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ACTUATION AND SYSTEM DESIGN AND EVALUATION

OMS ENGINE SHUTOFF VALVE

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VOLUME I

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LYNDON B. JOHNSON SPACE CENTER
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ABSTRACT

A technology program was conducted to identify and verify the optimum valve and actuation system concept for the Space Shuttle Orbit Maneuvering System (OMS) engine. Of major importance to the valve and actuation system selection was the ten-year, 100-mission, 10,000-cycle life requirement, while maintaining high reliability, low leakage, and low weight. Valve and actuation system concepts were comparatively evaluated against past valve failure reports and potential failure modes due to the Shuttle mission profile to aid in the selection of the most optimum concept for design, manufacture and verification testing. Two valve concepts were considered during the preliminary design stage; i.e., moving seat and lifting ball. The lifting ball valve concept was manufactured and tested to verify the operational characteristics. Two actuation systems were manufactured and tested; i.e., a pneumatic system and an ac motor drive system.

Test results demonstrated the viability of the lifting ball concept as well as the applicability of the ac motor actuation system to best meet the requirements of the Shuttle mission.

TABLE OF CONTENTS

TABLE OF CONTENTS (continued)

TABLE OF CONTENTS (continued)

LIST OF ILLUSTRATIONS

LIST OF ILLUSTRATIONS (continued)

LIST OF ILLUSTRATIONS (continued)

LIST OF TABLES

LIST OF TABLES (continued)

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FOREWORD

This final report is submitted by the Parker-Hannifin Corporation in accordance with the requirements of the NASA technology program contract NAS9-13442. The technology program was administered by the NASA Lyndon B. Johnson Space Center at Houston, Texas with Mr. J. Fries, the NASA Technical Project Monitor.

The program work was conducted at the Parker-Hannifin Corporation at Irvine, California and the Project Manager was Mr. V. B. Dunn assisted by Messrs. J. Presas, M. Cirilo, W. Johnson, and H. Lamb.

1.0 SUMMARY

The technology program conducted by Parker-Hannifin to identify and empirically verify a new valve and actuation system concept for the Space Shuttle Orbital Maneuvering System (OMS) 6,000-pound thrust rocket engine is reported herein. Although rubbing seal bi-propellant valves were used in the single-mission Apollo program, there was general agreement in NASA and the valve industry that these units would not be suitable for the 100-mission, 10,000-cycle, ten-year life design goal of the Shuttle. The program has resulted in the identification of a new valve and actuation system concept that will meet the design goal requirements for the Shuttle system.

The valve and actuation system concept that was developed provides a maximum configuration flexibility; i.e., series versus parallel, due to the planned modularity of the designed elements. The system consists of a valve, an a.c. motor, planetary gear train, electronic control module, linkage arrangement, and fail-safe feature. The valve is a ball-type with a unique operating action that lifts the ball straight off the seat, preventing seal rubbing and wearout. The valve action has resulted in the valve-type being referred to as a "lifting ball" valve. The valve seat and ball element demonstrated 20,000 cycles operation with no significant leakage. The a.c. motor is controlled by the electronic control module that provides the system logic framework for the valve assembly and enabled the a.c. motor to be operated from a d.c. input signal. This contributes to long life and improved reliability by eliminating the need for the sliding commutator brushes found in conventional d.c. motors. The a.c. motor also avoids the inherent propellant decontamination problems of the Apollo fuel-actuated valves, while providing the ease of maintainability essential for the long-life requirement of the Shuttle. The a.c. motor-operated actuation system was selected in lieu of a dedicated pneumatic system due to the weight advantage of the motor system. The system is so designed that the valve will drive to the closed position in the event of a power failure and is designed such that sufficient reliability provides confidence in the mission success capability of the assembly.

To provide this valve and actuation system concept, a series of tradeoff studies was conducted to assure that the most optimum system be selected. Design studies were performed to establish the most optimum detail design concepts for the system selected.

A prototype system was then manufactured and tested.

Parker-Hannifin is proud of the work accomplished on this program because, not only were systems identified and developed specifically for the Shuttle application, but these same concepts will be applicable to most future long-life valve requirements.

2.0 INTRODUCTION

The OMS engine valve and actuation system technology program described in this report was performed in support of the Space Shuttle Program. Due to stringent life requirements for the Space Shuttle hardware, technology programs were conducted to develop valve and actuation systems that will best meet these system requirements. The Space Shuttle vehicle is being designed to provide transportation to earth orbit to support a variety of missions. The Space Shuttle is being designed for 100 flights over a 10-year operational lifetime. The system is being designed to minimize post-flight refurbishment, maintenance, and checkout. For translational maneuvers, the Space Shuttle will employ two rocket propulsion systems. The valve and actuation system technology programs are being conducted to provide fuel and oxidizer shutoff valves for the OMS engine assembly.

Due to deficiencies in other spacecraft engine valves, during previous space programs, technology studies were appropriate to develop and identify long-life multi-usable concepts for the Space Shuttle Program. Major problems that had to be solved were long-term propellant-to-material compatibility, long cycle life, leakage, contamination sensitivity, marginal system reliabilities, and long term maintenance characteristics.

The technology program conducted at the Parker-Hannifin Corporation consisted of the following program tasks:

TASK I	ANALYSIS AND PRELIMINARY DESIGN
I-A	Review Valve Design Criteria with NASA
I-B	Conduct Survey to Determine Past Valve Problems
I-C	List Other Potential Valve Problems
I-D	Analyses and Preliminary Design Layout Drawings
I-E	Presentation of Task I Results and Recommended Design Concept to NASA.

TASK II	VALVE — DETAILED DESIGN
II-A	Conduct Optimization Studies
II-B	Complete Flight Weight Design Layout
II-C	Complete Prototype Design Layout
II-D	Preliminary Design Review
II-E	Design and Analysis (Lifting Ball Valve; (LBV))
II-F	Preliminary Design Review (LBV)
II-G	Detail Drawing Preparation (LBV)
II-H	Critical Design Review (LBV)
TASK III	PROTOTYPE VALVE FABRICATION
TASK IV	PROTOTYPE VALVE TESTING
IV-A	Prepare Prototype Test Plan
IV-B	Conduct Prototype Tests
IV-C	Prepare Test Report
TASK V	ACTUATION SYSTEM — DETAILED DESIGN
V-A	Design and Analysis
V-B	Preliminary Design Review
V-C	Prepare Detail Drawings
V-D	Critical Design Review
TASK VI	PROTOTYPE ACTUATION SYSTEM FABRICATION

TASK VII	PROTOTYPE VALVE AND ACTUATION SYSTEM TEST
VII-A	Prepare Prototype Test Plan
VII-B	Conduct Prototype Tests
TASK VIII	FLIGHT WEIGHT DESIGN UPDATE
TASK IX	DESIGN STUDIES/CONCEPT RE-EVALUATION
TASK X	ACTUATION SYSTEM SPARE PARTS SUPPORT
TASK XI	CONTAMINATION SUSCEPTIBILITY TESTING
TASK XII	VALVE EXTENDED PROPELLANT EXPOSURE TESTS
TASK XIII	ELASTOMERIC SEAT DESIGN STUDY
TASK XIV	REPORTS

The above tasks as listed, constituted a basic program plan with which to meet the requirements of the technology program contract. Throughout the program, emphasis was shifted from one task to another dependent upon program results. In some cases, tasks were cancelled prior to completion to divert the resources into more important areas.

During August 1973 the program, which had been completed through Task I and most of Task II, was redirected. The valve concept referred to as the "Moving Seat" poppet valve was discontinued due to the development risk of the design concept and the fact that preliminary analysis indicated the "lifting ball valve" had the potential of being significantly smaller, lower weight, and having a smaller pressure drop.

TASK I — The purpose of Task I was the determination of the most optimum valve and actuation system concept. Parker-Hannifin continually reviewed the OMS engine design criteria with NASA and potential OMS engine suppliers to assure all design considerations were made. A survey was conducted to identify failure modes, high risk areas, and operational problems in previously manufactured shutoff valve and actuation systems, principally those used in large earth-storable propellant systems. Additionally, LM Ascent, LM Descent, and Apollo Service Propulsion Module engine suppliers were contacted to more completely define previous failures and problems of large shutoff valves. Based on Parker-Hannifin's extensive valve and systems experience, a compilation of potential problem areas was prepared and combined with the survey data to provide a comprehensive definition of problem areas to be considered in the selection of the OMS valve and actuation system. Design approaches were developed or identified to resolve or minimize all the potential problem areas. Based upon the program technical guidelines and the results of the problem area study, Parker-Hannifin prepared analyses and preliminary design layout drawings of the candidate valve and actuation system concepts in both quad- and series-redundant configurations. Valve and actuation system concepts were analyzed and designed to provide installation requirements (weight, envelope, and electrical power), for equivalent performance (pressure drop, response time, leakage, and cycle life) and maintainability features. Relative particulate contamination tolerances, decontamination capability, reliability, failure modes, and propellant compatibility were also judged. The weight, envelope, and electrical power requirements for pneumatic operation were also approximated for comparison to the motor actuation systems.

TASKS II and V — During Task II and Task V, Parker-Hannifin conducted parametric studies on performance to optimize valve and actuation system configuration that was selected during Task I. The valve and actuation system was mathematically modeled and both static and dynamic performance studied. Additional design layout work was completed to determine the impact of "level of maintenance" on valve and actuation system weight. Alternate packaging arrangements were studied to achieve the lowest possible level of maintenance at the lowest weight and overall size. Special emphasis was placed on making the filters easily maintainable. In addition, effort was expended on maximizing commonality of maintainable subassemblies. During this task, a trade study of filter weight and size as a function of frequency of maintenance was conducted. A complete flight weight design layout, including assembly details, assembly methods, materials, and finishes was prepared. An installation drawing showing external envelope and all mechanical, electrical, and fluid interfaces was prepared. Also included in this task was the preparation of a prototype unit design layout of a single pair of mechanically-linked shutoff valves operated by a motor actuation system. Block-model shutoff valve construction, using breadboard electronics, standard RVDT's, and block-model motor and gear train were developed to facilitate manufacture and testing in a cost-effective manner. Preparation of detail drawings from which the valve and actuation system was manufactured were prepared during this task.

TASKS III and VI — Task III and Task VI were used to fabricate and assemble two prototype valves and actuation systems and to provide some select spares. Tooling was also fabricated under this task. A third prototype valve referred to as the alternate lifting ball valve was also fabricated under this task.

TASKS IV and VII — All test procedures as well as conducting the tests were accomplished under Task IV and Task VII. Test procedures were prepared for each specific test and these procedures are included in Appendix A. Refer to paragraph 8.2 for a list of all test procedures.

TASK VIII through TASK XIV — These tasks were only partially completed, then terminated due to program developments.

The subsequent sections of this report present the program technical requirements, design criteria, analysis and preliminary design, detail design, test section, and recommendations.

3.0 TECHNICAL REQUIREMENTS

3.1 General

This section consists of the program design requirements as specified in the NASA Statement of Work and also of the results of a Supplier Interface Plan.

The Program Design Requirements are presented in Table III-1. A copy of the technical requirements, as removed directly from the NASA Statement of Work, are included in Appendix A.

The Supplier Interface Plan was a program task conducted to assure that the valve and actuation system design would incorporate the requirements of all potential OMS engine suppliers. Table III-2 is a summary of the major technical interface information received from the potential engine suppliers.

Table III-1. Design Criteria

Parameter	Design Criteria
Compatibility	
1. Fluids	N_2O_4, MMH, 50-50 as liquids & vapors; H_2O at outlets; freon TF
Performance	
2. Pressures	
Nominal	205 psia N_2O_4, 208 psia MMH
Operating Range	172 to 265 psia
Max Surge	400 psia
Proof	400 psia
Burst	670 psia
Max Start	313 psia
3. Flow Rates	11.91 lb/sec N_2O_4
	7.22 lb/sec MMH
4. Pressure Drop	5 psid max (normal)
	"balanced" (fail close)
5. Response Time	100 - 1000 ms open
	100 - 1000 ms close
6. Response Repeatability	Important
7. Propellant Simultaneity	Design for simultaneous propellant delivery
8. Internal Leakage	10 scch GHe per seat (0 to 265 psid)
9. External Leakage	1.66×10^{-7} sccs GHe per joint
10. Electrical Supply	24 to 30.5 vdc (27.25 vdc nom)
11. Electrical Power Limit	To be determined

Table III-1. (Continued)

Parameter	Design Criteria
Life	
12. Cycles	4000 wet/pressurized, 6000 dry
13. Missions	500 missions
14. Time	10 years
15. Propellant Throughput	34,230 pounds per mission
Environmental	
16. Temperature	
Propellant	40 to 125°F
OMS Structure	40 to 120°F
Engine Soakback	200°F maximum
Transport/Storage	-55°F to +190°F
17. Random Vibration	20 to 2000 Hz, 15.3 g's rms, 231 hours
18. Shock	1.5 g maximum for 2.60 ms
19. Acceleration	Up to 4 g's
Maintainability	
20. General	Easily maintainable
21. Accessibility	To be determined
22. Filter Replacement	To be determined
Checkout	
23. General	Minimize valve actuations
24. Position Indication	Open and closed positions
Decontamination	
25. General	Easy to decontaminate
26. Fluid	Hot GN_2 purge

Table III-1. (Continued)

Parameter	Design Criteria		
Contamination			
27. Self generated	Minimize		
28. Propellant	0 - 25	not defined	
	25 - 50	1000 part.	500 ml sample
	50 - 100	100 part.	
	100 - 250	10 part.	
	250	0	
29. Filter Rating	Consistent with valve tolerance		
Construction			
30. Lubricants	Avoid if possible		
31. Dribble Volume	Not critical		
32. Failure Position	Close with loss of power		
33. Gas Pressure Source	Must be included in valve if used		
34. Motors	Brush type not allowed		
35. Force Margin	To be determined		
Installation			
36. Envelope	Minimize		
37. Mounting Provisions	On side of engine		
38. Porting	Parallel or counterflow		
39. Port Size	To be determined		
Weight			
40. General	Minimize		
Duty Cycle			
41. Maximum on-time	870 seconds		
42. Actuations per mission	20 maximum		

Table III-2. Summary - OME Suppli

Design Parameter	Technology Program Design Criteria Resulting from OME Supplier Interface	ALRC	
1. Soakback temperature	200°F Maximum	Valve Temperature TBD (330°F Chamber Ext)	<200°F
2. Engine Induced Pressure Spikes (During Engine Start)	400 psia maximum	<330 psia (<94 psi above S.S. Inlet)	<354 psi (<1.5 x
3. Response Repeatability	Strive for Repeatability; Predict Repeatability	≈± 10% (Including Valve to Valve, Voltage, Temperature and Pressure Variations)	≈± 10% (Includin at nomin
4. Propellant Lead (if any) (During Engine Start)	Design for Simul. Operation w/lead-lag if Possible	Up to 40% MMH Lead (Capability Desired)	Simultan
5. Vibration Amplification and/ or Attenuation	OMS Pod Spec Level (15.3 grms for 77 hr/axis)	TBD	Use OM
6. Weight Target	Minimum Consistent with Maintenance Objective	36 pounds	
7. Mounting Provisions	Provide Mounting Provisions on One Side of Valve	On Side of Engine	On Side
8. Porting	Parallel or Counterflow as Convenient for Valve	Layout Shows Counterflow	Most L Some S
9. Envelope	Minimize	Fit Within 28" O.D./13" I.D. ≈7" Along Thrust Axis (From Aerojet OME Drawing)	8" x 8. Valve C
10. Access for Maintenance	To be Established, Avoid Need for Access on Engine Mounting Side	TBD	

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ary - OME Supplier Interface Plan

Potential OME Supplier Interface Information			
	BAC	Rocketdyne	TRW
	<200°F	<130°F (Less than 10°F Above Propellant)	<200°F
	<354 psia (<1.5 x S.S. Inlet)	<316 psia (60 to 80 psi above S.S. Inlet)	418 psia (estimated) (Oxidizer Side)
tage,	≈± 10% (Including Valve to Valve Variations at nominal Test Conditions)	TBD	± 10 to 15%, or More, (Including Valve-to-Valve and Voltage Variations)
	Simultaneous Operation Desired	Simultaneous Operation Desired	May Desire 100 ms N_2O_4 Lead
	Use OMS Pod Spec Level	TBD	Use OMS Pod Spec Level (High Frequencies Attenuated)
	TBD	Minimize	32 pounds
	On Side of Engine	On Side of Engine	On Side of Engine
	Most Layouts Show Counterflow, Some Show Parallel Flow	Counterflow may be Desired	Layout Shows Parallel Flow
	8" x 8.5" x 10.88" (From Bell Valve Outline Drawing)	Minimize	Fit Within 28" O.D./15" I.D. ≈10" Along Thrust Axis (From TRW OME Drawing)
	TBD	Suggests we Work With MDAC and RI/SD	TBD

Table III-2

4.0 CONCEPT EVALUATION AND SELECTION

The object of this task was to generate a number of valve and actuation system concepts which meet the technical requirements described in Section 3.0 and to compare the candidate concepts and categorize them. This was accomplished by performing a preliminary valve tradeoff study considering valve concepts and actuation system concepts. The valve and actuation system concepts were rated for capability to avoid past problems, capability to provide reliability and long service life, and capability to satisfy the required design criteria. To ensure that the valve concepts considered during the tradeoff study would not repeat past valve weaknesses, two hundred and six failure reports of APS, DPS, and SPS engine propellant valves were reviewed to identify failure modes and frequency of failures. Table IV-1 presents a summary of the failure review. The table lists the failure mode, the valve system identification (APS, DPS or SPS), the number of failures, percentage of total failures, and the cause of the failures.

In addition to the possibility of recurrence of problems that have occurred in similar past valve applications, additional potential valve problems that may be caused by Shuttle long life and reusability criteria, as well as problems in meeting other OMS engine application design criteria were considered. Table IV-2 summarizes these three basic types of problem considerations. Additional potential problems which were a consideration are listed in Table IV-3.

Table IV-1. APS, DPS, and SPS Quad Valve Failu

Failure		APS				
		No.	%	Causes	No.	%
(Total Failures Reviewed)		47	-	-	128	-
Sliding Seals	Ball Seal Leakage	15	32%	Contamination, Wear, Scratches, Salting, Corrosion	61	48%
	Piston and Shaft Seal Leakage				24	19%
Poppet Seals	Pilot Valve Leakage	2	4%	Contamination	18	14%
	Erroneous Position Switch Output	8	17%	Solder Joints, Environ. Cond.	8	7%
	Low Electrical Resistances	2	4%	Damaged Wire, Faulty Diode	3	2%
	Sluggish Operation	4	9%		4	3%
	Hang-up	3	6%	Leaking Oxidizer Rusted Needle Bearings	4	3%
	Filter Collapse	6	13%	Inadequate Support	0	0
	Disconnect Leakage	-	-	-	-	-
	Miscellaneous	7	15%	-	6	5%

Quad Valve Failure Report Review Summary

DPS			SPS			Total	
No.	%	Causes	No.	%	Causes	No.	%
128	-	-	31	-	-	206	-
61	48%	Contamination, Wear, Scratches	4	13%	Contamination		
24	19%	Teflon Flaking	5	16%	Galling, Seal Shrinkage	109	53%
18	14%	Contamination, Motion of Solenoid	6	19%	Contamination, Assembly Error	26	13%
8	7%	Adjustment Sensitivity	0	0	-	16	8%
3	2%	Propellant Fumes, etc. (Sealing Problems)	5	16%	Moisture, Dirt, (Sealing Problems)	10	5%
4	3%		0	0	-	8	4%
4	3%	Leaking Oxidizer Reacts with Gear Lubricant	0	0	-	7	4%
0	0	-	0	0	-	6	3%
-	-	-	4	13%	Seal Handling Damage	4	2%
6	5%	-	7	22%	-	20	10%

Table IV-1

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Table IV-2. Past and Potential Valve Problems

Types of Problems	Problem, or Cause of Potential Problem
① Past Problems of the APS, DPS, and SPS	A. Sliding seal leakage (53% past failures) B. Teflon poppet leakage (13% past failures) C. Erroneous position switch signal (8% past failures) D. Low electrical resistances (5% past failures) E. Sluggish operation and hangup (8% past failures) F. Filter collapse (3% past failures) G. Disconnect Leakage (2% past failures)
② Problems will be caused by long life criteria	H. High cycle life (10,000 cycles) I. Long vibration time (231 hours at 15.3 grms) J. Long life requirement (10 years) K. Large number of missions (100) L. Avoidance of Liquid Flushing (GN_2) M. Ease of maintenance N. Ease of pre-flight checkout
③ There are problems meeting some of the other design criteria	See Table III-1 — Design Criteria

Table IV-3. Problems Meeting Other Design Criteria

Table III-1 Item No.	Design Criteria*	Potential Problems
3	Pressure Drop	There will be size and weight penalty to reduce the fail-closed quad valve ΔP below the present design value
5	Response Time	Closing response time as fast at 100 ms not feasible; 400 ms appears to be the practical minimum closing response
		The practical minimum opening response time is determined by the available power
6	Repeatability	The degree of closing repeatability achievable is of concern with the selected actuator concept. More design analysis is required
7	Propellant Simultaneity	Propellant lead is not possible with the existing design; design modification is needed to have propellant lead capability
16	Temperature	Thermal distortion of the drive train is a potential problem and requires more analysis
22	Maintenance	Accessibility for maintenance must be investigated to gain full maintenance flexibility
30	Lubricants	A considerable load-carrying penalty exists for non-lubricated gear teeth. The design goal to eliminate lubricants should be carefully reviewed
34	Envelope	Parker-Hannifin is concerned about the envelope of the present design. Coordination with MDAC and engine suppliers is required

*See Table III-1 for more complete design criteria

4.1 System Tradeoff Studies

The detail valve and actuation system trade study was accomplished in the following manner. A point rating figure of merit system was used and consisted of the data as presented in Table IV-1.

For "Capability to Avoid Past Problems"

A rating of 0	Will not avoid the problem
A rating of 1	Reduces severity of problem
A rating of 2	May virtually eliminate problem

For "Capability to Provide Reusability and Long Service Life Needed for 10-year/100-mission Shuttle Life"

and

"Capability to Meet Other Design Criteria"

A rating of 0	Does not appear to be a satisfactory concept
A rating of 1	Questionable capability to completely satisfy criteria/may have significant development test
A rating of 2	No obvious deficiencies/low development risk

Table IV-4 presents a matrix of thirteen shutoff valve types with the appropriate rating system for each evaluation criteria. The moving seat concept, using an elastomer seat, received the highest average rating (1.71).

Table IV-5 presents a matrix of nine actuation systems with the appropriate rating system for each evaluation criteria. The hermetically sealed motor actuation system concept, as applied to actuating the moving seat concept shutoff valve, received the highest average rating (1.74).

Table IV-6 presents a matrix of how the selected valve and actuation concept avoids or minimizes past problem areas. Table IV-7 presents an assessment of potential problem areas caused by long-life criteria versus the selected concept.

Table IV-4. Propellant Shuto

Percent of Past Failures	Shutoff Valve Type →	Sliding Seal Ball Valve (Using Bearing)	Retracting Seal Ball Valve (Using Bearing)		Sliding Seal Gate Valve (No Bearing)	Butte (Usin
	Seat Material →	Teflon Seat	Teflon Seat	Elastomer Seat	Teflon Seat	Tef
	<u>"Capability to Avoid Past Problems"</u>					
35%	1. Seat leakage caused by surface deterioration due to sliding contact (and contamination).	0	1	2	0	
	1a. Dry (checkout) cycling 1b. Propellant cycling					
13%	2. Seat leakage caused by surface deterioration due to contamination (and assembly sensitivity).		0	2		
4%	3. Sluggish Operation	0	1	1	0	
	<u>"Capability to provide reusability and long service needed in Shuttle;</u>					
	1. 4000 wet/6000 dry cycle life	0	1	2	0	
	2. 231 hour/15.3g(rms) vibration	1	1	1	2	
	3. 10 year operational service with propellant exposure, including moisture at outlets	1	1	1	1	
	4. Decontamination by GN_2	0	0	0	0	
	5. Ease of maintenance	0	0	0	0	
	6. 10 year filter life	0	0	1	0	
	<u>"Capability to Meet Other Design Criteria"</u>					
	1. Pressure (operating, surge, proof and burst)	2	2	1.5	1	
	2. Flow/Pressure Drop	2	2	2	2	
	3. Response Time	2	2	2	2	
	4. Temperature Resistance	2	2	2	2	
	5. External Ambient Resistance	2	2	2	2	
	6. Response Repeatibility	2	2	2	1	
	7. No Generated Contamination	1	2	2	1	
	8. No Lubricants Contact Propellant	2	2	2	1	
	9. Normally Closed	2	2	2	2	
	10. Weight and Envelope	2	2	2	2	
	Total Evaluation Points	21	25	29.5	19	
	Number of Evaluation Criteria	18	19	19	18	
	"Average" Rating	1.17	1.32	1.55	1.06	
	Disqualified by 0 rating?	Yes	Yes	Yes	Yes	

4. Propellant Shutoff Valve Comparison Matrix

...Seal ...e ...ing)	Sliding Seal Gate Valve (No Bearing)	Butterfly Valve (Using Bearing)	Swing Poppet (Using Bearing)		Moving Poppet Linear Poppet (No-sliding Contact)		Moving Seat Linear Poppet (No-sliding Contact)		Pinch Valve (No-sliding Contact)	
...tomer ...eat	Teflon Seat	Teflon Seat	Teflon Seat	Elastomer Seat	Teflon Seat	Elastomer Seat	Teflon Seat	Elastomer Seat	Teflon Mat'l	Elastomer Mat'l
[illegible]	0	0	1	2	1	2	1	2	1	2
[illegible]	X	X	0	2	0	2	0	2	0	2
	0	0	1	1	2	2	2	2	2	2
[illegible]	0	0	1	2	1	2	1	2	1	1
	2	1	2	2	1	1	1	1	1	1
	1	1	1	1	1	1	1	1	1	1
	0	0	0	0	1	1	1	1	2	1
	0	0	0	1	0	0	1	2	1	2
	0	0	0	1	0	1	0	1	0	1
5	1	2	1	1.5	2	1.5	2	1.5	1	1
	2	2	2	2	2	2	2	2	2	2
	2	2	2	2	2	2	2	2	2	2
	2	1	2	2	2	2	2	2	1	1
	2	2	2	2	2	2	2	2	2	2
	1	1	2	2	2	2	2	2	2	2
	1	1	2	2	2	2	2	2	2	2
	1	2	2	2	2	2	2	2	2	2
	2	2	2	2	2	2	2	2	0	0
	2	2	2	2	1	1	1	1	2	2
5	19	19	26	30.5	26	30.5	27	32.5	25	29
	18	17	10	10	19	19	19	19	19	19
55	1.06	1.12	1.37	1.60	1.37	1.60	1.42	1.71	1.32	1.53
	Yes	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes	Yes

Table IV-4

FOLDOUT FRAME 2

Table IV-5. Actuation System Comparison

% of Past Failures	Actuation System Description	Actuating Ball Type Closure Elem			
		Fuel Pressure Actuated		Gas* Pressure Actuated	
	Type of Ball Seal (ref)	Sliding	Retracting	Sliding	Retracting
	"Moving Part" Guidance	Sliding Fits	Flexures and Rotary Bearings	Sliding Fits	Flexures and Rotary Bearin
	Propellant Pressurized Seals	Rotary Shaft Teflon	Rotary Shaft Elastomers	Rotary Shaft Teflon	Rotary Shaft Elastomers
	"Actuator Piston Seal"	Sliding Seal Elastomer/Teflon	Bellows	Sliding Seal Elastomer/Teflon	Bellows
	Pilot Solenoid Seats	Teflon	AFE411	Teflon/Kaynar	Elastomer
	Seats used in Pressurization System Components			Teflon/Kaynar	Elastomer
	Electrical Component Sealing	Potting/O-rings	Welded Hermetic	Potting/O-rings	Welded Herme
	Motor Type				
	Position Sensing Type	Mechanical Contacts	Solid-State, with Electronics	Mechanical Contacts	Solid-State, wi Electronics
	Level of Maintenance	Actuation System	Actuation System	Actuation System	Actuation Syst
% of Past Failures	"Capability to Avoid Past Problems"				
≈18%	Rotary Shaft and Piston Seal Leakage (due to sliding, contamination, residual propellant, corrosion)	0	1	0	1
13%	Poppet Valve Leakage(s)	0	1	0	1
5%	Low Electrical Resistance (due to moisture)	0	2	0	2
4%	Sluggish Operation	0	1	1	2
4%	Hang-up	0	1	0	1
2%	Disconnect Leakage	2	2	0	1
8%	Erroneous Position Switch Output	0	1	0	1
54% ←	Total Failures Related to Actuation System				

*Dedicated Gas pressurization systems, either 2 or 4 assumed. Gas may be He or GN_2, but GN_2 requires component flow areas ≈2.6 times the size of He components.

System Comparison Matrix

all Type Closure Elements →				← Actuating Moving Seat Poppet Closures →		
Pressure Actuated →		← Motor Actuated →				Gas Pressure Actuated
	Retracting	Sliding	Retracting			
	Flexures and Rotary Bearings	Rotary Bearings	Rotary bearings	Rotary Bearings	Rotary Bearings	Rotary Bearings
	Rotary Shaft Elastomers	Rotary Shaft Teflon	Rotary Shaft Elastomers	Static O-ring Teflon	Static O-ring Elastomers	Static O-ring Elastomers
eflon	Bellows					Bellows
r	Elastomer					Elastomer
r	Elastomer					Elastomer
gs	Welded Hermetic	Potting/O-rings	Welded Hermetic	Potting/O-rings	Welded Hermetic	Welded/Hermetic
		d.c. Brush	Brushless, with Electronics	d.c. Brush	Brushless, with Electronics	
	Solid-State, with Electronics	Mechanical Contacts	Solid State, with Electronics	Mechanical Contacts	Solid State, with Electronics	Solid State, with Electronics
em	Actuation System	Actuation System	Actuation System	Actuation System	Actuation System	Actuation System
	1	1	1	2	2	2
	1	2	2	2	2	1
	2	0	2	0	2	2
	2	1	1	2	2	2
	1	1	1	2	2	2
	1	2	2	2	2	1
	1	0	1	0	1	1

component flow

Table IV-5
(Sheet 1 of 2)

FOLDOUT FRAME 2

Table IV-5. Actuation System Comparison Matri

Actuation System Description	Actuating Ball Type Closure Elements: Fuel Pressure Actuated		Gas* Pressure Actuated		
Type of Ball Seal (ref)	Sliding	Retracting	Sliding	Retracting	Slidi
"Capability to Provide Reusability and Long Service Needed in Shuttle"					
1. 4000 wet/6000 dry cycle life	0	0	0	0	0
2. 231 hour/15.3g (rms) vibration	0	1	0	1	0
3. 10-year Operational Service	0	0	2	1	2
4. Decontamination with GN_2	0	0	X	X	X
5. Ease of Maintenance	0	1	1	2	0
6. Ease of Pre-flight Checkout	1	1	0	0	2
7. Avoidance of Scheduled Maintenance	2	2	0	0	2
"Capability to Meet Other Design Criteria"					
1. Response Time	2	2	2	2	1
2. Response Repeatibility	1	2	1	2	1
3. Temperature Resistance	2	2	2	2	2
4. External Environment Resistance	2	2	2	2	2
5. No Generated Contamination	1	2	1	2	1
6. No Lubricants Contact Propellants	2	2	2	2	2
7. Close with Loss of Elect. & Pilot Power	2	2	2	2	2
8. Weight and Envelope	2	2	1	1	1
9. External Leakage	1	1	1	1	2
10. Electrical Power Consumption	2	2	2	2	1
Total Evaluation Points	22	33	20	30	2
Number of Evaluation Criteria	24	24	23	23	2
Average Rating	0.92	1.37	0.87	1.30	
Disqualified by O rating ?	Yes	Yes	Yes	Yes	Y

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FOLDOUT FRAME

em Comparison Matrix (Continued)

e Closure Elements			Actuating Moving Seat Poppet Closures		
re Actuated	Motor Actuated				Gas Pressure Actuated
Retracting	Sliding	Retracting			
0	0	0	1	2	2
1	0	1	0	1	1
1	2	1	2	1	1
2	0	2	0	2	2
0	2	2	2	2	0
0	2	2	2	2	0
2	1	1	1	1	2
2	1	2	2	2	2
2	2	2	2	2	2
2	2	2	2	2	2
2	1	2	1	2	2
2	2	2	2	2	2
2	2	2	2	2	2
1	1	1	1	1	1
1	2	2	2	2	1
2	1	1	1	1	2
30	28	35	33	40	35
23	23	23	23	23	23
1.30	0.85	1.52	1.43	1.74	1.52
Yes	Yes	Yes	Yes	No	Yes

Table IV-5
(Sheet 2 of 2)

Table IV-6. How the Selected Design Approach Avoids or Minimizes Past Problems

Valve Part	Applicable Past Problems*			Basic Method
	Avoids:	Severity of:	No Help:	
Bellows/Static Seals	A	E	-	Avoids pressurized sliding seals and avoids sliding parts in propellant
Seat/Poppet	-	B	-	Elastomer is more contamination-tolerant
Filter	-	B, F	-	Lower micron rating (B); positive structural backup (F)
Harmonic Drive, Shaft, Bearings	-	D	-	All-welded sealing of electrical parts
		E	-	Teflon bearings cannot corrode, are self-lubricating; other bearings are 100% sealed
Motor	G			No pneumatic disconnect required
		D	-	All-welded sealing of electrical parts
Position Indicator	-	C	-	All solid state construction

*See Table IV-2 for Alphabetical Key

Table IV-7. Assessment of Potential Problems Caused by Long-Life Criteria

Valve Part	Satisfies These Criteria with Little Development Risk	May Not Completely Satisfy These Criteria and Substantial Development Risk May Exist
Bellows	H K M N*	I - Cannot confidently analyze L - May not be able to completely clean convolutions J - No 10-year exposure data on Inconel 718
Static Seals	H I K M N	J - No 10-year exposure data on elastomers L - May not be able to clean seal crevices
Seat/Poppet	H I K L M N	J - No 10-year exposure data on elastomers/other materials
Filter	H L	I - Cannot confidently analyze J - No 10-year exposure data on 304L K - Requires periodic filter replacement N - Difficult to check for pressure drop/cont. level
Harmonic Drive, Shaft, Bearings	H J K L	I - Cannot confidently analyze
Motor	H J K L M N	I - Cannot confidently analyze
Limit Stop Shock Absorber	H J K L M N	I - Cannot confidently analyze
Electronics	H I J K L M N	

*See Table IV-2 for alphabetical key

4.2 Basic Concept Selection

The results of the tradeoff study were as follows:

Average Rating	Valve Concept
1.71	Moving Seat, Linear Poppet (Elastomer Seat)
1.60	Moving Poppet, Linear Poppet (Elastomer Seat)
1.60	Swing Poppet (Using Bearing)(Elastomer Seat)
1.55	Retractable Seal Ball Valve (Using Bearing) (Elastomer Seat)
1.53	Pinch Valve (Elastomer Material)
1.42	Moving Seat, Linear Poppet (Teflon Seat)
1.37	Moving Poppet, Linear Poppet (Teflon Seat)
1.37	Swing Poppet (Using Bearing)(Teflon Seat)
1.32	Retractable Seal Ball Valve (Using Bearing) (Teflon Seat)
1.32	Pinch Valve (Teflon Material)
1.17	Sliding Seal Ball Valve (Teflon Seat)
1.12	Butterfly Valve (Teflon Seat)
1.06	Sliding Seal Gate Valve (Teflon Seat)

Average Rating	Actuation System Concept
1.74	Motor Actuated, Hermetic Sealed (Moving Seat)
1.52	Motor Actuated Retracting, Hermetic (Ball Valve)
1.52	Gas Pressure Actuated (Moving Seat)
1.43	Motor Actuated (Moving Seat)
1.37	Fuel Pressure Actuated (Retracting)(Ball Valve)
1.30	Gas Pressure Actuated (Retracting)(Ball Valve)
0.92	Fuel Pressure Activated (Sliding)(Ball Valve)
0.87	Gas Pressure Activated (Sliding)(Ball Valve)
0.85	Motor Actuated (Sliding)(Ball Valve)

The tradeoff study resulted in a baseline selection of a linear poppet moving seat valve with an elastomer seat and hermetic sealed motor actuation system.

Once the system tradeoff had provided the baseline system, a redundancy study was conducted which considered quad and series-redundant valve and actuation system arrangements. Quantitative assessments of weight, power, and envelope were made. Also considered was development cost and risk, production cost, vibration resistance, contamination resistance, maintainability and ease of checkout.

The valve concept selected during the tradeoff study was subjected to design analysis and preparation for prototype manufacture. The subsequent information presented in Section 5.0 includes the preliminary design effort conducted on the selected valve and actuation system concept.

5.0 DETAIL DESIGN - MOVING SEAT POPPET VALVE AND ACTUATION SYSTEM

The motor-operated "moving seat valve" concept which was selected as a result of the tradeoff study was subjected to detail tradeoff studies to develop the most optimum configuration for manufacture.

Subsequent information presented in Section 5.0 provides detail design analysis on the valve, filter, actuation system, and a redundancy study.

5.1 Moving Seat Poppet Valve Description

The valve consists of a housing assembly, closure assembly, relief valve assembly and various seals. The heart of the assembly is the closure assembly. See Figure 5-1.

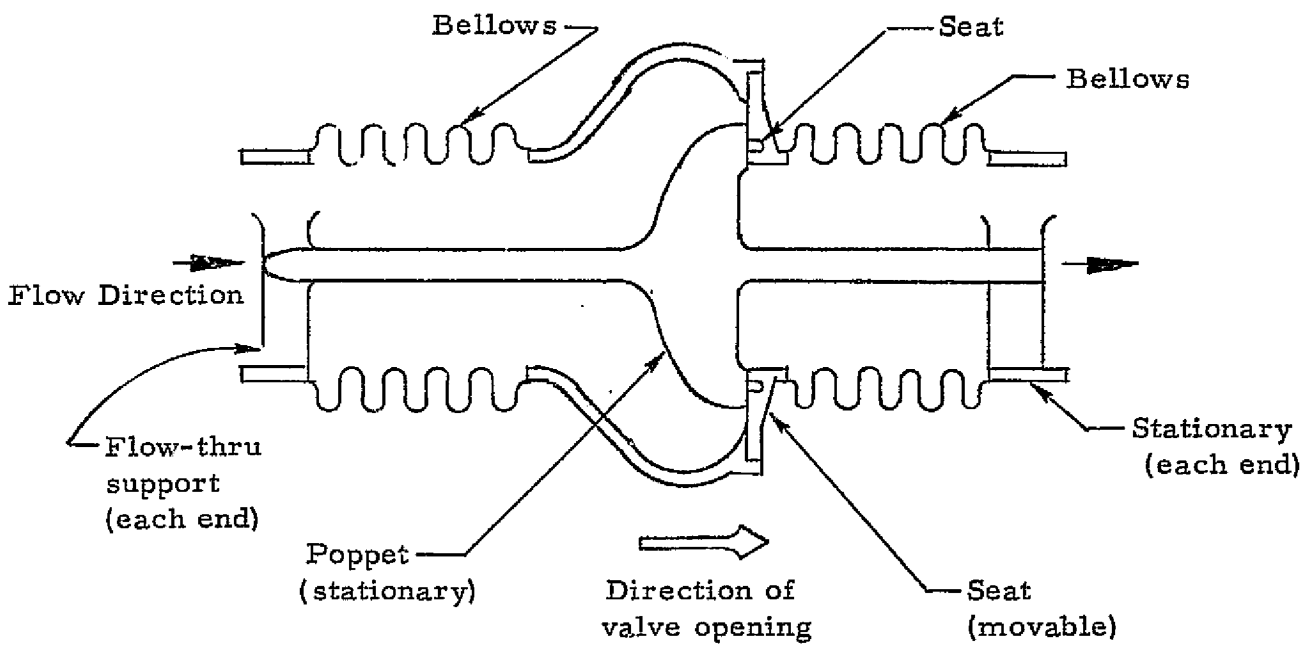

Figure 5-1. Valve Closure Element

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The valve is referred to as an "in-line, pressure balanced, moving seat, poppet valve," or "moving seat" valve for short. The valve poppet is stationary. The valve is opened by deflecting the seat in the direction indicated by the arrow. The actuator is located outside of the flow path, and connects to the outside of the seat housing. The bellows have three purposes: first, they allow the seat to be deflected with respect to the poppet, in order to open the valve. Second, they provide a hermetic seal between the propellant and the actuator. Third, they "pressure balance" the poppet valve. If one analyzes the forces acting on the valve, it will be found that there is virtually zero net area exposed to either inlet or outlet pressure. This minimizes the forces required for actuation.

Key features of the moving seat closure element are:

1. Streamlined flow path, which is easily decontaminated, and which has low flow resistance.

2. Inherent hermetic seal between actuator mechanism and propellant.

3. Low operating forces due to inherent pressure balance, thereby reducing actuation force and power requirements.

4. No sliding parts, so that no wear can occur and no lubrication is necessary.

5. No sliding contact of sealing surfaces — the wear that shortens seat life cannot occur.

6. Bellows provide spring force to close valve when electrical power is removed (fail-closed feature).

5.2 Valve Assembly — Analysis and Preliminary Design

5.2.1 Propellant Valve Seat Sizing — Shape and sizing studies were conducted to determine the valve internal configuration that would provide the minimum ΔP. "K" factors were calculated for two closure configurations and tabulated for stroke/diameter ratios.

Seat poppet ΔP analysis was also conducted as well as a valve ΔP budget. The analysis is included in Appendix B and the results of the pressure drop budget are included as Table V-1.

5.2.2

Table V-1. Pressure Drop Budget Quad Arrangement
(11.91 lb/sec N_2O_4 at 70°F)

Item Causing Pressure Drop		Pressure Drop, psid	
Filter	Inlet to Scroll	0.225	0.68
	Scroll to Housing	0.294	
	Filter Cloth, Backup	0.050	
	Outlet	0.113	
Inlet Manifold (Bends)			0.24
Valve Modules	Contraction	0.03	3.58
	Valve Bellows/Seat	3.52	
	Expansion	0.03	
Outlet Manifold (Bends)			0.48
Total Pressure Drop			4.98

5.2.2 Seat Stroke Versus Flow Diameter — A tradeoff study of valve seat stroke versus flow diameter was conducted. Refer to Figure 5-2 for the resulting plot. Due to improved lateral vibration resistance, a 45-degree seat/poppet angle was selected.

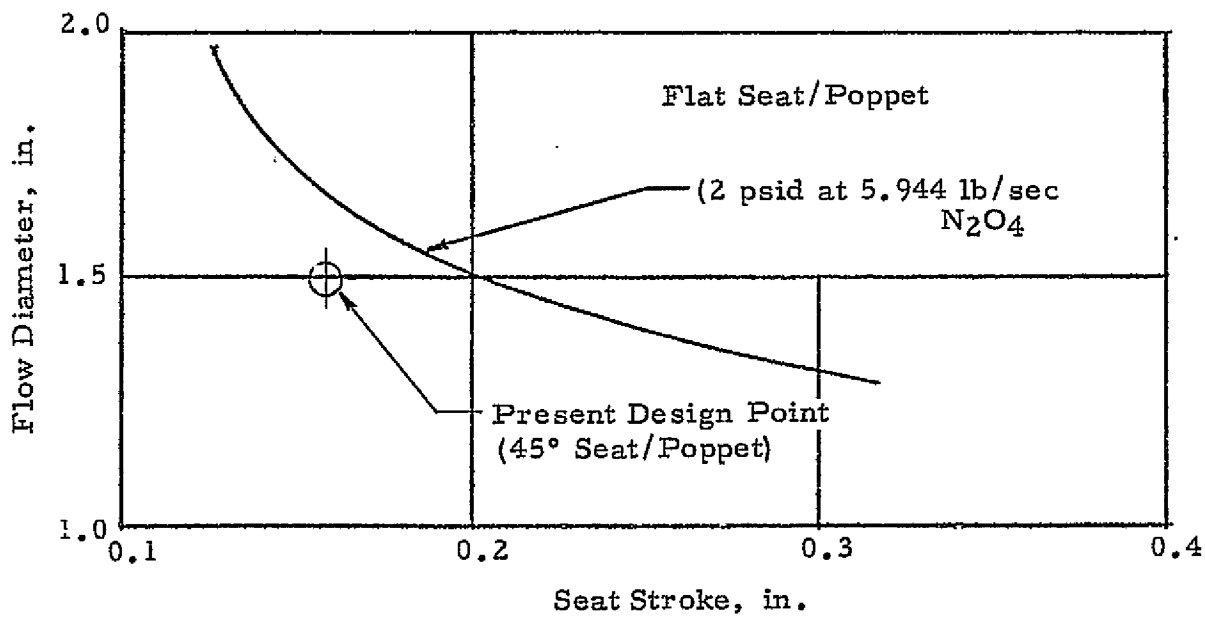


Figure 5-2. Seat Stroke versus Flow Diameter

5.2.3 Pressure Surge — The valve and actuation is so designed that the mechanical spring force obtained from deflecting the bellows back drives the entire actuation system to close the valves; this provides the inertia needed to control the closing response time, thereby limiting "fluid hammer" pressure surges to tolerable levels. The effect on valve closing time on peak surge pressure is shown in Figure 5-3. Note that the pressure surge is less than 30 psi for closing response in excess of 100 milliseconds. The analysis assumes a constant rate of effective area reduction during valve closure. The analysis is included in Appendix B.

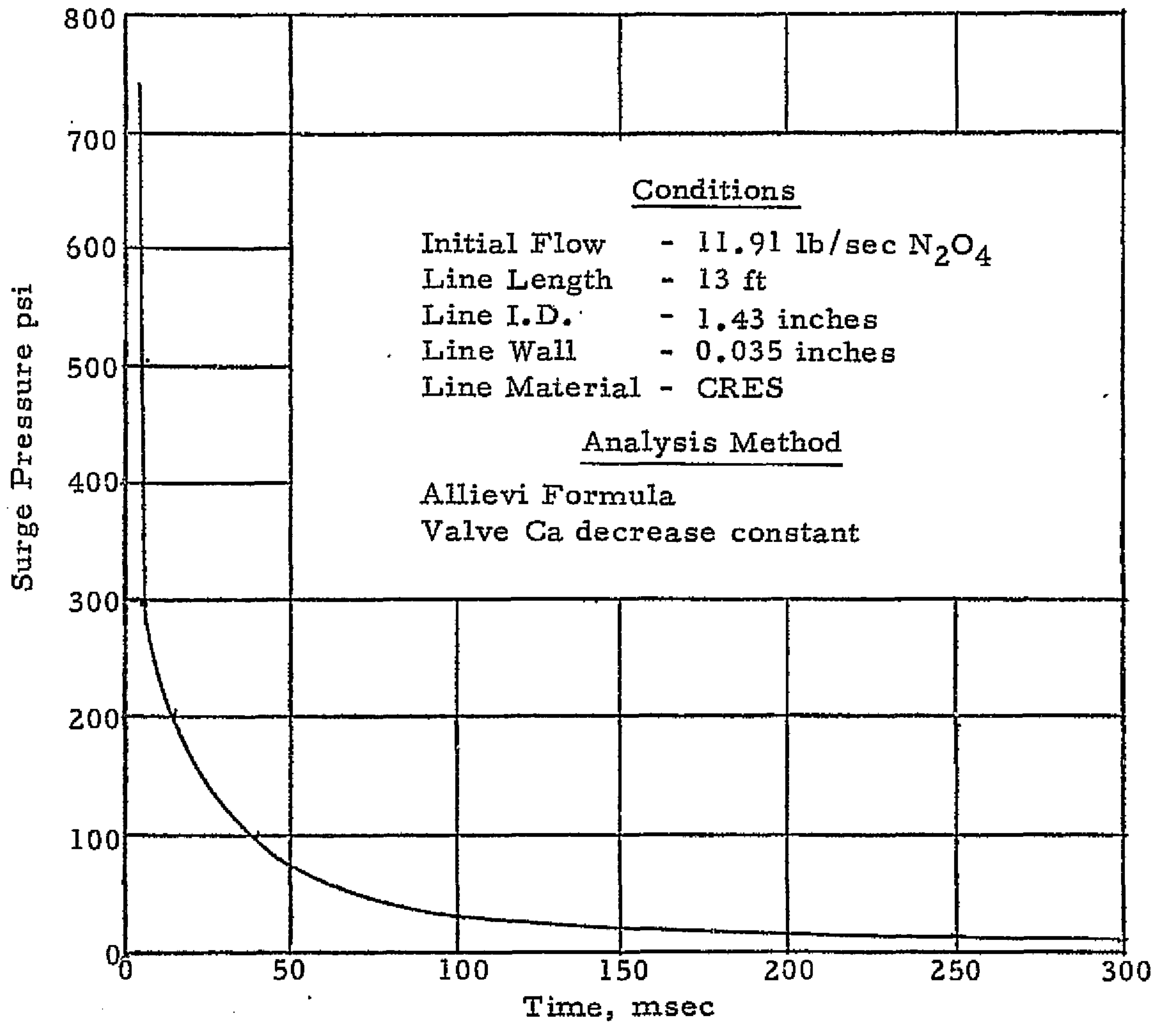


Figure 5-3. Closing Response versus Surge Pressure Study (for N_2O_4; MMH surge is lower)

5.2.4 Bellows Assembly Sizing · Several trial bellows designs were completed for different points on the curve of Figure 5-2 and it was determined that a flow diameter of 1.5 inches resulted in the least rigorous bellows design criteria. Bellows resonant frequencies for the axial direction and lateral direction, with and without propellant were calculated and are included in Appendix B. The results of the bellows assembly sizing effort are included in Table V-2.

Table V-2. Bellows Design Data (Quad Configuration)

Material: Inconel 718, Heat Treated

Basic Type: 3-ply, Hydroformed

Dimensions: OD = 1.96 in. No. of Convolutions = 10

ID = 1.53 in. Pitch = 0.224 in.

Thickness, each ply = 0.005 in. (Total Wall = 0.015 in.)

Spring Rate + 314 lb/in. ± 20%

Effective Diameter = 1.75 ± 0.020 in.

Stress Level: (270,000 psi allowed):

Condition (psi)		Upstream (psi)	Downstream (psi)
Installed	0	80,700	42,500
	400	187,400	149,200
Actuated	0	148,800	25,500 (extension)
	400	255,500	32,200 (extension)

Net Load (Both Bellows = 28.3 lb (Installed), 129 ± 20 lb (Open)

Life (Maximum Stroke and 400 psi Surge) = 11,500 Cycles

Resonant Frequency: 170 Hz (Dry, Axial), to 104 Hz (Wet, Lateral)

5.2.5 Final Configuration — The final valve configuration, resulting from the detail design effort is shown in Figure 5-4. The closure elements are of main concern for this report, inasmuch as the external configuration can be designed as required. The resulting detail poppet and seat design information is included in Table V-3.

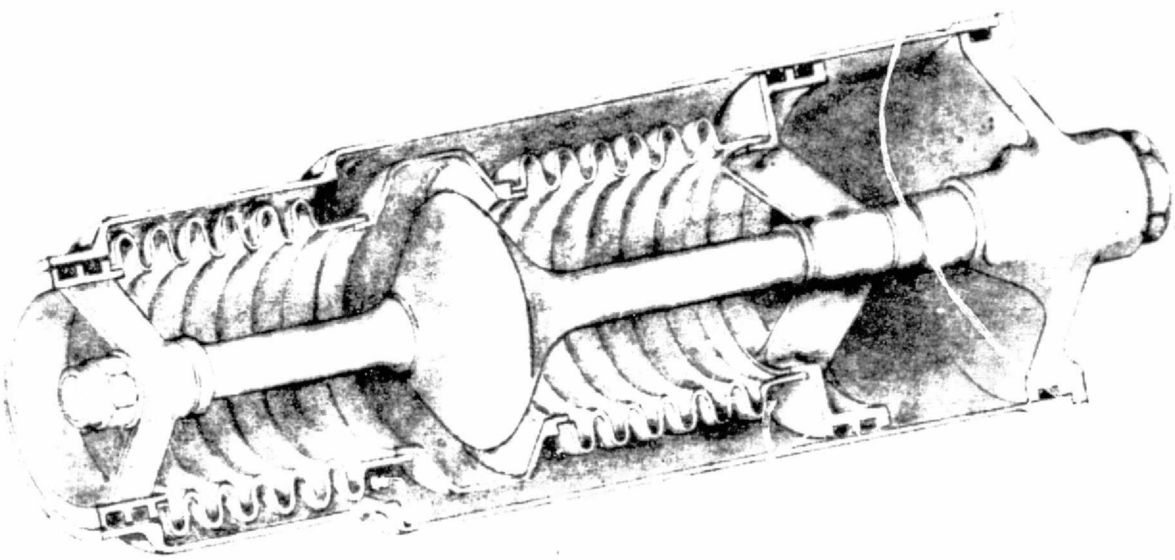


Figure 5-4. Valve Closure Element

Table V-3. Poppet and Seat Design Data (Quad Configuration)

Materials:	AFE411 (MMH): AFE124D(N_2O_4) for Poppet Seal, Inconel 718 for Seat Land
Basic Configuration:	Spherical, Mechanically Retained Seal, Metal-to-Metal Limit Stop (Bumper)
Dimensions:	1.70-inch Mean Diameter 0.03-inch Seat Land Width 90° Included Contact Angle 0.155 ± 0.005-inch Seat Stroke

Loads and Stresses:

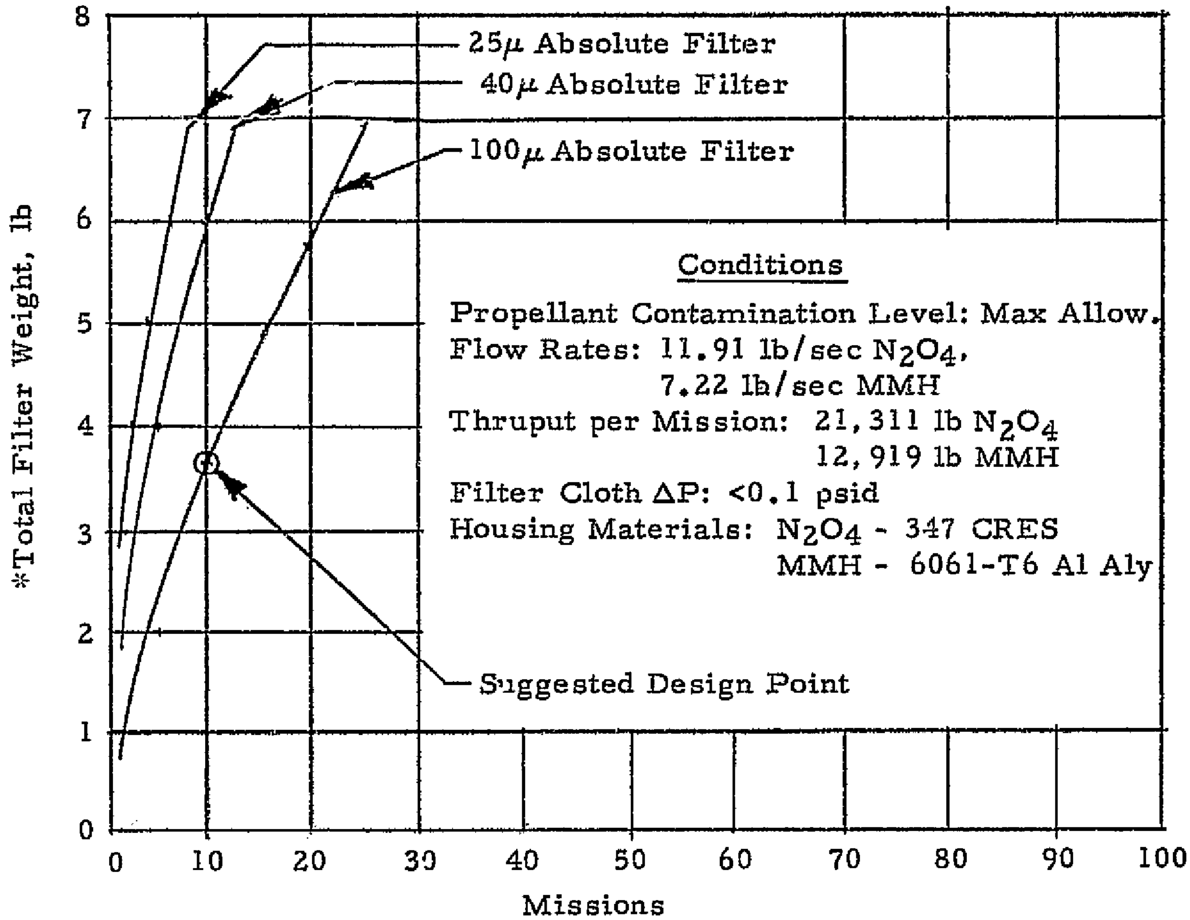
Pressure	Nominal Load (lb)	Maximum Stress* (psi)
Unpressurized	28.3	250
180 psid	48.7	430
265 psid	58.7	518

*Without Bumper Contact; any bumper contact will reduce seal stress.

