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DEVELOPMENT OF A LOW COST MOLDED PLASTIC MISSILE/RPV CONTROL SURFACE ACTUATOR

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FINAL REPORT FOR PERIOD DECEMBER 1973 to OCTOBER 1975

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FOR THE COMMANDER

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Hydraulic actuators used to control the fins of tactical missiles are typically fabricated from aluminum and steel alloys in a manner that requires external machining operations. These machining operations normally account for a large portion of the costs of such actuators. To reduce costs, Martin Marietta Aerospace investigated the feasibility of making such an actuator out of mo[illegible] plastic material by designing, fabricating and testing three complete hy[illegible]ulic actuators for the Air Force Flight Dynamics Laboratory.

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FOREWORD

This document is the final report on a study entitled, "Development of a Low Cost Molded Plastic Missile/RPV Control Surface Actuator". The work was performed under project 1987, task 01 from December 1973, to October, 1975, by Martin Marietta Aerospace Corporation, Orlando, Florida, under Air Force Contract No. F33615-74-C-3013 AFFDL.

The work was administered under the direction of the Air Force Flight Dynamics Laboratory, Wright-Patterson Air Force Base, Ohio, 45433, by Mr. Thomas D. Lewis, AFFDL/FGL, Project Manager.

This technical report was released in November, 1975.

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SECTION 1

INTRODUCTION

1. Introduction

This report describes the completed program which demonstrated the feasibility of the plastic actuator concept. The program involved a search for a suitable plastic material; design, fabrication and testing of the actuator; and a production cost analysis based on a quantity of 2000 units. Molding processes and techniques were investigated and pressure testing of the resulting parts was accomplished. Environmental testing was also performed and is described below.

Throughout the program, emphasis was placed on obtaining a moldable plastic material which could withstand +270°F temperatures and obtaining a design which would minimize cost and complexity while maximizing strength and overall actuator performance.

In the interests of economy on this experimental program, the three units fabricated for test were made with a steel shaft and steel pistons; however, all parts which were deemed critical to demonstration of feasibility were made from plastic (the fin locking mechanism was made from steel, as specified in the drawings).

The program was conducted in two phases. Phase I consisted of a study of candidate materials, design layout, design analysis and a production cost analysis. During Phase II, detailed design was accomplished, test equipment was fabricated, three units were fabricated and bench and pertinent environmental tests were performed.

SECTION 2

MATERIAL SELECTION AND ACTUATOR DESIGN

2.1 Material Selection Considerations

The selection of a plastic suitable for use as a structural material for molding hydraulic actuators required the investigation of many material properties. However, from the beginning of the program, one of the most difficult requirements was to find a material which could operate at +270°F since most moldable plastics suffer severe degradation in strength and creep resistance at this temperature. The allowable strength and creep properties were established by the critical area of the actuator which is the inner cylinder wall. Since size and kinematics dictated a cylinder bore of 1/2 inch I.D. and 1 inch O.D., the maximum allowable creep in the 1/2 inch dimension at an inner wall stress of 3900 psi over the period of operation at 270°F was estimated. The wall stress of 3900 psi was calculated based on the above dimensions and an operating pressure of 2700 psi. The required period of operation at 270°F consists of 24 five minute periods or 2 hours. The allowable total plastic creep was estimated at 0.002 inch based on a total allowable piston to wall distance of 0.008 inch (a practical maximum for reliable O-ring sealing at 2700 psi). This 0.008 inch is composed of 0.004 inch clearance and manufacturing tolerance, plus 0.001 inch elastic pressurization deflection, plus 0.001 inch swell allowance due to oil or environmental effects, plus the 0.002 inch plastic creep.

The above figures translate into an allowable plastic creep of 0.0004 inch during a 6 minute period over a 2 inch tensile specimen test length at 270°F at a tensile stress of 3900 psi.

2.1.1 Literature Search

The above material requirements revealed that a material must be found which combined high ultimate tensile strength with low creep characteristics at elevated temperature. Since data on high temperature creep is not normally found in the literature, heat deflection temperature was used as a guide to qualitative selection. Using the strength of polysulfone/glass as a baseline, (15,000 psi minimum room temperature ultimate strength) all compounds listed in the "Modern Plastic Encyclopedia," Volume 50, Number 10A, were reviewed (181 compounds) and those which had a 264 psi heat deflection temperature of 400°F or higher selected as candidates. (See Table 1.)

In addition, the manufacturer (Canadian) of mica filled plastics was contacted to investigate these materials. However, the only thermosets manufactured by them are phenolics or polyesters, both of which do not have the necessary strength at 270°F.

TABLE 1.

