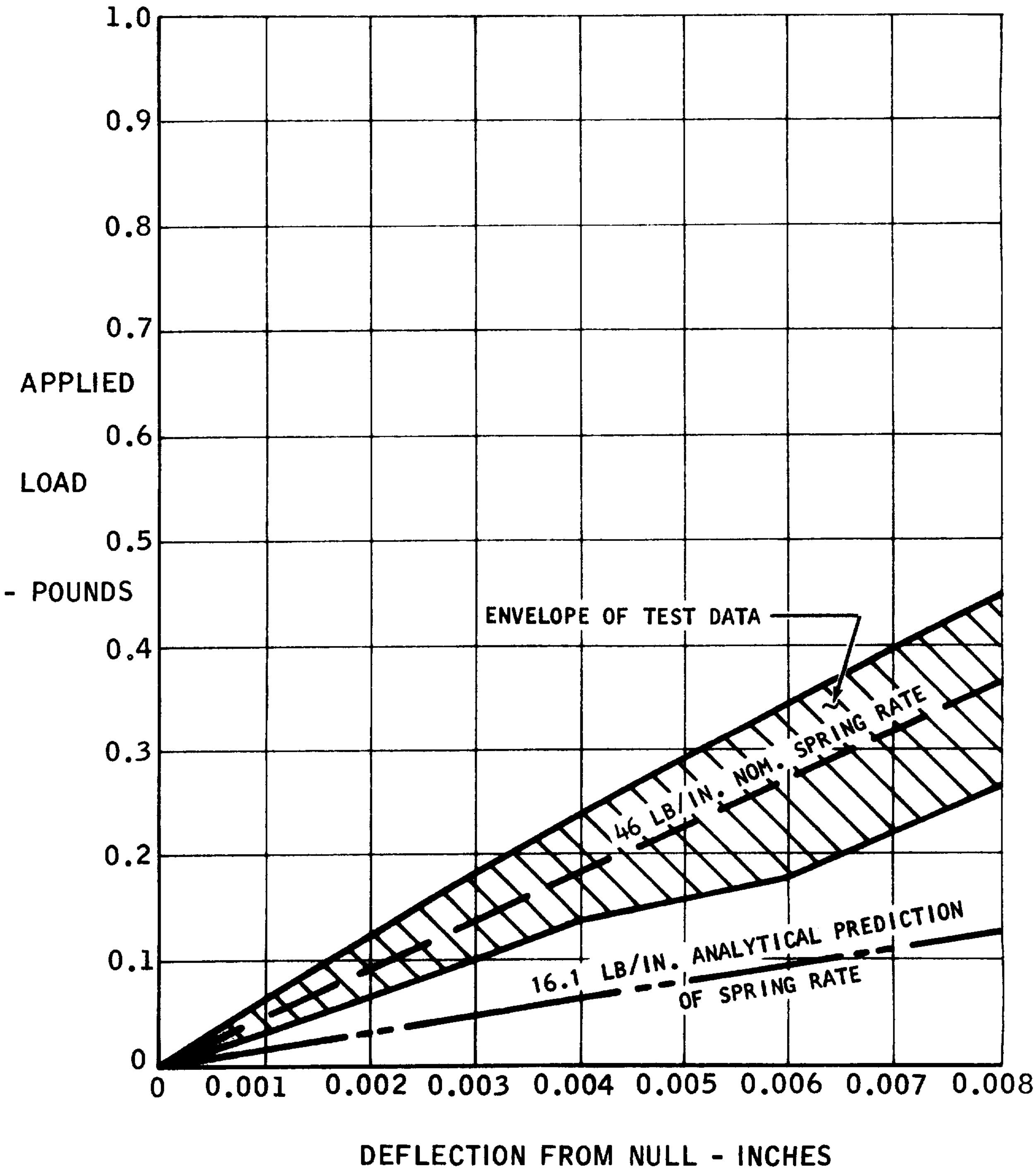


Figure 19

Valve Build-Up

Dimensional log forms were prepared and compiled for critical dimensions of each detail part of the valve. Detail parts were organized by valve serial number. Seal preload spring stack height was determined at the required preload using a spring tester. Detail parts were then machined, as required, to obtain the proper valve stroke, seal preload and valve seat location.

Coil resistance, coil to coil and coil to case insulation resistance at 500 vdc and dielectric strength at 600 vac rms were measured and recorded.

All detail parts, tooling and fixtures were ultrasonically cleaned, rinsed and vacuum dried.

The plugs and armature were then installed in their respective valve bodies and the coils simultaneously energized for 5 seconds at 35 vdc to polarize the permanent magnets. Measurements were then made of the open and close latch force, using a spring tester and applying an increasing load to the armature through a pin. The range of measured open position latch forces was 2.58 to 3.68 lb. while that of the closed position latch forces was 2.60 to 3.80 lb., approximately 150% of the design goal. Unlatching currents were then measured by monitoring current to the respective coils, while increasing applied voltage, until the armature translates to the commanded position. The range of measured magnetomotive force necessary to negate latching and translate the armature under zero applied load was 9.6 to 14.4 ampere turns, significantly less than the analytically predicted value (Ref. Figure 13).

Functional Testing

Each valve assembly was subjected to a functional checkout to substantiate capability to meet design goals. All tests were performed in the clean room of Bldg. 32 of the Marquardt test facility using filter GN_2 or filtered distilled water as a test media.

Valve seat nitrogen leakage was measured at inlet pressures of 60 ±5 psig and at 400±25 psig. Leakage was measured by monitoring displacement of a water column for 6 minutes while the valve was at ambient temperature and the inlet was pressurized at the required level. Of the four X28050 valves and three X28051 valves tested, all exhibited 0.0 SCCH leakage at 400 psig inlet. One exhibited 0.20 SCCH, one exhibited 0.3 SCCH leakage at 60 psig while the remaining five exhibited 0.00 SCCH leakage.

With the valve installed in a water flow system and inlet pressure at approximately 60 psig, a throttle valve was adjusted to obtain required test valve pressure drops and water flow rate at each test point was recorded. The envelope of the measured valve pressure drop characteristics is plotted in Figure 20. Measured values correlated well with the analytically predicted characteristic and demonstrated substantial margin with respect to the design goals of Tables II and III.

With the valve inlet pressurized with GN_2 at the required inlet pressure (60 ±5 psig and 400 ±25 psig static), the voltage to the opening coil was increased while monitoring the coil current draw. The current draw at the time of valve operation was recorded as the pull-in current. This procedure was repeated for the closing coil, with the current draw at time of actuation recorded as the drop out current. The measured values of pull-in and drop out current were converted to operating voltages by multiplying by the respective coil resistances at various coil temperatures. The envelope of the valve operating voltage is plotted in Figure 21. The demonstrated operating voltage represents a significant margin relative to the design requirement.

Valve response was measured at 60 psig and 400 psig GN_2 static inlet pressures and 25 vdc applied coil voltage. The unenergized coil induced voltage was monitored with a Polaroid equipped oscilloscope. Ambient temperature response times, for the two test pressures ranged from 3.8 to 6.3 milliseconds for opening and 3.3 to 5.3 milliseconds for closing. The response is interpreted as the time from electrical signal application until the armature had completed translation. The induced voltage traces obtained during this testing indicated a possible instability in the valve driver circuit. The characteristic of some traces evidenced a momentary random drop out of the command signal one to three milliseconds after initiation of signal. The drop out duration was generally of less than one milliseconds. Since the measured response times were well within the design requirement (Ref. Table II) including any error due to the suspected driver circuit anomoly, and planned development testing would provide an opportunity to verify response, an investigation into the anomoly was not undertaken.

Upon completion of functional testing, each valve was nitrogen purged and vacuum oven dried to remove all residual moisture.

Development Valve Configuration

During valve fabrication and build-up, the only change to the original valve design configuration was the modification to the covers and the addition of the permanent magnet retaining ring, found necessary to accomplish the coil assembly. Figure 22 presents the assembly drawing of the X28050 valve configuration, including changes made during valve build-up.