5.3 Filter Maintenance Study

A filter maintenance design study was performed to establish the tradeoff of filter weight versus the number of shuttle missions. Although it would be ideal to design filters for the full life of the Shuttle, the filter weight would be prohibitive in order to have the contaminant holding capacity needed to avoid excessive pressure drop.

Figure 5-5 presents results of the filter maintenance design study for 25-, 40-, and 100-micron absolute filter sizes. Extrapolation of the curves in Figure 5-5 predict extremely heavy filter weights for extended mission life requirements such as the Shuttle is anticipating. A recommended design point was selected which provides a reasonable maintainability time while not too severely impacting system weight or usable



*"Maintainable" Filter Subassemblies (2 Elements + 2 Housings)

Figure 5-5. Filter Maintenance Design Study

mission life. The recommended design point was for a 100-micron absolute filter which would be replaced after each ten missions, which is approximately once a year. The total system weight for this concept would be 3.3 pounds and includes two filter elements and two filter housings. A dutch weave filter element is recommended because it affords a degree of "depth" filtration; 100-micron dutch weave cloth should not permit particles larger than 250 microns (0.010 inch) in the "third" dimension to pass. Typical raw data used in filter maintenance study is shown in Figure 5-6.

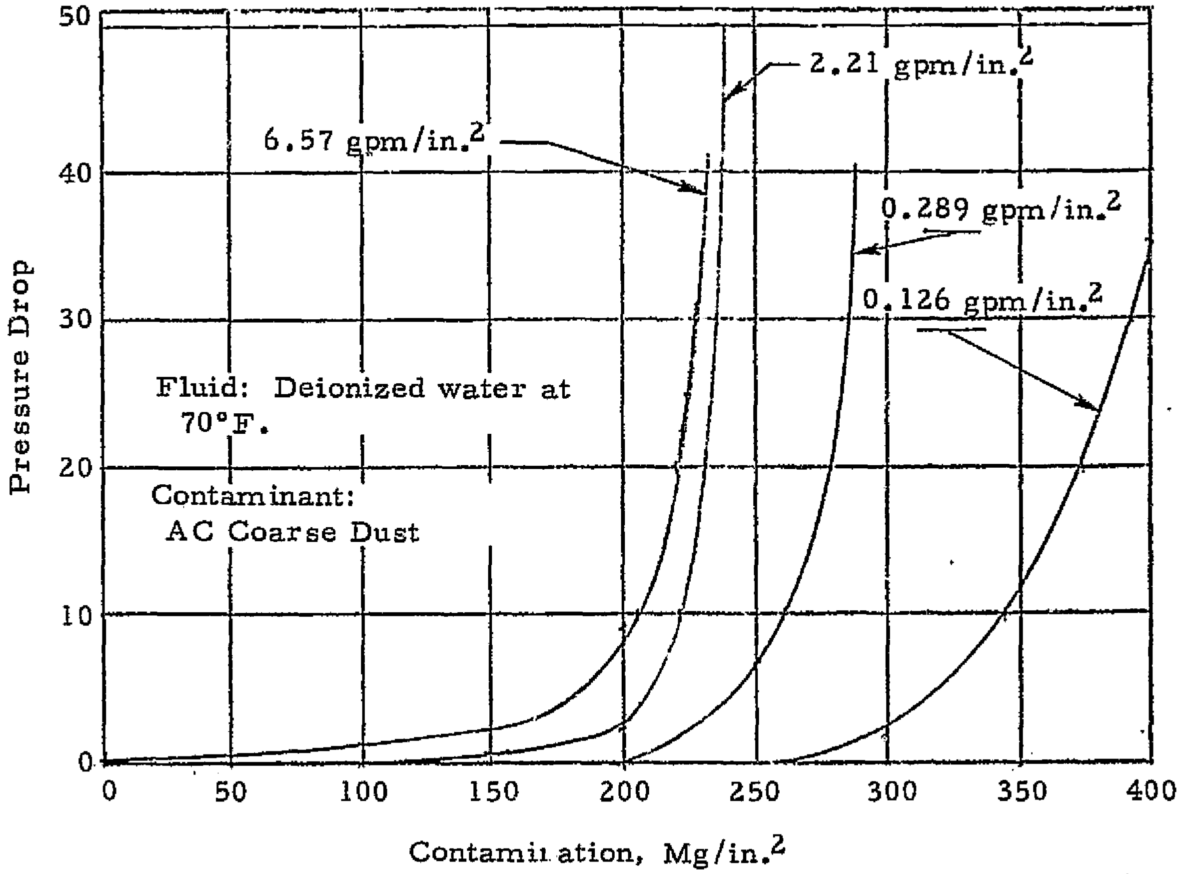


Figure 5-6. Contamination Tolerance of 30 x 160 PDSW Wire Cloth

5.4 Actuation System

The actuation system, selected as a result of the tradeoff study, was a hermetic sealed motor assembly. Additional tradeoffs were conducted on various motor assembly configurations that could be used to drive the valve. Figure 5-7 presents the system schematics of the evaluated systems. The selected actuation system consists of a motor, motor brake, single stage of gear reduction, an open position sensor, an electronic control, and a stop/shock absorber.

5.4.1 Motor Study — A motor concept study was conducted, comparing an induction motor, a stepper motor, and two types of brushless dc motors, i.e., one using Hall effect commutation and one using Light commutation. The results of the study are tabulated in Table V-4. Note that the selected design approach is a brushless ac motor utilizing "Light commutation." The other possible approach, using "Hall Effect" commutation was ruled out because of marginal characteristics at the

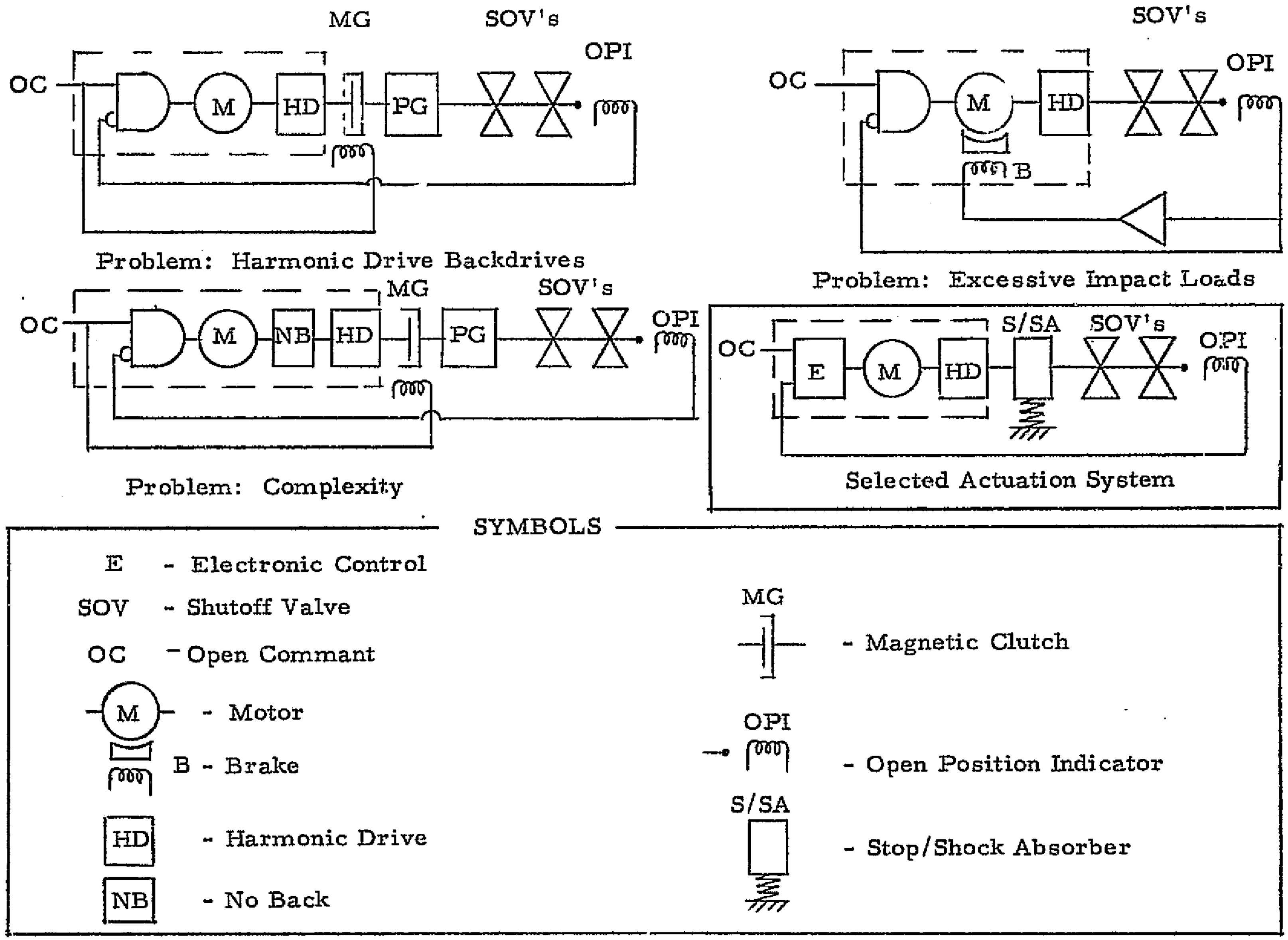


Figure 5-7. Motor Actuation Study

Table V-4. Alternate Motor Concept Design Study

Parameter		Induction Motor	Brushless DC Motors: Hall Effect Commutation	Brushless DC Motors: Light Commutation	Stepper Motor
Weight, lb	Motor	2.5	1.40	1.25	6.0
	Brake	0.5	-	-	-
	Electronics	0.55	0.55	0.55	0.55
	Total Weight	3.55	1.95	1.80	6.55
Inertia (oz-in.-sec^2)		1.15	0.160	0.124	2.0
Efficiency, Percent		30.0	40.0	34.0	-
Vibration (50 g's)		OK	OK	OK	OK
High Temperature (200°F)		OK	No	OK	OK
Cost (Relative)*		2	4	3	1
Procurement Time (Relative)*		2	4	3	1
				Selected Motor Concept	

* "1" is lowest cost/procurement time

(Above comparative figures were obtained based on a 500-rpm motor, 2-3/4 inches I.D. developing torque of 38 oz-in.)

maximum operating (engine soakback) temperature of 200°F. Both the induction and stepper motors have high weight when configured as multi-pole-pancake/low speed-high torque units. A brush-type motor was not included in the study because the OMS pod specification did not allow brush-type motors. The motor design data is provided in Table V-5.

5.4.2 Electronic Control — The electronic control schematic is provided in Figure 5-8. It consists of the electronics necessary to provide motor power, brake power, and position-sensing logic. The Shuttle dc buss voltage is conditioned as required, to operate the motor with valve position being monitored to allow reduction of holding power requirement when the valve is in the open position.

Table V-5. Motor Design Data (Quad Configuration)

Basic Type:	"Pancake" Brushless DC, 2-phase, 14-pole, Optical Commutation, Brake Winding
Materials:	Rotor Magnets, Alnico 5-7; Armature Laminations, Silicon Steel; Armature Windings, Class 155°C Copper; Lead Wires, Teflon Coated Copper; Rotor Bearings, 52100 using MIL-G-32778 Lubricant
Dimensions:	3.625 inches OD x 1.00 inches long; 2.500 inches ID
General Data:	Moment of Inertia, 0.124 oz-in.-sec^2; Total Number of Motor Revolutions to Open/Close, 4.86*; Average Motor Speed for 1/2-second Response 583 rpm*; Resistance of Each Winding, 3.1 ohms.*

*100:1 Gear Reduction

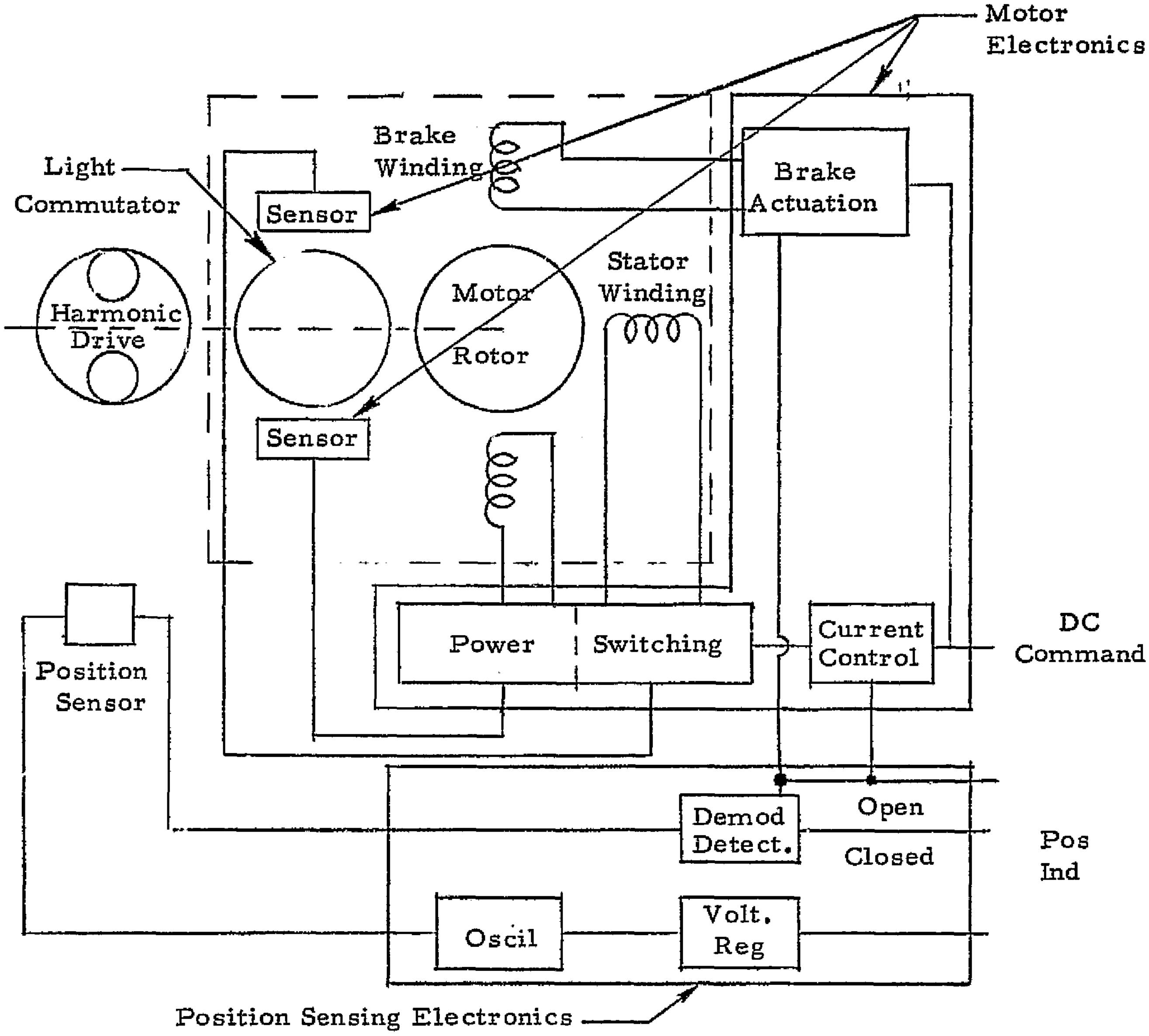


Figure 5-8. Electronic Control Diagram

5.4.3 Harmonic Drive — The harmonic drive unit provides a single stage of gear reduction to the actuation system. The unit will be hermetically sealed which isolates the motor and electronic control package from the shutoff valves. The Harmonic Drive design data is presented in Table V-6.

Table V-6. Harmonic Drive Design Data
(Quad Configuration)

Material:	321/347 CRES, Flexspline and Circular Spline; Gold-plated Gears; (Wave Generator Material TBD)
Basic Type:	Hermetic Version, External Flexspline
Configuration Details:	2-inch Pitch Diameter, 100 : 1 Reduction, 200 Teeth (Flexspline), 202 Teeth (Circular Spline), 30° Involute Teeth (Mod)
	Shaft Bearings
Material and Type:	Fiberglide, Woven TFE on Rigid Backing
Stress:	Approximately 1200 psi; (20,000 psi maximum allowed)

5.5 Valve and Actuation System Design Summary Results

The preliminary valve and actuation system configuration as selected in the tradeoff study was further defined as a result of the design study conducted. The basic design configuration selections are as listed.

- The valve closure element is a spherical elastomeric seat/poppet configuration with a 1.53-inch effective flow diameter.
- The filters are 100-micron absolute dutch weave cloth having a 10-mission service capability and weighing 3.3 pounds for two filter elements and two filter housings.
- The actuation system consists of a light commutated brushless dc motor, an electronic control circuit, a harmonic drive providing a single stage of gear reduction and hermetic sealing for the motor and electronics, a shock absorber, and an open position sensor to allow the reduction of holding power once the valve is in the true full open position.

A design summary of the prototype moving seat valve is included as Table V-7.

Table V-7. Design Summary — Prototype Moving Seat Valve

O-ring Cross Section - - - - - - - - -	= 0.070 inches
Groove Depth - - - - - - - - - - - - -	= 0.059 inches
Projection of Ring - - - - - - - - - -	= 0.011 inches
Compression of O-ring when Stop - - is Reached	= 15.8 percent
Compressive Force for 85 Shore Ring	= 3.9 lb/in. circumference
Force Required to Compress Ring - -	= 20.8 lb
Bellows Load Required to Compress Ring	= 14.7 lb
Poppet Stroke Required to Compress Ring	= 0.0156 inches

Bellows

OD	= 2.022"
ID	= 1.53"
D_R	= 1.566"
Pitch	= 0.25"
h	= 0.228"
$\frac{h}{t}$	= 38

10 Active Convolutions

Three-ply, 0.006" Thickness per Ply

Spring Rate: 368 lb/in.

Critical Pressure for Squirm = 964 psi

Table V-7. (Continued)

Upstream Bellows installed at 0.210 deflection results in an installed stress, δ_M = 82,600 psi.

At 0.170 stroke (0.380 total deflection), δ_M = 149,800 psi

At 0.200 stroke (0.410 total deflection), δ_M = 161,400 psi

At proof pressure of 400 psi, δ_B = 96,500 psi

Combined stress at 0.170 stroke, proof pressure = 246,300 psi

Half range repeated stress (stroke of 0.170", pressure change from 0 to 400 psi): 133,400 psi (17,000 cycle life)

Half range repeated stress (stroke of 0.200", pressure change from 0 to 268 psi): 107,900 psi (48,000 cycle life)

Downstream Bellows installed at 0.137" deflection results in an installed stress, δ_M = 54,000 psi. At 0.170" stroke bellows is in tension 0.033 inches.

Forces

Net Spring Load, Valve Module Closed:	27.0 lb
Net Spring Load, 0.170 Stroke:	152 ± 25 lb
Net Spring Load, 0.200 Stroke (0.030 Overstroke):	174 ± 29 lb
Crank Radius:	0.457 inches
Crank Rotation, 0.200 Stroke (0.030 Overstroke):	25.2°
Rotary Seal Friction:	0.6 in.-lb per Seal

Table V-7. (Continued)

Torque required to open single valve module	Start torque, unpressurized: 13.28 in.-lb Start torque, pressurized to 268 psi: 26.8 in.-lb Torque at 0.170 stroke: 69.1 ± 11.2 in.-lb Torque at 0.200 stroke: 79.0 ± 12.9 in.-lb
Torque required to hold single valve module open	At 0.200 stroke: 76.4 in.-lb ± 12.9 At 0.170 stroke: 66.7 ± 11.2 Just off seat, unpressurized: 10.88 in.-lb Just off seat, pressurized: 24.4 in.-lb
Torque required to open two valve modules	Start torque, unpressurized: 26.56 in.-lb Start torque, pressurized to 268 psi: 53.6 in.-lb Torque at 0.170 stroke: 138.2 ± 22.4 in.-lb Torque at 0.200 stroke: 158 ± 25.8 in.-lb
Torque required to hold two valve modules open	At 0.200 stroke: 152.8 ± 25.8 in.-lb At 0.170 stroke: 133.4 ± 22.4 in.-lb Just off seat, unpressurized: 21.76 in.-lb Just off seat, pressurized: 48.8 in.-lb
	<u>Stress Analysis</u>
Poppet Shaft (at root of threads)	Load at proof, valve closed: 2370 lb Root area, $\frac{5}{16}$ - 24 thd: 0.0524 in.^{2} Tensile stress: 45,300 psi

Table V-7. (Continued)

Actuator Shaft
Torque required to open two modules 0.200 = 184 in.-lb (max)
Minor diameter of serrations: 0.350
Torsional stress = 21,800 psi (at 0.200 stroke)
Maximum load on roller: 101.5 lb (max)
Center of serrations to center of bearing: 0.69
Moment: 70 in.-lb; bending stress: 13,500 psi
Roller
Roller diameter: 0.750
Roller width: 0.050
Maximum load: 101.5 lb
Contact Stress: 166,000 psi
Roller Bearing
Load: 101.5 lb maximum
Projected area: 0.0625 in.^{2}
Bearing stress = 1622 psi (max)
Transition Shaft Bearing
Load: 203 lb (max)
Projected area: 0.234 in.^{2}
Bearing stress: 868 psi (max)
Screws
Load per screw on cap: 395 lb at proof pressure
Boot stress, #8-32 thread; 32,900 psi

5.6 Redundancy Study

5.6.1 General — Quad and series redundant valve and actuation system arrangements were compared to quantitatively assess weight, power, and envelope. Also, these inherent characteristics were compared:

1. Development cost/risk
2. Production cost/problems
3. Vibration resistance
4. Contamination resistance
5. Maintainability
6. Ease of checkout

These common design conditions were imposed for this comparison:

1. Five psid total pressure drop
2. Five-tenths seconds opening response time
3. Seven-tenths seconds closing response time
4. Same level of maintenance

Table V-8 summarizes results of the redundancy study. It was somewhat surprising that the weight analysis indicates very little difference between the series and quad configurations. This is a result of smaller size valves in the quad-redundant unit. However, this does result in a pressure drop penalty in the event of a valve fail closed condition. Because the series configuration offered no substantial advantage over the quad configuration, with respect to weight, envelope, and power, NASA JSC directed the program effort to reflect the quad-redundant configuration only.

The subsequent paragraphs of this section present information pertaining to the detail studies, as well as outline and assembly drawings for both series- and quad-redundant configurations.

Table V-8. Comparison of Quad and Series Arrangements

Consideration	Level of Redundancy	
	Quad	Series
Weight	40 lb	41 lb
Envelope	Approximately Equivalent (See Dwgs 5736023 & 5736024)	
Power (Total Required)	168 watts (Peak Running)*	168 watts (Peak Running)*
	28 watts (Hold Open)*	14 watts (Hold Open)*

Relative Rating of Inherent Characteristics

Development Cost/Risk	Lower	
Production Cost/Problems		Lower (≈ 50%)
Life	Greater Potential	
Vibration Resistance	Fewer Problems	
Contamination Resistance		Better
Maintainability	-	Less Maintenance Required
Ease of Checkout	-	Minimum Testing Required

*Latest power analysis supercedes these preliminary design levels; however, power levels are the same for quad and series.

Note: Comments listed denote superior rating (relative to the other level of redundancy)

5.6.2 Weight Versus Pressure Drop Study — A sensitivity study was conducted to determine the effect of allowable pressure drop on the quad valve and actuation system weight. Three individual design points were studied and analyzed to establish the weight versus pressure drop. These data are plotted in Figure 5-9.

Note that Figure 5-9 shows pressure drop for both "Normal Operation" and "One Valve Failed Closed." The weight penalty necessary to reduce the "One Valve Failed Closed" pressure drop is evident.

5.6.3 Power Versus Response Time Study — Another sensitivity study was conducted that relates the effect of input electrical power on valve response time. Figure 5-10 shows the results of this study. The input power and current levels did not allow for an operating force margin. Thus, these levels were increased somewhat in the final design. Note that, as would be expected, a reduction in the transient opening current results in a longer opening response time. There is a similar relationship in hold-open power and closing response time; a reduction in the hold-open power results in longer closing times. This rather surprising relationship occurs because the lower hold-open power requires a higher gear reduction; this, in turn, results in the motor rotor accelerating to a lower speed because the mechanical energy being supplied from the bellows during back-drive of the actuation system is fixed.

5.6.4 Preliminary Design Layout Drawings -- Preliminary design layout drawings were prepared for the motor-operated moving seat valve arranged as (a) a quadredundant package having four motor-driven, mechanically linked pairs of moving seat shutoff valves and (b) a series-redundant package having two motor-driven, mechanically linked pairs of moving seat shutoff valves. Refer to Figures 5-11 through 5-14 for installation and assembly drawings of the valve and actuation system.

5.6.5 Weight Summary — A weight analysis was made for the valve and actuation system in both series and quad-redundant configurations. A summary of this analysis is presented in Table V-9. The table includes the basic component name, the material, a number referring to the item number of the part as called out on appropriate drawings, and the calculated weight of the components. The basic analysis indicates the series redundant configuration weight to be 40.67 pounds, which can be reduced to approximately 35.93 pounds if titanium is used. The quad-redundant weight is 39.71 pounds and can also be reduced to 35.60 pounds with the same material considerations for titanium.

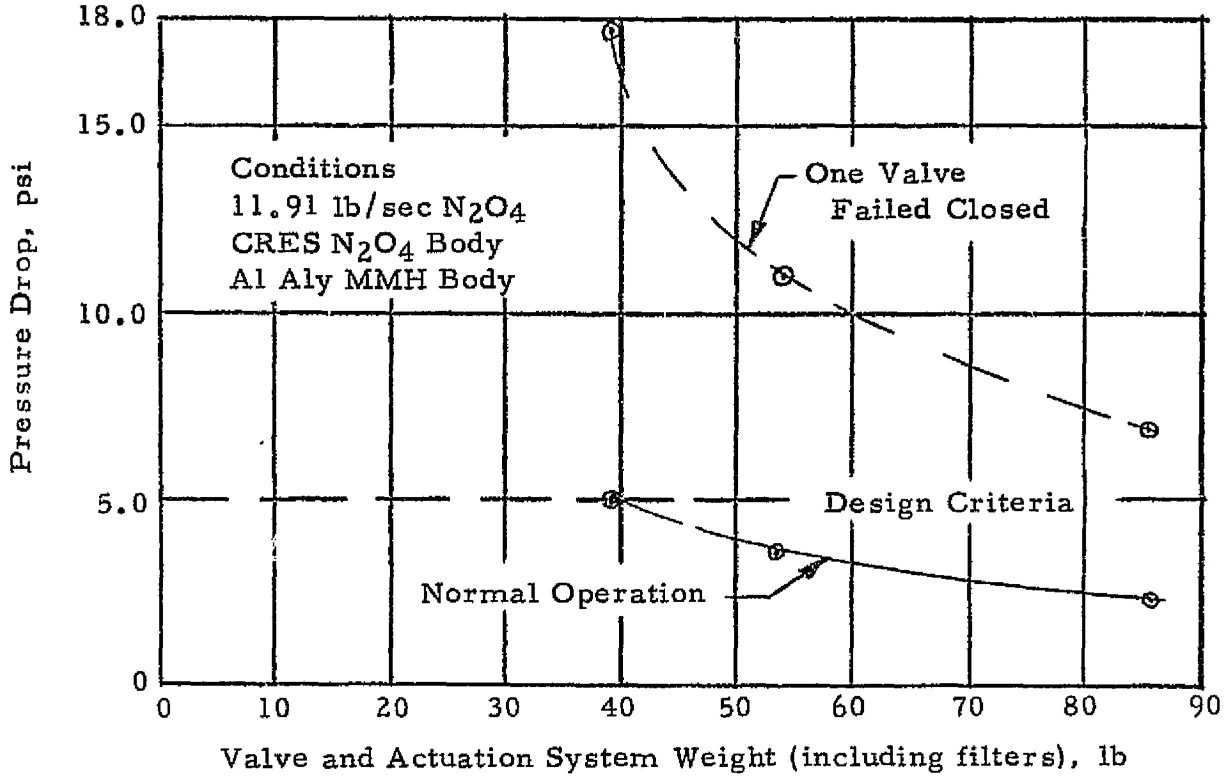


Figure 5-9. Weight versus Pressure Drop Sensitivity Study

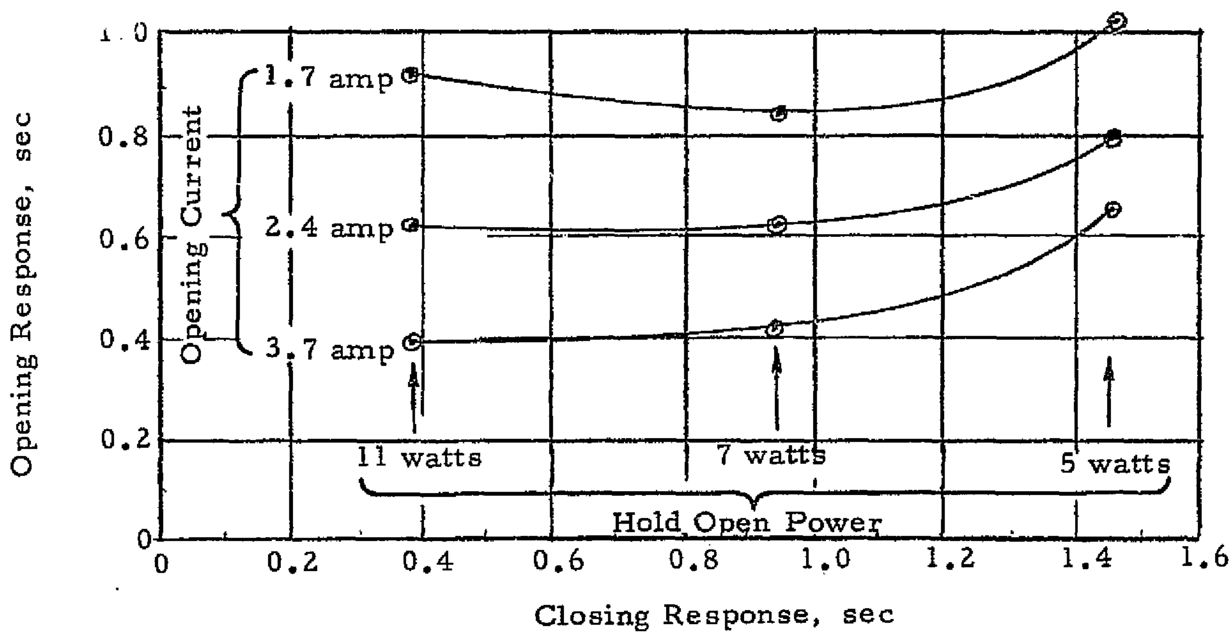


Figure 5-10. Power versus Response Time Sensitivity Study

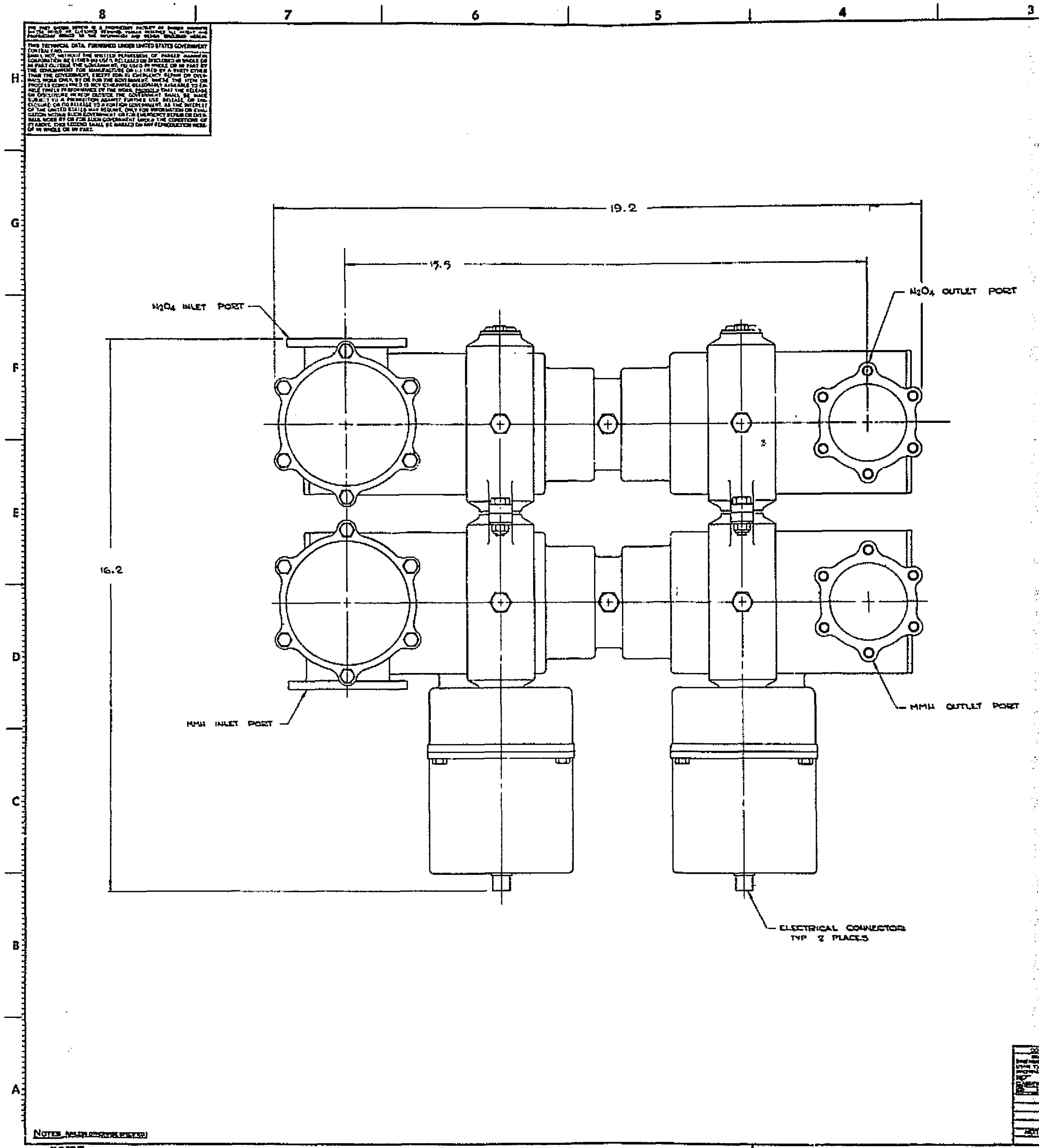


Figure 5-11. Preliminary Design Layout Series Redundant OMS Valve

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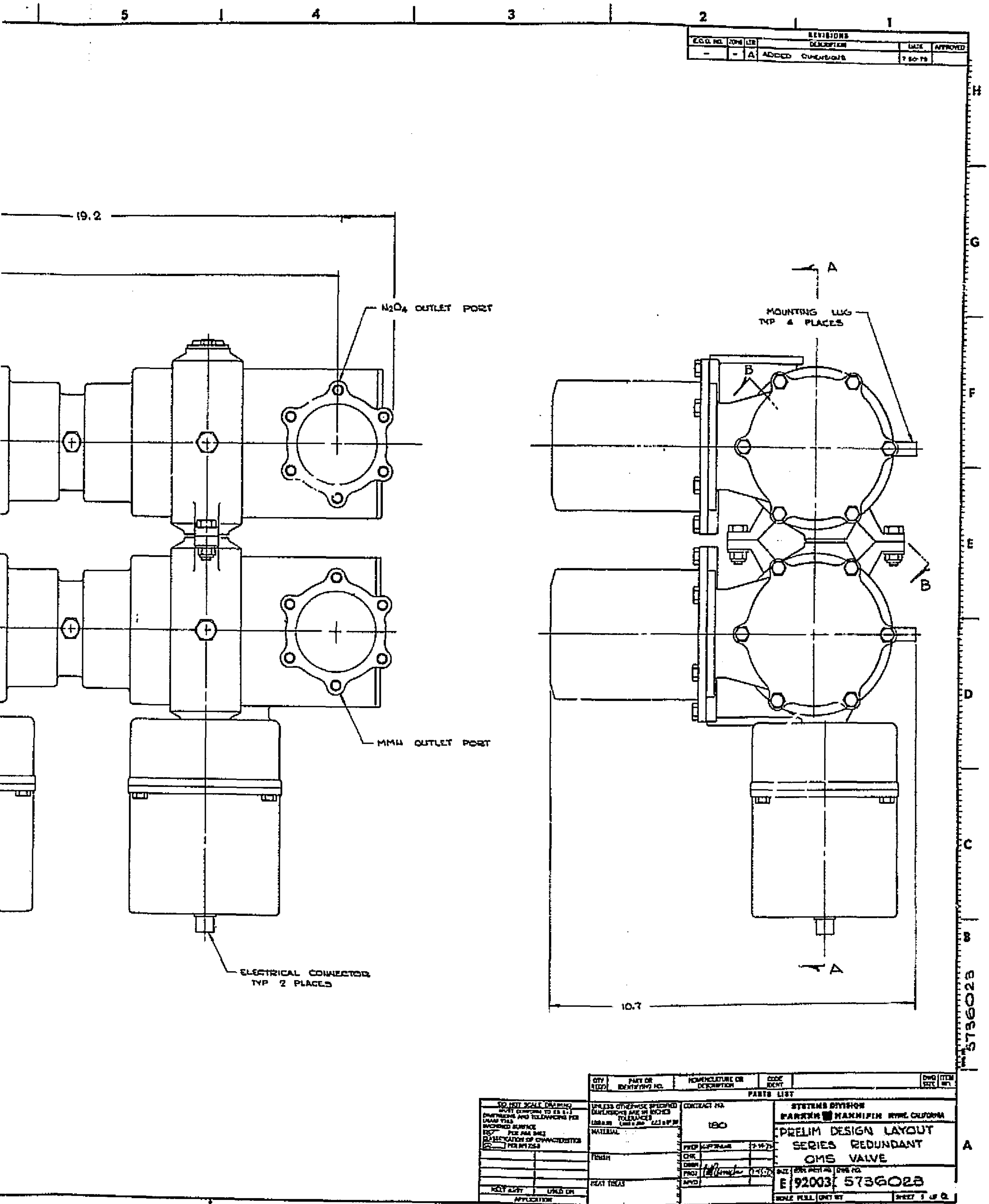

gn Layout Series Redundant OMS Valve

Figure 5-11
(Sheet 1 of 2)

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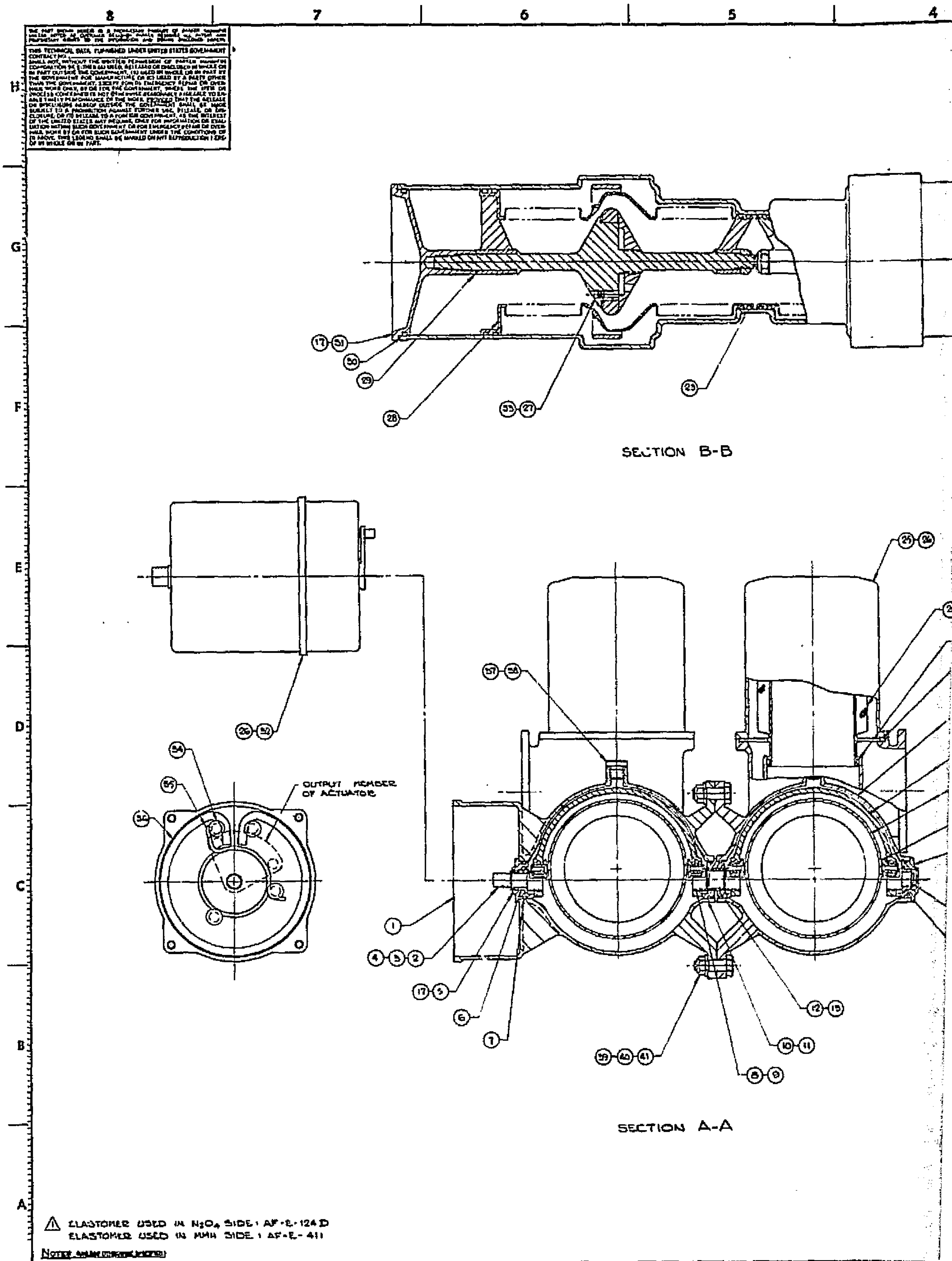
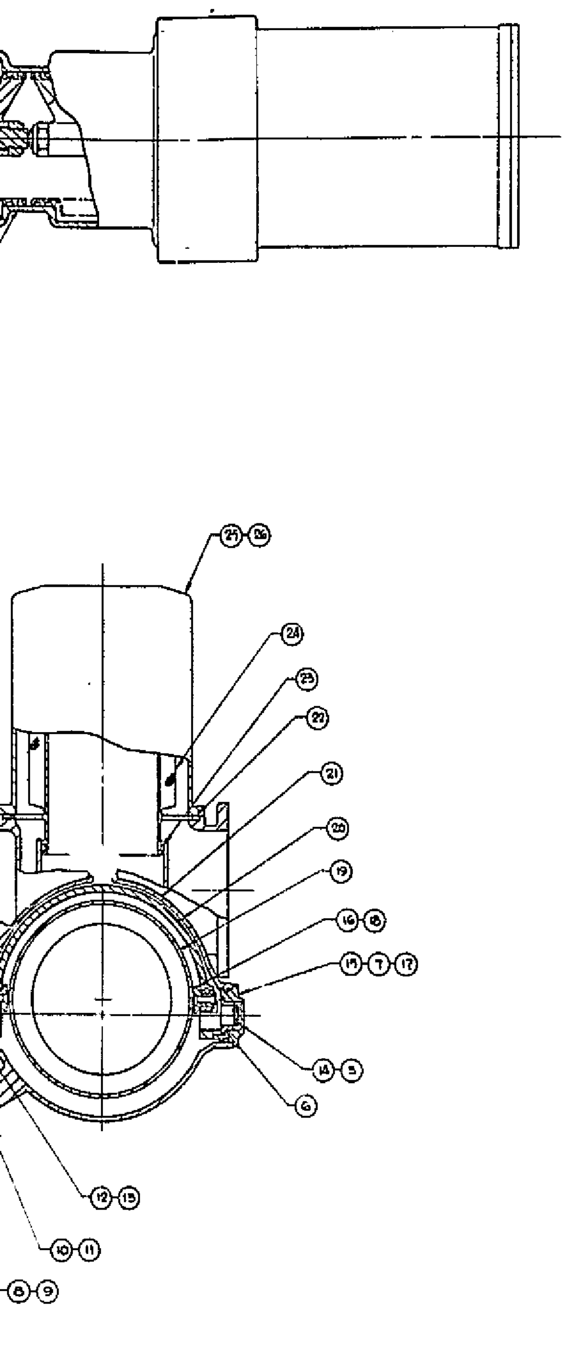


Figure 5-12. Preliminary Design Layout, Series Redund



QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	MATERIAL	ITEM NO.
4		NUT	CRES	41
4		WASHER	AL ALY	40
4		SHOULDER BOLT	CRES	39
6		PACKING	NOTED	38
6		PLUG	CRES	37
2		SHOCK ABSORBER	CRES	36
2		RETURN SPRING	CRES	35
8		GUIDE	CRES	34
2		PIN PLUG	CRES	33
2		ACTUATOR	[illegible]	32
4		CAP	CRES FOR N_2O_4 AL ALY FOR MMH	31
12		PACKING	NOTED	30
AR		SHIM	CRES	29
2		VALVE MODULE	BELLOWS & ATTACHMENTS: INCO-NEL 718; POPPET COMPONENTS: CRES	28
2		VALVE ASSY	BODY, BALL & SPRING: CRES SEAT: TEFLON	27
20		SCREW	CRES	26
2		HOUSING	CRES FOR N_2O_4 AL ALY FOR MMH	25
2		FILTER ASSY	CRES	24
10		PACKING	NOTED	23
2		PACKING	NOTED	22
1		BODY ASSY, N_2O_4	CRES	21
4		CRANK	CRES	20
2		VALVE MODULE	BELLOWS & ATTACHMENTS: INCO-NEL 718; POPPET COMPONENTS: CRES	19
8		SPIROL PIN	CRES	18
32		SCREW	CRES	17
8		ROLLER ASSY	[illegible]	16
2		CAP	CRES	15
		SHAFT	CRES	14
		PACKING	NOTED	13
		BEARING	TEFLON	12
		PACKING	NOTED	11
		SLEEVE	CRES	10
		RETAINING RING	CRES	9
		SHAFT	CRES	8
		PACKING	NOTED	7
		BEARING	TEFLON	6
		RETAINER, BEARING	CRES	5
		PACKING	NOTED	4
		RETAINING RING	CRES	3
		SHAFT	CRES	2
		BODY ASSY, MMH	AL ALY	1

PARTS LIST

CONTRACT NO.	180
	SYSTEMS DIVISION PARKER HANNIFIN IRVINE, CALIFORNIA
TITLE	PRELIM DESIGN LAYOUT, SERIES REDUNDANT OMS VALVE
SIZE	E
CODE IDENT NO.	92003
DWG NO.	5736023

out, Series Redundant OMS Valve

Figure 5-12
(Sheet 2 of 2)

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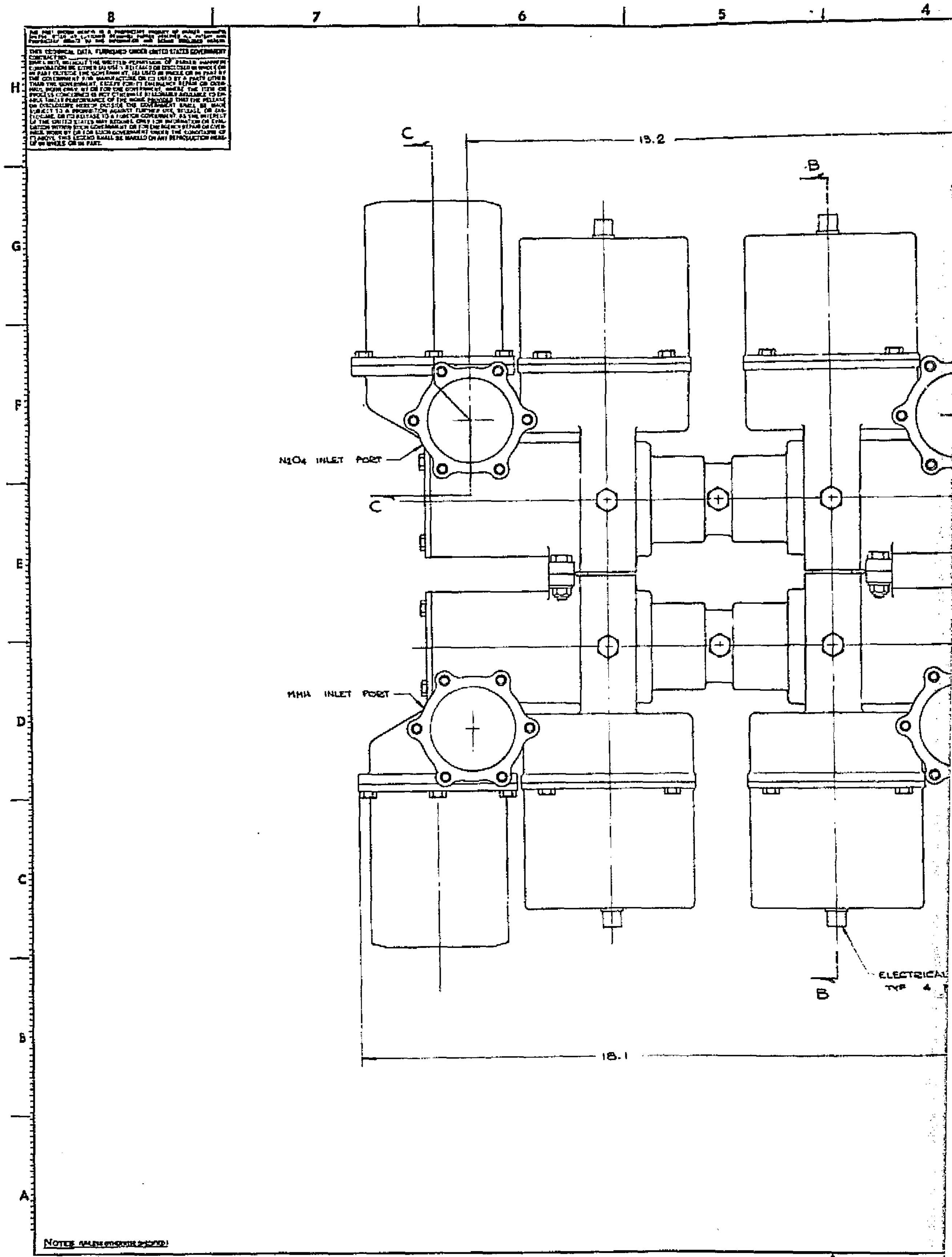

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Figure 5-13. Preliminary Design Layout Quad Redundant OM

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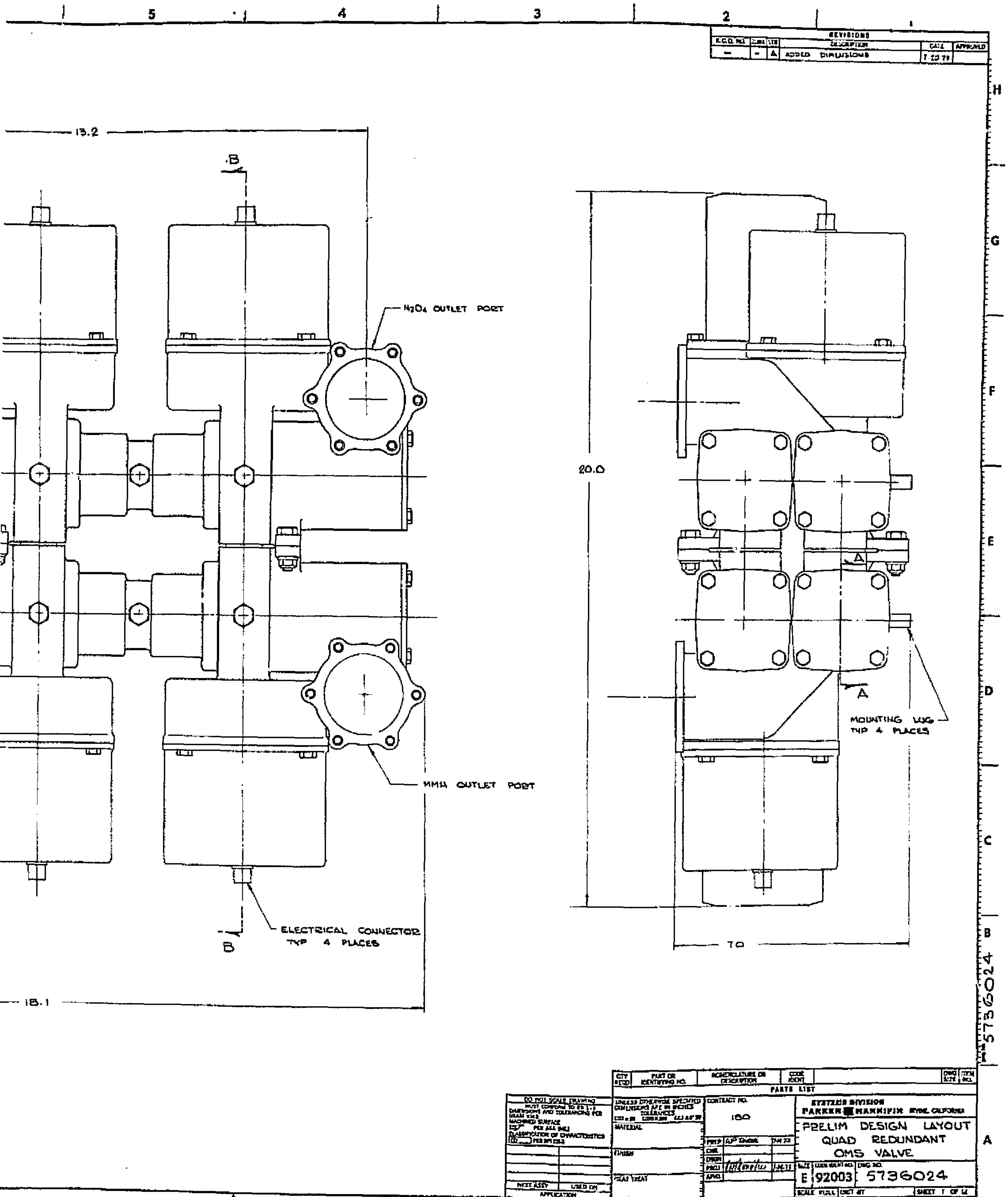

