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ACTUATION AND SYSTEM DESIGN AND EVALUATION

OMS ENGINE SHUTOFF VALVE

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PREPARED FOR

**NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
LYNDON B. JOHNSON SPACE CENTER
HOUSTON, TEXAS 77058**

PARKER HANNIFIN

AIR AND FUEL DIVISION

18321 JAMBOREE BLVD., P.O. BOX 2050, IRVINE, CALIF. 92664

APPENDIX A

TECHNICAL REQUIREMENTS

A.1 TECHNICAL CHANGES DURING THE PROGRAM

A.2 The number of valve and actuation system concepts to be selected for experimental evaluation and design verification was changed from one to two.

3.0 TECHNICAL REQUIREMENTS

3.1 GENERAL

3.1.1 Study Requirements

The contractor shall conduct trade studies and conceptual design efforts to identify advanced valve and actuation system design approaches suitable to meet the long life, maintainability and economic development requirements for the OMS engine bipropellant shutoff valve. One valve and actuation system concept will be selected for experimental evaluation and design verification. The selected valve and actuation system concept shall be suitable for packaging and use in a mechanically linked quadredundant shutoff valve configuration and also in a valve configuration, which incorporates only series redundancy. A detailed design with appropriate analysis and drawings will be established for a flight weight shutoff valve. A prototype test valve design shall also be established which contains all the essential design features to experimentally verify fabrication and operation of the flight weight shutoff valve configuration. The prototype test valve need not be to the same level of redundancy as the flight weight valve design, if analysis indicates that all essential valve and actuation system features can be experimentally verified with a simpler test configuration.

3.1.2 Design Requirements

The contractor will define in detail the concepts and theories emanating from the study effort. Environmental conditions under which the valves and actuation systems will satisfactorily operate and the performance and detailed characteristics of the equipment will be clearly specified.

3.1.3 Development Requirements

The contractor will specify those special factors that must be considered in translating design data into tangible end items. The contractor should identify any problems which become evident and might potentially affect manufacturing processes and techniques. The solutions to these problems should identify what must be developed in order to facilitate manufacturing of the end product. The contractor will conduct testing and prepare test documentation to verify that the performance design requirements of the valve(s) and actuation system(s) meet the requirements of this SOW.

3.1.4 Technical Guidelines

The following guidelines, with a few noted exceptions, are not to be considered firm requirements. They are intended as optimum design objectives and are subject to change in accordance with technology limitations and reliability considerations. One of the primary objectives of this contractual effort is to define the realistic and obtainable requirements that should be imposed on a valve and actuation system for the space shuttle OME and thus hopefully avoid development problems that may result from initially unrealistic performance requirements.

3.1.4.1 Application - The valve and actuation system technology and design recommendations developed as a result of this contractual effort will be utilized in defining the recommended design, operational capabilities, and requirements for the OME propellant valve and actuation system.

3.1.4.2 Fluid Media Compatibility - The valve for this program must be compatible for exposure to the following propellant vapors, liquids, and combinations of oxidizer and fuel vapors. The propellants will be nitrogen tetroxide (N_2O_4), 50/50 blend of hydrazine and unsymmetrical dimethylhydrazine (50% N_2H_4 - 50% UDMH), and monomethylhydrazine (MMH). The contractor will have conclusive compatibility data on each material recommended for usage. In evaluating propellant compatibility, the contractor will also evaluate propellant moisture combinations since once a valve is exposed to propellants it is unreasonable to assume that the unit will remain free of moisture for the remaining service life.

The contractor will not consider propellant deconamination of components to extend the service life, since cleaning of hardware between missions is improbable and will result only when required to insure personnel safety during system repairs. The valve must also be compatible with anticipated cleaning and flushing fluids.

3.1.4.3 Lubricants - Due to propellant compatibility, low temperature operation, and extended service life, total exclusion of lubricants is a desirable design goal.

3.1.4.4 Maintainability - The valve must be designed to be easily maintained. Replaced detail part of the valve must not affect the operational characteristics of the valve.

3.1.4.6 Cycle Life - A design goal of 4000 wet cycles and 6000 dry cycles will be used for this program.

3.1.4.7 Internal Leakage - A leak rate of 10 standard cubic centimeters per hour (scch) of helium will be used as a goal.

3.1.4.8 Pressure Drop - A maximum pressure drop of 5 psid from the valve inlet to the valve outlet which will include all filters and redundant valves. The valve design shall provide for a balanced pressure drop in the event of a failure in one of the parallel flow paths to minimize the resulting engine mixture ratio shift.

3.1.4.9 Response - The valve opening and closing times shall have absolute actuation times in the range of 100 to 1000 milliseconds. The actual times will be established through trade studies considering valve actuation approaches and the effects of valve actuation times on the engine start and shutdown transient.

3.1.4.10 Response Repeatibility - Response repeatability should be considered an important factor in the design of the valve and actuation system.

3.1.4.11 Filters - Filters used in the valve and actuation system should be consistent with the contamination tolerance of the valve and actuation system.

3.1.4.12 Fabrication Limitations - In the process of designing a prototype valve to satisfy the requirements of this SOW, the contractor should maintain an awareness of the design requirements that will be imposed on a "flight-type" design to insure that the prototype will be adaptable.

3.1.4.13 Weight and Envelope - Minimum weight and envelope are important design considerations not to be overlooked by the contractor.

3.1.4.14 Contamination - Contamination tolerance will be a major design objective for this program. Limitation of self-generated contamination shall also be a primary design goal.

3.1.4.15 Decontamination - Dead-ended passages, crevices, and other possible areas in which contaminations could collect and hinder a decontamination process should be avoided.

3.1.4.16 Other Requirements - The design should be consistent with the Orbit Maneuvering System Pod procurement specification, as defined in specification MC621-0002 "Orbital Maneuvering Subsystem - Technical Requirements" incorporated herein by reference.

APPENDIX B

MOVING SEAT PRELIMINARY DESIGN ANALYSIS

1. Seat Poppet ΔP Analysis
2. Propellant Valve Seat Sizing
3. Pressure Surge Versus Closing Time
4. Bellows Resonant Frequency

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Seat/Poppet ΔP Analysis - Quad Configuration

Conditions:
1191 lb/sec N_2O_4 total flow
7.22 lb/sec MMH total flow
4 psi allocated to valve seat/poppets
($\therefore$ 1 psi allowed for filter and other package losses)

Procedure:

a. Calculate velocity pressure versus line sizes (1.0 to 2.0 inches)

b. Calculate Reynolds number versus line size

c. Set up a K-factor budget based on preliminary design sketch

d. Calculate poppet stroke needed, as a function of ΔP.

Velocity Pressure:

$$q = \frac{1}{\rho}\left(\frac{w}{0.669A}\right)^2 = \frac{1}{\rho}\left(\frac{\dot{w}}{0.669 \times 0.7854\, d^2}\right)^2$$

for N_2O_4, $\rho = 90.2$ lb/ft^3 at 70°F, $\dot{w} = \frac{11.91}{2}$

$= 5.955$ lb/sec (for each parallel leg)

$$q = \frac{1}{90.2}\left(\frac{5.955}{0.669 \times 0.7854\, d^2}\right)^2 = \frac{1.4240}{d^4}$$

(continued)

for MMH, $\rho = 54.8\ \text{lb/ft}^3$ at 70°F, $\dot{w} = \dfrac{7.22\ \text{lb/sec}}{2} = 3.61\ \text{lb/sec}$
(for each parallel leg)

$$q = \frac{1}{54.8}\left(\frac{3.61}{0.669 \times 7854d^2}\right)^2 = \frac{0.86139}{d^4}$$

Reynolds Number:

$$Re = \frac{15.28\dot{w}}{\mu d}$$

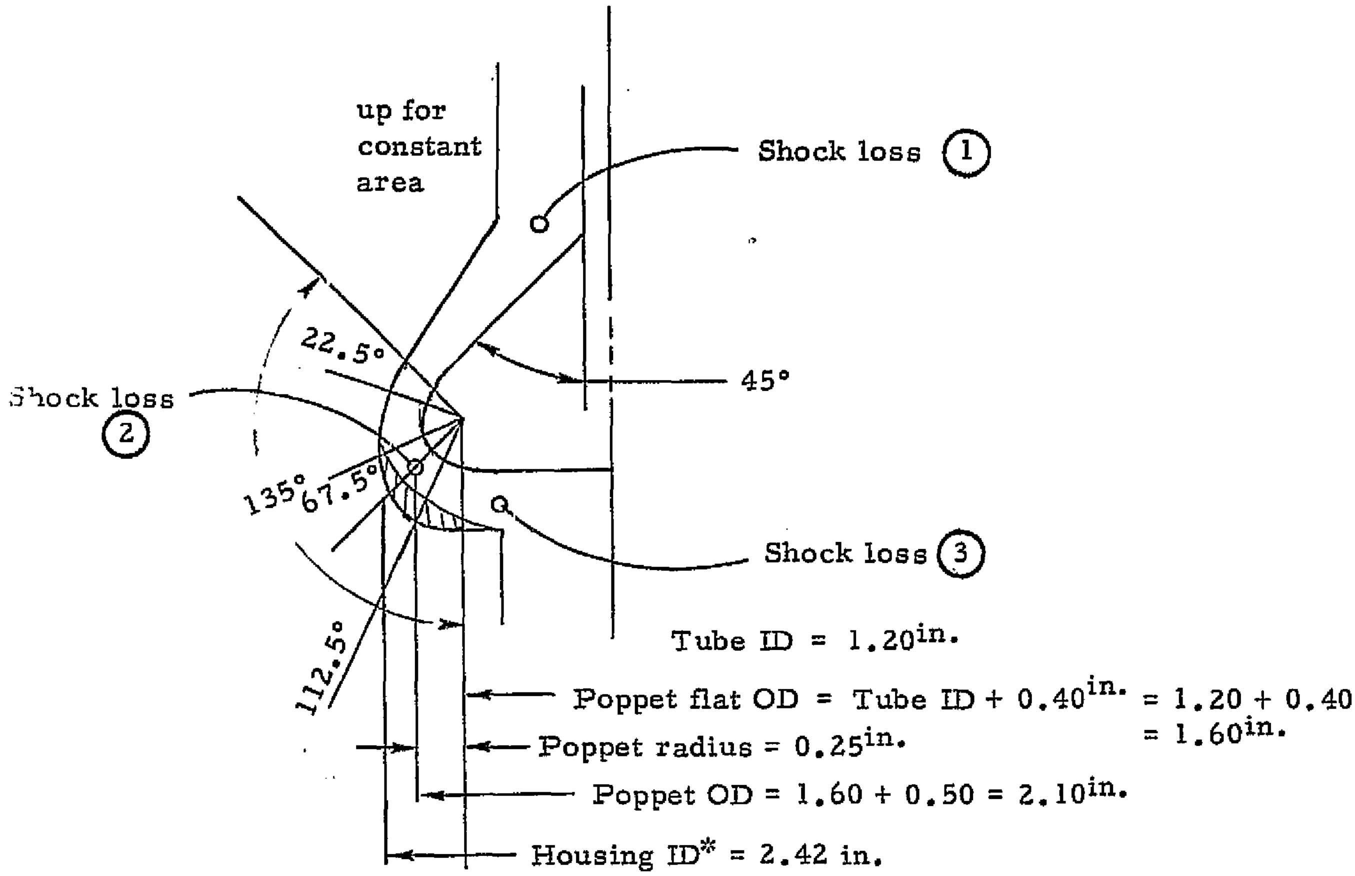
for N_2O_4, = 0.413 centipoise at 70°F
$= (6.7249 \times 10^{-4})(0.413) = 2.7774 \times 10^{-4}$ lbm/sec-ft

$$Re = \frac{(15.28)(5.955)}{2.7774 \times 10^{-4}\ (d)} = \frac{3.27617 \times 10^5}{d}$$

for MMH, μ = 0.85 centipoise at 68°F = 6.7249×10^{-4} (0.85)

$$Re = \frac{(15.28)(3.61)}{(5.7162 \times 10^{-4})(d)} = \frac{9.6499 \times 10^4}{d} = 5.7162 \times 10^{-4}\ \text{lbm/sec-ft.}$$

Line Size (inches)	N_2O_4 q (psi)	N_2O_4 Re	MMH q (psi)	MMH Re
1.0	1.4240	3.27617×10^5	0.86139	9.6499×10^4
1.1	0.9726	2.9783	0.5883	8.7726
1.2	0.6867	2.7301	0.4154	8.0415
1.3	0.4986	2.5201	0.3016	7.4230
1.4	0.3707	2.3401	0.2242	6.8928
1.5	0.2813	2.1841	0.1702	6.4332
1.6	0.2173	2.0476	0.1314	6.0311
1.7	0.1705	1.9272	0.1031	5.6764
1.8	0.1357	1.8201	0.0820	5.3610
1.9	0.1093	1.7243	0.0661	5.0789
2.0	0.0890	1.6381	0.0538	4.8249



Annular area, from poppet OD to housing ID, = line area,

then,

$$(0.7854)(1.20)^2 = 0.7854\left[\text{housing ID}^2 - 2.10^2\right]$$

$$\text{housing ID}^* = \sqrt{1.20^2 + 2.10^2} = 2.419 \text{ inches}$$

It appears that this K-factor (based on valve inlet flow area) can be drastically reduced with little penalty on the design, e.g., doubling the annular area would reduce the K-factor to less than 0.1 ($\approx \frac{0.3}{4} = 0.075$)

Shock Loss (3) K-Factor Calculations

Line Size	Total K-factor (for 2 psi ΔP)		K-factor Allowed for Shock (3)	
	N_2O_4	MMH	N_2O_4	MMH
1.0	1.404	2.320	0.954	
1.1	2.056	3.400	1.606	
1.2	2.912	4.815	2.462	
1.3	4.011	6.631	3.561	
1.4	5.395	8.921	4.945	
1.5	7.110	11.751	6.660	
1.6	9.204	15.220	8.754	
1.7	11.730	19.399	11.280	
1.8	14.738	24.390		
1.9	18.298	30.257		
2.0	22.472	37.175		

Shock Loss (1)

Treat this as a 45-degree miter joint flow change in a circular duct, K = 0.35.

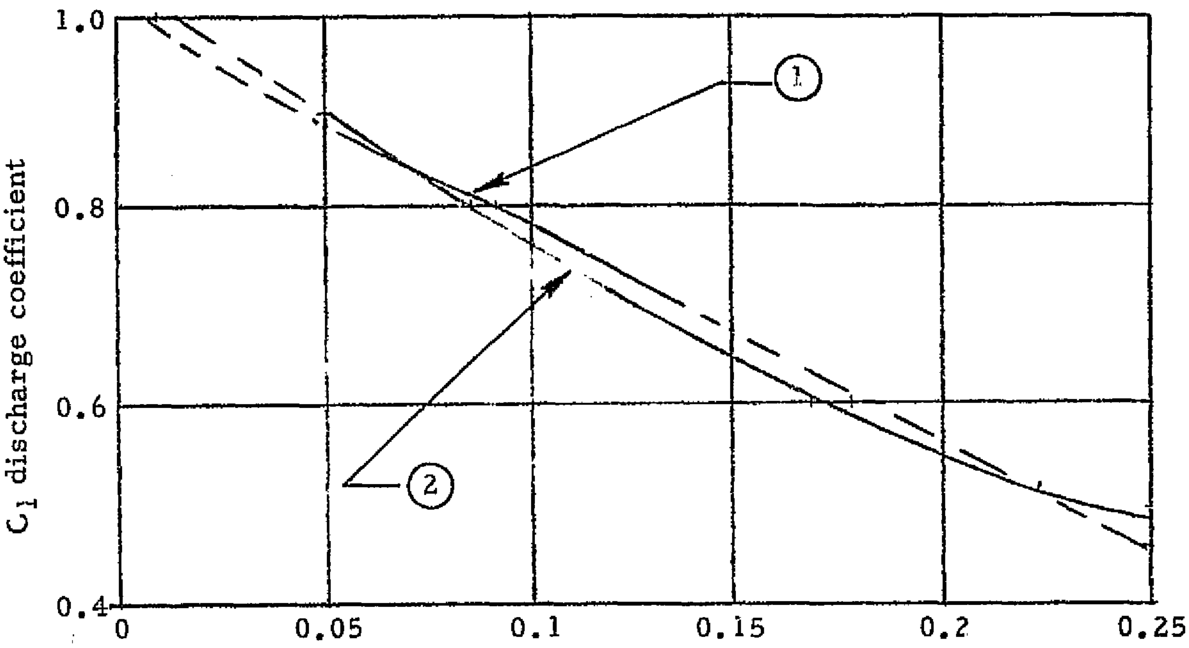
Shock Loss (2)

Treat this as a 135-degree flow change in a rectangular duct. Look at three different points; 22.5°, 67.5°, and 112.5°.

	225°	675°	112.5°
a = duct width = mean circumference =	$(\pi)(2.22)$	$(\pi)(2.26)$	$(\pi)(1.92)$
	6.97	7.10	6.03
b = duct thickness = radial thickness of path =	0.16	0.21	0.35
r = mean duct radius = mean flow radius =	0.33	0.36	0.42
a/b = aspect ratio =	43.5	33.8	17.2
r/b =	2.1	1.7	1.2
K*, for 90° bend	≈0.1	≈0.15	≈0.2

*The very high aspect ratios greatly exceed the maximum a/b ratio of 10 on the SAE graph (Figure 1A-19). These K-factors are therefore approximate.

Comparing some discharge coefficient data for flat poppet valves -

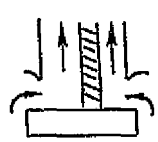


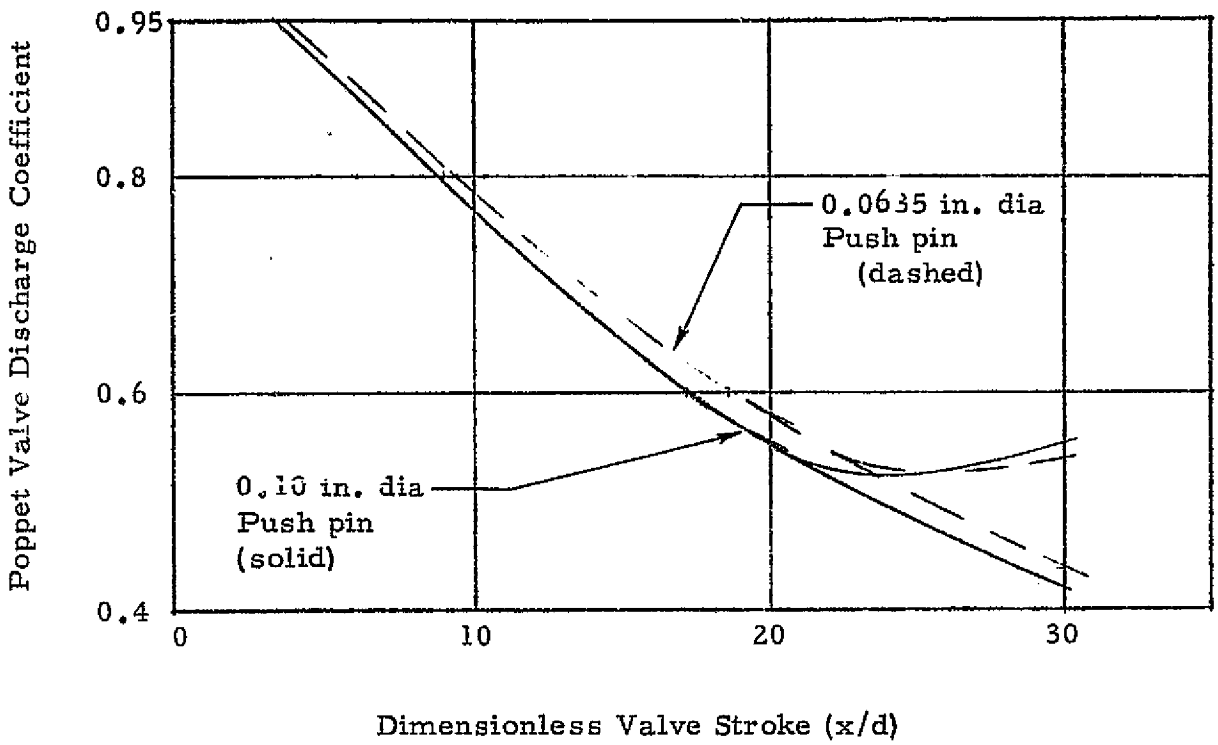
x/d

Valve Stroke/Seat Diameter

1. From Tsai, "Dynamic Behavior of Simple Pneumatic Pressure Reducer." Extrapolated for Pressure Ratio = 1.0.
   (Average for both flow directions) K = 1.6.

2. From Kenyon, 132 block model test data, pressure ratio = 0.862

(Flowing like )



Poppet Valve Discharge Coefficient versus Dimensionless Valve Stroke

Notes:

1. Inlet Pressure 305 psia
   Outlet Pressure 263 psia
   Pressure Ratio 0.862

2. Seat diameter 0.325 inches
   Push Pin Diameter noted

3. 132 Block Model Regulator
   Main Valve, Flat Poppet

Developing matrix to plot K for seat, based on seat area, versus X/d

$$K = \frac{1}{C_d^2}$$

x/d	C_d	K
0.05	0.9	1.235
0.10	0.77	1.686
0.15	0.64	2.441
0.20	0.54	3.429
0.25	0.47	4.527

Determine allowable K for seat, based on seat area, versus x/d

$$K \ \alpha \ a^2 \text{ (for constant } \Delta P\text{)}$$

$$K_{\text{based on seat area}} = K_{\text{based on inlet area}} \left(\frac{\text{a of seat}}{\text{a of inlet}}\right)^2$$

also,

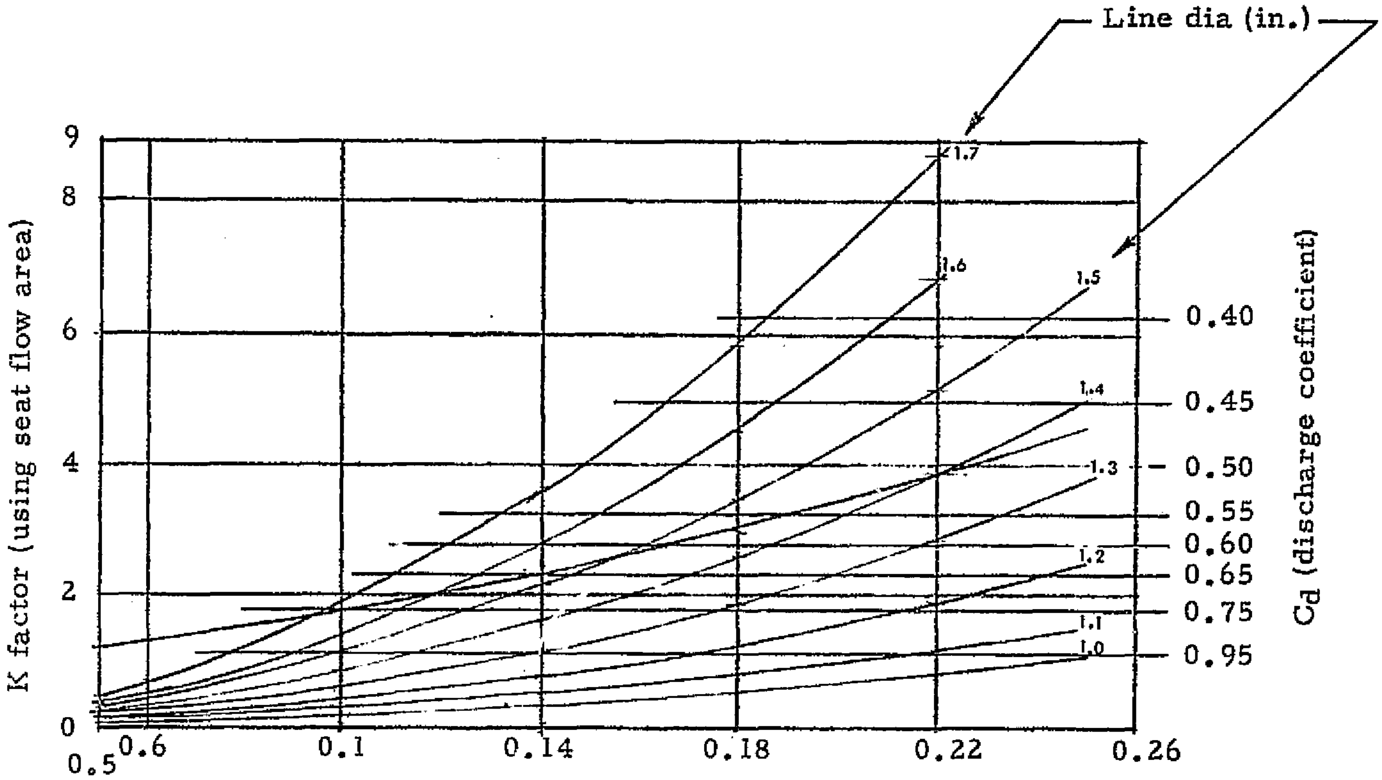
$$\frac{a_{\text{of seat}}}{a_{\text{of inlet}}} = 4\left(\frac{x}{d}\right)$$

then,

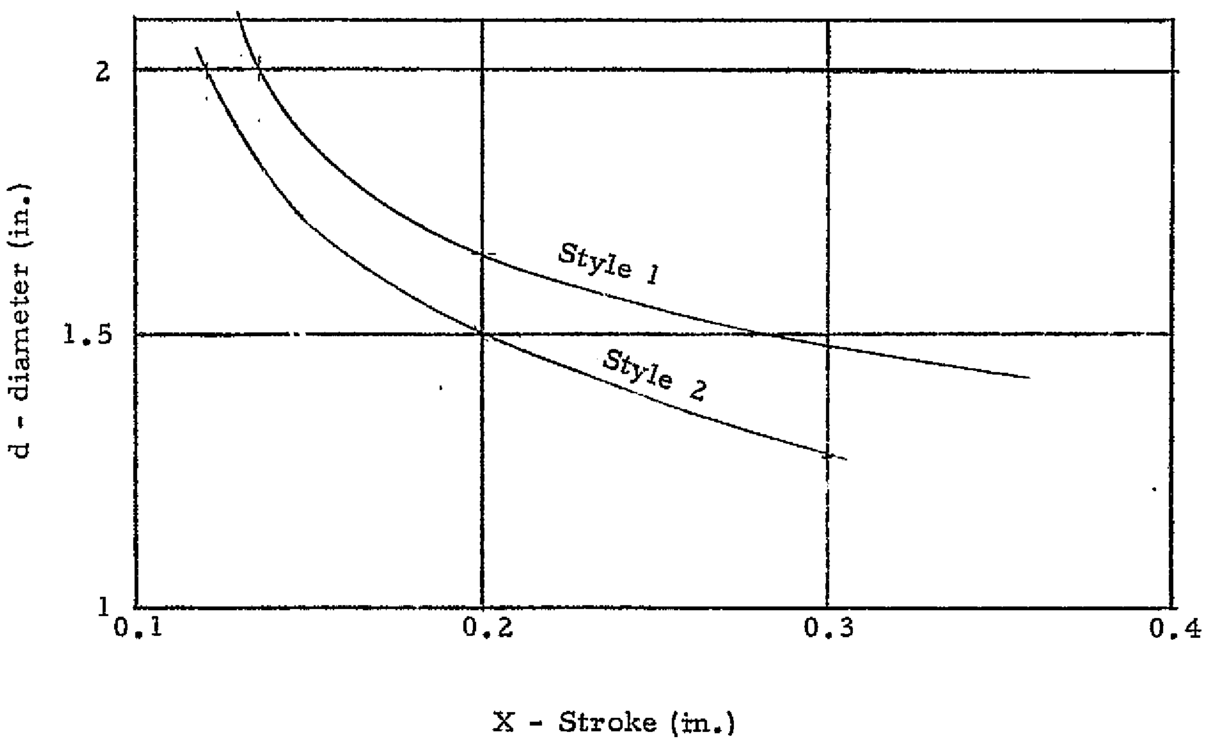
$$K_{\text{based on seat area}} = \left(K_{\text{based on inlet}}\right) (16) \left(\frac{x}{d}\right)^2$$

Inlet Diameter, (inches)

	1.0	1.1	1.2	1.3	1.4	1.5	1.6	1.7
$K_{\text{based on inlet}}$ →	0.954	1.606	2.462	3.561	4.945	6.660	8.754	11.28
x/d ↓	$K_{\text{based on seat area}}$							
0.05	0.0382	0.0642	0.0985	0.142	0.198	0.266	0.350	0.451
0.10	0.153	0.257	0.394	0.570	0.791	1.066	1.401	1.805
0.15	0.343	0.578	0.886	1.282	1.78	2.398	3.151	4.061
0.20	0.611	1.028	1.576	2.279	3.165	4.262	5.602	7.219
0.25	0.954	1.606	2.462	3.561	4.945	6.660	8.754	11.28



x/d (valve stroke/seat diameter)



Diameter versus Stroke Tradeoff Summary Graph

Propellant Valve Seat Sizing

Valve Assembly Requirements (complete quad)

11.91 lb/sec N_2O_4 = 90.2 lb/ft^3 at 70°F

7.22 lb/sec MMH = 54.2 lb/ft^3 at 70°F

5 psid (of which 1 psi is allocated to the filter)

Individual Seat Requirements

$$\Delta P = \frac{5 - 1}{2} = 2 \text{ psid per seat}$$

$$w = \frac{11.91}{2} = 5.955 \text{ lb/sec } N_2O_4 \text{ per side}$$

since $Ca \sim \frac{\dot{w}}{\sqrt{\delta}}$

and $\frac{w}{\sqrt{\delta}} = \frac{11.91}{\sqrt{90.2}} = 1.254$ for N_2O_4

$= \frac{7.22}{\sqrt{54.8}} - 0.975$ for MMH

So that the N_2O_4 requirement governs.

In terms of K factor:

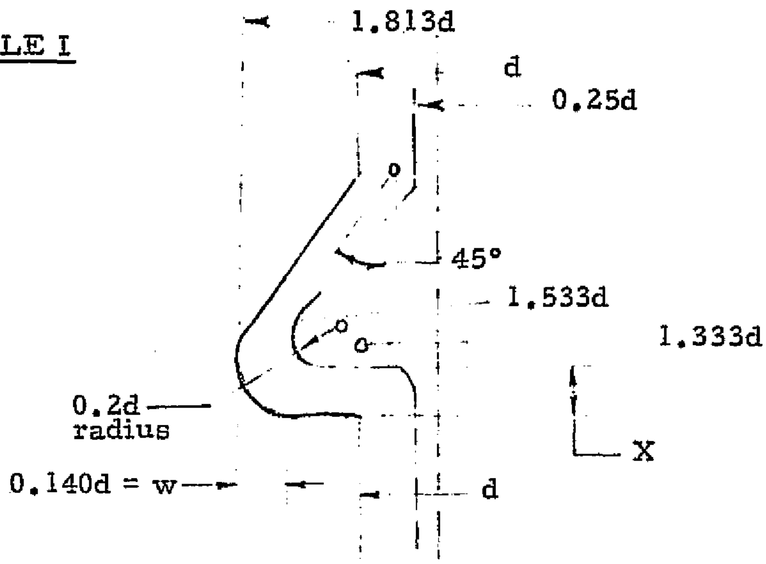
$$\Delta P = Kq = \frac{K}{\delta}\left(\frac{\dot{w}}{0.699a}\right)^2$$

$$2.0 = \frac{K}{90.2}\left(\frac{5.955}{0.669a}\right)^2$$

$$180.4 = K\left(\frac{8.90}{a}\right)^2$$

$$a = 8.90\sqrt{\frac{K}{180.4}} = 0.633\sqrt{K}$$

STYLE I



Find w for constant flow area:

$$(1.533d + 2w)^2 - (1.533d)^2 = d^2 - (0.25d)^2$$

try w = 0.2d

1.3864d ≠ 0.9375d

try w = 0.1d

653d ≠ 0.9375d

w = 0.15d

1.009d ≠ 0.937

w = 0.140d

0.937

Therefore w = 0.140d

Bellows Convolutions

$$K = N\ 1 - \left[\left(\frac{D_1}{D_1 + 0.438S}\right)^2\right]^2$$

corrugation pitch

Typically

$N = 11$ per bellows (22 total)

$D_1 = 1.2$

$S = 0.1875$ (pitch)

Therefore

$$K = 22\left\{1 - \left[\frac{1.2}{1.2 + 0.438(0.1875)}\right]^2\right\}^2 = 0.338 \text{ typical}$$

This will vary with final details of the design.