COMPOUND CANDIDATES

		Ftu @ 75°F	264 psi Heat Deflection Temp., °F
1.	Polysulfone/Glass	17,000	500
2.	Nylon/Glass	30,000	500
3.	Polyester/Glass	17,300	420
4.	Polyimide/Glass	21,000	660
5.	Polyphenylene Sulfide	23,000	450
6.	Polybutadiene	18,000	500
7.	Epoxy/Glass	25,000	480
8.	Phenol-Formaldahyde	5,000 - 18,000	300 - 600

Compounds 1 through 4 were tested with the results reported herein. Compounds 1 through 3 exhibited excessive creep at 270°F. Compound 5 was found to have a severe degradation in tensile strength at 270°F (Ftu = 8000 psi), based on information from the vendor (Figure 1.).

Compounds 6 and 7 were subjected to reduced-effort testing. Both compounds have coefficients of thermal expansion which are large in comparison to compound 4. No vendor source could be found for compound 8.

Compound 4 was selected for exhaustive testing because of its 660°F heat deflection temperature and its reported insensitivity to chemical attack as reported in AFML-TR-73-277, "Exploratory Development of Chemical Mechanisms of Aging on Polymers used as Resins for Structural Composites."

2.1.2 Testing

All specimens which were tensile tested were pulled on either an Instron model TTD or a TTC test machine. Stress/strain charts were obtained which allowed determination of both ultimate and proportional limits. To acquire simultaneous plastic creep data, the specimens were loaded in approximately 4000 psi increments. After each increment of load was applied, a 6 minute waiting period was initiated during which the Instron chart recorded specimen plastic creep. The 6 minute period was chosen to approximate the 5 minute, 270°F duty cycle with an additional minute added for machine adjustment and settling allowance.

2.1.2.1 Polysulfone/30 Percent Glass Fiber

A total of 10, 7 inch tensile specimens were tested. Table 2. presents the results of these tests. Six specimens were tested at room temperature