Design Layout Quad Redundant OMS Valve

Figure 5-13

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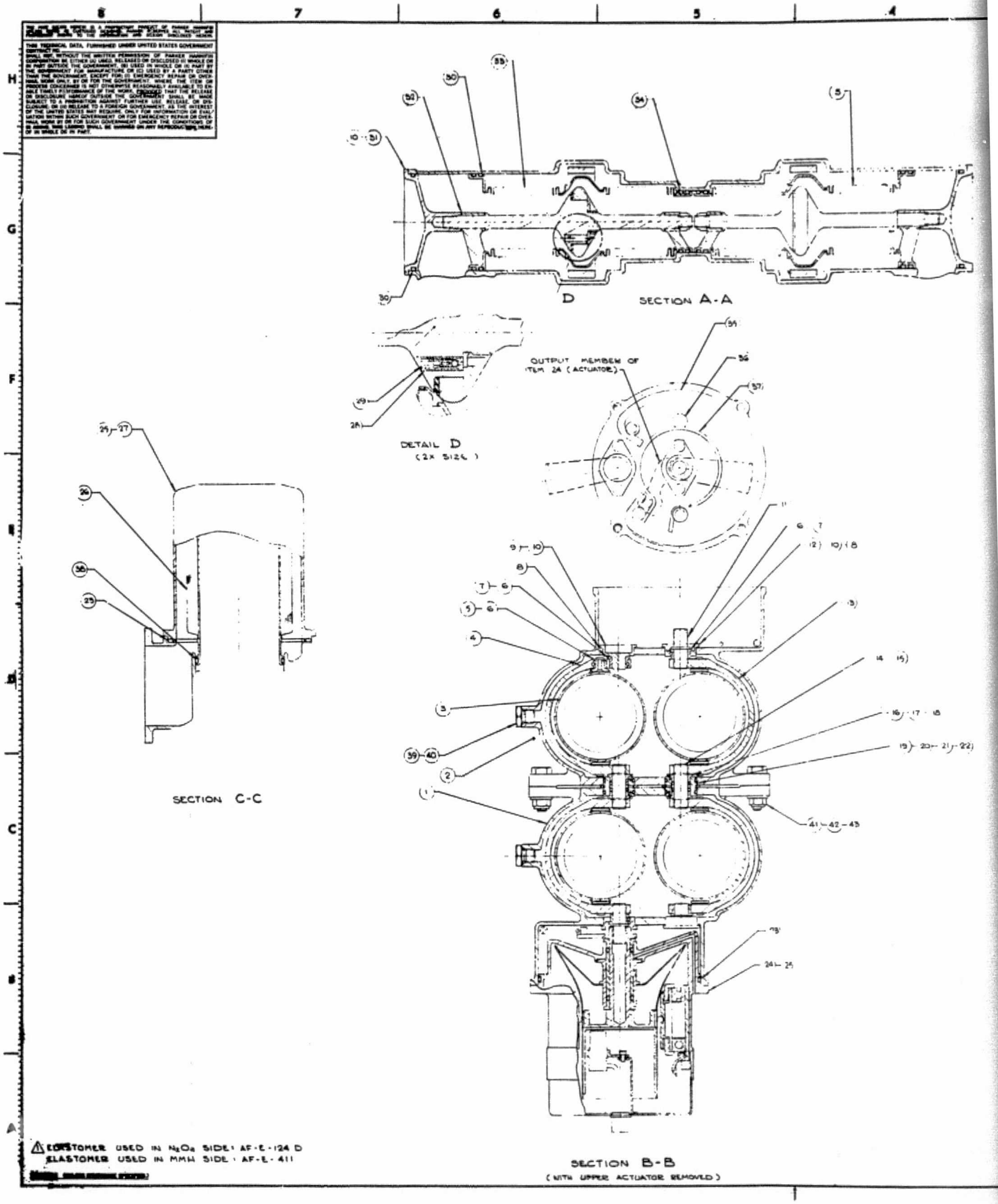

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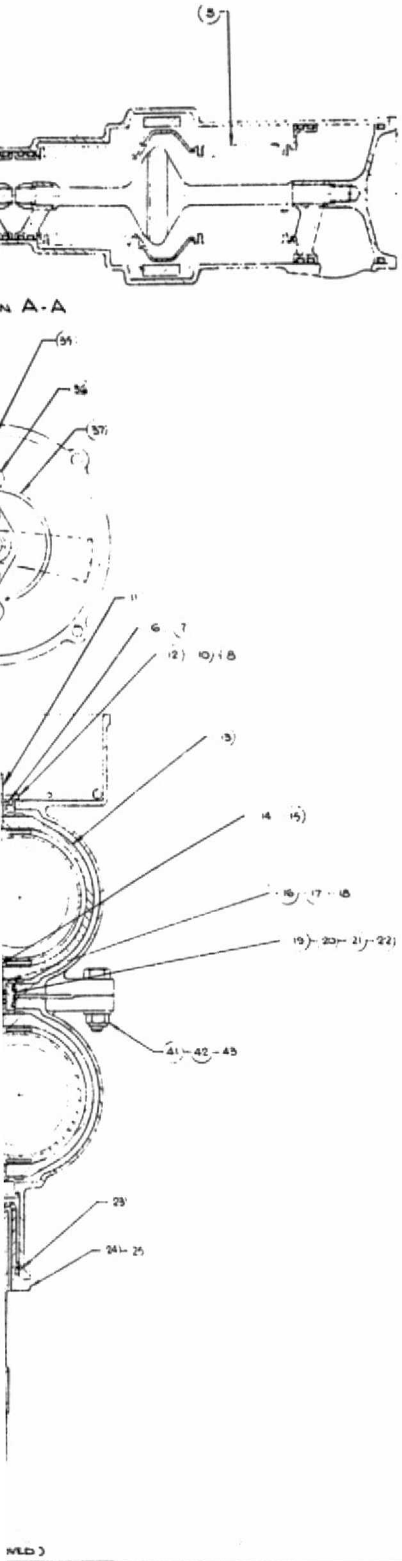
Figure 5-14. Preliminary Design Layout Quad Redund



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QTY REQD CONFIG NO	QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	CODE IDENT		DWG SIZE	ITEM NO.
4			NUT		CRES		43
4			WASHER		AL ALY		42
4			SHOULDER BOLT		CRES		41
3	3		PACKING		NOTED		40
6			PLUG		CRES		39
1	1		PACKING		NOTED		38
4			SPRING		CRES 18-8		37
6			GUIDE		CRES		36
4			SHOCK ABSORBER		CRES 18-8		35
8	8		PACKING		NOTED		34
2	2		VALVE MODULE		SEE ITEMS		33
AR			SHIM		CRES		32
8			CAP		CRES FOR N_2O_4 AL ALY FOR MMH		31
12	12		PACKING		NOTED		30
4			PIN		CRES		29
4			VALVE ASSY		BODY, GUIDE, SPRING, BALL: CRES; SEAT: TEFLON		28
2			HOUSING		CRES FOR N_2O_4 AL ALY FOR MMH		27
2			FILTER ASSY		CRES		26
28			SCREW		CRES		25
4			ACTUATOR		CORE: IRON SILICON STL; MOTOR MAGNET: ALNICO 5-7; [illegible]		24
1	1		PACKING		NOTED		23
4	4		PACKING		NOTED		22
4			RETAINING RING		CRES		21
8			SPIROL PIN		CRES		20
4			BRG RETAINER		CRES		19
4			SEPARATOR		TEFLON		18
4	4		PACKING		NOTED		17
8			BEARING		TEFLON		16
8			RETAINING RING		CRES		15
4			SHAFT		CRES		14
4			CRANK		CRES		13
4			BRG RETAINER		CRES		12
4			SHAFT		CRES		11
32			SCREW		CRES		10
4			CAP		CRES		9
AR			SHIM		CRES		8
4			BEARING		TEFLON		7
10			SPIROL PIN		CRES		6
16			ROLLER ASSY		RIVET, WASHER, ROLLER: CRES; BEARING: TEFLON		5
4			CRANK		CRES		4
2	2		VALVE MODULE		BELLOWS, BELLOWS ATTACHMENTS: INCONEL 718; POPPET COMPONENTS: CRES		3
1			BODY ASSY, N_2O_4		CRES		2
1			BODY ASSY, MMH		AL ALY		1

PARTS LIST

DO NOT SCALE DRAWING MUST CONFORM TO ES 1-1 DIMENSIONS AND TOLERANCING PER USASI Y14.5 MACHINED SURFACE 125 PER ASA B46.1 CLASSIFICATION OF CHARACTERISTICS CD PER SPI 735-3	UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES .XX ± .03 .XXX ± .010 ∠ ± 0° 30' MATERIAL	CONTRACT NO. 180	SYSTEMS DIVISION PARKER HANNIFIN IRVINE, CALIFORNIA
NEXT ASSY / USED ON APPLICATION	FINISH HEAT TREAT	PREP / CHK / DSGN / PROJ / APVD	PRELIM DESIGN LAYOUT QUAD REDUNDANT OMS VALVE
			SIZE E	CODE IDENT NO. 92003	DWG NO. 5736024
			SCALE FULL	UNIT WT	SHEET 2 OF 3

5736024

y Design Layout Quad Redundant OMS Valve - Assembly

Figure 5-14

Table V-9. Weight Analysis Breakdowns

Series Redundant (PN 5736023)	
Item Name/No.	Weight, lb
Al Aly Body/1	2.95
CRES Body/21	7.28
Al Aly Cap/31	0.245 each
CRES Cap/31	0.715 each
Al Aly Filter Housing/25	0.32
CRES Filter Housing	0.93
CRES Filter/24	0.855 each
Valve Module/28	3.71 each
Cranks, Roller/16, 18, 20	0.265 each
Shafts, Bearing Supports	1.580 total
Shock Absorber/36	0.16
Motor Actuator/32	3.46 each
Fasteners, Misc	1.00
Total	40.67

Quad Redundant (PN 5736024)	
Item Name/No.	Weight, lb
Al Aly Body/1	2.295
CRES Body/2	6.660
Al Aly Cap/31	0.078 each
CRES Cap	0.242 each
Al Aly Filter Housing/27	0.322
CRES Filter Housing	0.932
Filter/26	0.852 each
Valve Module/3	1.107 each
Cranks & Rollers/5, 6, 13	0.176 each
Shafts 0	0.840 total
Shock absorber/35	0.160
Motor Actuator/24	3.457 each
Miscellaneous Fasteners	1.500
Total	39.71

6.0 DETAIL DESIGN – LIFTING BALL VALVE AND ACTUATION SYSTEM

6.1 General

Although extensive design and analysis of the moving seat poppet valve showed it to be the most promising approach to avoid past ball valve problems, it became obvious that the valve concept was somewhat beyond the accepted state-of-the-art. Extensive demonstration testing would be necessary to qualify the concept to potential users. It was felt by both NASA and Parker-Hannifin that there was not time in the program to accumulate sufficient data to offset the extensive experience base accumulated in the APS, DPS, and SPS Ball Valve Programs, and that an alternate valve concept should be investigated for consideration. The concept selected was a "lifting ball valve" which appeared to have some advantages over the "moving seat valve," i.e., weight, size, and pressure drop.

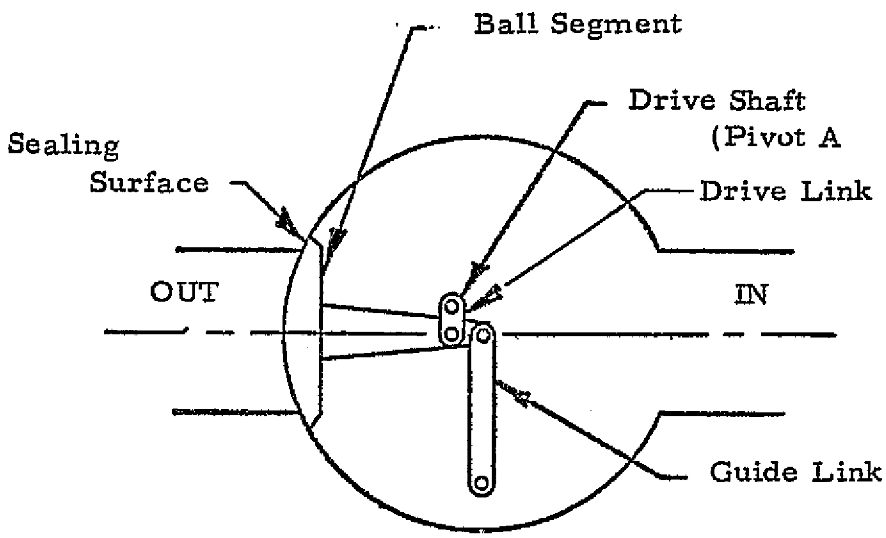
6.2 Lifting Ball Valve Description

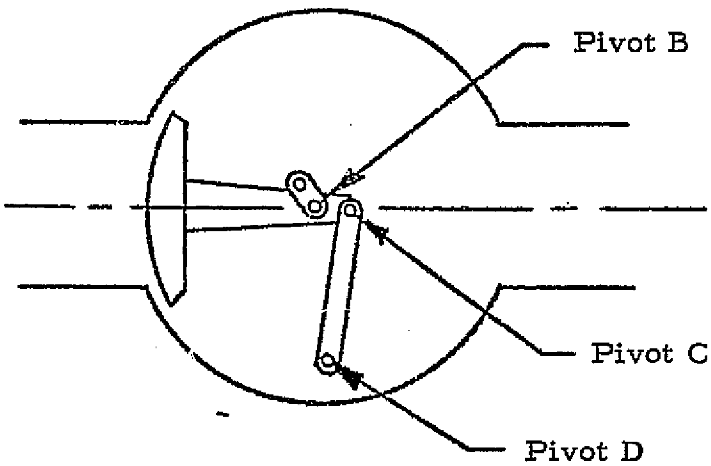
The basic principle of the lifting ball valve is to move the ball straight back off the valve seat, thereby eliminating any seat scrubbing. This action was accomplished by utilizing a four-bar linkage action within the valve to rotate the ball from the seat. The basic ball motion is described in Figure 6-1. With the valve in the closed position, (Figure 6-1, View A), the sealing surface of the ball rests on the seat and is both force- and pressure-loaded to the ball stop. With the actuation device off, a spring maintains the ball in the seated position. To open the valve, the actuation system rotates the drive shaft in the counterclockwise direction, which results in lifting the ball from the seat in a straight line, thereby eliminating any serious scrubbing of the seat. (See Figure 6-1, View B.) As the drive shaft continues to rotate, the ball is driven clear of the flow path. (See Figure 6-1, View C.) With the drive shaft rotated in the clockwise direction, the ball swings into the seating position and moves straight into the seat stop.

The basic lifting ball valve concept consists of an inlet port, housing, outlet port containing the valve seat, a ball that seals on the seat, internal linkage with which to position the ball, an input drive shaft which drives the linkage, and appropriate shaft and port seals. The detail design analysis was devoted to selecting the most optimum technique for operating the valve ball and also to make detail configuration and sizing decisions.

View A

Seated position - segment of ball rests on seat



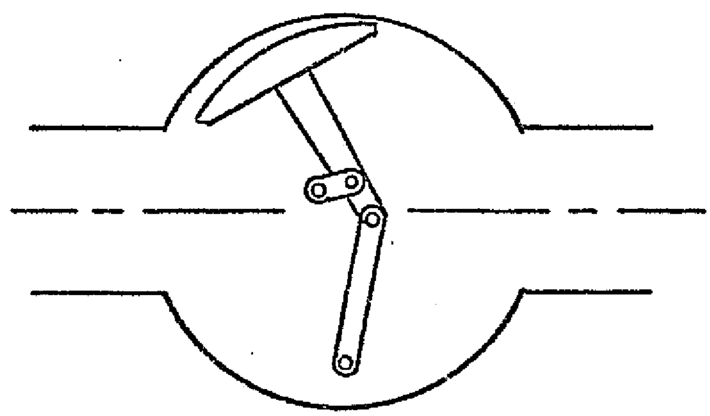


View B

Drive shaft starts to rotate, causing ball to lift clear of seal without rotation

View C

Drive shaft rotation complete. Ball is clear on flow path.



A four-bar linkage first lifts the ball clear of the seat, then rotates it out of the way.

Figure 6-1. Principle of Operation of the "Lifting Ball"

It was realized that it was not necessary to use a whole ball as a poppet; however, a study was performed to determine if a whole ball, partial ball, or some special hybrid configuration would be most optimum. As a result of this study, the hybrid configuration was selected. See Table VI-1 for a summary of the results.

Table VI-1. Ball versus Visor Design Study Results

Flow Control Element Concept	Peak Torque, in.-lb	K Factor	Weight	Manufacturing Cost
Ball	43.0	0.056	Higher	Lowest
Visor	29.0	0.263	Lower	Intermediate
Hybrid	29.0	0.056	0.3 lb	Highest

6.3 Lifting Ball Valve Assembly — Analysis and Preliminary Design

6.3.1 Ball/Poppet Shape Study — The most dramatic result of this study was the significantly lower peak operating torque for the visor approach: 29.0 in.-lb versus 43 in.-lb for the ball. Figure 6-2 is a computer plot showing these peak torques, as well as the entire operating torque profile throughout the valve opening rotation. The hybrid approach had a peak torque equal to the visor concept. (The analysis results are from a computer program that considered the following design and operating parameters:)

1. All linkage dimensions
2. The ball (or visor)
3. Friction of all bearings
4. Shaft seal friction
5. Pressure drop force
6. Aerodynamic (flow) forces

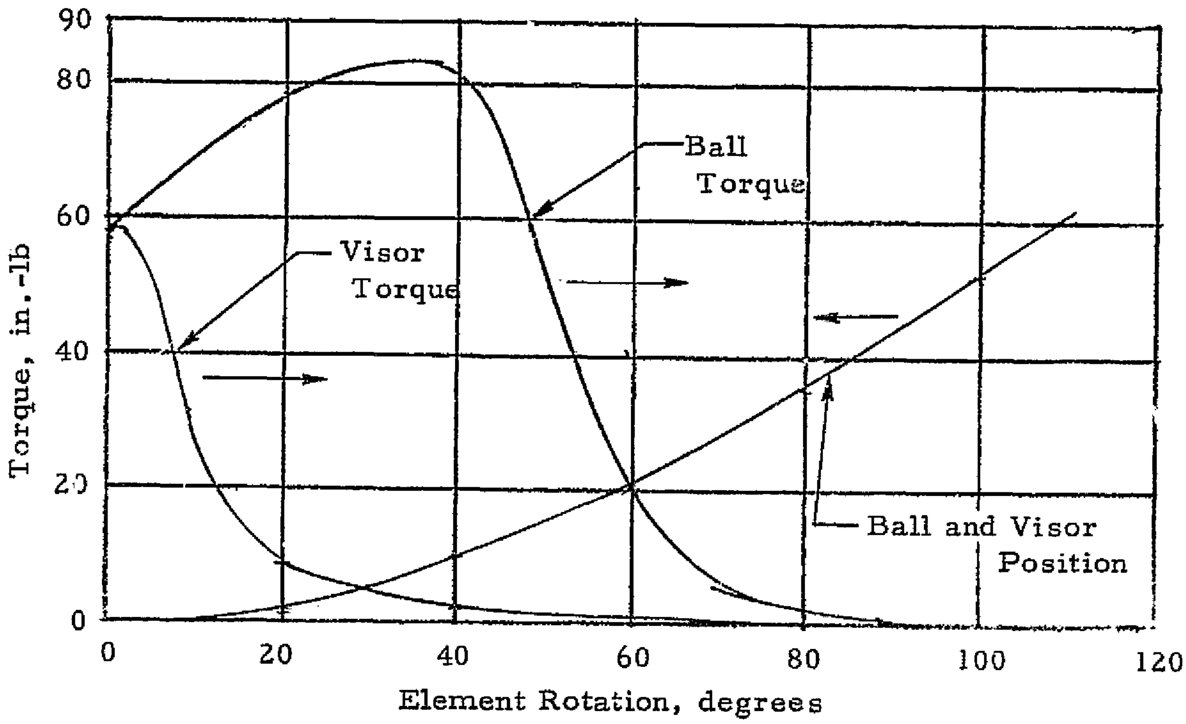


Figure 6-2. Ball Versus Visor Operating Torque and Linkage Gain

As shown in Table VI-1, the visor type flow control element has a significantly higher K-factor than that of the ball. By using the hybrid approach, where a flow tube is placed in the center of the visor, the much lower pressure drop characteristic (K-factor) of the ball valve can be achieved, while maintaining the low operating torque of the visor.

The deficiencies of the hybrid concept are minor compared to the operating torque and pressure drop advantages. As shown in Table VI-1, the hybrid weight is higher than a visor (but not much - only that contributed by the thin-walled flow tube), and the manufacturing cost is highest.

The hybrid configuration is shown in Figure 6-3. The action of the valve ball seating stroke automatically positions the flow-through tube in the valve.

6.3.2 <u>Valve Sizing Analysis</u> — A preliminary valve sizing analysis was performed to determine the approximate valve size needed to satisfy the valve pressure drop criteria. Table VI-2 summarizes the basic criteria and results of this study. The analysis is included in Appendix C.

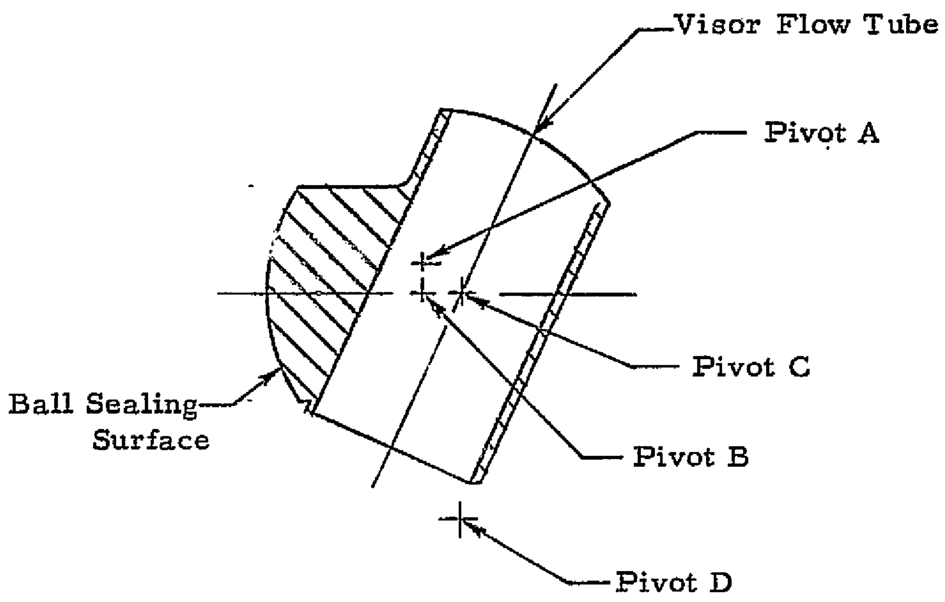


Figure 6-3. Hybrid Configuration

Table VI-2. Preliminary Valve Sizing Study

Condition	Total N_2O_4 Flow Rate (lb/sec N_2O_4)	Allowable ΔP (psid max)	Valve Size
1. All Valves Open	11.91	5	0.625
2. 1 leg of Quad Open	11.91	10	0.750
3. 1 leg of Quad Open	11.91	5	0.900

Following this study, the valve (internal diameter) size of 0.900 inches was tentatively selected for subsequent design and analysis tasks. This was done since it is quite desirable to maintain the valve pressure drop at or about 5 psid in the failed condition to minimize the change in engine performance due to valve failure.

6.3.3 Preliminary Configuration Layout — A basic configuration layout was made of the preliminary design configuration. A schematic of the valve configuration is included as Figure 6-4. Also included on the drawing is a N_2O_4 valve and an MMH valve mechanically linked together using a Geneva mechanism. A spring/pressure energized Teflon seat and redundant spring/pressure energized Teflon shaft seals are used. Rulon journal bearings are used to take all rotating bearing loads. Only one separate "link" is used (this is the "guide link") as other "linkages" are provided by eccentricities in the shaft and rotor. External coil springs are used to provide mechanical seat preload.

A static torque analysis computer program was used to establish the motion characteristics of the rotor, to determine the input shaft versus rotor position characteristic, and to establish the input shaft operating forces.

The program output described the input shaft to rotor position characteristic, the opening and closing direction torques as measured at the valve input shaft. Torques could be determined with or without the effect of bearing friction.

Figure 6-5 shows the input shaft position versus rotor position characte[illegible]. This shows that 90 degrees of valve shaft rotation causes the rotor [illegible]ate 65 degrees, which is sufficient to completely open the valve. Note that the "shaft to rotor" rotation gain is very high at valve closed position (at the instant of closure it is infinite) and reduces as the valve opens, until, at the valve open position, the gain falls off to 0.6. This characteristic eliminates seal scrubbing, since the rotor has a negligible rotational component as it enters the seat. Note that as the valve shaft rotates its final five degrees, the rotor rotates only 0.2 degrees. Although five degrees of input shaft rotation represent a rotor lift of 0.011 inch, the next side motion of the rotor with respect to the seal is less than 0.005 inch. Since the seat deflection will amount to less than 0.011 inch, the total "sliding contact" on the seat will be well under 0.005 inch. This is negligible when compared to the sliding contact of conventional ball valves, and is probably no more than occurs in many equivalent size poppet valves during the seat alignment process.

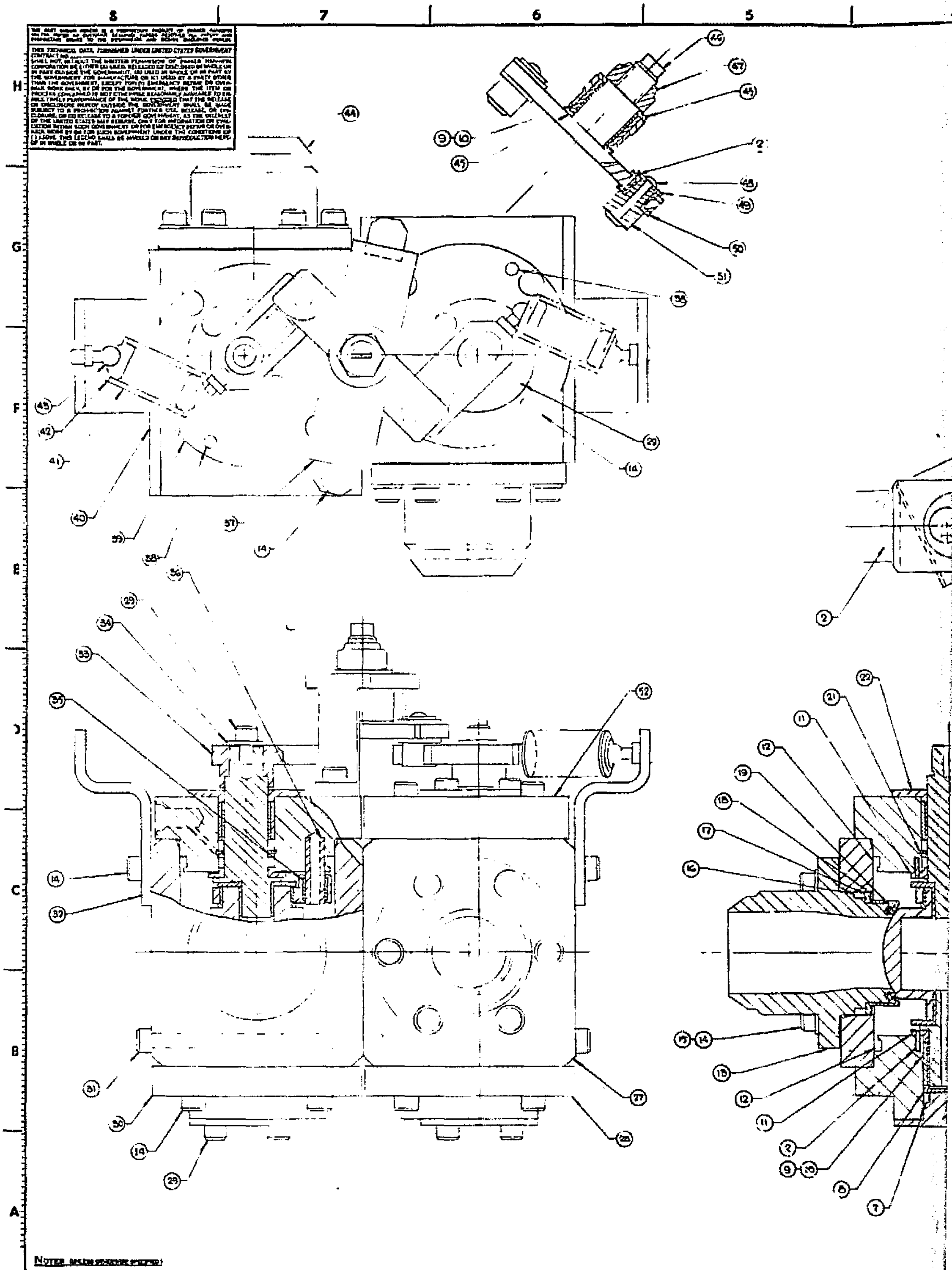


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Figure 6-4. Preliminary Design Lay

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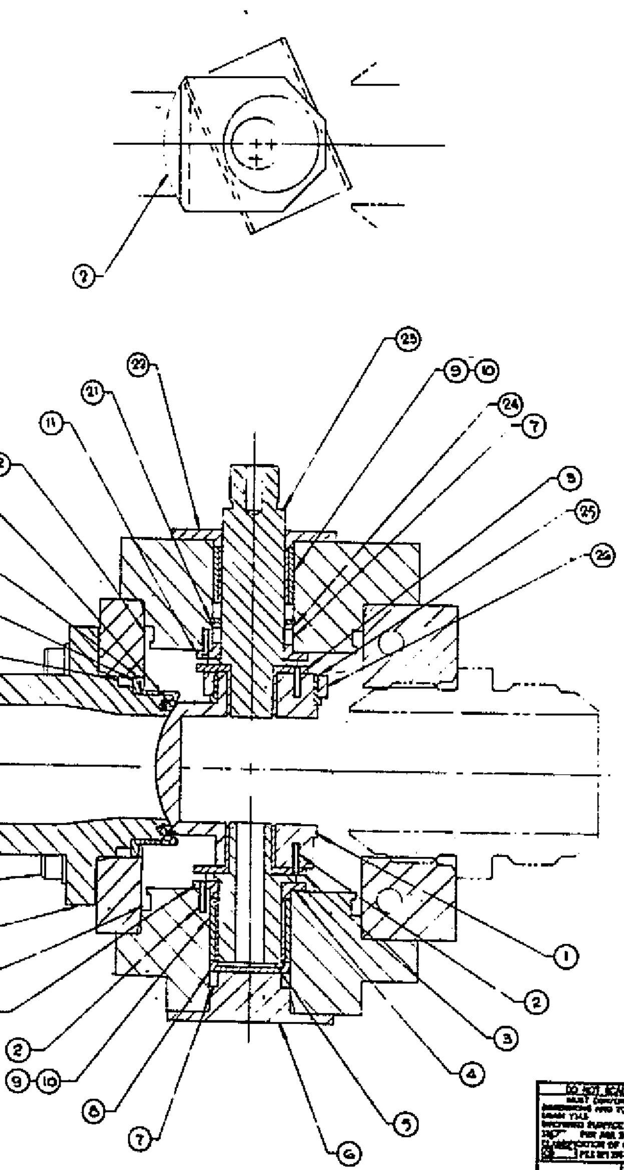

QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	CODE IDENT		DWG SIZE	ITEM NO.
1		COVER, UPPER	CRES			52
2		ROLLER	CRES			51
4		BEARING	RULON			50
2		WASHER	CRES			49
2		RIVET	CRES			48
2		NUT	CRES			47
1		YOKE	CRES			46
2		THRUST WASHER	RULON			45
1		FITTING	AL ALY			44
4		BALL SCREW	CRES			43
4		RETAINER	AL ALY			42
2		SPRING	CRES			41
1		BODY	AL ALY			40
1		COVER, UPPER	AL ALY			39
4		DOWEL	CRES			38
1		BRACKET	AL ALY			37
2		PIN	CRES			36
2		BEARING	RULON			35
2		WASHER	CRES			34
2		CAM	CRES			33
2		BRACKET	AL ALY			32
4		SCREW	CRES			31
1		COVER, LWR	AL ALY			30
18		SCREW	CRES			29
1		COVER, LWR	CRES			28
1		BODY	CRES			27
2		LINK	CRES			26
2		BEARING	RULON			25
2		WASHER	CRES			24
2		SHAFT, UPPER	CRES			23
2		RETAINER	CRES			22
2		RETAINING RING	CRES			21
						20
2		SEAL	TEFLON W CRES SPRING			19
AR		SHIM	CRES			18
2		SEAL RETAINER	CRES			17
2		SEAL	TEFLON W CRES SPRING			16
34		WASHER	CRES			15
34		SCREW	CRES			14
1		FITTING	CRES			13
4		SEAL	TEFLON W CRES SPRING			12
4		BRG, THRUST	RULON			11
5		BRG RETAINER	CRES			10
5		BEARING	RULON			9
2		RETAINER	CRES			8
6		SEAL	TEFLON W CRES SPRING			7
2		CAP	CRES			6
2		SHAFT, LWR	CRES			5
AR		SHIM	CRES			4
4		BEARING	RULON			3
10		SPIROL PIN	CRES			2
2		VISOR	CRES			1

PARTS LIST

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MATERIAL

FINISH

HEAT TREAT

NEXT ASSY | USED ON

APPLICATION

CONTRACT NO. 180

PREP		
CHK		
DSGN		9-21-73
PROJ		9-21-73
APVD		

PARKER HANNIFIN IRVINE, CALIFORNIA

PRELIM DESIGN LAYOUT PROTOTYPE LIFTING BALL VALVE

SIZE	CODE IDENT NO.	DWG NO.
E	92003	5730025

SCALE | UNIT WT | SHEET 1 OF 1

5730025

...ary Design Layout Prototype Lifting Ball Valve

Figure 6-4

LDOUT FRAME 2

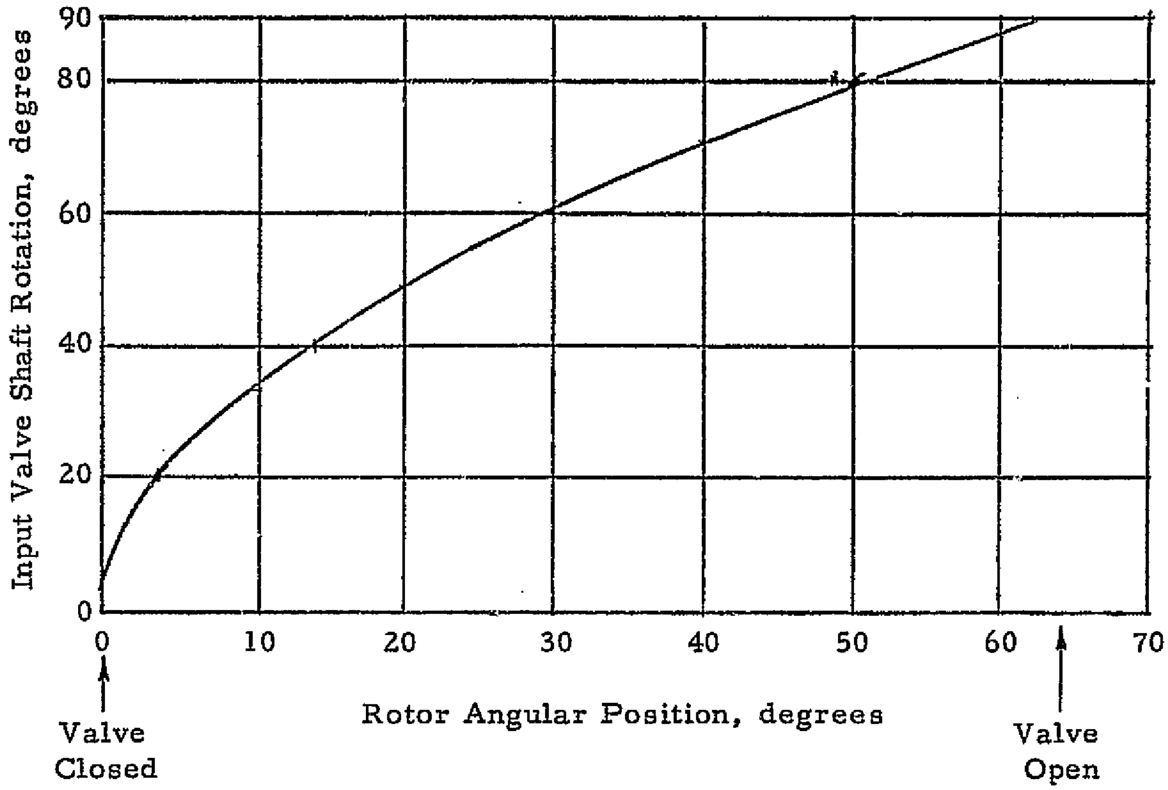

Figure 6-5. Input Valve Shaft Versus Rotor Rotation

Figure 6-6, resulting from the same computer program, shows torque acting on the rotor during opening and closing. These torques are generated by the total of pressure unbalance, flow forces, bearing friction, and shaft seal friction. Figure 6-7 again shows the valve torques, except as acting on the valve input shaft. Therefore, the valve actuator must generate opening torques in excess of those shown in Figure 6-7.

Subsequent analysis of the valve configuration resulted in minor modifications to the preliminary design concept. The Geneva type mechanism presented previously, to link the two valves together, has proven to have an unsatisfactory mechanical advantage condition that requires high power. A cam design was studied and although it cut the required motor power, it was bulky and difficult to package. The final solution is a 4-bar linkage configuration. It was also necessary to drive both rotor-to-body bearings from the input shaft to evenly distribute the bearing loads. The best approach for this was to run the drive shaft through the

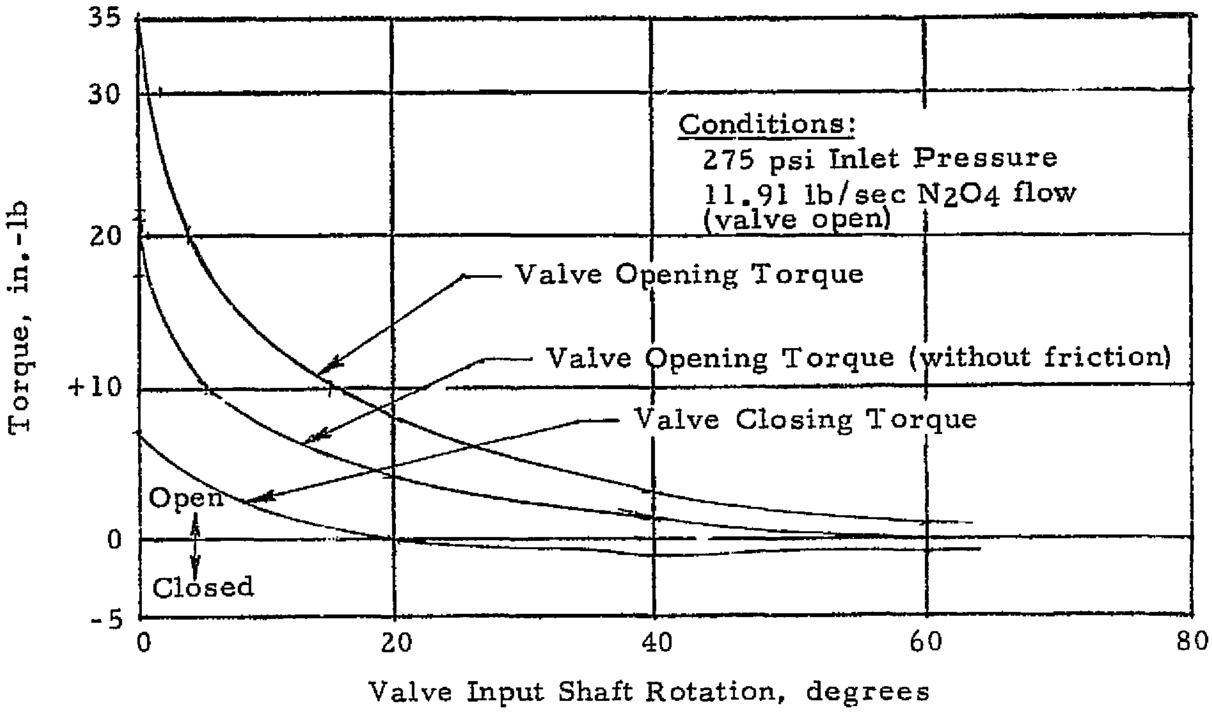

Figure 6-6. Rotor Torque

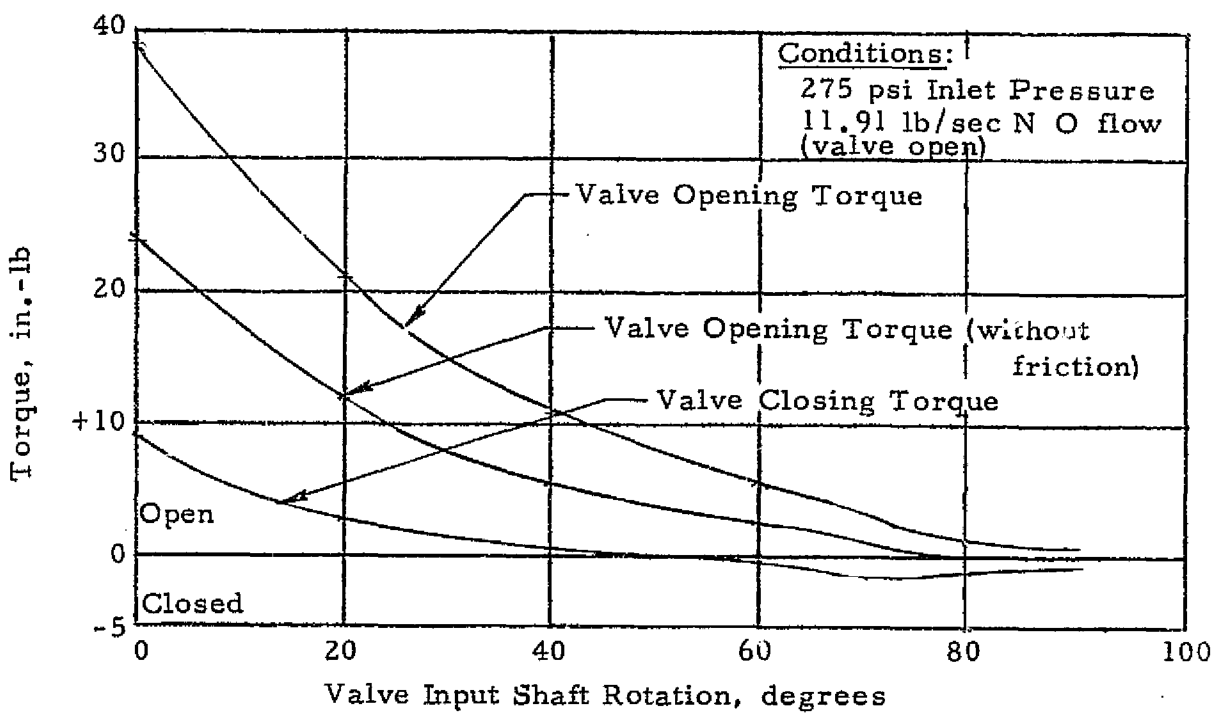

Figure 6-7. Input Valve Shaft Torque

flow path. To accomplish this, a fluid flow path had to be provided in the valve drive shaft. This was accomplished by eloxing a hole through the shaft in such a manner that the shaft would meet the required torque loading and maintain a minimum ΔP for the valve. Refer to Appendix C for shaft ΔP calculations. The valve drive shaft change was also accompanied by the replacement of the Rulon journal bearings with a Duplex pair of ball bearings on the drive side of the shaft, and an electronize treatment of all rotating, rubbing, surfaces when friction is encountered. The basic lifting ball valve design is shown in Figure 6-8.

6.3.4 Valve Pressure Drop Analysis — Valve pressure drop was calculated and is presented in Table VI-3.

Table VI-3. Valve Pressure Drop

Condition	Fuel (MMH at 7.22 lb/sec)	Oxidizer (N_2O_4 at 11.91 lb/sec)
Both parallel legs open	0.647 psid	1.071 psid
Only 1 leg open	2.591 psid	4.283 psid

Pressure Drops as listed in Table VI-3 are for the following conditions:

1. The pressure drop is the total for two valves in series.
2. The pressure drop does not include the manifold which connects the two legs of the "quad" together in parallel.
3. The pressure drop is that which would be obtained when tested with inlet and outlet lines equal to the valve bore diameter (0.9 inch).

Complete details of the above pressure drop analysis are given in Appendix C.

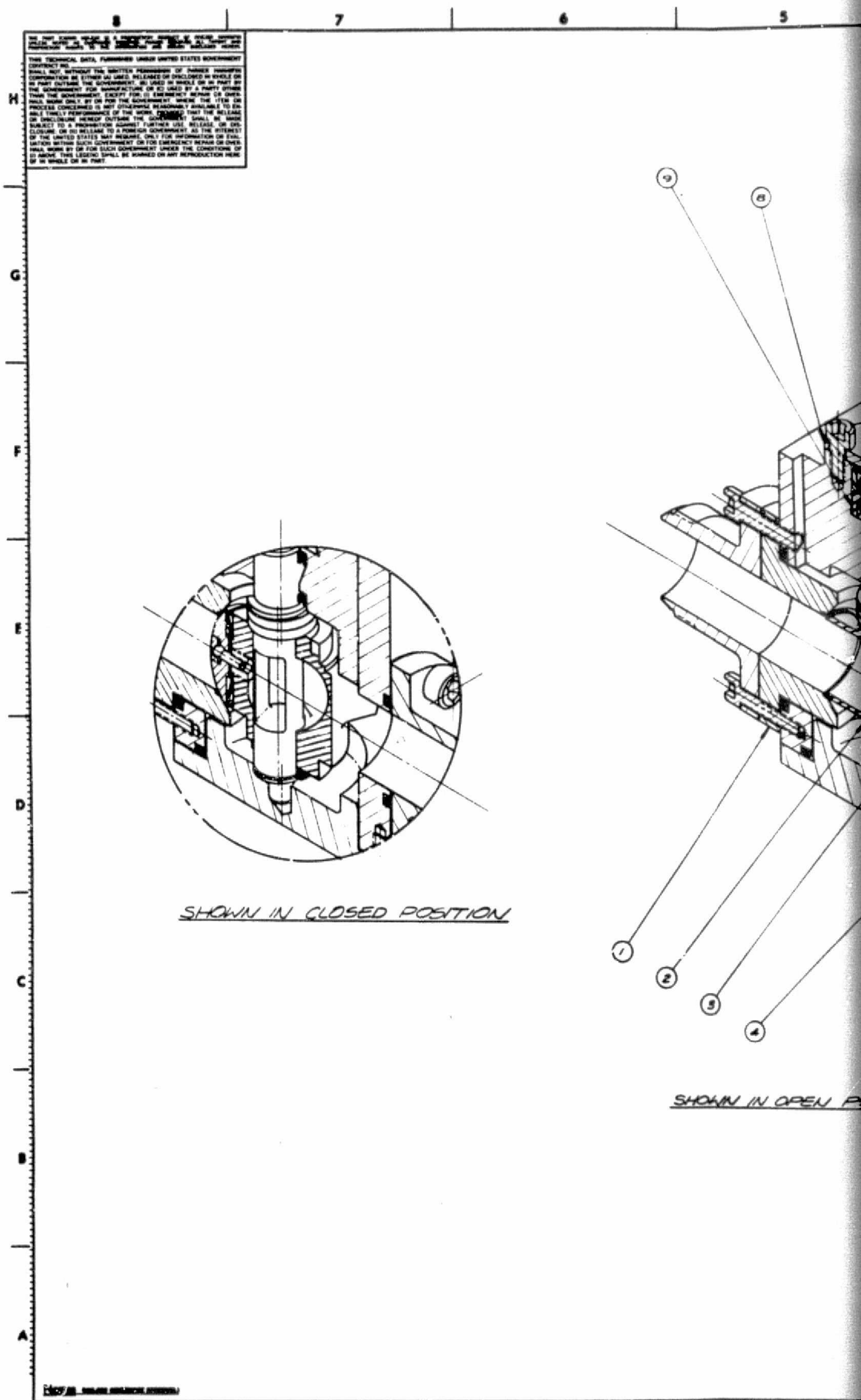

Figure 6-8. Lifting Ball Valve (P

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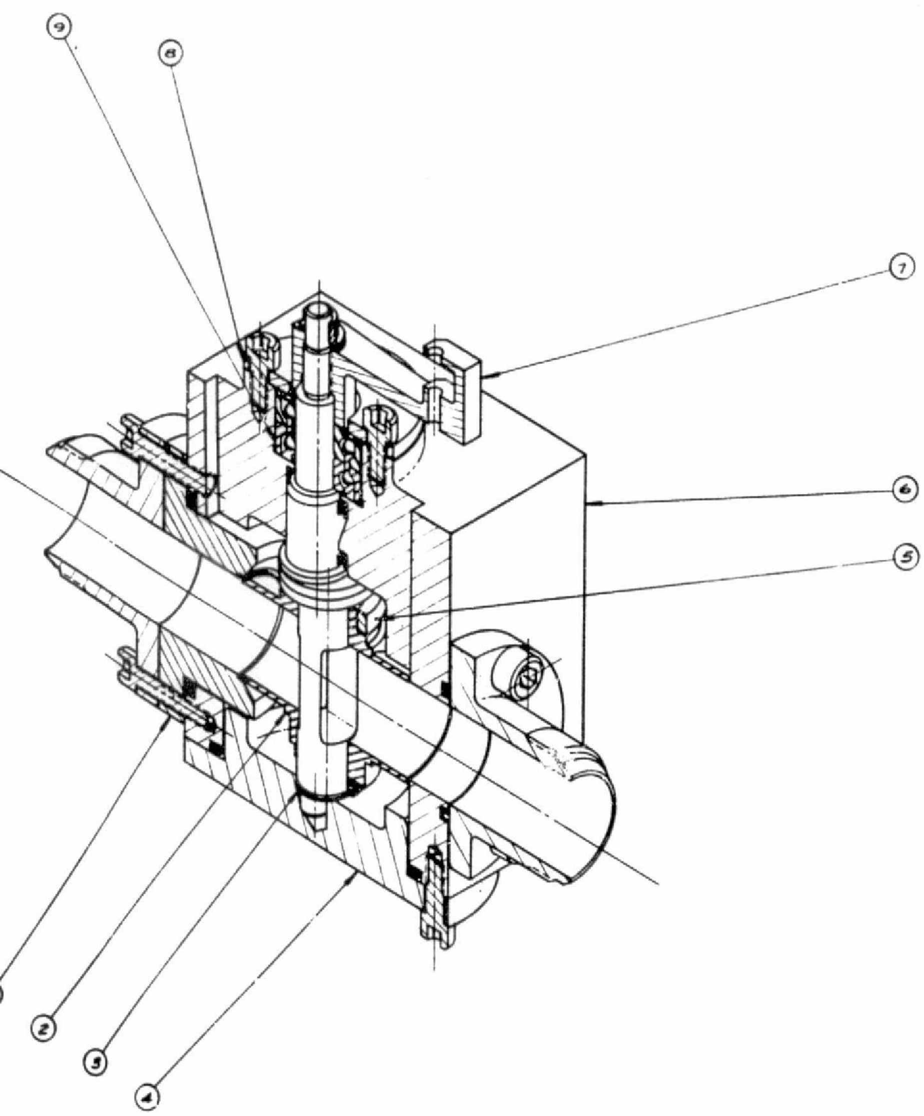
—		RELEASED		



SHOWN IN OPEN POSITION

QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	CODE IDENT		DWG SIZE	ITEM NO.
2	OLOODTL	BEARING		NEW DEPARTURE	—	9
1	5736080-1	BEARING RETAINER				8
1	5736081-1	LEVER				7
1	5736071-1	HOUSING				6
1	5736089-1	LINK				5
1	5736075-1	COVER				4
1	5736072-1	SHAFT				3
1	5736061-101	VISOR ASSY				2
1	5736070-101	SEAT ASSY				1
—	5739001M1	BALL VALVE, LIFTING			E	—

PARTS LIST

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES TOLERANCES

MATERIAL

FINISH

HEAT TREAT

CONTRACT NO.

PREP | CHK | DSGN | PROJ | APVD

NEXT ASSY | USED ON

APPLICATION

BALL VALVE, LIFTING (PROTOTYPE)

SIZE E | CODE IDENT NO. 92003 | DWG NO. 5739C01M1

SCALE | UNIT WT | SHEET 1 OF 1

5739001M1

ifting Ball Valve (Prot otype)

Figure 6-8

FOLDOUT FRAME 2

There is some plumbing required for operation of the valves as a complete "quad." This plumbing consists of:

1. Diverging "Y."

2. Reduction in area from system tube I.D. to the 0.900-inch I.D. of the valve

3. Expansion in area from the 0.900-inch I.D. of the valve to the system tube I.D.

4. Converging "Y."

The pressure drop of this plumbing will vary with the details of its design, but it can easily have much more pressure drop than the valve itself. As an example, we analyzed the plumbing arrangement shown in Figure 6-9 and calculated the pressure drop with the results presented in Table VI-4.