45-degree Annular Mitre Bend

No data for this shape - consider similar flow paths

Mitre Bends

Source	Round	Square
BHRA	0.24	
SAE	0.35	0.50

Use K = 0.43 based on approach area

135-degree Annular Bend

$$\frac{r}{b} = \frac{1/2(0.2d + 0.140d)}{0.140d} = 1.21$$

$$\frac{a}{b} = \text{Aspect Ratio} = \frac{\pi(1.533d + 0.2d)}{0.140d} = 38.89$$

Use $\frac{r}{b} = 1.25$, $\frac{a}{b} = 40$, to avoid useless complications.

From SAE book, for rectangular ducts K = 1.35 for the above conditions for 90-degree bend, this is the only reasonable similar case for which data is available (Ref Figure 1).

From SAE book (Page A20) 135-degree bend has 18 percent more loss than 90, $\therefore$ K = 1.18 (1.35) = 1.593.

<u>STYLE I</u>

Reference Area $= 0.785(d^2 - 0.25d^2) = 0.736\,d^2$

Curtain Area $= \pi dx = \pi d^2\left(\frac{x}{d}\right)$

Area Ratio $= \dfrac{\text{Ref Area}}{\text{Curtain Area}} = \dfrac{0.736d^2}{\pi d^2\left(\frac{x}{d}\right)} = \dfrac{0.234}{\left(\frac{x}{d}\right)}$

Seat K Factor

$\frac{x}{d}$	1/4	1/8	1/16
Area Ratio	0.937	1.874	3.75
C_T	0.48	0.69	0.84
K_T	4.34	2.10	1.41
K_A	3.20	1.815	1.34
K_A'	2.80	6.37	18.84

← Curtain area as reference (K_A)

← Standard reference area (K_A')

Body K Factors

Element	K
45° bend	0.43
135° bend	1.59
2 bellows	0.34
Total	2.36

Standard reference area for all K factors

Total K Factor and Stroke

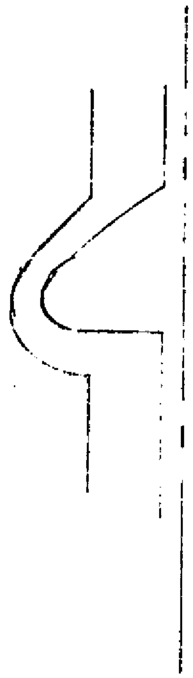
Recall that $a = 1.325\sqrt{K}$ from a previous page, and $a = 0.736d^2$, so

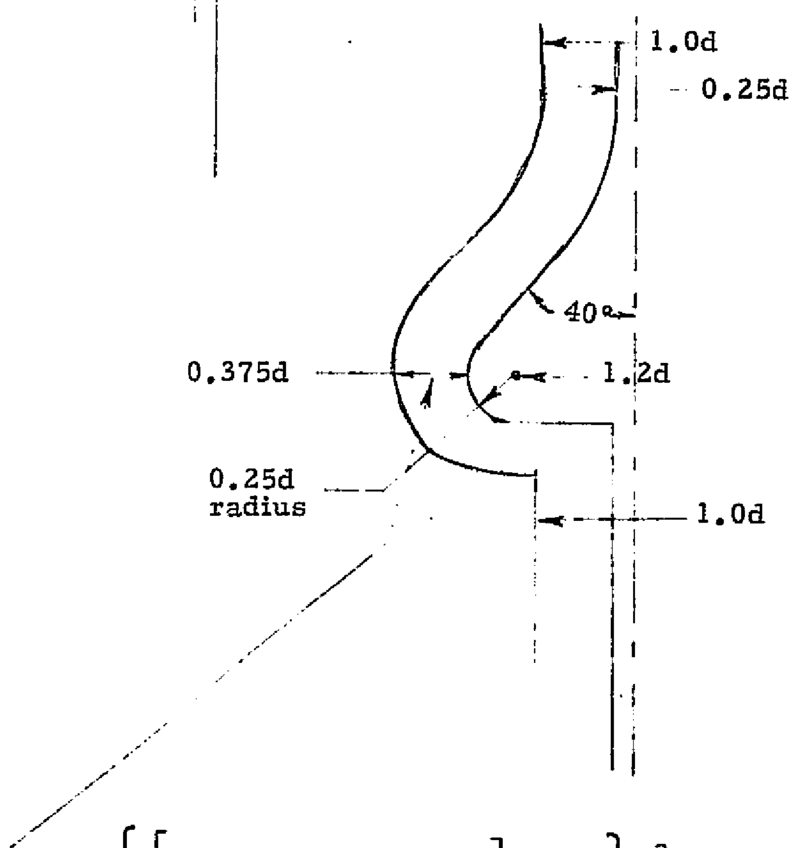
$$0.736d^2 = 0.663\sqrt{K}$$

$$d^2 = 0.900\sqrt{K}$$

$$d = 0.949(K)^{1/4}$$

$\frac{x}{d}$	1/4	1/8	1/16
Body K	2.36	2.36	2.36
Seat K	2.80	6.37	18.84
Total K	5.16	8.73	21.20
$K^{1/4}$	1.507	1.719	2.146
d	1.43	1.63	2.036
x	0.358	0.204	0.127





Flow area $\left\{\pi\left[1.2 + 2(0.25) + 0.375\right]\ 0.375\right\} d^2 = 2.44d^2$

$$\text{Aspect ratio} = \frac{\pi 1.2 + 2(0.25) + 0.375}{0.375} = 17.36$$

$$\frac{r}{d} = \frac{0.25 + \frac{0.375}{2}}{0.375} = 1.16$$

Scale: d = 1.00 inch

STYLE II

Reference area = $0.736\,d^2$ as before

Seat K factors: as before

Body K Factors

40-degree bend use rectangular duct data, Page A-29, SAE book

$$\frac{r}{b} = \frac{0.500}{0.375} = 1.333$$

$$\frac{a}{b} = \frac{\pi(1.0)}{0.375} = 8.37 \text{ (average)}$$

For 90° K = 0.205

For 40° K = 0.44(0.205) = 0.0902 Standard reference area

140° bend use rectangular duct data

$$\left.\begin{array}{l} \dfrac{r}{b} = 1.16 \\ \\ \dfrac{a}{b} = 17.36 \end{array}\right\} K = 0.7 \text{ for } 90° \text{ (local reference area)}$$

For 140°, K = 1.42 (0.7) = 0.994

$$\text{Area ratio} = \frac{0.736}{2.44} = 0.302$$

$K = (0.302)^2\, 0.994 = 0.09$ standard reference area

Bellows

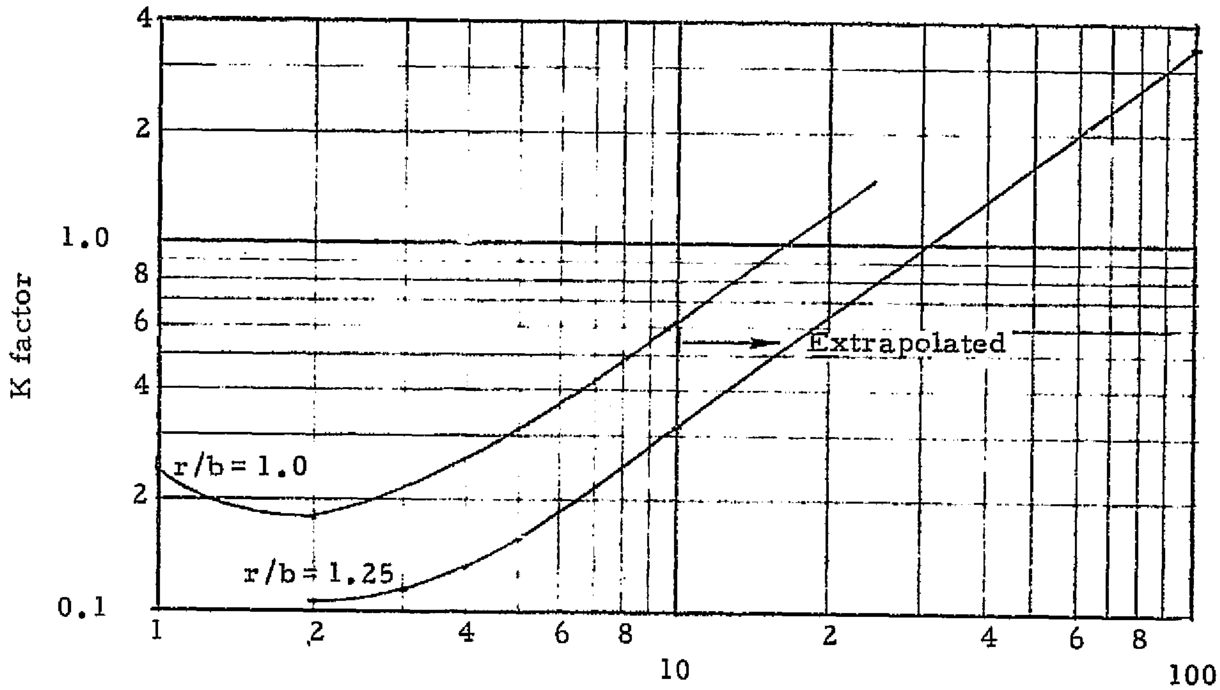
K = 0.34 as before

Body Total 0.09 + 0.09 + 0.34 = 0.52

Total K factor and Stroke

Same method as before $d = 0.949(K)^{1/4}$

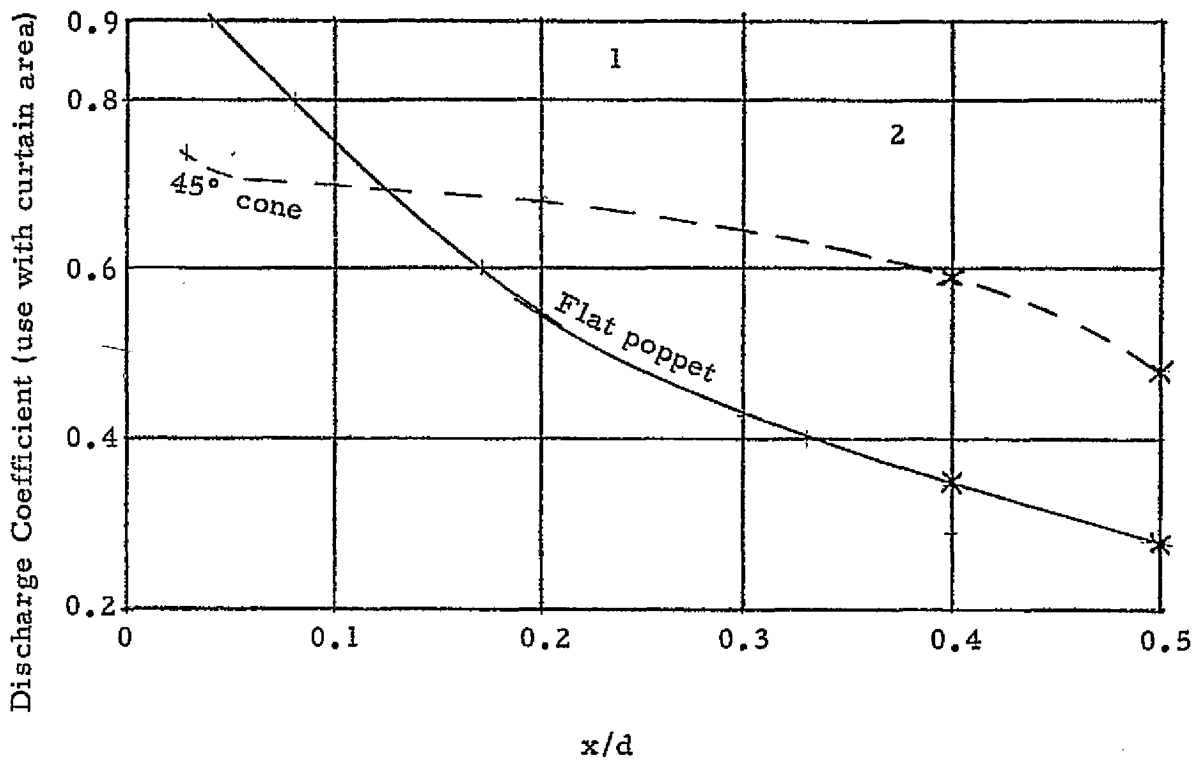
$\frac{x}{d}$	1/4	1/8	1/16
Body K	0.52	0.52	0.52
Seat K	2.80	6.37	18.84
Total K	3.32	6.89	19.36
$K^{1/4}$	1.34	1.62	2.09
d	1.27	1.53	1.98
x	0.317	0.191	0.123



Aspect Ratio = $\frac{\text{width}}{\text{height}}$

Re = $0.6(10)^6$

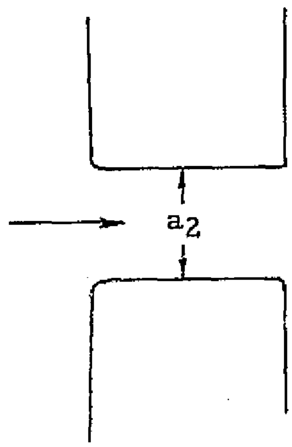
Ref SAE, Page 29



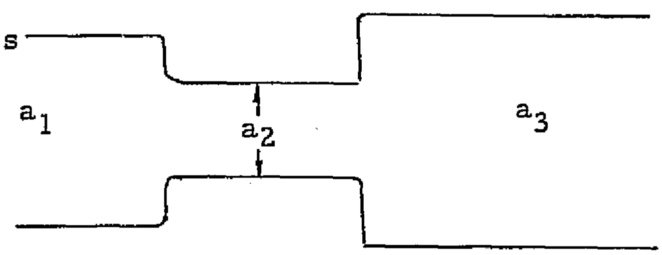
Poppet Valve Discharge Coefficients
(large approach and exit areas)

Application of Poppet Valve Test Data to the Present Application:

Test data for this case:



Application is this case



The loss factor consists of

K_1 = entrance loss

K_2 = passage loss

K_3 = exit loss

For the test case, assuming a rounded entranc3:

$K_1 = 0$ $\qquad$ $K_2 = K_T - 1.0$ $\qquad$ Ref area = hole area

$K_3 = 1.0$ $\qquad$ K Measured in test

The application case also has a rounded entrance, therefore, $K_1 = 0$.

The exit loss K_3 is no longer 1, since the exit area is a_3, not α. There is an addition expansion in the test case, which may be considered to be from a_3 to α, for which

$$K_3 = 1.0 \qquad a_3 = \text{reference area}$$

or,

$$K = 1.0\left(\frac{a_3}{a_2}\right)^2 \text{ where } a_2 = \text{reference area.}$$

This loss is saved in the application, therefore

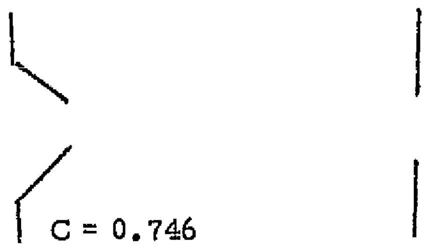
$$\underbrace{K_a}_{\text{application K factor}} = \underbrace{K_T}_{\text{Test K factor}} - 1.0\left(\frac{a_3}{a_2}\right)^2$$

provided the entrance loss coefficients are both zero, or equal for the two cases, as would be the case with slight edge rounding.

Discharge Coefficients for Large x/d Valves

For the 45-degree cone seat, no test data is available for $x/d > 1/8$.

When $x/d \gg 0.25$, the situation reduces to flowthrough at 45-degree chamfered criteria.



C = 0.746 C = 0.611

(data from Rouse)

The pushrod blocks some area - for Burns case, the fraction of open area was

$$0.785\,(0.10)^2 = 0.00785$$

$$0.785\,(0.325)^2 = 0.0829$$

$$\text{Net fraction open} = \frac{0.0829 - 0.00785}{0.0829} = 0.905$$

Therefore discharge coefficient based on hole area for x/d 0.25 is

$$C_{hole} = 0.746\,(0.905) = 0.675 \quad \text{40 degrees}$$

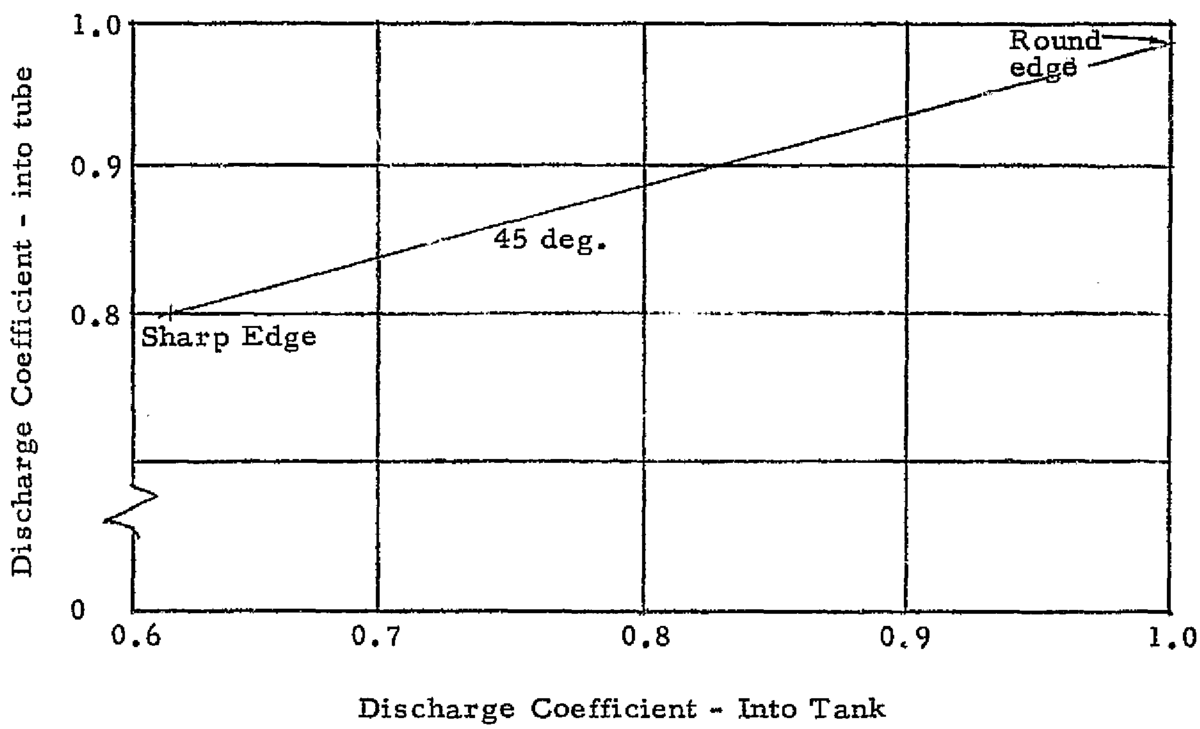
$$0.553 \quad \text{90 degrees}$$

Assuming x/d is large enough so that poppet stroke no longer has any effect, but that curtain area is still used as the reference, then

$$C = \frac{\text{actual flowrate}}{\text{curtain area}}$$

For the flat seat

$$C = \frac{C_{hole}\,\pi d^2}{\pi d \times 4} = \frac{C_{hole}}{4(x/d)} = \frac{0.553}{4(x/d)} = \frac{0.138}{\frac{x}{d}}$$


1.0
0.9
0.8
0
Round edge
45 deg.
Sharp Edge
0.6
0.7
0.8
0.9
1.0
Discharge Coefficient - into tube
Discharge Coefficient - Into Tank

For the 45-degree core seat

$$C = \frac{C_{hole}\,\pi\,d^2}{0.707\,\pi d \times 4} = \frac{C_{hole}}{0.707(4)\left(\frac{x}{d}\right)} = \frac{0.675}{0.707(4)\left(\frac{x}{d}\right)} = \frac{0.238}{\frac{x}{a}}$$

Subject: Pressure Surge versus Closing Time for Quad Bipropellant Valve

The "water hammer" pressure surge due to valve closure was calculated. The following results were obtained for valve closure in less than the critical time:

Fluid	(lb/sec)	Line	Critical Time(1) (sec)	Surge(3) (psi)	Wave Velocity (ft/sec)
N_2O_4	11.91	1.43 I.D. Rigid Wall	0.005	883	3834
N_2O_4	11.91	1.43 I.D. CRES, 0.035 wall	0.006	744	3230
MMH	7.22	1.43 I.D. Rigid wall	0.004	736	5268
MMH	7.22	1.43 I.D. Aluminum(2) 0.035 wall	0.006	473	3385

(1) Calculated for 10' length. For 20' multiply by 2, etc.

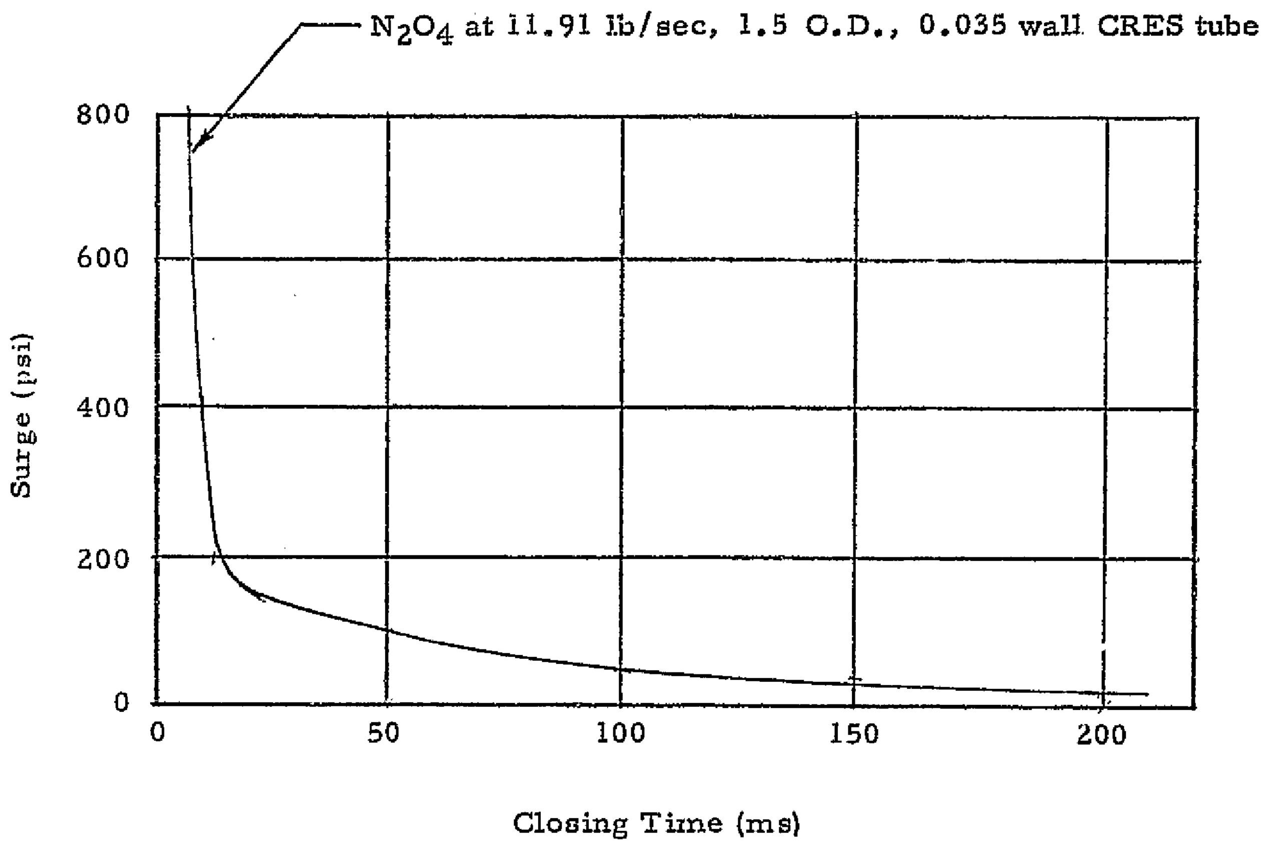
Line lengths, from tank outlets to valve inlets:

MMH = 12.5'
N_2O_4 = 12.9'

(2) Per telecon with Paul Felise at Space Division on 6-27-73, 300 series CRES will be used for both propellants; 1-1/2" O.D.; 0.035" wall.

(3) Independent of line length, when valve closes in less than the critical time.

Results for slower valve closure (assuming a constant rate of decrease of ca) were calculated by the Allievi formula, and are plotted in Figure 1.



Constant Rate of Decrease of Valve CA

Note: Curve plotted for 10-foot long line. Multiply closing time by 2 if line is twice as long, etc.

Closing Response versus Surge Pressures

Line Size = 1.5 in. O.D., 0.035 wall

$$a = 0.785\,[1.5 - 2(0.035)]^2 = 1,605 \text{ in.}^2$$

Velocity

N_2O_4

$$\rho = 90.2 \text{ lb/ft}^3$$

$$\dot{w} = 11.91 \text{ lb/sec}$$

$$v = \frac{w}{\rho a} = \frac{11.91(144)}{90.2(1.605)} = 11.84 \text{ ft/sec}$$

MMH

$$\rho = 54.8 \text{ lb/ft}^3$$

$$w = 7.22 \text{ lb/sec}$$

$$v = \frac{\dot{w}}{\rho a} = \frac{7.22(144)}{54.8(1.605)} = 11.82 \text{ ft/sec}$$

Bulk Modulus Source (psi)	MMH	N_2O_4
Aerojet	$3.28(10)^5$	$2.86(10)^5$

Acoustic Velocity

$$C = \sqrt{\frac{BG}{\rho}} =$$

for N_2O_4

$$C = \sqrt{\frac{2.86(10)^5(144)\ 32.2}{90.2}} = 3,834 \text{ ft/sec}$$

for MMH

$$C = \sqrt{\frac{3.28(10)^5(144)32.2}{54.8}} = 5,268 \text{ ft/sec}$$

Maximum Surge (for rigid pipe)

$$\Delta P = \rho \frac{vC}{g}$$

for N_2O_4

$$\Delta P = \frac{90.2(11.84)3.834}{(32.2)144} = 883.0 \text{ psi}$$

for MMH

$$\Delta P = \frac{54.8(11.82)5,268}{(32.2)144} = 735.9 \text{ psi}$$

Maximum Surge (for elastic pipe)

The effective bulk modulus becomes

$$B' = \frac{BEt}{Et + Bd}$$

For MMH

Aluminum tube $E = 9.9(10)^6$ psi

$t = 0.035$ $d = 1.5$

$$B' = \frac{3.28(10)^5\ 9.9(10)^6\ 0.035}{9.9(10)^6\ 0.035 + 0.328(10)^6(1.5)} = 1.355(10)^5 \text{ psi}$$

$$C' = \sqrt{\frac{1.355(10)^5}{3.28(10)^5}}\ 5.268 = 3385 \text{ ft/sec}$$

$$\Delta P' = \sqrt{\frac{1.355(10)^5}{3.28(10)^5}}\ 735.9 = 473 \text{ psi}$$

For N_2O_4

CRES tube, $E = 30(10)^6$ psi

$t = 0.035 \qquad d = 1.5$

$$B' = \frac{2.86(10)^5 30.0(10)^6 0.035}{30(10)^6 0.035 + 0.286(10)^6 1.5} = 2.03(10)^5$$

$$C' = \sqrt{\frac{2.03(10)^5}{2.86(10)^5}} 3,834 = 3230 \text{ ft/sec}$$

$$\Delta P' = \sqrt{\frac{2.03(10)^5}{2.86(10)^5}} 883 = 744 \text{ psi}$$

Critical Time

$$T = \frac{2\ell}{C}$$

$$T' = \frac{2\ell}{C'}$$

For MMH, $\ell = 10$ ft

$$T' = \frac{2(10) \text{ ft}}{3385 \text{ ft/sec}} = 5.9 \text{ ms}$$

For N_2O_4, $\ell = 10$ ft

$$T' = \frac{2(10)}{3230} = 6.19 \text{ ms}$$

For 200 ms closing time, N_2O_4

$$\Delta P' \frac{(774)(6.19)}{(200)} = 24 \text{ psi}$$

As an aid to preliminary design, the surge may be calculated for assumed uniform rate of change of value Ca, using the allievi method.

From Rich, Hydraulic Transients

$$\rho \equiv \frac{aV_0}{2gH_0}$$

$$\theta = \frac{aT}{2}$$

Converting from Rich's nomenclature

Rich	Kenyon	
H_0	P/ρ	static head
V_0	v	initial velocity
g	g	gravitational constant
a	C'	wave velocity
T	T	valve closing time
ℓ	ℓ	duct length.

Then for the case

N_2O_4

$C' = 3230$ ft/sec

$P = 270$ psi 38,880 psf $\}$ $H_0 = 431$ ft

$\rho = 90.2$ lb/ft^3

$T = 0.2$ sec

$\ell = 10$ ft

$v = 11.84$

$$\rho = \frac{3230(11.84)}{2(32.2)431} = 1.38$$

$$\theta = \frac{aT}{2\ell} = \frac{3230(0.2)}{2(10)} = 32.3$$

From the alleivi chart, page 27 Rich. $3^2 = 1.04$, so

$\Delta P = 0.04\,(270 \text{ psi}) = 10.8$ psig (graph hard to read for such small values of ρ and large θ)

Alternately, the (approximate) equation of alleivi may be used when $\rho > 1$.

$$3^2 - \rho\frac{3}{\theta} - 1 = 0.$$

For our example, we may assume a 3 and calculate the corresponding θ. This will permit calculation of ΔP versus T.

$$\frac{\rho 3}{\theta} = 3^2 - 1$$

$$\frac{\theta}{\rho 3} = \frac{1}{3^2 - 1}$$

$$\theta = \frac{\rho 3}{3^2 - 1} = \frac{1.383}{3^2 - 1}$$

$$T = \frac{\theta 2\ell}{a} = \frac{2(10)}{3230}\,\theta$$

$$T = 0.00619\theta$$

3^2	θ	T	ΔP
1.04	35.18	0.217	10.8
1.15	9.86	0.061	40.5
1.25	6.17	0.38	67.5
1.50	3.38	0.21	135.0
2.00	1.95	0.012	270.0

Bellows Resonant Frequencies:

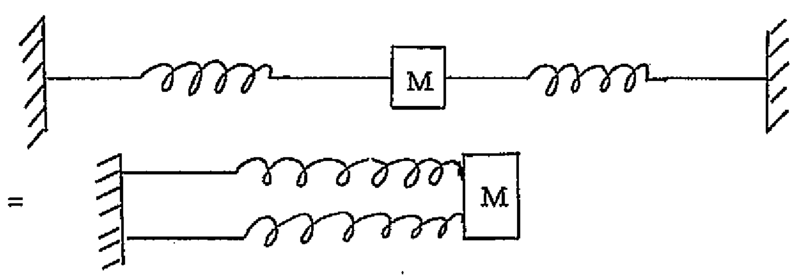
I. Axial Direction:

a. Without Propellant

Weight of each bellows: 0.133 lb

Weight of the seat: 0.125 lb

Spring Rate: 314 lb/in. per bellows



$\therefore k = 314 \times 2 = 628$ lb/in.