X28050 AND X28051 VALVE PRESSURE DROP CHARACTERISTIC

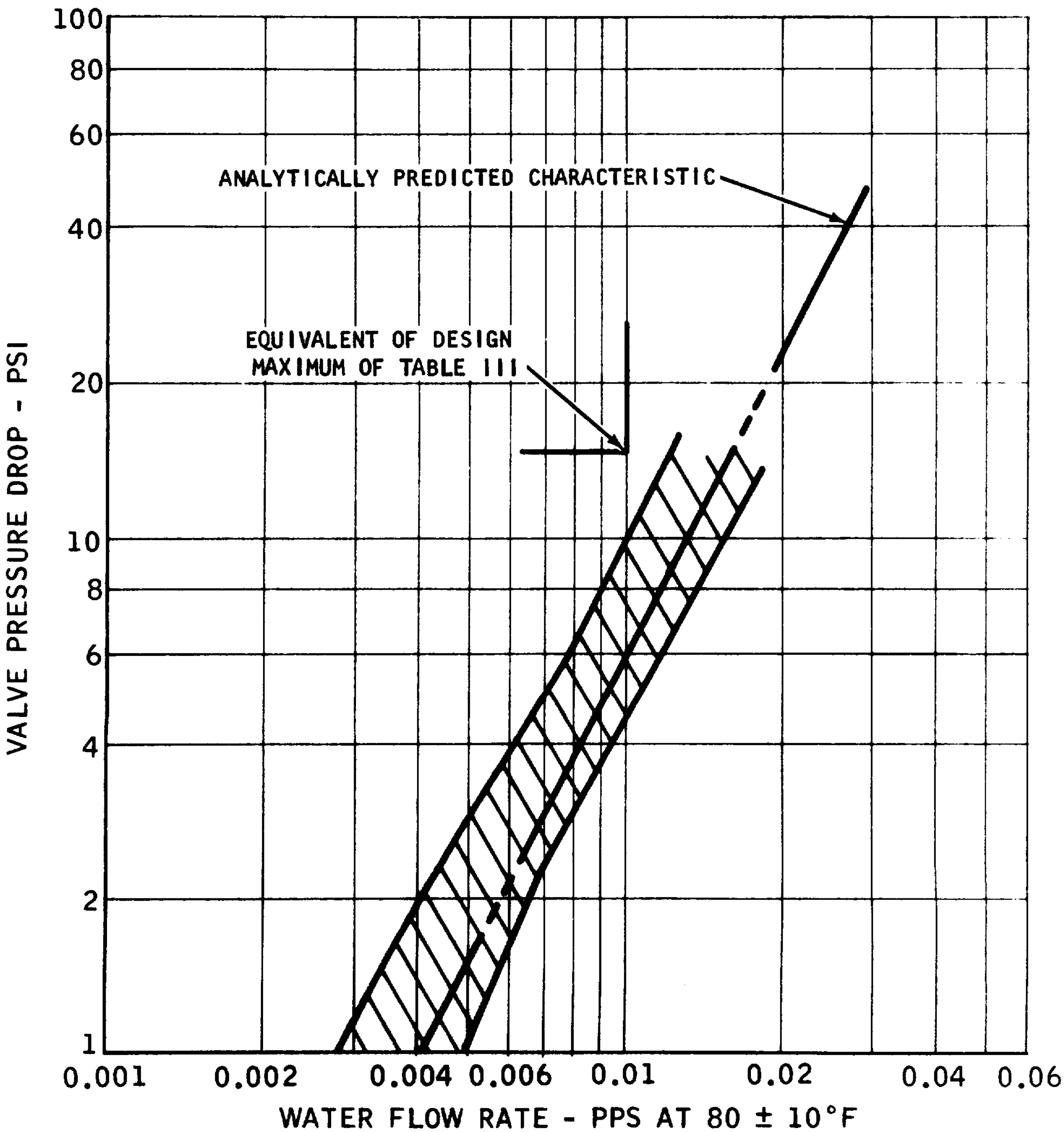


Figure 20

X28050 VALVE "PULL-IN AND DROP-OUT" VOLTAGE vs TEMPERATURE

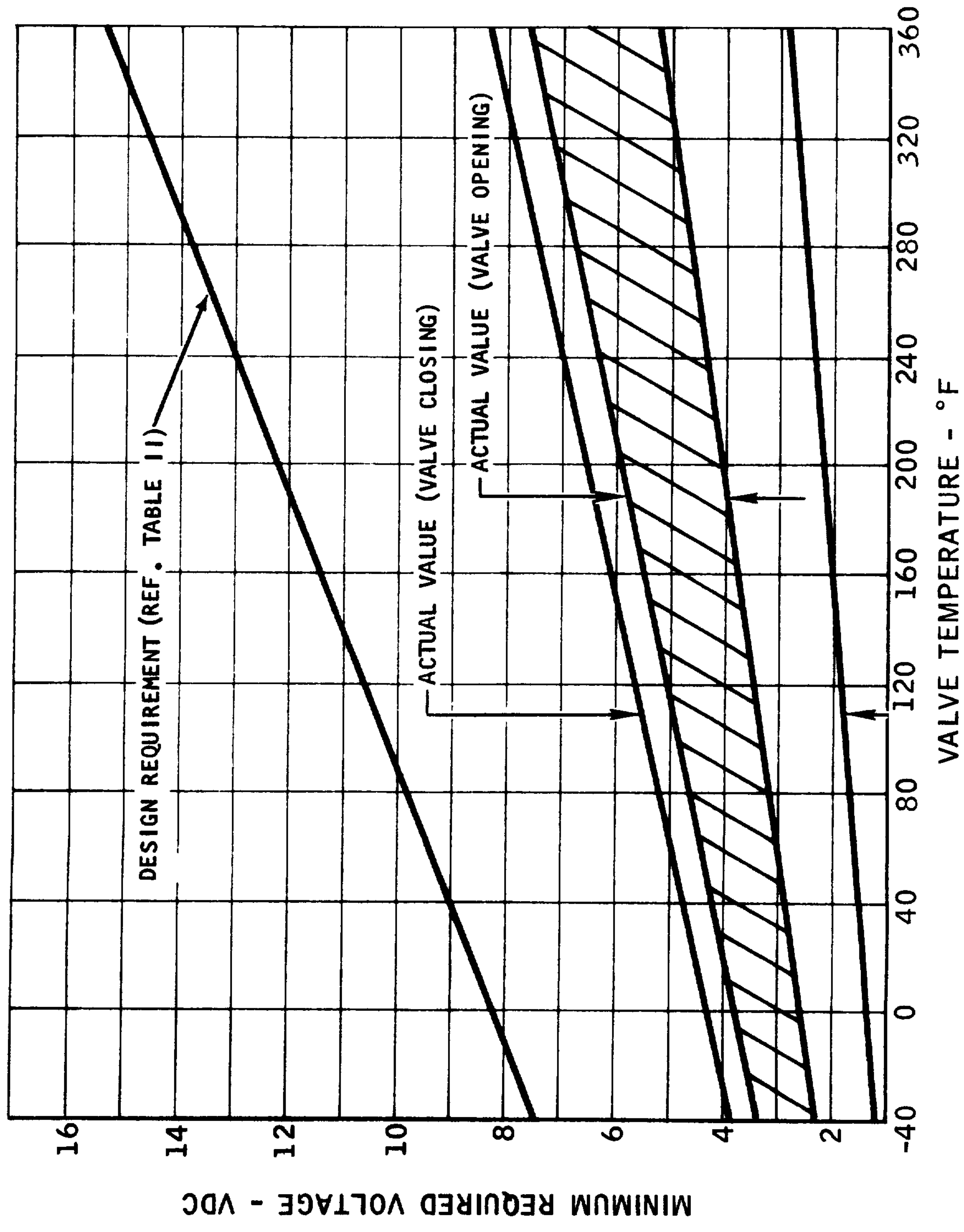
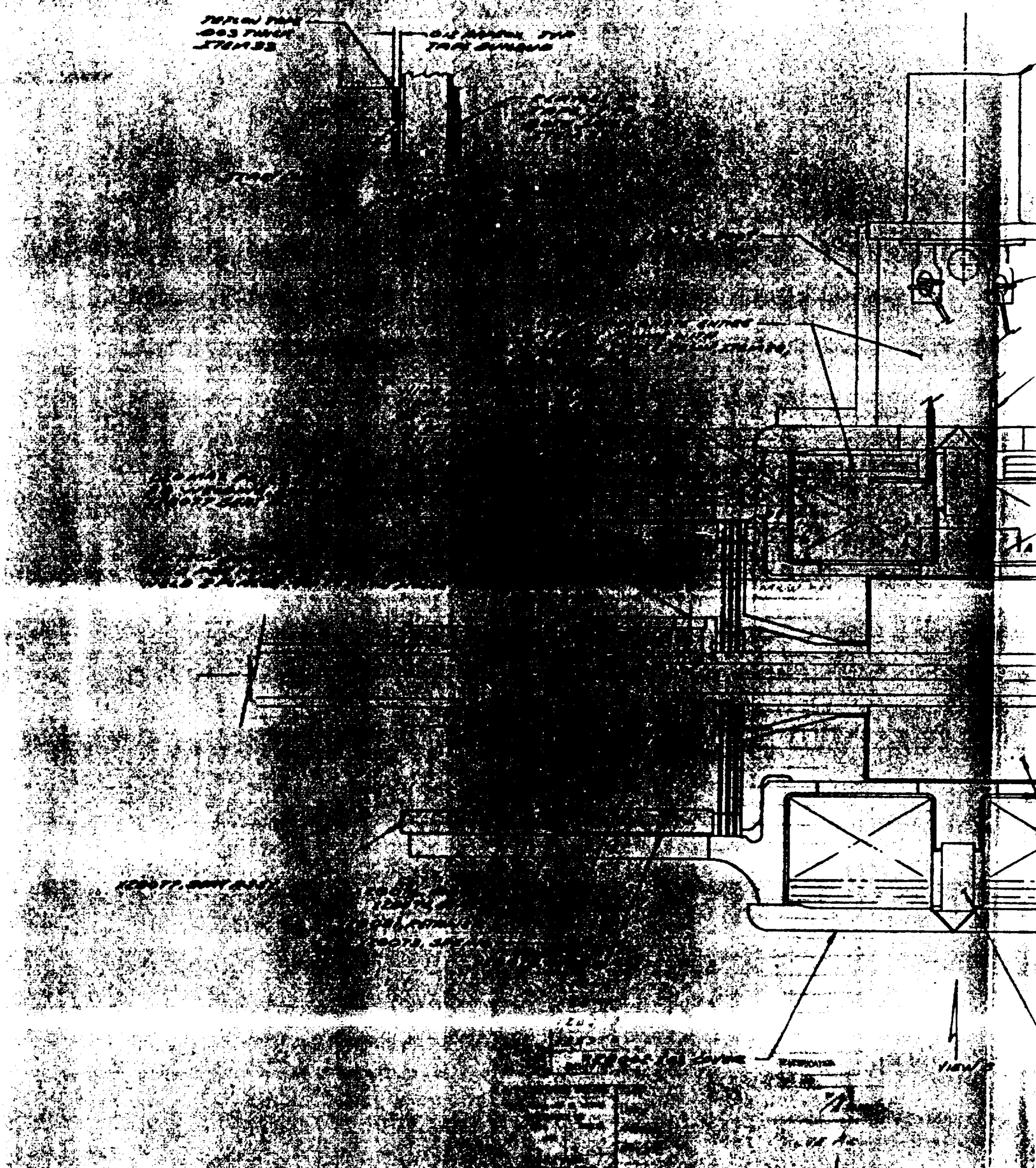
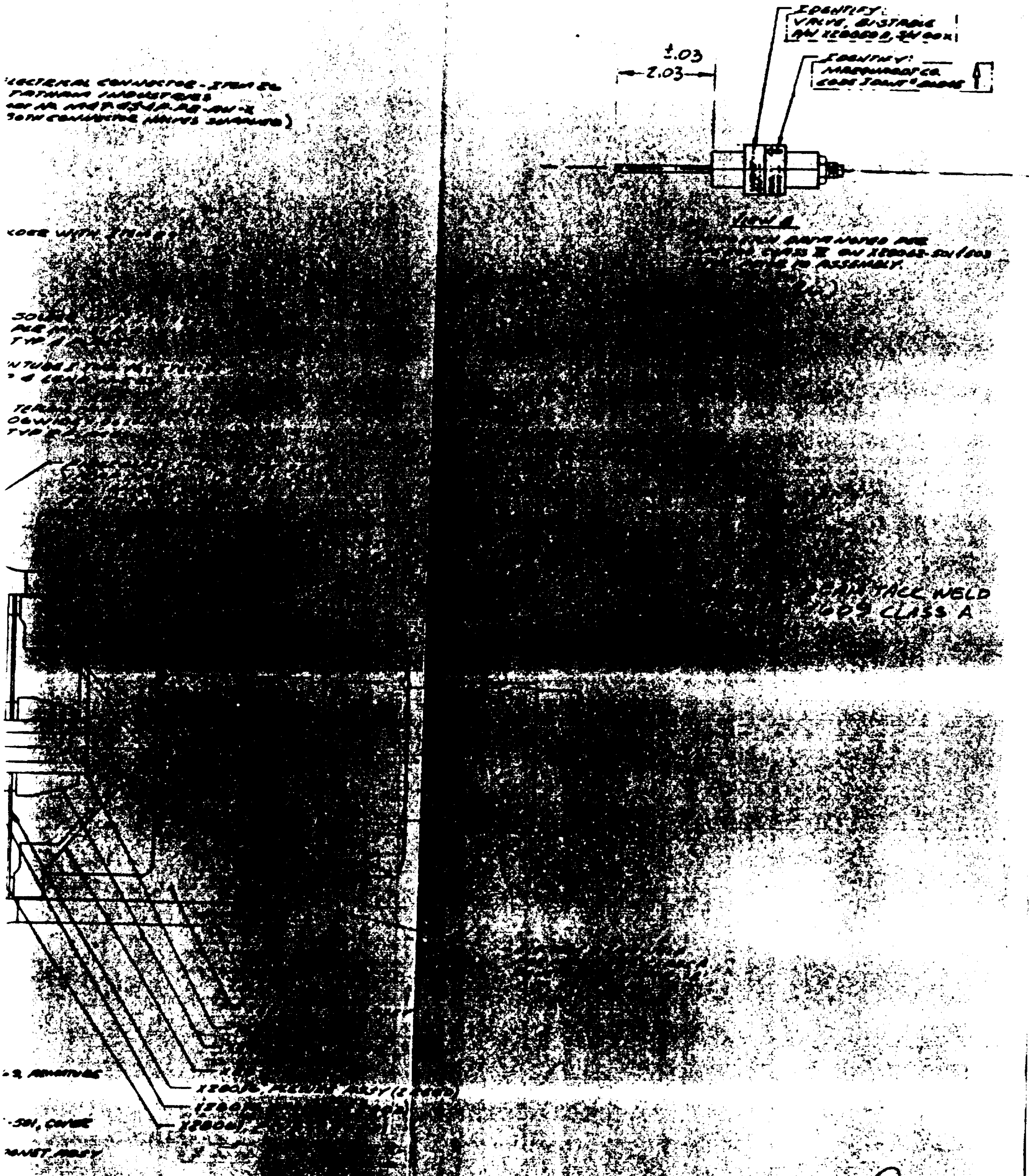