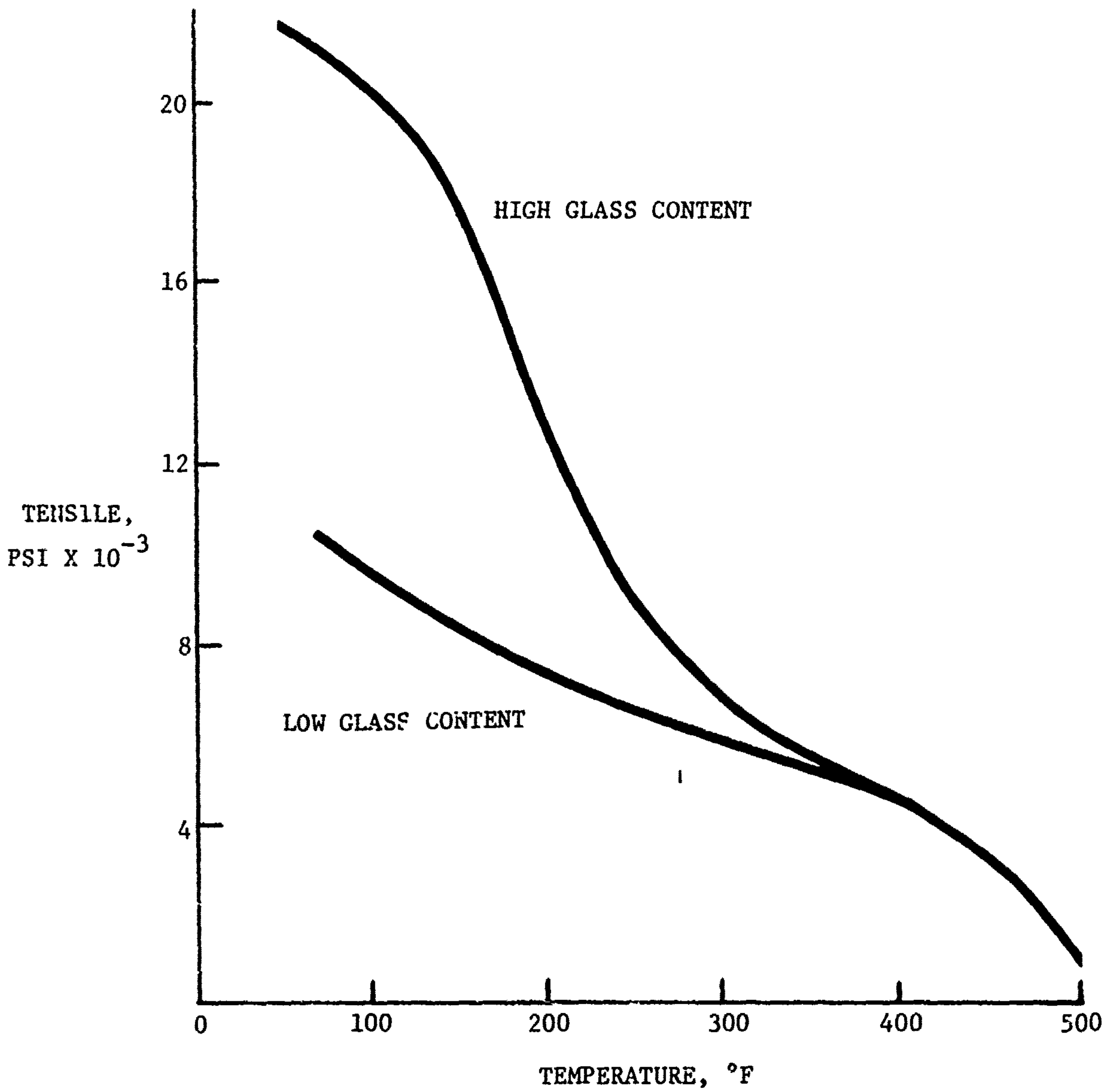


Figure 1. Tensile vs Temperature Polyphenylene Sulfide

TABLE 2

TENSILE TESTS OF POLYSULFONE/GLASS

Specimen Number	Pre-Conditioning	Test Temp.	Proportional Limit(psi)	Ultimate Limit(psi)	Creep(in)*
P1	Room Temp.	Room Temp.	6200	17300	0.0003
P2	Room Temp.	Room Temp.	8700	17900	0.0003
P3	66 Hr in MIL-H-5606 @ Room Temp.	Room Temp.	12500	18100	0.0003
P4	66 Hr in MIL-H-5606 @ Room Temp.	Room Temp.	12500	18100	0.0003
P5	66 Hr in Water @ Room Temp.	Room Temp.	6800	15600	0.0004
P6	66 Hr in Water @ Room Temp.	Room Temp.	6200	15400	0.0006
P7	5 Min @ 270°F	270°F	None Apparent	5900	0.0380
P8	5 Min @ 180°F	180°F	None Apparent	14900	0.0006
P9	66 Hr in MIL-H-5606 @ Room Temp. then 5 Min @ 180°F	180°F	None Apparent	14500	0.0007
P10	66 Hr in MIL-H-5606 @ Room Temp. then 5 Min @ 180°F	180°F	None Apparent	12300	0.0007

* Creep = Plastic deformation in 6 Min. in a 2 inch test length at a tensile stress equivalent to 3900 psi

TABLE 3

TENSILE TESTS OF NYLON/GLASS

Specimen Number	Pre-Conditioning	Test Temp.	Proportional Limit (psi)	Ultimate Limit (psi)	Creep (in)*
NX1	Room Temp.	Room Temp.	9800	30900	0.0002
NX2	Room Temp.	Room Temp.	10800	29100	Not Tested
NX3	5 Min @ 270°F	270°F	None Apparent	15900	0.0012
NX4	5 Min @ 180°F after 6 Hr in Water @ Room Temp.	180°F	None Apparent	19500	0.0009
NX5	5 Min @ 180°F	180°F	None Apparent	19700	0.0012
NX6	5 Min @ 270°F	270°F	None Apparent	14400	0.0025
NX7	66 Hr Soak in MIL-H-5606 @ Room Temp.	Room Temp.	9800	29400	0.0004
NX8	66 Hr Soak in MIL-H-5606 @ Room Temp.	Room Temp.	9600	29100	0.0008
NX9	66 Hr Soak in MIL-H-5606 @ Room Temp.	Room Temp.	9800	29400	0.0008

* Creep = Plastic deformation in 6 Min in a 2 inch test length at a tensile stress equivalent to 3900 psi

and four specimens were tested at high temperature. Exposure to various environments (specimen conditioning) prior to tensile testing is noted in the table.

Table 2. shows that at 270°F, the specimen crept plastically 0.038 inch in 6 minutes at a tensile stress of 3900 psi, which was grossly unacceptable (0.0004 inch required). This occurred even though glass reinforced polysulfone has a published heat distortion temperature of 365°F at 264 psi, one of the highest of the available thermoplastics. Also, this material exhibited a relatively high ultimate tensile strength at room temperature of 18,000 psi. The data also indicates failure to meet the creep requirement at 180°F although the margin of failure is greatly reduced. Based on the poor creep and tensile strength properties exhibited at elevated temperature, further testing of this material was discontinued.