$M_{sp} = 2 \times 0.133 = 0.266$ lb

$M = 0.125$ lb

$$\therefore f_n = 3.13 \sqrt{\frac{k}{M + \frac{M_{sp}}{3}}}$$

$$= 3.13 \sqrt{\frac{(628)}{0.125 + \frac{0.266}{3}}} = 169.7 \text{ cps}$$

b. With Propellant

Additional weight of the propellant

$$= \frac{\pi}{8}\left(1.96^2 - 1.53^2\right)(20 \times 0.224 - 0.29)\left(\frac{90.2}{1728}\right) = 0.129$$

$$f_n = 3.13 \sqrt{\frac{628}{0.125 + \frac{0.266 + 0.129}{3}}} = 154.8 \text{ cps}$$

II. Lateral Direction

a. Without Propellant

$$f_t = 2.52 \frac{D_M}{\ell} f_n$$

$$= 2.52 \times \frac{1.96 + 1.53}{2 \times (20 \times 0.224)} (169.7)$$

$$= 166.6 \text{ cps}$$

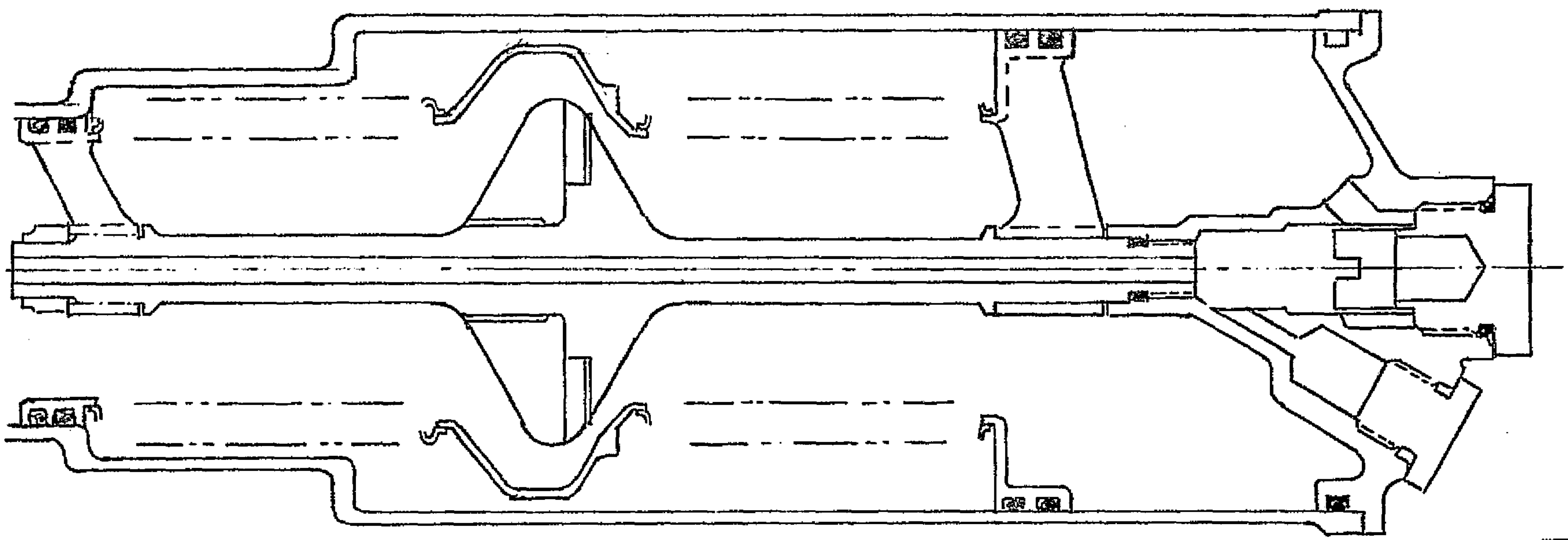
b. Without Propellant

$$f_t = 24.85 \frac{D_M}{\ell} \left(\frac{k}{w}\right)^{1/2}$$

$$k = 107 \text{ lb/in.}$$

$$w_{prop} = \frac{\ell}{8}\left(1.96^2\right) + 1.53^2 \quad (20 \times 0.224 - 0.29) \frac{90.2}{1728} = 0.531$$

$$f_t = 104.27$$



Bellows: O.D. = 1.96"
I.D. = 1.53"
Pitch = 0.224"
Convolutions = 10 (each bellows)
Number of ply = 3 0.005"/ply
Spring rate = 314 lb/in. (each bellows)
Material = Inconel 718 (ρ = 0.3 lb/in.^{3})

Needed: Resonant frequency of spring/mass system for

1. Axial direction
2. Lateral direction

with and without propellant (ρ = 90.2 lb/ft^3) inside bellows

APPENDIX C

LIFTING BALL VALVE PRESSURE DROP CALCULATIONS

Rated Flows

N_2O_4 $\dot{w}$ = 11.91 lb/sec ρ = 90.2 lb/ft^3

MMH $\dot{w}$ = 7.22 lb/sec ρ = 54.8 lb/ft^3

Since the velocity pressure is proportional to $w^2/$, we may identify the worst case for design by using this figure of demerit:

$$\left(\frac{w^2}{\rho}\right)_{N_2O_4} = \frac{11.91^2}{90.2} = 1.572$$

$$\left(\frac{w^2}{\rho}\right)_{MMH} = \frac{7.22^2}{54.8} = 0.951$$

Thus the N_2O_4 requirement will produce the largest ΔP, and governs the design.

One leg only obviously gives the largest ΔP and governs the design

NOTE: All K factors are for stagnation pressure loss

Values Only

The velocity pressure for the design case is

$$q = \frac{1}{\rho}\left(\frac{w}{0.6690}\right)^2$$

$$w = 11.91 \text{ lb/sec}$$

$$\rho = 90.2 \text{ lb/ft}^3$$

$$a = 0.785\ (0.900)^2 = 0.6358 \text{ in.}^2$$

$$q = \frac{1}{90.2}\left[\frac{11.91}{0.669(0.6358)}\right]^2 = 8.692 \text{ psi}$$

The Reynolds number is:

$$Re = \frac{3.29(10)^{-3}\,\dot{w}}{\mu d}$$

$$\dot{w} = 11.91 \text{ lb/sec}$$

$$d = 0.900 \text{ in.}$$

$$\mu = 5.95(10)^{-9}\ \frac{\text{lb/sec}}{\text{in.}^2}$$

$$Re = \frac{3.29(10)^{-3}11.91}{5.95(10)^{-9}(0.900)} = 7.32(10)^6$$

Wall Friction

From the Moody diagram:

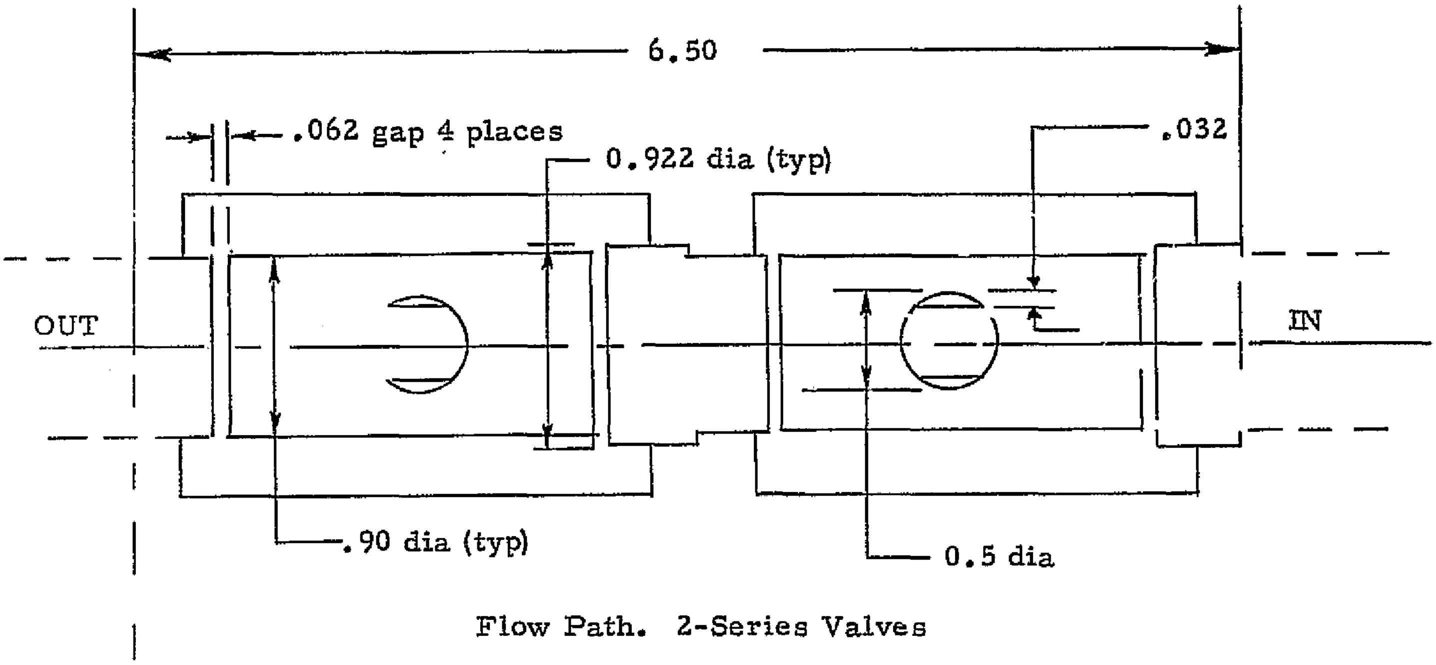
$f = 0.0085$

also

$\ell = 6.50$ in.

$d = 0.900$

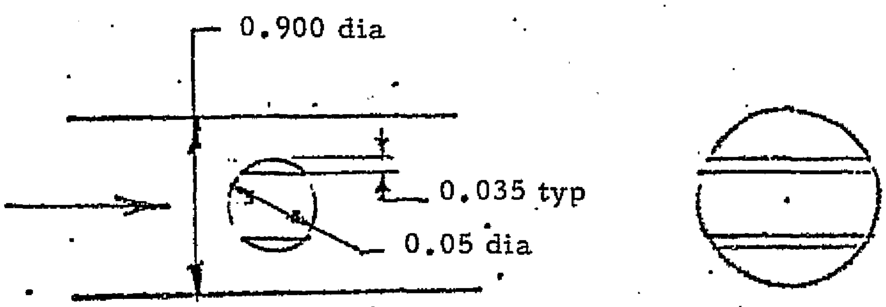
$$k = \frac{f\ell}{d} - \frac{0.0085(6.5)}{0.900} = 0.0614$$



Flow Path. 2-Series Valves

Torque Shaft

There is also an obstruction in the flow path due to the shaft that transmits torque to the lower end of the ball:



This may be analyzed as a venturi flow area:

Area of top section:

$$a = R^2 \cos^{-1} \frac{(R - h)}{R} - (R - h) \sqrt{2h(R - h)}$$

$$R = 0.450$$

$$h = 0.450 - 0.250 = 0.200$$

$$R - h = 0.250$$

$$a = (0.45)^2 \cos^{-1} \frac{0.250}{0.450} - 0.250 \sqrt{2(0.200)\ 0.250}$$

$$a = 0.1198 \text{ in.}^2$$

Half area of midsection

$$a = \frac{\pi R^2}{2} \left[R^2 \cos^{-1} \frac{(R - h)}{R} - (R - h) \sqrt{2h(R - h)}\right]$$

$$R = 0.450$$

$$h = 0.450 - 0.250 - 0.035 = 0.165$$

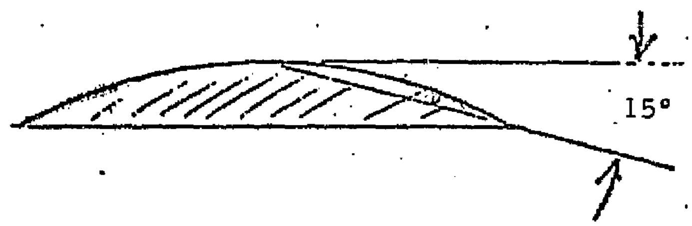
$$r - h = 0.285$$

$$a = \frac{\pi}{2} R^2 \left[(0.45)^2 \cos + \frac{0.285}{0.450} - (0.285) \sqrt{2(0.165)(0.285)}\right] = 0.1389$$

$$\text{Flow area} = 2(0.1198 + 0.1389) = 0.5174 \text{ in.}^2$$

Bore Area = 0.6358 in.^{2}

Area Ratio = $\frac{a_1}{a_2} = \frac{0.5174}{0.6358} - 0.8137$



The diffuser is of non-standard shape, but surely can have no more loss than a standard diffuser of 15 degrees semi-angle, so

$$K = C\left(1 - \frac{a_1}{a_2}\right)^2$$

C = 0.58 from CRC Handbook

$$\frac{a_1}{a_2} = 0.8137$$

$K = 0.58\,(1 - 0.8137)^2 = 0.0201$ for each valve or 0.0402 for both

Gaps

The gaps at the ends of each flow tube may be analyzed by adapting the equation of Hawthorne and Helms (R.E. 10 June '63)

$$K = N\left[1 - \left(\frac{g}{d + 0.876g}\right)^2\right]^2$$

N = 4

d = 0.900

(continued)

$$g = 0.062 \text{ each}$$

$$K = 4\left[1 - \left(\frac{0.900}{0.900 + 0.876/0.062}\right)^2\right]^2 = 0.0484$$

Diversion

For purging purposes, 5 percent of the flow is diverted around the flow tube. The diversion itself does not involve any loss in the main stream, but there are two losses.

a. Diameter step up, upstream of diversion

$$d_1 = 0.900$$

$$d_2 = \sqrt{1.05}\,(0.900) = 0.922$$

for 5 percent diversion

$$K = \left[1 - \left(\frac{d_1}{d_2}\right)^2\right]^2 = \left[1 - \left(\frac{0.900}{0.922}\right)^2\right]^2 = 0.002, \text{ each valve}$$

b. Re-entry of diverted stream from the conservation of momentum:

$$K = 2\left[2\mu - \mu^2(\sigma + 1)\right]$$

$$\mu = 0.05 \text{ (fraction diverted)}$$

$$\sigma = \frac{a_{bore}}{a_{slot}} = \frac{0.6358}{0.7(\ \ )(0.800)\ 0.062} = 5.18$$

$$K = 2\left[2(0.05) - (0.05)^2(5.18 + 1)\right] = 0.1691 \text{ each valve}$$

Summary of K Factors (2 valves)

Element	K
Wall friction	0.0614
2 shafts	0.0402
4 gaps	0.0489
2 diameter step ups	0.0040
2 recombine diverted flow	0.3382
Total	0.4927

ΔP of 2 Valves

$K = 0.4927$ (above)

$q = 8.692$ psi N_2O_4 at 11.91 lb/sec

$\Delta P = Kq = 0.4927(8.692) = \underline{4.283}$ psi for governing case.

For other cases of interest:

a. Both parallel paths, N_2O_4 at 11.91 lb/sec

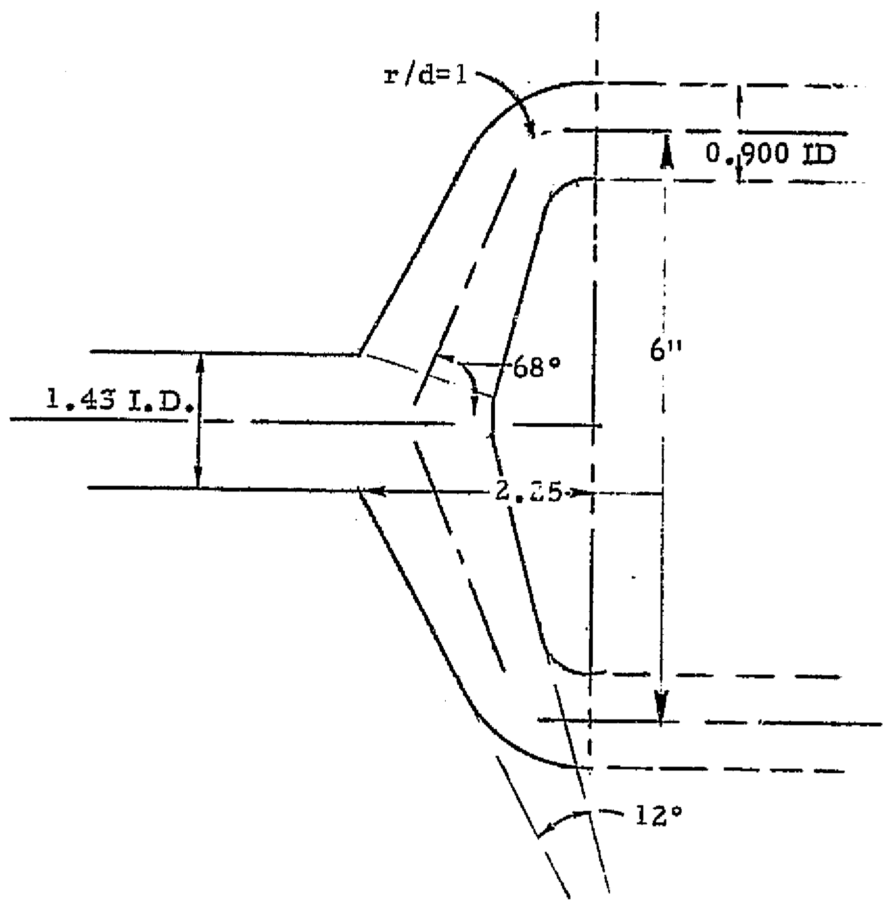
$$\Delta P = \frac{4.283}{4} = 1.071 \text{ psi}$$

b. MMH at 7.22 lb/sec, one leg only

$$\Delta P = \left(\frac{7.22}{11.91}\right)^2 \left(\frac{90.2}{54.8}\right) 4.283 = 2.591, \text{ psi}$$

c. MMH at 7.22 lb/sec, both parallel legs

$$\Delta P = \frac{2.591}{4} = 0.647$$



Flow Path Through Manifold

Manifold + Valves

At the governing case of 11.91 lb/sec N_2O_4 via one side only, the velocity pressure is:

$$q = \frac{1}{\,}\left(\frac{\dot{w}}{0.669a}\right)^2 = \frac{1}{90.2}\left(\frac{11.91}{0.669(1.605)}\right)^2 = 1.364 \text{ psi}$$

68° Diverging Y

From SAE Book, page A-58, for $\frac{V_2}{V_1} = 1.0$, 68° angle:

$K = 1.13$

Converging Cone

$K = 0$

Bend, 68°, R/d = 1

From SAE book, page A-19 for 90°, R/D = 1 and $K_e = 7.32\,(10)^6$

$K = 0.15$

Correction factor for 68° bend, from page A-20:

$C = 0.85$

$\therefore K = 0.85(0.15) = 0.128$

Valve Assembly (2 in series)

As previously derived

$K = 0.493$

Bend, 68°, R/d = 1

As before, K = 0.128

Diverging Cone, 10° included angle

$d_1 = 0.900$

$d_2 = 1.430$

For 12° included angle, from CRC handbook page 256:

C = 0.10

$$K = C\left[1 - \left(\frac{d_1}{d_2}\right)^2\right]^2 = 0.10\left[1 - \left(\frac{0.9}{1.43}\right)^2\right]^2 = 0.036$$

Converging Y, 68°

From SAE book, page A-54

K = 1.07

Summary	Local K	Reference Diameter	K Referred to d = 1.43
Diverging Y	1.130	1.430	1.113
Converging Cone	0.000	1.430	0.000
Bend	0.128	0.900	0.815
Valve Assembly	0.493	0.900	3.142
Bend	0.128	0.900	0.815
Diverging Cone	0.036	0.900	0.229
Converging Y	1.07	1.430	1.070
			7.184

ΔP = kq = 7.184(1.364) = 9.80 psi. N_2O_4 at 11.91 lb/sec via one side only

APPENDIX D

PLANETARY GEAR TRAIN ANALYSIS

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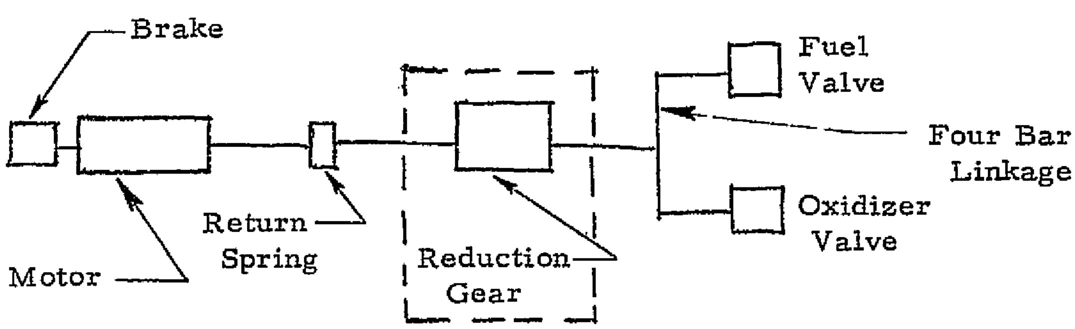
Actuation System

Motor/Valve Reduction System Selection

Initially, the design requirement was identified as a single electric motor operating a set of two identical lifting ball valves via valve shaft mounted arms. Several mechanisms were evaluated for this actuation train; the considerations were:

- Reliability
- Weight
- Technical Risk
- Packaging

The above resulted in the following, typical to each 1/4 of total quad:



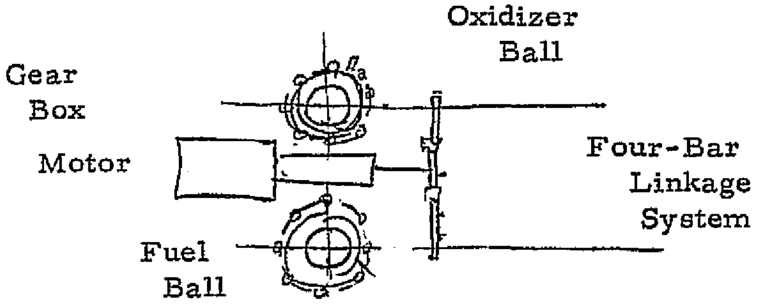
Reliability and repeatability of actuation required the use of a high rpm induction motor for 250 ms closing and 500 ms opening.

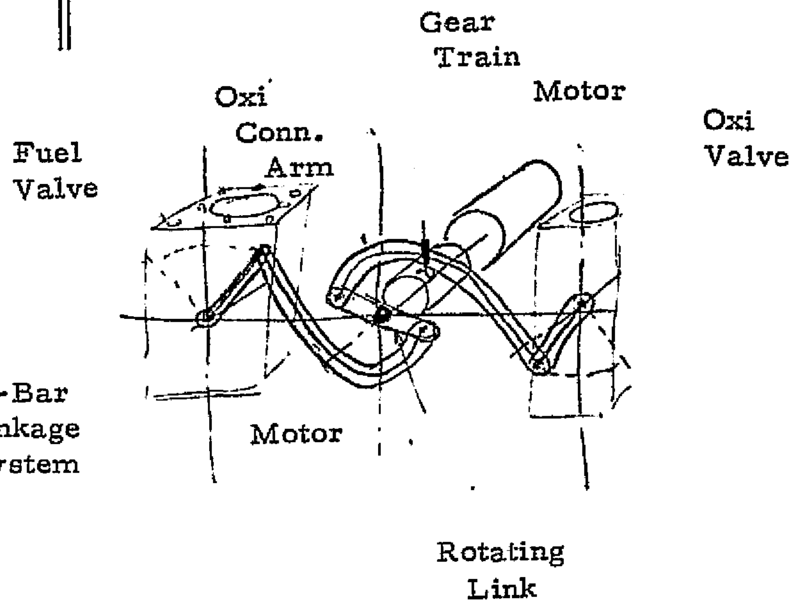
	Gear Train Rate 72:1	77:1	120:1	
Motor Speed (rpm) Closing	7680	8200	12,800	
Motor Speed (rpm) Opening	3840	4100	6,400	
Required Motor Torque (τ_R)	0.66	0.63	0.372	Opening Mode
Designed Motor Torque (τ_D)	1.32	1.26	0.744	Opening Mode

Gear Train Output Shaft

Above based on $\left(\dfrac{160°}{0.25 \text{ sec}} \times \dfrac{R}{360°} \times \dfrac{60 \text{ sec}}{1 \text{ min}}\right) \dfrac{\text{Gear}}{\text{Ratio}} = \dfrac{\text{Motor}}{\text{rpm}}$

Total acceptance package and weight plus maximum allowed pressure drop across (total) quad (contraction, expansion, joints, etc) required the valve ℄ to be close to each other. Since this optimum mechanical advantage was obtained with a four-bar linkage, the motor and gear box ℄ had to be parallel and equidistant to the valve ℄ .



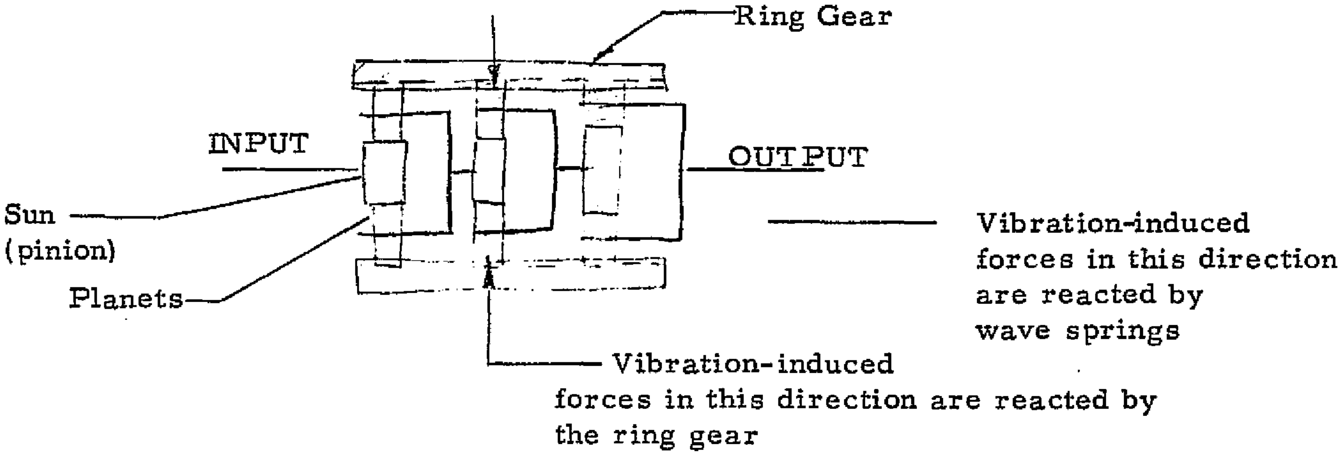


All of these requirements limit reduction as follows:

1. Reliable
2. Small diameter (gear box between valves)
3. High reduction (77 : 1 ratio)
4. Coaxial with motor
5. Non-reversing (driven not 'going' train)
6. Not vibration sensitive

- Requirements 1, 3, 4, and 5 can be met by most (involute) spur gear systems.
- Requirement 6 requires heavy bearings and a large package.
- Requirement 2 may be met by harmonic drive but 1 and 6 plus price and delivery are a concern.

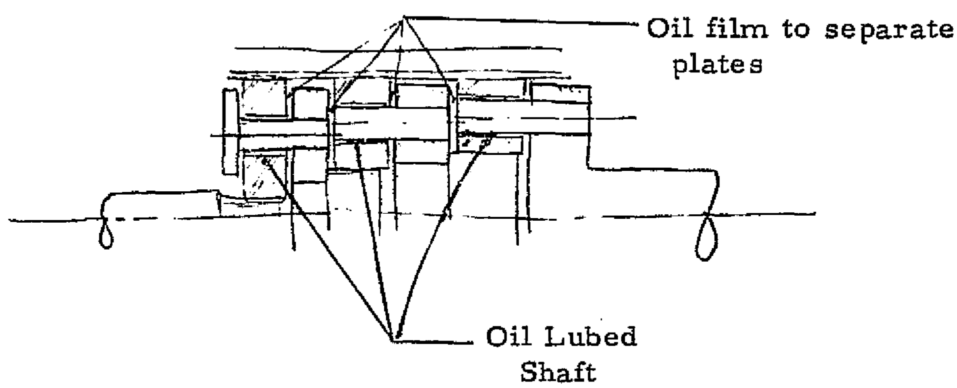
All the above requirements are met by a multiple-stage planetary drive in which the planets are held within a common, stationary ring gear (integral with the case)



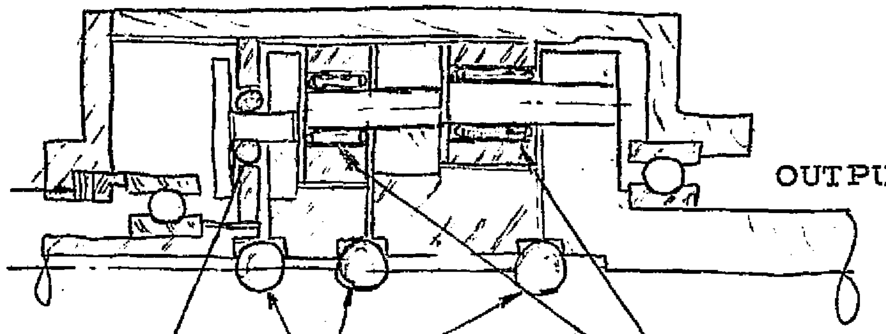
(For detail design see Engineering Records 4.5.11 Dec '73, 3 July '74)

Once the planetary gear train was tentatively selected, the materials, efficiency, strength, size and vibration sensitivity were considered:

Environment: Most multiple stage gear trains rely on an oil film to control contact between plates and to lubricate the planets.



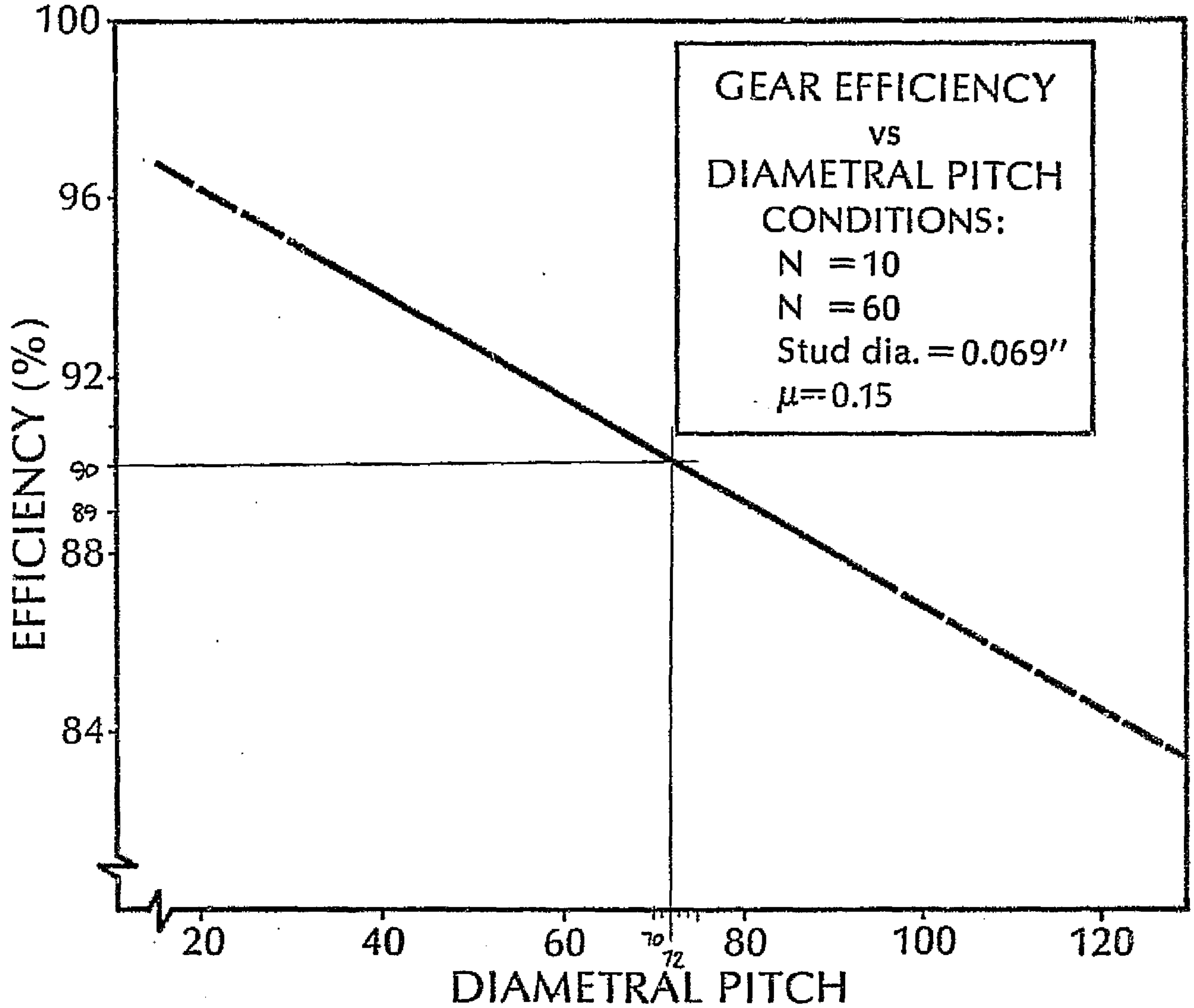
Our reduction requirements (77 : 1) could have been satisfied by a gear train as sketched above, but due to environmental consideration, no conventional lubricants were used. Instead, our design was modified as shown on the following page.