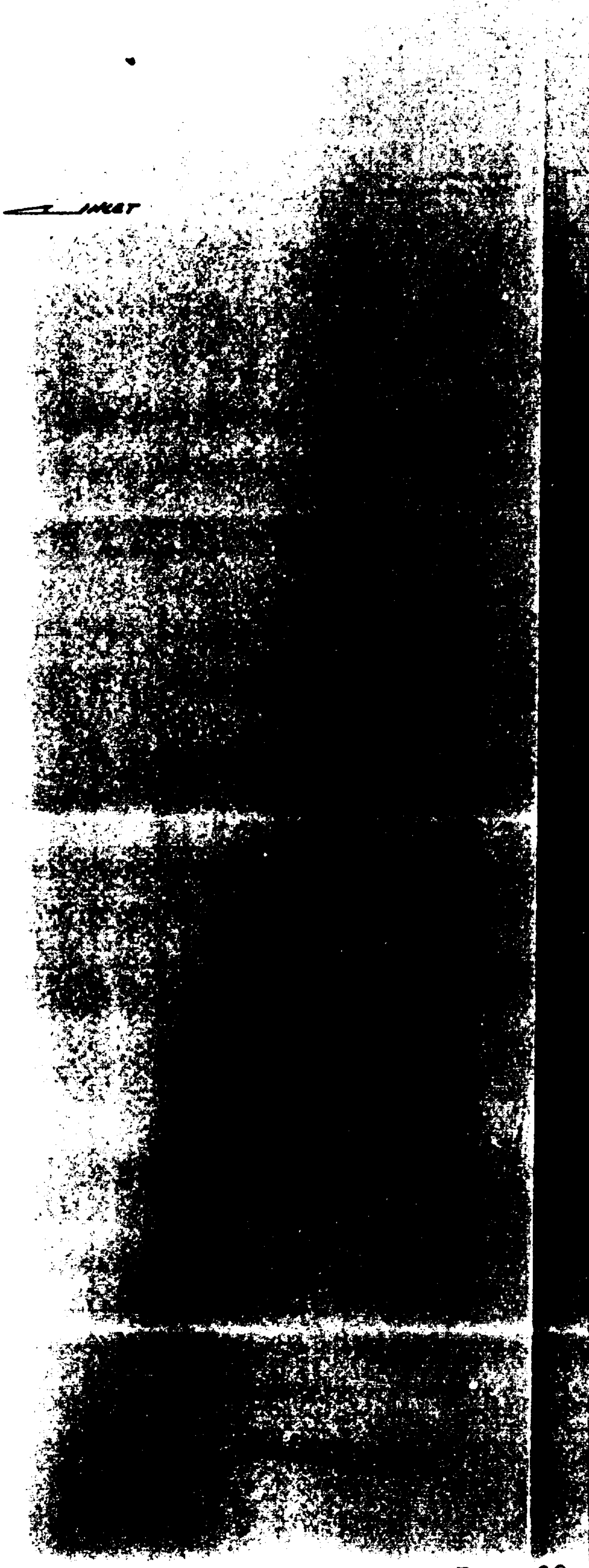
Figure 21



FOLDOUT FRAME



FOLDOUT FRAME 2



Nomenclature or Description	Specification	Material or Note	Zone	Item No.
[illegible]		[illegible]		33
[illegible]		[illegible]		32
[illegible]		[illegible]		31
[illegible]				30
[illegible]	[illegible]	[illegible]		29
[illegible]	[illegible]	[illegible]		28
[illegible]		[illegible]		27
[illegible]		[illegible]		26
[illegible]	MK-W-583	[illegible]		25
				24
				23
				22
				21
				20
[illegible]				19
[illegible] ASSY				18
[illegible]				17
SEAT ASSY				16
CAP				15
SEAL				14
SPRING				13
SHAFT				12
RETAINER				11
SPACER				10
[illegible]				9
MAGNET ASSY				8
BOOT				7
COVER				6
[illegible]				5
PLUG				4
FILTER				3
BODY				2
[illegible] ASSY				1

LIST OF MATERIALS OR PARTS LIST

[illegible] BI-STABLE

88848

X28050

PROJ 5078

SCALE 10 X

SHEET 1 OF 1

FIRST APPL.

LAYOUT

Figure 22

DEVELOPMENT TESTING

Acceptance Testing

After completion of valve build-up, functional checkout and final closure welding, each valve assembly was subjected to an acceptance test to verify its structural and performance integrity.

With the outlet capped and the valve in the "latched open" position, each X28050 valve was pressurized with helium at 200 ±20 psig for five minutes, while the valve was immersed in distilled water. During the test time, there was no evidence of leakage observed from the four valves tested. The three JPL valves were tested in a similar fashion except the valves were pressurized with nitrogen at 800 ±20 psig. No evidence of leakage was observed. Following the testing, valve exterior surfaces were blown dry with gaseous nitrogen.

Valve internal (seat) leakage was measured by monitoring water displacement in a burette while pressurizing the valve inlet with helium. X28050 valves were pressurized to 60 ±5 psig for six minutes while monitoring seat leakage. No evidence of leakage (0.0 SCCH) was detected from any of the four valves tested. The JPL valves were tested at inlet pressures of 20 ±5 psig and 400 ±20 psig. At each inlet pressure, burette displacement was monitored every three minutes for a period of 30 minutes. No evidence of leakage (0.0 SCCH) was observed during the testing of these three valves.

Valve pull-in and dropout current were measured by the method employed during functional testing. Values for the X28050 valves were measured at an inlet pressure of 60 ±5 psig. Average pull-in current for the four valves, increased 11 milliamps over that measured during functional testing. Average dropout current for the four valves increased 5 milliamps over the value measured during functional testing. Pull-in and drop out currents for the three JPL valves were measured at an inlet pressure of 400 ±20 psig. Average pull-in current decreased 19 milliamps from functional test values and average dropout current decreased 2 milliamps from that measured during functional testing.

Valve cleanliness was then verified by performance of a distilled water rinse test wherein a 100 ml sample was taken as the valve was cycled 5 times with the inlet pressurized at 100 ±5 psig. Particle counts of the effluent samples were made to determine relative cleanliness. The two X28050 valves scheduled for build-up into resistojet engines and the three JPL valves were tested in this manner. Figure 23 presents the average particle distribution of the five rinse samples taken. Also shown are the allowable particle distributions in samples to meet Marquardt MPS 210 cleanliness requirements and JPL FS 503574 cleanliness requirements.

RINSE TEST PARTICLE DISTRIBUTION

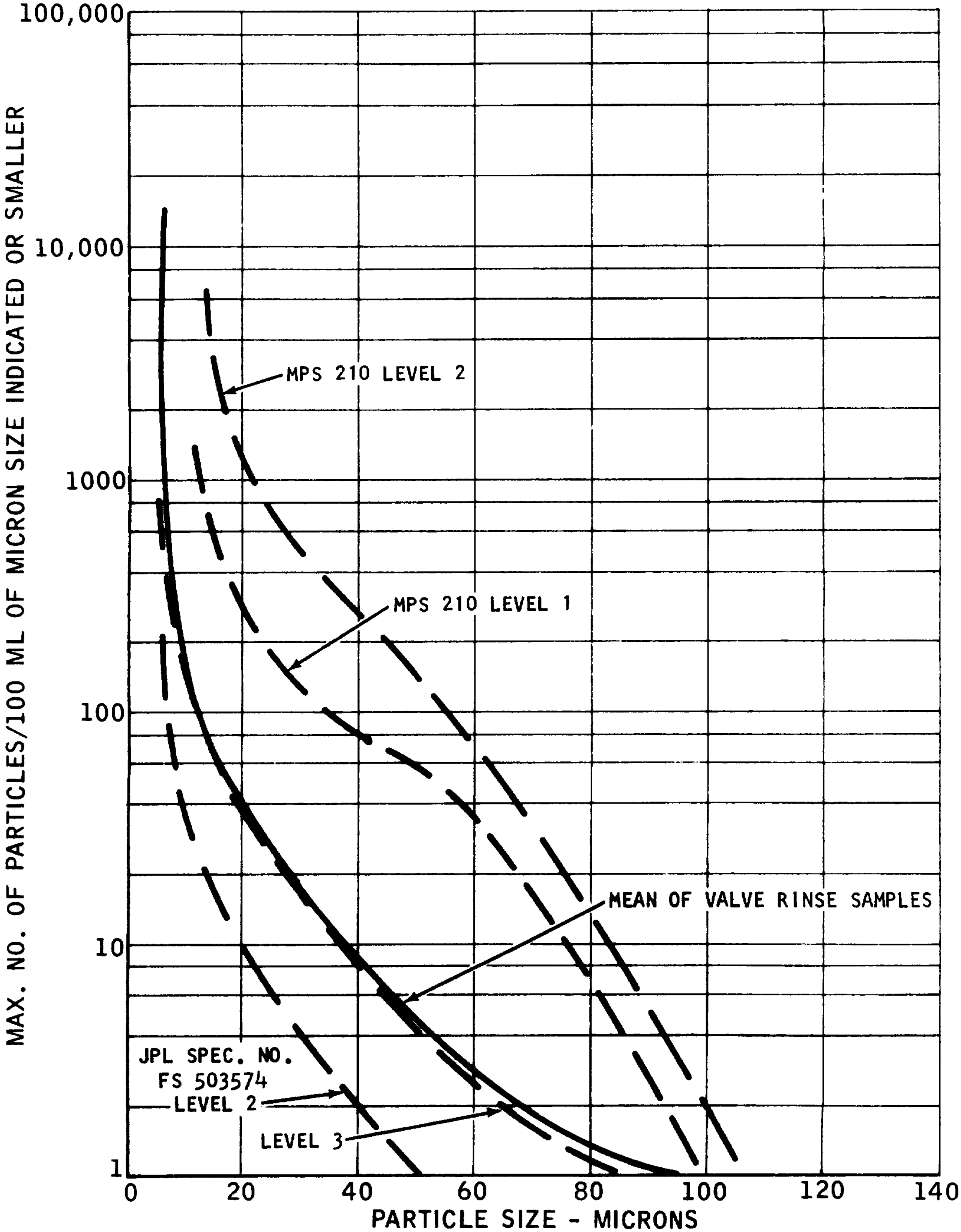


Figure 23

Upon completion of this acceptance test procedure, the valves were nitrogen purged, vacuum oven dried and packaged in double bags. Two X28050 valves were delivered for assembly onto resistojet thrustors for subsequent testing (S/N's 003 and 004). The remaining two X28050 valves (S/N's 001 and 002 were delivered for bench testing in the Marquardt test facility. The three X28051 valves were delivered to J.P.L. for evaluation.

Bench Testing

S/N 001 and 002 X28050 valves were subjected to a series of bench tests to establish performance characteristics over the total design range, demonstrate life cycle capability and determine off-limits capability of the design by testing at levels in excess of the design requirements. All testing was performed in Bldg. 37 of the Marquardt test facility using GN_2 and He as the test fluids, and using instrumentation which had been calibrated and certified for the appropriate range of interest and required accuracy.

After conditioning the valves at the required temperature, pull-in and drop-out current, valve response at 20, 25, 28 and 32 vdc, and valve helium leakage were measured. Test inlet pressures were 30, 60, 300 and 400 psig over the temperature range of +40 to +160°F and 30 and 60 psig over the temperature range of -40 to +300°F. The sequence of testing was such that thermal cycling was accomplished to impose maximum design loads on the valve seat from the standpoint of sealing. Table VII and VIII present the performance data, for S/N 001 and S/N 002 valve respectively, resulting from this testing.

Valve response repeatability over the total test temperature and pressure range was ±0.1 ms for S/N 001 and ±0.2 ms for S/N 002 valve. For the range of temperatures and pressures of the test, valve response time can be considered virtually independent of these two parameters. For the two valves tested, opening and closing response over the total temperature and pressure range, and at an applied voltage of 20 to 32 vdc, was 2.5 ±0.7 milliseconds.

Pull-in and drop-out current measurements, in general, were constant over the total range of test conditions and were consistent with the values measured during valve acceptance testing. Eight of the forty pull-in current measurements and one of the forty drop-out current measurements made, deviated beyond what would be considered normal variability. A portion of this deviation can be attributed to the test equipment since the low current valves hence voltage were on the low threshold of the control band capability of the voltage control rheostat and also were in the low range of the ammeter used to monitor valve current draw (0 to 1.5 amp full scale meter). The magnitude of the pull-in current variation did not manifest itself in a similar wide deviation in valve response. Therefore, it is concluded that this deviation was a random occurrence and does not affect valve performance.