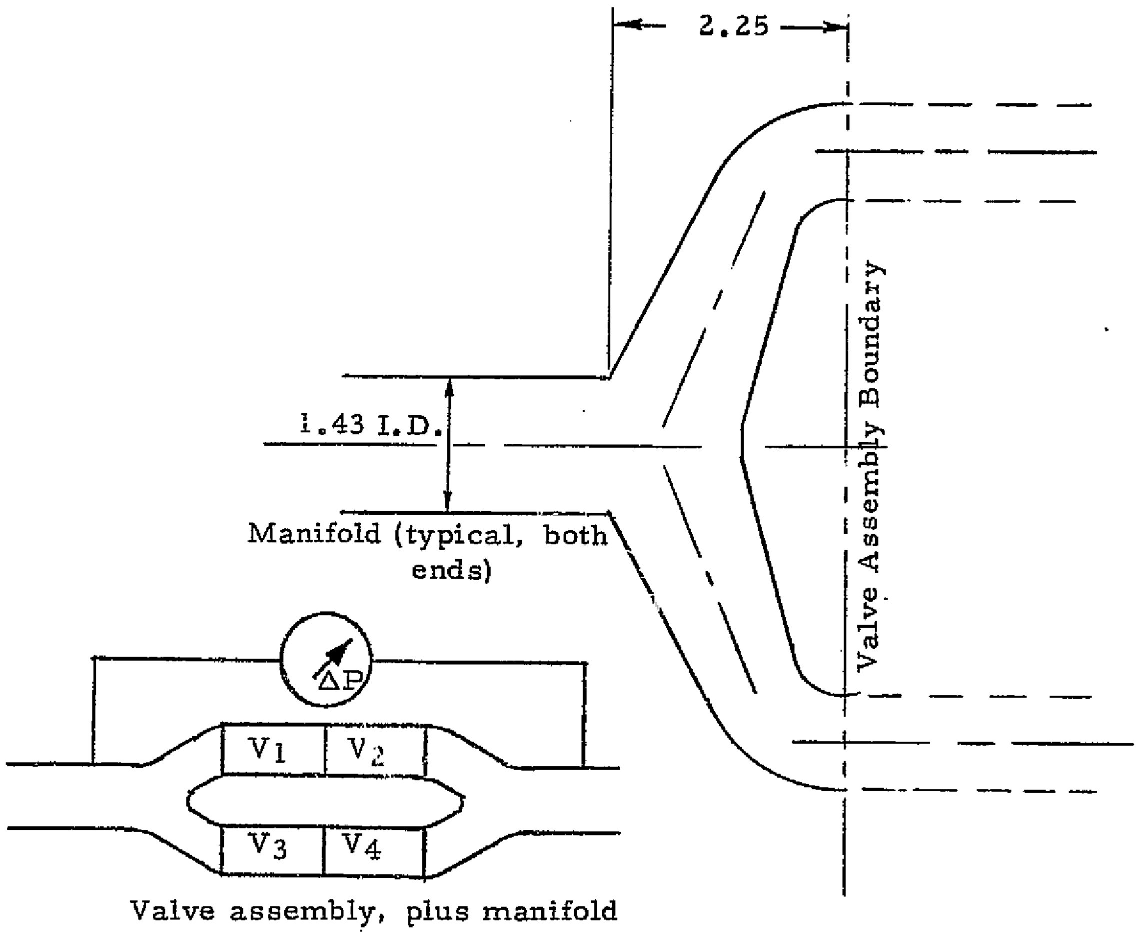


Figure 6-9. Manifold Configuration

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Table VI-4. Pressure Drop Valve and Manifold Assembly

Condition	Fuel (MMH at 7.22 lb/sec)	Oxidizer (N_2O_4 at 11.91 lb/sec)
Only one leg open	5.92 psid	9.80 psid
Both parallel legs open (approximately)	1.48 psid	2.45 psid

It is seen that the pressure drop of the complete quad is roughly twice that of the valve alone, for the example analyzed. A more compact or less streamlined manifold than shown in Figure 6-9 would increase this difference. A more streamlined and less compact manifold would, of course, reduce the pressure drop.

6.3.5 Visor Angle Versus Valve ΔP — The new valve design, as shown in Figure 6-8, was subjected to the computer program analysis to evaluate the visor angle versus valve pressure drop for the first six degrees of valve opening. The analysis was performed for 173, 211, and 275 psig. The resulting data is shown in Figure 6-10.

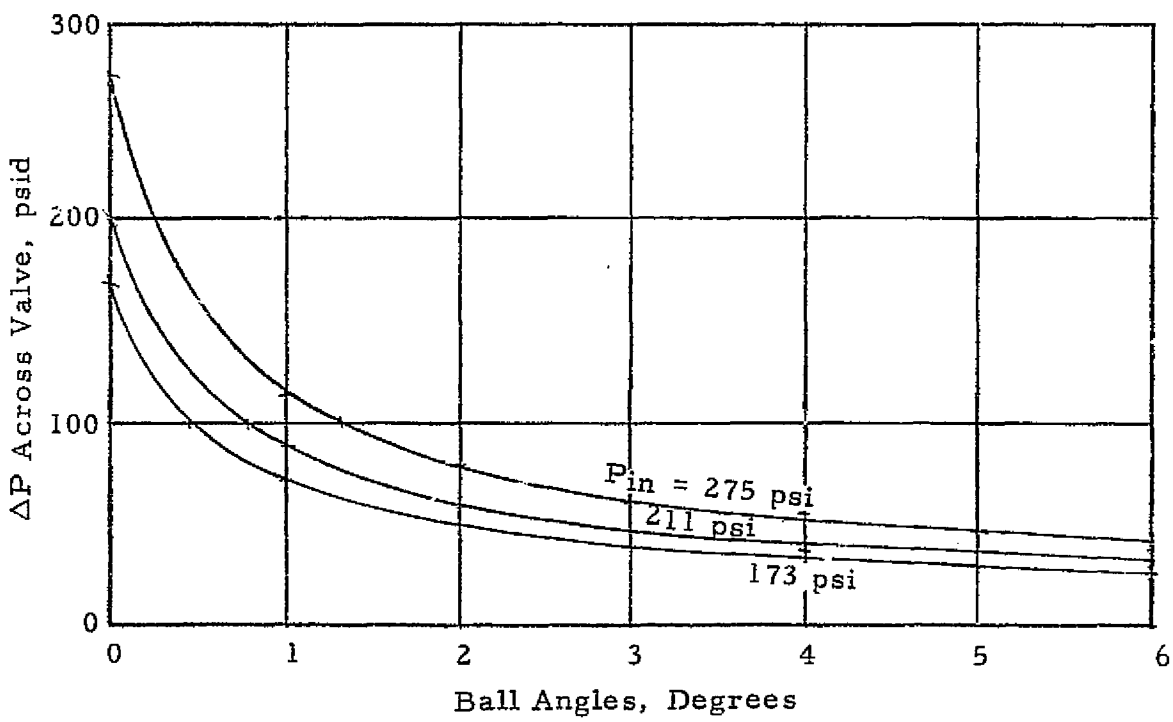
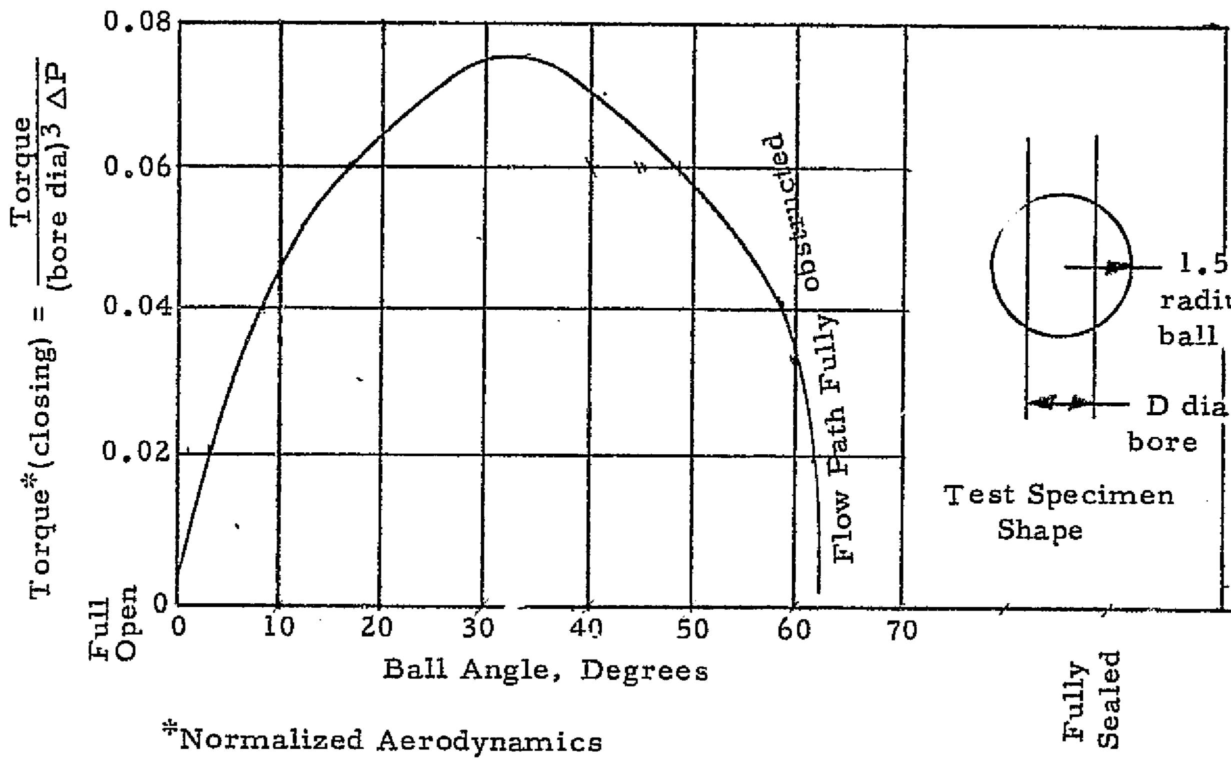


Figure 6-10. Valve ΔP versus Visor Angle

6.3.6 Valve Torque Analysis — A valve torque analysis was conducted and consists of the following:

The basic forces acting on the valve are:

a. Aerodynamic torque. This torque varies with valve angle and tends to rotate the "ball" to the closed position. Typical aerodynamic torque data (from Parker-Hannifin tests) are shown in Figure 6-11.



In addition to a thrust force tending to seat the ball, ball valves have an aerodynamic torque tending to rotate the ball closed. Data from Parker-Hannifin Report S62R9521, illustrated in this graph, was used to analyze the aerodynamic torque acting on the proposed valve.

Figure 6-11. Ball Valve Aerodynamic Torque Data

b. Thrust. This force acts through the center of rotation of the ball and tends to close the valve. The thrust force is obviously equal to times the square of the seat radius times the pressure drop, when the "ball" has not yet rotated. When the ball has fully rotated out of the stream, the thrust force is obviously zero. For intermediate angles, the thrust force was estimated by assuming the valve pressure drop to act over the area of the "ball" exposed to the flowing stream, with the direction of the force acting through the center of pressure.

Neither thrust force nor torque can be computed without knowledge of the pressure drop across the valve. This depends on the flow resistance of upstream and down stream elements in the system as well as the valve itself. For example, the flow rate can certainly not be assumed to be rated flow when the valve is in a partly open position.

For purposes of analysis, we assumed a constant inlet pressure of 275 psig. We assumed that the thrust chamber and injector could be represented by a fixed orifice which causes the specified rated flow when the valve is in the fully open position. We believe this assumption to be accurate enough for all practical purposes.

The maximum valve pressure drop, and therefore maximum torque, occurs when opening one valve, with the other valve in that leg already open, and the valves in the parallel leg still closed. We used this worst case for analysis of required torques.

Forces are applied to open the valve by means of a four-bar linkage. Note that the linkage must first lift the ball and then rotate it, in one continuous motion. The equilibrium of forces acting on the linkage was analyzed by means of Parker-Hannifin computer program S297 and graphs of the results prepared.

The first graph, Figure 6-12, shows how the "ball" angle varies with the valve shaft angle. Note that there is almost no rotation of the ball for the first 10 degrees of rotation of the input shaft. This is the time when the ball is being lifted from the seat. The basic torque required at the valve shaft is shown in Figure 6-13. This torque overcomes the thrust and aerodynamic torque acting on the ball. Notice that the valve is basically self-closing, since an opening force is always required to hold the valve in equilibrium. Figure 6-14 shows ball side motion for lift positions. This graph indicates that the ball has only 0.00025-inch seat scrubbing action in the first 0.005 inch of lift.

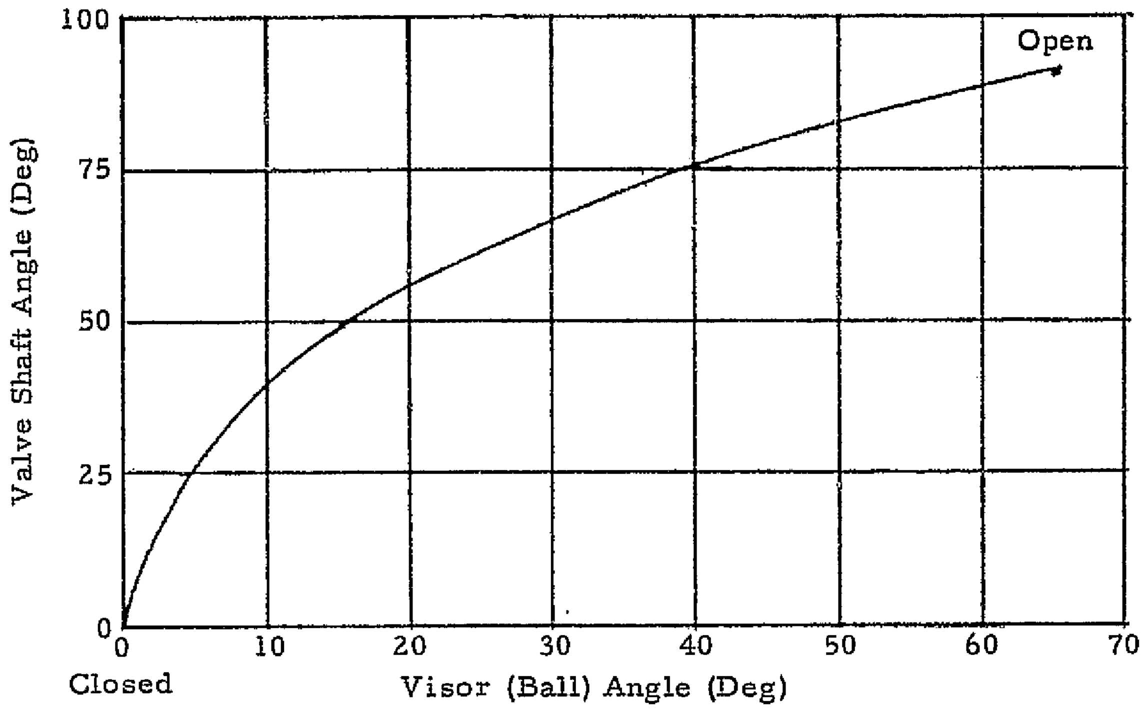


Figure 6-12. Valve Shaft Angle versus Visor Angle

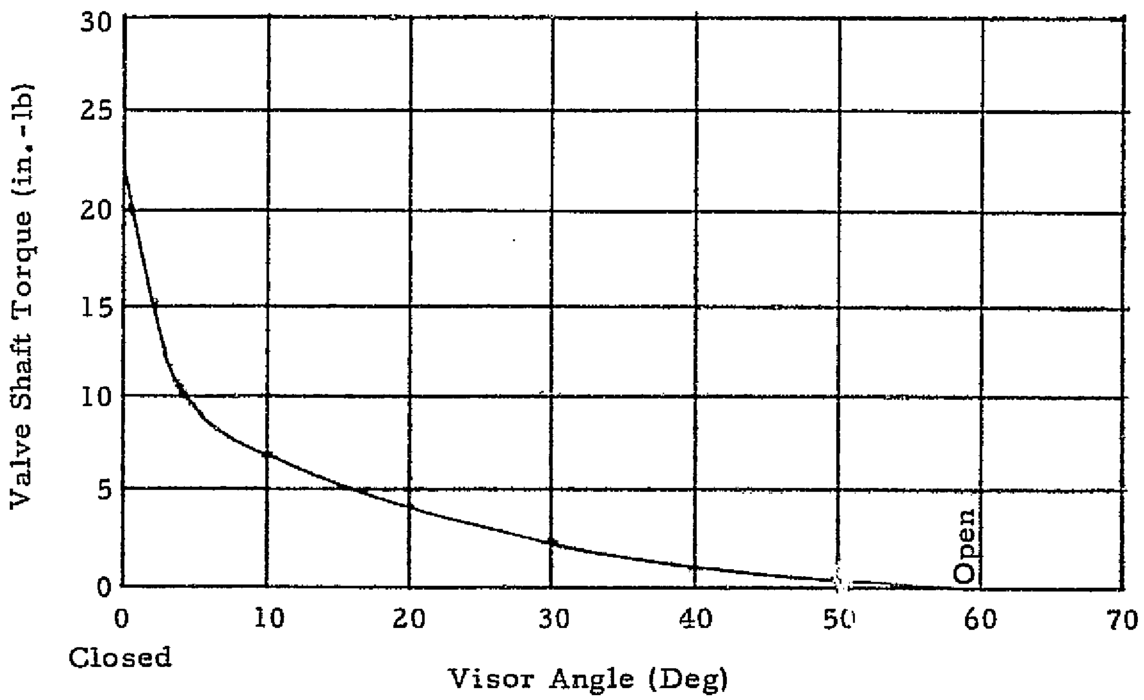


Figure 6-13. Valve Shaft Torque versus Visor Angle

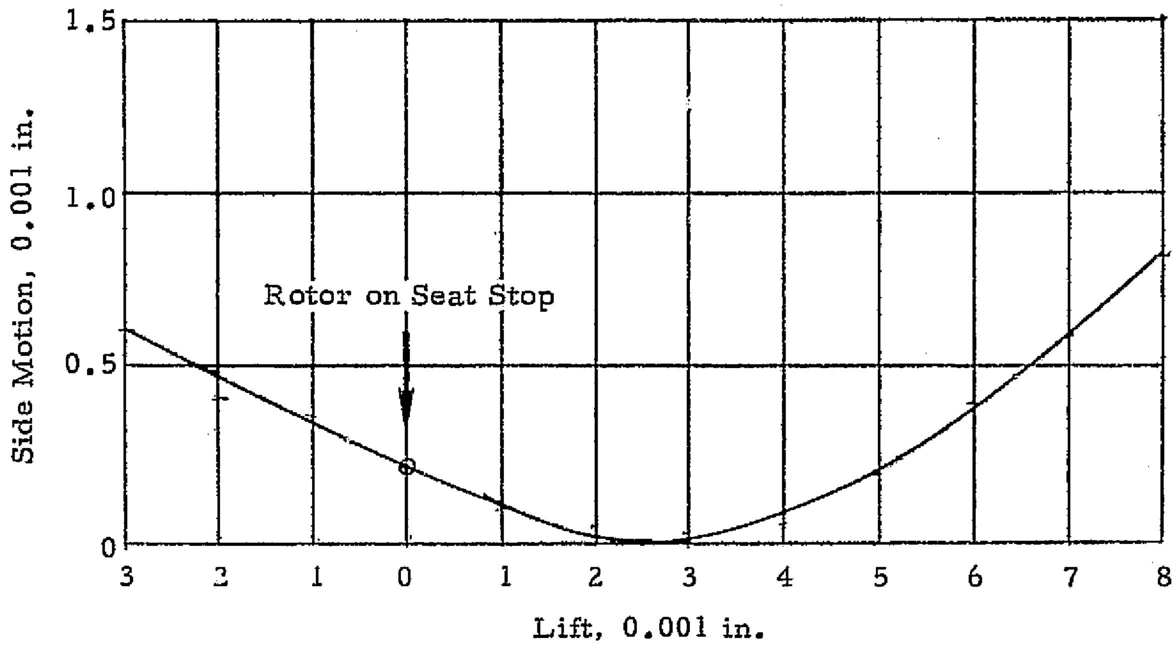


Figure 6-14. Ball Side Motion versus Lift Position

Friction forces were then included and torque computed with the same program. The results are shown in Figure 6-15. Note that the curve contains a "hysteresis" loop due to the fact the friction forces oppose motion in either direction. The bearings were assumed to have a coefficient of friction of 0.2, a conservative value for any bearing suited to the cycle requirements. The coefficient of friction of the ball bearings, etc, is negligible by comparison and was not included. The shaft seal friction torque used was 1.04 in.-lb.

As seen in Figure 6-15, the valve is still basically self-closing, although a slight closing force must be applied to assist it if the friction forces are as large as assumed.

The results of the analysis as shown in Figure 6-15 indicate that the valve driving torque requirement must be approximately 35.5 lb-inches. This torque must be available for the initial rotation of the valve drive shaft; however, the requirement reduces to 8.5 lb-inches within the first 20 degrees of drive shaft rotation.

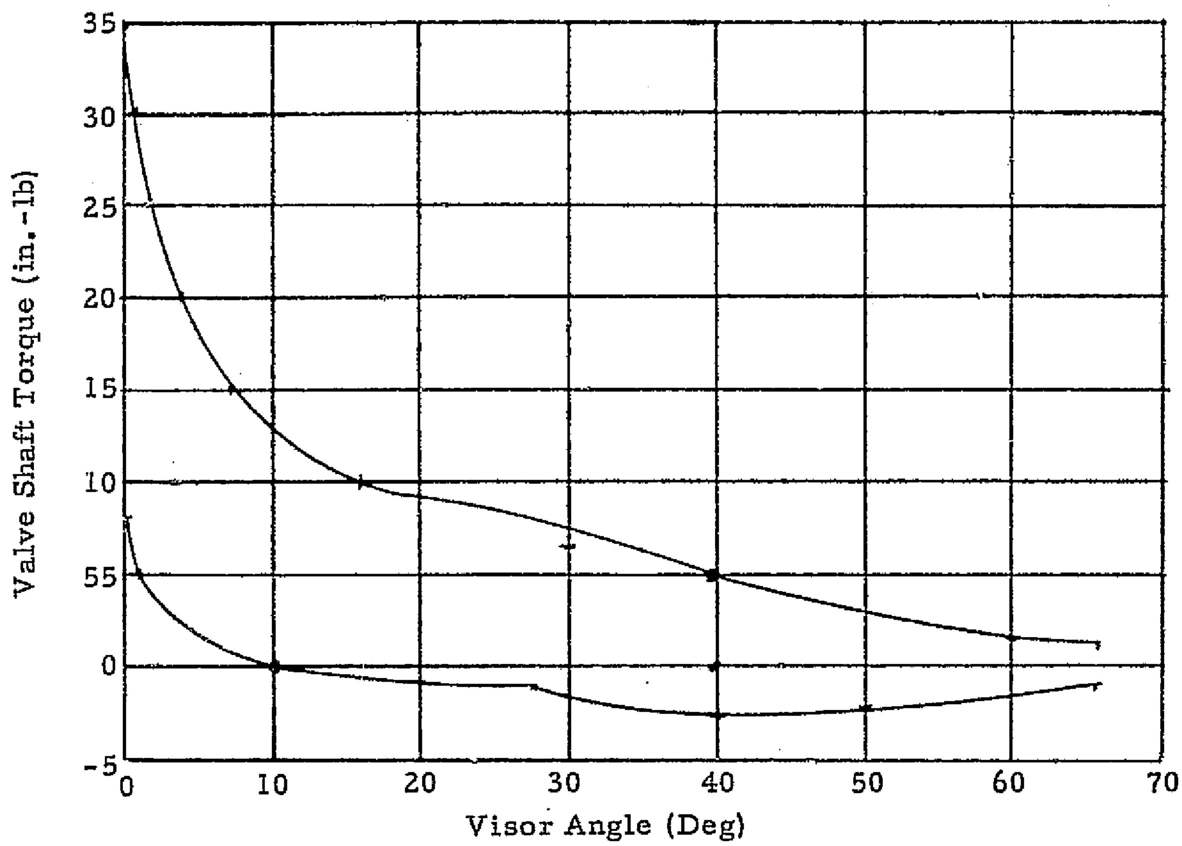


Figure 6-15. Valve Shaft Torque versus Visor Angle (with Maximum Friction)

6.4 Actuation System for Lifting Ball Valve

6.4.1 Actuation System Concept Evaluation — Lifting Ball Valve. The actuation system concept, as developed for the moving seat valve, was reassessed for application with the lifting ball valve. Figure 6-16 presents three candidate configurations considered. Concept (a) is similar to that used to actuate the moving seat poppet valve. However, a substantial return spring must be added (to insure valve "fail-closed" position with loss of electrical power) for use with the lifting ball valves; the moving seat concept utilizes bellows which inherently supply this torque (up to about 160 in.-lb). It became obvious that this spring would be quite large and would prove very difficult to damp under vibration.

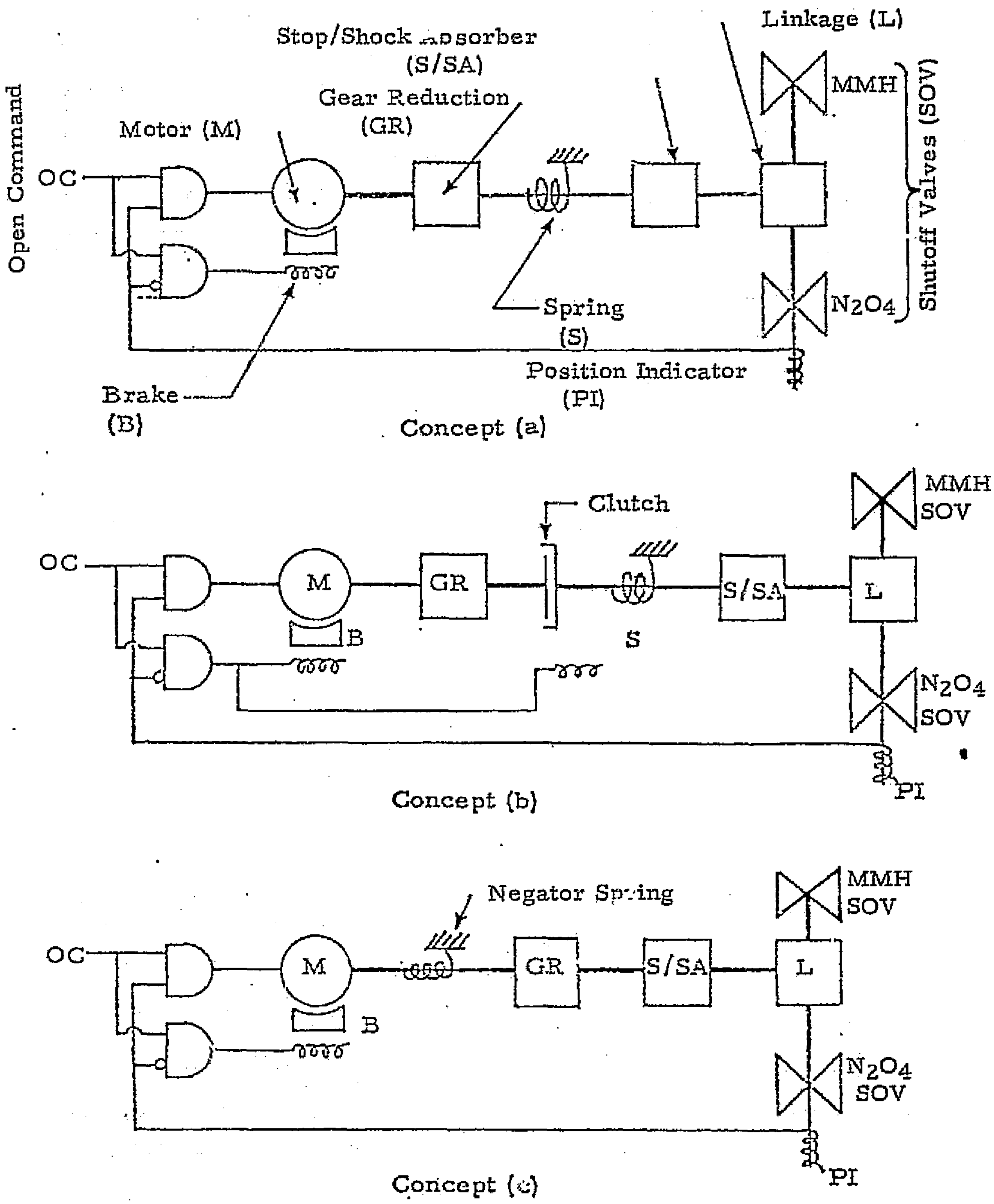


Figure 6-16. Motor Actuation System Concepts

Concept (b) shown on Figure 6-16 incorporates an electromagnetically operated mechanical clutch to avoid backdriving the motor and gears from the low speed side during valve closure. Although this substantially reduces the size of the return spring (one preliminary design has a 13-inch-lb spring), the addition of an in-line clutch increases power consumption and increases the system weight. Also, although the clutch is needed only to insure valve "fail-closed" after loss of electrical power, it must reliably operate each valve cycle and provide successful valve operation. Concept (b) was set aside because of these weight, power, and reliability considerations.

The selected actuation system is identified as Concept (c) on Figure 6-16. This approach uses a small negator spring located on the motor shaft. This approach (1) avoids the need for a large spring with its inherent vibration sensitivity problems, while (2) avoiding the weight and power requirements of an in-line clutch. Locating the spring on the low torque side of the gear reduction minimizes spring size because it uses the mechanical advantage afforded by the gear reduction. The negator spring provides a nearly constant torque over a large number of turns (like a "clock" spring), but has all its turns in contact for good vibration resistance (unlike a "clock" spring).

6.4.2 Actuation System Motor — To minimize technical risk, an AC induction ("squirrel cage") type motor will be used. Although previous analysis in this program has shown that the DC brushless torque motor has very good performance characteristics, it requires either optical or magnetic commutation and uses permanent magnets in the rotor. These requirements introduce technical risk because of temperature sensitivity (especially of the Hall effect devices if temperatures near 200°F are contemplated), demagnetization sensitivity of the magnets (due to current surges), and vibration sensitivity (of the magnets under sudden stop/start operation and during the long-term/high-level vibration of the Space Shuttle).

6.4.3 Drive Linkage — As previously noted in this report, the lifting ball valve operating torque is maximum at the start of valve opening and then drops off as the valve opens. If two valves are linked directly to the actuation system, the motor actuation system must be sized to develop twice this peak torque. However, if the valves are linked together in a manner that provides additional mechanical advantage at the start of opening, and a reduced mechanical advantage as the valve opens, the peak torque required to actuate the valves is reduced, resulting in a smaller motor and less power.

A Geneva mechanism was first analyzed for its capability to link the valves and reduce the peak operating torque. Figure 6-17 shows the valve shaft torque, as a function of valve shaft rotation, for a single valve. Figure 6-18 is a graph of the net torque when the Geneva mechanism is used to link two valves together. Although the very high initial mechanical advantage of the Geneva mechanism reduced the operating torque at the start of valve closure, a peak torque of 70 in.-lb is required after 33 degrees of input rotation to the Geneva. This is due to the mechanical advantage of dropping off to 0.6 at this point. Thus, the desired "torque-smoothing" effect was not obtained, and the motor requirement remained high; with 100 : 1 gear reduction, the inrush current was in excess of 15 amps at the 100 percent torque margin design point.

Next, a cam was devised to operate the two valves. A dramatic reduction in operating torque occurred as shown in Figure 6-19, although the cam required a higher rotation to open the valves (148 degrees).

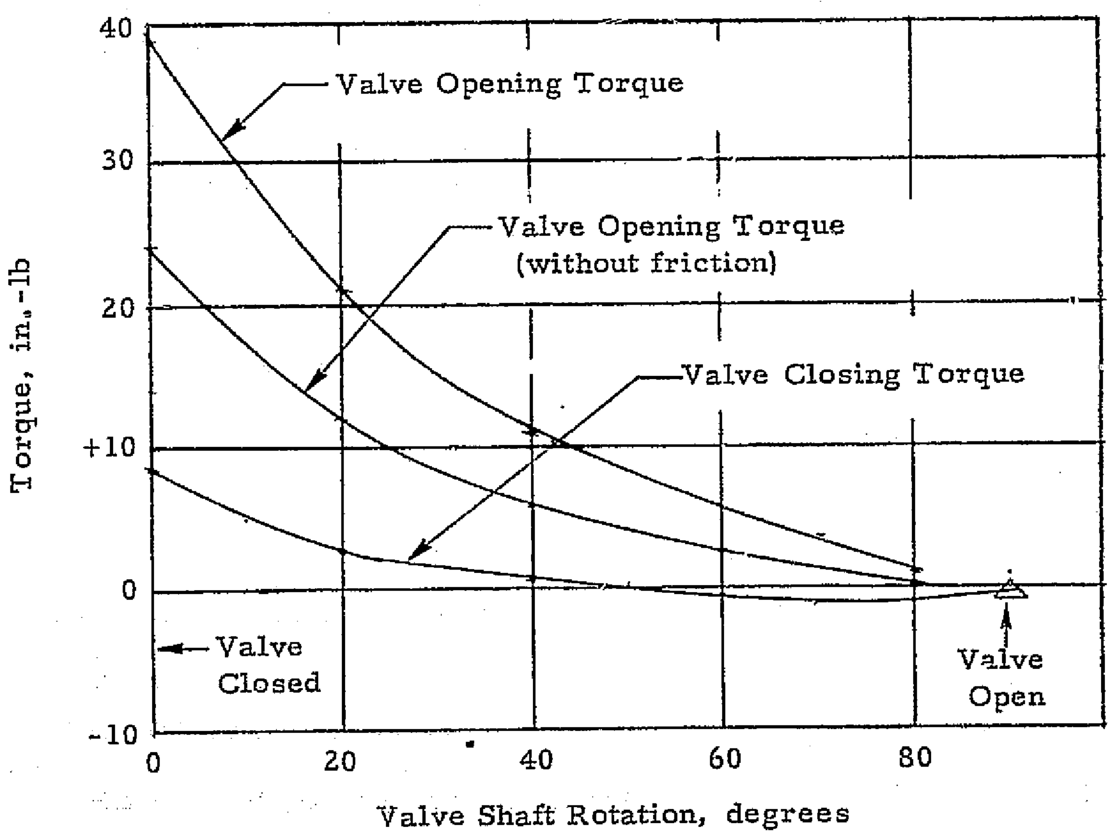


Figure 6-17. Valve Shaft Torque (one valve)

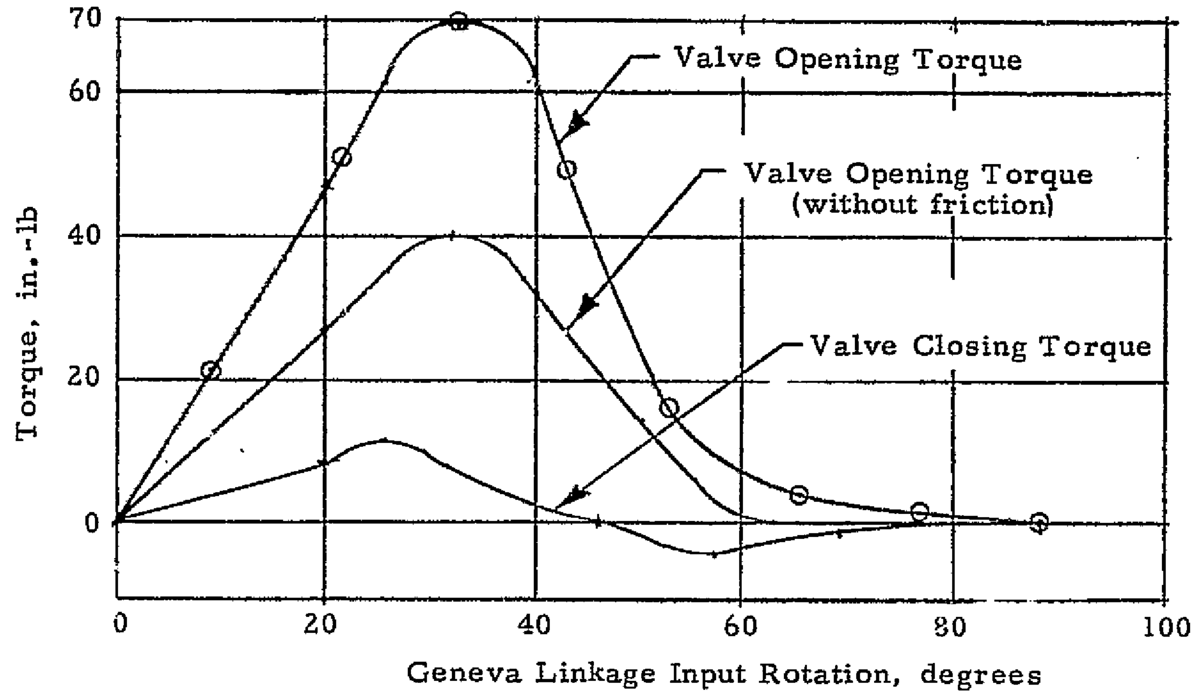


Figure 6-18. Linked Valves Torque, Using Geneva Linkage

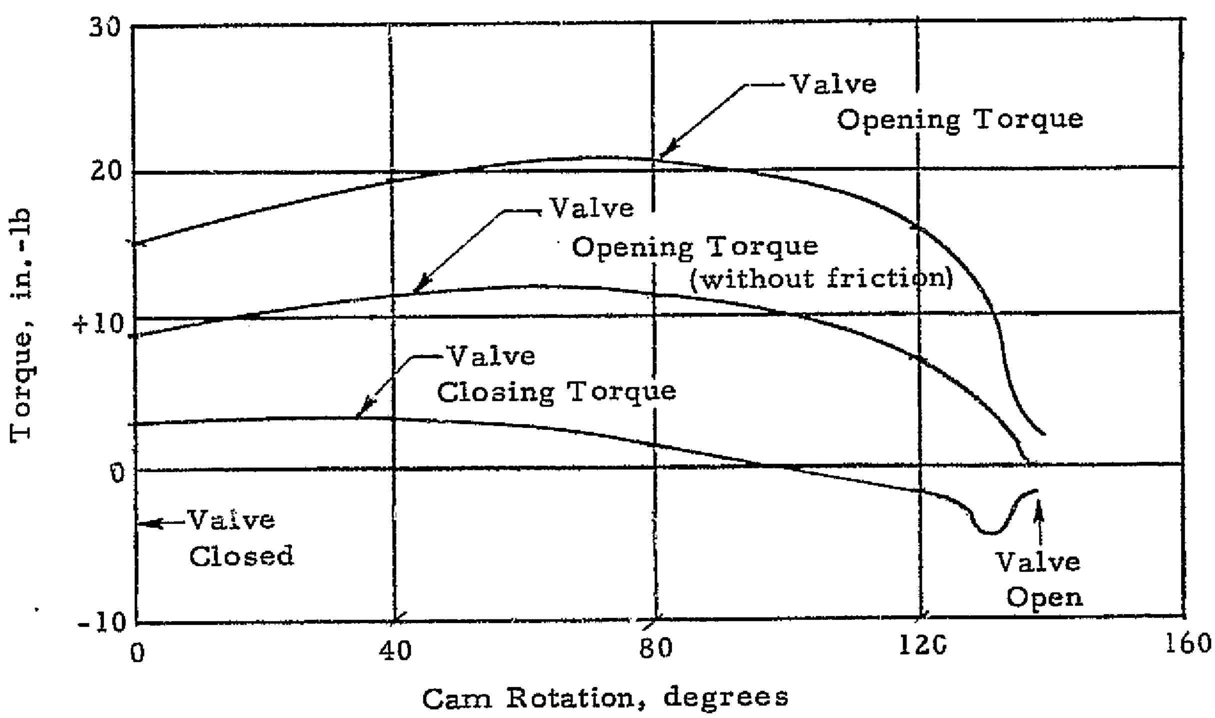


Figure 6-19. Linked Valves Torque, Using Cam

Although the inrush current at the 100-percent torque margin design point was reduced to less than 100 amps, the cam proved difficult to package in a reasonable envelope and there was concern about supporting the cam under vibration.

Next, the synchronizing linkage was analyzed. An objective of the analysis was to synthesize a four-bar linkage such that the motor would be required to put out as nearly constant a torque as possible, since this obviously results in the minimum size motor. The torque required at the valve shaft has previously been shown to have a peak at the valve-closed position, and to drop off continuously as the valve opens. Ideally, the linkage would have a variable mechanical advantage that was maximum at the closed position, and fell off as the valve opened. It is also desirable that the mechanical advantage rise again at the open position, not because high torque is needed, but because some overtravel of the actuator is required (i.e., a region where actuator rotation produces little rotation of the valve shaft). This overtravel assists in stopping the valve in the exact open position desired. Another requirement is that the linkage rotates the valve shaft 90 degrees, since this is what is required to fully open the valve. A linkage was synthesized to meet these requirements.

The linkage was analyzed from the principles of statics using computer program S298. The first graph, Figure 6-20, shows that the required 92-degree valve shaft rotation is attained by 160 degrees rotation of the gear output shaft. The next graph, Figure 6-21, shows the required gear output shaft torque versus valve angle. Note that the large peak in valve shaft torque has been reduced by the high mechanical advantage of the linkage in this region. The peak torque required at the gear train output shaft is 25.7 in.-lb, which occurs at a valve angle of 2 degrees.

6.4.4 Gear Train — The Harmonic Drive originally considered for use for the required rotational reduction necessary, provided a user risk inasmuch as there was only a sole supplier and the question of qualification for use might have presented a problem. It was decided to pursue a planetary gear train which is well established in the industry.

A gear ratio of 77:1 was selected for the planetary gear box, and a minimum efficiency of 73 percent was established. We selected a return spring torque (acting on the motor shaft) of 0.17 in.-lb. The required torque at the motor shaft was then computed as a function of valve angle, using the computer program. The results show that the peak torque required of the motor during opening is 0.66 in.-lb. In order to maintain the specified force margin of two, this torque is doubled for motor design purposes. Thus, the motor develops not less than 1.32 in.-lb torque during opening, which is twice the maximum torque required.

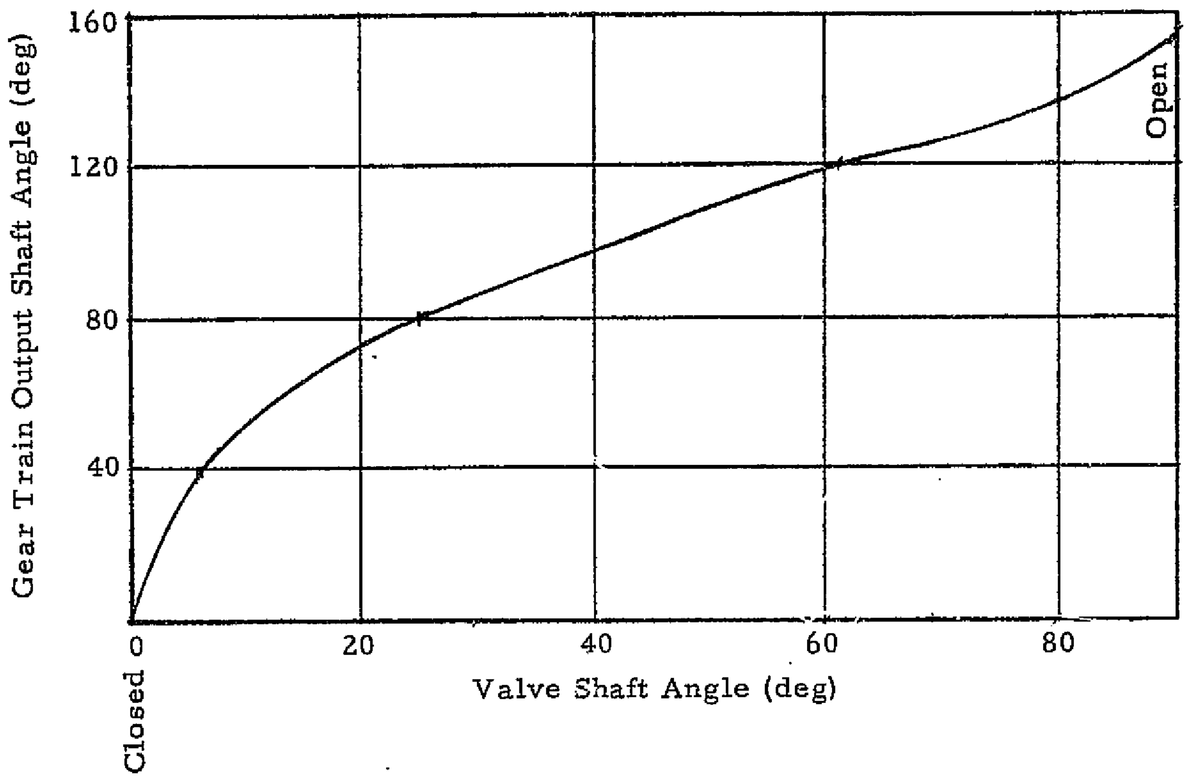


Figure 6-20. Gear Train Output Shaft Angle versus Valve Shaft Angle

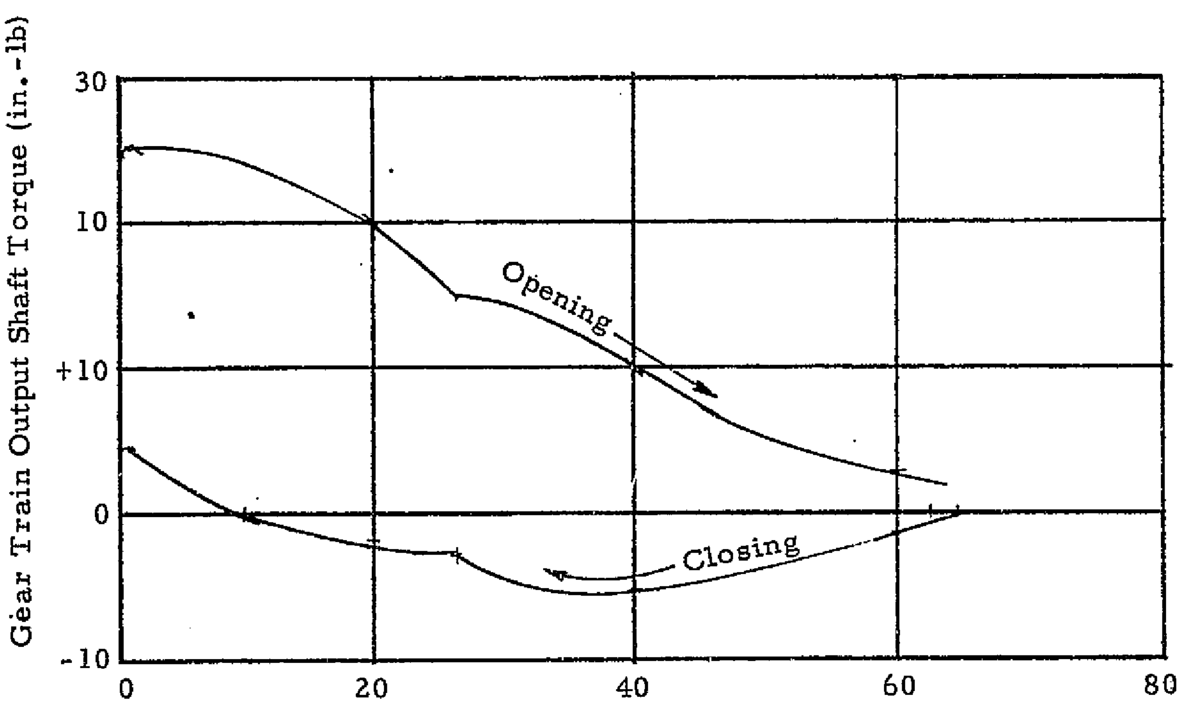


Figure 6-21. Gear Train Output Shaft Torque versus Valve Angle

C-2

The analysis also shows that the torque required at the motor shaft is always in the opening direction and is never less than 0.07 in.-lb. When the torque applied by the motor and brake is zero, as in the case when electrical power fails off, there is no opening torque applied to the motor shaft. Therefore, the valve will accelerate closed by itself.

Refer to Appendix D for Planetary Gear Train Analysis.

6.4.5 Electronic Control Circuit — Due to the selection of an AC induction type motor being made, the spacecraft dc buss voltage must be processed. Also, the system logic must be provided to monitor the valve position and reduce brake power when in the full open position. Because the motor speed is controlled by frequency and not amplitude, the electronic control must provide the required opening and closing motor frequencies upon demand. A functional diagram of the electronic control is included as part of Figure 6-22. All electronic components used in the electronic control will be solid state and provide the utmost in reliability and long life.

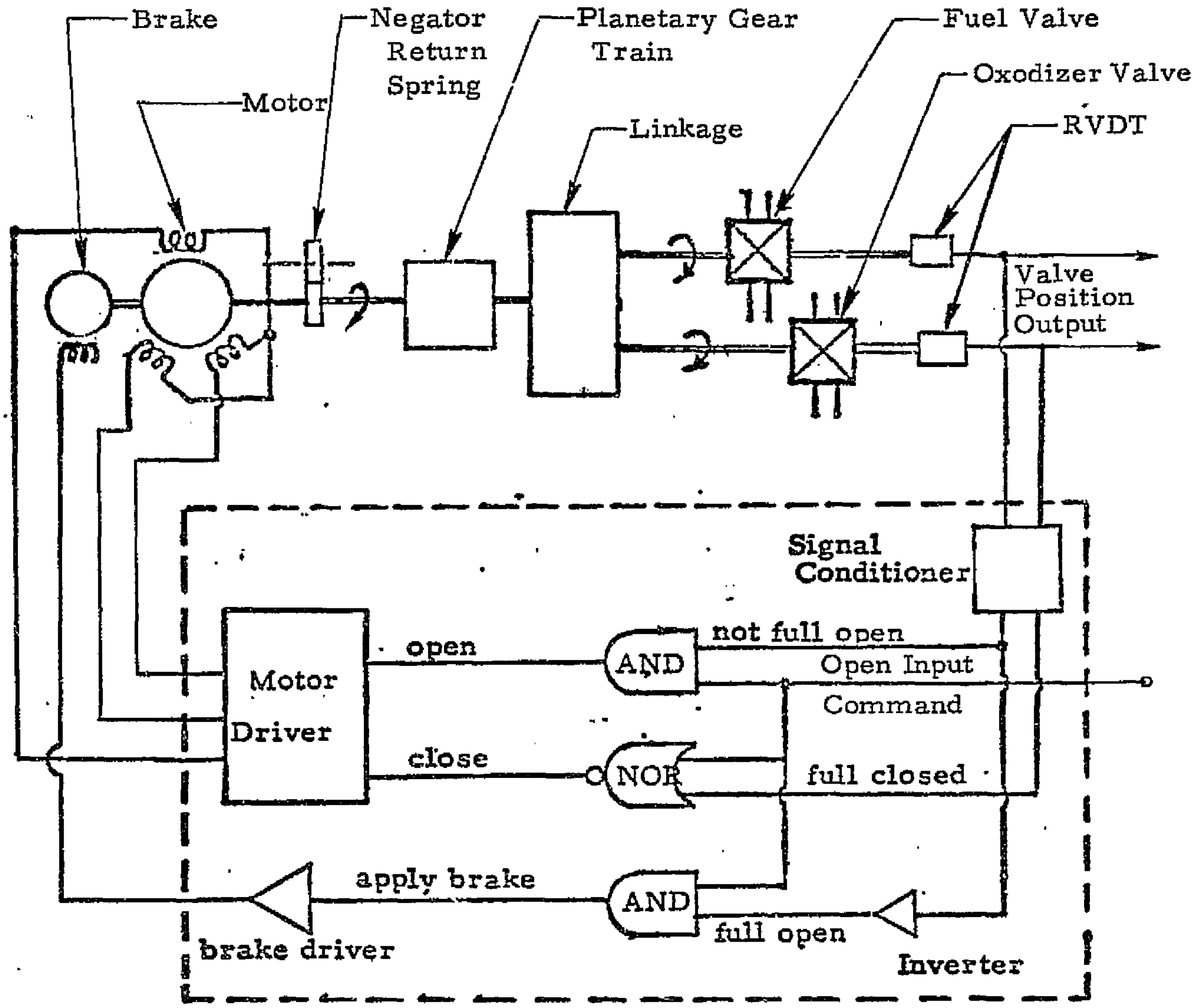
6.4.6 Rotational Variable Differential Transducer (RVDT) — The planetary gear train output shaft, which rotates 62 degrees, is monitored by a RVDT which provides the valve positioner signal to the electronic control circuit. Detail selection was accomplished by comparison testing of two candidate RVDTS's, with the test results provided in the test section.

6.5 Lifting Ball Valve and Actuation System Design Summary Result

6.5.1 General — The selected design approach, which provides the most convenient packaging, size, and operational confidence is schematically described in Figure 6-22 and briefly defined in the following.

1. Motor. The motor is a three-phase, squirrel cage, induction motor. It, of course, has no brushes. It operates on an alternating current supplied by the electronic control.

2. Electronic Control. The electronic control is a solid state inverter, which converts the DC vehicle power into a 3-phase AC voltage for motor operation. The driver receives control signals which turn the motor on and off and control its direction of rotation.



Solid State Electronic Control Package

Valve status is monitored by the valve angle transducer. When commanded to open, the solid state logic circuits verify that the valve is not already full open, and then energize the electric motor to rotate in the opening direction. When the full open status is detected, the logic circuits de-energize the motor and energize the electromagnetic brake. The brake holds the valve open with reduced power consumption as long as the open command is maintained. When the open command is removed, the motor drives the valve closed in a similar manner to opening. In case of electrical power failure, the valve is closed by energy stored in the return spring.

Figure 6-22. Schematic Diagram of Actuation System

3. Brake. The brake is an electromagnetic device (built into the motor) which supplies a torque to hold the valve full open with reduced power consumption. The brake torque is created by electromagnetic (not mechanical) means.

4. Brake Driver. The brake driver is simply a solid state switch which energizes the brake with DC power on receipt of low power level control voltage.

5. Return Spring. The return spring on the motor shaft stores energy to drive the valve closed in case of electrical power failure. The spring winds up as the valve opens, and unwinds as the valve closes.

6. Planetary Gear Reduction. The motor operates most efficiently at a relatively high rpm and low torque. The planetary gear reduction converts the motor power output to the relatively high torque at low rpm that is required for valve operation.

7. Linkage Assembly. The link assembly synchronizes the motion of the fuel and oxidizer valve driven by a particular motor. An additional function of the linkage assembly is to provide a variable mechanical advantage. The mechanical advantage is high in the open and closed positions, and lower in the intermediate positions. The arrangement flattens peaks in valve actuation torque, and provides for necessary overtravel in the open and closed positions.

8. RVDT Angular Position Transducer. This component is an angular position transducer with no wipers and no mechanical surface contact. It provides an AC "analog" output proportional to the angular position of the actuator. This signal is used for two purposes:

   a. For valve performance monitoring

   b. To turn off the motor (via the signal conditioner) when the valve is full open or full closed.

9. Signal Conditioner. This component is a solid state switch which receives the analog position signals from the RVDT's and puts out two electrical control signals:

   a. Valve in full open position.

   b. Valve in full closed position.