In addition, all gears are coated with molybdenum disulphide based solid lubricant. Some Krytox 260K grease (vacuum grease) is used at the roller bearings.

Efficiency:

Gears with diametral pitch of 72 were chosen for good efficiency, smooth run and accuracy of manufacturing. See figure below for efficiency curve.

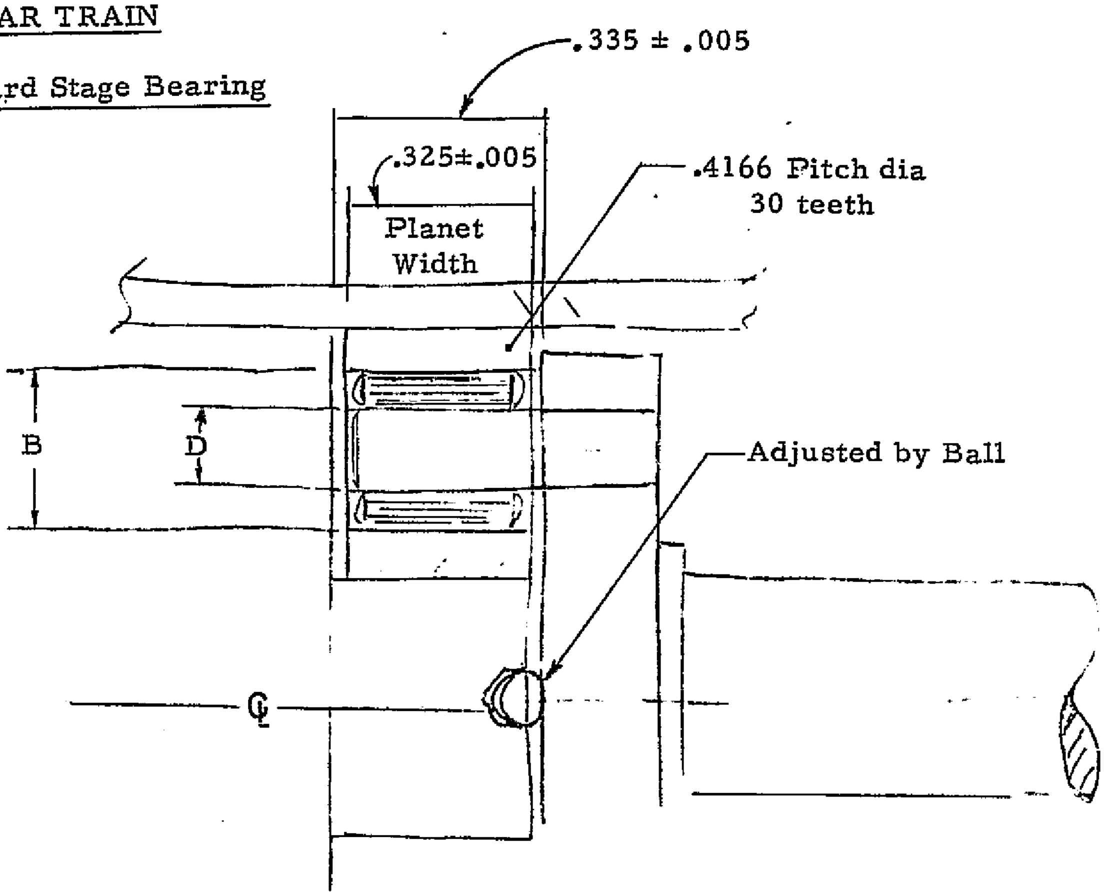


Three stages = 0.9 x 0.9 x 0.9 = 0.73 efficiency of our box.

A 72 dimetral pitch with 20° also results in a very favorable strength-to-weight ratio tooth form which contributes toward a lighter overall weight.

GEAR TRAIN

Third Stage Bearing



For 0.0625d 12 Rollers

$$h = 12; \; K = \frac{1}{\sin \frac{180°}{12}} = 3.863703$$

$$P.D. = Kd + 0.001 = 3.863703 \times 0.0625 + 0.001 = 0.24248 \text{ nom dia}$$

$$B = P.D. + d = 0.24248 + 0.0625 = 0.30498 \approx 0.3050^{+0.005}_{-0.000}$$

$$D = P.D. - d - d.c. = 0.24248 - 0.0625 - 0.0005$$

$$= 0.17948 \approx 0.1795^{+0.0000}_{-0.0003}$$

where d.c. = 0.0005 min

Load $= \frac{322 \text{ lb}}{3 \text{ brgs}} = 74$ lb/bearing

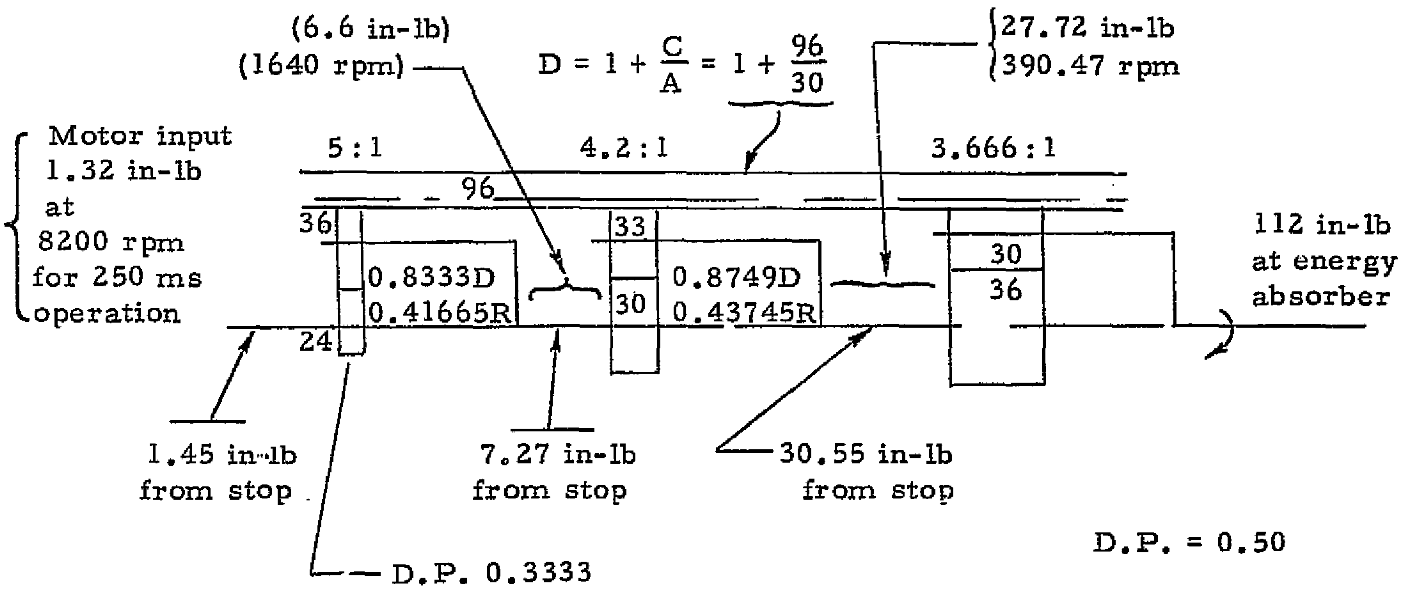
Speed $= \frac{8200\ (36)}{(5)(4.2)(30)} = 468$ rpm SF = 2.2 (page 119 torr)

L.F. = 0.38 (for 20 hrs p 119 torr)

RF = 1.1 (page 120 torr)

HF = 7.15 (R_c 35) (page 120 torr)

Req. BD_c = (74)(2.2)(0.38)(1.1)(7.15) = 486 lb



Class 3 (precision) gears, diametrical pitch 72

Maximum error in action = 0.0005 (ref Mach. Handbook Page 718)

Material: 440C R_c 33-35 (Brinnell 302-331)

Tensile 142-155 ksi

GEAR TRAIN

Rollers - Dynamic Capacity

Second Stage Planets - 12 rollers 0.0469 dia x 0.166 lg Q - 8271

Basic Dynamic Capacity (BDC) = $A\, d^{\frac{29}{27}}\, \ell^{\frac{7}{9}}$ (Torrington Cat 567 pp 118-123)

Where A = function of number of rollers

d = roller diameter, inches

ℓ = effective length of roller contact, inches

$A = 36,700$

$d^{\frac{29}{27}}\, \ell^{\frac{7}{9}} = 0.00705$ for spherical end rollers

BDC = (36,700)(0.00705) = 258 pounds available

Load on bearings - $\frac{64 \text{ lb}}{3 \text{ brgs}} = 21.4$ lb/brg

Speed is 1640 rpm x $\frac{30}{33}$ = 1490 rpm; SF = 3.1 (from p 119 torr)

Hours of use = (10,000 cycles) $\left(\frac{34}{5}\right)\left(\frac{\text{rev}}{\text{cycle}}\right)\left(\frac{1 \text{ min}}{1640 \text{ rev}}\right)\left(\frac{1 \text{ hr}}{60 \text{ min}}\right) = 0.71$ hrs

Use 20 hours as safety factor; LF = 0.38 (from P 119 torr)

Rate rotation factor, RF = 1.1 (from p 120 torr)

Hardness factor, HF = 7.15 (R_C 35)

The Required Basic Dynamic Capacity, BDC = Load on bearing x SF x LF x RF x HF

= (21.4)(3.1)(0.38)(1.1)(7.15) = 198 pounds

Third Stage Planets - 12 rollers 0.0625 dia x 0.310 lg S-1414-Q

$d^{\frac{29}{27}}\, \ell^{\frac{7}{9}} = 0.0178$; A = 36,700; BDC = 36,700 x 0.0178 = 653 pounds.

Basis of Calculations:

$$F_t = \frac{S_w f y}{p} \left(\frac{1200}{1200 + V} \right)$$

This is the Lewis equation with zero service and lubrication factors. (Reference Design of Machine Elements, Vallance and Doughtie)

F_t = tangential force at the pitch line

S_w = Safe Stress (approximately 1/3 of ultimate strength)

f = gear width

Y = form factor

V = velocity at pitch line in f_{psn}

F_w = limit load for wear

S_{es} = surface endurance limit

$E = 29 \times 10^6$

N_p, N_g = number of teeth in pinion, p or gear, g

$\phi = 20°$

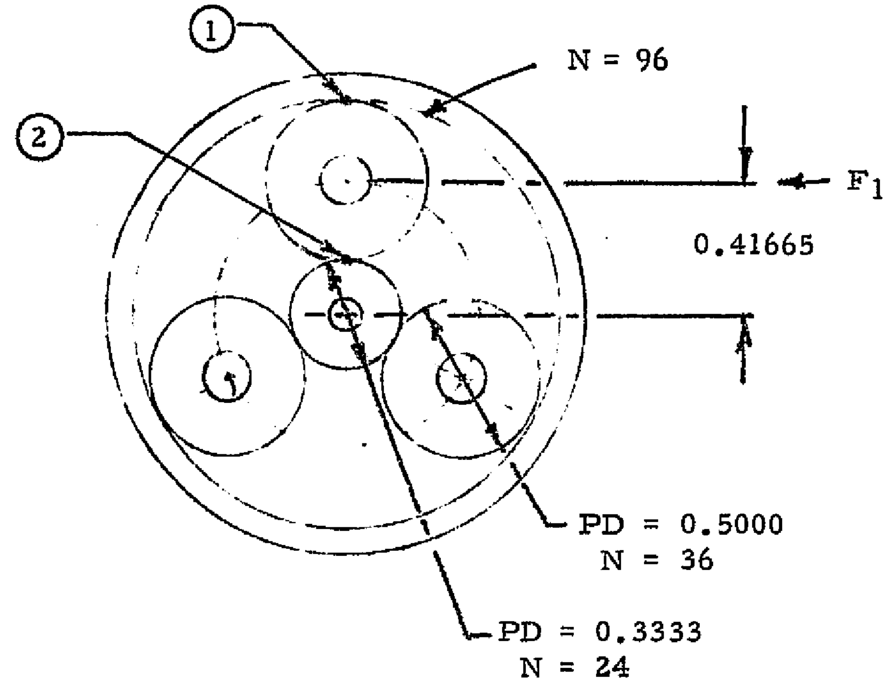
For wear:

$$F_w = \frac{(D_p)(f)(S_{es})^2(\sin\phi)}{1.4}\left(\frac{2 n_g}{N_p + N_g}\right)\left(\frac{1}{E_p} + \frac{1}{E_g}\right)$$

S_{es} = (400)(Brinell No.) - 10,000 = (400)(302) - 10,000 = 110, 800

S_w = 67,000 (Ref Des. of Mach p 382)

FIRST STAGE



$$F_1 = \frac{6.6 \text{ lb-in.}}{0.41665} = 15.8406$$

(1) (2) Tangential force on input pinion (sun gear) is:

$$F_{(1)(2)} = \frac{1.32 \text{ lb-in.}}{\frac{0.3333}{2}} = 7.9208 \text{ pounds.}$$

For calculated pinion wear is can be assumed that the load is distributed among three surfaces with a constant full contact of at least 70 percent or equivalent of

$$F_w = \frac{7.9208}{2} = 3.9604 \text{ pounds}$$

Considering higher loads imposed by stop, i.e., 112 lb-in. at 3rd stage

output $\rightarrow \dfrac{112 \text{ lb-in.}}{(3.666)(4.2)} = 7.274$ lb-in. at first stage output

then $F_1 = \dfrac{7.274}{0.41665} = 17.458$ pounds; because two surfaces in contact

$$F_1 = \frac{17.458}{2} = 8.729 \text{ pounds.}$$

Assuming a minimum of two contact surfaces out of three planets

$$F_t = \frac{8.729}{2} = 4.3645 \text{ pounds}$$

For pinion strength, and assuming a service factor = 0.80 and a lubrication factor = 0.50 (medium shock p 383 Design of Machine Elements)

$$F_t = \frac{S_w \, f \, Y}{P}\left(\frac{1200}{1200 + V}\right)(SF)(LF) \; ; \quad Y = 0.336, \; V = 715.58$$

Solve equation for f which gives minimum gear width.

$$F = \frac{(F_t)(P)}{(S_w)(Y)}\left(\frac{1200 + V}{1200}\right)\frac{1}{(SF)(LF)}$$

$$= \frac{(3.96)(72)}{(67,000)(0.336)} \quad \frac{1200 + 715.58}{1200} \quad \frac{1}{(0.80)(0.50)} = 0.05054 \text{ inch}$$

For pinion wear:

$$F_w = \frac{(D_p)(f)(S_{es})^2 (\sin \phi)}{1.4}\left(\frac{2 N_g}{N_p + N_g}\right)\left(\frac{1}{E_p} + \frac{1}{E_g}\right)$$

(continued)

Solve equation for f which gives minimum gear width.

$$f = \frac{(f_w)(1.4)(N_p + N_g)}{(D_p)(S_{es})^2 (\sin\phi)(2N_g)} \left(\frac{E_p}{1} + \frac{E_g}{1}\right)$$

$$= \frac{(3.96)(1.4)(24 + 36)}{(0.3333)(110800)^2(\sin 20°)(2)(36)} \left(\frac{29 \times 10^6}{1} + \frac{29 \times 10^6}{1}\right) = 0.0479 \text{ in.}$$

This was based on only two surfaces in constant contact. For planet strength, f will be smaller for larger Y's.

For planet wear

$$f = \frac{(3.96)(0.3333)}{(0.500)(82.73)} = 0.032\text{-inch minimum width}$$

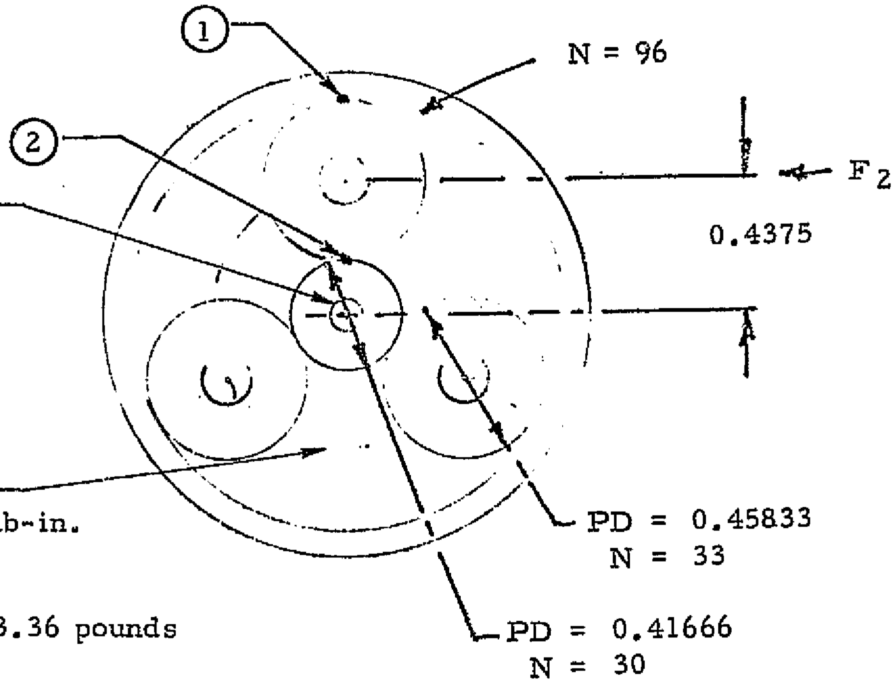
Considering stop loads into pinion strength requirement.

$$F_t = 4.365$$

$$4.365 = 78.344\, f; \quad f = 0.0557\text{-inch minimum}$$

NOTE: Based on antifriction design and calculated size, planet width will be 0.060 inch.

SECOND STAGE



$$F_2 = \frac{27.72 \text{ lb-in.}}{0.4375} = 63.36 \text{ pounds}$$

(1)(2) Tangential force on input pinion (sun gear) is:

$$F_{(1)(2)} = \frac{6.6 \text{ lb-in.}}{\frac{0.4166}{2}} = 31.68 \text{ pounds}$$

For calculated pinion wear it can be assumed that the load is distributed among three surfaces with a constant full contact of at least 73 percent or equivalent of

$$F_w = \frac{31.68}{2} = 15.84 \text{ pounds.}$$

Considering higher loads imposed by stop, i.e., 112 lb-in. at 3rd stage

Output → $\dfrac{112 \text{ lb-in.}}{(3.666)}$ = 30.55 lb-in. at second stage output

then $F_2 = \dfrac{30.55}{0.4375} = 69.83$ pounds; because two surfaces in contact

or

$$F_t = \frac{69.83}{2} = 34.915 \text{ pounds.}$$

Assuming a minimum of two contact surfaces

$$F_t = \frac{34.915}{2} = 17.457 \text{ pounds.}$$

SECOND STAGE

For pinion strength

$$Y = 0.358$$

$$V = \frac{\pi}{12} \times \frac{30}{72} \times 1640 = 178.89$$

$$F_t \approx 15.84 \text{ pounds}$$

$$15.84 = \frac{77,800 \; f \; 0.358}{72} \quad \frac{1200}{1200 + 179} \quad \times 1.00 \times 0.50$$

$$15.84 = 168.27 \; f$$

$$f = 0.094 \text{ inches minimum}$$

For Pinion Wear

$$F_w = 15.84 \text{ pounds}$$

$$15.84 = \frac{0.4166 \ f \ (118.400)^2 \sin\phi}{1.4} \left(\frac{2 \times 33}{30 + 33}\right) \quad 6.896 \times 10^{-8}$$

$$15.84 = 103.07 \ f$$

$$f = 0.1537 \text{ in. min width}$$

For planet strength, f will be smaller for larger Y.

For planet wear

$$15.84 = 103.07 \ f \times \frac{0.4583}{0.4166}$$

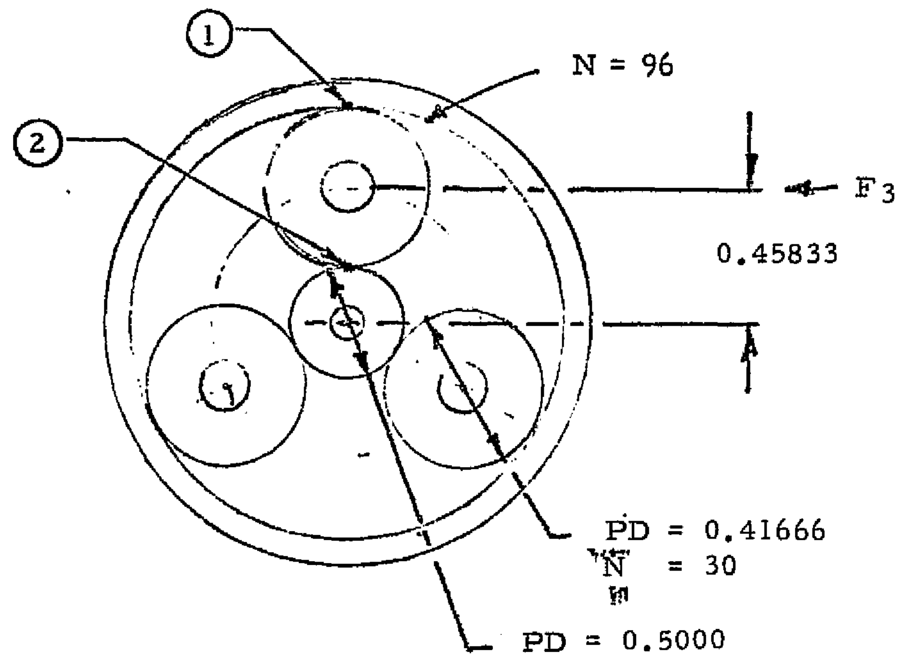
$$15.84 = 113.387 \ f$$

$$f = 0.139 \text{ inches minimum}$$

Considering higher loads imposed by stop, new minimum width of f shall be

$$17.457 = 168.27 \ f$$

$$f = 1.037 \text{ in. minimum}$$



Third Stage Input
(1.32)(5)(4.2) = 27.72 lb min

Third Stage Output
(1.32)(5)(4.2)(3.666) = 101.62 lb-in.

$$F_3 = \frac{101.62 \text{ lb-in.}}{0.45833 \text{ in.}} = 221.717 \text{ pounds}$$

(1) (2) Tangential force on input pinion

$$F_{(1)(2)} = \frac{27.72 \text{ lb-in.}}{\frac{0.500}{2}} = 110.88 \text{ pounds}$$

For calculated pinion wear, it can be assumed that the load is distributed among three surfaces with a constant full contact of at least 70 percent or equivalent of

$$F_w = \frac{110.88}{3} = 36.96 \text{ pounds}$$

Considering higher loads imposed by stop, i.e., 112 lb-in. at 3rd stage

Output → $\frac{112 \text{ lb-in.}}{0.45833}$ = 244.365 lb-in. at first stage output.

then $F_3 = \frac{244.365}{2}$ = 122.182 pounds; because three surfaces in

$$F_3 = \frac{122.182}{3} = 40.7275 \text{ pounds}$$

APPENDIX E

PNEUMATIC ACTUATOR DESIGN DATA

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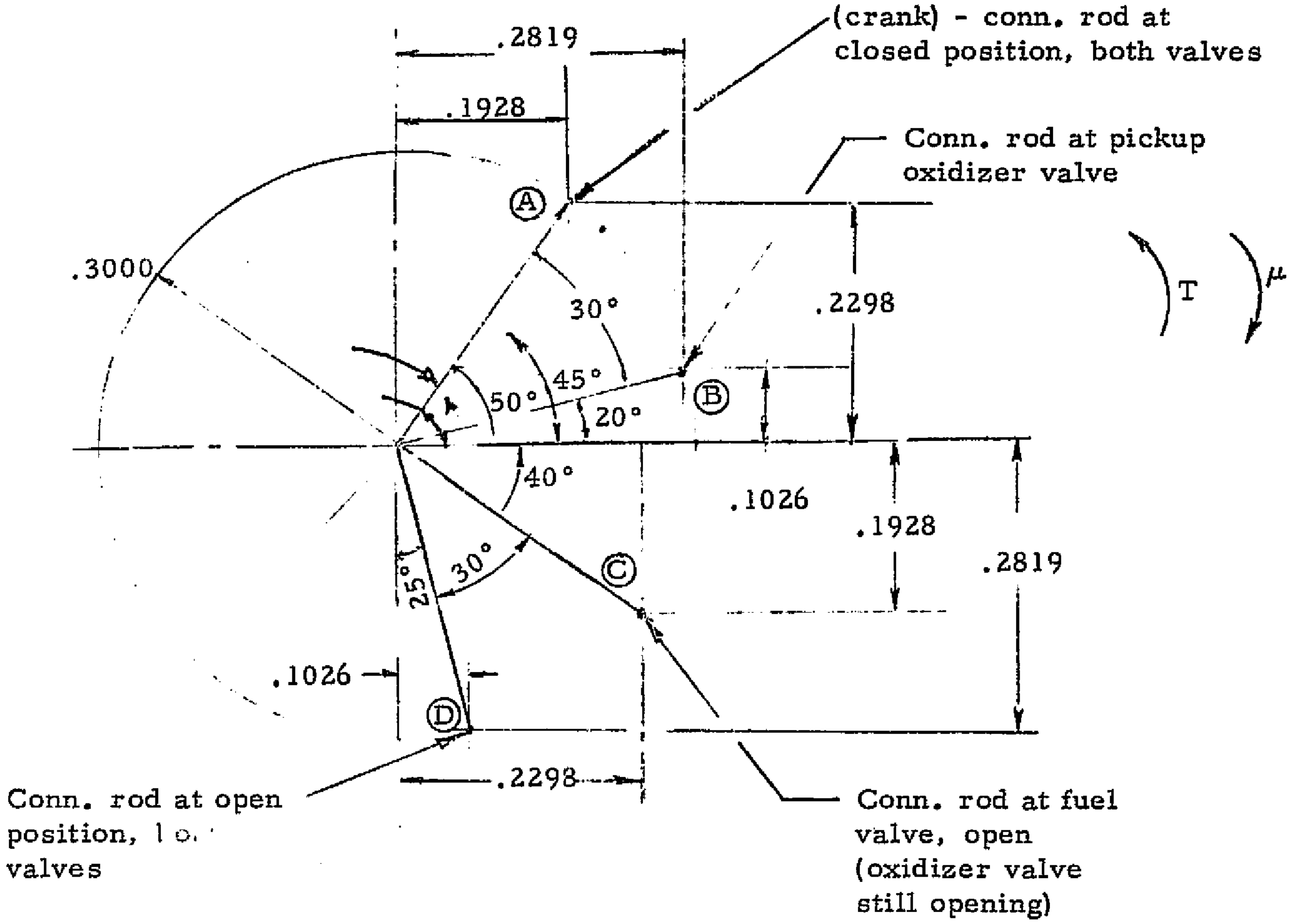
PNEUMATIC ACTUATOR



Actuator connecting rod de-energized position will be approximately

$$\frac{0.3000 - 0.1026}{2} + .1026 = 0.2013$$

use 0.1928 nominal

Actuator Effective Area

$$D_m = 2.0 \text{ in dia.} ; A_{eff} = \frac{\pi}{4}(2)^2 = \pi$$

$$\text{Actuator Forces} = (200 \text{ PSIA})(A_{eff}) = 200\pi = 628 \text{ pounds}$$

$$\text{Displacement} = (A_{eff})(\text{travel}) = (\pi)(.444) = 1.39 \text{ in}^3$$

<u>OPENING MODE</u>

From valve full closed position the initial required torque at the crank shaft is:

$$T_A = (\underbrace{35.5 \text{ lb-in}}_{\text{Valve opening torque req.}} + \underbrace{6 \text{ pounds}}_{\text{valve bi-stable spring force}}) = 41.5 \text{ lb-in}$$

$$\text{Available Torque } M_A = (628 \text{ lbs} - 50 \text{ lbs})(.193) = 111.55 \text{ lb-in}$$

After initial 30° motion at oxidizer valve fork pick-up

$$T_B = (18 + 6)(2) + (35.5 + 6)(1) = 89.5 \text{ lb-in}$$

$$M_B = \left\{628 - \left[400\,(.2298 - .1026)\right] - (50)\right\}(.2819) = 148.59 \text{ lb-in}$$

At the position immediately prior to fuel valve opening point

$$T_{(c-1^\circ)} = (9 - 6)(1) + (18 + 6)(2) = 51 \text{ lb-in}$$

$$M_{(c-1^\circ)} = \left\{628 - \left[400\,(.2298 + .1888) + 50\right]\right\}(.2331) = 95.7$$

At the position where fuel valve is completely open and oxidizer valve is partially open.

$$T_C = (18 + 6)(2) = 48 \text{ lb-in}$$

$$M_C = \left\{628 - \left[400 - (400)(.2298 + .1928) + 50\right]\right\}(.2298) = 102.7 \text{ lb-in}$$

At the position where the oxidizer valve is one degree from opening

$$T_{(B-1°)} = (18 - 6)(1) = 12 \text{ lb-in}$$

$$M_{(B-1°)} = \left\{628 \left[400 (.2298 + .2800) + 50\right]\right\} (.1075) = 40.21 \text{ lb-in}$$

At 100% open position the remaining available torque is:

$$M_B = \left\{628 - \left[400 (.2298 + .2819) + 50\right] (.1026) = 38.30 \text{ lb-in}$$

CLOSING MODE

Closing From full open at point D

$$T_D = (8 + 6)(1) = 14 \text{ lb-in}$$

$$M_D = \left[400 (.2298 + .2819) + 50\right] .1026 = 26.13 \text{ lb-in}$$

If bi-stable valve spring is used then

$$M_D = (373.32)(.1026) = 38.30 \text{ lb-in}$$

Closing, at point when fuel valve is picked up, point C.

$$T_C = (4+6)(2) + (8 + 6)(1) = 34 \text{ lb-in}$$

$$M_C = \left[400 (.2298 + .1928) + 50\right] (.2298) = 50.335$$

Closing, at the point when the oxidizer valve is almost closed (B+1°) and fuel valve is approx 2/3 closed.

$$T_{(B+1°)} = (4-6)(2) + (4+6)(2) = 20 \text{ lb-in}$$

closing at point B = same as $T_{(B-1°)}$

closing at point A = (50)(.1928) = 9.64 lb-in

APPENDIX F

TEST PROCEDURES

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PARKER HANNIFIN • 18321 JAMBOREE BOULEVARD • IRVINE, CALIFORNIA 92664

CONTROLLED DOCUMENT

NUMBER: DVT 5739000

TITLE: Design Verification Test Procedure, Orbiting Maneuvering Engine Propellant Valve

RELEASE HISTORY							
DATE	REVISION	E.O. NO.	MICROFILM	DATE	REVISION	E.O. NO.	MICROFILM

REFERENCE:

PREPARED BY: Vance D. Dunn
Vance D. Dunn
Project Engineer

APPROVED BY: Vance D. Dunn
Vance D. Dunn
Project Engineer

1.0 SCOPE

This procedure outlines the verification testing procedure to be performed on the Orbiting Maneuvering Engine (OME) Propellant Valve Assembly.