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TABLE VII - PERFORMANCE DATA

[illegible] D X28050 S/N 001

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS PSIG	VOLTAGE vdc		t_{OPEN} ms	t_{CLOSE} ms		I_{PI} AMPS	$I_{D.O.}$ AMPS		HELIUM LEAKAGE SCCH				
70	30						.115	.120		0.0				
		20		2.6	2.5									
		25		2.3	2.2									
		28		2.1	2.1									
		32		2.0	1.9									
	60						.120	.120		0.0				
		20		2.5	2.4									
		25		2.2	2.3									
		28		2.1	2.0									
		32		2.0	2.0									
	300						.125	.125		0.0				
		20		2.6	2.6									
		25		2.3	2.3									
		28		2.1	2.1									
		32		2.0	2.0									
	400						.132	.125		0.0				
		20		2.6	2.6									
		25		2.3	2.3									
		28		2.1	2.1									
		32		2.0	2.0									

GENERAL CALCULATION SHEET

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TABLE VII - PERFORMANCE DATA
(Continued)

MTP 0191 APPENDIX D X-28050 S/N 001

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS. PSIG	VOLTAGE Vdc		t_{OPEN} ms	t_{CLOSE} ms		$I_{P.I}$ AMPS	$I_{D.O.}$ AMPS.		HELIUM LEAKAGE SCCH				
165	30						.120	.120		0.0				
		20		2.7	2.6									
		25		2.4	2.3									
		28		2.2	2.1									
		32		2.0	2.0									
	60						.120	.122		0.0				
		20		2.7	2.6									
		25		2.4	2.3									
		28		2.2	2.1									
		32		2.0	2.0									
	300						.130	.115		0.0				
		20		2.7	2.6									
		25		2.4	2.4									
		28		2.2	2.2									
		32		2.0	2.0									
	400						.130	.115		0.0				
		20		2.7	2.7									
		25		2.4	2.4									
		28		2.3	2.2									
		32		2.1	2.0									

GENERAL CALCULATION SHEET

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TABLE VII - PERFORMANCE DATA
(Continued)

MTP 0191 APPENDIX D x 28050 S/N 001

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS PSIG	VOLTAGE vdc		t_{OPEN} ms	t_{CLOSE} ms		$I_{P.I}$ AMPS	$I_{D.O.}$ AMPS		HELIUM LEAKAGE SCCH				
25-35	30						.118	.115		0.0				
		20		2.5	2.5									
		25		2.3	2.2									
		28		2.0	2.0									
		32		1.9	1.9									
	60						.115	.110		3.0				
		20		2.6	2.6									
		25		2.3	2.3									
		28		2.0	2.1									
		32		1.8	2.0									
	300						.120	.085		0.5				
		20		2.7	2.5									
		25		2.3	2.3									
		28		2.0	2.0									
		32		1.9	2.0									
	400						.120	.110		0.0				
		20		2.7	2.5									
		[illegible]		2.3	2.2									
		28		2.1	2.1									
		32		1.8	2.0									

GENERAL CALCULATION SHEET

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TABLE VII - PERFORMANCE DATA
(Continued)

MTP 0191 APPENDIX D x 28050 S/N 001

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS PSIG	VOLTAGE vdc		t_{OPEN} ms	t_{CLOSE} ms		$I_{P.I.}$ AMPS	$I_{D.O.}$ AMPS		HELIUM LEAKAGE SCCH				
262	30						.190	.115		0.0				
		20		2.6	2.7									
		25		2.2	2.2									
		28		2.1	2.1									
		32		1.9	2.0									
262	60						.190	.110		0.0				
		20		2.7	2.6									
		25		2.4	2.2									
		28		2.2	2.0									
		32		2.0	1.9									
300	30						.120	.115		0.0				
		20		2.7	2.6									
		25		2.3	2.3									
		28		2.1	2.1									
		32		1.9	1.9									
300	60						.120	.115		0.0				
		20		2.6	2.6									
		25		2.3	2.2									
		28		2.2	2.1									
		32		1.9	1.9									

MAR 21

		VCC	MA	MA	AMPS	AMPS	
0	30				120	110	2.7
		20	2.6	2.4			
		25	2.3	2.2			
		28	2.2	2.0			
		32	2.0	[illegible]			
	60				.120	.110	0.5
		20	2.6	2.4			
		25	2.3	[illegible]			
		28	2.2	[illegible]			
		32	2.0	[illegible]			
-20	30				.125	.110	1.3
		20	2.3	2.3			
		25	2.2	2.1			
		28	2.1	2.0			
		32	2.0	[illegible]			
	60				.130	.105	[illegible]

GENERAL CALCULATION SHEET

TABLE VIII - PERFORMANCE DATA

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MTP 0191 Appendix D X280050 S/N 002

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Temp °F	Inlet Press PSIG	Voltage vdc		t_{open} ms	t_{close} ms		I_{PI} Amps	$I_{D.O.}$ Amps		Helium Leakage scch				
70	30						.190	.147		0.5				
		20		2.9	3.0									
		25		2.6	2.6									
		28		2.4	2.4									
		32		2.1	2.2									
	60						.188	.148		0.0				
		20		3.0	3.0									
		25		2.5	2.6									
		28		2.4	2.4									
		32		2.2	2.2									
	300						.205	.145		0.0				
		20		3.0	3.0									
		25		2.6	2.5									
		28		2.4	2.4									
		32		2.2	2.2									
	400													
		20		3.0	3.0		.210	.140		0.0				
		25		2.6	2.5									
		28		2.4	2.4									
		32		2.3	2.2									

GENERAL CALCULATION SHEET

MAC 8830 REV. 4-57

TABLE VIII - PERFORMANCE DATA
(Continued)

MTP 0191 Appendix D x28050 S/N 002

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Temp °F	Inlet Press. PSIG	Voltage vdc		t_{open} ms	t_{close} ms		$I_{P.I.}$ Amps	$I_{D.O.}$ ms		Helium Leakage scch				
165	30						.130	.140		0.0				
		20		2.9	3.1									
		25		2.4	2.7									
		28		2.3	2.5									
		32		2.1	2.4									
	60						.130	.140		0.0				
		20		3.0	3.1									
		25		2.6	2.7									
		28		2.4	2.5									
		32		2.3	2.3									
	300						.140	.125		0.0				
		20		3.1	3.0									
		25		2.7	2.5									
		28		2.5	2.4									
		32		2.3	2.3									
	400						.142	.125		0.0				
		20		3.1	3.0									
		25		2.6	2.5									
		28		2.5	2.5									
		32		2.4	2.3									

GENERAL CALCULATION SHEET

MAC B830 REV. 4-57

TABLE VIII - PERFORMANCE DATA
(Continued)

MTP 0191 APPENDIX D

x 280SD S/N 002

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS PSIG	VOLTAGE vdc		t_{OPEN} ms	t_{CLOSE} ms		$I_{P.I}$ AMPS	$I_{D.O.}$ AMPS		HELIUM LEAKAGE SCCH				
25-40	30						.132	.140		0.0				
		20		2.9	2.8									
		25		2.6	2.5									
		28		2.2	2.4									
		32		2.0	2.1									
	60						.130	.140		2.0				
		20		3.0	2.9									
		25		2.6	2.6									
		28		2.4	2.3									
		32		2.1	2.2									
	300						.140	.135		1.0				
		20		2.9	2.9									
		25		2.6	2.5									
		28		2.4	2.3									
		32		2.3	2.2									
	400						.140	.135		0.0				
		20		3.1	2.8									
		25		2.7	2.4									
		28		2.5	2.3									
		32		2.3	2.2									

GENERAL CALCULATION SHEET

MAC 8830 REV. 4-57

TABLE VIII - PERFORMANCE DATA
(Continued)

MTP 0191 Appendix D x 28050 S/N 002

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
Temp °F	Inlet Press PSIG	Voltage vdc		t_{open} MS	t_{close} MS		$I_{P.I}$ AMPS	$I_{D.O.}$ AMPS		Helium Leak SCCH				
263	30						.220	.140		0.0				
		20		3.2	3.2									
		25		2.6	2.7									
		28		2.5	2.6									
		32		2.3	2.4									
	60						.220	.135		0.0				
		20		3.1	3.2									
		25		2.7	2.7									
		28		2.5	2.6									
		32		2.3	2.4									
300	30						.135	.135		0.0				
		20		3.2	3.2									
		25		2.7	2.8									
		28		2.5	2.6									
		32		2.3	2.4									
	60						.135	.125		0.0				
		20		3.2	3.1									
		25		2.7	2.7									
		28		2.5	2.5									
		32		2.3	2.4									

GENERAL CALCULATION SHEET

MAC B830 REV. 4-57

TABLE VIII - PERFORMANCE DATA
(Continued)

MTP 0191 APPENDIX D X-28050 S/N 002

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS, PSIG	VOLTAGE vdc		t_{open} ms	t_{close} ms		$I_{P.I.}$ AMPS	$I_{D.O.}$ AMPS		HELIUM LEAKAGE scch				
0	30						.138	.130		0.3				
		20		3.0	2.8									
		25		2.6	2.4									
		28		2.4	2.2									
		32		2.2	2.0									
	60						.140	.130		0.2				
		20		3.1	2.7									
		25		2.6	2.4									
		28		2.4	2.2									
		32		2.2	2.0									
-50	30						.145	.130		0.8				
		20		3.0	2.7									
		25		2.6	2.4									
		28		2.3	2.2									
		32		2.1	2.0									
	60						.160	.130		1.7				
		20		3.0	2.7									
		25		2.6	2.4									
		28		2.3	2.2									
		32		2.1	2.0									

Helium leakage measurements made over the temperature and pressure test range, resulted in occasional evidence of very low magnitude leakage. All instances of indicated leakage occurred during testing at temperatures below ambient. The test method, wherein the test valve is submerged in a temperature controlled bath and the leakage monitoring burette is at ambient temperature, is susceptible to false leakage indication by virtue of the warming, and expanding of the trapped gas volume between the valve seat and the water column. Since the test set-up resulted in a significantly large trapped gas volume and insulation was not used to protect the tubing from room temperature gradients, leakage measurement errors were a possibility. Nevertheless, the magnitude of the observed leakages were such that valve seat seal integrity was maintained and subsequent testing would offer opportunities to again verify leakage characteristics at similar thermal conditions.

Valve cycle life was demonstrated by the completion, by each test valve, of 100,000 cycles of operation, with 60 psig GN_2, over the temperature range of -40 to +300°F. The cycle life test was performed in 10 increments of 10,000 cycles each. Each increment was performed after conditioning the valve at a temperature within the test range such that the total range was covered. After each increment of cycling, valve response at 28 vdc and helium leakage at 60 psig was measured and recorded at the respective test temperature. A summary is presented in Tables IX and X and the cycle life temperature and leakage histories are plotted in Figure 24. Throughout the cycle life test, valve leakage was virtually zero as indicated by water displacement monitoring. Two occurrences of leakage were noted. In both cases, both valves indicated leakage and though some difference in leakage magnitude resulted, the origin of the noted leakage is thought to be thermal instability of the trapped gas volume rather than seat leakage.

With cycle life testing completed, the valves were subjected to an off-limits test series to establish design margins and demonstrate maximum valve capability. To define a baseline and to substantiate post life cycle test valve condition, the initial test series was performed at ambient temperature. Response at 20, 25, 28 and 32 vdc, pull-in and drop-out current, and helium leakage were measured at 20 ±5 psig and 400 ±25 psig. After cycling the valve 1,000 cycles with 60 psig, these parameters were again measured. The valve was then conditioned at +350°F. During this conditioning, oven-overshoot to a valve temperature of +400°F occurred. After re-establishing valve temperature at +350°F, valve performance was measured. During pull-in current measurement, the S/N 002 exhibited a high unstable value upon testing at 20 psig inlet pressure. After several attempts to obtain a stable measurement, the valve ultimately failed to open. The S/N 001 valve exhibited a high unstable drop-out current measurement and though the valve did not fail to close, current draw was erratic and excessive. Both valves were removed from test and allowed to cool to ambient temperature. Using a Wheatstone Bridge, coil resistance and coil to case resistance was measured. The S/N 001 valve closing coil resistance was

GENERAL CALCULATION SHEET

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TABLE IX CYCLE LIFE TEST DATA

MTP 0191 APPENDIX G X28050 S/N 001

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS PSIG	CUMULATIVE CYCLES		to 28vdc 60 PSIG MS		tc 28vdc 60 PSIG MS								HELIUM LEAKAGE SCCH
72	60	10,006		—		—								0.0
215		20,009		2.2		2.2								0.0
90		30,015		2.2		2.2								0.0
25		40,022		2.0		2.2								0.0
250		50,026		2.3		2.3								0.0
70		60,043		2.2		2.3								0.0
-20		70,057		2.0		2.0								0.8
305		80,072		2.2		2.5								0.0
-40		90,077		1.9		2.0								0.0
70		10081		2.5		2.4								0.2

GENERAL CALCULATION SHEET

MAC B830 REV. 4-57

TABLE X — CYCLE LIFE TEST DATA

MTP 0191 — APPENDIX G — X28050 — S/N 002

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
TEMP °F	INLET PRESS PSIG	CUMULATIVE CYCLES		t_o 28vdc 60 PSIG ms		t_c 28vdc 60 PSIG ms								HELIUM LEAKAGE SCCH
72	60	10,009		—		—								0.0
215		20,015		2.5		2.5								0.0
90		30,021		2.3		2.3								0.0
25		40,030		2.3		2.3								0.0
250		50,062		2.4		2.5								0.0
70		60,086		2.4		2.4								0.0
-20		70,112		2.3		2.2								0.9
305		80,131		2.4		2.6								0.0
-40		90,142		2.3		2.2								0.0
70		100,148		2.4		2.5								0.5