10. Valve Assembly. The valve assembly is a lifting ball valve that provides a minimum of 10,000 operational cycles.

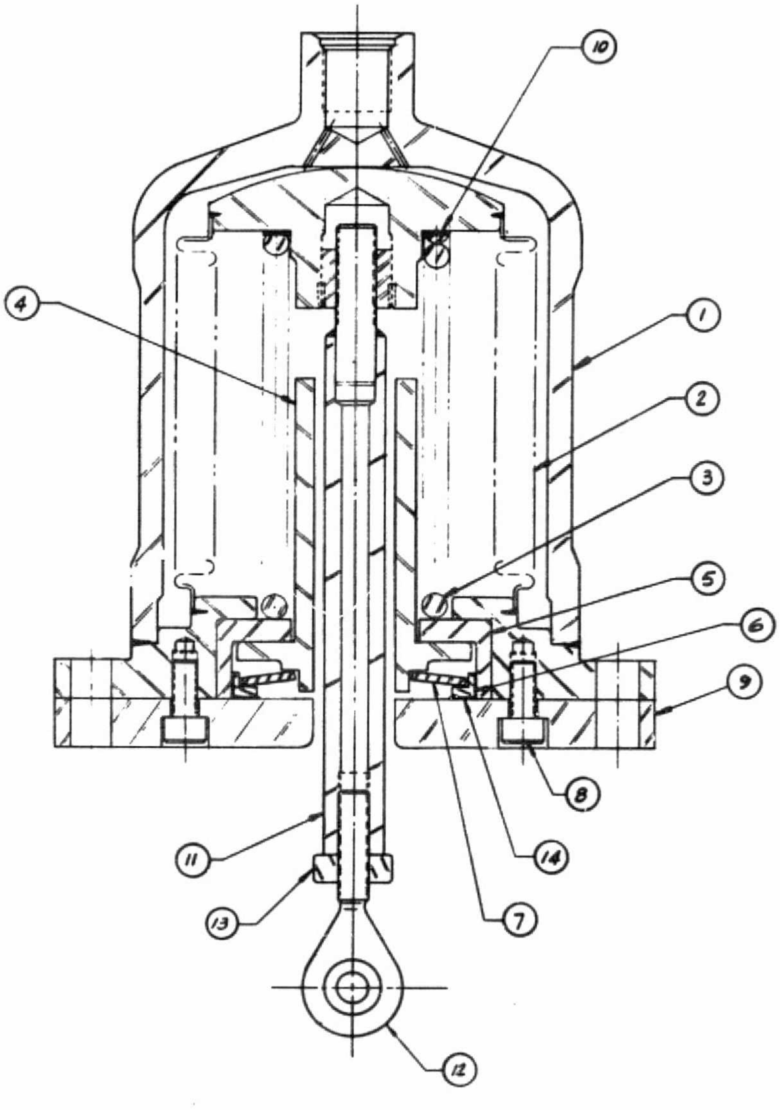
6.6 Alternate Ball Valve Concept

6.6.1 General — The lifting ball valve concept as originally designed had somewhat higher initial operating torque, and occasional sticking, at both open and closed positions. When the rotating surfaces were coated with a light film of lubricant, the valve performed as predicted. However, an analysis was performed to determine the cause of the problem. The basic analysis is included in the test section of this report. The conclusions of the analysis and detail tests performed directed the alternate ball valve concept. Because the valve performs as predicted with a light coat of lubricant on the rotating surface, it was reasonable to assume that if the rotational surfaces can be more closely controlled, or predicted, the valve concept and operational validity will be upheld. To accomplish this, new valve linkage dimensions were established which allowed the required control. A minimum operational prototype valve model was manufactured to demonstrate the viability of the 4-bar linkage concept. The valve consisted of a modified drive shaft and ball assembly; however, the rotational parameters and associated components were maintained as would be required to test the concept. The alternate ball valve detail configuration is included in Section 7.0.

6.7 Pneumatic Actuation System

6.7.1 General — A Pneumatic Actuation System was designed to provide the valve operating force, in lieu of the motor-driven actuation system. Two prototype actuators were manufactured and tested. A prototype valve and actuation system outline drawing was also prepared which provided the maximum fuel lead linkage specified; i.e., 30 degrees. Figure 6-23 and 6-24 show the details of the pneumatic actuator assembly and the pneumatic actuator valve drive system.

To design the pneumatic actuator, the required stroke had to be determined. Therefore, the valve an actuation system concept layout was prepared.



1 MAY BE PURCHASED FROM
NMB CORP. /AERO DIVISION
12057 W JEFFERSON BLVD
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(213) 390 4401

NOTES:

Figure 6-23. Pneumatic Actuator

REVISIONS					
E.C.O. NO.	ZONE	LTR	DESCRIPTION	DATE	APPROVED
—	—	NC	RELEASE FOR E.D.O.	29 MAR '74	

QTY REQD	PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	CODE IDENT	REMARKS	DWG SIZE	ITEM NO.
AR	5739048	SHIM, B.V.				14
1	AN320 C3	NUT			—	13
1	AHM-3	BEARING, ROD END		NMB CORP.	—	12
1	5739040	ROD, CONNECTING			C	11
A/R	5739047	SHIM				10
1	5739034	PLATE, MTG			D	9
4	#8-36UNF, 3/8 LONG	SCREW, CAP		(NAS1361-08-6 OR EQUIV)	—	8
1	5739042	SPRING, BELLEVILLE			C	7
1	5739043	GUIDE, INNER			C	6
1	5739041	RETAINER, SPRING			C	5
1	5739039	STOP, BELLOWS			C	4
1	5739027	SPRING, RETURN			B	3
1	5739020	BELLOWS ASSY			D	2
1	5739038	SHELL, ACTUATOR			D	1
	-101	ACTUATOR ASSY	—			

PARTS LIST

DO NOT SCALE DRAWING

MUST CONFORM TO ... 1-1
DIMENSIONS AND TOLERANCING PER USASI Y14.5
MACHINED SURFACE ... PER ASA B46.1
CLASSIFICATION OF CHARACTERISTICS ... PER SPI ...

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN INCHES
TOLERANCES
.XX ± .03 .XXX ± .010 ∠ ± 0° 30'

MATERIAL

FINISH

HEAT TREAT

CONTRACT NO. 5180

PREP PIERSAS
CHK
DSGN
PROJ
APVD

PARKER HANNIFIN IRVINE, CALIFORNIA

ASSEMBLY, ACTUATOR

SIZE D | CODE IDENT NO. 92003 | DWG NO. 5739036

SCALE 2/1 | UNIT WT — | SHEET 1 OF 1

5739036

Pneumatic Actuator Assembly

Figure 6-23

FOLDOUT FRAME 2

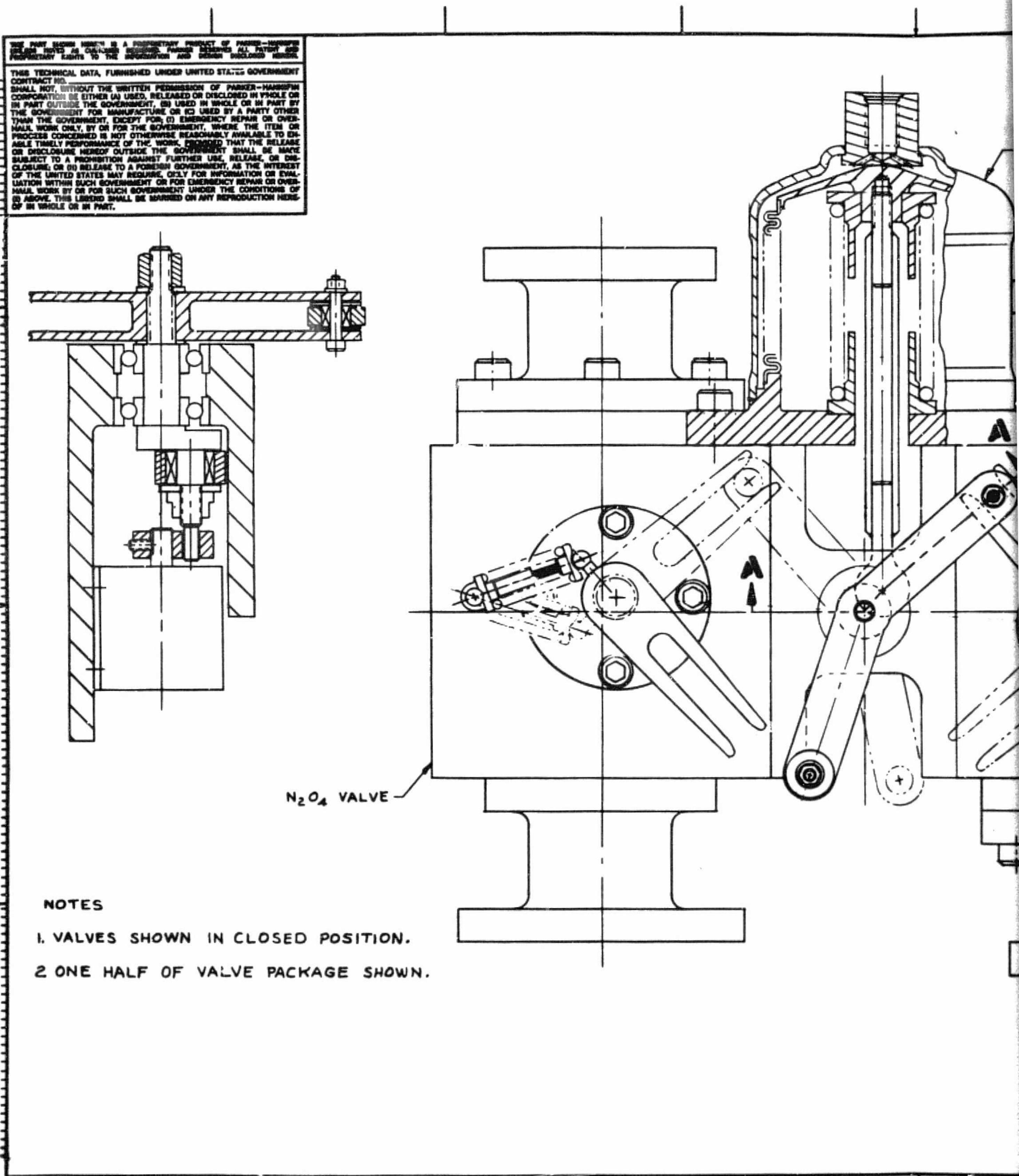
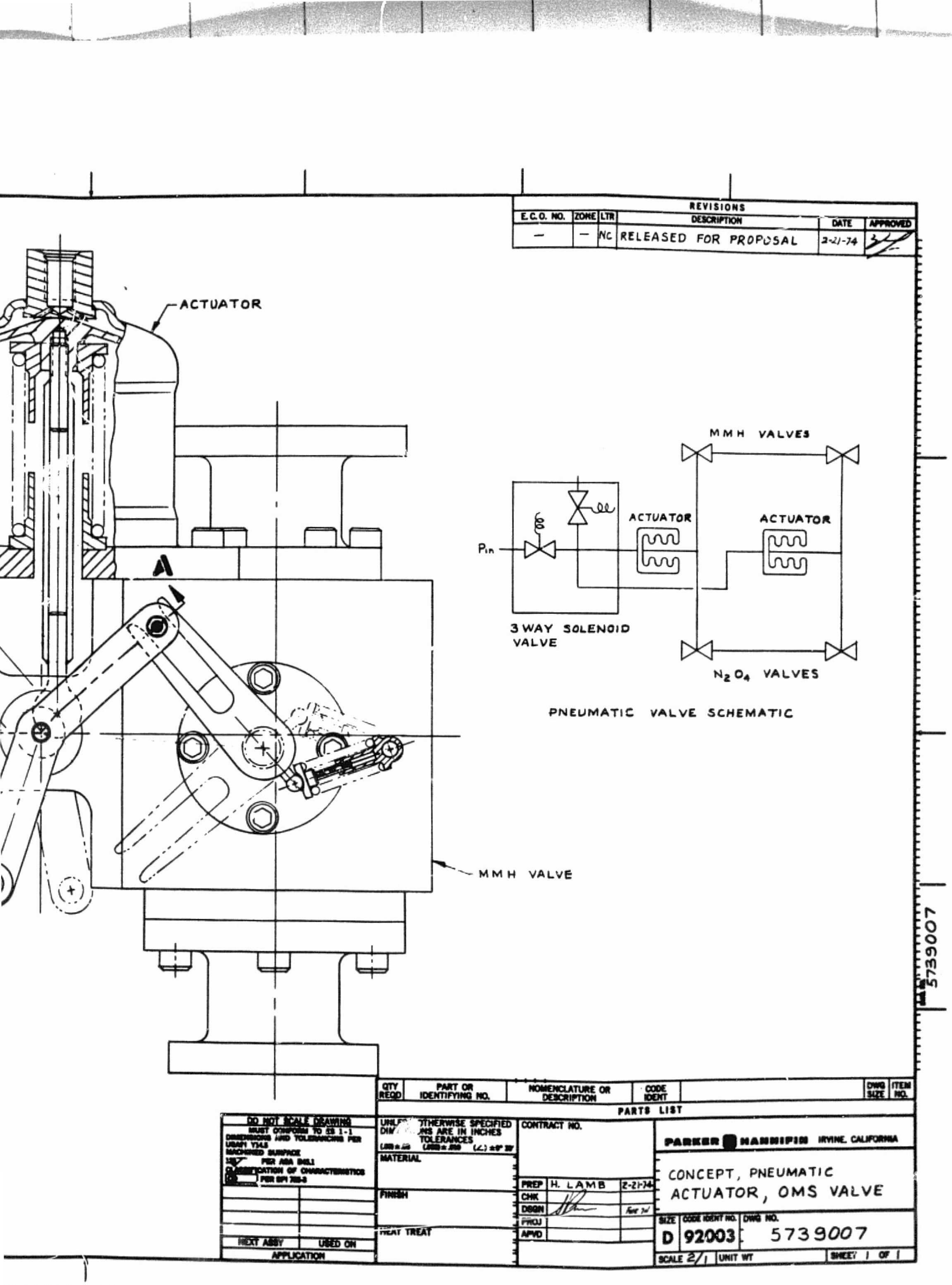


Figure 6-24. OMS Valve Pneuma



OMS Valve Pneumatic Actuator Concept

Figure 6-24

6.7.2 Prototype Pneumatic Actuator — The pneumatic actuator consists of a multiple-ply hydraformed bellows assembly, mounting plate, shell, bellows stop, return spring, and belleville spring. Refer to Figure 6-23. Preliminary design data is included in Appendix E.

6.7.3 Prototype Pneumatic Actuation System — The detail system as shown in Figure 6-24 consists of a pneumatic actuator, a drive assembly, a linkage assembly, and a valve seat loading spring. The system was designed with a 30-degree fuel lead, an actuator stroke of 0.5-inch, and a 120-degree linkage rotation.

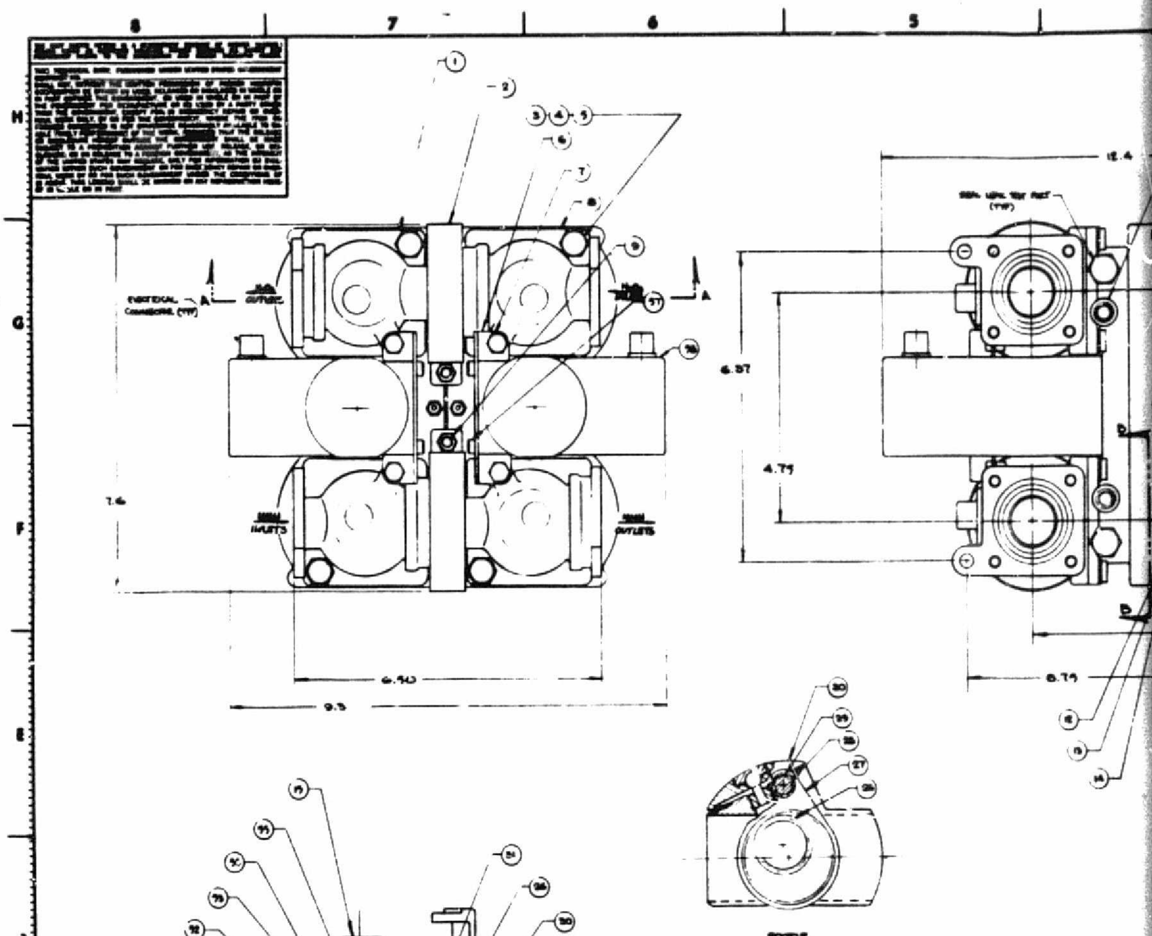


Figure 7-1. OME Quad Valve and A

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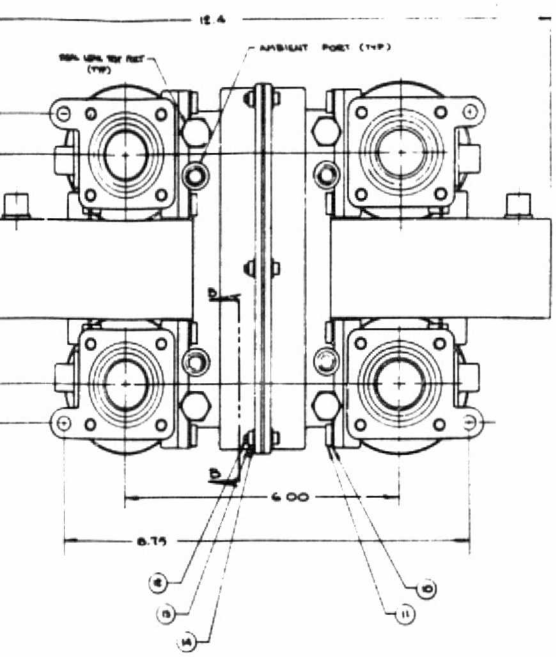

Qty	Nomenclature or Description	Item No.
4	ACTUATOR ASSY (SEE SHEET 2 FOR SECTION)	58
8	SCREW	57
8	PACKING (SEALS JOINT BETWEEN VALVE AND ACTUATOR)	56
8	ASM. VALVE	55
8	NUT	54

Qty	Nomenclature or Description	Item No.
8	DUPLEX BEARING SET	53
AR	SHIM	52
16	RING, RETAINING	51
24	SEAL	50
16	RETAINER	49
8	CAP	48
8	SEAL ASSY	47
8	SEAL	46
AR	SHIM	45
8	RING, RETAINING, BEVELED	44
8	BUSHING	43
8	ADAPTER	42
8	RING, RETAINING	41
AR	SHIM	40
8	SHOULDER WASHER	39
8	WAVE WASHER, SPRING	38
8	WASHER	37
4	SEAL	36
4	SEAL	35
8	SEAL	34
8	SEAL	33
8	BUSHING	32
8	SHAFT	31
8	VISOR ASSY	30
8	BUSHING	29
8	PIN	28
8	LINK	27
16	BUSHING	26
8	LINK ASSY - CONTAINS TWO NEEDLE BEARINGS	25
8	CLEVIS	24
AR	SHIM	23
8	BUSHING	22
8	SPRING, OVERTRAVEL	21
4	ASM ACTUATOR	20
24	NUT	19
16	PIN	18
4	SPRING, ENERGY ABSORBING	17
4	NUT	16
8	BOLT	15
24	WASHER	14
12	NUT	13
12	SCREW	12
32	BOLT	11
32	WASHER	10
4	NUT	9
4	BODY	8
8	BOLT	7
4	BRACKET	6
16	PACKING	5
16	PACKING	4
16	PLUG (SEALS LEAKAGE CHECKOUT PORT)	3
4	V-BAND COUPLING	2
4	BODY	1
-101	VALVE ASSY	

PARTS LIST

OME QUAD VALVE AND ACTUATION SYSTEM

E 92003 5730026

SHEET 1 OF 2

Quad Valve and Actuation System

Figure 7-1
(Sheet 1 of 2)

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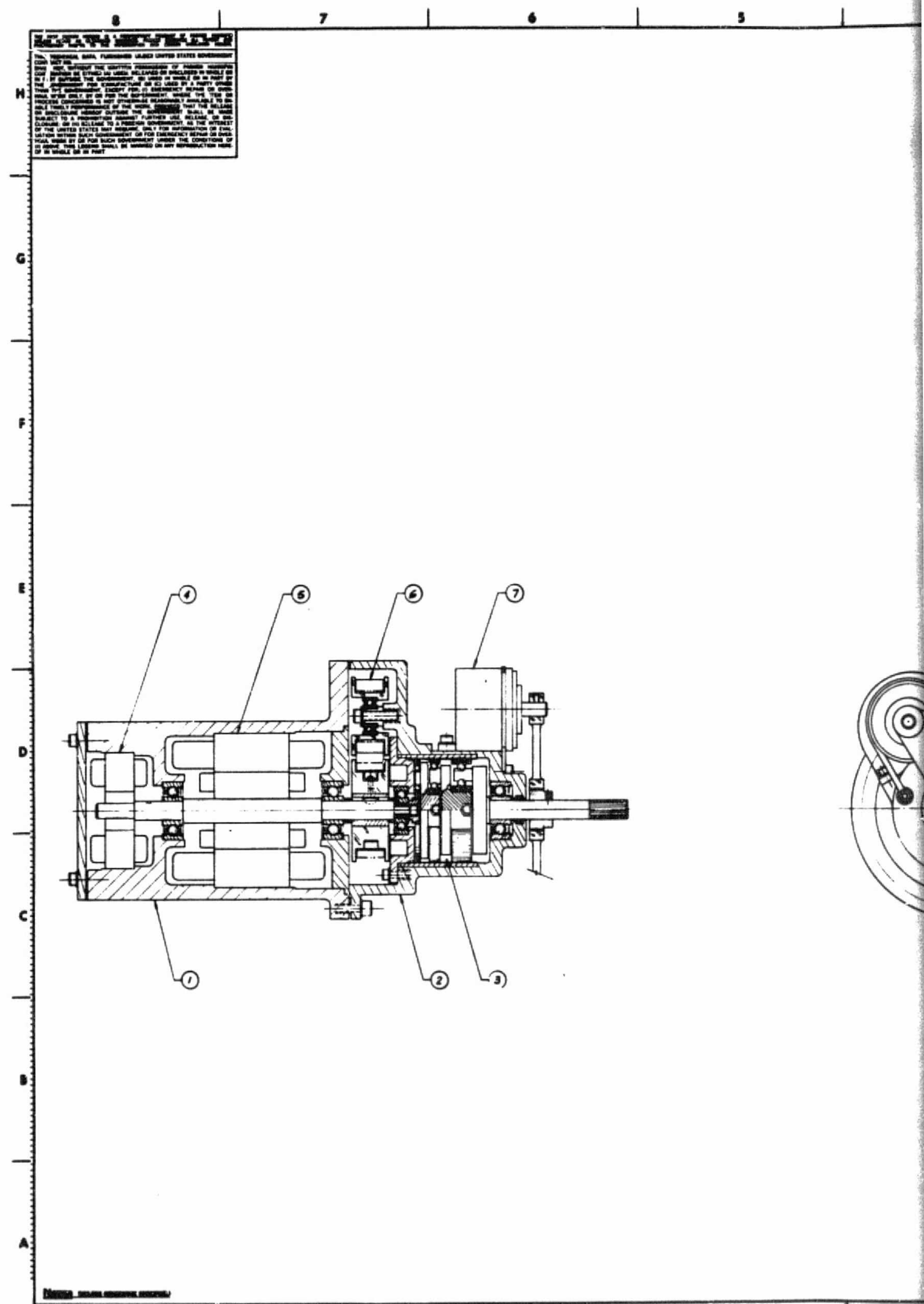

Figure 7-1. OME Quad Valve and Actua

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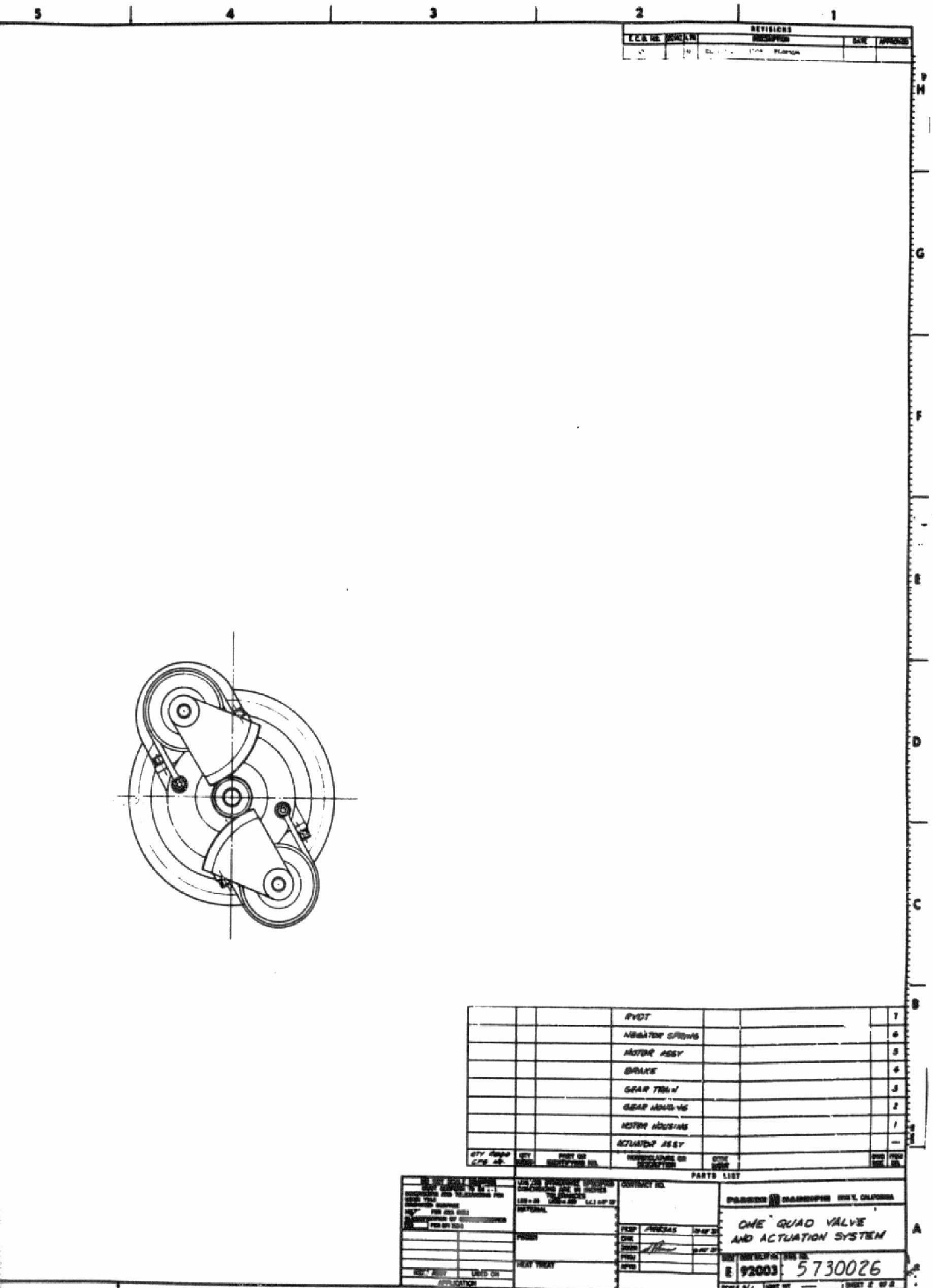

...d Valve and Actuation System (Continued)

Figure 7-1
(Sheet 2 of 2)

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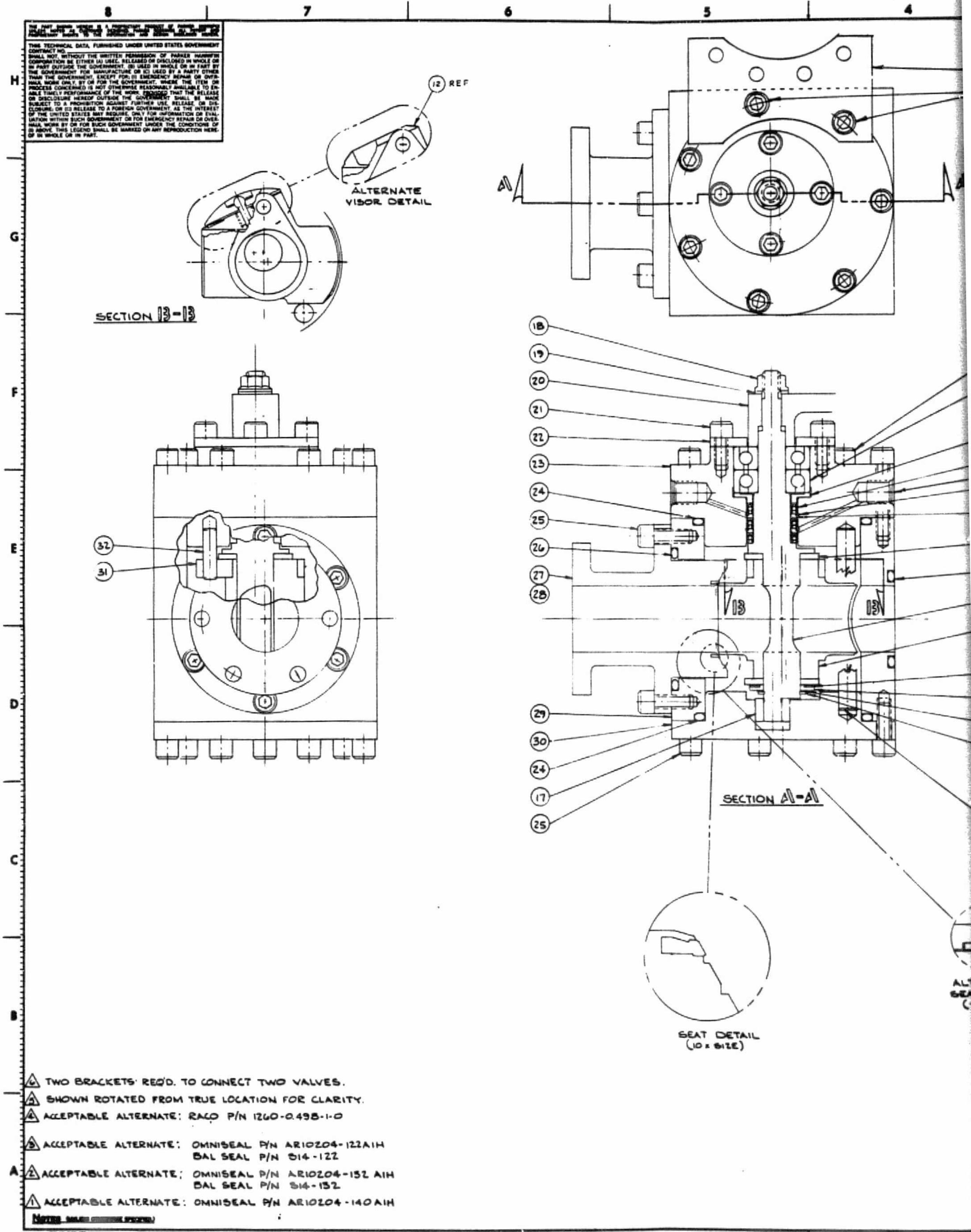

Figure 7-2. Lifting Ball Valve (Prototype)

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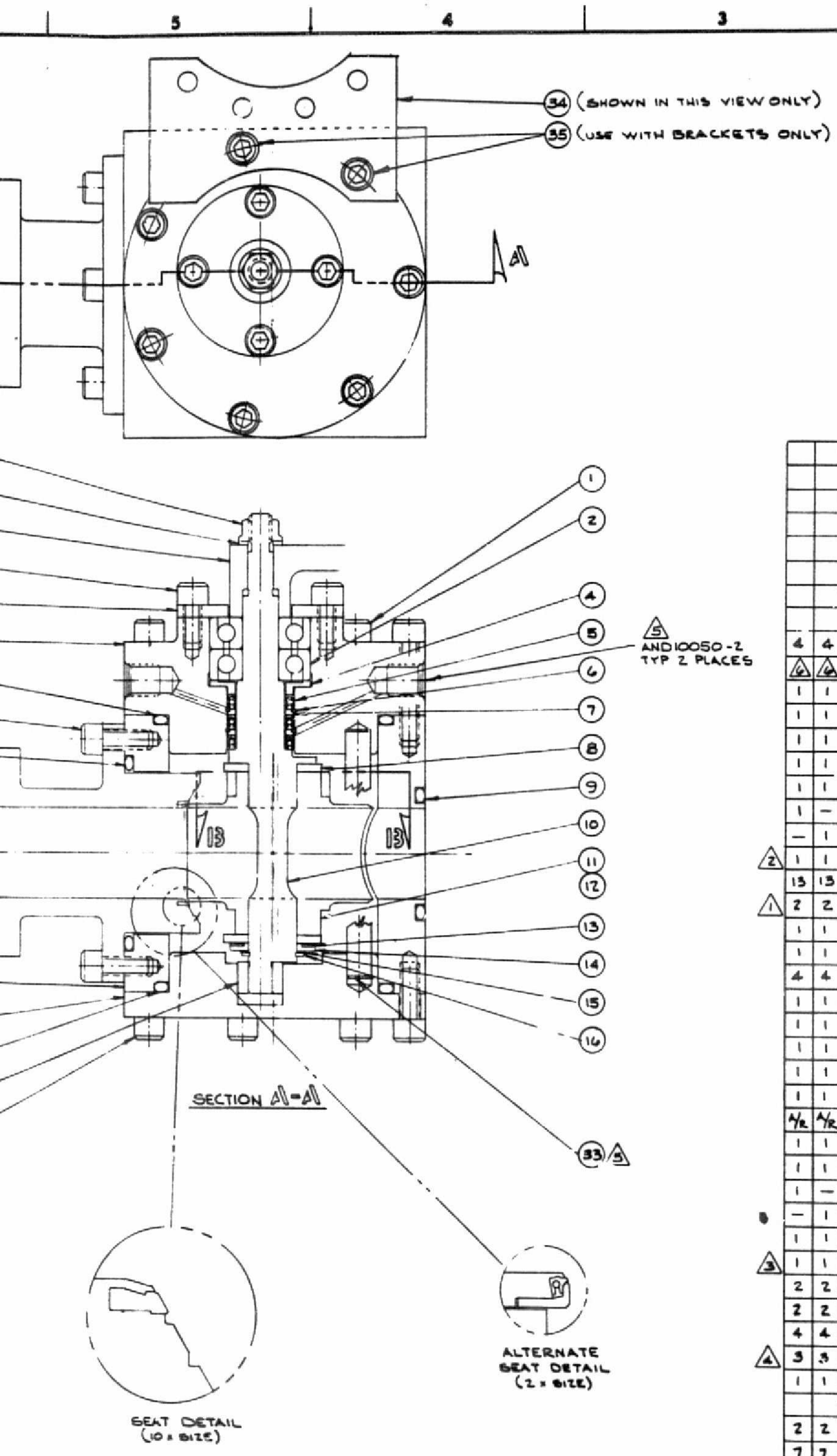

E.C.O. NO.		REVISIONS DESCRIPTION	DATE	APPROVED
-	- INC	RELEASED		

	4	4	NAS1351C3-20	SCREW		CRES		35
	6	6	5736092-1	BRACKET		300 SERIES CRES		34
	1	1	5736091-1	PIN		15-5 PH CRES		33
	1	1	5736090-1	PIN		15-5 PH CRES		32
	1	1	5736089-1	LINK		PH 13-8 MO CRES		31
	1	1	5736075-1	COVER		PH 13-8 MO CRES		30
	1	1	5736071-1	HOUSING		PH 13-8 MO CRES		29
	1	—	5736070-101	SEAT ASSY		PH 13-8 MO CRES		28
	—	1	5736067-101	SEAT ASSY		PH 13-8 MO CRES		27
2	1	1	2-132	O' RING				26
	13	13	NAS1351C3-8	SCREW		CRES		25
1	2	2	2-140	O' RING				24
	1	1	5736076-1	COVER, UPPER		PH 13-8 MO CRES		23
	1	1	5736080-1	RETAINER, BRG.		300 SERIES CRES		22
	4	4	NAS1351C3-6	SCREW		CRES		21
	1	1	5736081-1	LEVER		300 SERIES CRES		20
	1	1	AN960C416L	WASHER		CRES		19
	1	1	H46-3	NUT	15653	KAYNAR (CRES)		18
	1	1	5736079-1	BUSHING		15-5 PH CRES		17
	1	1	5108-50-H	RING, RETAINING		WALDES. TRUARC 15-7 PH CRES		16
	A/R	A/R	5736082 -1 -2	SHIM		15-5 PH CRES		15
	1	1	5736083-1	SPACER		15-5 PH CRES		14
	1	1	5736093-1	WASHER, WAVE		15-7 PH CRES		13
	1	—	5736089-101	VISOR ASSY		PH 13-8 MO CRES		12
	—	1	5736061-101	VISOR ASSY		PH 13-8 MO CRES		11
	1	1	5736072-1	SHAFT		15-5 PH CRES		10
3	1	1	2-122	O' RING				9
	2	2	5736084-1	WASHER		15-5 PH CRES		8
	2	2	5736088-1	SPACER		15-5 PH CRES		7
	4	4	5736086-1	RING, CRESCENT		15-5 PH CRES		6
4	3	3	AR10104-112AH	SEAL	00624	AEROQUIP, OMNISEAL		5
	1	1	5736087-1	RETAINER, SEAL		15-5 PH CRES		4
								3
	2	2	OLOODTL	BEARING, BALL		NEW DEPARTURE		2
	7	7	NAS1351C3-16	SCREW		CRES		1
	-102	-101		ASSEMBLY				
	QTY REQD		PART OR IDENTIFYING NO.	NOMENCLATURE OR DESCRIPTION	CODE IDENT			ITEM NO.

PARTS LIST

CONTRACT NO. 180	SYSTEMS DIVISION PARKER HANNIFIN IRVINE, CALIFORNIA	
PREP CLEETON	BALL VALVE, LIFTING (PROTOTYPE)	
	E 92003 5739001	
	SCALE 2X	SHEET 1 of 1

5739001

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Table VII-1. Prototype Lifting Ball Valve Parts List

Title Ball Valve, Lifting
(Prototype)

Sheet 1 of 2

Final Assembly P/N 5739001-101, 102 Alternate

Line No.	Part Number	Qty EI	Description	Material
1	5739001-101,-102	1	VALVE, LIFTING BALL	
2	NAS1351C3-12	12	SCREW	CRES
3	OLOODTL	2	BEARING, BALL	New Departure
4	AR10104-112A1H	2	SEAL	Omniseal, Aeroquip 00624
5	5736084-1	2	WASHER	PH13-8MO CRES
6	2-122	1	O-RING	
7	5736072-1	1	SHAFT	PH13-8MO CRES
8	5736061-101	1	VISOR ASSY, SWIVEL	
9	5736061-1	1	SUBASSEMBLY	
10	5736062-1	1	BALL	PH13-8MO CRES
11	5736063-1	1	SEGMENT	PH13-8MO CRES
12	5736064-1	1	RETAINER	PH13-8MO CRES
13	5736074-1	1	ADAPTER	PH13-8MO CRES
14	5736094-2,-3,-4,-5	AR	SHIM	302 CRES
15	5736060-1	1	VISOR, SWIVEL	PH13-8MO CRES
16	5736077	1	BELLEVILLE	
17		AR	WELD WIRE	PH13-8MO CRES
18	5736059-1	1	VISOR (ALTERNATE)	PH13-8MO CRES
19	5736093-1	1	BELLEVILLE	301 CRES 3/4-FULL HD
20	5736083-1	1	SPACER	PH13-8MO CRES
21	5736082-1,-2	AR	SHIM	PH13-8MO or 302 CRES
22	5736085-1	1	RETAINER, VISOR	PH13-8MO
23	H46-3	1	NUT	CRES KAYNAR 15653
24	AN960C416L	1	WASHER	CRES
25	5736081-1	1	LEVER	304 CRES

Table VII-1. Prototype Lifting Ball Valve Parts List (Continued)

Title Ball Valve, Lifting (Prototype)

Sheet 2 of 2

Final Assembly P/N 5739001-101, 102 Alternate

Line No.	Part Number	Qty EI	Description	Material
26	NAS1351C3-6	4	SCREW	CRES
27	5736080-1	1	RETAINER BUSHING	304 CRES
28	2-140	1	O-RING	
29	NAS1351C3-8	7	SCREW	CRES
30	2-132	1	O-RING	
31	5736067-101	1	SEAT ASSY	
32	5736068-1	1	SEAT, BLANK	13-8 MO
33	5736069-1	1	INSERT (S.G : 2.17 min)	Molded Teflon per ES5-11A GR A
34	5736070-101	1	SEAT ASSY (Alternate)	
35	5736070-1	1	SUBASSEMBLY	
36	5736088-1	1	FLANGE	PH13-8MO CRES
37	5736092-1	1	RETAINER, SEAL	PH13-8MO CRES
38	MS171440	2	PIN, SPRING	CRES
39	5736065-1	1	SEAL	TFE Teflon
40	5736071-1	1	HOUSING	PH13-8MO
41	5736075-101	1	COVER ASSY	
42	5736075-1	1	COVER	PH13-8MO
43	5736090-1	1	PIN, LINK	PH13-8MO
44	5736089-1	1	LINK	PH13-8MO CRES
45	5736090-1	1	PIN, LINK	PH13-8MO CRES
46	5736091-1	1	PIN, VISOR STOP	PH13-8MO CRES

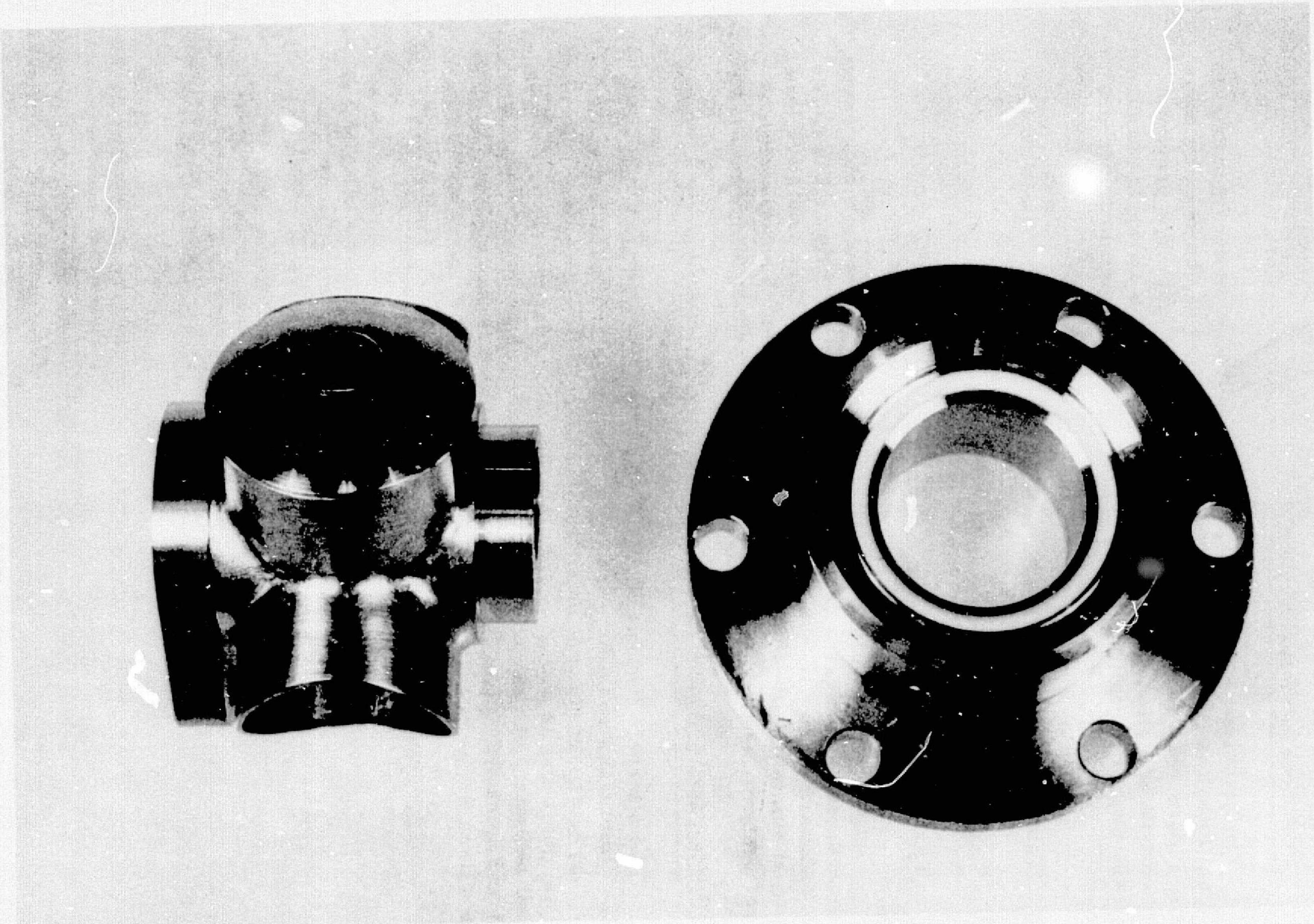


Figure 7-3. Ball and Seat Assembly

7.3 Prototype Actuation System

The prototype actuation system configuration is considerably different from the flightweight version would be as can be seen by comparing Figure 7-1, Sheet 2 of 2 and Figure 7-4. Figure 7-4 presents the prototype actuation system drawing with the valve shown in phantom as a reference. Parts lists of the major assemblies are provided as follows.

Table VII-2. Actuation System

Table VII-3. Motor Assembly

Table VII-4. Gear Train Assembly

Figure 7-5 shows the test valve housing, ball, seat, motor assembly, brake, and a disassembled planetary gear train. Figure 7-6 shows the breadboard electronic control system complete with a resistive load used to simulate a motor during preliminary testing. The electronic control system was packaged in a small control module for system testing.

7.4 Alternate Lifting Ball Valve

The alternate lifting ball valve was manufactured such that valve linkage dimensions could be evaluated. The valve configuration was similar to the alternate concept as shown in Figure 7-7. Differences between what was built and as shown is only in areas not under question, such as the ball and seat. An exploded view of the valve as manufactured is provided in Figure 7-8. The Parts List for the valve is included as Table VII-5.