2.0 APPLICABLE DOCUMENTS

The Parker-Hannifin documents listed below form a part of this document, except if conflicts exist, this document takes precedence.

QCPM610 Calibration Records and Controls
QCPM620 Calibration of Test Equipment
1PMP5730018 Program Plan, OMS Engine Propellant Valve Technology Program.

3.0 GENERAL REQUIREMENTS

3.1 Environmental

Unless otherwise specified all testing shall be conducted within the following environmental conditions:

Temperature 70°F ± 10°F.
Relative Humidity 90% maximum.
Barometric Pressure: Local Atmosphere.

3.2 Tolerances

Unless otherwise specified, the following tolerances apply to test parameters:

Temperature ± 5°F.
Electrical ± 1% F.S.
Weight - Secondary standard requirements.

3.3 Test Facilities

All test facilities to be used for these tests must be approved by the project engineer.

3.4 **Failure Reporting**

Any test result that does not meet the specified requirement constitutes a failure and terminates the test immediately until the cause of the variance is determined and/or the project engineer authorizes continuation.

3.5 **Test Equipment**

Test equipment, as called out in test setup figures, is identified in detail in Table 3-1. The test equipment shall be capable of producing, maintaining, and indicating the specified test conditions. Applicable instrumentation shall be subjected to periodic calibration in accordance with QCPM610 and QCPM620 and shall be marked with calibration due dates.

3.6 **Test Sequence**

The sequence of testing is optional and any order can be accomplished with approval of the project engineer.

3.7 **Photographs**

Photographs of good resolution shall be taken of all test setups.

3.8 **Test Results Data**

Complete test results shall be recorded for each test. Data will be maintained on supplied data sheets or strip chart outputs; refer to Paragraph 4.3.

3.9 **Disposition of Data**

At the conclusion of testing all data will be provided for the project engineer.

3.10 **Test Report**

A final test report shall be prepared by engineering for inclusion in an overall project report.

3.11 **Disposition of Test Specimen**

Test specimens shall be cleaned, packaged, and turned over to engineering at conclusion of testing.

TABLE 3-1

TEST EQUIPMENT

Description	Range
Voltmeter	1. 0-32 Vdc 2. 0-32 Vdc 3. 0-32 Vac 4. 0-10 Vac
Ammeter	0-20 Amps
Power Supply	1. 0-36 Vdc 2. 0-10 Vac, 400 hz
Recorder	8 Channel
Gauges	1. 0-500 psig - 2 req.

4.0 TEST PROCEDURE

4.1 General

System testing consists of those tests necessary to demonstrate the performance of specific areas of concern within the subject valve design. Detail test instructions will outline procedures to allow evaluation and definitization of the following functions:
DEFINITION

1. Operational Verification - with all detail components assembled verify the operation of the valve assembly, i.e., cycle open to closed and closed to open.

2. Response Time - Measurement of time to open and time to close valve.

3. Response Repeatability - Measurement of difference between response time cycles.

4. RVDT Position Indication - Testing of valve position monitoring capability.

5. Logic Verification - Demonstration of electronic control logic circuits to perform as specified.

6. Power Loss Valve Response - Verification of valve closing with power loss.

7. Knetic Energy Absorbing Spring Test - Verification of spring action.

8. Brake Test - Demonstration of brake operation, including holding torque at both power levels.

4.2 Detail Components

Table 4-1 presents a list of the detail components to be used for system testing. Also included in the table is the Parker-Hannifin drawing number.

TABLE 4-1

COMPONENT LIST

Part No.	Description
5736122-102	Motor
-	Electronic Control
5736114-101	Planetary Gear Box
-	Linkage
5730025-101, 102	Valve Assembly
23501 Pickering	RVDT
Hunter Spring Corp	Negator Spring
5736119-1	K. E. Absorbing Spring

Prior to initiati-g system testing assemble the system components in ac.ordance with the assembly instructions included in document AIP 5739000.

4.3 Data Recovery

All pertinent parameters will be monitored and rocorded on a strip chart recorder. A complete set of data will be recorder for each test conducted thereby minimizing instrumentation time. Table 4-2 presents the critical parameters to be recorded and describes the significance of each.

TABLE 4-2

RECORDED PARAMETERS

Parameter	Range/Units	Significance
RVDT Signal Out, S1	0.05-0.15 Vac	This is a continuous signal of amplitude versus time. The two end points represent valve closed position (0.05 Vac) and open position (0.15 Vac). Valve response time can also be established.
Input Current	0-15 Amps	Motor current draw.
Motor Voltage	0-36 Vac	This represents the voltage supplied by the electronic control to the motor as measured across A to B inputs.
Brake Voltage	0-36 Vdc	Brake voltage will provide holding power information.
Battery Supply Voltage	24-36 Vdc	The required voltage design point for the motor was 24-30.5 Vdc and this output will verify application of input voltages of the required magnitudes.
Valve 1 inlet pressure Valve 2 inlet pressure	0-265 psia 0-265 psia	This represents an aerodynaamic torque loading for the system.

4.4 Test Instructions

Connect the system to a test setup as shown in Figure 1. Use test equipment called out on Table 3-1. Proceed with detail operational verification test as outlined in the subsequent procedure.

a. Apply 27Vdc battery input voltage.

b. Apply 6Vac 400 hz input voltage to RVDT.

NOTE

Valve is in the closed position and should not change.

c. Depress activate switch (S1).

d. Valves should actuate to full open position and remain there.

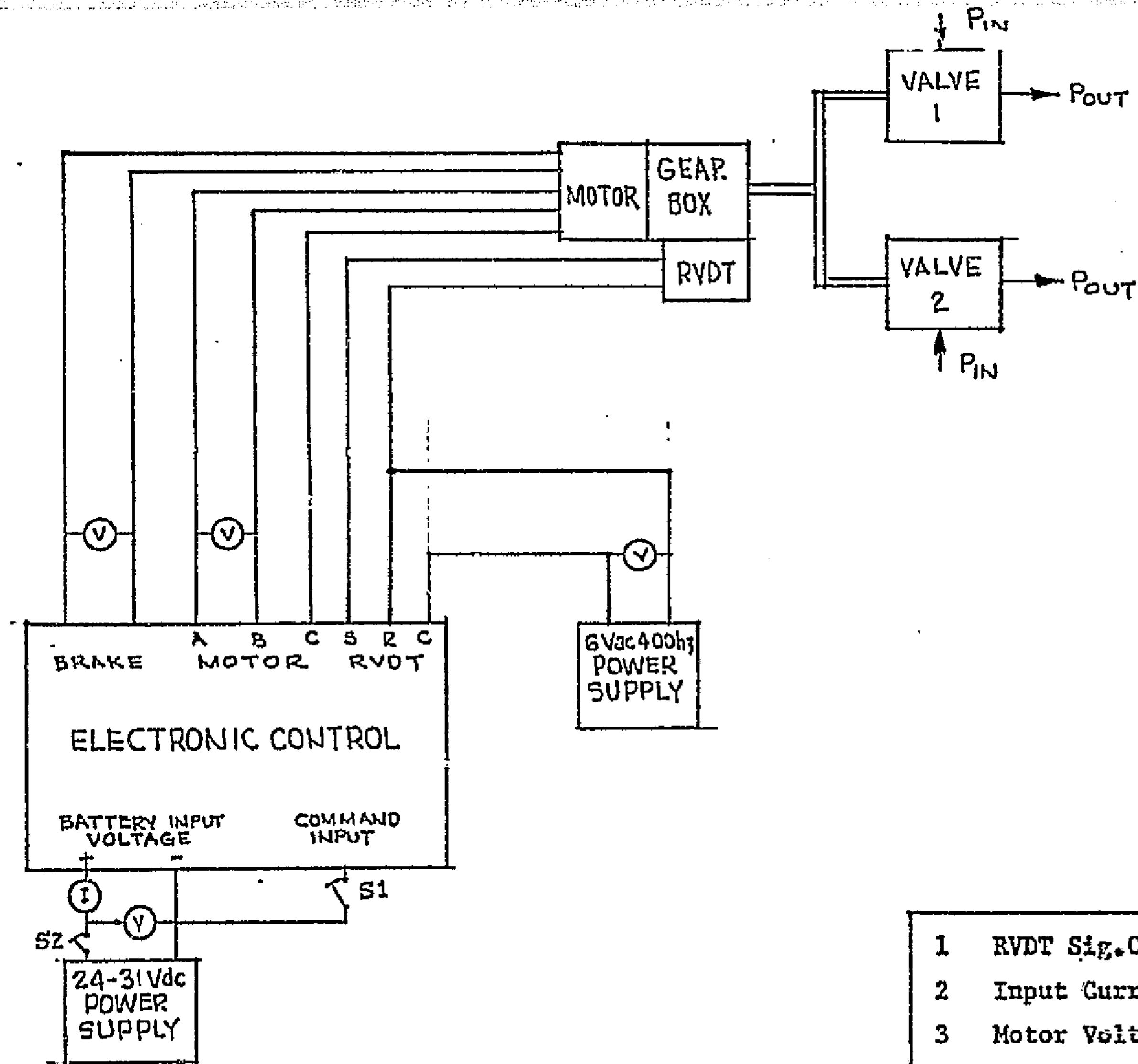
e. Depress activate switch (S1).

f. Valves should actuate to full closed position and remain there.

g. Apply 206 psia GN_2 to valve inlet port.

h. Depress activate switch (S1).

i. Valves should actuate to full open position and remain there.



1	RVDT Sig.Out 'S'
2	Input Current 'I'
3	Motor Voltage 'A-B'
4	Brake Voltage
5	Input DC Voltage
6	Valve 2 Pin
7	Valve 2 Pin
8	Channel Recorder

FIGURE 1 - TEST SETUP

NOTE

Observe that the valve assembly operated smoothly with no signs of binding.

j. Depress valve activate switch (S1).

k. Valve should actuate to full closed position and remain there.

l. Reduce valve inlet pressure to ambient.

m. Reduce battery input voltage to 24 Vdc.

n. Apply 265 psia GN_2 to valve inlets.

o. Depress valve activate switch (S1).

p. Valve should actuate to full open position and remain there.

q. Perform tests as listed in Table 4-3.

TABLE 4-3

ACTUATION TESTS

Test	Apply Battery Voltage, Vdc	Valve inlet Pressure, Psia	Cycles
1	24	265	1) 5 cycles, open to closed 2) Final cycle remove inlet power (open switch S2).
2	26	265	
3	28	265	
4	30.5	265	
5	32	265	
6	32	172	
7	30.5	172	
8	28	172	
9	26	172	
10	24	172	

r. Remove system power.

s. Disassemble system from test setup.

5.0 HARDWARE EVALUATION

At the end of all testing, disassembly the components as necessary to inspect for any operational wear or damage.

ENGINEERING ORDER

SYSTEMS DIVISION

FUNCTION	APPROVALS DATE	NAME print	DISTRIBUTION HARD COPY	MICRO PRINT
PRINCIPAL ENGR	18 Feb 74	V.D. dunn	1	
RESPONSIBLE ENGR	18 Feb 74	M. Cirillo	2	
RELIABILITY ENGR				
PLANNER	18 Feb 74	B. McCaa	2	
PROCESS ENGR				
QUALITY CONTROL				
CONTRACTS	18 Feb 74	J. Denny	4	
CHECK				
RELEASE	2-21-74	[illegible]	1	

DOCUMENT NUMBER: DVT 57390C

CODE IDENT	SIZE	SHEETS	REV LTR
92003	A	51	NC

EO SERIAL NO: ______ SHEETS: ______

EO PREPARED BY:

EO TYPE
- [x] INITIAL RELEASE
- [ ] INCORPORATED CHANGE
- [ ] NON-INCORPORATED CHANGE
- [ ] STOP ORDER
- [ ] MEMO
- [ ] ADVANCE MATERIAL ORDER

ECP NO.

SCN NO.

CHANGE CONTROL
- [ ] NOT APPLICABLE
- [ ] CLASS I
- [ ] CLASS II

TITLE

DESIGN VERIFICATION TEST PROCEDURE

REPLACES ______ REPLACED BY ______

REASON FOR CHANGE

REFERENCED SPECS:

USAGE REQUIREMENTS

PROG NO.	CUSTOMER	SPEC IDENT NO.	DWG CFG	REV	NEXT ASSEMBLY	END ITEM DWG NO.	EI EFFECTIVITY FIRST	EI EFFECTIVITY LAST	QTY NEXT ASSY	QTY EI
S/80										

PARTS STATUS (Planning Use Only)	
In Assy	
Stock	
In Process	
Purch	
Raw Matl	

10

SYSTEMS DIVISION
PARKER HANNIFIN ■ 18321 JAMBOREE BOULEVARD ■ IRVINE, CALIFORNIA 92664

CONTROLLED DOCUMENT

NUMBER: DVT5739001

TITLE: Design Verification
Test Procedure,
Orbiting Maneuvering Engine
Propellant Valve

RELEASE HISTORY							
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PREPARED BY: M. Cirilo
M. Cirilo

APPROVED BY: Vance B. Dunn
V. Dunn, Project Engineer

PHS 1831 (8-70)

LIST OF EFFECTIVE PAGES

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1.0 SCOPE

This development test procedure details the design verification testing to be performed on the Orbiting Maneuvering Engine (OME) Propellant Valve, Parker-Hannifin PN5739001, and its components. This plan covers Task IV under Phase II of Parker-Hannifins Program Plan, 1PMP5730018.

2.0 APPLICABLE DOCUMENTS

The documents listed below form a part of this document to the extent specified.

a. Military Specifications

MIL-P-27401B, Nitrogen, Propellant Pressurizing Agent

b. Military Standards

MSFC 237 Freon, Cleaning Solvent

c. Parker-Hannifin

ES5-13	Laminar Flow Clean Room Facility
ES5-15	Quality Control Procedures and Practices, Contamination Control Area and Components
QCPM610	Calibration Records and Controls
QCPM620	Calibration of Test Equipment
5739001	Ball Valve, Lifting (Prototype)
1PMP5730018	Program Plan, OMS Engine Propellant Valve Technology Program
EPS5736146	Process Specification, Ultrasonic Cleaning.

3.0 GENERAL REQUIREMENTS

3.1 Environmental - Unless otherwise specified all testing shall be conducted within the following environmental conditions:

Contamination Control:	ES5-13 (Clean Room)
Temperature:	70°F ± 5° F
Relative Humidity:	90 Percent or less
Barometric Pressure:	Local Atmosphere

3.2 Test Media - The following test fluids will be used:

Nitrogen in accordance with MIL-P-27401

Helium in accordance with MIL-P-27407

Distilled or de-ionized water

Isopropyl alcohol per TT-I-735

Freon per MSFC 237

3.3 Tolerances - Unless otherwise specified, the following tolerances apply to the application of test requirements.

Weight:	± 1 percent
Temperature:	± 5°F
Pressure:	To be measured on gages with range and accuracy necessary to assure design requirement compliance.
Flow Rate:	± 1 percent
Leakage Rate:	± 10 percent

3.4 Test Facilities - All tests shall be conducted at the Parker, Irvine, facility. Should the necessity arise for additional test facilities, these facilities shall be selected from Parker-approved test laboratories. All testing shall be under the supervision of Parker project engineering personnel.

3.5 Failure Reporting - Failure is defined as the inability of the valve to meet specified performance requirements of this procedure. In the event of a failure or out-of-tolerance condition, it shall be the responsibility of Parker test engineering to concurrently and immediately notify Project Engineering. The test setup, equipment and test specimen shall be examined in order to provide the basis for subsequent failure verification and assurance that an actual valve failure has occurred.

3.6 Test Equipment - Test equipment shown in test Figures (or equivalent) shall be used. (The percent shown is maximum percent of full scale accuracy). Test equipment shall be capable of producing, maintaining and indicating the specified test conditions. Applicable instruments shall be subjected to periodic calibration in accordance with QCPM610 and QCPM620 and shall bear a current calibration decal at time of implementation.

3.7 Testing

3.7.1 Component Testing - The basic concepts of the valve will be tested at the valve component level. Tools and fixtures will be used to isolate failure modes for the purpose of establishing the optimum conditions which will prevent failures in the valve. Component tests are divided into three main catagories - (1) seat tests, (2) torque tests and (3) visor motion or linkage tests. Individual tests are included in Appendix B.

3.7.2 Valve Tests - Testing of the complete valve shall be undertaken after component testing is in progress or has been completed. The sequence below will be followed to the extent permitted by hardware availability and success encountered.

3.7.3 Test Sequence - Testing will follow the sequence below unless otherwise directed by Project Engineering.

TEST	Paragraph	Component	VALVE	
			SN01	SN02
Cleaning & Particle Count Parts	4.1.4	1	1	1
Cleaning & Particle Count Valve	4.1.5	-	2	2
Valve Seat Tests	4.2.1.1	~~2~~	2½	-
Torque Measurement	4.2.1.2	3	-	-
Linkage	4.2.1.3	4	-	-
Proof Test	4.2.2.1	-	3	3
Torque Tests	4.2.2.2	-	4	4
External Leakage Test	4.2.2.3	-	5	5
Internal Leakage Test	4.2.2.4	-	6	6
Flow - ΔP Tests	4.2.2.5	-	7	7
Temperature Leakage Tests	4.2.2.6	5	8	-
Vibration Test	4.2.2.7	-	-	8
Life Test	4.2.2.8	-	-	9
Hardware Evaluation	5.0	6		

3.8 Photographs - Photographs of good resolution shall be made of each valve test setup for inclusion in the test report.

3.9 Test Results Data - Complete test results data shall be recorded for each test using reproducible copies of the data sheets provided in Appendix A. Where appropriate, test results data shall be recorded as exact observed values. Entries on data sheets shall be made clearly and legibly.

3.10 Disposition of Data - At the conclusion of testing, the original data sheets shall be retained by Parker project engineering.

3.11 Test Report - A final test report shall be prepared by engineering. The report shall be complete within itself. The test report shall include, as a minimum, the following information:

a. Introduction, purpose, summary and conclusion.

b. Detailed report of each test.

c. Exact test methods and results.

d. Detail schematic and/or photograph of each test setup with a list of all test equipment and instrumentation used, the manufacturer's model number and accuracy. The serial numbers of equipment which are critical for repeating tests and/or calibration shall also be logged.

e. Data sheets recording all raw data as actual values.

f. Traceability to test equipment, operators and witnesses.

g. Location and time of test performance.

h. Required traces to verify levels for all vibration tests.

i. Detailed parts list containing part number, drawing revision letter and lot number for each part in the test article.

3.12 Deviations and/or Variations - Any deviations and/or variations from the procedures specified herein shall require approval of Parker project engineering prior to testing. Any changes to this procedure after its approval may be initiated in the form of an amendment, change order or revision to the procedure.

3.13 Disposition of Test Specimen - Test specimen SN01 and 02 shall be placed in bonded stores at Parker-Hannifin for system testing which shall follow.

3.14 Success Criteria - The success criteria for the evaluation program shall be the capability of the valve to withstand all environments imposed upon it and meet all post-environment functional checks.

4.0 ASSEMBLY AND TEST

4.1 ASSEMBLY PROCEDURE

4.1.1 Contamination Control - the following procedure shall be used for contamination control during full or partial assembly of the OME valve or its componants. All assembly or disassembly will be done in the clean room environment.

4.1.2 Parts Inspection - Inspect all parts to be assembled, under 20 X magnification for loose burrs, burrs in corners, passages, edges, etc., for contamination and for nicks and other damage.

4.1.3 Preclean - Preclean all parts, tools and fixtures as follows:

(a)* Sonic clean in Turco 3878, minimum of five (5) minutes.
(b) Rinse in de-ionized water.
(c) Clean in Turco 4215.
(d) Rinse in de-ionized water.
(e)* Sonic clean in isopropyl alcohol, minimum of five (5) minutes.
(f) Blacklite inspect parts.
(g) Place in clean poly protective container.

4.1.4 Clean and Particle Count (Parts and Tools).

(a)* Ultrasonic clean parts, tooling and fixtures for approximately five (5) minutes.
(b) Wash in alcohol.
(c) Flow 500 ml of freon over hardware and particle count per ARP598. Particle count for a 500 ml sample shall not exceed the limits below. Use visual count unless automatic counter is specifically authorized by Engineering.

PARTICLE SIZE	NO. OF PARTICLES
0-25 µ	1000
26-50 µ	50
51-100 µ	5 (No metal particles)
~~101-200 µ~~	~~1 (No metal particles)~~
7-200 µ	0

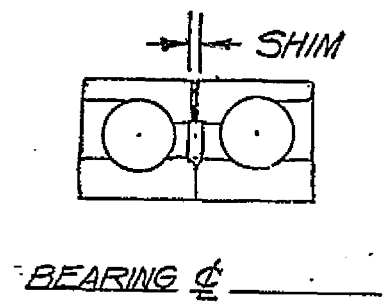
* Clean per EPS5736146 as applicable.

(d) Vacuum oven dry parts at 150°F at 25" Hg minimum for thirty (30) minutes, minimum.

4.1.5 Valve Assembly - All parts, tools and fixtures shall be cleaned, particle counted and dried per para. 4.1.4 and all valve assembly and dis-assembly shall be done in a cleanroom environment. An assembly procedure shall be developed, incorporating any special techniques, tools and changes to the drawing 5739001.

(a) Examine each part under 20X magnification for burrs, contamination, finish on bearing surfaces and other damage or conditions which can affect performance.

(b) Ball Bearing Installation - Install the duplex bearings back to back (DB) as shown in assembly print PN 5739001. Place a TBD* inch shim between the two bearing outer races only as shown in sketch. (Ball bearings are to be protected from solvents and water by the use of Tool No. F59-0-316 when cleaning the valve and by use of light lubricants during normal use to prevent corrosion).



(c) Install the two shaft seals per Drawing 5739001.

(d) Visor Installation - Install the visor on the shaft prior to shaft ~~prior to shaft~~ installation into housing. Measure the tension of the Belleville spring, PN 5736093, on the visor in the assembled position.

(e) Shaft Installation - Install the shaft-visor-link assembly making sure that the link fits on the 5736090 pin. Install the 5736081 lever and apply 30 to 40 in-lb of torque to the shaft nut.

* To be determined at Test Series 180-200.

(f) Cover Installation - Install the 5736075 Cover. Check the shaft and visor for freedom of movement, and for vision orientation. Measure the shaft angle rotation.

(g) Install the seat assembly and the F65-0-2233 Det -3 outlet fitting. Measure the shaft angle rotation.

4.1.6 Cleaning and Particle Count - Install the valve as shown in Figure 1. Open and close the valve while flowing freon at 100 psi through the valve. Catch a 500 ml sample and particle count per ARP598. Use the limits in para. 4.1.4 (c). In cycling the valve DO NOT exert a torque greater than 50 lb-in at the end of the opening or the closing stroke.

4.2 TEST PROCEDURE

4.2.1 COMPONENT TESTS

4.2.1.1 Seat Tests - Seat testing at the component level shall be per Appendix B, Test Number (TN) 180-100 series. The object of this testing is to develop alternative seat configurations, to evaluate the effect of valve variables on seat performance, and to be able to isolate valve variables for evaluation and development purposes.

4.2.1.2 Torque Tests - Torque tests will be conducted to establish optimum bearing preload, to determine bearing friction loads and to establish a reference design level to evaluate performance changes. TN180-200 series of Appendix B cover torque testing.

4.2.1.3 Linkage Tests - Linkage and relative motion of the shaft, visor and linkage are evaluated together with the torque test in TN180-200 series of Appendix B.

4.2.2 VALVE TESTS

4.2.2.1 Proof Pressure Test

4.2.2.1. Procedure - Connect the OME valve as shown in Figure 1. Open the visor and pressurizing to 435 ± 10 psig. Hold pressure for five (5) minutes, minimum. Then close the visor, and loosen one of the down stream AN929 caps to bleed the downstream pressure and remove the AN929 cap. Hold inlet pressure five (5) minutes, minimum. Actuate the visor after depressurization.

4.2.2.1.1 Requirements - There shall be no visual evidence of cracks, warpage or any other pressure induced damage nor impairment of movement of the shaft and visor. Record data in Data Sheet A-1.

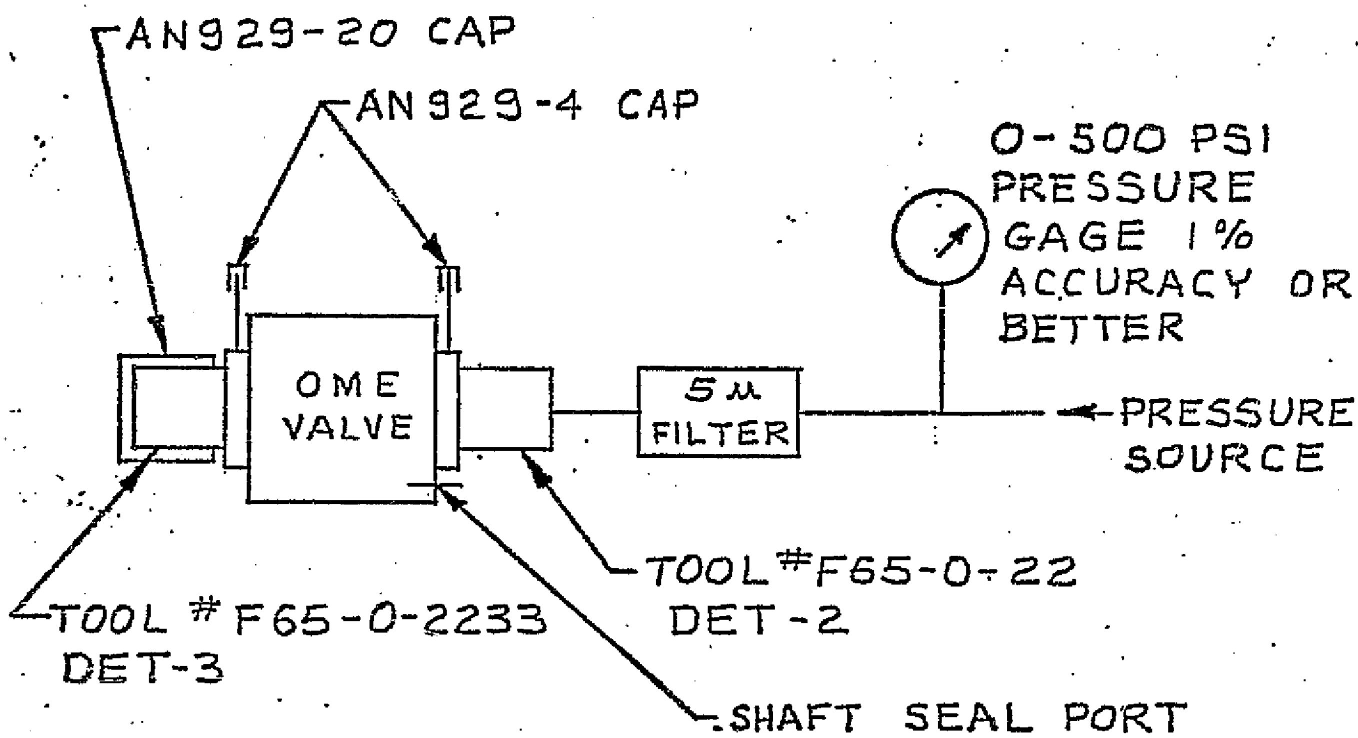


FIGURE 1

PROOF PRESSURE

4.2.2.2 Torque Measurement

4.2.2.2.1 Procedure - With the valve depressurized, make an X-Y plot of shaft torque VS positon. Make a complete hysteresis curve. Pressurize OME valve with 290 psig of freon per Figure 2, at the start of the test, and actuate the visor three (3) times.

4.2.2.2.2 Requirement - Compare torquehysteresis curves with data obtained during component tests. Evaluate and investigate any changes to the reference criteria. Record data in Data Sheet A-2.

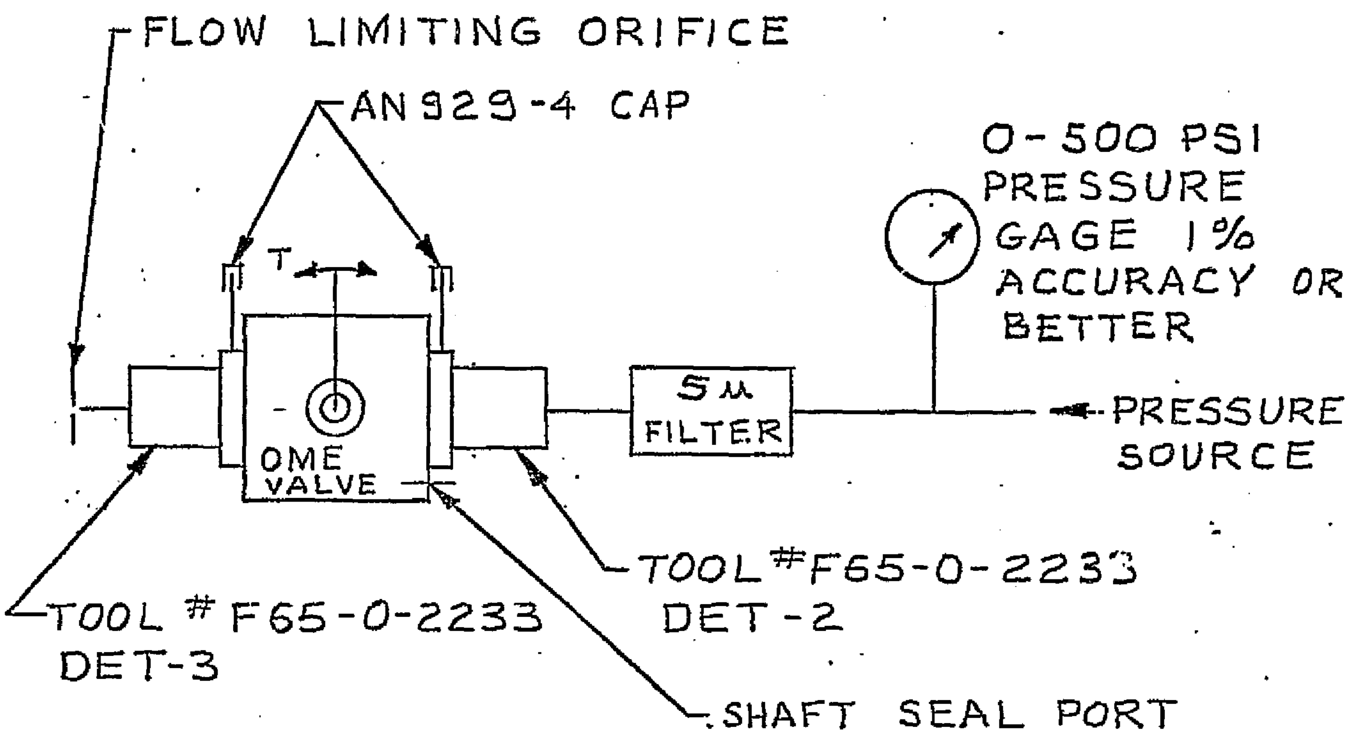


FIGURE 2
TORQUE MEASUREMENT

4.2.2.3 External Leakage

4.2.2.3.1 Procedure

(a) Connect the OME valve per Figure 3A. Open the visor partially and pressurize the valve with 300 ± 10 psig of helium. Hold the pressure for a minimum of five (5) minutes before making a reading and until the mass spectrometer detector stabilizes. Record the body leakage and then reduce the pressure to 5 psig and repeat the above procedure. Make sure fittings and fixture seals are not leaking during test.