X28050 CYCLE LIFE TEST HISTORY

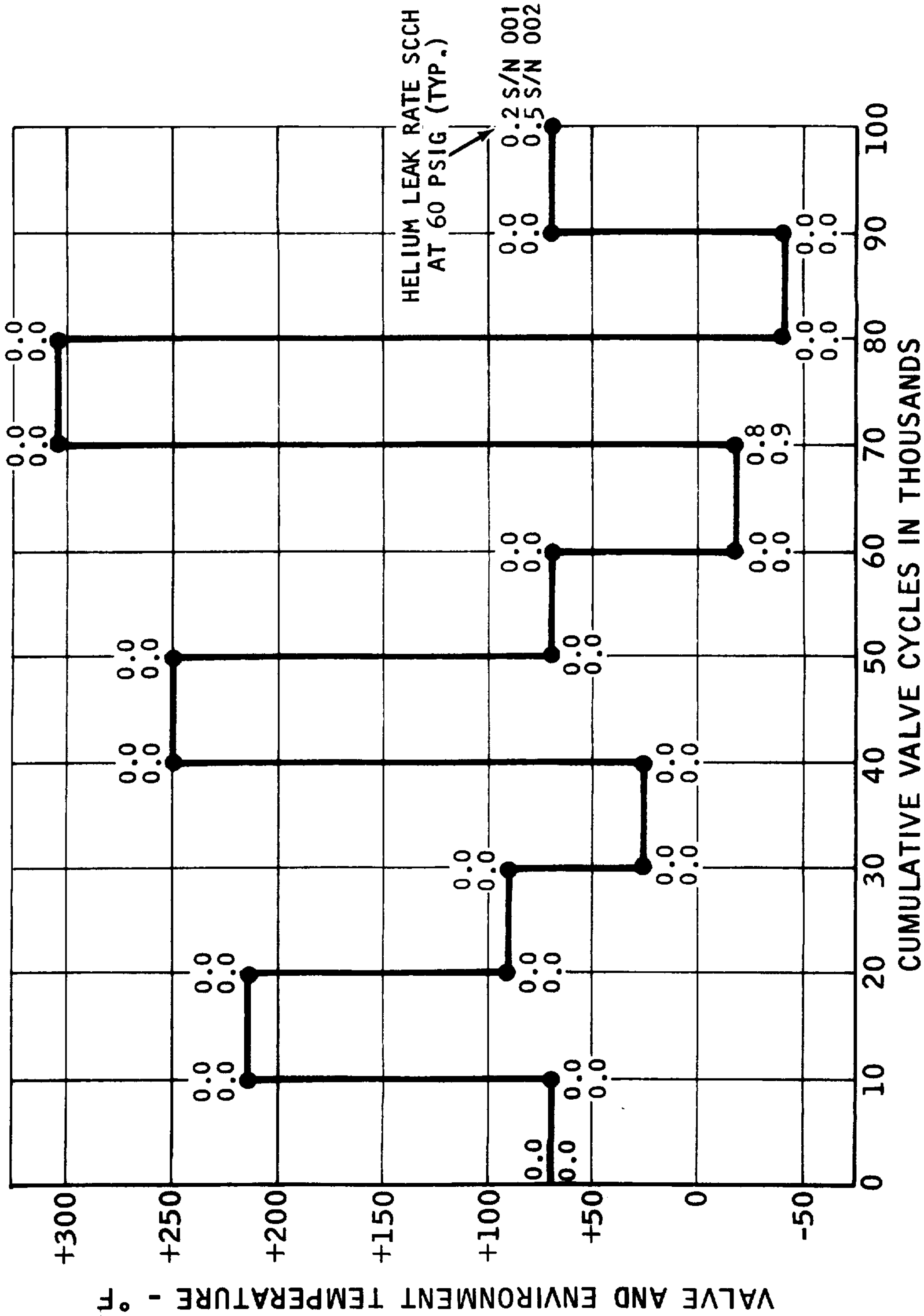


Figure 24

found to be 29.75 ohms (35.68 ohms acceptance test value) but no short to the case was evidenced. The S/N 002 opening coil resistance was measured as 30.03 ohms (35.36 ohms acceptance test value) and resistance to case indicated a short to the case near the middle of the coil (start wire to case resistance = 21.12 ohms, finish wire to case resistance = 10.24 ohms). Since testing to this time had successfully demonstrated capability to meet all design goals, and the observed failure had occurred as a result of an off-limits condition, the test program was terminated. Investigation of the coil failure was initiated, recognizing that rework of the failed coil could be accomplished without jeopardizing the integrity of the valve's internal components and that testing of the valve could be continued once the coil failure was identified and appropriate rework performed.

To obtain access to the coils, the covers were split (axial slots milled) and removed leaving the coils intact (connector, connector boot and the potted leads were removed by cutting the coil leads where they egressed from the basic coil cavity). Since the failure had occurred in only one coil of each valve, the unfailed coil of each valve was manually unwound, and the magnet assembly carefully removed to expose as much of the failed coil, in an intact condition, as possible. As each element was removed, careful visual examinations were made for any potential cause of failure. After removal of the permanent magnet segments, leaving the glass laminate split washers covering the inner face of the failed coil, exposed failed-coil-magnet wire was observed in the gap between the I.D. of the glass laminate split washer and the body (Ref. Figure 25). Under the impetus of the coil winding wire tension required to properly nest the turns, and aggravated by the additional stresses and loads of reversing the lay of the turns in transitioning to the next layer of turns, coil wire occasionally extruded into this gap. After potting and subsequent thermal cycling, the movement, due to differential coefficients of expansion of the respective materials, eventually chafed through the insulating materials until the coil wire made electrical contact with the body (S/N 002 valve) or shorted to another portion of the coil wire, isolating a portion of the coil (S/N 001 valve). In particular, the 400°F exposure of these valves had created sufficient displacement of the coil cavity elements to create the observed condition.

This failure was determined to be assessable to a design deficiency, in that tolerance stack-up allowed sufficient gap to occur for extrusion of the coil wire into the gap, and offered a potential electrical shorting path to the body. In the S/N 001 valve, wherein no short to the body was evident, the extruding wire made contact with the coil start wire which enters the coil cavity radially along this surface. Corrective action, to preclude this type of failure on all subsequent valves, will be;

a) After magnet assembly to body and glass laminate split washer installation, fill the gap between the split washer and body with epoxy resin to form a smooth, totally insulated surface adjacent to the coil windings.

b) Use insulation sleeving over the coil start wire where it radially enters the coil cavity.

VALVE COIL FAILURE ORIGIN

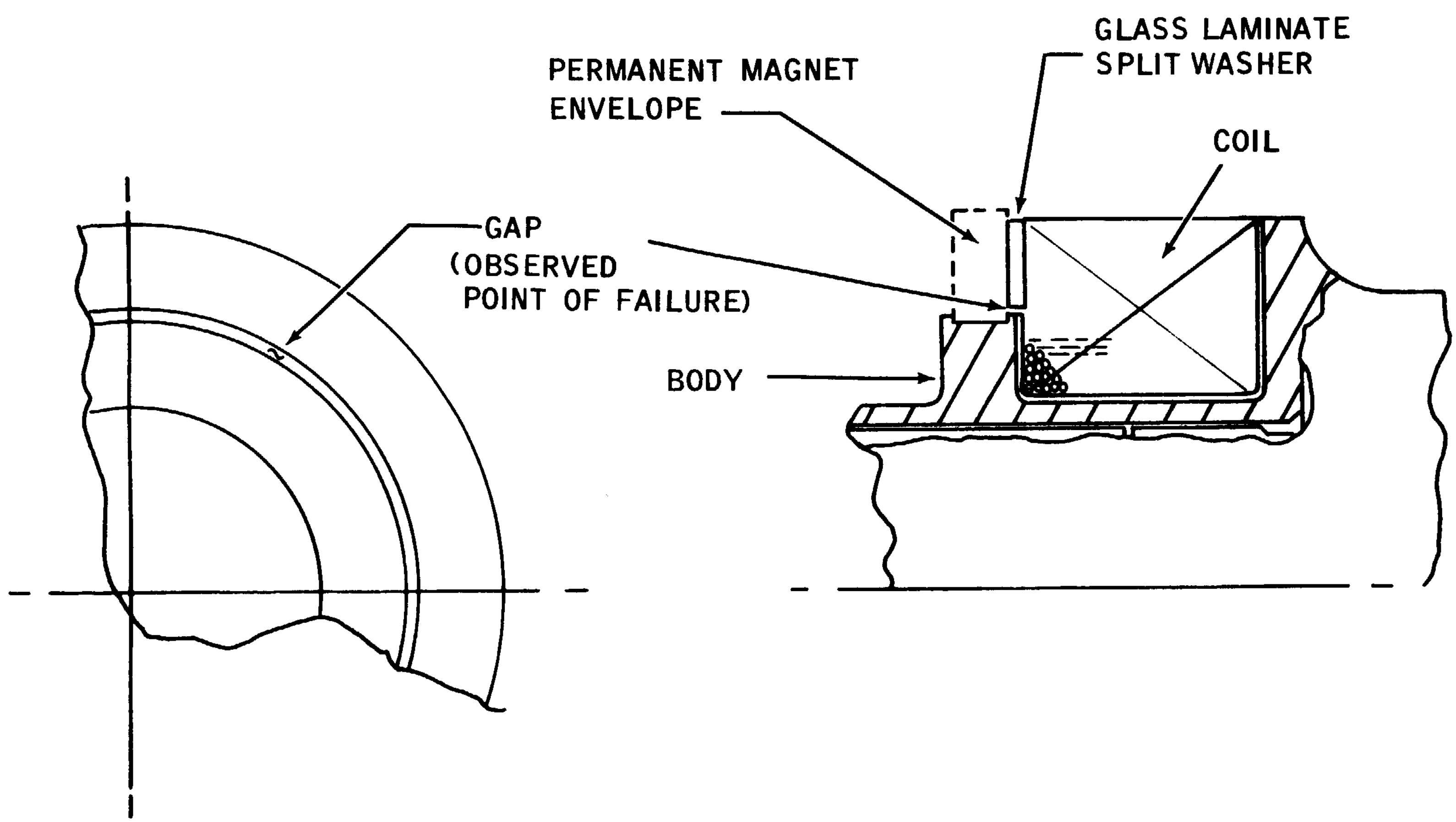


Figure 25

Resistojet Thruster Testing

S/N 003 and 004 X28050 valves were assembled into resistojet thruster assemblies to perform the function of controlling propellant flow to the thruster (Ref. Figures 26, 27 and 28). The completed thruster assemblies were "hot fire" acceptance tested to substantiate performance and integrity, one using H_2 as the propellant, the other using NH_3 propellant. Propellant inlet pressure during these tests was in the range of 25 to 35 psia.