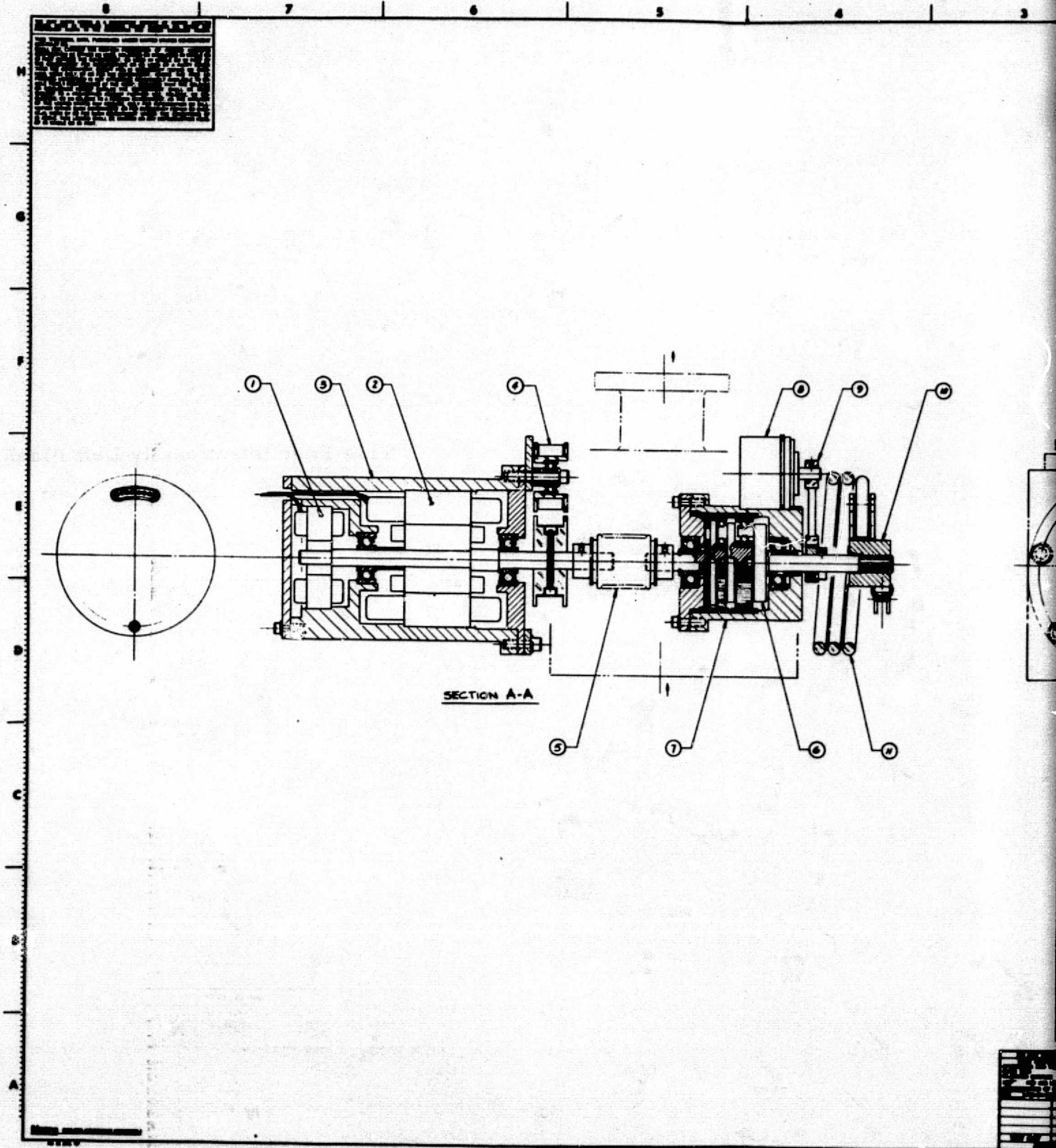
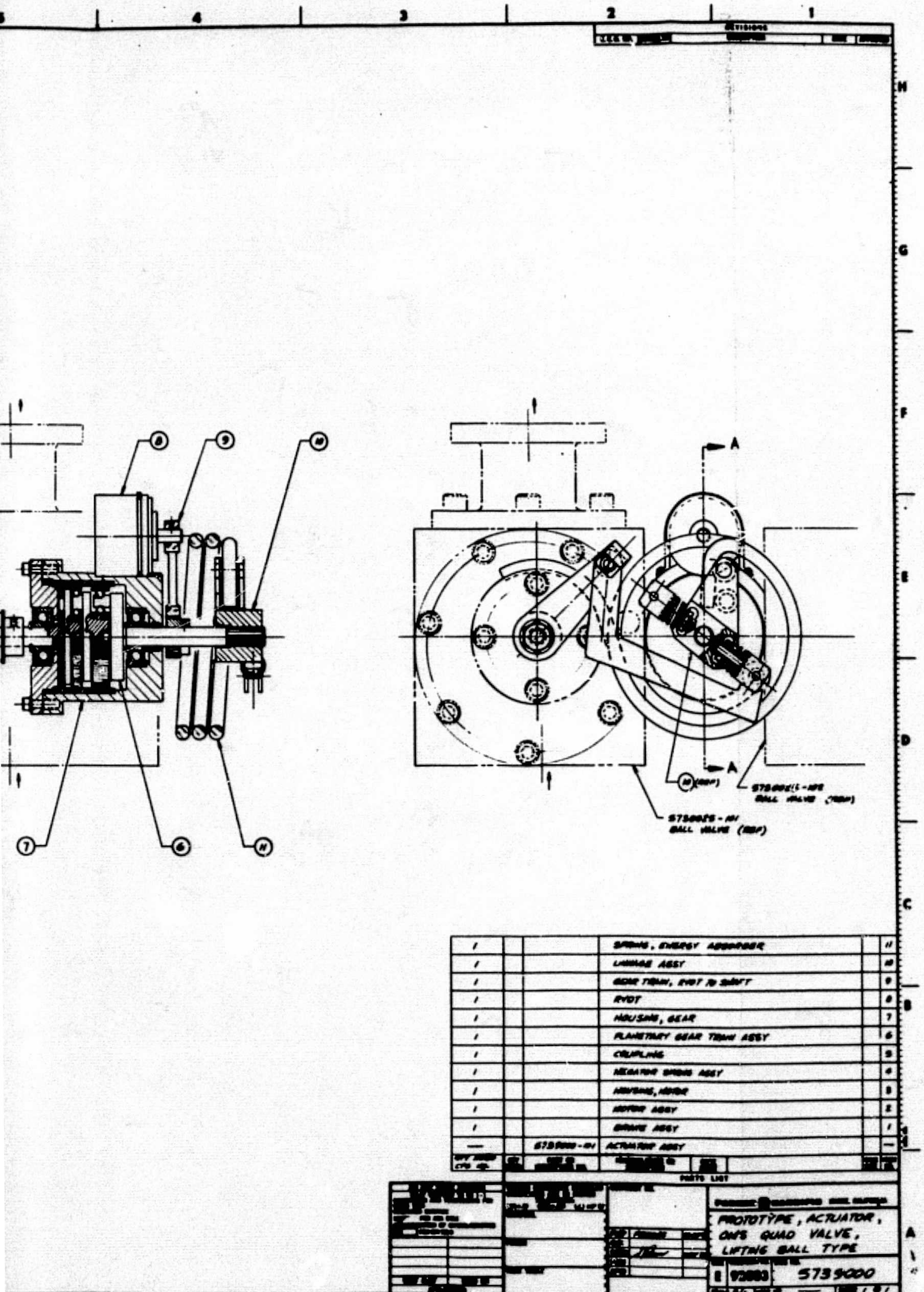


Figure 7-4. Lifting Ball Valve Actuator (Prototype)



ing Ball Valve Actuator (Prototype) Figure 7-4

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Table VII-2. Actuator Assembly Parts List

Title Prototype, Actuator, OMS Quad Valve, Lifting Ball Type

Final Assembly P/N 5739000-101

Line No.	Part Number	Qty EI	Description	Material
1	5736122-101	1	ASSY, MOTOR, ACTUATOR	See PL5736122
2	NA5620C8L	5	WASHER	CRES
3	NAS1352C08-6	4	SCREW	CRES
4	CS-2	1	SET SCREW	CRES PIC
5	5736113-1	1	OUTPUT DRUM	
6	MS51957-12	1	SCREW	
7	5736114-101	1	GEAR TRAIN ASSY	SeePL5736114
8	R30A	1	RVDT	Schaevitz
9	MS35842-13	1	CLAMP	
10	5736116-1	1	GEAR SECTOR	Delrin (25% Glass)
11	CS-1	2	SET SCREW	CRES PIC
12	5736119-1	1	SPRING	302 CRES
13	5736117-1	1	GEAR	300-Series CRES
14	5736112-1	1	STORAGE DRUM	Delrin or Celcon
15		1	NEGATOR SPRING	Hunter Spring Corp
16	SR3SSTA	1	BEARING	Barden
17	H46L06	1	NUT	Kaynar
18	5736109-1	1	STUD	303 CRES
19	5736118-1	1	PIN	410 CRES
20	5736121-1	1	RETAINER	440C CRES
21	5736120-1	1	PIN	410 CRES
22	5736095-1	1	LINK	410 or 416 CRES
23	5736124-1	2	DRAG LINK	7075, 2024-T6 or 6061-T651 Al aly

Table VII-2. Actuator Assembly Parts List (Continued)

Title Prototype, Actuator, OMS Quad Valve, Lifting Ball Type

Sheet 2 of 2

Final Assembly P/N 5739000-101

Line No.	Part Number	Qty EI	Description	Material
24	58 FT	2	NUT	SPS
25	5736125	2	SPRING	FS 302 CRES
26	5736126-1	2	BUSHING	410 CRES
27	5736127-2, -3, -4	AR	SHIM, WASHER	300 Series CRES
28	5736128-1	2	CLEVIS	410 CRES
29	5736129-1	4	PIN	440C CRES
30	58FM-44D	4	NUT	SPS
31	GB-24	4	BEARING	Torrington

Table VII-3. Motor Assembly Parts List

Title Assembly, Motor, Actuator Sheet 1 of 2

Final Assembly P/N 5736122-101

Line No.	Part Number	Qty EI	Description	Material
1	5736100-101	1	HOUSING & STATORS ASSY	
2	5736101-101	1	HOUSING ASSY	
3	5736101-1	1	HOUSING	7075 or 2024-T6 Al Aly
4	5736101-2	1	INSERT	300 Series CRES
5	NAS1394C-08L	5	INSERT (8-32 Unc)	303 CRES
6	NAS13g4C-3L	8	INSERT (10-32 Unc)	303 CRES
7	5736102-1	1	STATOR ASSY, MOTOR	
8	5736131-1	AR	STATOR LAMINATIONS	TRANCOR "T"
9	E-626	AR	BONDING AGENT	Bondmaster or Chrysler Cycleweld 55-6
10	Epoxylite 5403	AR	BONDING AGENT	Comp. G
11		2	END RING	400 Series CRES
12	5736103-1	1	STATOR ASSY, BRAKE	
13	5736105-1	1	CAP, BEARING - MOTOR	300 Series CRES
14	SR4SSTA	1	BEARING, BALL	Barden
15	5736130-101	1	ROTOR ASSY	
16	5736106-101	1	SHAFT & LAMINATIONS	
17	5736131-2	AR	ROTOR LAMINATIONS	Trancor "T"
18	5736107-1	1	SHAFT	
19	5736131-3	AR	END RING	Cartridge Brass
20	5736131-4	23	BAR	No. 1 Brass
21	SIL-FOS	AR	BRAZING ALLOY	Handy Harman Corp
22		AR	SILICONE VARNISH	Dow Corning Corp
23	5736108-1	1	TUBE	304 CRES or 300 Series

Table VII-3. Motor Assembly Parts List (Continued)

Title Assembly, Motor, Actuator Sheet 2 of 2

Final Assembly P/N 5736122-101

Line No.	Part Number	Qty EI	Description	Material
24	SR4ASSTA	2	BEARING	Barden
25	5736115-1	1	ROTOR, BRAKE	B1113 Steel
26		AR	SILICONE VARNISH	Dow Corning Corp
27	5736111-1	1	SPACER	
28	5736110-1	1	WASHER, WAVE	
29		4	SCREW, FLAT HD	8-32
30		4	SCREW, CAP	10-32
31		4	WASHER	#10
32	5736104-1	1	CLOSURE, MOTOR	7075, 2024 or 6061-T6 AlAly
33	PT02E-10-6P	1	CONNECTOR	Bendix
34		4	SCREW, CAP	4-40
35	SN63	AR	SOLDER	QQ-S-571

Table VII-4. Gear Train Parts List

Title Assembly, Gear Train, Actuator

Sheet 1 of 2

Final Assembly P/N 5736114-101

Line No.	Part Number	Qty EI	Description	Material
1	5739008	1	ASSY AND MFG	
2	5739009	1	HOUSING	
3	5739010	1	CAP	
4	5739011	1	SHAFT	440C CRES
5	5736110-1	1	SPRING, WAVE WASHER	
6	5736111-1	1	SPACER	
7		3	BALL	Tungsten Carbide
8	5739014	1	RING GEAR	440C CRES
9	5739015-101	1	GEAR ASSY	
10	5739015-1	1	CARRIER	400 Series CRES
11	5739015-2	1	GEAR	440C CRES
12	5739015-3	3	PIN	440C CRES
13	5739015-4	3	GEAR	440C CRES
14	0.0625 Dia	24	BALL	Tungsten Carbide
15		1	PIN, DOWEL	
16		4	SCREW CAP	8-32 CRES
17	2-011	1	O-RING	BUNA-N
18	5739016	3	PLANET GEAR	440C CRES
19	5739012	36	ROLLER	440C CRES
20	5739018-101	1	GEAR ASSY	
21	5739018-1	1	CARRIER	400 Series
22	5739018-2	1	GEAR	440C CRES
23	5739018-3	3	PIN	440C CRES
24	5739017	3	PLANET	440C CRES

Table VII-4. Gear Train Parts List (Continued)

Title Assembly, Gear Train, Actuator

Sheet 2 of 2

Final Assembly P/N 5736114-101

Line No.	Part Number	Qty EI	Description	Material
25	5739013	36	ROLLER	440C CRES
26	5739019	1	SHAFT ASSY	
27		1	SHAFT	PH13-8MO
28		3	PIN	440C CRES
29	S38SS	1	BEARING	Barden
30	KNL-1032T-SP	1	INSERT	Keenserts
31	KNCAL 0832T-SP	4	INSERT	Keenserts
32	SR4ASSTA	1	BEARING	Barden
33	S12562-011	1	SLIPPER SEAL	Teflon

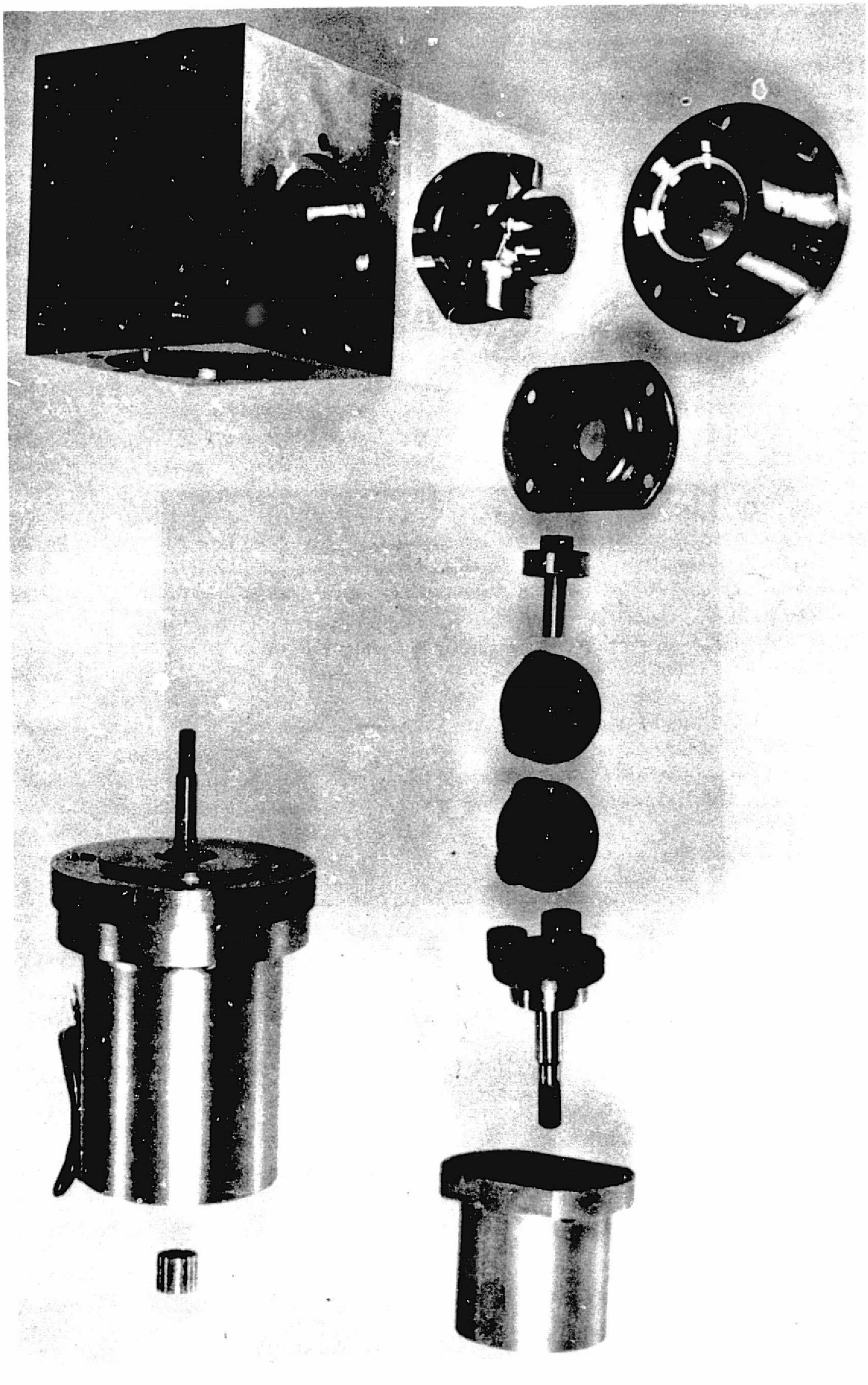

Figure 7-5. System Components

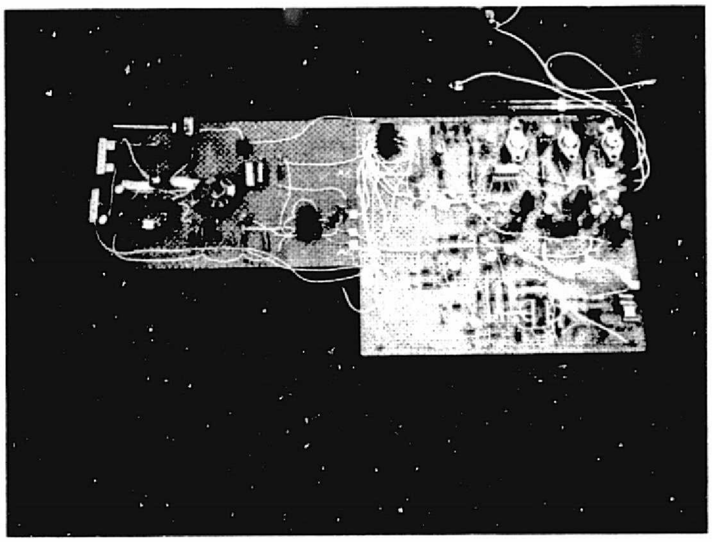

Figure 7-6. Electronic Controls Assembly

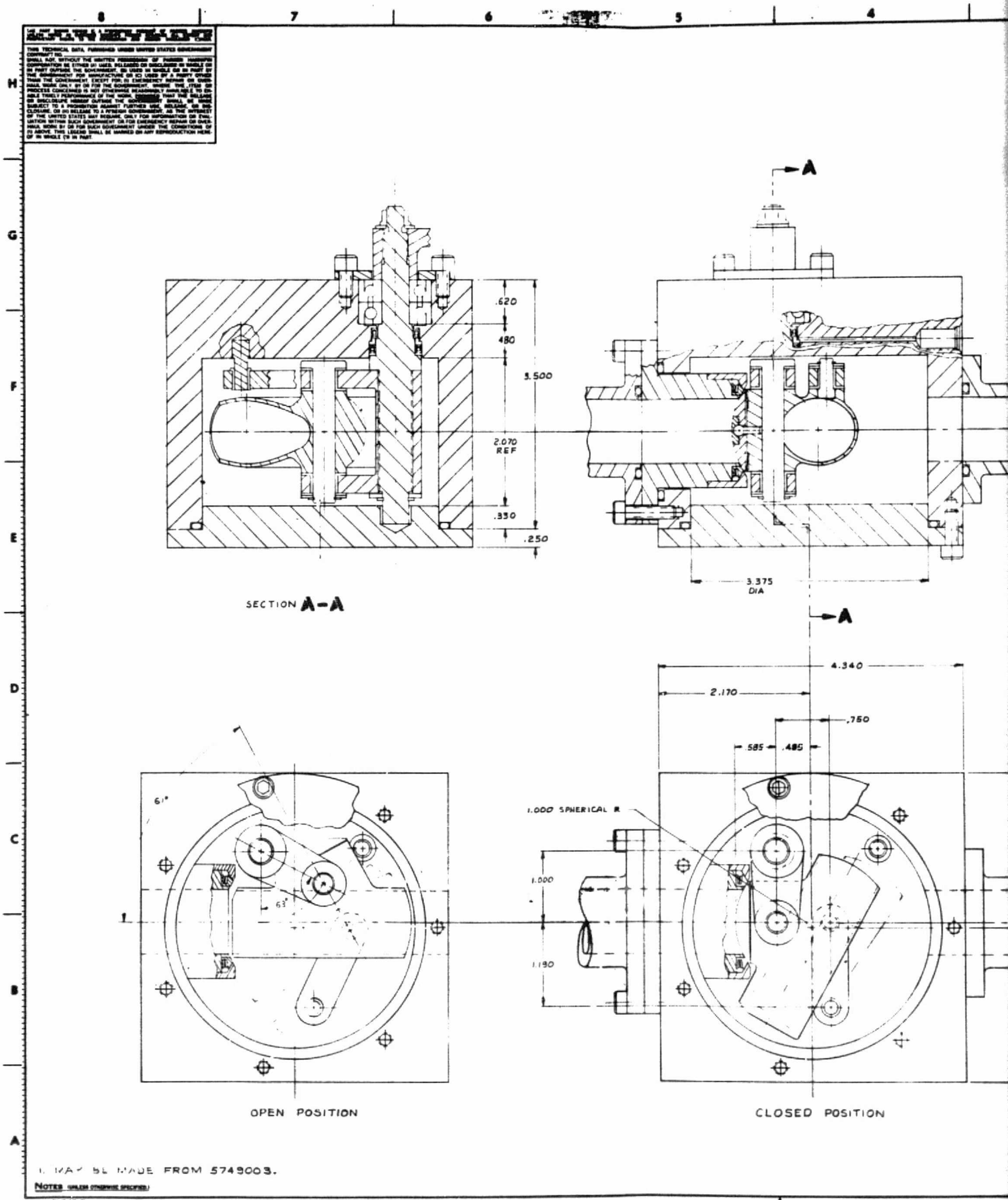

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Figure 7-7. Alternate Lifting Ball Valve

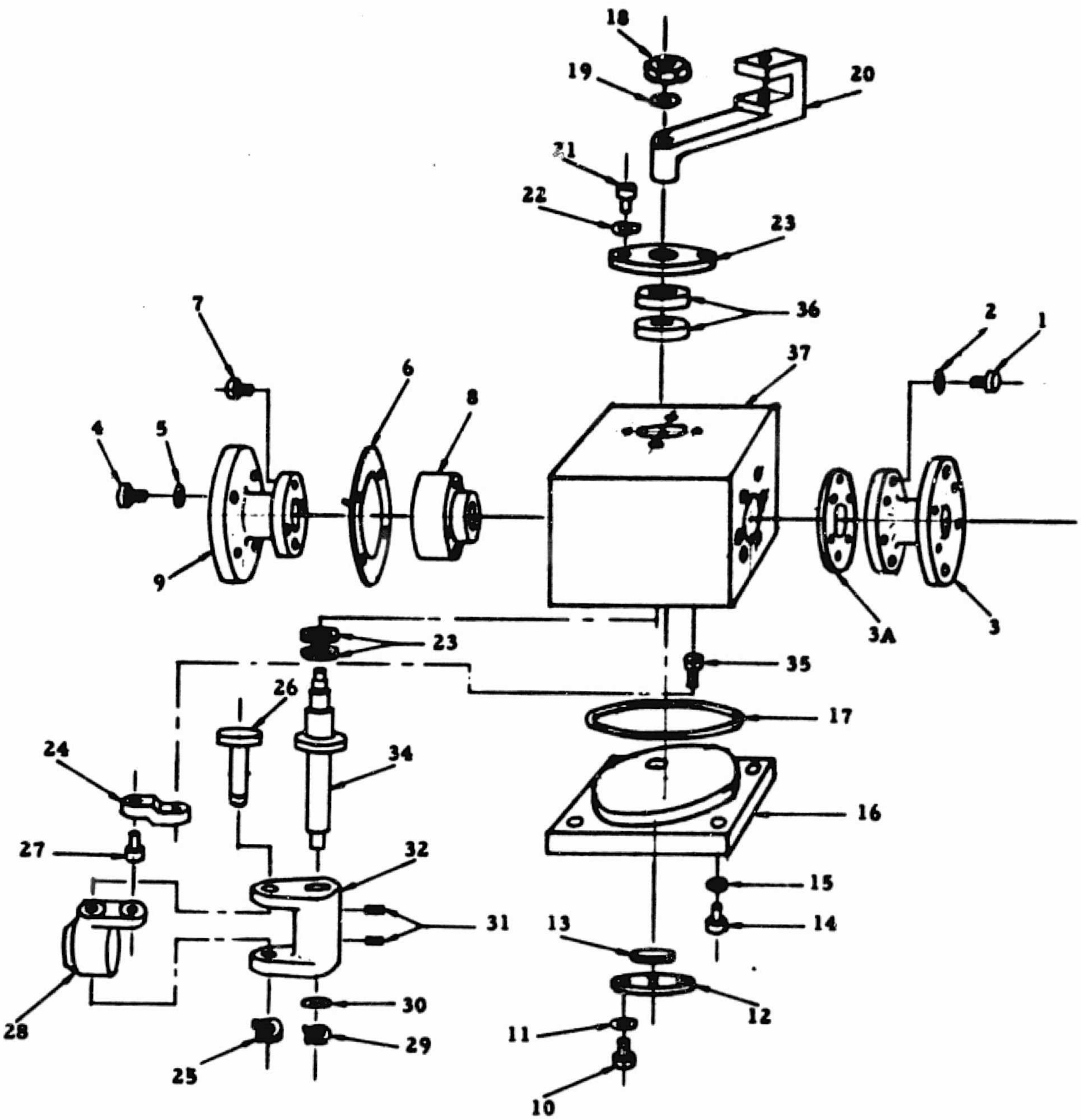


Figure 7-8. Alternate Prototype Lifting Ball Valve

Table VII-5. Parts List

Title Valve, Alternate Lifting Ball

Sheet 1 of 2

Final Assembly P/N 5749030-101

Line No.	Part Number	Qty EI	Description	Material
1	10-32 x 1/2	6	SCREW	CRES
2	0.2 ID	6	WASHER	CRES
3	F65-0-2233	1	TEST FIXTURE	304 CRES (Pass)
3A	5749025-1	1	GASKET	Teflon
4	6-32 x 1/2	4	SCREW	CRES
5	0.15 ID	4	WASHER	CRES
6	5749025-1	1	GASKET	Elastomer
7	6-32 x 1/4	4	SCREW	CRES
8	5749023	1	SEAT ASSY	6061T6 (Anodize)
9	5749022	1	FLANGE	6061T6 (Anodize)
10	6-32 x 3/8	4	SCREW	CRES
11	0.15 ID	4	SCREW	CRES
12	5749024	1	COVER	6061T6 (Anodize)
13	5749025-2	1	GASKET	Elastomer
14	10-32 x 5/8	4	SCREW	CRES
15	0.2 ID	7	WASHER	CRES
16	5749021	1	COVER, VALVE HOUSING	6061T6 (Anodize)
17	2-153	1	O-RING	Rubber (Parker Seal)
18	H46-3	1	NUT (1/4-28)	CRES (any standard nut will do)
19	0.275 ID	1	WASHER	CRES
20	5736081-1	1	LEVER	304 (Pass)
21	10-32 x 3/8	4	SCREW	CRES

Table VII-5. Parts List (Continued)

Title Valve, Alternate Lifting Ball

Sheet 2 of 2

Final Assembly P/N 5749030-101

Line No.	Part Number	Qty EI	Description	Material
22	0.2 ID	4	WASHER	CRES
23	5736080-1	1	RETAINER, BEARING	304 CRES (Pass)
24	5749017	1	LINK	303
	5749026	1	LINK	304 CRES (Alternate Part)
25	MS16624-4031	1	SNAP RING	CRES
26	5749018	1	PIN	303 (Electrolize)
27	5749019-1	1	PIN	303 (Electrolize)
28	5749016	1	BALL ASSY	303
29	MS16624-4037	1	SNAP RING	CRES
30	0.4 ID	1	WAVE WASHER OR SHIM	CRES
31	2/56 x 1/4	2	SET SCREWS	CRES
32	5749015	1	LINK, MAIN DRIVE	303
33	AR10104-112A1H	2	SEALS	Teflon (Aeroquip, Omniseal)
34	5749014	1	SHAFT, VALVE DRIVE	303 (Electrolize)
35	5749019-2	1	PIN	303 (Electrolize)
36	100H	2	BEARINGS	CRES (Barden)
37	5749020	1	HOUSING, VALVE	6061T6 (anodize)

8.0 TESTING

This section presents the development test philosophy, detail test procedures, a comparison of test results versus design requirements, and detail test results for all tests conducted. Testing was divided into functional categories and will be reported in that order. Testing categories are included as follows:

a. Valve Component Tests

b. Valve Tests

c. Electric Meter Testing

d. Electronic Control Testing

e. Planetary Gear Train Testing

f. Rotational Variable Differential Transducer Testing

g. Pneumatic Actuator Testing

8.1 Development Test Philosophy

Testing was accomplished in all functional categories with regard to qualifying the performance of specific design concepts. The valve assembly, planetary gear train assembly, motor assembly, and electronic control assembly were constructed to a testing prototype design and did not represent a flight weight configuration. However, the basic operational concepts, i.e., seat configuration, ball and visor configuration, drive shaft, linkage, gear ratios, control logic, motor operation, were all flight weight versions.

Comparative tradeoff tests were also conducted in applicable functional categories where more than one candidate design or technique was evident. Concept selections were made as a result of these comparative tests.

8.2 Test Procedures

Tests were conducted in accordance with formal test procedures and these are included in Appendix F. The test procedures used are listed as follows:

Procedure Number	Description
DVT5739001	Design Verification Test Procedure, Lifting Ball Valve Assembly, Orbiting Maneuvering Engine Propellant Valve
DVT5739006	Design Verification Test Procedure, Actuator Assembly, Orbiting Maneuvering Engine Propellant Valve
DVT5739000	Design Verification Test Procedure, Orbiting Maneuvering Engine Propellant Valve
DVT5739036	Pneumatic Actuator Test Procedures
DVT5749030	Alternate Valve Assembly Test Procedures

8.3 Test Results Versus Design Requirements

Critical operational parameters were demonstrated during testing. In the event specific verification was not demonstrated, analytical verification was either accepted or rationale developed as to how the goals could be achieved. Comparison of Table III-1 and the data included in Section 8.0 will verify that the important operational goals have been met.

8.4 Test Results

8.4.1 Valve Component Tests — Three valve seat configurations were evaluated for seat loading versus pressure loading and for cycle life capability. Ten thousand cycles, using GN_2 pressure, was established as the cycle life goal.

The three seat configurations tested are shown in Figure 8-1. Figure 8-1(a), a part number 5736067, is a TFE Teflon seat mechanically swaged into a groove contained in the outlet flange of the valve. For element test purposes, the Teflon was swaged into a 300-series stainless

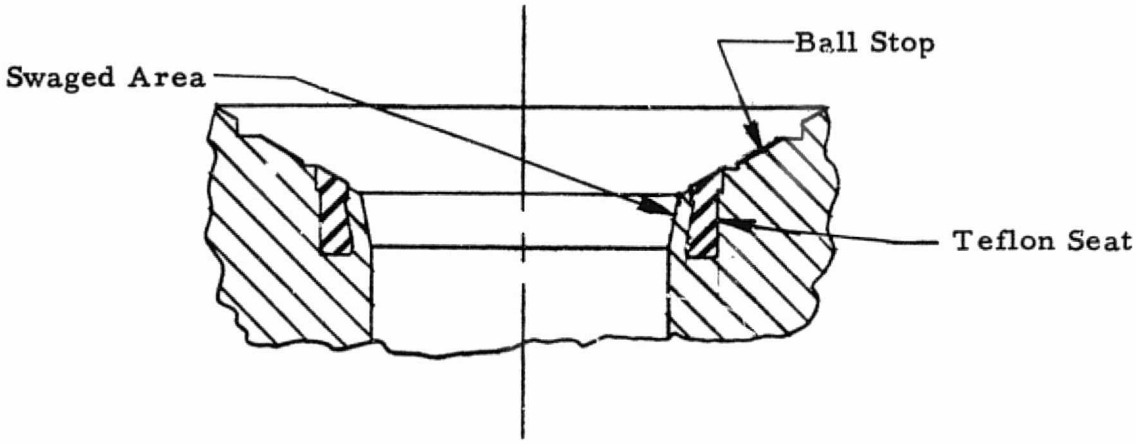


Figure 8-1(a) Swaged Seat

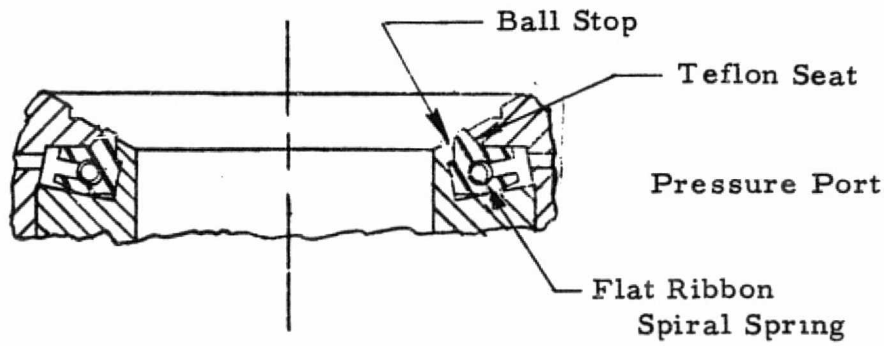


Figure 8-1(b) Flat Ribbon Spiral Spring Seat

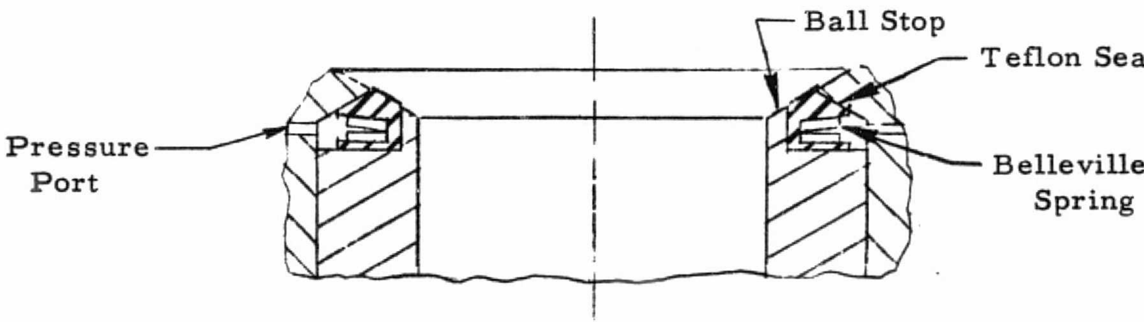


Figure 8-1(c) Belleviile Spring Seat

Figure 8-1. Valve Seat Configurations

steel manifold rather than the approved PH 13-8 MO material. This was done due to the questionableness of swaging the precipitation hardened material. Inasmuch as this testing was primarily conducted to evaluate a seat design, the material was not considered a pertinent difference. Figure 8-1(b), part number 5736070, is a TFE Teflon-formed seat with a flat spiral ribbon spring contained in the seat. This assembly consists of a two-piece manifold secured together after installing the seat assembly. Figure 6-1(c), part number 5736076, is a TFE Teflon-formed seat with face-to-face belleville springs incorporated in the assembly. This assembly also consists of a two-piece manifold secured together after installing the seat assembly.

Major differences in the three seat assemblies are as follows:

a. The swaged seat requires extensive stress relieving prior to usage.

b. The swaged seat design incorporates the ball stop upstream of the seat; the other two configurations have the stop downstream.

c. The swaged seat will cold-flow if unit pressure exceeds stress/strain limits, and it also depends upon the Teflon recoverability to provide adequate sealing. The other two configurations do not cold-flow because the spring is designed to deflect at a specific pressure loading. This always ensures uniform seat loading regardless of system pressure above spring preload.

d. During purging, the swaged seat configuration presents a more convenient contamination control profile than the other two candidate concepts.

e. The two spring-loaded seat configurations contain upstream pressure ports to inside seal areas.

8.4.1.1 Seat Deformation Test — All three seat assemblies were tested to determine the force required to seat a two-inch diameter ball onto the ball stop. An Instron force-measuring machine with a 100-pound load cell was used for this test. A chart of ball movement versus force was recorded. Detail inspection of seat dimensions did not indicate any seat deformation.

8.4.1.2 Seat Leakage Versus Inlet Pressure — To demonstrate preliminary valve seat sealing capability, each seat configuration was tested for leakage at various GN_2 inlet pressures. Seats were installed in a test setup as shown in Figure 8-2. and seat leakages were recorded for inlet pressures of 5, 40, 100, 200, and 300 psig. No spring force preload was applied to the ball for these tests. Table VIII-1 summarizes the results of these tests. All seats performed well within desired limits.

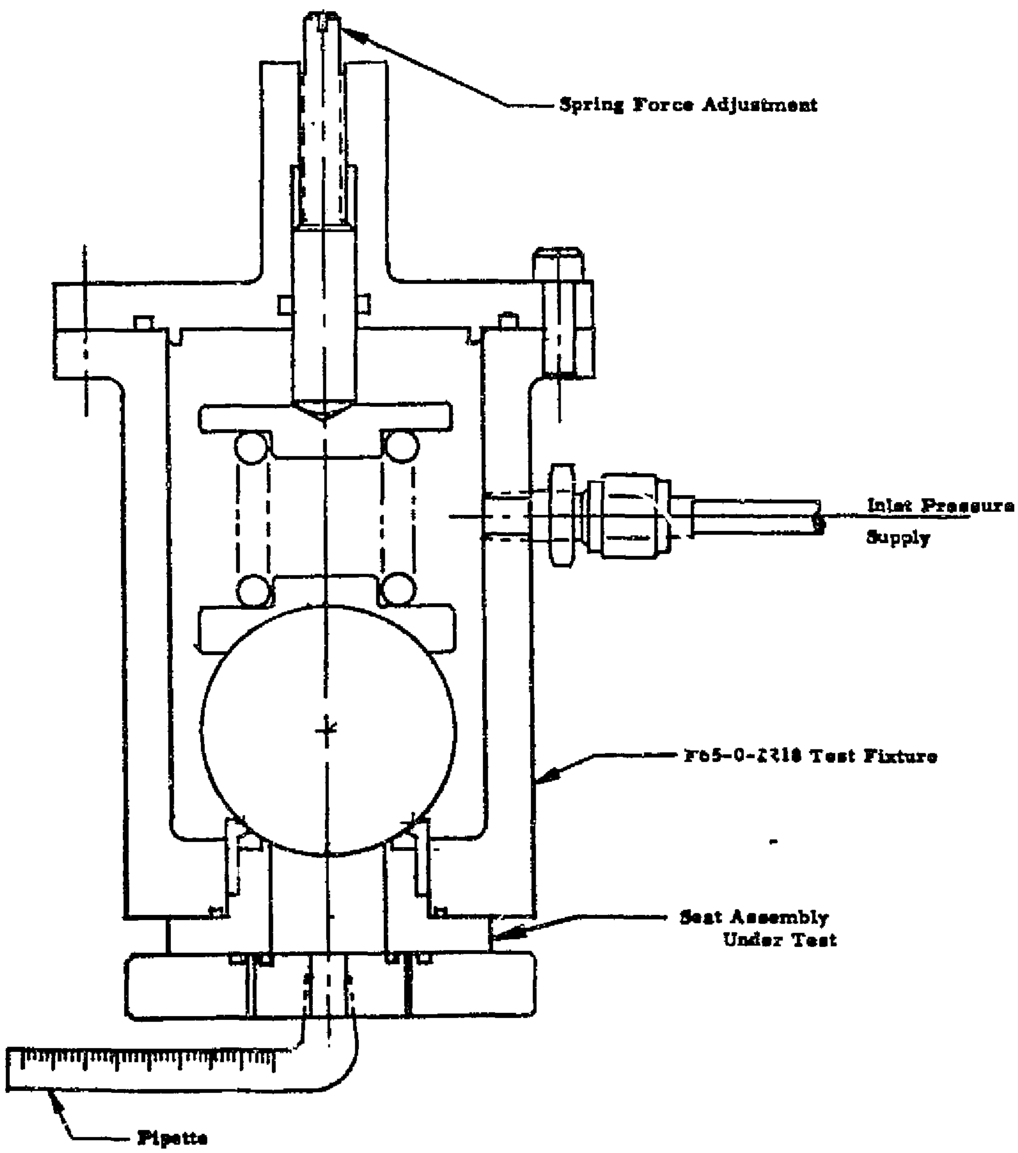


Figure 8-2. Leakage versus Inlet Pressure Test Setup

Table VIII-1. Seat Leakage versus Inlet Pressure

Seat Configuration	Seat Leakage, scc/hr of N_2				
	5 psig	40 psig	100 psig	200 psig	300 psig
Swaged Seat					
SN 01	0.1	0.0	0.2	0.4	0.0
SN 02	0.0	0.0	0.3	0.0	0.0
Flat Ribbon Spiral Spring Seat					
SN 01	0.8	0.0	0.0	0.0	0.0
SN 01A (New Seat)	0.2	0.1	0.1	0.0	0.0
SN 02	*	0.0	0.0	0.0	0.0
SN 02A (New Seat)**	6.2	3.1	5.3	6.6	6.6
Belleville Spring Seat					
SN 01	1.4	0.0	0.0	0.0	0.0
SN 02	1.2	0.0	0.0	0.0	0.0

*Leakage beyond range of pipette. Zero leakage at 10 psig.

**New seat height above bumper only 0.0008 inch, which is less than necessary.

Results published and acceptable even in this worst condition.

8.4.1.3 Life Cycle Tests — Valve seat cycle life was demonstrated in a test fixture setup with the ball stroke achieved with a pneumatic actuator. Refer to Figure 8-3 for a picture of the test setup. One of each seat type was subjected to the cycle testing with the ball stroke set at 0.010 inch. The test system was cycled, with pressure applied, and leakage measurements were conducted at selected intervals and at inlet pressures of 5, 40, 100, and 275 psig.

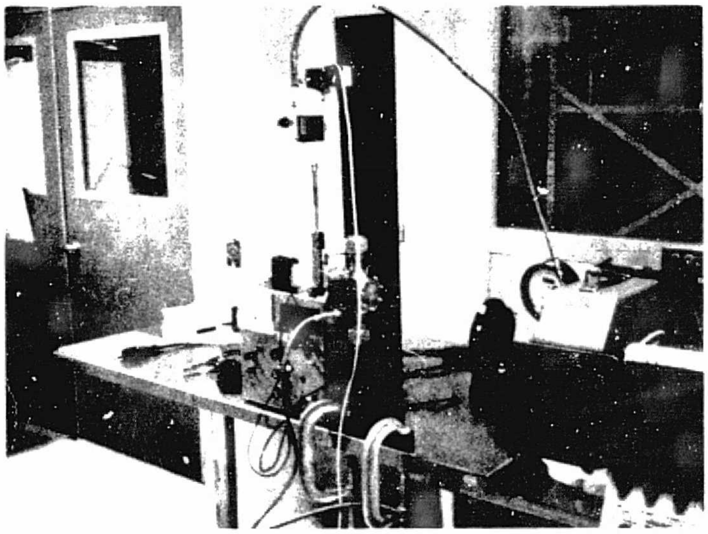


Figure 8-3. Life Cycle Test Setup

Table VIII-2 summarizes the results obtained with leakage rates prior to start, after 6000 cycles, and at the conclusion of 10,000 cycles. All cycles were run dry using GN_2 as the pressurant.

8.4.1.4 Swivel Versus Fixed Visor Tests — A study was conducted between the swivel and the fixed visor (ball) poppet designs, using leakage as the criteria for comparison. This testing was conducted to determine if either design was sufficient to eliminate special valve poppet alignment. While a test fixture was available for this testing, all the valve components were available; therefore the valve was used to perform this testing. Each seat configuration was tested against both visor configurations at 5, 40, 100, 200 and 300 psig. Leakage was measured at the valve outlet with a pipette. Results are summarized in Table VIII-3. Results conclusively support the swivel visor design for use in the valve configuration.

Table VIII-2. Life Cycle Tests

Seat Configuration	Number Cycles	Seat Leakage, scc/hr N_2 5 psig	5 psig	40 psig	275 psig
Swaged Seat AN 02	0	0.0	0.0	0.0	0.0
	6,000	0.8	0.4	0.2	0.2
	10,000	1.4	0.6	0.4	0.4
Ribbon Spring Seat SN 01	0	0.0	-	-	0.2
	6,000	0.3	0.1	0.2	0.1
	10,000	1.0	0.5	0.2	0.4
Belleville Spring Seat SN 02*	0	0.5	-	-	0.8
	6,000	6.6	3.6	0.6	0.0
	10,000	6.3	4.2	3.8	0.0

*Filter installed backward and seat imbedded with many metal particles after 300 cycles. However, seat continued to function properly.

All three seat configurations performed satisfactorily and post-test inspections indicated no signs of seat degradation, except for metal particles in one seat as noted in the table above.

8.4.1.5 Torque Tests — Valve operating torque versus inlet pressure was measured and compared with calculated values. The test was conducted by applying GN_2 pressure to the valve and manually measuring operating torque using a torque wrench. The test was repeated four times with values recorded. The torque versus inlet pressure band is presented in Figure 8-4. In the tests, the visor-to-link area and the shaft-to-visor area were lightly lubricated with lanolin. The data appears to indicate the torque versus inlet pressure to be slightly higher than predicted. With the valve in the clean, unlubricated condition, it was noted that jamming could occur when operating from the open-to-closed position. The jamming condition occurs between the visor and link assembly bearing interface.

Table VIII-3. Swivel versus Fixed Visor Tests

Seat Configuration	Swivel Visor Seat Leakage scc/hr N_2					Fixed Visor Seat Leakage scc/hr N_2				
	Test Pressure, psig									
	5	40	100	200	300	5	40	200	300	
Swaged Seat SN 01	0	0	0	0	0.0	*	*	*	*	*
SN 02	0	0	0	0	0.0	*	*	*	*	*
Ribbon Spring Seat SN 01A	0	0	0	0	0.0	*	*	*	*	*
SN 02A(1)	78	0	0	0	33.0	*	*	*	*	*
SN 02A(2)	0	0	0	0	13.2	*	*	*	*	*
Belleville Spring Seat SN 01	200	-	-	-	0.0	*	*	0.4	-	0.45
SN 01(3)	0	0	0	0	0.0	0	0	0.0	0	0.0
SN 02	-	-	-	-	-	*	*	*	*	*
SN 02(3)	-	-	-	-	-	0	0	0.0	0	0.0

*Leakage beyond range of pipette

(1)Bumper height only 0.0009 inch

(2)Twenty-pound torque applied to valve shaft

(3)Seventeen-pound torque applied to valve shaft

A comprehensive analysis was conducted on the linkage configuration, and although it is impossible to determine the exact jamming position, the analysis demonstrates many contact angles that would result in no driving forces to the visor, regardless of visor-to-link friction level. Figure 8-5 presents a schematic of the critical valve rotational intefaces, along with significant associated dimensions. Table VIII-4 provides the actual dimension tolerances and the overview description of the figure.

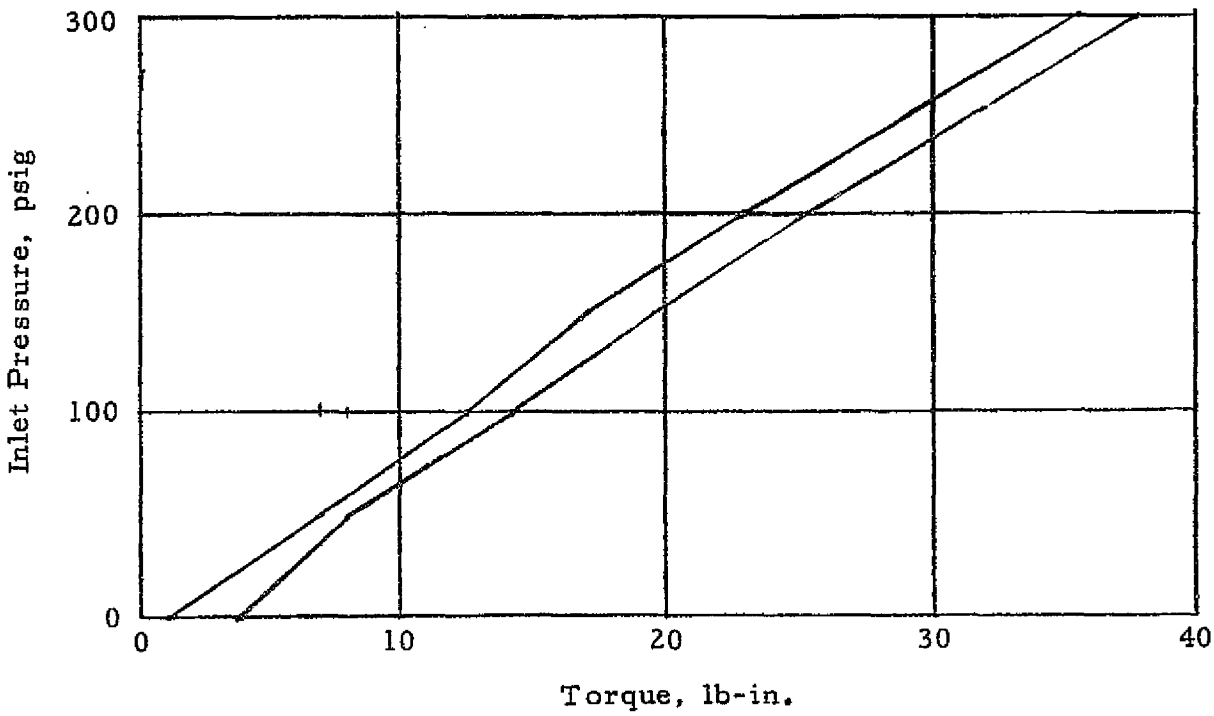


Figure 8-4. Valve Torque versus Inlet Pressure

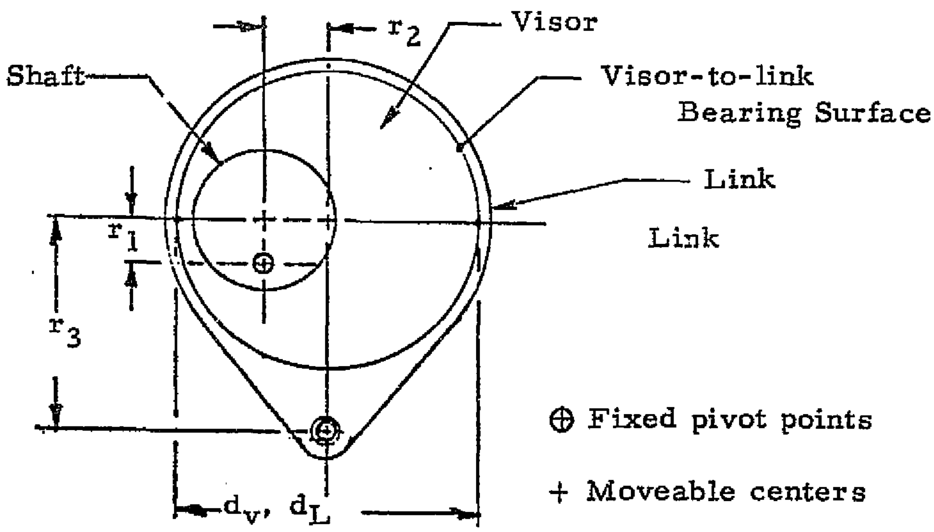


Figure 8-5. Linkage Definition

Table VIII-4. Dimensions

Reference	Dimension	Description
d_V	0.8750 ± 0.0002	Visor-bearing surface diameter
d_L	0.8766 ± 0.0001	Link-bearing surface diameter
r_1	0.125 ± 0.001	Drive shaft offset
r_2	0.138 ± 0.001	Shaft-to-visor centerline offset
r_3	0.750 ± 0.001	Link pivot-to-visor centerline length

The valve motion is described by rotating the shaft about the fixed pivot point in the clockwise direction. This motion lifts the ball from the valve seat and initiates the visor rotation (CCW) inside of the link. This motion is continued until the drive shaft has rotated 90 degrees.

The point of contact between the visor and the link assembly is dependent upon a number of variables; i.e., direction of loading, the real center of the shaft, the center of the visor, and the center of the link-bearing diameter. The point of contact can be found on the line of contact that passes through the drive shaft centerline and the visor drive shaft bore centerline. The position and angle of the line of contact is random, inasmuch as it depends on the tolerances of the manufactured components.

At the point of contact, there is a common normal and a common tangent to the two surfaces. Refer to Figure 8-6. The velocity components along the common normal of the two bodies at the point of contact are equal and in the same direction; i.e., zero relative velocity along the common normal. This definition is essential to satisfy the physical constraint imposed at the point of contact; i.e., no separation of visor and link. See Figure 8-7.

$$\left(\vec{V}_{A1N} = \vec{V}_{A2N} \text{ at point of contact.}\right)$$

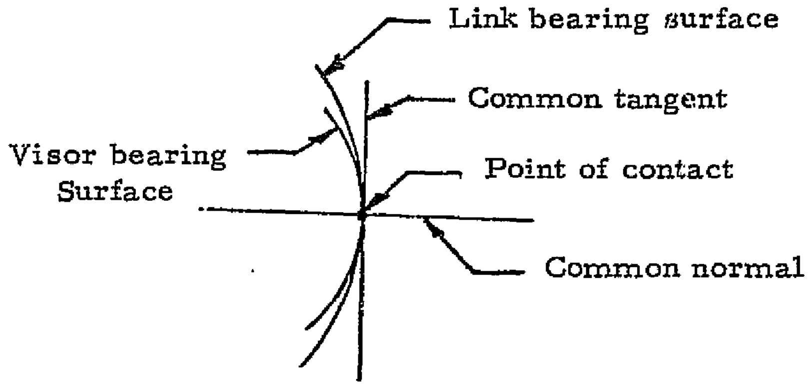
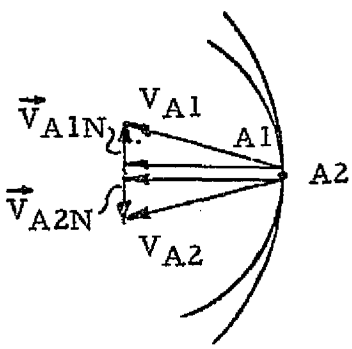


Figure 8-6. Visor Bearing versus Link Bearing Definition



A1 = Contact point on visor

A2 = Contact point on link

Figure 8-7. Contact Angle Definition

The velocity at any point of contact on the visor-to-link assembly can be calculated knowing the "instantaneous center" of rotation of the body. The "instantaneous center" of a rotating body is the intersection of two lines drawn perpendicular to the velocity vectors at the contact point.

The input work to the rotating body must be equal to or greater than the sum of the output work and the work done against the friction force.

$$\underbrace{F_{A1}V_{A1}}_{\text{driver side}} \geqslant \underbrace{F_{A2}V_{A2} + F_f V_{slip}}_{\text{driven side}}$$

Imposing the criteria of the previous discussion only indicates whether there can be relative motion or not. It does not, however, provide the overall force balance and torques involved, which can be calculated using similar principles. A graphic solution can also be used to obtain various velocity components and qualified for system condition using the following equations. Subscripts for the following equations are:

1 for the driven member (link)

2 for the driver member (visor)

$$F_2V_2 = F_1V_1 + F_f \; V_{2T} - V_{1T}$$

$$\mu \geqslant \frac{V_1^2 V_{2T} - V_2^2 V_{1T}}{\left| V_{1N}\left[V_1^2 - V_1^2 \;\; (V_{1T} - V_{2T})^2\right]\right|}$$

If it is desired to neglect friction work,

$$F_2V_2 = F_1V_1$$

$$\mu \geqslant \frac{V_1^2 V_{2T} - V_2^2 1T}{\left| V_{1N}\left(V_1^2 - V_2^2\right)\right|}$$

In order to have motion in the system, the above inequality must be satisfied. Two basic assumptions are used, as shown on the following page.

1. V_{2T} and V_{1T} are in the same direction. If not, replace V_{1T} with $-V_{1T}$.

2. $V_{2T} > V_{1T}$. If not, rearrange numerator as:

$$\left(V_2^2\ V_{1T} - V_1^2\ V_{2T}\right).$$

The subsequent figures and tables present the analytical results of estimating the dynamics of the critical rotating surfaces for the 4-bar linkage configuration of the current prototype valve design. A summary of the results is provided in Figure 8-8.

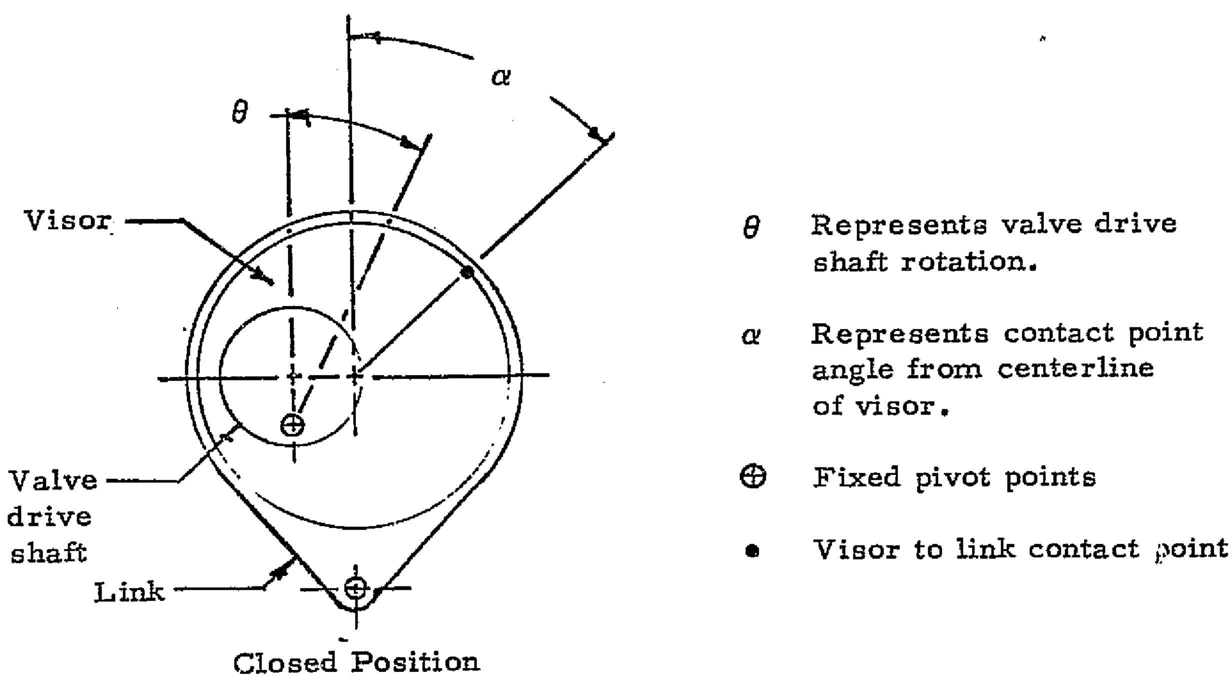


Figure 8-8. Valve Shaft Angle versus Contact Point Angle Definition

The valve linkage was oriented in the closed position and then rotated to pre-selected angles, arbitrary contact points chosen, and figures of merit calculated at those points. Negative values of figures of merit indicate linkage will not rotate, regardless of the friction coefficient between the two parts, if the contact angle is as estimated.

Table VIII-5 presents the results of the analysis and Figure 8-9 defines the parameters.

8.4.1.6 Valve Cycle Life Tests — Because the valve will rotate the full 90 degrees when lubricated between the visor and link assembly interface, it was possible and feasible to continue with certain valve tests. The previous analysis provides certainty that the 4-bar linkage concept is valid if the contact angle variance is reduced by pivoting on smaller diameters. With this in mind, tests were conducted that will demonstrate other concepts than the linkage.