(b) Connect the OME valve per Figure 3B. The visor will be partially open. Measure leakage out of shaft seal #1 by repeating the procedure above, except use N_2 leakage measurement.

(c) Connect the OME valve per Figure 3C. The visor will be partially open. Measure leakage out of shaft seal #2 by repeating procedure (a) above, except use N_2 leakage measurement.

4.2.2.3.1 Requirement - Record body leakage, shaft seal port leakage and shaft seal (out of tool No. F59-0-316) both at high and at low pressures. Helium leakage shall not exceed 1×10^{-4} cc/sec and nitrogen leakage shall not exceed 10 scc/hr. Record data in Data Sheet A-3.

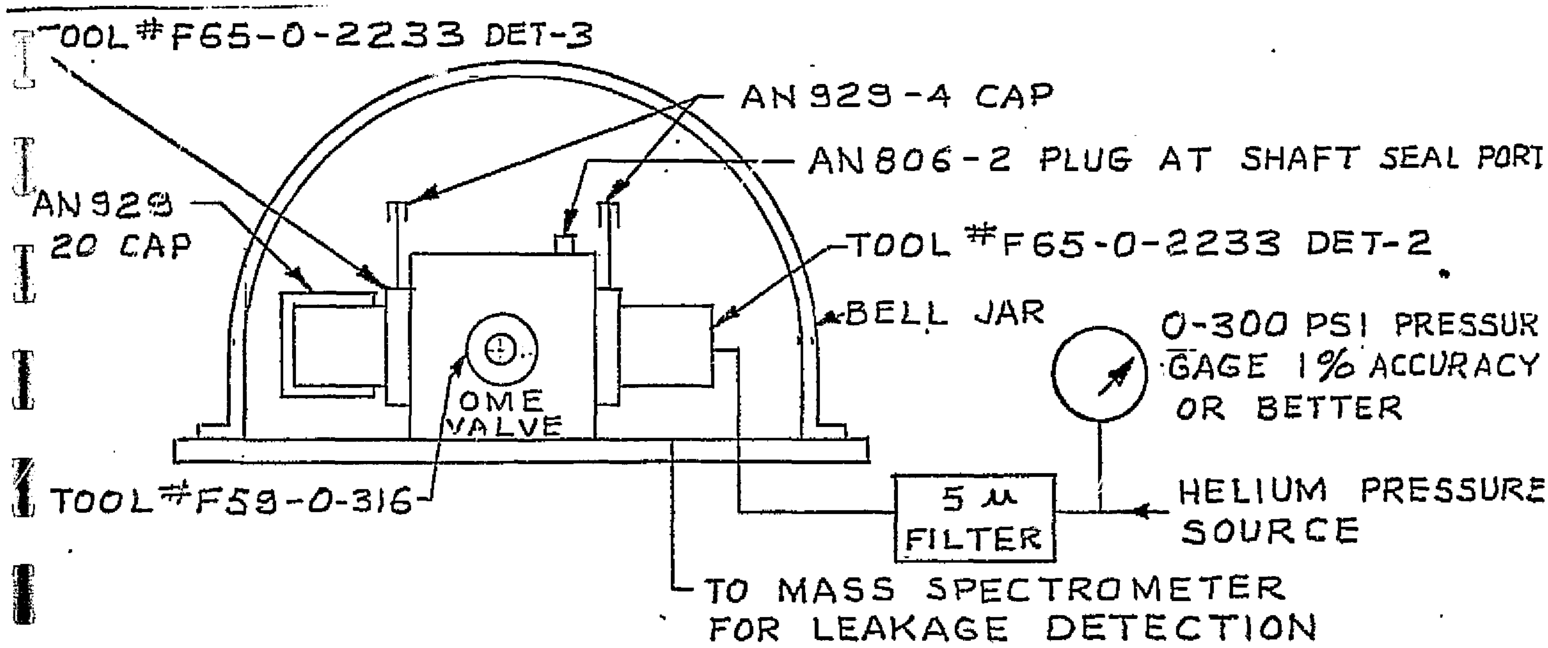


FIGURE 3A
BODY LEAKAGE

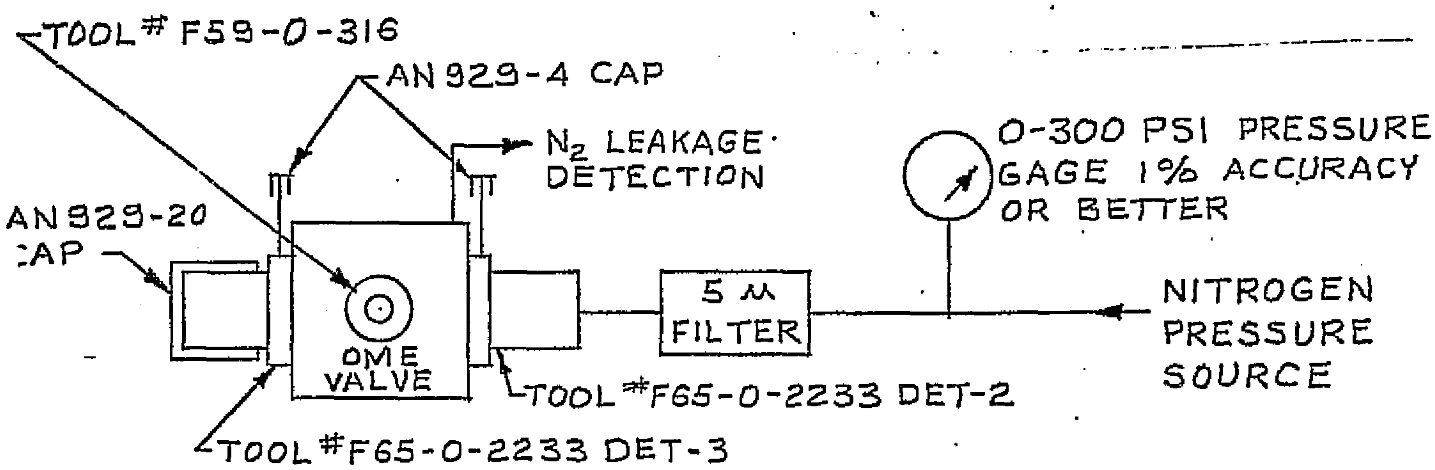


FIGURE 3B
#1 SHAFT SEAL LEAKAGE

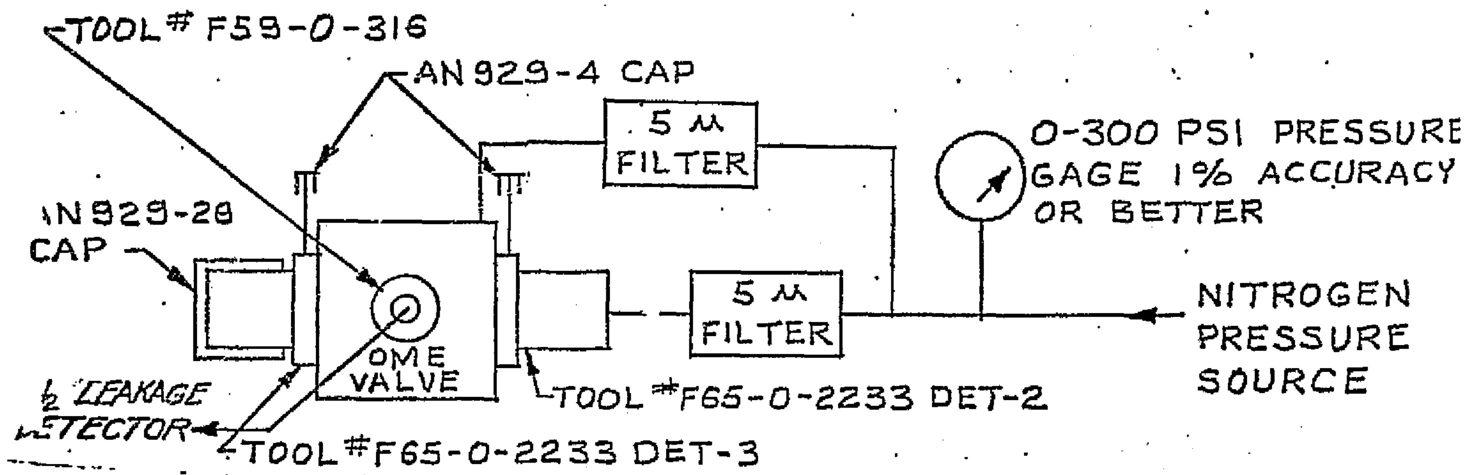


FIGURE 3C
#2 SHAFT SEAL LEAKAGE

4.2.2.4 Internal Leakage

4.2.2.4.1 Procedure - Connect the OME valve per Figure 4. Pressurize the closed valve to 265 ± 5 psig. Allow to stabilize for five (5) minutes, minimum, then measure leakage at outlet port while maintaing a torque of *TBD in-lb on the shaft, clockwise. Reduce the inlet pressure to 5 spig and repeat the leakage test, while maintaining the clockwise torque on the shaft.

4.2.2.4.2 Requirements - Record leakage and any test condition which influenced leakage rates. Leakage shall not exceed 10 scc/hr of GN_2.

* To be determined at component testing test series 180-100.

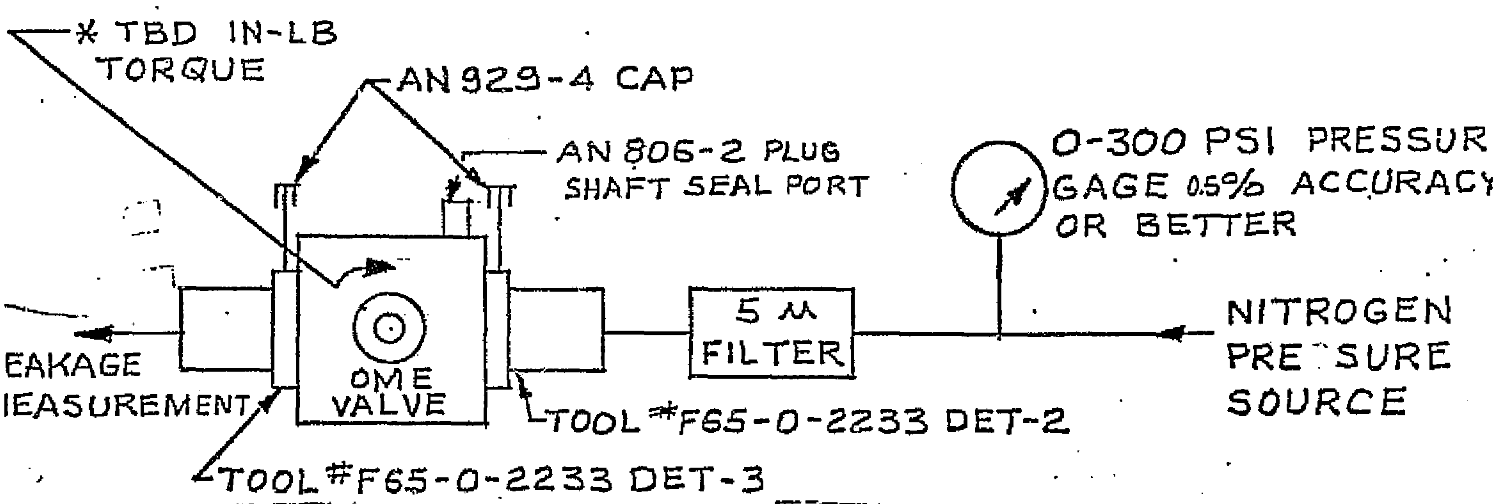


FIGURE 4

INTERNAL LEAKAGE

4.2.2.5 FLOW-ΔP TEST

4.2.2.5.1 Procedure

(a) Single Valve - Connect the OME valve as shown in Figure 5A. Flow 72 gpm of water with an inlet pressure of 290 psig. A flow limiting orifice shall control the water flow. Repeat the test three (3) times continuously recording ΔP, Pinlet, and shaft torque as a funtion of shaft angular position through3 complete cycle s.

(b) Series Valves - Connect the valves as shown in Figure 5B. Repeat procedure (a) above, except that the two valves will be actuated simultaneously.

4.2.2.5.2 Requirements - The presure drop shall be 5 psid, maximum. Record data in Data Sheet A-6.

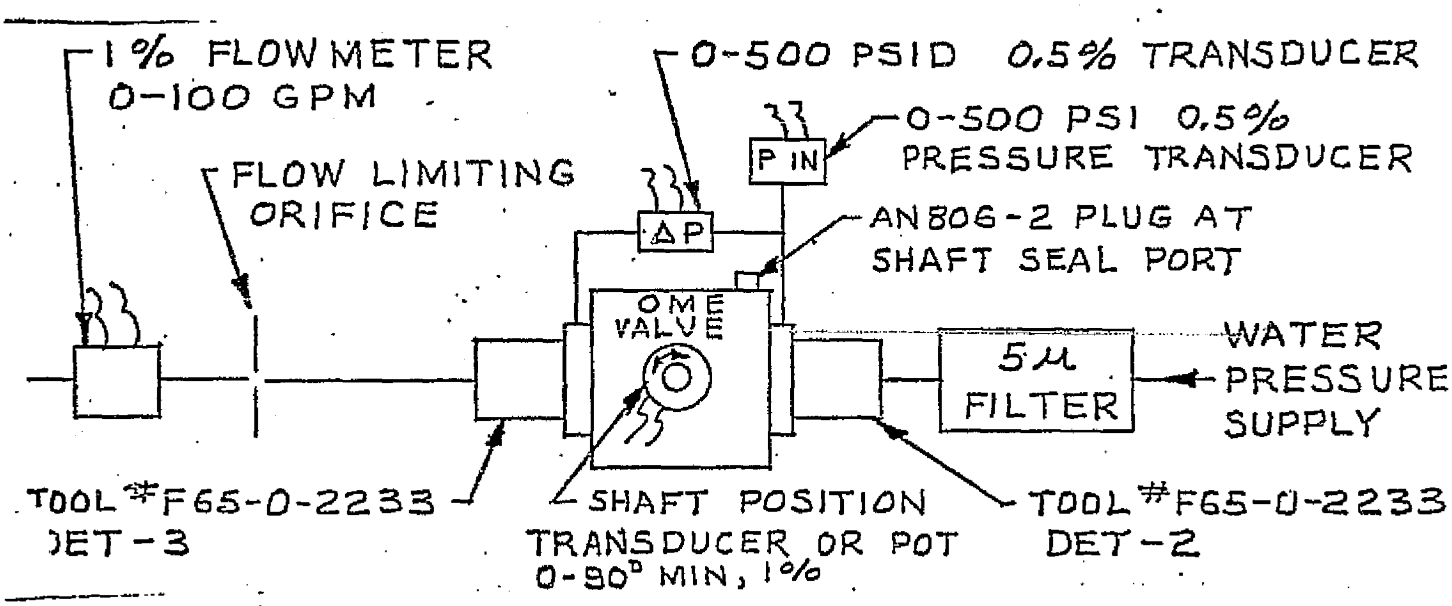


FIGURE 5 A
SINGLE VALVE

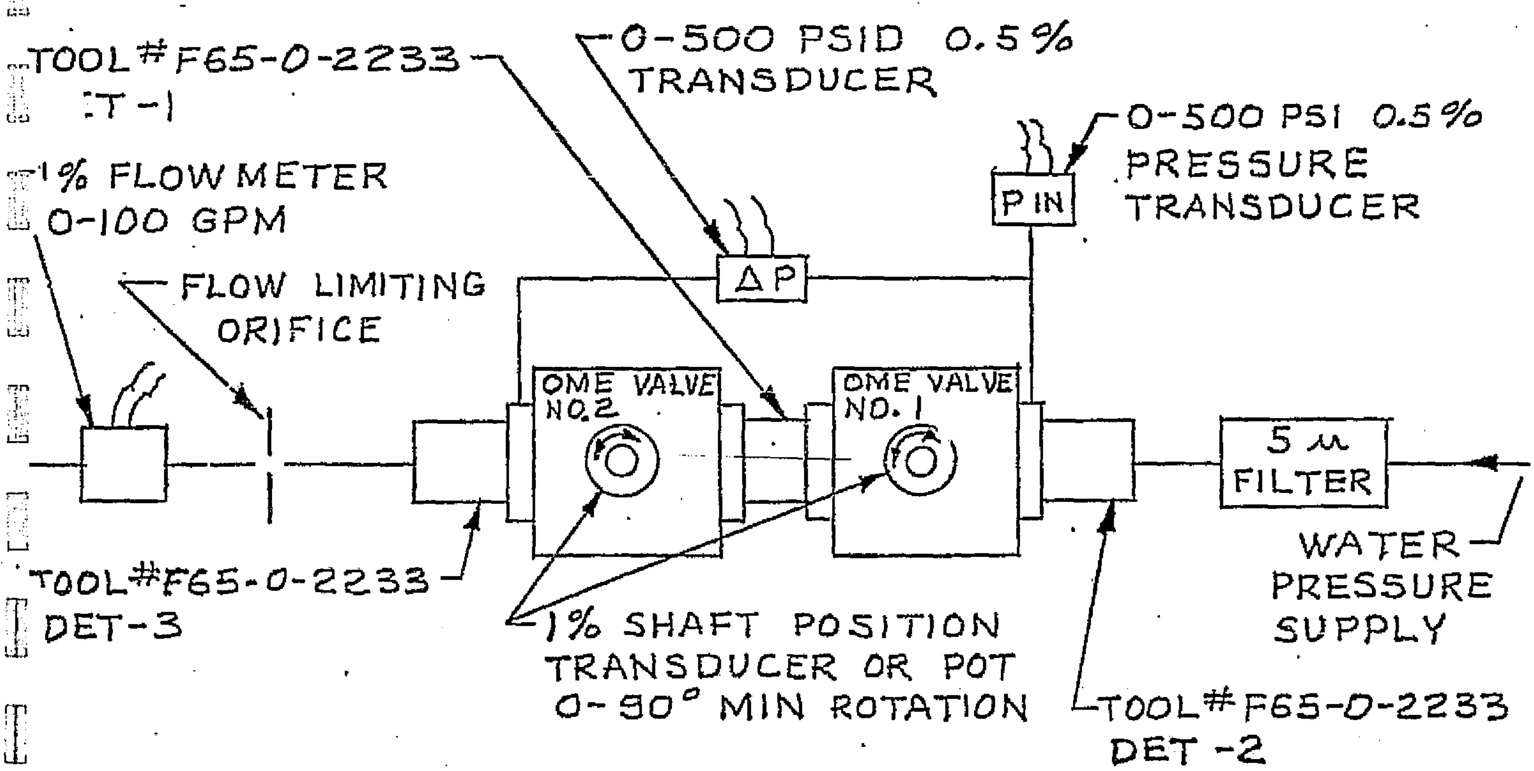


FIGURE 5 B
SERIES VALVE
FLOW VS ΔP TESTS

4.2.2.6 Temperature Internal Leakage

4.2.2.6.1 Procedure - (a) Connect the OME valve per Figure 6, and condition the setup for 2 hours, min. or until it stabilizes at 40°F. Pressurize the closed valve to 265 ± 5 psig. . Allow to stabilize for five (5) minutes, minimum, then measure leakage at outlet port while maintaining a torque of *TBD in-lb on the shaft, clockwise. Reduce the inlet pressure to 5 psig and repeat the leakage test, while maintaining the clockwise torque on the shaft. (b) Condition the setup as above, except at 200°F and repeat the internal leak tests.

4.2.2.6.2 Requirements - Record leakage and any test condition which influenced leakage rates. Leakage shall not exceed 10 scc/hr of GN_2.

* To be determined at component testing test series 180-100.

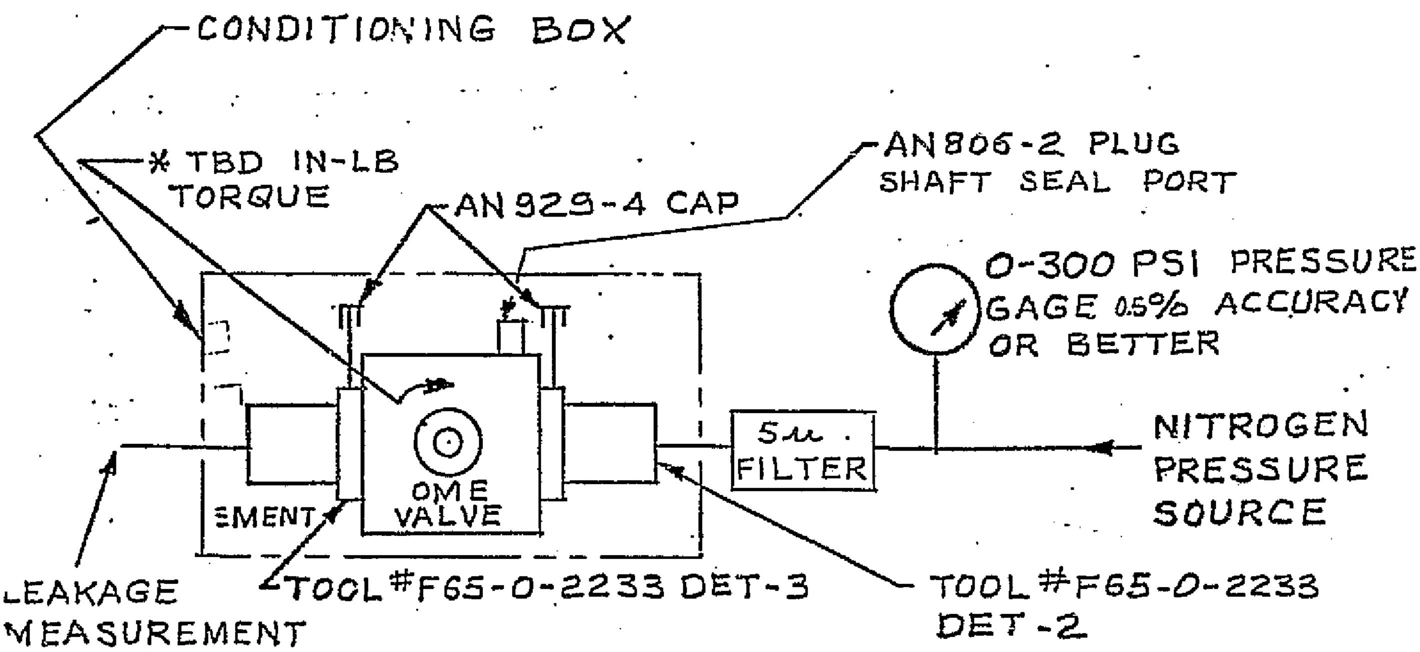


FIGURE 6
TEMPERATURE INTERNAL LEAKAGE

4.2.2.7 VIBRATION

4.2.2.7.1 Procedure - Mount the valve on a vibration fixture. Position the driving accelerometer on the valve body, a monitor on the end of the shaft, one on the valve outlet port and one on the test fixture (See Fig. 7).

Random vibrate for five (5) minutes on each of the three (3) orthogonal axes at the following levels when the valve is unpressurized:

20 to 286 Hz	$0.025g^2/Hz$
286 to 700 Hz	6 db/octive rise
700 to 2000 Hz	$0.15g^2/Hz$

Perform the following after each axis is tested:

(1) Torque test per 4.2.2.2
(2) Leak test per 4.2.2.4
(3) Visual examination for effects of seat and on other parts (without disassembling)

4.2.2.7.2 Requirements

(a) Torque per 4.2.2.2.2
(b) Leakage per 4.2.2.4.2

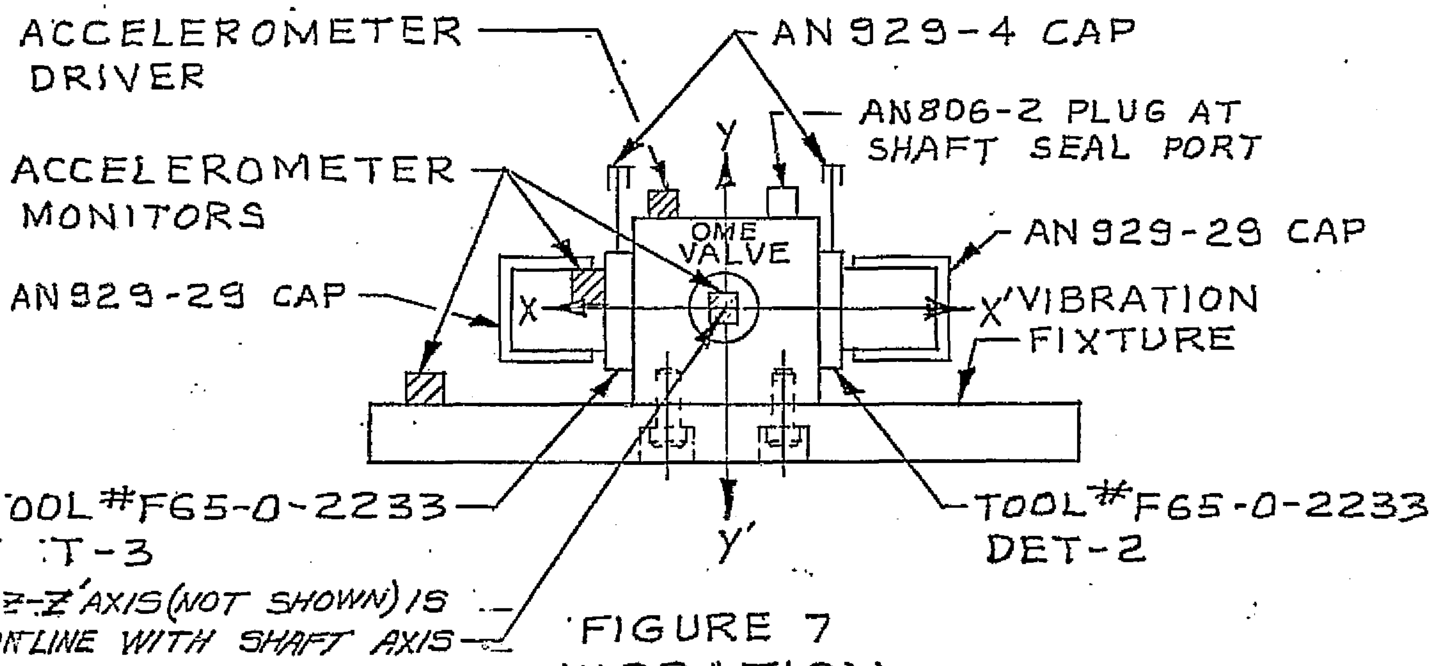


FIGURE 7
VIBRATION

4.2.2.8 LIFE TEST

4.2.2.8.1 Procedure - Cycle the OME valve open and closed at the rate of 1 to 2 cycles per second. Cycle life of 6,000 dry cycles and 4,000 wet cycles shall be performed for a total of 10,000 cycles. Two duty cycles shall be used. Install the valve as shown in Figure 6. The object of the downstream filter is to examine for particles which may indicate wear in the valve. This filter shall be examined after each cycle period. For the first cycle, performance (c), (d) and (e) after (a) and before (b). The first four cycle periods shall consist of the following operation in sequence:

(a) Cycle 300 times dry with no pressure.
(b) Cycle 200 times with water at 250 psig per Fig. 5.
(c) Torque tests per 4.2.2.2
(d) Valve outlet filter operation
(e) Leak test per 4.2.2.4.

If there is no change in (c), (d) and (e), above for the 3rd and 4th cycle periods, increase (a) from 300 cycles to 600 cycles and (b) from 200 cycles to 400 cycles per cycle period. Continue at the 600 and 400 cycle level until there are significant changes in the torque per para (c) above or in the filter contamination per para (d) above and (e). Record finds in Data Sheet.

4.2.2.8.2 Requirements - Torque changes shall comply to 4.2.2.2.2 and leakage shall comply to 4.2.2.4.2.

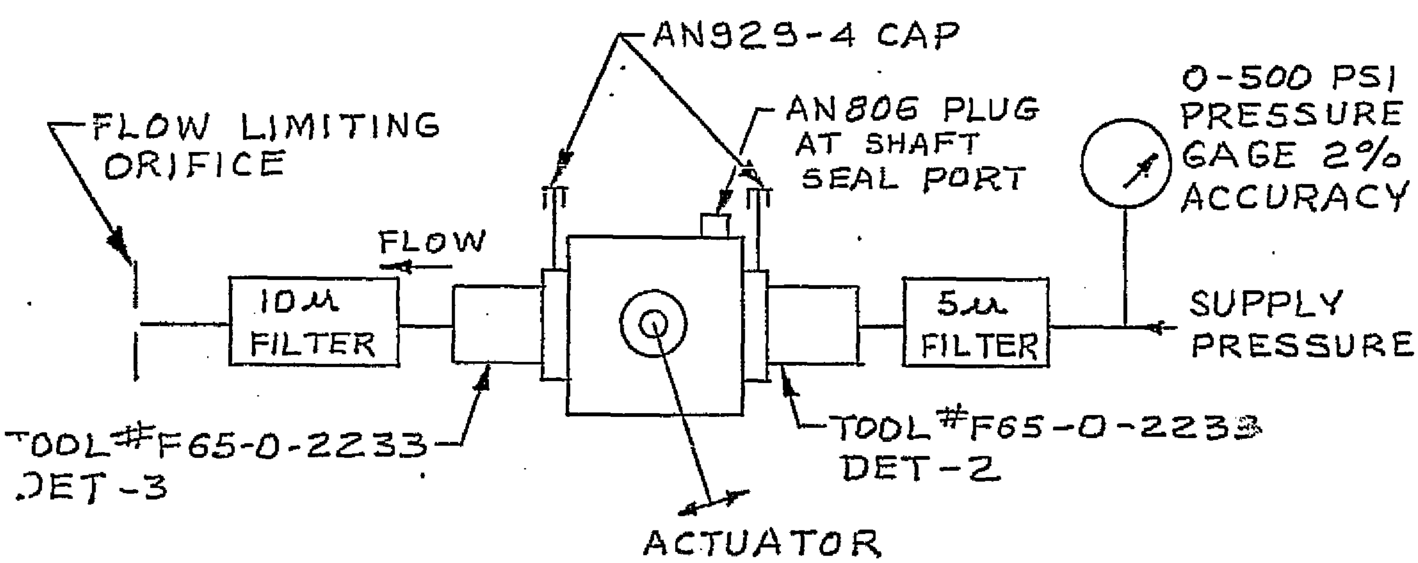


FIGURE 8
CYCLE LIFE

5.0 HARDWARE EVALUATION

Disassemble the valves or components as required to analyze for wear, bending, warping and for any other visual damage.