After completion of acceptance testing the thrusters were delivered to Durkee Environmental Laboratories of Torrance, California, for environmental testing. During the following environmental testing, the thruster was not operated and the valve and thruster were not pressurized. Test axes are as shown in Figure 26 and sequence of testing was as presented herein.

a. Pretest Sinusoidal Sweep

With the thruster assembly rigidly mounted to the shaker, the following sine sweep, in each of the three axes, at one octave/minute, was made to define natural frequencies and amplification factors for the various components;

20 to 60 Hz at 0.13 g peak
60 to 235 Hz at 0.00071 inch double amplitude
235 to 2000 Hz at 2.0 g peak

Accelerometers located on the valve attachment clamps provided indications of the amplification factors to the valves. Table XI presents a tabulation of the resultant inputs to the valve from the above inputs to the thruster mounting bracket.

b. Flight Environment Sinusoidal Sweep

The following flight environment sinusoidal spectrum was swept at 3 octaves/ minute:

1.5 to 8.5 Hz at 0.12 inch double amplitude
8.5 to 30 Hz at 0.48 g peak

c. Ground Handling Sinusoidal Sweep

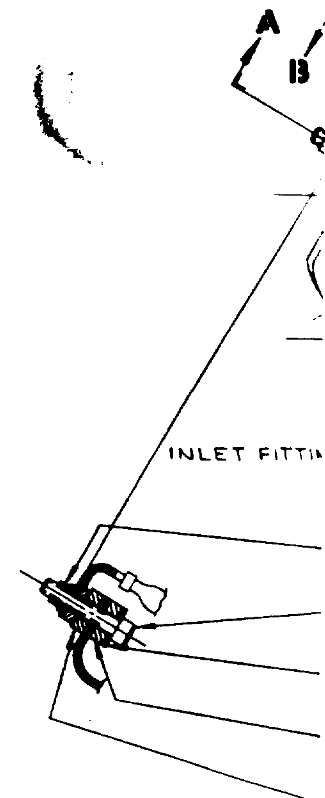
Each thruster assembly was subjected to the following sinusoidal sweep spectrum at a sweep rate of 1 octave/minute:

5.0 to 27.5 Hz at ±1.56 g
27.5 to 53 Hz at 0.043 inch double amplitude
52 to 500 Hz at ±6 g

14 13 12

NOTES:- UNLESS OTHERWISE SPECIFIED

1. ADD CAP (ITEM 20) TO FITTINGS INDICATED WHEN UNIT IS NOT IN USE. TORQUE - FINGER TIGHT.
2. MINOR BEND ADJUSTMENTS ARE PERMISSIBLE TO ALIGN TUBE TO FITTING.
3. ALL TORQUES ARE IN ADDITION TO RUNNING TORQUE. LUBRICATE ALL THREADS WITH KRYTOX 143AC LUBRICANT.
4. TO INSTALL AND REMOVE TERMINAL CLAMP, APPLY PARALLEL JAW PLYERS ACROSS BOTH SURFACES INDICATED ON BOTH SIDES AND SPRING TO A SOLID STOP. DO NOT APPLY PRESSURE BEYOND THE SOLID STOP POSITION.



SECTION A-A

FOLDOUT FRAME

14 13 12

13 12 11 10 9

OTHERWISE SPECIFIED

1 20) TO FITTINGS INDICATED WHEN UNIT

E. TORQUE - FINGER TIGHT.

DJUSTMENTS ARE PERMISSIBLE TO ALIGN

G.

ARE IN ADDITION TO RUNNING TORQUE.

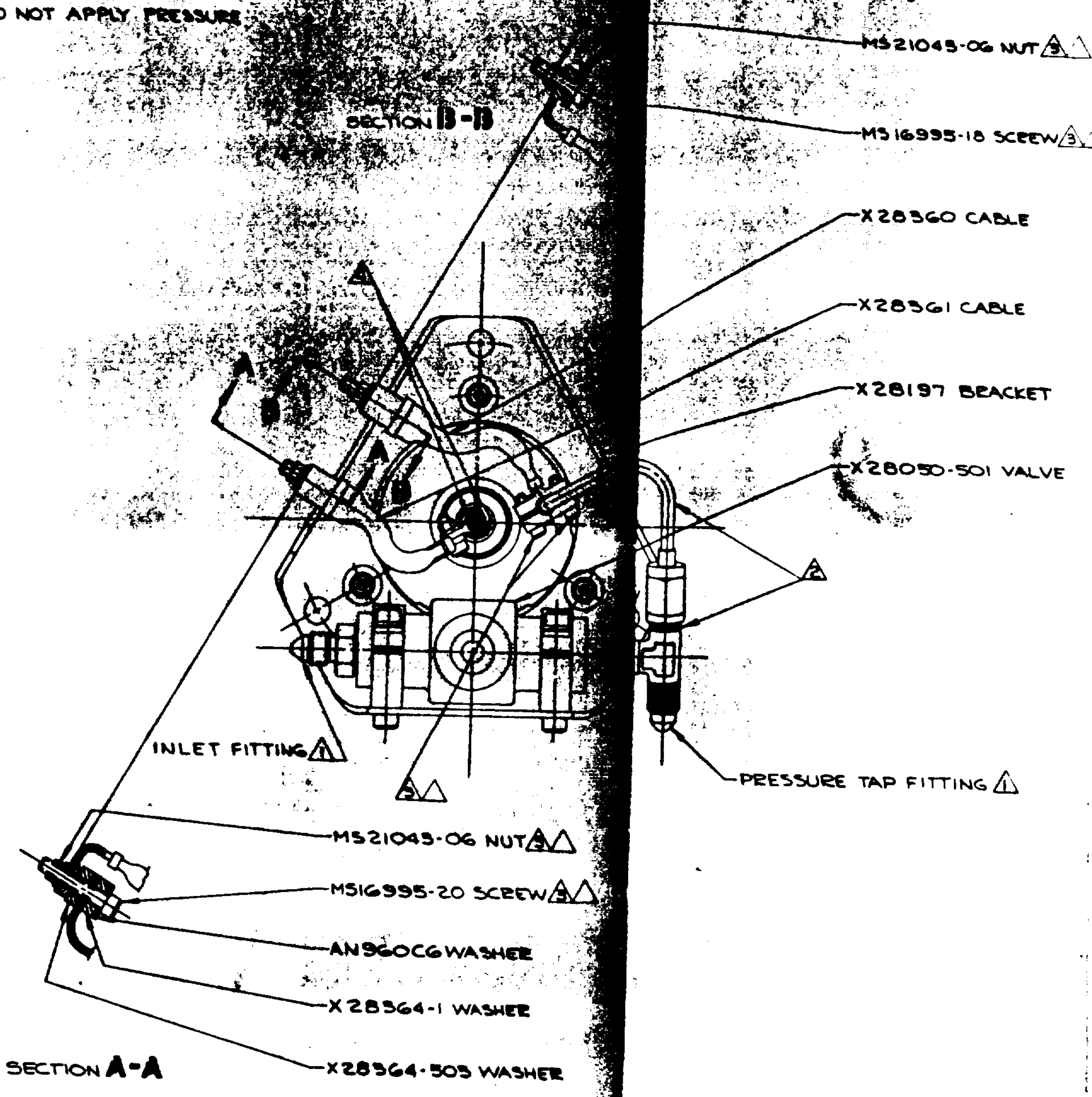
THREADS WITH KRYTOX 143AC LUBRICANT.

O REMOVE TERMINAL CLAMP, APPLY PARALLEL

ACROSS BOTH SURFACES INDICATED ON BOTH

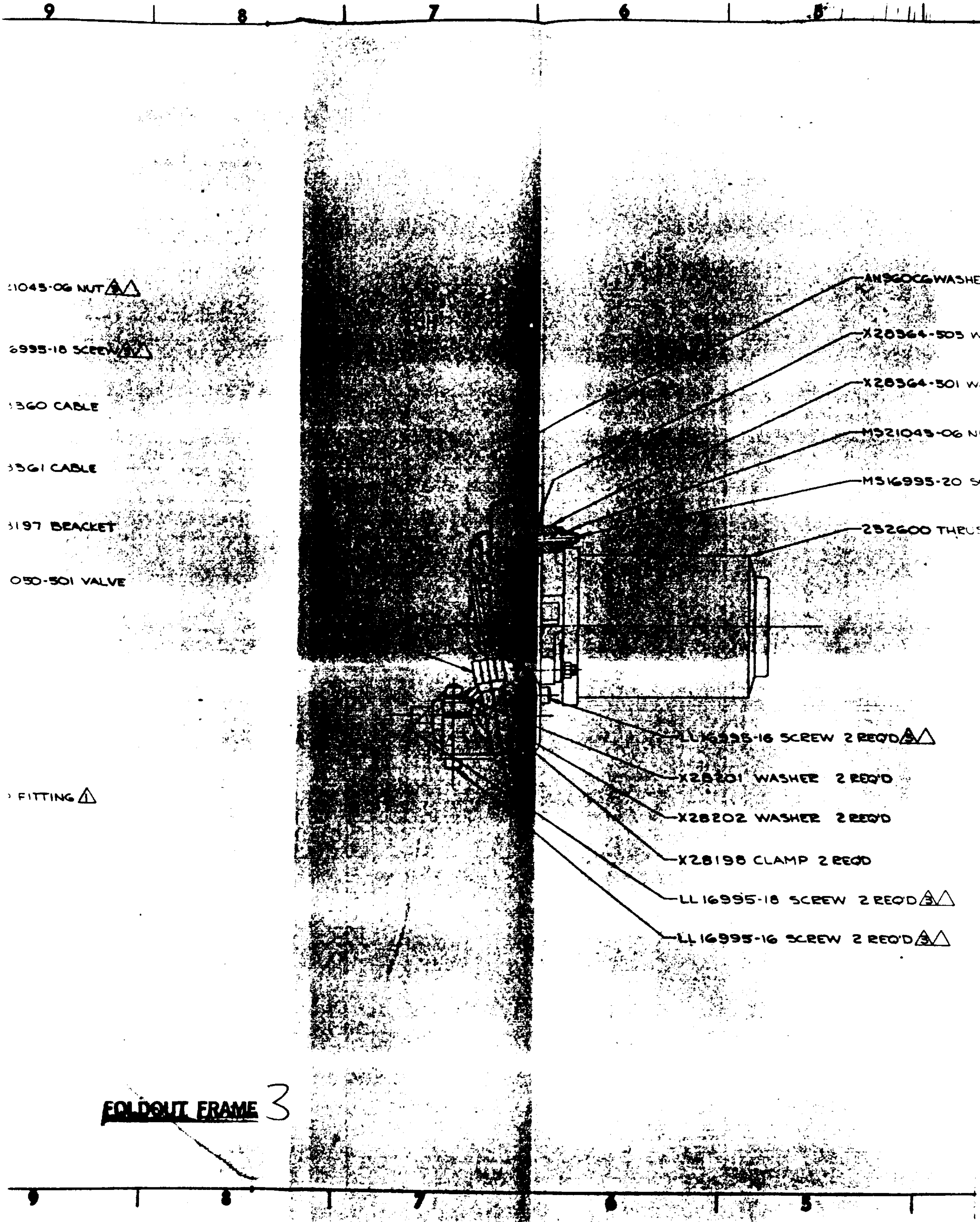
RING TO A SOLID STOP. DO NOT APPLY PRESSURE

OLID STOP POSITION.



FOLDOUT FRAME 2

13 12 11 10 9



FOLDOUT FRAME 3

LTR	DESCRIPTION
A	(1) INITIAL DRAWING ISSUE

QTY		PART NO.	NOMENCLATURE	SPECIFICATION	MATERIAL	ZONE	ITEM
2		LL16995-16	SCREW, KEL-F			[illegible]	25
4		LL16995-16	SCREW, KEL-F			[illegible]	24
							23
							22
							21
2		AN929-2	CAP			[illegible]	20
[illegible]		AN960C6	WASHER			[illegible]	19
[illegible]		[illegible]	NUT			[illegible]	18
[illegible]		[illegible]	SCREW			[illegible]	17
[illegible]		MS16995-16	SCREW			F/5	16
							15
							14
							13
							12
4		X28364-503	WASHER			[illegible]	11
3		X28364-501	WASHER			F/5	10
1		X28364-1	WASHER			[illegible]	9
4		X28361	CABLE			E/5	8
1		X28360	CABLE			F/5	7
2		X28202	WASHER			C/6	6
2		X28201	WASHER			C/6	5
2		X28198	CLAMP			C/6	4
1		X28197	BRACKET			E/5	3
1		X28050-501	VALVE			D/5	2
1		232600	THRUSTOR			E/5	1