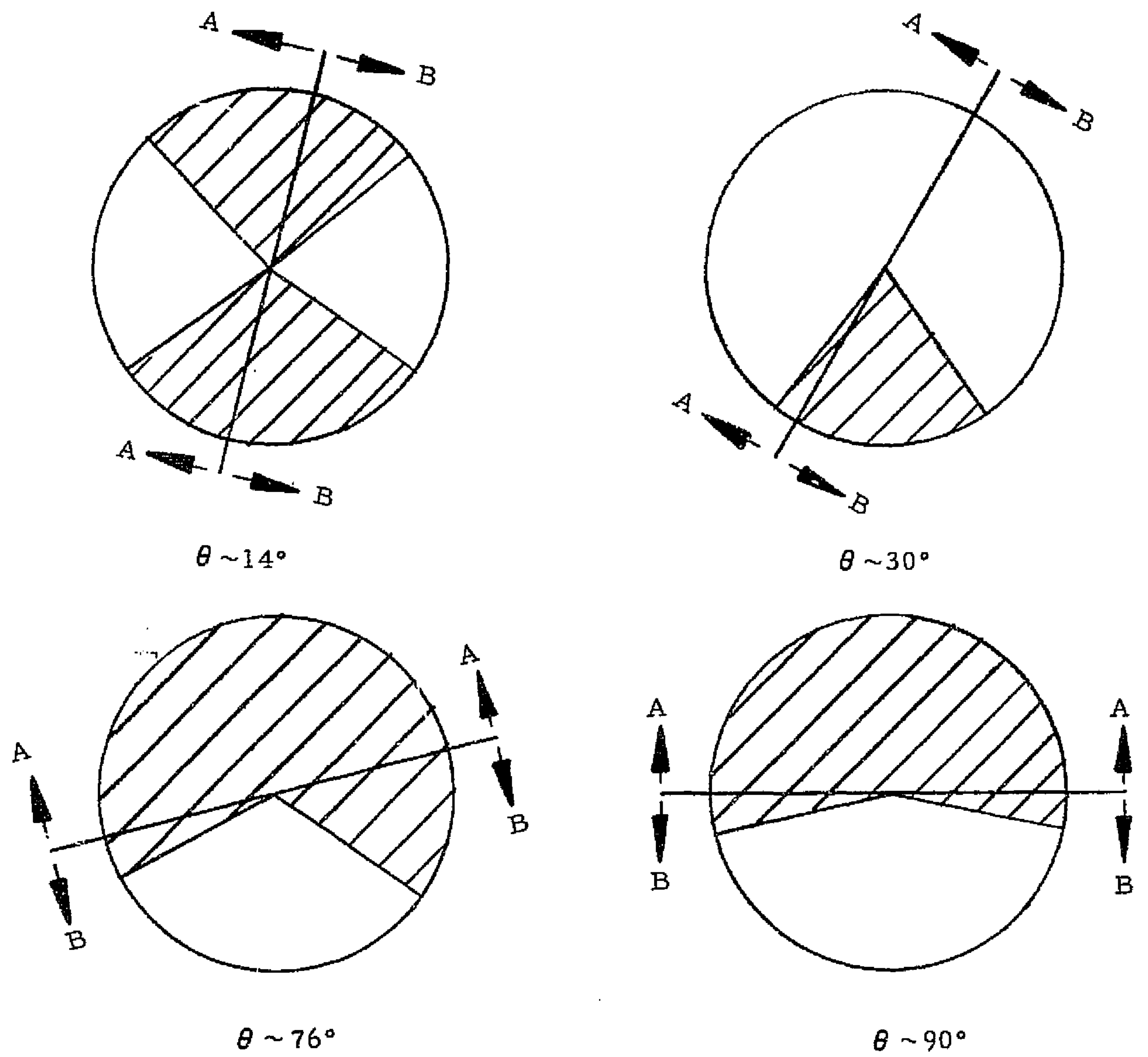
An additional 10,000-cycle life test was conducted on the valve assembly. The belleville spring seat and adjustable visor were installed in the valve and the valve was pneumatically actuated from open to closed with torque measurements taken periodically, as was leakage, at three inlet pressures. The cycles were operated with a valve inlet GN_2 pressure of 285 psig. The torque measurements were made using strain gauges mounted on the actuator drive shaft. The strain gauges used were a metal film type calibrated for both compression and tension. The rationale for measuring operating force was not to demonstrate correlation with predicted values, but instead to determine if operating forces, in a valve that exhibits high torque requirements, becomes excessive. If torque (or operating force) increased from the baseline value sufficiently to merit concern, the valve was to be disassembled and lubricated in the critical areas. It was not required to disassemble and relubricate during the 10,000 cycles. The valve torque was calculated from the valve lever angle and the strain gauge value. Figure 8-10 represents the test schematic.

Torque = (Force) (cos 45°)(1.33). Refer to Table VIII-6 for force and leakage data. The initial valve starting torque can be calculated with the valve shaft lever at 45 degrees and the moving torque can be approximated to be a maximum at the force times lever length.

A typical copy of the oscilloscope Lissajous traces taken during testing is shown in Figure 8-11 with a brief explanation of the waveform meaning.

Table VIII-5. Valve Shaft Angle versus Contact Point Angle

Contact Point Angle, α°	Valve Drive Shaft Angle, θ				
	17°	30°	75°	90°	95°
5		+0.0			
15		+4.930			
30		+2.060	-1.920	-1.660	-1.475
32	+0.006				-0.530
47	+0.0				
60		+1.090	-0.688	-0.580	
62	+2.760				
90	+1.080	+0.658	-0.284	-0.190	-0.147
120			-0.034	+0.068	
127					+0.162
129				+0.140	
130	-0.304	+0.187			
143			+0.054		
150		+0.023	+0.242		+0.364
152	-0.069				
165		-0.473		+0.556	
208				+0.930	
210		-0.292	+0.469		+0.427
212	-0.189				
228				+0.244	
236					+0.157
238				+0.146	
239	+0.311				
240		+0.191	+0.063		
254				+0.019	
262				-0.044	
270	+0.884	+0.530	-0.149	-0.116	-0.126
285				-0.263	
300		+0.930	-0.167		-0.481
302	+2.870				
322	+0.0				
325				-1.04	
330		+1.580	-1.095		-1.31



Shaded area potentially impractical to achieve desired motion when contact occurs in this area.

θ = valve input shaft rotation, zero indicates open and 90° closed valve position

AA = potential contact region for valve closing mode

BB = potential contact region for valve opening mode

Figure 8-9. Summary of Linkage Analysis

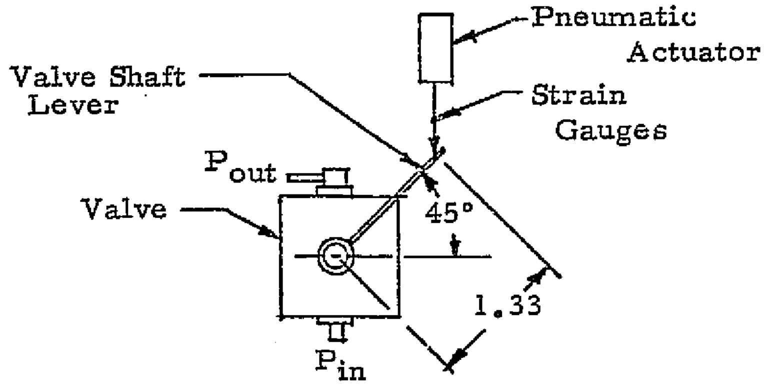


Figure 8-10. Valve Torque Test Drawing

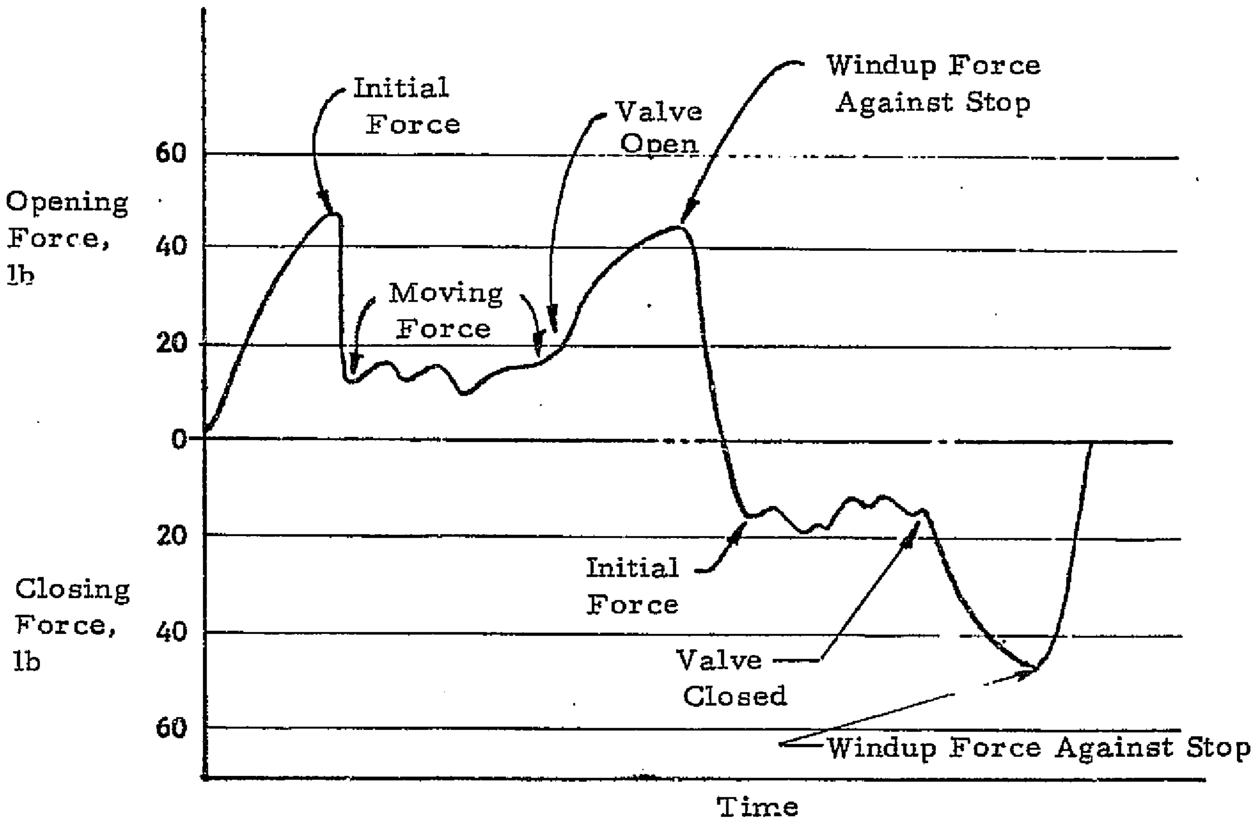


Figure 8-11. Typical Valve Force Trace

As can be noted from Table VIII-6, the seats performed exceptionally well in the valve assembly with no indications of leakage at any time. Torque was relatively consistent through the test. At completion of testing and post calibration, the valve was subjected to shaft seal leakage tests. Both seals leaked in excess of the allowable amount. The valve was disassembled and all parts inspected. Both shaft seals were heavily contamination with iron oxide, and the downstream seal was severely damaged. Upon critical inspection it was determined that the seal was damaged during installation. The valve shaft area between the two shaft seals was covered with Teflon flakes from the seals. The duplex bearing pair used were severely rusted, and this rust is what appeared on the seals. The bearings were nearly locked up, and it was evident that the valve shaft had been rotating inside the bearing bore for some time. The bearings used were a lower grade bearing than called out; however, a 22-week lead time on the required bearings caused us to compromise for

Table VIII-6. Valve Cycle Test

	Seat Leakage at Pressure (psig)			*Initial Force (lb)		*Moving Force (lb)		
Cycles	5	80	285	Open	Closed	Open	Closed	Remarks
0	-	-	-	13	13	12	12	Pin = 0
0	-	-	-	32	18	14	10	Pin = 100
0	0	0	0	51	16	14	11	Pin = 285
100	0	0	0	53	16	16	11	Pin = 285
400	0	0	0	44	22	12	18	Pin = 285
425	-	-	-	53	18	16	18	Pin = 285
550	-	-	-	53	18	16	16	Pin = 285
1,000	0	0	0	51	20	16	16	Pin = 285
1,2[illegible]0	-	-	-	50	16	16	16	Pin = 285
2,200	-	-	-	46	18	13	14	Pin = 285
3,400	-	-	-	48	14	14	12	Pin = 285
4,000	-	-	-	48	16	14	13	Pin = 285
6,000	0	0	0	44	16	13	14	Pin = 285
8,000	-	-	-	48	18	13	12	Pin = 285
9,000	-	-	-	46	14	10	13	Pin = 285
10,000	0	0	0	42	16	10	12	Pin = 285

*Force values are estimated averages.

these tests. Shaft seal installation tools were made to preclude the damage to the seals in the future. No rust or Teflon flakes were found on the seat, visor, ball assembly, or in the main chamber of the valve body. It was evident the seat had been through many cycles; however, no damage was observed. A small scratch was noted on the ball and the adjoining seat stop. No problem is envisioned with this condition.

8.4.1.7 Link Tests — The prototype unit was designed to allow comparison of using two visor guide links versus one located on top of the visor and versus one located on the bottom of the visor link. The valve was assembled to the correct configuration and torque measured as outlined previously. For this test the valve used the swivel visor and the flat ribbon spiral seat. A four-pound preload was applied to the visor with the Belleville spring retained with a snap ring. Again, and as before, there was lubricant in the valve which obviously nullifies the absolute data but still allows for comparison. Table VIII-7 summarizes the data as measured; it appeared fairly arbitrary which configuration should be used Therefore, one lower link was selected.

Table VIII-7. Link Torque versus Pressure

Applied Pressure (psig)	Torque, lb-in.			
	Two Links No Pre-load	Two Links Pre-loaded	Lower Link	Upper Link
0	1.0	4	2.0	2.0
50	7.0	7	8.0	8.0
100	12.5	14	12.5	13.5
150	17.0	19	18.0	19.5
200	23.0	25	23.5	25.0
250	29.5	32	30.5	31.5
300	38.0	38.0	36.0	38.5

8.4.1.8 Flow ΔP Testing — The valve ΔP was verified on a flow stand at 50, 90, and 100 gpm flow. The flow medium was a solvent with a specific gravity of 0.793 at 75°F. The valve Ca was calculated for the test conditions and this Ca used to determine N_2O_4 ΔP. Table VIII-8 summarizes the reduced data. This appears to be consistent with the predicted ΔP.

Table VIII-8. Flow — ΔP Test Data

Test Data		Calculated Data
Specific Gravity = 0.793 Temperature = 75°F Inlet Pressure = 90 psig		$Ca = \frac{w}{(0.6687)\sqrt{\rho \Delta P}}$
Flow Rate, gpm	ΔP, psi	$Ca = \frac{5.502}{(0.6687)\sqrt{(49.389)(0.881)}} = 1.247$
50 (5.502 lb/sec)	0.8810	
90 (9.904 lb/sec)	2.4770	$\Delta P_{N_2O_4} = \left[\frac{11.91}{(1.247)(0.6687)}\right]^2 \frac{1}{90.2}$
100 (11.005 lb/sec	3.2195	= 2.261 psi

8.4.1.9 Liquid Cycle Test — The valve assembly was subjected to twenty cycles on a liquid test stand. The flow medium was a solvent with a specific gravity of 0.793 at 75°F. The valve was manually actuated using a torque wrengh and torque values were noted. Rotating surfaces of the valve were lubricated for this test. Table VIII-9 summarizes the test data.

8.4.1.10 Temperature Tests — The three candidate seat configurations, i.e., swaged seat, flat ribbon, spiral seat, and belleville spring seat, were tested in the valve for GN_2 leakage at 70 and 200°F. Tests were conducted at pressures of 5, 40, 100, 200, and 300 psig with no leakage greater than 0.2 scch noted.

8.4.1.11 Proof Pressure Tests — The valve assembly was subjected to proof pressure testing of 435 ± 10 psig for 5 minutes with no indication of material damage or deformation. Valve seat leakage was measured subsequent to proof pressure tests; no leakage was observed at 5, 40, 100, 200 or 300 psig.

Table VIII-9. Liquid Cycle Test Data

Specific Gravity	= 0.793
Temperature	= 75°F
Inlet Pressure	= 91 psig
ΔP	= 2 psi
Flow Rate	= 70 gpm (7.70 lb/sec)

Torque Data:

Initial torque (close-to-open)	= 2 - 3 lb-in.
Maximum torque to open	= 12 - 18 lb-in.
Holding torque (open)	= 12 - 15 lb-in.
Initial torque (open-to-close)	= 5 lb-in.

Valve goes to close position from system pressure after first 10 degrees rotation.

Leakage subsequent to test at 5, 80, and 285 psig was zero in all cases.

8.4.1.12 Electronic Control Testing — The breadboard electronic control circuit was tested using a resistive load in place of the three-phase motor. The circuit optimization process resulted in some minor circuit value adjustment, and the addition of a small circuit to prevent initial power-on current drain. A schematic is included as Figure 8-12.

Performance tests were conducted with input voltages up to 40 volts and current levels up to 14 amps into a resistive load. Saturation of the switching transistors was satisfactory during testing.

Exact tracings of Lissajous patterns recorded from the oscilloscope at significant nodes are included as Figures 8-13 through 8-17.

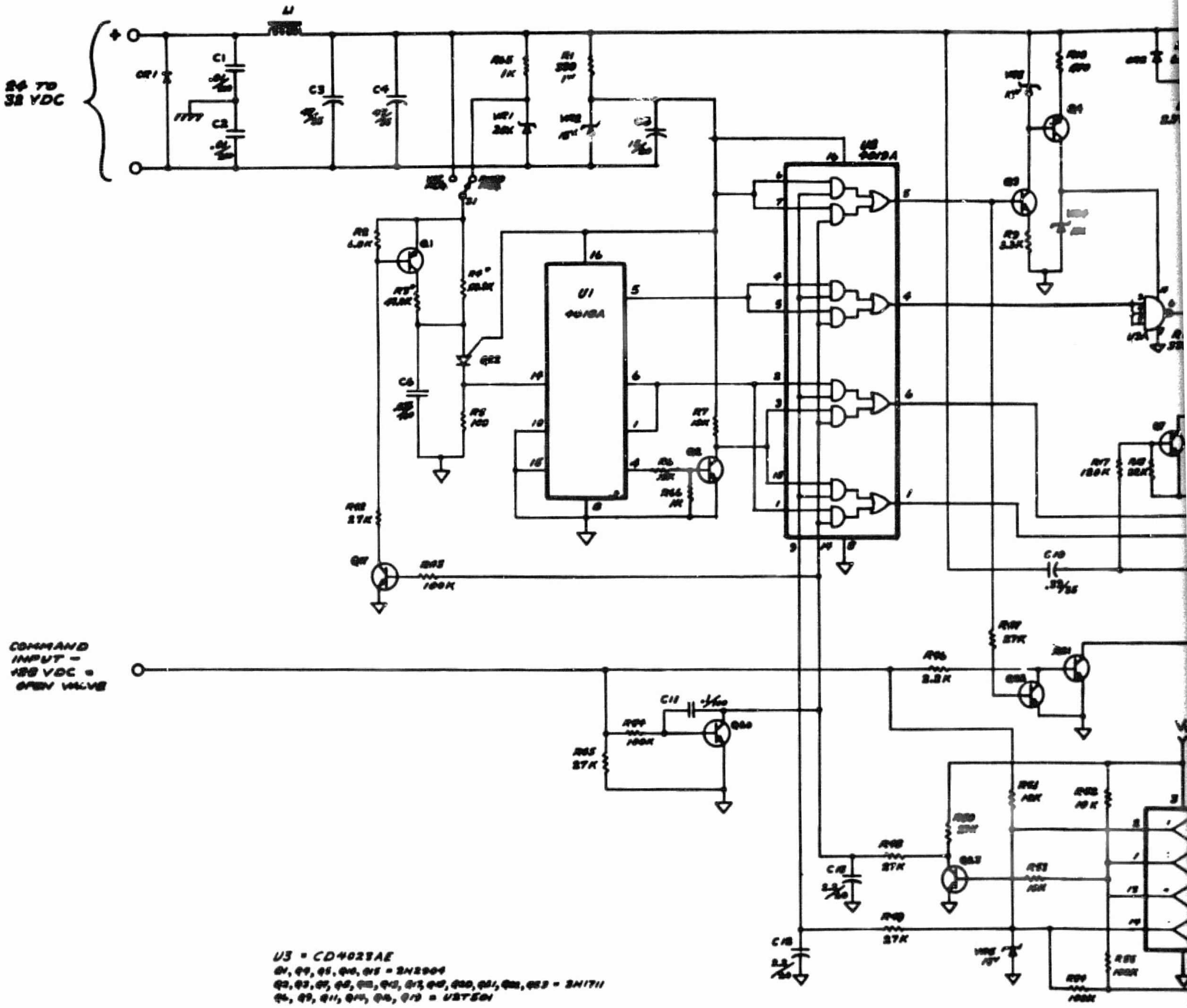


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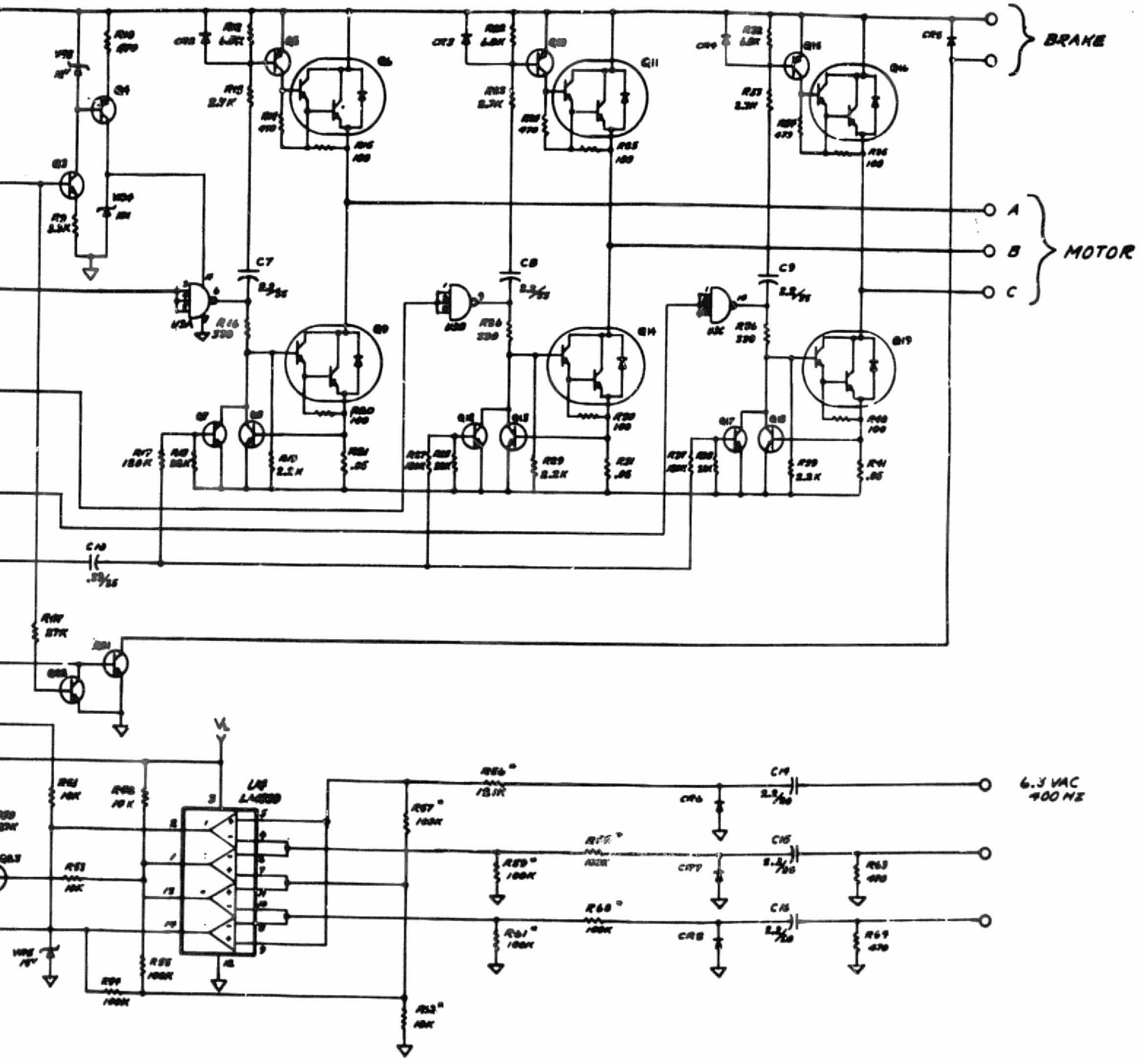
Figure 8-12. Electronic Control Pa

FOLDOUT FRAME



Electronic Control Package Schematic

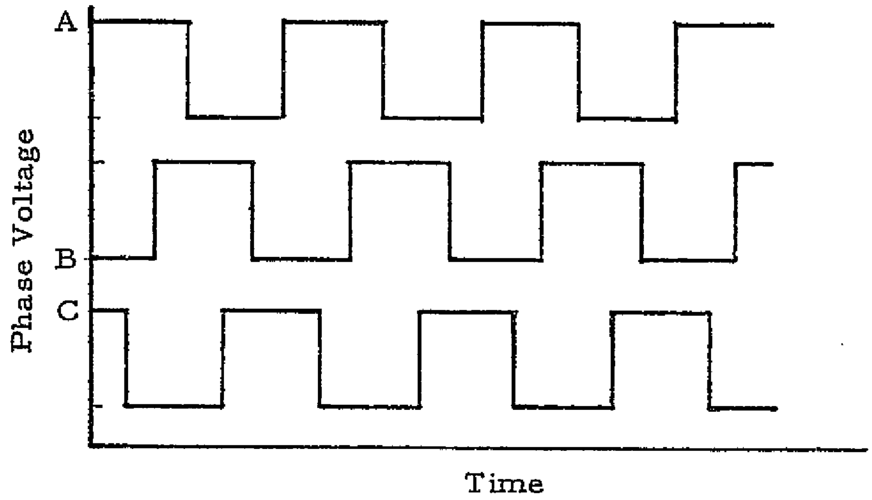


Figure 8-13. Electronic Control Output Voltage to Motor

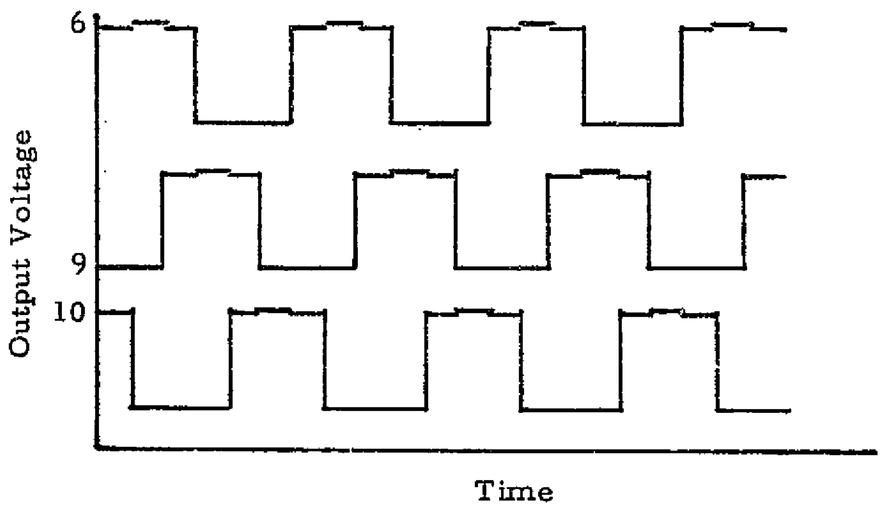


Figure 8-14. Output Voltage from U3 Pins 6, 9 and 10

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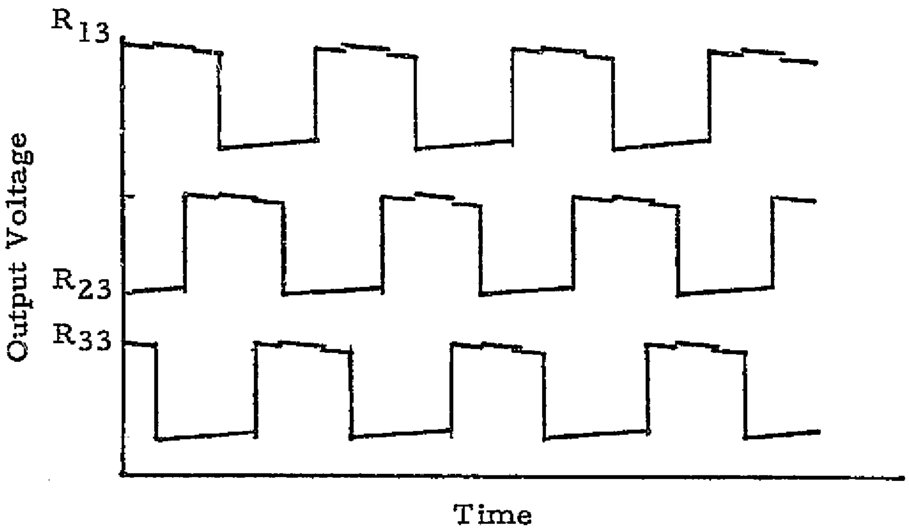

Figure 8-15. Output Voltage from Junction R_{13}-C7, R_{23}-C8, R_{33}-C9

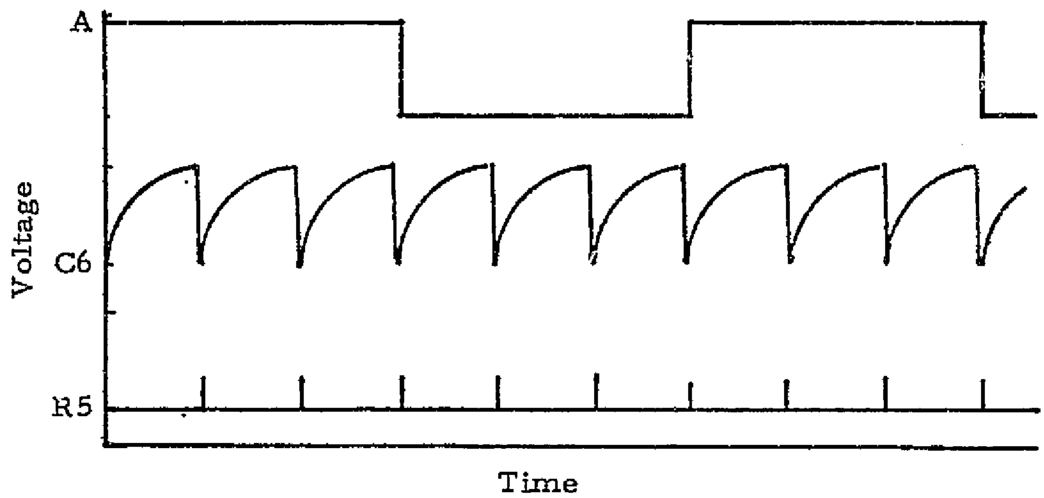

Figure 8-16. Output A, Voltage Across C6, Voltage Across R5

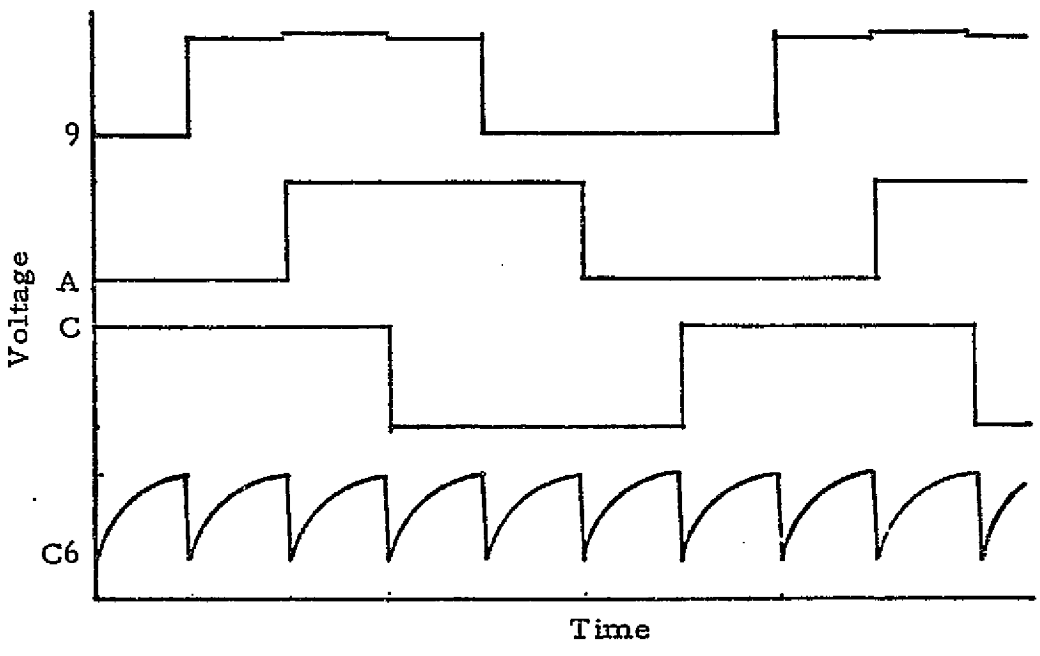

Figure 8-17. Output U3 Pin 9, Voltage A, Voltage C, Across C6

The input circuit was tested to demonstrate a 3-phase output variation with a change of input voltage. Typical data recorded during testing is shown in Figure 8-18. Figure 8-18(a) shows zener control output for valve close demand as a function of input voltage. Figure 8-18(b) shows frequency of oscillator output for valve close demand as a function of input voltage. Figure 8-18(c) shows 3-phase output with variation of input voltage. Figure 8-18(d) shows variation of oscillator output for valve open demand as a function of input voltage.

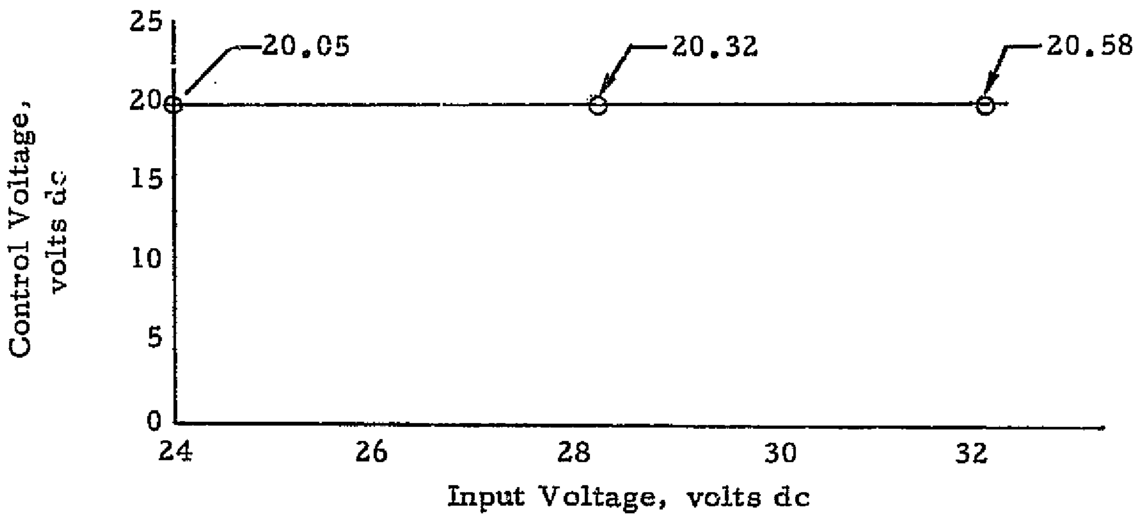

Figure 8-18(a). Zener Control Output with Input Voltage Variation

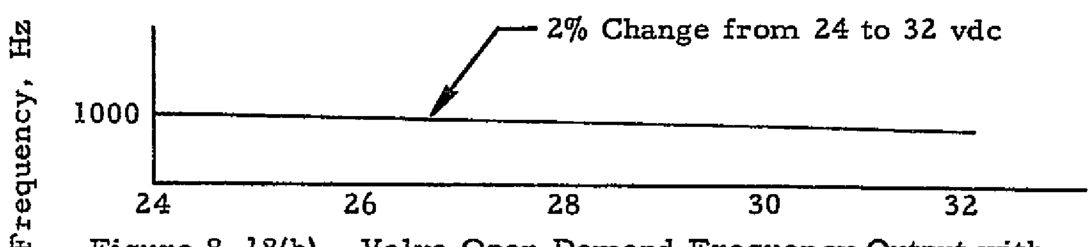

Figure 8-18(b). Valve Open Demand Frequency Output with Input Voltage Variation

Figure 8-18. Electronic Control Test Data

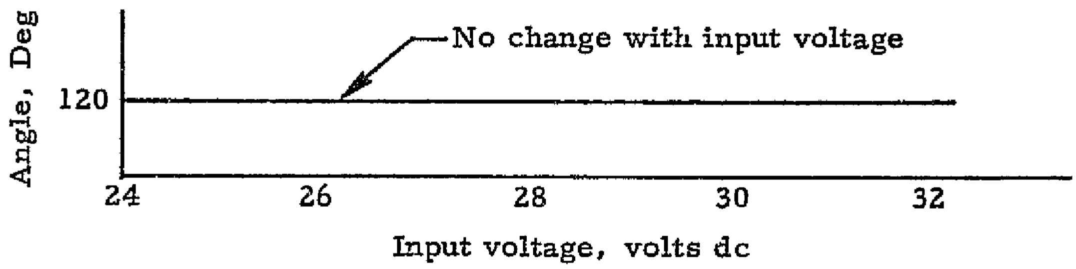

Figure 8-18(c). Three-phase Angular Variation with Input Voltage Variation

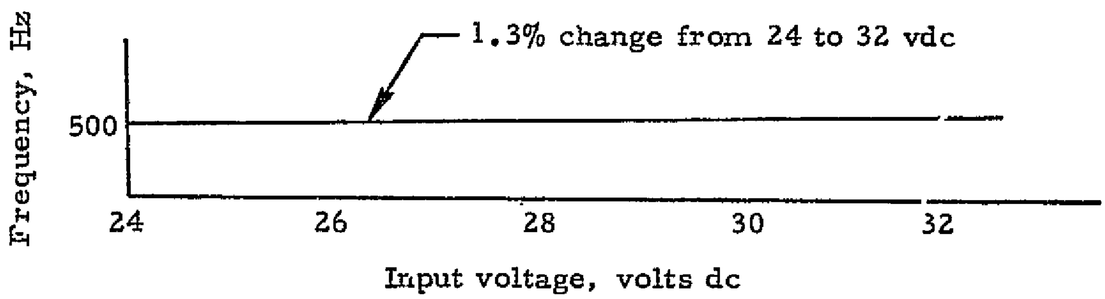

Figure 8-18(d). Valve Close Demand Frequency Output with Input Voltage Variation

Figure 8-18. Electronic Control Test Data (Continued)

8.4.2 Alternate Lifting Ball Valve Concept Testing — The alternate valve concept, as explained in the detail design section, requires a minimum of testing due to the major differences only being in the linkage dimensions and not in the sealing concept. The major consideration for testing demonstration is the operating torque and the 10,000 cycles life capability.

8.4.2.1 Torque Measurement — The torque to operate the valve was manually measured using a chatillon scale. Closed-to-open, then open-to-closed, were measured ten times each. The valve stroke was 65 degrees. Tables VIII-10 and VIII-11 summarize torque measurements. No pressure was applied and no shaft seals were installed.

Table VIII-10. Closed-to-Open

Test	Force, lb	Torque lb-in.
1.	0.40	0.532
2.	0.40	0.532
3.	0.40	0.532
4.	0.40	0.532
5.	0.40	0.532
6.	0.35	0.4655
7.	0.40	0.532
8.	0.35	0.4655
9.	0.40	0.532

Table VIII-11. Open-to-Closed

Test	Force, lb	Torque lb-in.
1.	0.8	1.064
2.	0.8	1.064
3.	1.0	1.330
4.	1.0	1.330
5.	0.8	1.064
6.	0.8	1.064
7.	0.8	1.064
8.	0.9	1.197
9.	0.9	1.197

The valve shaft seals were installed and torque measurements again taken. Tables VIII-12 and VIII-13 summarize the results of these tests.

Table VIII-12. Closed-to-Open

Test	Force, lb	Torque lb-in.
1.	0.70	0.9310
2.	0.70	0.9310
3.	0.80	1.0640
4.	0.80	1.0640
5.	0.75	0.9975
6.	0.80	1.0640
7.	0.75	0.9975
8.	0.82	1.0906
9.	0.80	1.0640
10.	0.80	1.0640

Table VIII-13. Open-to-Closed

Test	Force, lb	Torque lb-in.
1.	1.10	1.463
2.	1.10	1.463
3.	1.10	1.463
4.	1.15	1.5295
5.	1.10	1.463
6.	1.20	1.596
7.	1.18	1.5894
8.	1.20	1.596
9.	0.90	1.197
10.	1.15	1.5295

Valve linkage operation was extremely smooth and maintained a consistent feed throughout testing. The torque increase resultant from installing two seals is 0.532 lb-in. or 0.266 lb-in. per seal. The shaft seal friction torque estimated originally was 0.52 lb-in. per seal.

The valve was pressurized with GN_2 to 100 psig, then opening and closing torques measured without pressure; no changes from original values were noted. The valve was pressurized at 50 and 100 psig and torque measurements taken. Table VIII-14 summarizes the test data.

Table VIII-14. Torque versus Pressure Measurements

Force, lb	Torque, in.-lb	Pressure, psig	Measurement Direction
3.8	5.054	50	Close-to-open
4.0	5.320	50	Close-to-open
4.0	5.320	50	Close-to-open
4.0	5.320	50	Close-to-open
3.8	5.054	50	Close-to-open
2.0	2.660	50	Open-to-close
2.2	2.926	50	Open-to-close
1.8	2.394	50	Open-to-close
2.2	2.660	50	Open-to-close
2.0	2.660	50	Open-to-close
14.0	18.620	100	Close-to-open
13.0	17.290	100	Close-to-open
14.0	18.620	100	Close-to-open

Subsequent to pressurized torque measurements, unpressurized torque values were measured; no change from the original values was observed.

8.4.2.2 Valve Cycle Testing — The alternate valve assembly was cycled 10,000 cycles with a pneumatic actuator. The valve was unpressurized throughout this testing. Subsequent to the test all moving parts were examined for signs of wear or electrolize chipping. No signs of surface deterioration was evident. Torque values of the valve were measured subsequent to the life testing with no changes noted from the original values.

The alternate valve assembly tests demonstrate the linkage concept to be the most viable configuration and with linkage length optimization being performed probably could result in a very superior type valve. The torque values measured do not appear to be excessive and due to the manufacturing control implemented, this torque probably represents the worst case condition. On all parts there were no concentricities, co-axialism, normality, flatness, or parallelism called out. In most cases, open tolerances were provided and no inspections were conducted on any parts. With production type controls implemented on this design, the operating torque could probably be further reduced. The bearings used were somewhat lower quality than would be selected for a production design, due to a 22-week procurement time.

8.4.3 Electric Motor Testing — Motor testing was conducted on 5736112-102 Motor Assembly in accordance with the instructions contained in test document DVT5739006, Revision A, dated 11 October 1974.

The philosophy of testing was to demonstrate design point operation only as a reference with selected parametric scanning being the main consideration. From this data most efficient motor operating points can be selected. The electronic control module was used in conjunction with the motor tests to provide the design operating signal.

8.4.3.1 DC Resistance Test of Primary Windings — The dc resistance between motor terminals was measured to assure resistance balance of the coils. A wheatstone bridge was used to perform the measurement. All coils were within 1.3 percent of each other. Table VIII-15 summarizes the results.

Table VIII-15. Motor Resistance

Connection, Wire Color	Resistance, ohms
Red to Black	0.699
Red to Green	0.696
Green to Black	0.690

8.4.3.2 Locked Rotor Torque Test — Locked rotor torque tests were conducted to evaluate steady state torque requirement for the motor-locked rotor condition.

Reduction of the data indicates that at design point operation the motor develops sufficient torque to meet the design requirement; i.e., 28 vdc input, 175 Hz, 1.349 lb-in. A curve of torque versus input voltage is provided in Figure 8-19. With a torque apportionment as shown

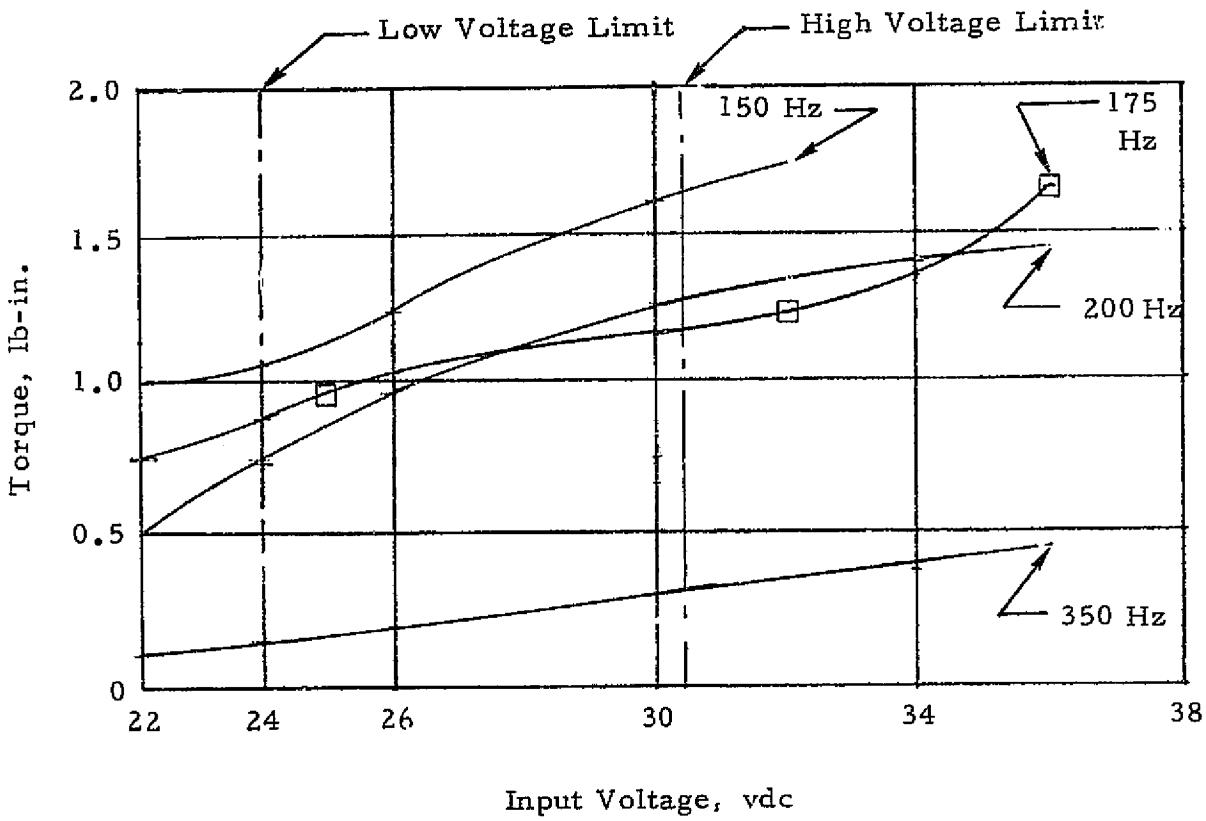

Figure 8-19. Locked Rotor Torque versus Input Voltage

in Figure 8-20, the 2 : 1 torque margin for all conditions does not exist in this design. It is, of course, no trouble to provide this margin with the information obtained from motor testing. Table VIII-16 shows calculated torque required for valve.

The maximum torque required to operate the valve is at the closed-to-open condition. At this condition, the actual torque requirement is 219 lb-in. resulting in a motor requirement of 0.77 lb-in. With the battery voltage depleted to the lower limit of 24 vdc, a margin of 1.14 is realized. This represents the worst possible condition the system could realize as presently designed.

To perform the testing, the motor was held in a bench vise and a 4.5-inch long arm attached to the output shaft. An 0- to 1000-gram tension scale was suspended from the arm and a tare established. The motor

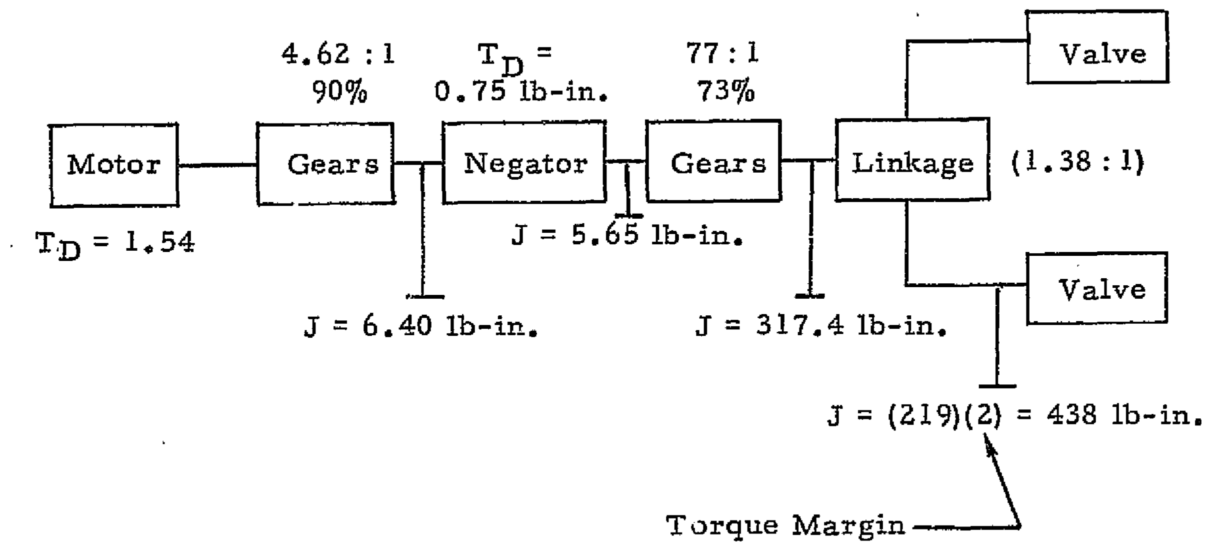

Figure 8-20. Torque Apportionment

Table VIII-16. Calculated Valve Torque

Seat Diameter	= 1.026 inch
Seat Area	$= \frac{\pi(1.026)^2}{4} = 0.83$ in.^{2}
Link Length	= 1.00 inch
System Pressure	= 265 psig
Torque Required	= (265)(0.83)(1) = 219 lb-in.

was connected to the electronic control and operated to determine locked rotor torque at various conditions of frequency and input voltage. The torque value at 150 Hz and 36 vdc input voltage was not run to avoid possible excessive loading on the electronic control. At 150 Hz and 28 vdc, the motor seems to saturate rapidly.

8.4.3.3 Shaft Speed versus Frequency at No-Load Condition — Speed versus frequency plot is presented in Figure 8-21. The motor plot is compared with the theoretical speed and data correlates well with both

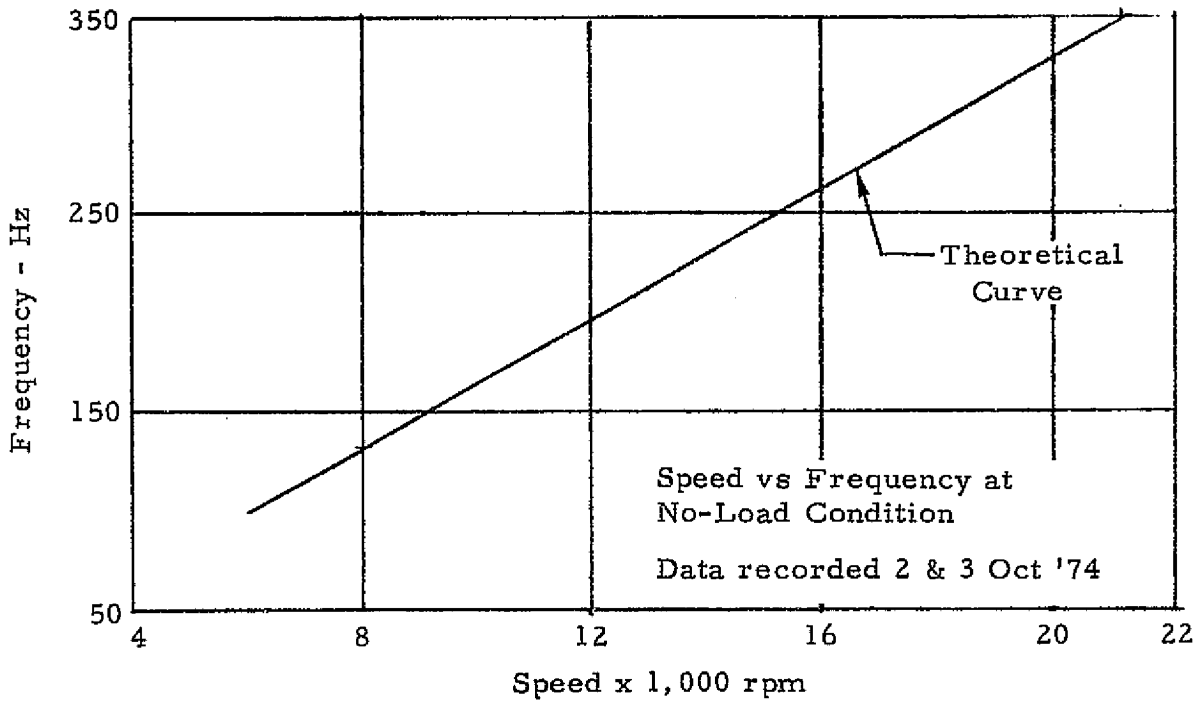

Figure 8-21. No-load Motor Speed versus Frequency

slope and amplitude. It is noted that the measured motor speed is slightly less than the theoretical speed. This is due to the rotor slip which is characteristic of this type motor.

All data was obtained by either running specific tests or by transferring the initial "no-load" test point at the beginning of each run. The test setup is the same as used for the "speed versus torque at variable load" test. Motor speed was monitored by a steel rotor with six magnetic lobes acting in conjunction with an electromagnetic pickup connector to an electronic cycle counter with reference to the facility supply power (115 vac, 60 Hz). Motor rpm was displayed as [(rpm)(6)]/60.

8.4.3.4 Speed versus Torque at Variable Load (Dyno Test) — Speed versus torque curves for variable voltage input and electronic control frequencies are included at Figures 8-22 through 8-26. Curves present typical speed/torque type curves for this type motor.