APPENDIX A
TEST DATA SHEETS

SYSTEMS DIVISION
PARKER HANNIFIN

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REV LTR	NC				
DATE	2-8-74				

OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO. ______ DATE. ______

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
	Proof Pressure Valve Open	No visual distortion damage, leakage, or shaft-visor freedom of motion.		
.2.1	Proof Pressure Valve Closed			

Test Comments: ______

SYSTEMS DIVISION
PARKER HANNIFIN

NO. DVT5739001	BY MC	PAGE A-3			
REV LTR	NC				
DATE	2-8-74				

OME VALVE DEVELOPMENT TEST DATA SHEET

P/N 5739001 SN ____________ TEST DATE ____________

TEST COMMENTS: ____________

PARA	TEST	TEST REQUIREMENTS	TEST DATA	TECH.
	Torque Measurement, Depressurized	Hysteresis Curve #1		
		Hysteresis Curve #2		
4.2.2.2		Hysteresis Curve #3		
	Torque Measurement Pressurized to 290 psig	Hysteresis Curve #1		
		Hysteresis Curve #2		
		Hysteresis Curve #3		

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SYSTEMS DIVISION
PARKER HANNIFIN

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DATE	2-8-74			

OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO. ______________ DATE. ______________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
	External leakage, body only Fig. 3A	300 ± 10 psig 1×10^{-4} scc of helium per second, max.	______ scc/sec	
		5 psig 1×10^{-4} scc of helium per second, max.	______ scc/sec	
2.2.3	External leakage, Shaft sec. #1 Fig. 3B	300 ± 10 psig of GN_2 10 scc/hr. max.	______ scc/hr	
		5 psig of GN_2 10 scc/hr. max.	______ scc/hr	
	External leakage, shaft sec #2 Fig. 3C	300 ± 10 psig of GN_2 10 scc/hr. max.	______ scc/hr	
		5 psig of GN_2 10 scc/hr. max.	______ scc/hr	

Test Comments ____

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OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO. __________ DATE. __________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
4 2.2.4	Internal Leakage	265 ± 5 psig inlet pressure leakage shall be 10 scc/hr. maximum	________ scc/hr	
		5 psig inlet pressure leakage shall be 10 scc/hr, maximum	________ scc/hr	

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OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO.__________ DATE.__________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
5	FLOW-ΔP Single Valve	There shall be a ΔP = 5 psid when 72 gpm of water flow through the valve per Figure 5A.	Test #1 ΔP=____psid	
			Test #2 ΔP=____psid	
			Test #3 ΔP=____psid	
	FLOW-ΔP Series Valves	There shall be a ΔP = 5 psid when 72 gpm of water flow through the valves per Figure 5B.	Test #1 ΔP=____psid	
			Test #2 ΔP=____psid	
			Test #3 ΔP=____psid	

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SYSTEMS DIVISION
PARKER HANNIFIN

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OME VALVE DEVELOPMENT TEST DATA SHEET

P/N 5739001 SN ______________ TEST DATE ______________

TEST COMMENTS: ______________

PARA	TEST	TEST REQUIREMENTS	TEST DATA	TECH.
4.2.2.6	Temp. Internal Leakage 40°	265 ± 5 psig inlet pressure. Leakage shall be 10 scc/hr. maximum Shaft Torque TBD* in-lb	______ scc/hr	
		5 psig inlet pressure. Leakage shall be 10 scc/hr. maximum Shaft Torque TBD* in-lb	______ scc/hr	
	Temp. Internal Leakage 200°F	265 ± 5 psig inlet pressure. Leakage shall be 10 scc/hr. maximum Shaft Torque TBD* in-lb	______ scc/hr	
		5 psig inlet pressure. Leakage shall be 10 scc/hr. maximum Shaft Torque TBD* in-lb	______ scc/hr	

* To be determined at Test Series 180-100.

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OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO. ____________ DATE. ____________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
4. .2.7	Vibration X Axis axis for	Vibrate each of the three (3) mutually perpendicular axis for five (5) minutes per the following levels : 20 to 286 Hz .0259^2/Hz 286 to 700 Hz 6db/oct. rise 700 to 2000Hz .15g^2/Hz	______in-lb torque dry ______in-lb torque 250psig ______scc/hr at 5 psig ______scc/hr at 275 psig	
	Vibration Y Axis	There shall be no significant change in torque requirement nor in seal leakage	______in-lb torque dry ______in-lb torque 250 psig ______scc/hr at 5 psig ______scc/hr at 275 psig	
	Vibration Z Axis		______in-lb torque dry ______in-lb torque 250 psig ______scc/hr at 5 psig ______scc/hr at 275 psig	

Test Comments____

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.OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO.__________ DATE.__________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
	Cycle Life / Cycle No. 1	300 dry cycles 200 wet cycles at 250 psig torque per 4.2.2.2 Leakage per 4.2.2.4	______ dry cycles ______ wet cycles ______ in-lb torque dry ______ in-lb torque at 250 psig ______ scc/hr at 5 psig ______ scc/hr at 275 psig	
4.2.2.8	Cycle Life Cycle No. 2		______ dry cycles ______ wet cycles ______ in-lb torque dry ______ in-lb torque at 250 psig ______ scc/hr at 5 psig ______ scc/hr at 275 psig	
	Cycle No. 3		______ dry cycles ______ wet cycles ______ in-lb torque dry ______ in-lb torque at 250 psig ______ scc/hr at 5 psig ______ scc/hr at 275 psig	

Test Comments____________________

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DATE	2-8-74				

OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO.________ DATE.________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
	Cycle No. 4	300 Dry Cycles 200 Wet Cycles at 250 psig Torque per 4.2.2.2 Leakage per 4.2.2.4	_____ dry cycles _____ wet cycles _____ in-lb torque dry _____ in-lb torque at 250 psig _____ scc/hr at 5 psig _____ scc/hr at 275 psig	
4. .2.8	Cycle Life Cycle No. 5	_____ dry cycles _____ wet cycles	_____ dry cycles _____ wet cycles _____ in-lb torque dry _____ in-lb torque at 250 psi _____ scc/hr at 5 psig _____ scc/hr at 275 psig	
	Cycle No. 6	_____ dry cycles _____ wet cycles	_____ dry cycles _____ wet cycles _____ in-lb torque dry _____ in-lb torque at 250 psi _____ scc/hr at 5 psig _____ scc/hr at 275 psig	

T st Comments -

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NO. DVT5739001	BY MC	PAGE A-			
REV LTR	NC				
DATE	2-8-74				

OME DEVELOPMENT TEST DATA SHEET

PART NO. 5739001 VALVE NO. ________ DATE. ________

PARA	TEST	REQUIREMENTS	ACTUAL	TECH.
	Cycle No. ____	____ dry cycles ____ wet cycles	____ dry cycles ____ wet cycles ____ in-lb torque dry ____ in-lb torque at 250 psig ____ scc/hr at 5 psig ____ scc/hr at 275 psig	
4.2.2.8	Cycle No. ____	____ dry cycles ____ wet cycles	____ dry cycles ____ wet cycles ____ in-lb torque dry ____ in-lb torque at 250 psig ____ scc/hr at 5 psig ____ scc/hr at 275 psig	
	Cycle No. ____	____ dry cycles ____ wet cycles	____ dry cycles ____ wet cycles ____ in-lb-torque dry ____ in-lb torque at 250 psig ____ scc/hr at 5 psig ____ scc/hr at 275 psig	

Test Comments ________

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REV LTR	NC				
DATE	2-8-74				

APPENDIX B
COMPONENT TESTS

180-100 Seat Tests, Swaged
180-110 Seat Tests, Floating, Helical Spring
180-120 Seat Tests, Floating, Belleville Spring
180-200 Torque Test

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NO. 180-100 BY MC PAGE 1

REV LTR.	NC				
DATE	2-7-74				

180 DEVELOPMENT TEST PROGRAM

TEST ITEM: PN5736067 SEAT ASSY, SWEGED

T. V.	TEST OBJECTIVE	TEST DESCRIPTION
101	Test seat integrity with perfect, floating ball	Use test fixture to test for: (1) Pre-load vs leakage (2) Leakage repeatability (3) Seat deformation (4) Leakage vs Temp. (5) Seat movement vs temp & pressure
102	Same as 101 except PN5736061, swivel visor, floating	Test the surface of the swivel visor
103	Same as 102 except PN5736061 swivel visor will be restrained and allowed to swivel	Test the ability of the swivel to correct misalignment.
104	Same as 101 except PN5736059 visor will be used to test fixed seal & fixed visor	Visor will be tested in a free position and constrained and loaded as required in 101 to form seal.

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PARKER HANNIFIN

NO. 180-101 BY MC PAGE 2

REV LTR	NC				
DATE	1-3-74				

DEVELOPMENT TEST

TEST: Seat, Swaged, PN5736067SK

OBJECTIVE: To establish the integrity of the 5736067 seat assembly when a two inch diameter, perfectly shaped, floating ball is used as a function of seat finish, after lap clearance, pressure level, forming conditions, seat movement.

TEST HARDWARE: 5736067SK Seat Assem., Swaged
F65-0-2218 Test Fixture (all parts must be particle free)

PROCEDURE:

1. Inspect surface finish of seat, the 1.033±.003 dia, .043±.002 dim. & .0024±.002 dim. for actuals. Record.
2. Assemble the 5736067SK into the F65-0-2218 Fixture and insure the O-ring seal on the bottom plate (item 10) is fully compressed with the bolts (item 11). (Make sure all parts are particle free)
3. With "Zero" spring load on the ball apply the following N_2 pressure and measure seal leakage at each pressure in psig; after measuring the position of the 2" ball from the bottom of item 10: 5, 10, 20, 50, 75, 100, 150, 175, 200, 250, 300, 350 & 400
4. Depressurize the fixture and measure the position of the ball as in 3 above.
5. Slowly pressurize the fixture and determine the compression of the teflon seat as a function of pressure.
6. Inspect the part as in 1 above.

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NO. 180-102 BY MC PAGE 3

REV LTR	NC				
DATE	1-2-74				

DEVELOPMENT TEST

TEST: Seat, Sweged PN5736067SK With Swivel Visor PN5736061 with visor unrestrained

OBJECTIVE: To establish the integrity of the sweged seat and unrestrained swivel visor using the best test conditions established in test -101.

TEST HARDWARE: 5736067SK seat assem., sweged, 5736061 Swivel Visor Assem. F65-0-2218 Fixture (all parts must be particle free)

PROCEDURE:

1. Use the 5736067 seat assy and conditions which produced the best leakage results in test 180-101.

2. Use the same test conditions and compare performance of both tests.

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NO. 180-103 BY MC PAGE 4

REV LTR	MC				
DATE	1-2-74				

DEVELOPMENT TEST

TEST: Seat, swaged, PN5736062SK, with restrained Swivel Visor Assem, PN5736061

OBJECTIVE: To evaluate the effect of seat-visor misalignment on leakage as a function of the variables in test 180-101, using the best practices established in -101 & -102.

TEST HARDWARE: 5736062SK Seat Assy, 5736061 Swivel Visor Assy F65-0-2229 Fixture (all parts must be particle free)

PROCEDURE:

1. Align the visor assem and the seat assem so that the center lines of both coinside.
2. Use the optimum test conditions of test 180-102 and pressurize.
3. Induce side motion to misalign the two ℄'s up to the maximum without exceeding leakages in test 180-102.

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NO. 180-104	BY MC	PAGE 5		
REV LTR	NC			
DATE	1-3-74			

DEVELOPMENT TEST

TEST: Seat swaged, P/N5236067SK, with 5236059 Visor

OBJECTIVE: To evaluate the effect of a fixed swaged seat and a constrained visor on leakage at varying pressures & other conditions as established in tests -101, 102 & -103.

TEST HARDWARE: 5236067SK Seat Assem., 5236059 visor and F65-0-2229 fixture (all parts must be particle free)

PROCEDURE:
Follow the best practices established through test 180-103 to produce the lowest possible leakages.

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NO. 180-TIO BY MC PAGE 1

REV LTR.	NC				
DATE	1-3-74				

180 DEVELOPMENT TEST PROGRAM

TEST ITEM: PN5736065, SEAT FLOATING, FLAT RIBBON SPRING

T.V.	TEST OBJECTIVE	TEST DESCRIPTION
1.1	Test seat with perfect ball to establish seat integrity	Install seal in test fixture #F65-0-2215 and test for leakage rate, deflection vs inlet pressure & permanent set.
1.2	Test the ability and extent to which the seal can seal with seat-visor misalignment	Install visor, PN5736059, in test fixture and measure leakage as a function of inlet pressure and extent of misalignment.

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NO. 18C-111 BY M.C PAGE 2

REV LTR	NC				
DATE	1-3-74				

DEVELOPMENT TEST

TEST: Seat, Floating, Flat Ribbon Spring, PN5736070

OBJECTIVE: To establish the integrity of the 5736070 Seal when a 2" dia. ball is used as in test -101.

TEST HARDWARE: 5736070 Seat Assy., Flat Ribbon Spring. FGS-0-2218 Fixture (all parts must be particle free.)

PROCEDURE:
Use the best procedure established in tests -101 through -104 to obtain the lowest possible seal leakage.

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NO. 180-112 BY MC PAGE 3

REV LTR	N/C				
DATE	1-3-76				

DEVELOPMENT TEST

TEST: Seat, Flat Ribbon Spring, PN5236070 with 5236059 Visor

OBJECTIVE: To establish conditions and procedures by which these parts can be used without exceeding leakage of test -111.

TEST HARDWARE: 5236070 Seal Assem., 5236059 Visor and F65-0-2229 Fixture (all parts must be particle free)

PROCEDURE:
Use the procedures used in test 180-104 to produce the minimum possible leakage under the conditions used in the 100 series tests.

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NO. 180-200 BY MC PAGE 1

REV LTR	NC				
DATE	1-2[illegible]				

180 DEVELOPMENT TEST PROGRAM

TEST ITEM: BEARING, BALL, TORQUE

T. I.	TEST OBJECTIVE	TEST DESCRIPTION
201	Establish bearing friction, and shaft axial movement as a function of bearing preload and axial load	Use valve housing (5736071), retainer (5736080), Shaft (5736072), bearings (OL600TL), shim stock and Instron Tester, to vary bearing face "stickout" or dimension "A". See NO bearing cat[illegible]
202	To establish torque requirement as a function of valve operating pressure load	Using the same set above, with visored seal assy, best settings of preload and shaft axial travel, pressurize the valve to 25psig, 50psig, 100psig 150psig, 200psig & 250psig and measure break away torque.
203	Total Torque & Link Evaluation	Repeat above test with one link connected and with two links connected.

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NO. 180-271	BY MC	PAGE 2			
REV LTR	NC				
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DEVELOPMENT TEST

TEST: BEARING, BALL, TORQUE, Axial Load & Preload only

OBJECTIVE: To determine the torque requirements between the shaft and the ball bearings as a function of preload on the bearings and the axial movement of the shaft as a function of axial load.

TEST HARDWARE: Ball bearings, Shaft (5236072), Housing (5236071), retainer (5236080), Cover (5236075), Shim stock .0010, .0015" & .0020" thick (use either 5913013 or 3012 shims from 154 stock).

PROCEDURE:

1. Assemble the shaft & bearings (without visor) per dwg 5739001 without shims.
2. Tighten the screws on bearing retainer item (22) on 5739001
3. Tighten the shaft nut, item 18, 50 in-lb.
4. Slowly load the shaft with the Instron and record force vs shaft displacement up to 100 lb. Record data
5. Measure "breakaway" torque with shaft "loaded" at 100 lb. and "unloaded".
6. Add shims between the outer races of the bearings and increase the torque until axial travel of the shaft at 100 lb is .0002" or less. Measure torque for each setting as in 5. above.

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NO. 180-202 BY M.C PAGE 3

REV LTR	NC				
DATE	1-25-74				

DEVELOPMENT TEST

TEST: BEARING, BALL TORQUE
loading on shaft-Housing ball & Journal bearing

OBJECTIVE: To determine the torque required due to pressure loading on shaft-housing bearings only.

TEST HARDWARE: Same as 201 plus 5236059 visor without the l[illegible] (5236069).

PROCEDURE: 1. Assemble the valve with a visor only to make a seal and to obtain pressure loading effects "break-away" torque measurements without visor shaft friction.
2. Measure [illegible] "break away" torque without pressure at 25 psig, at 50 psig and up to 250 psig in 50 psi increments.

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PARKER HANNIFIN

NO. 180-203 BY MC. PAGE 4

REV LTR	NC				
DATE	1-25-74				

DEVELOPMENT TEST

TEST: BEARING, BALL TORQUE
Total Torque

OBJECTIVE: To determine the total torque required due to all bearings under pressure loading.

TEST HARDWARE: Same as in 202 plus Link #5736089

PROCEDURE: 1. Using the same set-up as in Test 202, install one link on the visor, per 5739001 and repeat test 202.
2. Install the two links on the visor and repeat test 1. above.

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NO. 180-T20	BY MC	PAGE 1			
REV LTR.	A				
DATE	1-14				

180 DEVELOPMENT TEST PROGRAM

TEST ITEM: PN5736076 SEAT, FLOATING BELLEVILLE SPRING

T.V.	TEST OBJECTIVE	TEST DESCRIPTION
1.1	Test seat with perfect 2" dia. free floating ball to establish integrity of seal	Use the F65-0-2216 fixture to establish the forces required and the procedures necessary to obtain a perfect seal.
122	Establish the integrity of a seat-visor combination under conditions of misalignment.	Use the F65-0-2229 fixture
1.3		
1.4		

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NO. 180-121 BY MC PAGE 2

REV LTR	NC				
DATE	1-3-74				

DEVELOPMENT TEST

TEST: Seat, Belleville Spring, PN5736076

OBJECTIVE: To establish the integrity of the 5736076 Belleville seal assem when a 2" dia. ball is used as in tests -101 and -111

TEST HARDWARE: 5736076 Belleville Spring Seat Assem, 1 and - F65-0-2218 Fixture (all parts must be particle free)

PROCEDURE:
Use the methods and procedures developed in tests -101 through -104 to obtain the lowest possible seal leakage.

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NO. 180-122 BY MC PAGE 3

REV LTR					
DATE					

DEVELOPMENT TEST

TEST: Seat, Belleville Spring PN5236067 with 5236059 Visor

OBJECTIVE: To establish the worst possible conditions using the procedures of test -121 without exceeding the leakage in -121.

TEST HARDWARE: 5236067 Belleville Spring Seal Assem, 5236059 Visor and F65-0-2229 Fixture (all parts must be particle free).

PROCEDURE:
Use the same procedure used in tests -109 and -112 to obtain the minimum possible leakage under the most severe alignment conditions.

7 October 74 ①
Vance G. Dunn

DVT 5749030

TEST INSTRUCTIONS

The subsequent instructions provide detail requirements regarding testing of the prototype lifting ball valve. This valve has been manufactured to demonstrate the feasibility of the alternate concept 4-bar linkage design. The valve prototype was manufactured in two stages,

1) a no-flow model just to prove operation,
2) modification of the no-flow model to allow pressurization and approximate aerodynamic torque accessment.

The flow type valve is by no means a pressure tight valve however sufficient pressure loading will be provided to show slope of aerodynamic loading.

Because this valve, and the testing thereof is being accomplished on a minimum cost basis much of the normal structured formality has been waivered and additional information can be obtained from the writer, i.e., Vance G. Dunn X347.

Assemble valve in accordance with assembly instructions dated 25 Sept and revised 5 Oct.

Testing for the valve consists of cycle testing and torque measurements.

NO LOAD TORQUE MEASUREMENTS

a. Orient valve to closed position and rotate shaft 63 degrees to the open position. Measure torque required to rotate valve shaft.

b. Close valve and measure torque required.

c. Repeat test 10 times and record all values.

AERODYNAMIC TORQUE MEASUREMENTS (See figure 1)

a. Connect GN_2 supply to the inlet valve test fixture (3).

b. Connect valve to a test setup as shown in figure 2.

c. Adjust actuator for a ~~stroke that~~ will drive the valve 63°.

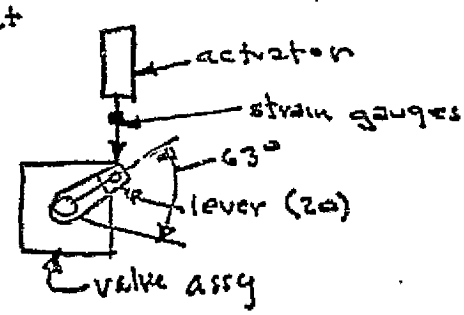

Figure 2.

d. Apply 200 psig to valve inlet and stroke valve measuring and recording strain gauge outputs. Torque is calculated as $(F_{strain\ gauge})(\cos \frac{63}{2})(1.33)$.

· Repeat stroking for 10 cycles.

NOTE

Pressure will leak from valve thru unsealed areas.

e. Apply 300 psig to valve inlet and stroke valve measuring and recording strain gauges.

Cycle valve 10 times.

f. Reduce data and compare for torque repeatability. If torque does not increase for 10 cycles continue stroking for 10000 cycles.

WARNING

If gas leakage is excessive or torque increases above 250 lb-in shut down test and contact this writer for disposition.

DATA REDUCTION

At conclusion of testing provide valve and all test data to vance b. dunn X347. Valve will be disassembled and inspected for wear. Pictures will be taken and a brief summary prepared.

DUT 5739036

S. J. PRESAS

SEP 30 1974

PNEUMATIC ACTUATOR TESTING

BELLOWS STROKE VS. PRESSURE –

THIS TEST IS TO GIVE US DATA ON THE BELLOWS ASSY. USE THE SUB ASSEMBLY FORMED BY 5739038 -1 SHELL WELDED TO 5739020 – 101 BELLOWS ASSEMBLY.

APPLY GAS PRESSURE AT PRESSURE PORT AND MEASURE BELLOWS STROKES VS PRESSURE. RECORD BELLOWS FREE END TO FLANGE DIMENSIONS AT VARIOUS PRESSURES TO OBTAIN STROKES BETWEEN .05 TO .65 INCH MAX AT .05 IN. INCREMENTS. CAUTION: DO NOT EXCEED .65 MAX STROKE. (REF: SEE ATTACHED SHEET)

ACTUATOR STROKE VS. PRESSURE –

THIS TEST IS TO VERIFY ACTUAL STROKE VS. PRESSURE.. FOR THE COMPLETE ACTUATOR AND, BY PLOTTING THE CURVES, SHOW THE B.V. SPRING ACTUATION POINT AND COMBINED BELLOWS AND RETURN SPRING FORCES.

MEASURE AND RECORD AVAILABLE RETURN SPRING CAVITY AND SPRING & BELLOWS STROKE.
COMPLETE ACTUATOR ASSEMBLY PER DWG. NO. 5739036 AND THE MANUFACTURING ROUTINGS.

APPLY GAS PRESSURE AT THE ACTUATOR PRESSURE PORT AND MEASURE ROD END STROKE WITH RESPECT TO PRESSURE, RECORD AT EVERY .05 IN. MOTION.

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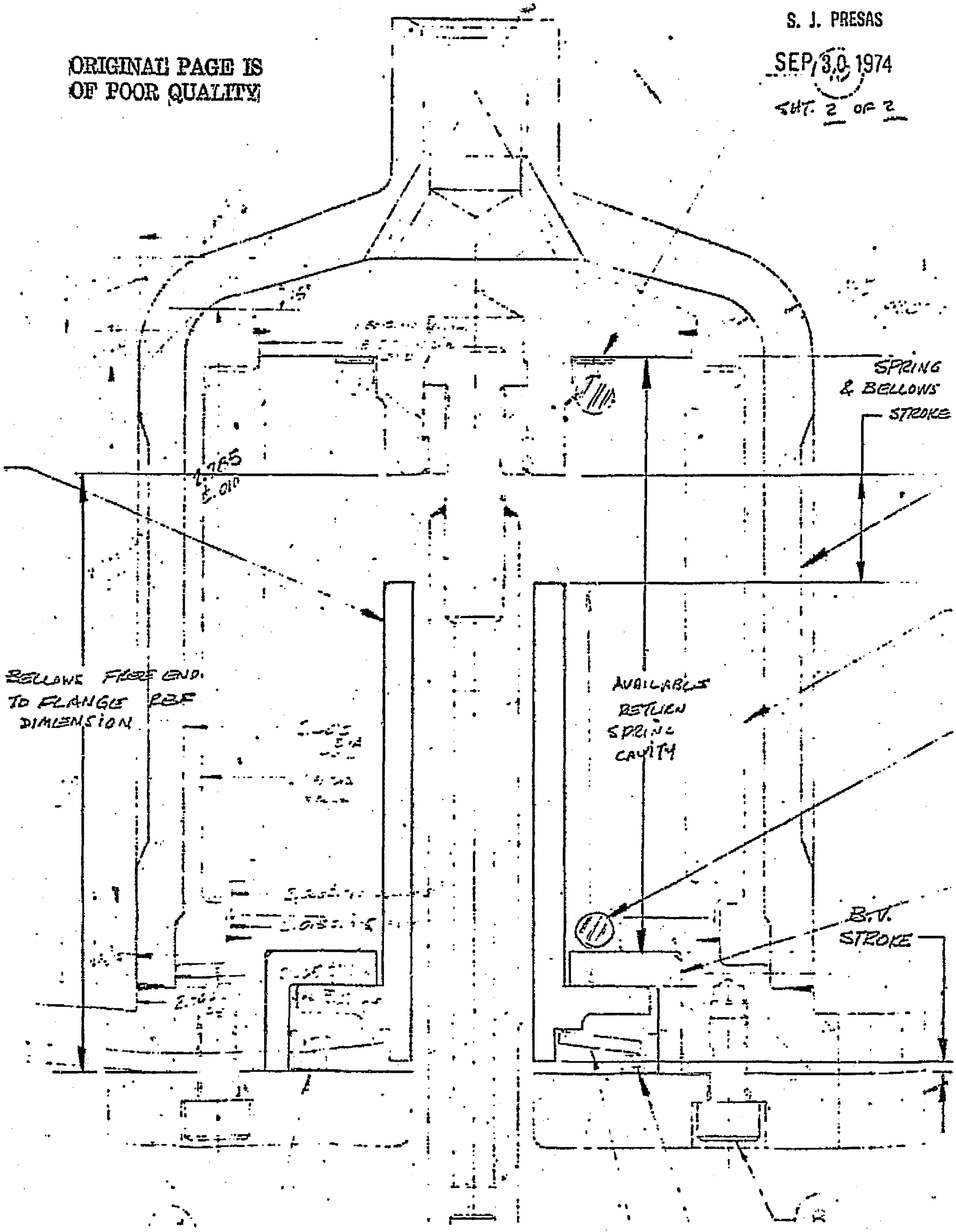
S. J. PRESAS

SEP 30 1974

SHT. 2 OF 2



TITLE		DATE	NAME	COPY	PRINT
DESIGN VERIFICATION TEST PROCEDURE ACTUATOR ASSY	PRINCIPAL ENGR	14 OCT 74	DUNN	10	—
	RESPONSIBLE ENGR	14 OCT '74	PESAS	1	—
	RELIABILITY ENGR				
REPLACES ______ REPLACED BY ______	PLANNER	—	—	—	—
REASON FOR CHANGE: TO REVISE MOTOR TEST SECTION & GEAR ASSY TEST SECTION.	PROCESS ENGR				
	QUALITY CONTROL				
	CONTRACTS				
	CHECK				
REFERENCED SPECS:	RELEASE	10.14.74		1	

CODE IDENT 92003 | SIZE A | SHEETS 24 | REV LTR A

EO SERIAL NO: 02 SHEETS: 1

EO PREPARED BY: PESAS

EO TYPE
- [ ] INITIAL RELEASE
- [x] INCORPORATED CHANGE
- [ ] NON INCORPORATED CHANGE
- [ ] STOP ORDER
- [ ] MEMO
- [ ] ADVANCE MATERIAL ORDER

ECP NO. —

SCN NO.

CHANGE CONTROL
- [x] NOT APPLICABLE
- [ ] CLASS I
- [ ] CLASS II

USAGE REQUIREMENTS

PROG NO.	CUSTOMER	SPEC IDENT NO.	DWG CFG	REV	NEXT ASSEMBLY	END ITEM DWG NO.	EFFECTIVITY FIRST	EFFECTIVITY LAST	QTY NEXT ASSY	QTY EI
180	NASA (EDO)	—	—	—	—	—	—	—	—	—

PARAGRAPHS 4.1 & 4.3 ENTIRELY REVISED

PARTS STATUS (Planning Use Only)	
In Assy	
Stock	
In Process	
Purch	
Raw Matl	

SYSTEMS DIVISION
PARKER HANNIFIN ■ 18321 JAMBOREE BOULEVARD ■ IRVINE, CALIFORNIA 92664

CONTROLLED DOCUMENT

NUMBER: DVT5739006

TITLE: Design Verification
Test Procedure,
Actuator Assembly
Orbiting Maneuvering Engine
Propellant Valve

RELEASE HISTORY							
DATE	REVISION	E.O. NO.	MICROFILM	DATE	REVISION	E.O. NO.	MICROFILM
20 June 74	NC	01	[illegible]				
10-14-74	A	02					

REFERENCE: NASA
S180

PREPARED BY: S.J. Presas
Senior Design Engineer

APPROVED BY: Vance E. Dunn
Vance Dunn
Project Engineer

LIST OF EFFECTIVE PAGES

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TABLE OF CONTENTS

1.0 SCOPE

This development test procedure details the design verification testing to be performed on the Actuator Assembly for the Orbiting Maneuvering Engine (OME) Propellant Valve, Parker-Hannifin PN5739006. This plan covers Task VII under Phase II of Parker-Hannifin's Program Plan, 1PMP5750018.

2.0 APPLICABLE DOCUMENTS

The Parker-Hannifin documents listed below form a part of this document to the extent specified.

QCPM610	Calibration Records and Controls
QCPM620	Calibration of Test Equipment
5739006	Prototype Assembly, Actuator and Valves
IPMP5730018	Program Plan, OMS Engine Propellant Valve Technology Program

3.0 GENERAL REQUIREMENTS

3.1 Environmental - Unless otherwise specified all testing shall be conducted within the following environmental conditions:

Temperature:	70°F ± 5° F
Relative Humidity:	90 Percent or less
Barometric Pressure:	Local Atmosphere

3.2 Tolerances - Unless otherwise specified, the following tolerances apply to the application of test requirements.

Weight:	± 1 percent
Temperature:	± 5° F
Electrical:	± 1 percent F.S.

3.3 Test Facilities - All tests shall be conducted at the Parker, Irvine, facility. Should the necessity arise for additional test facilities, these facilities shall be approved by Parker Project engineering personnel. All testing shall be under the supervision of Parker Project engineering personnel.

3.4 Failure Reporting - Failure is defined as the inability of the actuator assembly or component to meet specified performance requirements of this procedure. In the event of a failure or out-of-tolerance condition, it shall be the responsibility of Parker test engineering to notify Project Engineering. The test setup, equipment and test specimen shall be examined in order to provide the basis for subsequent failure verification and assurance that an actual failure has occurred.

3.5 Test Equipment - Test equipment shown in test Figures (or equivalent) shall be used. Test equipment shall be capable of producing, maintaining and indicating the specified test conditions. Applicable instruments shall be subjected to periodic calibration in accordance with QCPM610 and QCPM620 and shall bear a current calibration decal at time of implementation.

3.6 Test Sequence - The sequence of testing is optional, except Negator Spring Response and Inertia Absorbing Spring Operation shall be the last tests conducted.

3.7 Photographs - Photographs of good resolution shall be made of each valve test setup for inclusion in the test report.

3.8 Test Results Data - Complete test results data shall be recorded for each test using standard laboratory data sheet paper or equivalent. Where appropriate, test results data shall be recorded as exact observed values.

3.9 Disposition of Data - At the conclusion of testing, the original data sheets shall be retained by Parker project engineering.

3.10 Test Report - A final test report shall be prepared by engineering. The report shall be complete within itself. The test report shall include, as a minimum, the following information:

a. Introduction, purpose, summary and conclusion.

b. Detailed report of each test.

c. Exact test methods and results.

d. Detail schematic and/or photograph of each test setup with a list of all test equipment and instrumentation used, the manufacturer's model number and accuracy. The serial numbers of equipment which are critical for repeating tests and/or calibration shall also be logged.

e. Data sheets recording all raw data as actual values.

f. Traceability to test equipment, operators and witnesses.

g. Location and time of test performance.

h. Detailed parts list containing part number, drawing revision letter and lot number for each part in the test article.