LIST OF MATERIALS OR PARTS LIST

X28200 A

RESISTOJET ASSY, MODEL R-110

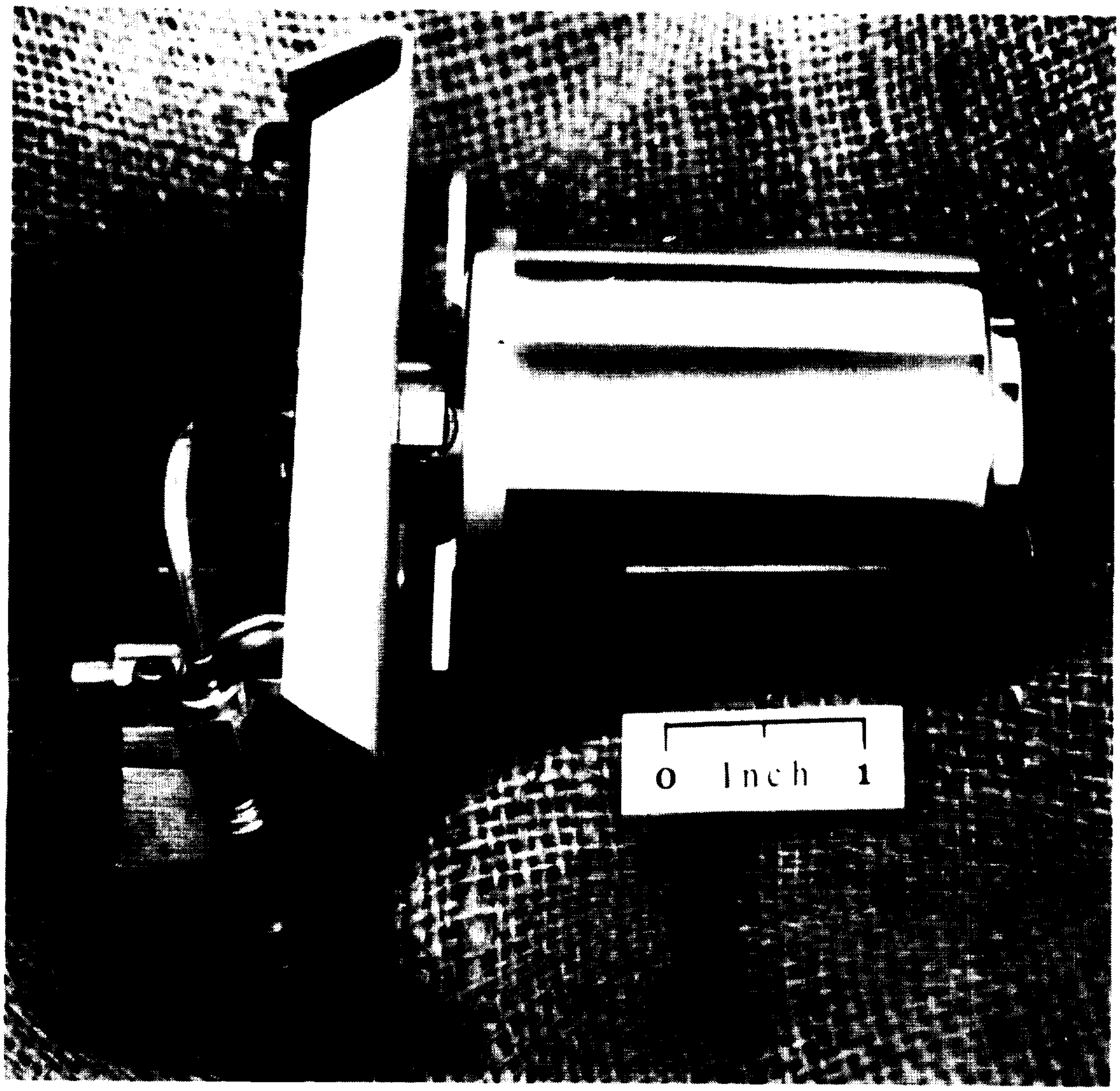
J [illegible] X28200

SCALE 2/1

FOLDOUT FRAME 4

Figure 26

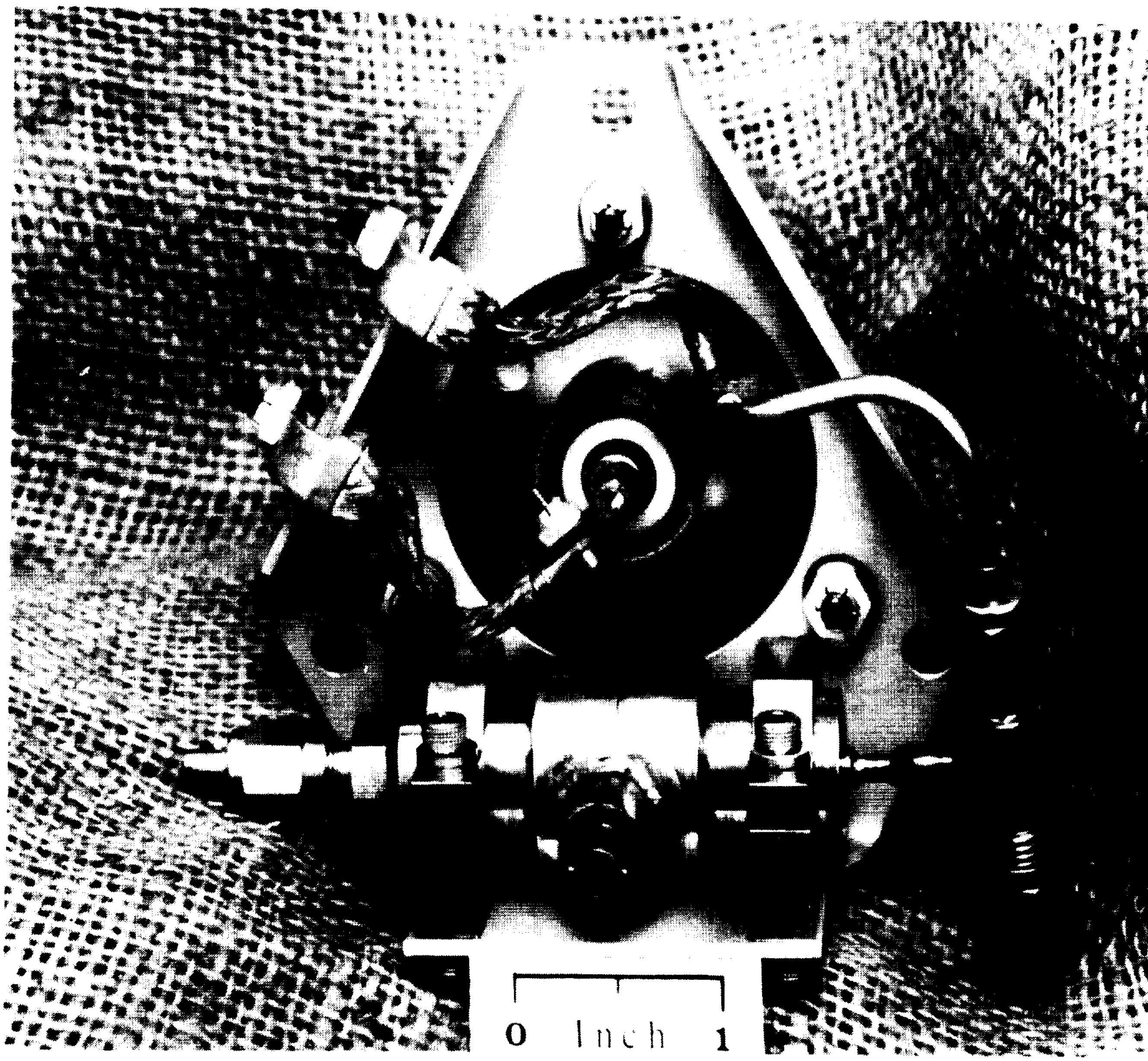
RUGGEDIZED RESISTOJET - MODEL R-110



NEG. 9939-29

Figure 27

RUGGEDIZED RESISTOJET - MODEL R-110



NEG. 9939-30

Figure 28

TABLE XI

PRETEST SINUSOIDAL SWEEP - THRUSTER VALVE INPUTS

Thruster Axis	Valve Axis	Input to Thruster Mounting Bracket	Valve Mounted Accelerometer Values: S/N 001 Thruster Freq. H_z	S/N 001 Thruster $\pm$ g's	S/N 003 Thruster Freq. H_z	S/N 003 Thruster $\pm$ g's
X	Perpendicular to Flow Axis	20 to 60 H_z at 0.13 g	420	4	240	10.0
		60 to 235 H_z at .00071 in. D.A.	720	2	560	5.5
			950	3.5	980	4.5
		235 to 2000 H_z at 2.0 g	1000	4.5	1150 to	
			1080	4.5	2000	3.0
		↓	1375	3.0		
			1750	2.5		
			2000	1.5		
Y	Axis of Flow	↓	550	2	580	3
			630	2.5	780	4
			760	3	1180	10
			1250	4	1680	5
					2000	6.5
Z	Perpendicular to Flow Axis		320	4	260	10
			780	1.5	590	2
			1280	1	820	2.5
			1850	3.5	1230	1.5
					1550	2.5

d. Shock

Each thruster was subjected to one terminal peak sawtooth pulse shock, of 30^{+5}_{-0} g over an 11 ±1 millisecond duration, in each direction on each of the three orthogonal thruster axes.

e. Acceleration

Mounted on the arm of a centrifuge, each thruster was subjected to an acceleration of 8 g for 120 seconds duration in each orthogonal axis.

f. Flight Random Vibration

Both thruster assemblies were exposed to the flight random vibration spectra shown in Figure 29. One minute/axis high level vibration was performed, followed by the 2 minutes/axis low level vibration exposure. During the latter testing of the S/N 003 thruster assembly a structural failure of the valve outlet tube, at the weld joint to the instrumentation fitting, occurred (see Figure 30). This failure is not attributable to the valve since its origin lies in the large unrestrained mass of the instrumentation fitting, welded to the valve outlet tube, installed during thruster assembly build-up.

g. Off-Limits Random Vibration

The S/N 003 thruster assembly was vibrated at the off-limits spectrum of Figure 29. Since the outlet tube to instrumentation fitting structural failure of the previous test existed, the valve was not directly coupled to the unsupported mass of the fitting during this test.

h. Post-Test Sinusoidal Sweep

The S/N 001 thruster assembly was subjected to the following sinusoidal spectrum at a sweep rate of one octave/minute;

20 to 60 Hz at 0.13 g peak
60 to 235 Hz at 0.00071 inch double amplitude
235 to 2000 Hz at 2.0 g peak

No significant shifts in dynamic response characteristics from the pre-test data (a. above) was noted.