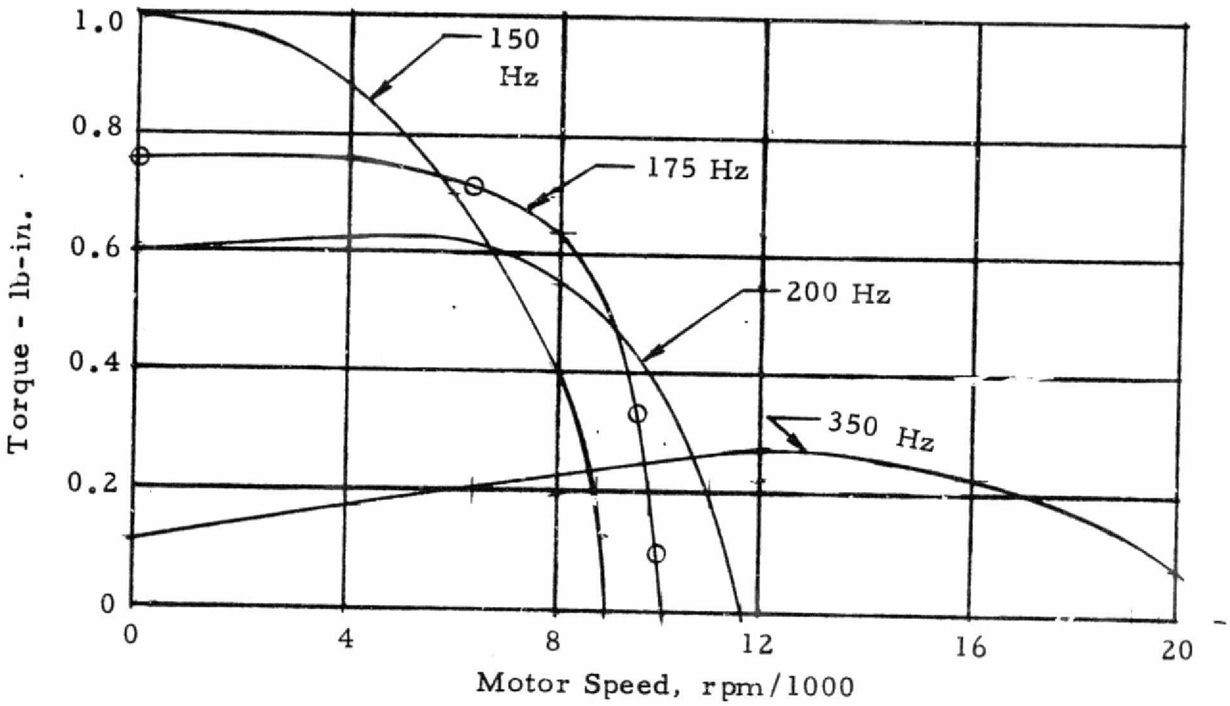

Figure 8-22. Motor Speed vs Torque Curve at 22 vdc Supply Voltage

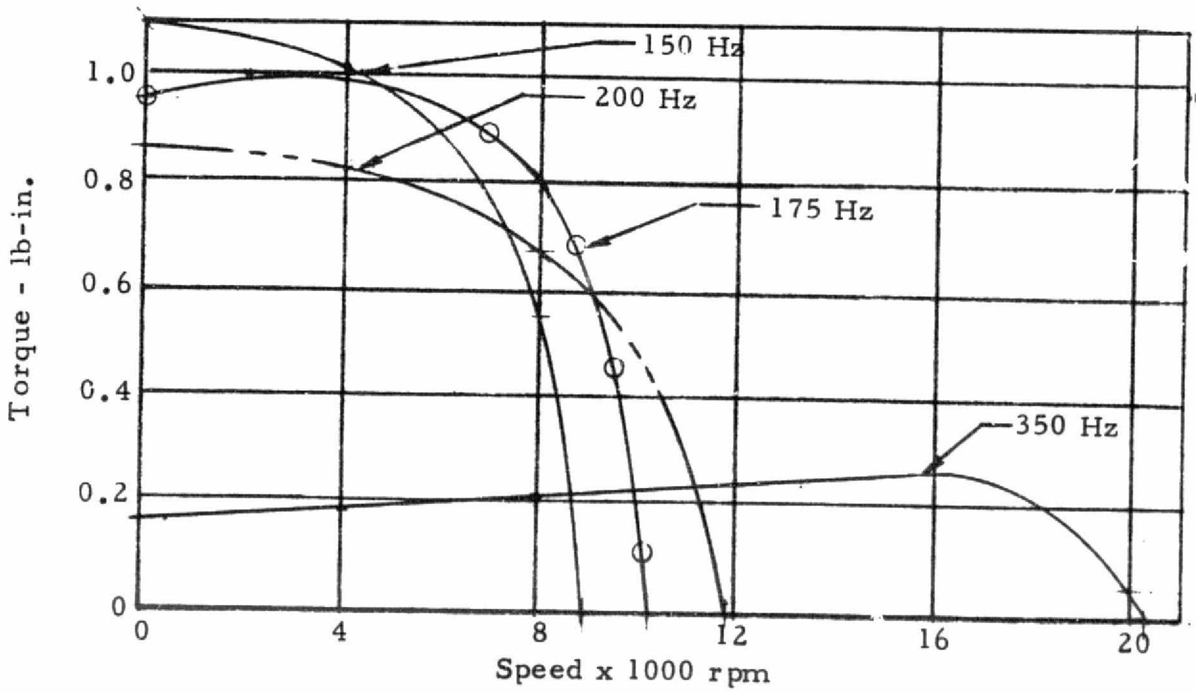

Figure 8-23. Motor Speed versus Torque Curve at 25 vdc Supply Voltage

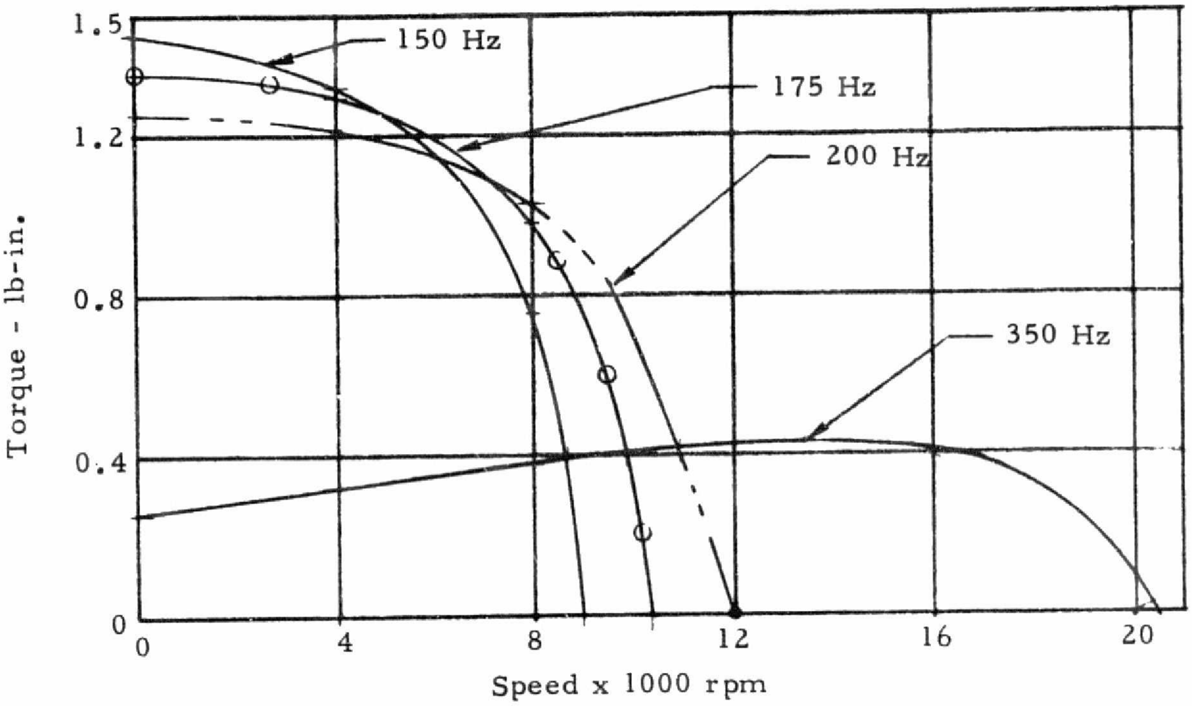

Figure 8-24. Motor Speed versus Torque Curve at 28 vdc Supply Voltage

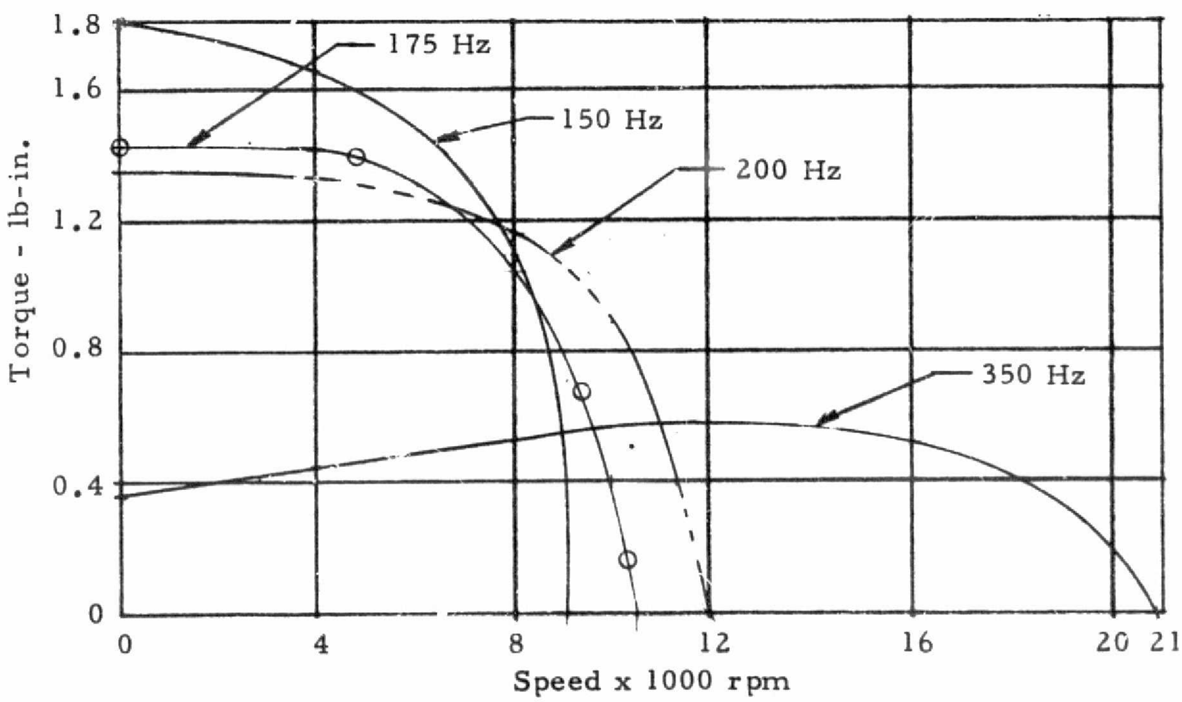

Figure 8-25. Motor Speed versus Torque Curve at 32 vdc Supply Voltage

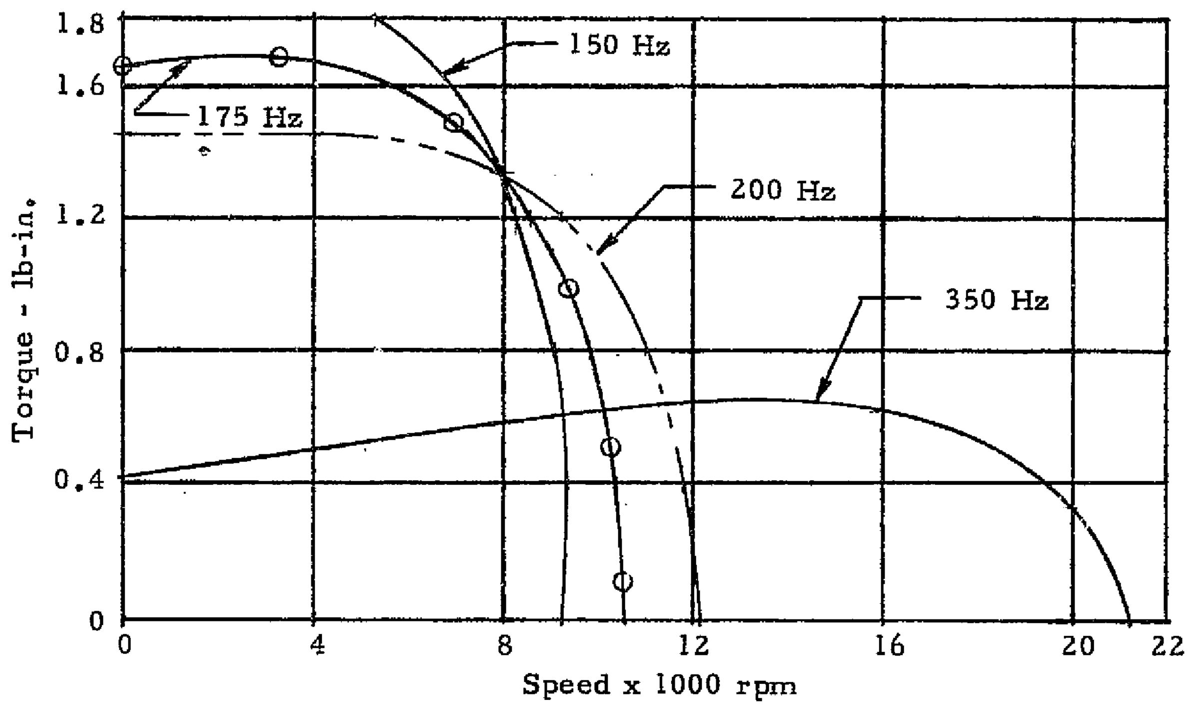

Figure 8-26. Motor Speed versus Torque Curve at 36 vdc Supply Voltage

The design point for electronic control frequency is also the best suited frequency as indicated by the test data. The design point operation is best supported by this frequency at all stages of operation. At an initial valve torque, to the motor, of 0.77 lb-in. and decreasing with opening angle, the maximum operating time is as follows:

$$\left(\frac{8090 \text{ Rev}}{\text{min}}\right)\left(\frac{\text{min}}{60 \text{ sec}}\right)\left(\frac{1 \text{ Rev Gear Box}}{77 \text{ Rev Mtr}}\right) = 1.751 \ \frac{\text{Rev Gear Box}}{\text{Sec Mtr}}$$

$$\left(\frac{132°}{\text{Valve Stroke}}\right)\left(\frac{\text{Rev}}{360°}\right) = 0.3666 \ \frac{\text{Rev}}{\text{Valve Stroke}}$$

$$\left(\frac{0.3666 \text{ Rev}}{\text{Valve Stroke}}\right)\left(\frac{\text{Sec}}{1.751 \text{ Rev Gear Box}}\right) = 0.209 \ \frac{\text{Sec}}{\text{Valve Stroke}}$$

With a motor startup time of 30 msec maximum, the maximum valve operation time is 209 + 30 = 239 msec. This is, of course, a maximum time calculation due to the valve torque requirement decreasing and motor speed increasing.

This valve speed operation was calculated using motor speed values from Figure 8-23, which is the minimum battery supply voltage. As supply voltage increases, speed increases for like torques thereby ensuring a faster valve speed.

The motor was tested in a test setup as shown in Figure 8-27. The motor was connected to a transfer ring and attached to the frame of an air bearing type dynamometer. A rubber coupling was used to join the motor output shaft with the dynamometer generator rotor. The motor loading was achieved by applying a dc power to the dynamometer stator, which then acted as a clutch. The torque developed was determined by monitoring a water manometer which was calibrated for specific loading. Calibration was accomplished by placing known values of weights at a point twelve inches from the dynamometer rotor centerline on the torque bar and recording the water manometer meniscus height. Refer to Figure 8-28. Calibration curves were run prior to each test and are included as Figures 8-29 and 8-30.

8.4.4 <u>Planetary Gear Train Tests</u> — Two planetary gear trains were tested, one SN01 was lubricated with Krytox 240AC grease and SN02 with gears and races coated with microseal 200 (molybdenum disulphide) and lubricated with Krytox 260AC.

8.4.4.1 <u>Reduction Ratio</u> — This test was conducted by rotating input shaft by hand and counting n[illegible]er of turns required for a complete output shaft turn. The ratio obtained was 77 : 1 for both gear train assemblies.

8.4.4.2 <u>Efficiency</u> — This test was conducted with an output shaft reaction torque of 11.618 lb-in. Both SN01 and SN02 required a slow running torque of 4 oz-in. to overcome this weight. Efficiency was then estimated as

$$\mu = \frac{11.618}{0.25 \times 77} = 0.60$$

which is below the 0.73 minimum design goal. The low efficiency was attributed to the Krytox grease.

$$\mu = \frac{\text{output torque}}{\text{input torque x reduction ratio}}$$

Serial No. 02 was disassembled and all grease removed from the gear train. The tests were repeated with a resulting efficiency of μ = 0.80 which is above the design point.

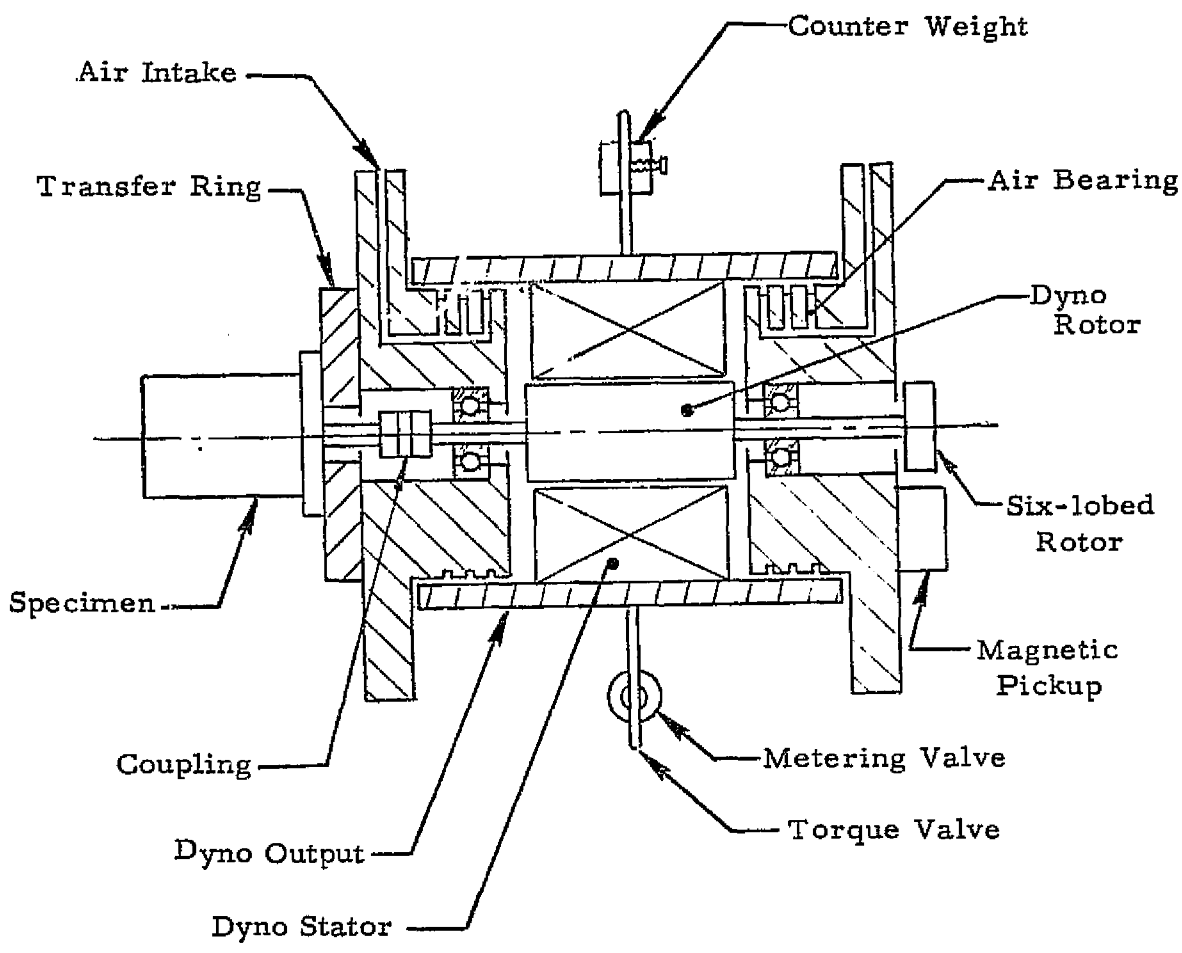

Figure 8-27. Dynamometer Schematic

A sheave, cable and weight were placed at the output shaft as shown in Figure 8-31. The weight was allowed to free-fall and the rate of sescent measured with results as shown in Table VIII-17.

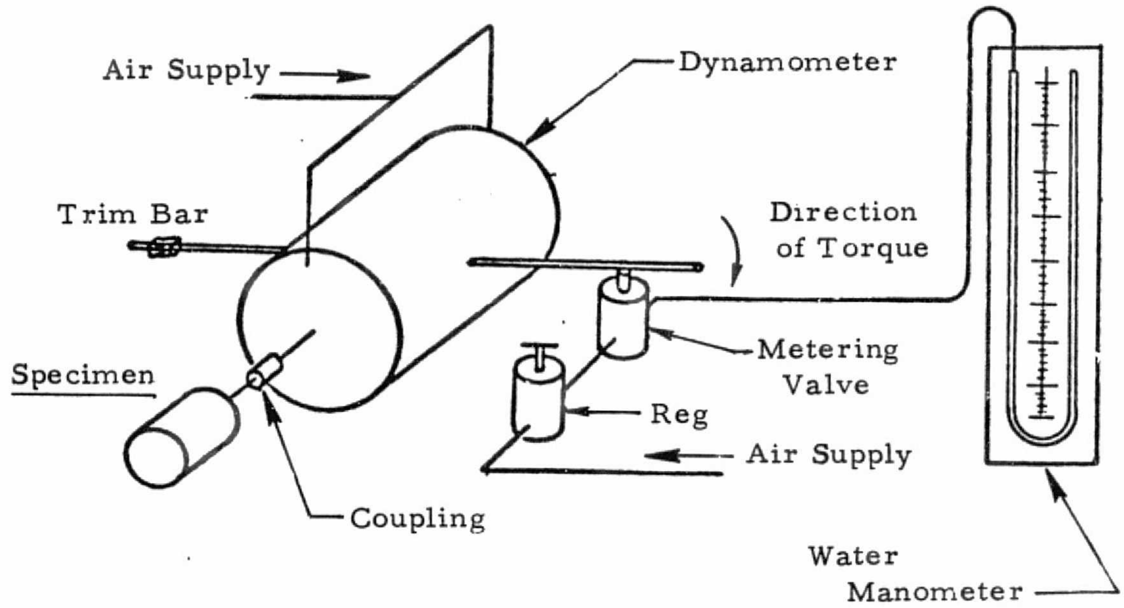

Figure 8-28. Motor Test Setup

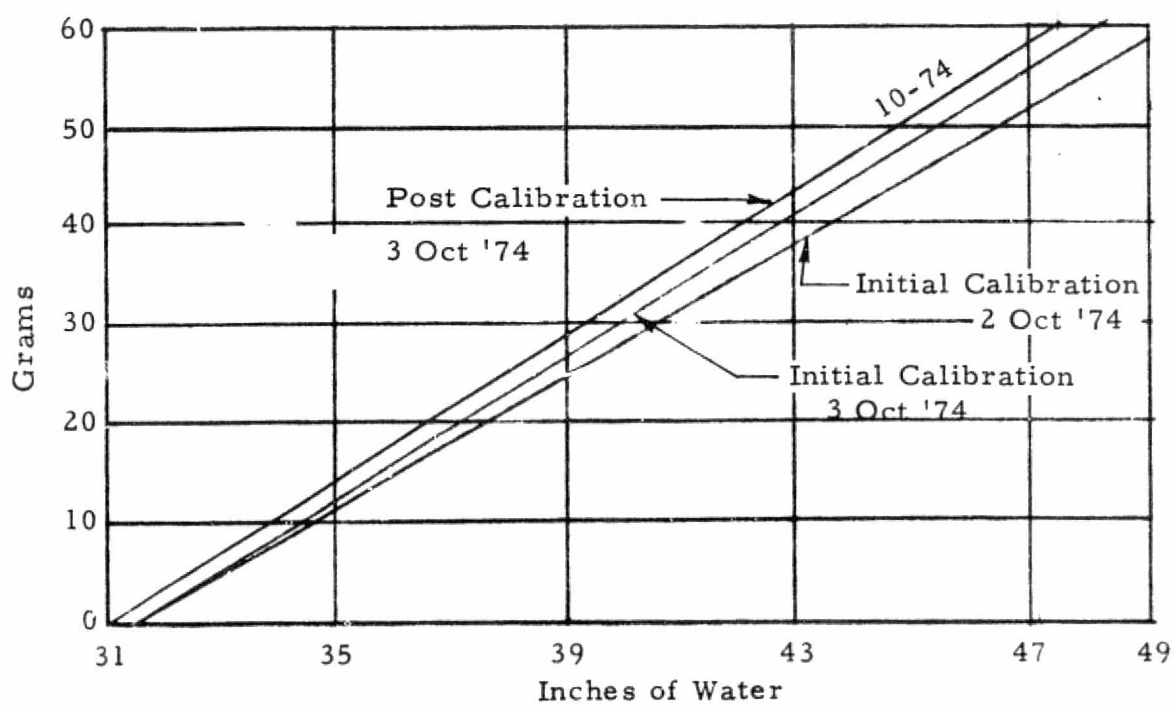

Figure 8-29. Dynamometer Calibration, Curve A

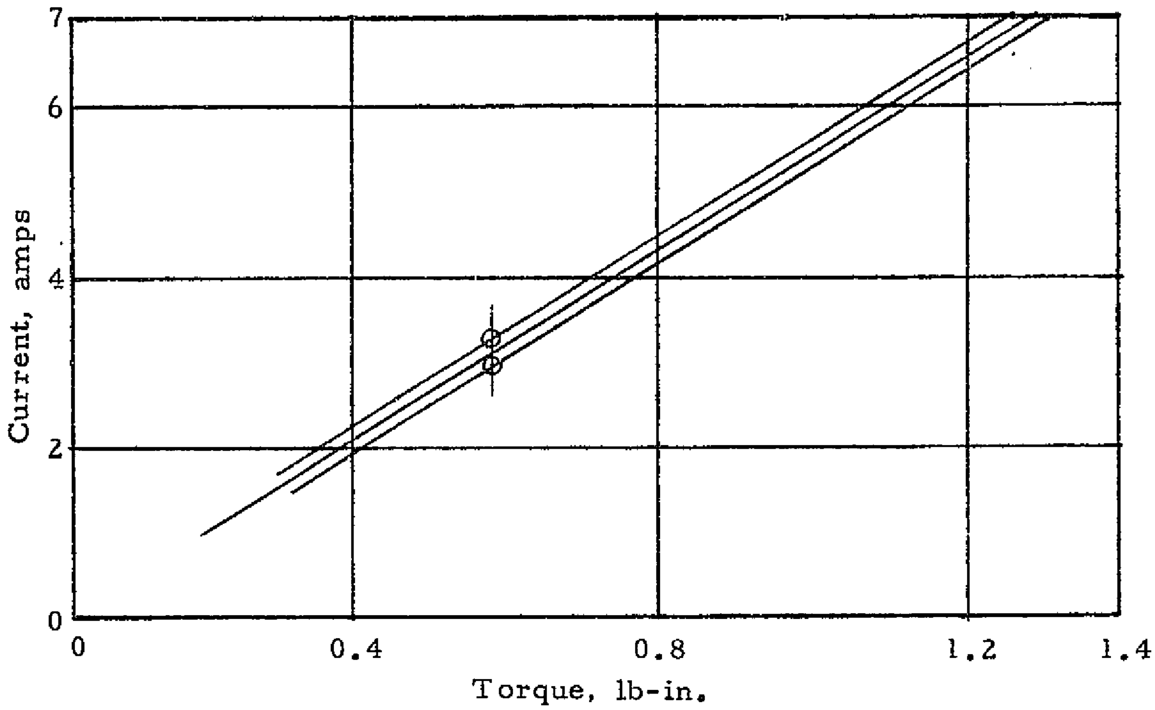

Figure 8-30. Dynamometer Calibration, Curve B

Drop Tests

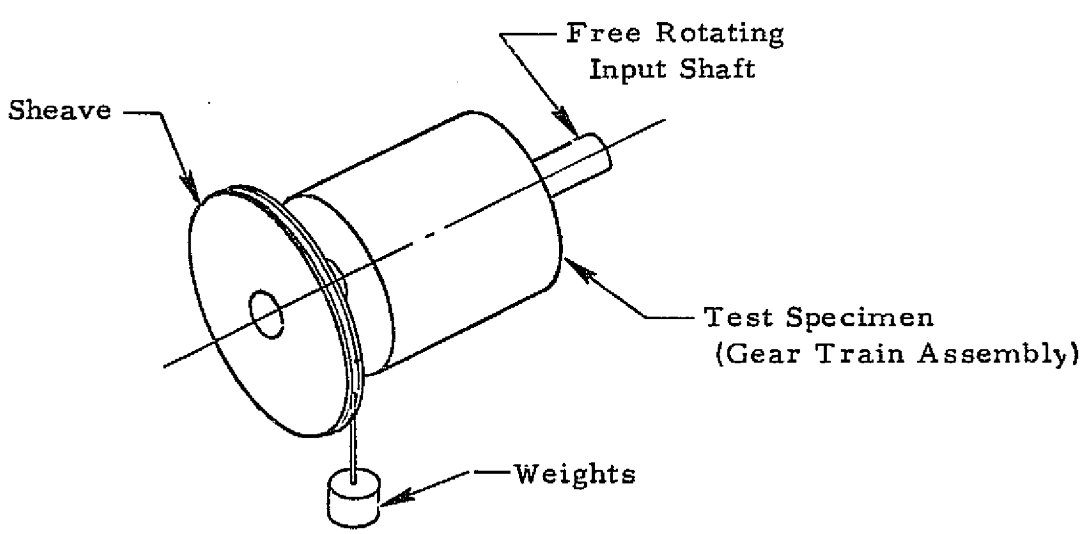

Figure 8-31. Gear Train Test Setup

Table VIII-17. Drop Tests

Specimen	Weight (lb)	Equiv Torque (lb-in.)	Length of Drop (inches)	Time (sec)	Output Shaft Speed, (rpm)	Input Shaft Speed (rpm)
SN 01 with Krytox 240AC	11.70	11.618	12	7.0	16.59	1,277.43
				7.0	16.59	1,277.43
SN 02 with Krytox 260AC	11.70	11.618	12	4.2	27.52	2,119.04
				3.2	36.11	2,780.47
				3.2	36.11	2,780.47
SN 02 without grease	11.70	11.618	12	0.05	231.14	17,797.75
				0.05	231.14	17,797.75

Our first tests verified the reduction ratio is indeed 77 : 1 with no sign of backlash. The second test indicates that gear train efficiency is severely affected by addition of grease; however, with microseal only, gear train surpasses the design point.

The third test (drop tests) shows that the speed of actuation of the gear trains is adversely affected by the addition of Krytox grease, also to be noted is the very favorable average speed attained by the input shaft while back-driven from the output shaft.

8.4.5 RVDT Testing — RVDT testing was accomplished in accordance with the instructions contained in test document DVT5739006, Revision A, dated 11 October 1974. Two specimens were tested, a Pickering Number 23501, and a Schaevite No. R30A. The Pickering provides a more linear output over the required range and appears to be more suitable for our application.

Pickering Test Data	Primary coil resistance	70.2 ohms
	Secondary coil resistances	302.4 ohms, 303.3 ohms

A three (3) Vac 400-Hertz input signal was applied to the primary windings of the RVDT and output voltages recorded for every 2 degrees rotation up to 190 degrees and every 5 degrees rotation up to 360 degrees. Test data is shown in Table VIII-18 and plotted data presented in Figure 8-32. A schematic of the test setup is shown in Figure 8-33.

Table VIII-18. Pickering Test Data

Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)
0	0.005	66	0.159	132	0.112	210	0.075
2	0.006	68	0.163	134	0.107	215	0.087
4	0.010	70	0.168	136	0.102	220	0.100
6	0.014	72	0.172	138	0.098	225	0.110
8	0.018	74	0.176	140	0.092	230	0.122
10	0.024	76	0.179	142	0.088	235	0.135
12	0.028	78	0.183	144	0.082	240	0.147
14	0.034	80	0.186	146	0.078	245	0.157
16	0.040	82	0.188	148	0.073	250	0.167
18	0.044	84	0.190	150	0.068	255	0.177
20	0.050	86	0.191	152	0.063	260	0.186
22	0.053	88	0.191	154	0.058	265	0.192
24	0.058	90	0.192	156	0.053	270	0.194
26	0.063	92	0.191	158	0.048	275	0.190
28	0.067	94	0.188	160	0.044	280	0.186
30	0.073	96	0.187	162	0.040	285	0.175
32	0.078	98	0.184	164	0.035	290	0.165
34	0.083	100	0.182	166	0.030	295	0.154
36	0.088	102	0.178	168	0.024	300	0.142

(Continued)

Table VIII-18. Pickering Test Data (Continued)

Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)
38	0.093	104	0.174	170	0.020	305	0.130
40	0.098	106	0.170	172	0.015	310	0.118
42	0.103	108	0.167	174	0.010	315	0.106
44	0.108	110	0.163	176	0.007	320	0.095
46	0.113	112	0.158	178	0.005	325	0.082
48	0.118	114	0.153	180	0.005	330	0.070
50	0.123	116	0.149	182	0.007	335	0.058
52	0.128	118	0.144	184	0.012	340	0.046
54	0.133	120	0.140	186	0.016	345	0.034
56	0.137	122	0.136	188	0.020	350	0.022
58	0.142	124	0.131	190	0.026	355	0.010
60	0.147	126	0.126	195	0.038	360	0.005
62	0.151	128	0.122	200	0.050		
64	0.155	130	0.117	205	0.062		

Shaevite Test Data { Secondary Coil Resistance 228.5 ohms, 228.2 ohms

A three (3) Vac 400-Hertz input signal was applied to the primary windings of the RVDT and output voltages recorded for every 5 degrees rotation up to 360 degrees. Test data is shown in Table VIII-19 and plotted data presented in Figure 8-34. The test setup is as shown in Figure 8-33.

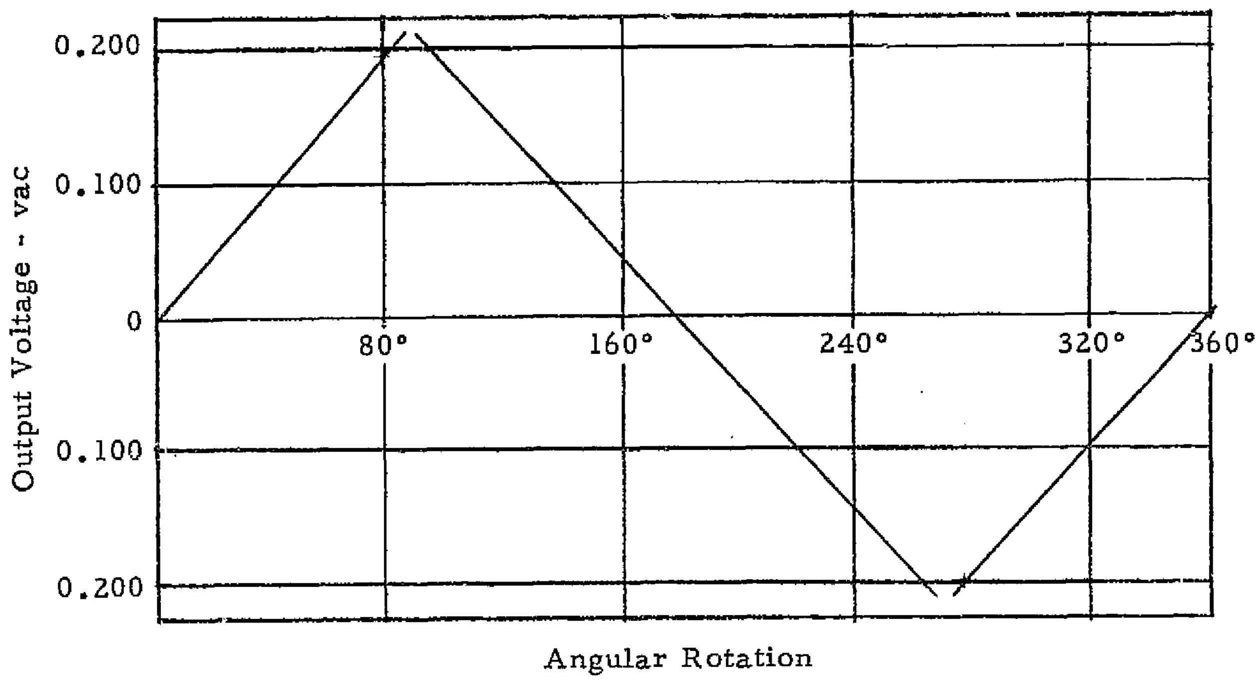

Figure 8-32. Pickering RVDT Voltage versus Angle Plot

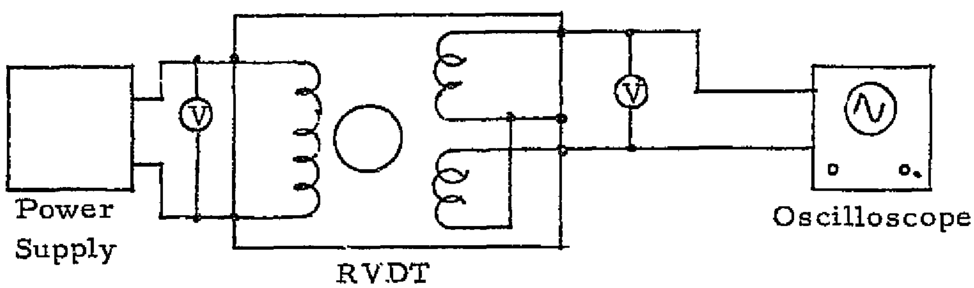

Figure 8-33. Schematic of RVDT Setup

Table VIII-19. Schaevitz RVDT Test Data

Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)
0	0.005	155	0.005	310	0.123
5	0.010	160	0.014	315	0.112
10	0.023	165	0.027	320	0.098
15	0.035	170	0.039	325	0.086
20	0.047	175	0.050	330	0.073
25	0.059	180	0.063	335	0.060
30	0.071	185	0.076	340	0.047
35	0.082	190	0.088	345	0.035
40	0.092	195	0.100	350	0.022
45	0.103	200	0.113	355	0.012
50	0.110	205	0.124	360	0.005
55	0.117	210	0.136		
60	0.123	215	0.147		
65	0.128	220	0.157		
70	0.130	225	0.167		
75	0.135	230	0.175		
80	0.135	235	0.182		
85	0.132	240	0.188		
90	0.127	245	0.192		
95	0.123	250	0.196		
100	0.117	255	0.197		
105	0.110	260	0.197		
110	0.100	265	0.195		
115	0.091	270	0.193		

(Continued)

Table VIII-19. Schaevitz RVDT Test Data (Continued)

Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)	Shaft Angle (Deg)	Output Volts, (Volts)
120	0.080	275	0.188		
125	0.069	280	0.182		
130	0.057	285	0.175		
135	0.045	290	0.166		
140	0.032	295	0.157		
145	0.020	300	0.147		
150	0.007	305	0.133		

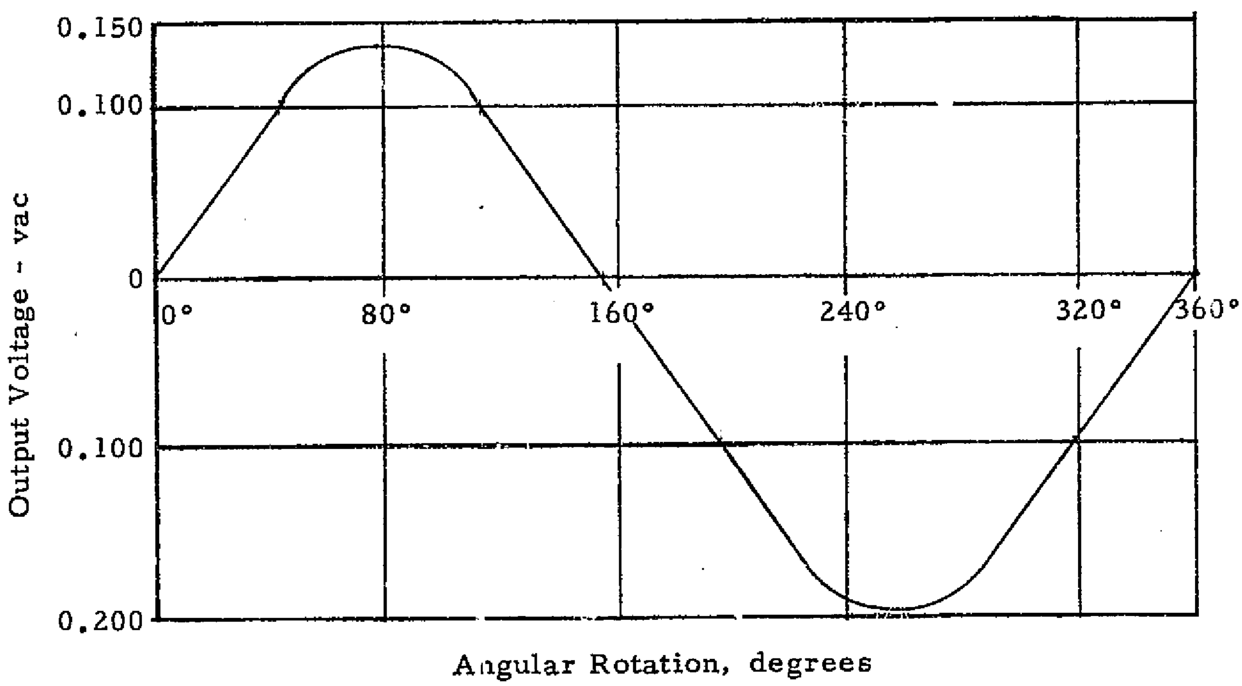

Figure 8-34. Schaevitz RVDT Voltage versus Angle Plot

8.4.6 Pneumatic Actuator Assembly — The pneumatic actuator assembly was tested in accordance with the instructions contained in DVT5739036. Testing consisted of two basic tests; Bellows Stroke versus Pressure, and Actuator Stroke versus Pressure.

The Bellows Stroke versus Pressure test provided bellows force rate, length of stable stroke, and maximum predicted life. See Figure 8-35. The bellows was pressurized with GN_2 with the bellows free-end-to-flange dimension and gas pressure recorded at stroke increments of approximately 0.050 inch.

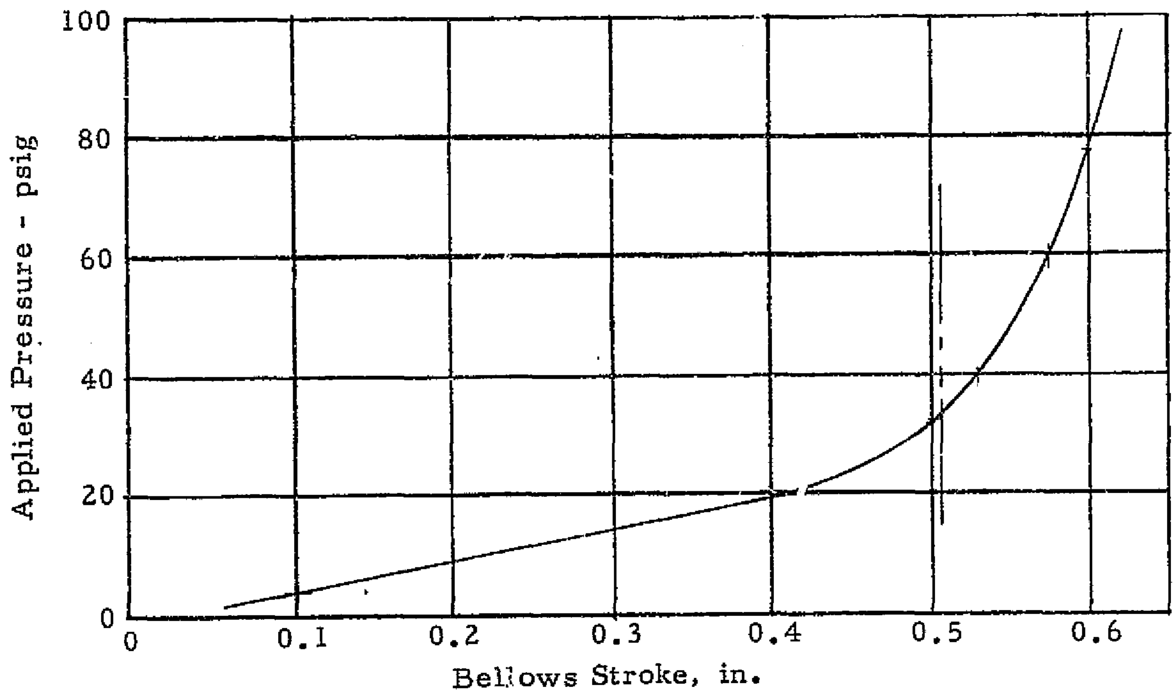

Figure 8-35. Bellows Stroke versus Pressure

The bellows rate was linear between 0 and 0.40 inch, slowly increasing between 0.40 and 0.50 inch, and increasing rapidly between 0.50 and 0.60 inch. When the stroke exceeded 0.60 inch, the bellows was no longer stable and experienced a preset of 0.014 inch. Repeated actuation between 0 and 0.50 inch resulted in a bellows preset of 0.005 inch after which the bellows became repeatable. The stroke requirement for the system is much less than the 0.50-inch test value.

The Actuator Stroke versus Pressure test was performed to determine the proper length of the bellows stop and the required shimming for the unit. The design requirements were to provide a stroke of 0.50 inch at 250 psig with the final 0.050 inch of stroke requiring only 60 percent pressure loading. All detail components for the Actuator Assembly were measured, the spring cage length and internal stroke calculated, and the thickness of shims required. The bellows stop was longer than desired but the unit was still used to select a belleville spring with the most desirable ramp characteristic. See Figures 8-36 and 8-37.

The bellows stop was reworked for the required stroke and retest data is presented in Figure 8-38.

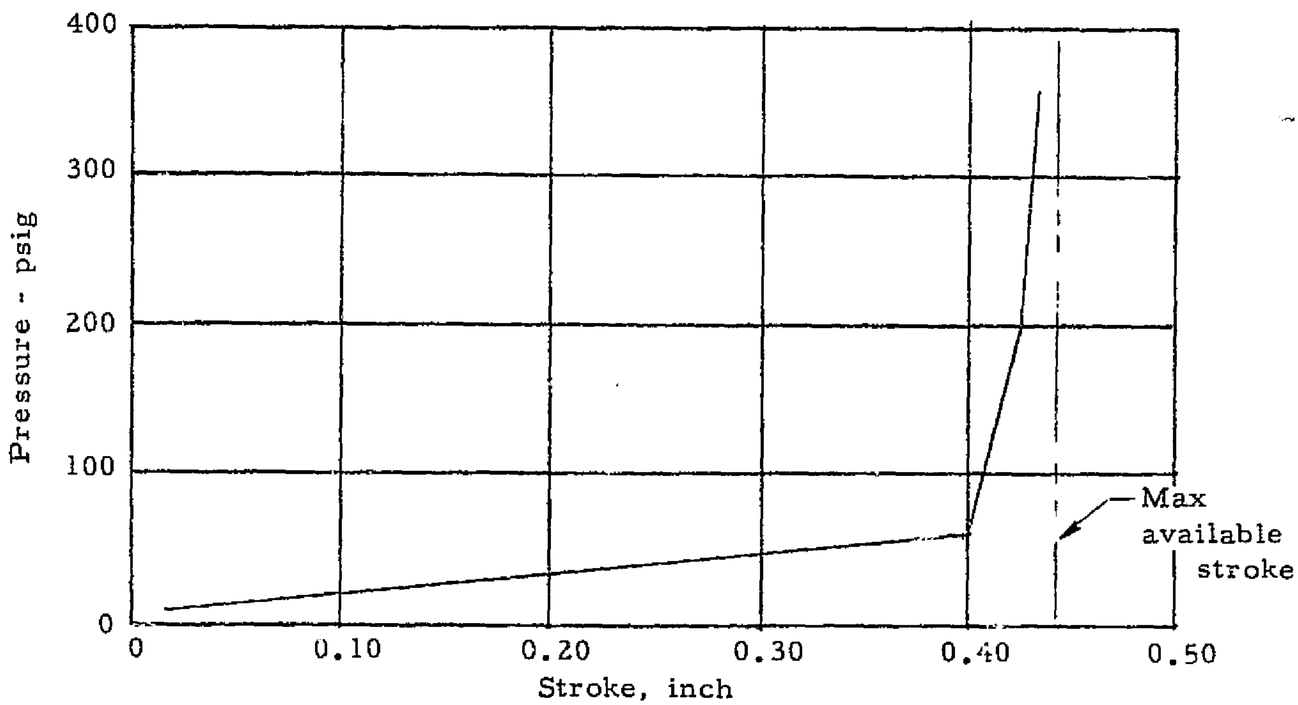

Figure 8-36. Actuator Assembly Stroke vs Pressure, 0.019 Shims

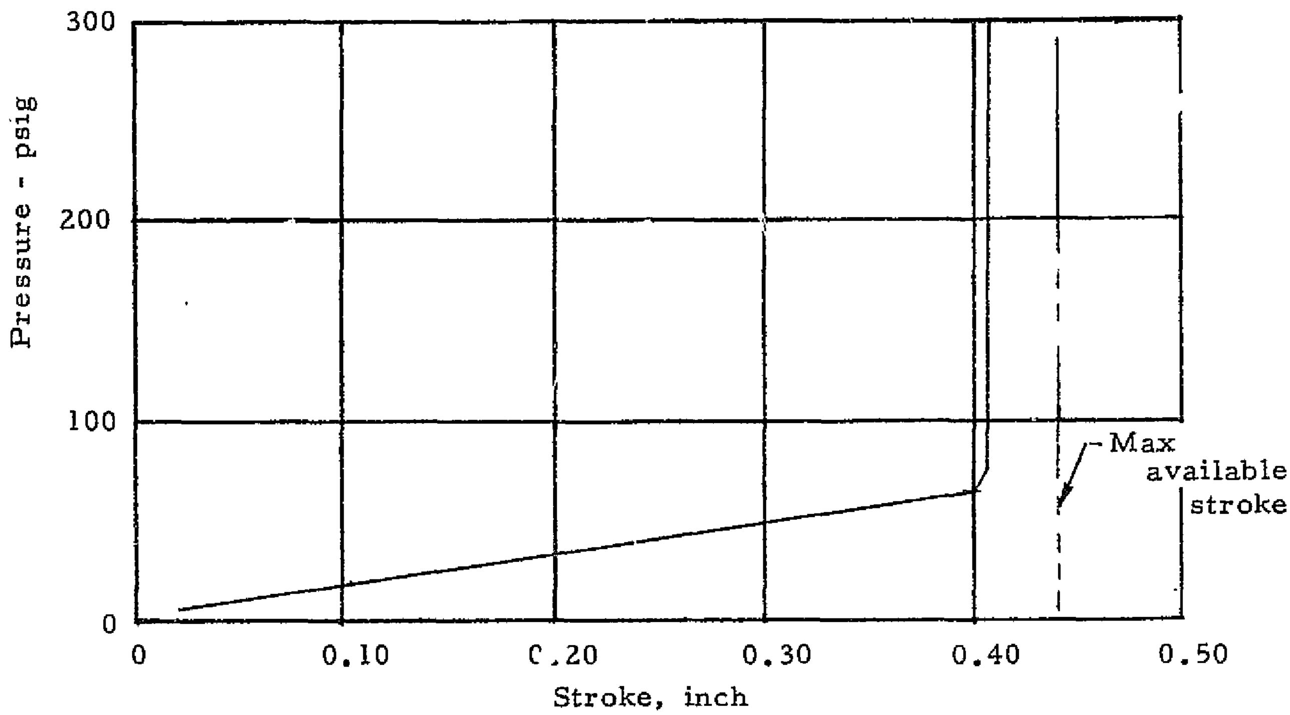

Figure 8-37. Actuator Assembly Stroke vs Pressure, 0.008 Shims

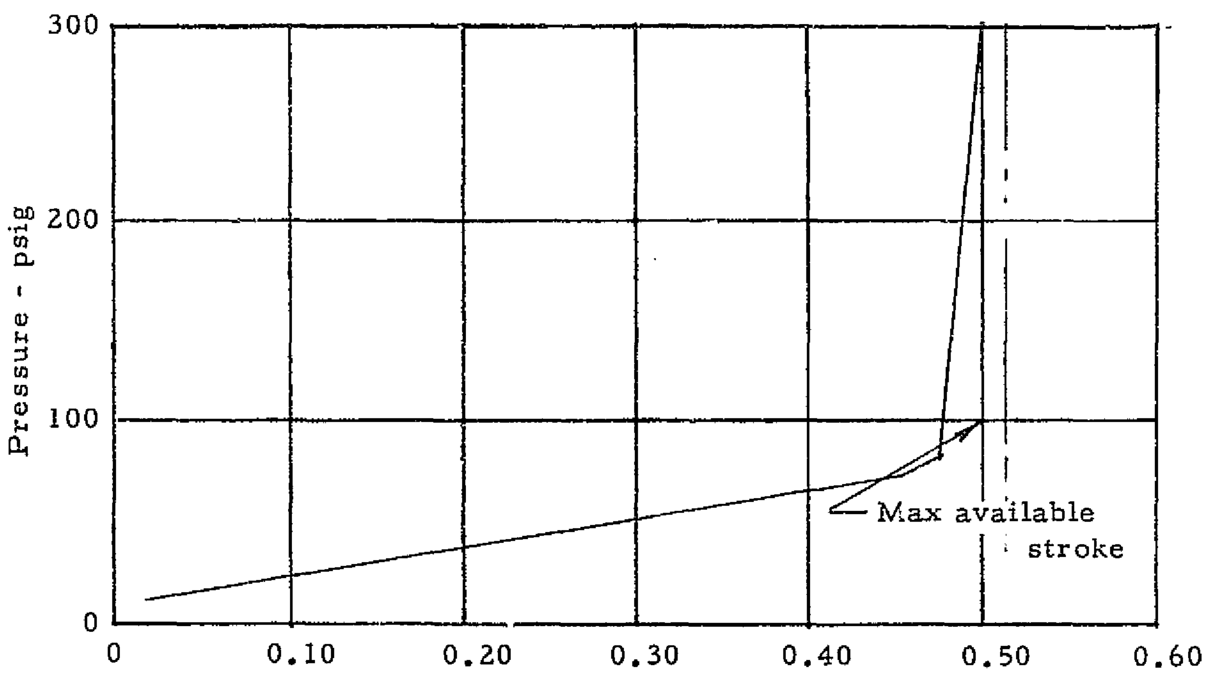

Figure 8-38. Actuator Assembly Stroke versus Pressure

9.0 RECOMMENDATIONS AND CONCLUSIONS

The intrinsic feasibility of the new a.c. motor-operated lifting ball shutoff valve concept was demonstrated in the Parker-Hannifin program. Most significantly, the prototype valve easily demonstrated the required 10,000-cycle life while maintaining tight closure characteristics. This new design approach shows definite promise to be the ultimate solution for the OMS engine valve, as well as for other long-life, low leakage, low pressure drop valve applications.

The Parker-Hannifin program focused on basic concept selection and long-life feasibility testing of detail prototype units of all major detail assemblies. The lifting ball valve was manufactured and tested to demonstrate the major operating parameters such as cycle life testing, torque tests, pressure drop testing, temperature tests, and proof pressure tests. The electronic control breadboard was manufactured and all control functions demonstrated. A.C. motor tests were conducted on two prototype motors to demonstrate locked rotor torque, motor speed versus control frequency input signal, and motor speed versus torque. Planetary gear train and RVDT tests were also accomplished to ensure compatible operational modes. The program has been basically involved in the testing of components and additional work remains to qualify the design approach for practical application in a specific flight configuration. For example, we recommend that the following tasks be completed on the a.c. motor-operated lifting ball valve for the OMS engine application:

On the Existing Prototype

1. Response Time Tests
2. Long Term Propellant Compatibility Tests
3. Propellant Decontamination Tests
4. Vibration Test
5. Fail-safe Closure Test

Analysis and Design

1. Refine and optimize linkage lengths and bearing sizes for minimum unit weight and size.

2. Complete the flight weight design for installation in the Orbiter OMS engine.

3. Refine the modular assembly for ease of maintenance.

Based on the results of the original system concept tradeoff study, Parker-Hannifin recommends the design, manufacture, and testing of the "moving seat valve" concept. This concept was subordinated to the lifting ball valve in the technology program due to the advanced state-of-the-art os the design, thereby presenting Parker-Hannifin with a questionable user acceptability in the limited time frame available. The design, as presented at the outset of this program most certainly provides sufficient credibility to further justify verification of the concept.