3.11 Deviations and/or Variations - Any deviations and/or variations from the procedures specified herein shall require approval of Parker project engineering prior to testing. Any changes to this procedure after its approval may be initiated in the form of an amendment, change order or revision to the procedure.

3.12 Disposition of Test Specimen - Test specimens shall be placed in bonded stores at Parker-Hannifin for system testing which shall follow.

4.0 TEST PROCEDURE

4.1 PLANETARY GEAR TRAIN OPERATION

The purpose of this test is to determine gear train efficiency and reduction ratio.

4.1.1 Reduction Ratio - The gear train assembly is to be held against a flat surface (workbench), cycle counters may be attached to the input and output shafts. (Refer to Figure 1.) Rotate the gear train input shaft by hand, using a serrated shaft and count input turns required for one complete output turn. Repeat test a minimum of three times.

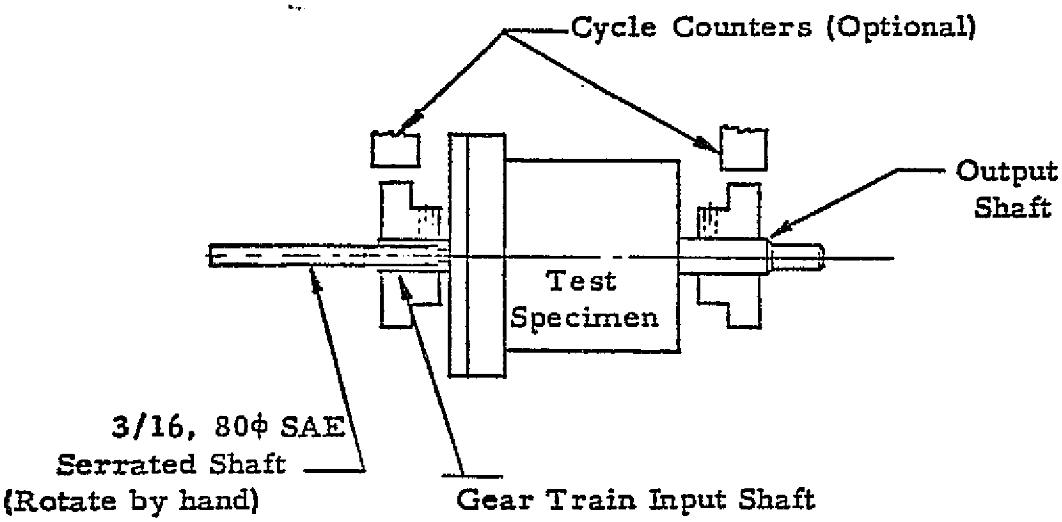

Figure 1

Requirements: The reduction ratio between input shaft and output shaft shall be 77:1.

Feel smoothness of gears during rotation; note any rough spots on data sheets.

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4.1.2 Efficiency - Install output shaft hub as shown and hold the gear train assembly against a flat surface. (Refer to Figure 2)

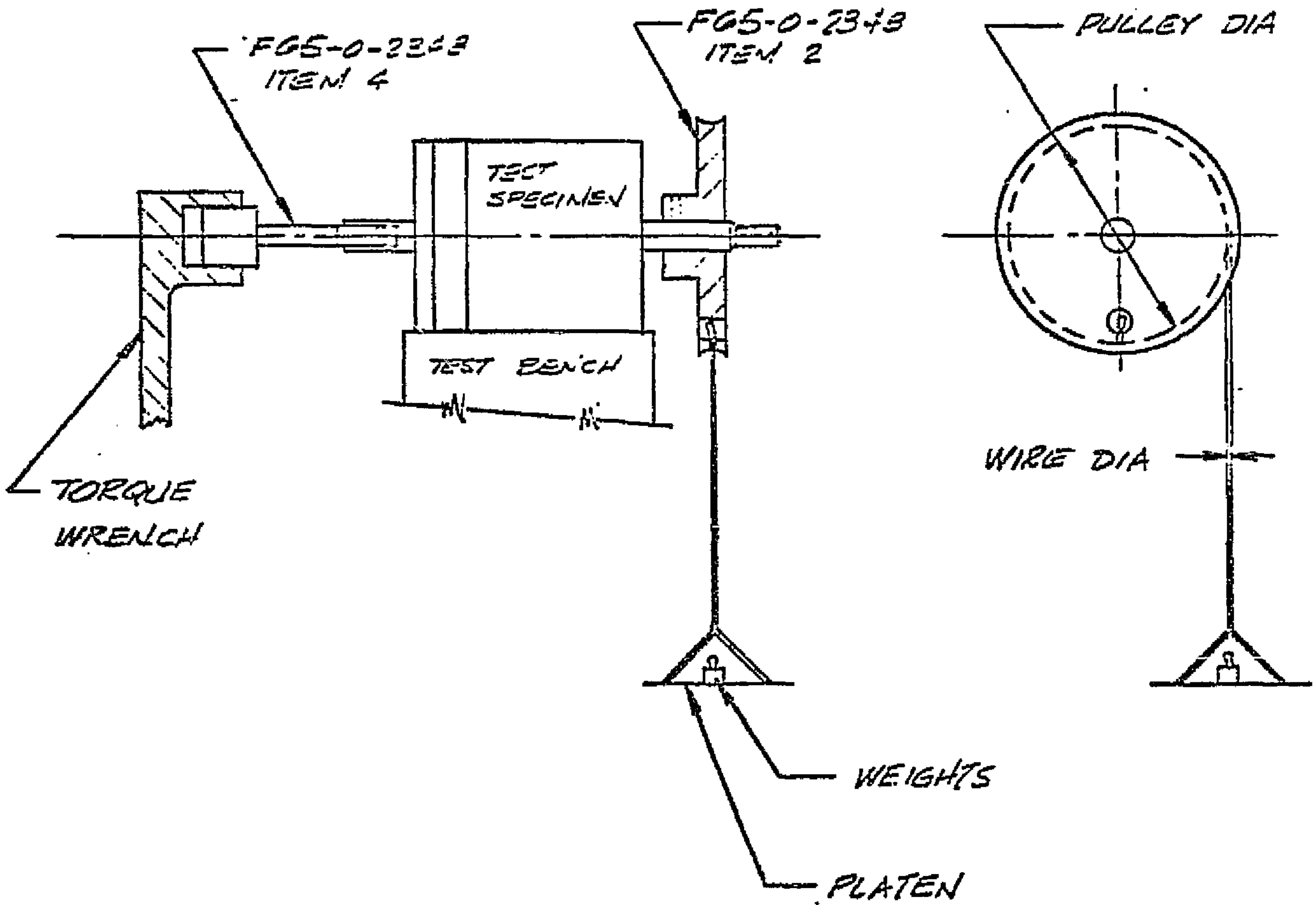
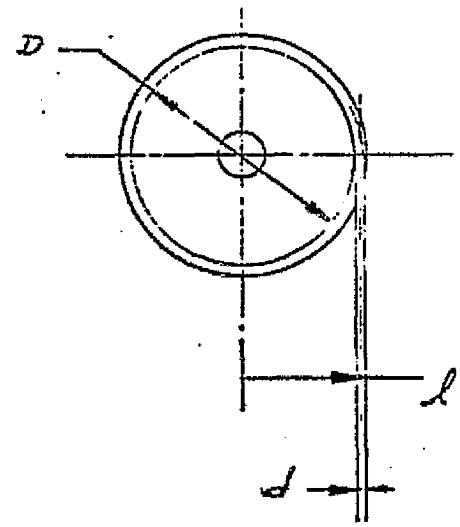


Figure 2

Place a weight of approximately 10 pounds at the platen shown and lift it by rotating the input shaft with a torque wrench. Record input shaft torque, weight at output shaft platen, pulley diameter, wire diameter, etc.

EFFICIENCY CALCULATIONS:



$$l = \frac{D}{2} + \frac{d}{2}$$

WHERE: D = PULLEY DIA
d = WIRE DIA

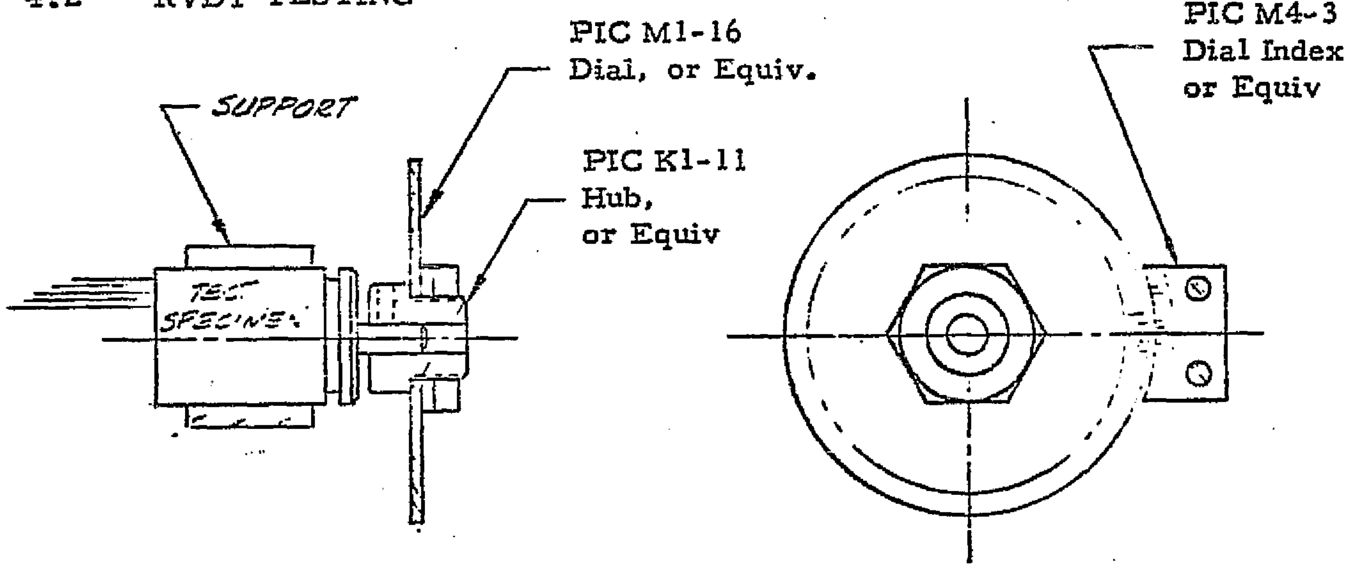
OUTPUT TORQUE
T_o = WEIGHTS AT PLATEN X l

EFFICIENCY IS THEN:

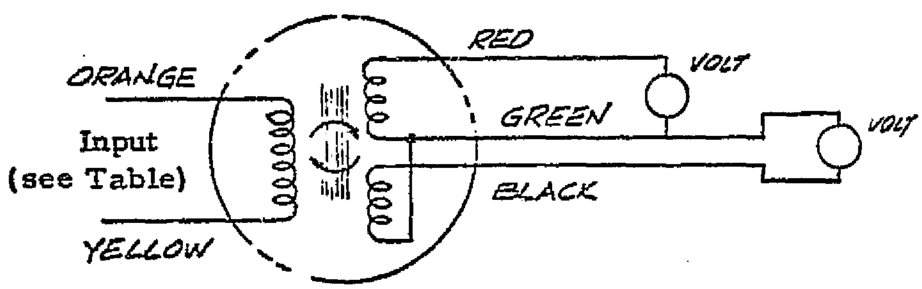
$$\mu = \frac{\text{OUTPUT TORQUE}}{\text{INPUT TORQUE X REDUCTION RATIO}}$$

Requirements: The gear train efficiency must be 0.73 minimum.

4.2 RVDT TESTING



Unit Type	Input
[illegible]haevite [illegible]0A	10 vac 400 Hz
Pickering 23501	6 vac 400 Hz



WIRING DIAGRAM

Figure 3

With the RVDT held within soft (rubber) jaws in a vise, or equivalent, attach the disc dial shown above. Wire the unit as shown in Figure 3 and rotate shaft to null position.

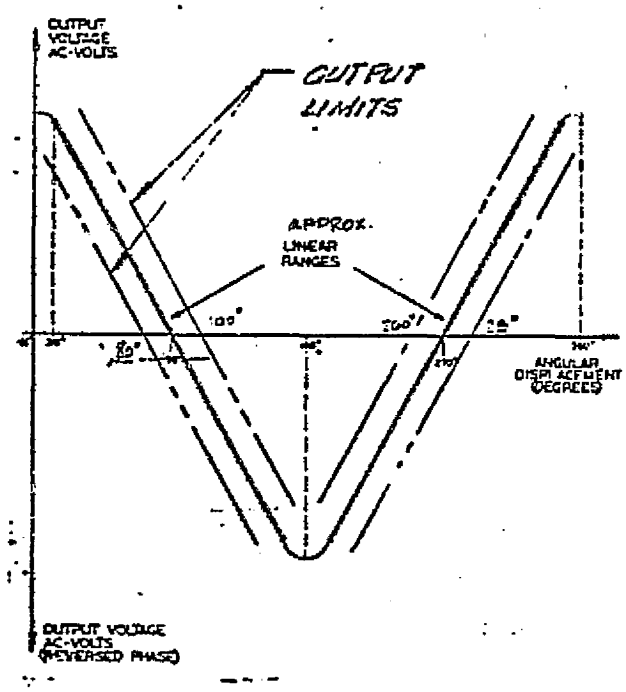
Rotate the shaft 360° and measure and record output (E) versus angle in 2° increments. Rotate the shaft 360° in the reverse direction; measure and record output (E) versus angle in 2-degree increments.

Repeat tests three times; record all data.

Plot voltage output curve with respect to shaft position.

Requirements:

Pickering
Model 23501



Requirements:

Schaevitz
Model R30A

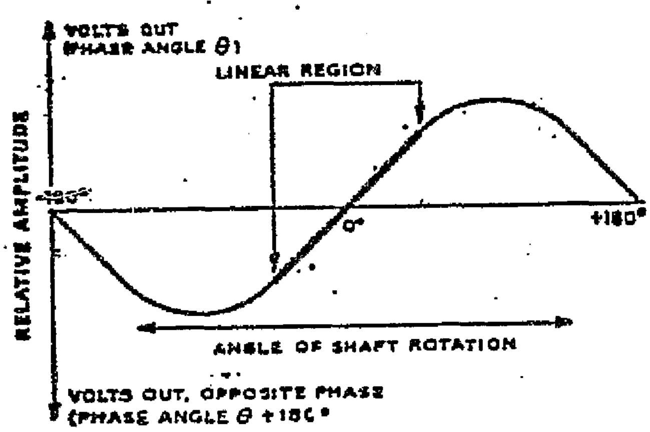


Figure 4

4.3 MOTOR OPERATION VERIFICATION

NOTE: If instrumentation and test equipment type and layout allow it; some of the following tests may be combined, provided the data obtained is equivalent to that required here.

4.3.1 dc Resistance Test of Primary Winding - Measure and record dc resistance between terminals. Effective motor resistance is to be taken as the average of the results obtained from the three possible paired combinations of the three terminals.

4.3.2 Locked Rotor Torque Tests

Attach an arm to the motor output shaft and set up torque measuring apparatus as shown in Figure 5. Connect the motor to the Parker-Hannifin engineering-supplied electronic control and operate to determine locked rotor torque at the various input conditions shown in Data Sheet 1. Perform tests on both motor configurations. Record results.

Power to inverter between 22 and 32 vdc (see note below*) Frequencies of 50, 75, 100 and 150 Hz for 01 motor and frequencies of 150, 175, 200 and 350 for 02 motor. Record data on data sheet 1.

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*Continue increasing E_{in} dc until limit of electronic package is reached or until either the torque value reaches a peak point or the motor current drawn shows sizes of saturation.

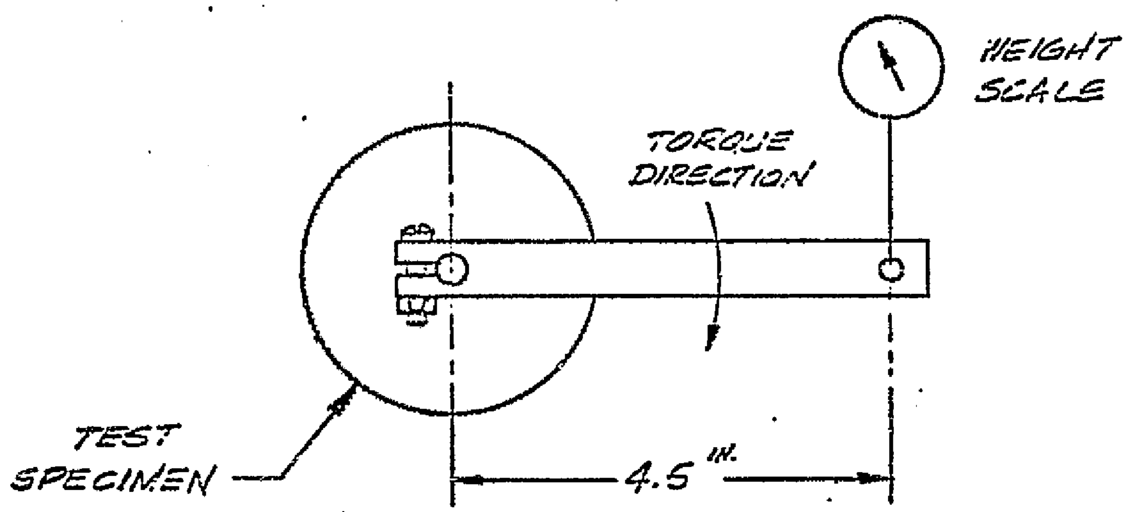

FIGURE 5

4.3.3 Shaft Speed Versus Frequency at No-load Condition - Set up the motor assembly in a test bench approximately as shown in Figure 6 and run the motor with power obtained from Parker-Hannifin engineering-supplied electronic control. This test setup may be combined with the setup for speed versus torque (dyno) test.

Test Specimen: Two electric motors 5739122 SN -01 and -02.

Run the motor with power to electronic control between 22 and 36 vdc. Hold frequencies at 50, 75, 150 and 200 Hz for SN -01 motor and at 150, 175, 200 and 350 Hz for SN -02 motor.

Record test data on data sheet 2.

4.3.4 Speed Versus Torque at Variable Load (Dyno Test)

Set up the motor assembly for dynamometer testing approximately as shown in Figure 6. The dynamometer should be calibrated against a set of known weights prior to each series of tests. Run the motor with power obtained from Parker-Hannifin engineering-supplied electronic control. Conduct tests with both SN -01 and -02 motors.

Power to inverter between 22 and 32 vdc (see note below*). Frequencies of 50, 75, 100 and 150 Hz for -01 motor and frequencies of 150, 175, 200 and 350 for -02 motor. Record data on Data Sheet 3.

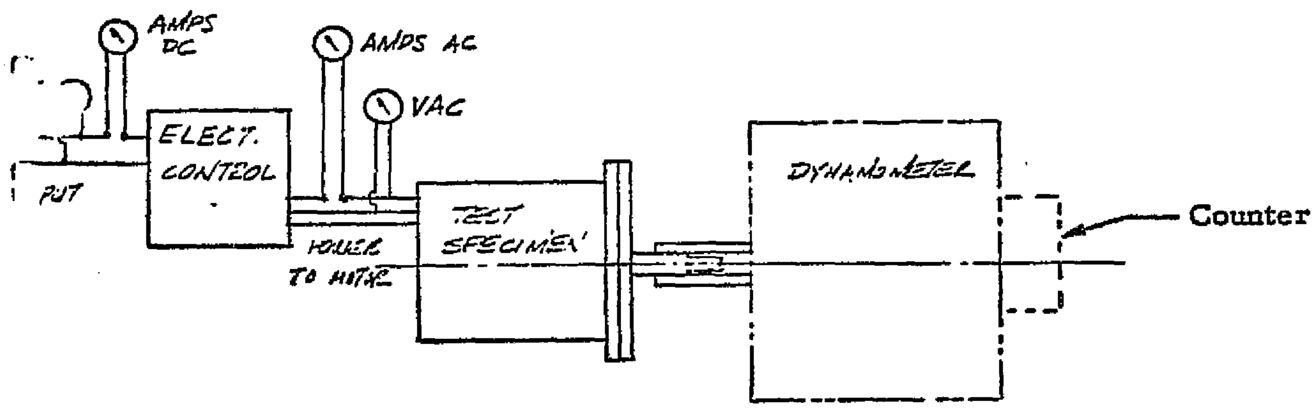

Figure 6

*Continue increasing E_{in} dc until limit of electronic package is reached or the motor current drawn shows sizes of saturation.

4.4 Brake Operation

With the motor held on a test bench, install test tool as shown in Figure 7. Energize the motor brake coil with 22 vdc through 31 vdc and place small laboratory weights (or equivalent) in platen until the motor shaft rotates (approximately equivalent to 8-oz/in. torque).

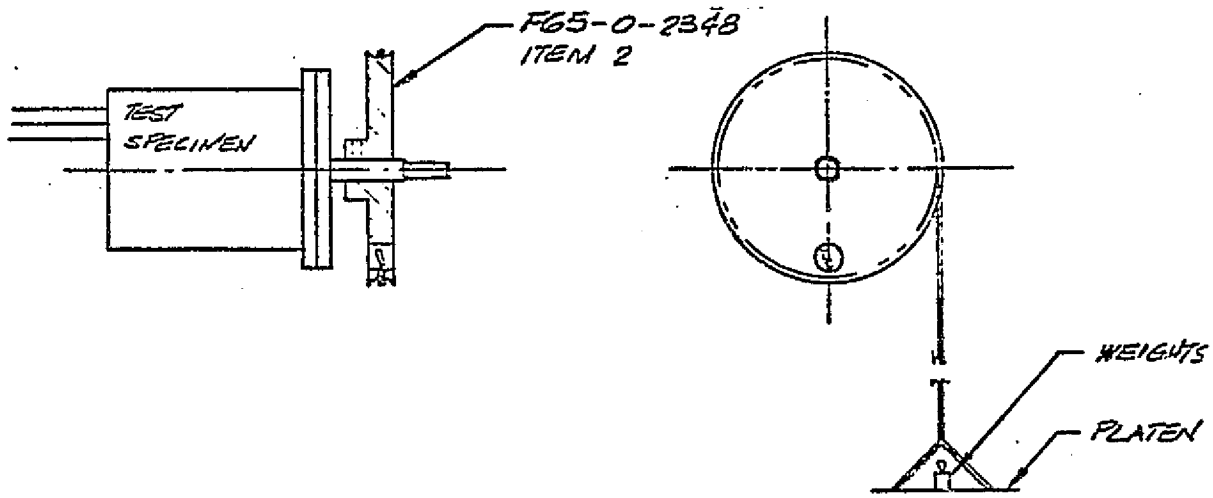

Figure 7

Record voltage, ampere draw and equivalent torque.

Requirements: The brake shall hold 8-oz-in. at 28 vdc with a power consumption of 5 watts maximum.

4.5 NEGATOR SPRING OPERATION

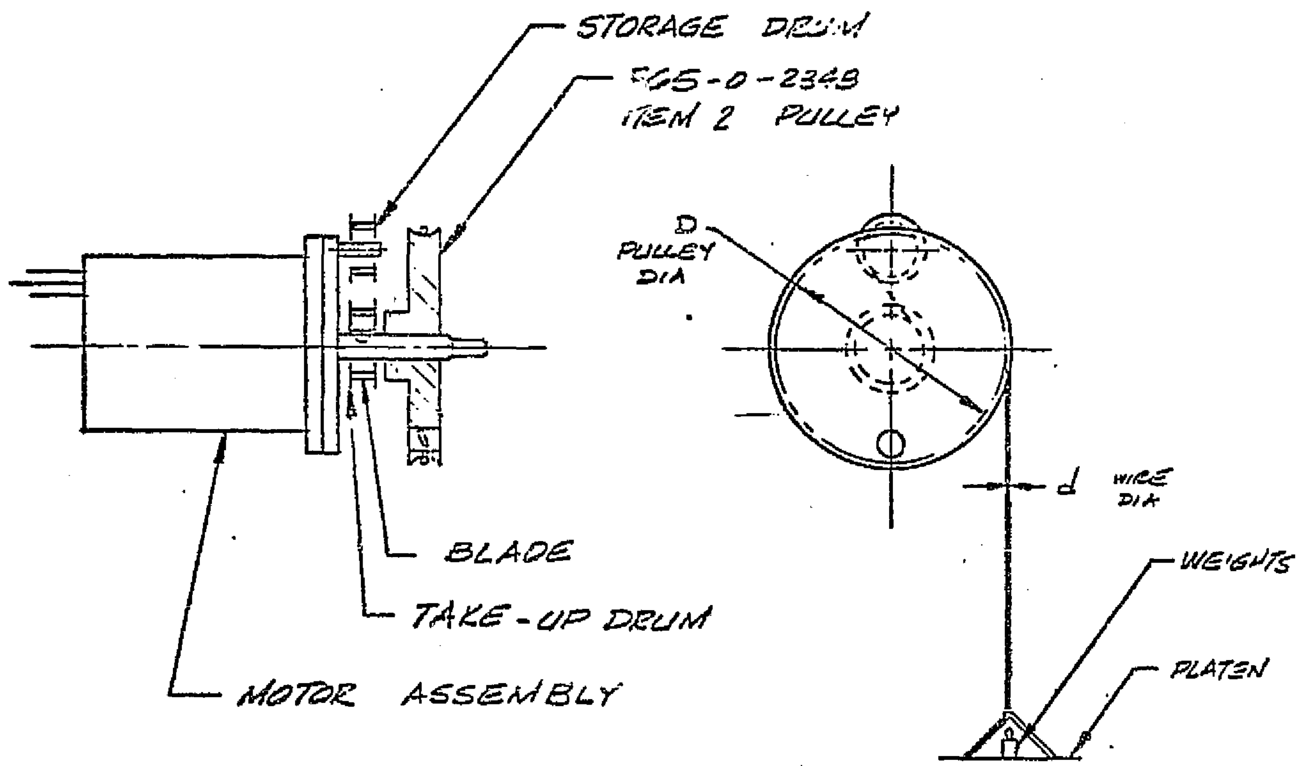

Figure 8

TORQUE AT MOTOR SHAFT IS

$$T_o = \left(\frac{D}{2} + \frac{d}{2}\right)(\text{WEIGHTS} + \text{PLATEN})$$

4.5.1 Torque - Set up motor assembly, negator spring and drums as shown in Figure Figure 8.

With all of the blade in the storage drum (approximately 2 coils at take-up drum, valve closed position) and the motor not energized, measure the effective torque at the motor shaft by means of calibrated weights at the test pulley. Record pulley diameter, wire diameter, weights and calculated torque.

Rotate the motor shaft by hand, 34 turns winding the negator spring blade into the takeup drum. Measure the effective torque at the motor shaft by means of calibrated weights at the test pulley. Record pulley diameter, wire diameter, weights and calculated torque.

4.5.2 Response

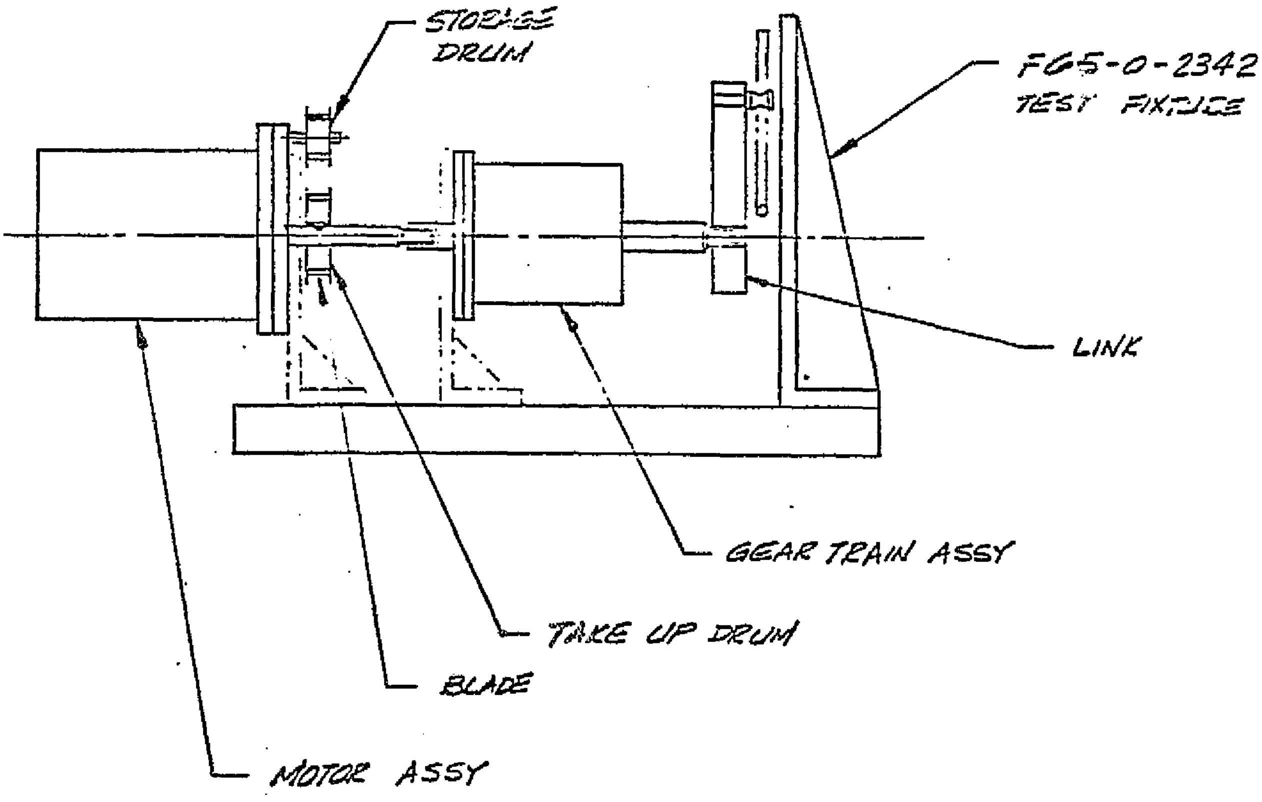

Figure 9

Prepare test setup as shown in Figure 9. This test is to be conducted with a partially completed actuator assembly assembled per Drawing No. 5739006. (Note: It is required that the motor stops at the end of the stroke or the return spring ends may be damaged.)

Actuate the motor between open and closed position and observe the negator spring blade.

Requirements: The motion should be smooth; the blade should not jump the spool sides.

4.5.3 Return Action - With the actuator in the valve open position, remove electrical power and observe.

Requirements: The negator spring must return the actuator to valve closed position within 250 ms.

4.6 INERTIA ABSORBING SPRING OPERATION

Use actuator assembly per Drawing No. 5739006, except modify with test fixture F65-0-2342 special test tool to provide a shock absorbing action in the valve opening direction (required for tests conducted without a functional valve and corresponding linkage). Valves not required.

Actuate motor in the closing mode and observe for overtravel.

Repeat test actuating with the negator spring only.

Record both results.

Requirements: The spring shall be able to absorb the inertia from the motor assembly rotor and other rotating parts within 6° to 10° of motion past closed position.

4.7 MOTOR REVOLUTIONS VS GEAR TRAIN SHAFT OUTPUT ROTATION

Use Actuator Assembly per Drawing No. 5739006 modified per Paragraph 4.6 above. Attach a magnetic or optical target to the motor assembly shaft and to the gear train assembly output shaft. Assemble a special test pulley at the gear train assembly output shaft to create artificial torques between 30 and 80 Lbs-in by means of weights. (Note: a pony brake or equivalent laboratory equipment is also acceptacle.)

Actuate motor under load and measure motor output shaft and gear box output shaft revolutions in a one-minute run. Record.

Requirements: The ratio shall be 77:1.
Record actual torques used.