RANDOM VIBRATION SPECTRA

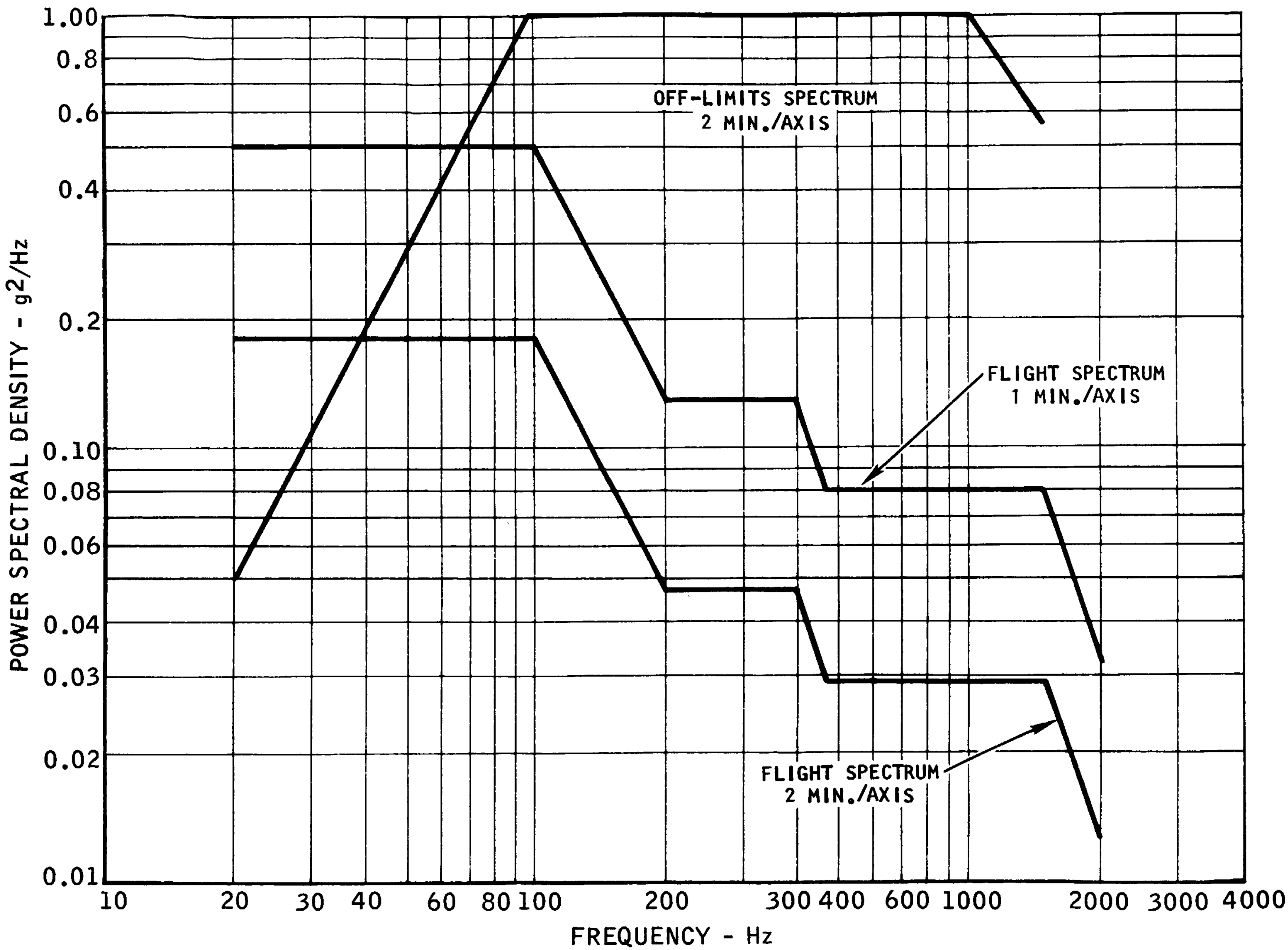
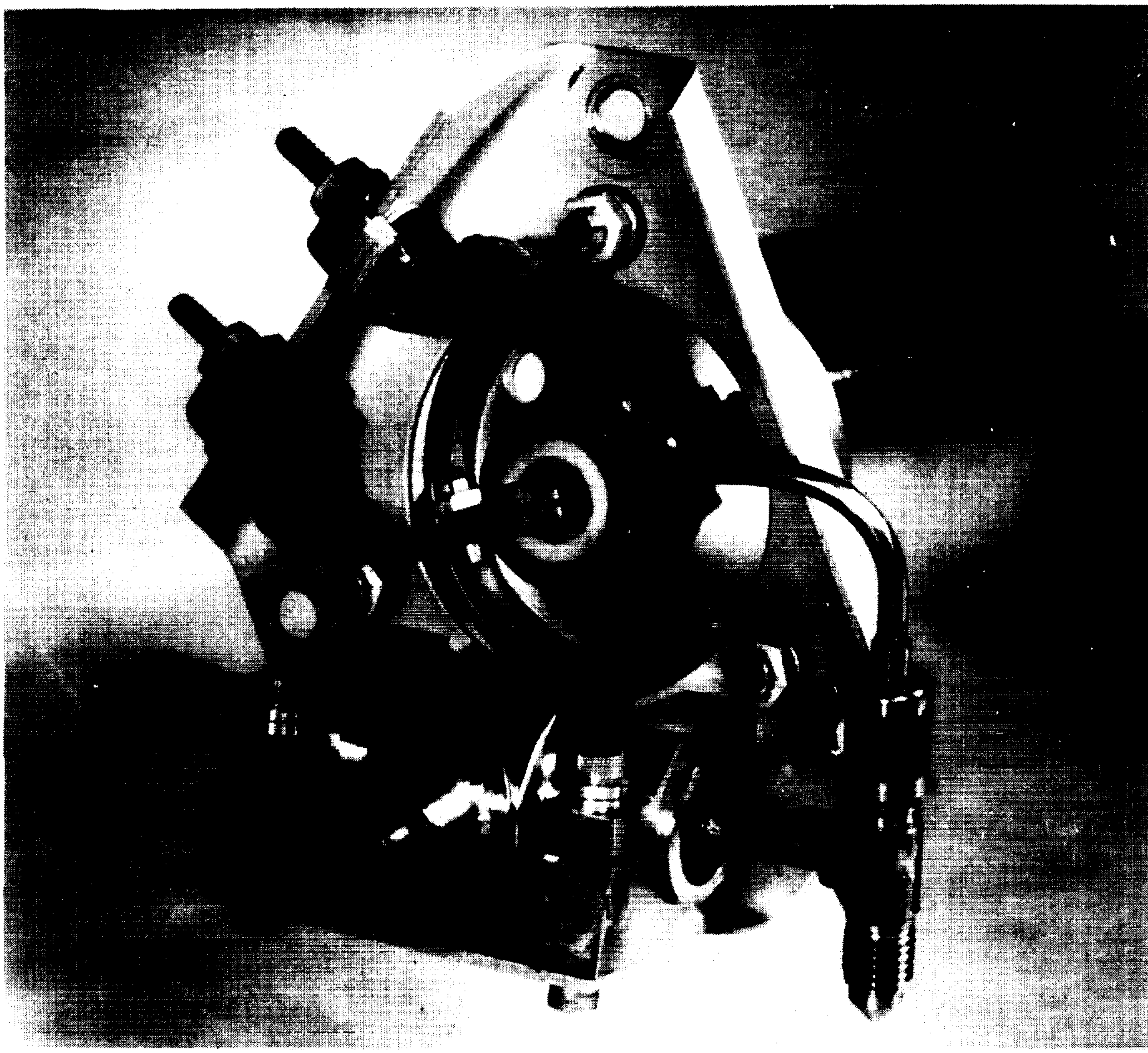


Figure 29

RUGGEDIZED RESISTOJET THRUSTER AFTER DYNAMIC ENVIRONMENT TESTS



NEG. 9939-35

Figure 30

i. Post-Test Evaluation

No evidence of failure of the valve to withstand the environmental loads was evidenced during the testing. After completion of the environmental testing, the valves were helium leak tested at 20 and 100 psig inlet pressures and no evidence of leakage was apparent (0.0 SCCH). Pull-in and drop-out current was measured and found to agree with acceptance test values for the respective parameters within 10%.

Following vibration testing and the post test leakage check, the resistojet thrusters were installed in the Marquardt high vacuum test facility (Building 42) where they were subjected to a 720 hour life test. During this test, the thrusters were in an environment of 10^{-5} Torr to 100 micron with the engines operated for 0.5 hours every 1.0 hour for a total of 720 cycles. During engine operation, the valve outlet achieved a temperature of up to 140°F and the valve mounting bracket achieved a peak temperature of 160°F. The S/N 003 valve controlled hydrogen flow to one thruster while the S/N 004 valve controlled flow of ammonia to the other thruster.

At the completion of the 720 hour life test, both valves were helium leak checked at 50 psig inlet pressure, and exhibited 0.0 SCCH leakage. The valves were then cycled with their respective propellants, and periodically leak checked with 50 psig helium, until each had accumulated a total of 100,000 cycles of operation. A summary of this test history is presented in the following tabulation.

Accumulated Cycles	Measured Helium Leakage	
	S/N 003 (Cycled in H_Z)	S/N 004 (Cycled in NH_3)
Approx. 1,000 (post 720 hour life test)	0.0 SCCH	0.0 SCCH
10,000	0.4	0.15
15,184	0.0	-
20,000	-	0.85
50,000	0.0	0.50
75,000	0.0	0.0
100,000	0.0	0.0

Supplementary Valve Tests

Three valves, P/N X28051, were received at J.P.L. on 27 September 1971. These valves were procured to provide an appropriate latching type valve for use in APS testing and to provide evaluation hardware for determination of latching valve capability in the APS size range. The valves are essentially identical to the valve, Marquardt P/N X28050, that is used on the Resistojet which Marquardt is developing for Langley Research Center.

The planned testing for the three valves was as follows;

S/N 001	N_2H_4 Cycle Test at Pit G (25,000 cycles)
S/N 002	N_2H_4 Exposure Test at ETS (2 years)
S/N 003	Vibration and H_2O Leakage Test
	N_2H_4 System Test in Bldg. 117

The vibration and leakage test (S/N 003) was concluded on 2 November 1971 and the results (see Table XII) were satisfactory. There was no evidence of leakage during the test and valve performance after exposure was unchanged. S/N 003 valve was then installed into the thruster test feed system at Bldg. 117. The hydrazine cycle test (S/N 001) is in progress at Pit G with valve performance satisfactory during and after the first 10,000 of the scheduled 25,000 cycles. S/N 002 is installed in a hydrazine tank-test setup at ETS where performance during and after long-duration exposure to hydrazine, with intermittent operation when samples of hydrazine are removed for analysis, can be evaluated.

The radiated magnetic fields for the energized (open and close) coils and the de-energized (opened and closed) positions were measured in the J.P.L. Magnetic Laboratory. Results of the field mapping are shown in Table XIII. Two factors are evident from this measurement of latching solenoid magnetic fields.

1. The magnitude of the radiated fields in the energized condition can be minimized by reducing the solenoid power level and increasing the efficiency of the solenoid magnetic circuit.

2. The residual magnetic fields after the solenoid power is removed is a function of the amount of "hard" magnetic material in the valve design and the efficiency of the magnetic circuit and shielding.

Considerable design margin in the solenoid coil power requirement (25 watts) and the amount of latching force (two pounds) provided by the permanent magnets was provided in the current valve configuration to meet Resistojet criteria. Both of these characteristics would be optimized in any valve procured for APS flight-type hardware.

TABLE XII

VIBRATION SUMMARY

I. Required Sinusoidal Vibration

Freq. Range (Hz)	Amplitude (g Pk)	Sweep Rate (oct/min)	Notes
5 2000	1.0	Not specified	X, Y & Z Axes.
5 - 30	4.0	2.0	X, Y & Z Axes. Low frequency requirement allowed maximum capability of the exciter
30 - 2000	10.0		

II. Actual Sinusoidal Vibration

Freq. Range (Hz)	Amplitude (g Pk)	Sweep Rate (oct/min)	Notes
5 - 2000	1.0	2.0	X, Y & Z Axes.
5 - 10	0.8" D.A.		
10 - 30	4.0	2.0	X, Y & Z Axes.
30 - 2000	10.0		

III. Required Random Vibration

Freq. Range (Hz)	Power Spectral Density (g^2/Hz)	Wide Band Level (GRMS)	Duration (sec)	Notes
25 - 50	Increasing at 12 dB/oct.			
50 - 600	1.0	27.8	300	X, Y & Z Axes.
600 - 2000	Decreasing at 12 dB/oct			

IV. Actual Random Vibration

Freq. Range (Hz)	Power Spectral Density (g^2/Hz)	Wide Band Level (GRMS)	Duration (sec)	Notes
Same as above				X, Y & Z Axes.

TABLE XIII

RADIATED MAGNETIC FIELD

P/N X28051 LATCHING SOLENOID VALVE

Valve Condition	Maximum Radial Field At 12 Inches (Nanotesla)
Initial De-energized Residual Field	230
Open Coil Energized	1300
Valve Open and De-energized	240
Close Coil Energized	1250
Valve Closed and De-energized	175

CONCLUSIONS AND RECOMMENDATIONS

Conclusions

As a result of the development effort reported herein, the following conclusions have been drawn:

- The valve design has a demonstrated capability to meet all design requirements for the current resistojet thruster application.

- Design margins were demonstrated which substantiate the valve design's capability to operate at pressures from 0 to 400 psia and temperatures from -40°F to +300°F.

- Demonstrated design margins indicate capability to produce the valve with a coil which will draw power of not greater than 8 watts at 70°F and 30 vdc if energized continuously. This capability can be accomplished without altering the present valve envelope and with only a slight increase in valve response time.

- The flow characteristic of the valve makes the design suitable for applications in propellant systems as an isolation valve as well as an injector valve.

- Design principles demonstrated are:

  * Magnetic bi-stable actuator

  * Moving element flexure guidance

  * Spring loaded floating soft seal

  * AF-E-102 ethylene propylene terpolymer seat seal

- Based on the number of valve cycles demonstrated by the seat design with essentially liquid leak-tight sealing, the calculated mean cycles between failure for the design is 3.184×10^6 cycles.

- The valve coil failure during off-limits testing was a design deficiency of the development configuration. Corrective action to alleviate this failure mode will not negate the significance of the capability demonstrated.

- The valve configuration can be readily adapted to a metal-to-metal poppet/seat seal.

- The design has demonstrated capability to withstand the dynamic environments of most current and anticipated spacecraft propulsion application.

Recommendations

Based upon the level of maturation of the design, demonstrated by the successful completion of the tasks described herein, the following recommendations are proposed:

- Incorporate the valve into the biowaste thruster and thruster system for evaluation testing.

- Conduct long term tests under hard vacuum conditions, with usage propellants.

- Determine potential applications of the valve and variations of the basic design such that benefits of commonality can be realized.

- Evaluate hard seat configuration of valve.

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