2.1.2.2 Nylon/30 Percent Glass Fiber

A total of 9 Nylon 66, 40% fiberglass filled tensile test specimens were tested. Table 3. lists the results of these tests. Test rationale and methods were the same as those described in Section 2.1. Nylon/glass exhibited very high ultimate strengths at room temperature (29,000 psi) with acceptable proportional limits (10,000 psi). However, at elevated temperatures, the creep characteristics were not acceptable. At 270°F and 6 minutes, creep values of 0.0012 and 0.0025 inch were recorded and, at 180°F, 0.0009 and 0.0012 inch were recorded, all of which exceed the 0.004 inch allowable. Although the creep characteristics of nylon were superior to polysulfone at 270°F, they were similar at 180°F. (See Tables 2 and 3.)

Further testing of nylon was discontinued due to its high temperature creep characteristics.

2.1.2.3 Polyester/Glass

Two specimens of polyester/glass were tested at 180°F. The 264 psi heat deflection temperature for the polyester/glass compound was reported at 415°F. When tested at 180°F, however, both samples exhibited proportional limits of 3200 psi and ultimates of 8800 psi. Further testing was abandoned when both samples crept 0.003 inch at 3200 psi in 6 minutes at 180°F.

2.1.2.4 Polyimide/Glass

The polyimide/glass compound selected for testing was Rhodia Corporation Kinel 5504. This compound (65%, 1/4 inch chopped glass reinforced) was selected because of its physical properties, low cost, and reported inherent characteristic of emitting no gases or water vapor during the compression molding process. The latter characteristic is reported by the manufacturer and was at least partially substantiated by test. (This characteristic strongly enhances moldability.)

The Rhodia literature gives the following salient physical properties for Kinel 5504:

1. F_{tu} = 21,000 psi @ 77°F

2. F_{cu} = 32,500 psi @ 77°F

3. Heat distortion temperature = 660°F @ 264 psi

4. Coefficient of linear expansion = 7.8×10^{-6} in/in/°F

5. Density = 119 lb/cu ft

6. Impact strength = 17 ft-lb/in, notched I_{ZOD} at 77°F.

The Rhodia literature also indicates that 5504 retains approximately 90% of its room temperature tensile strength at 270°F. Also, Rhodia personnel affirmed that no significant change in physical properties occurs at -65°F.

Rhodia's literature indicates no change in flexural strength of specimens tested at 77°F after 500 hours exposure to 250°F, 30 psi steam. When tested at 480°F, however, the flexural strength fell from 57,000 psi to 33,000 psi. (See Section 2.2.4 regarding humidity test.)

Report AFML-TR-73-277 states that ultraviolet light can cause chemical changes in polyimides (prevented or mitigated by blocking agents or shading). It states that chemical attack, with the exception of water, presents no problems with known organic solvents. It also states that "since there are no known organic solvents for polyimides, we would expect no degradation to occur as a result of contact with cleaning compounds, fuel, etc."

Molding techniques for Kinel 5504, as with most thermosetting resins, are restricted to compression molding, and possibly for small simple parts, transfer molding. Injection molding is not feasible with existing molding techniques.

A 1/8 x 12 x 12 inch plaque of Kinel 5504 was obtained from Rhodia Corporation and three, 4-inch test specimens (highly necked) were cut out and subjected to tensile tests. The ultimate strengths were: 16,880, 26,990, and 22,770 psi. These values illustrate the spread in tensile values found to be typical of plastics with large percentages of glass reinforcing fiber.

2.2 Phase I Testing of Polyimide/Glass

All testing done on Kinel 5504 polyimide/65%, 1/4 inch chopped glass material was accomplished with specimens molded at Martin Marietta. Molds were designed and fabricated and molding processes were developed in the most expeditious and economical manner since no Phase I funds were available for hardware fabrication. Much experience was gained in the field of economic molding of thermosetting plastics, however, optimum physical properties (to match manufacturer's data) were not always obtained. The conclusions reached regarding physical properties are, therefore, considered conservative and these conservative figures were subsequently used in the design rather than the manufacturer's published data.

Although some experience was gained in making plaques from preforms, no preforms were used in molding specimens used for test data. Preforms should enhance strength and product consistency.

2.2.1 Tensile Testing

A total of 10 polyimide, 7 inch specimens were tensile tested using the same testing rationale stated in section 2.1. The results of these tests are listed in Table 4. An average room temperature ultimate tensile strength of 14,820 psi and an average proportional limit of 10,650 psi were obtained for the four specimens tested. At 270°F these averages were 12,400 psi and 11,200 psi, respectively, for the specimens tested. At 270°F, the maximum creep in 6 minutes at proportional limit stress (11,200 psi average) was 0.0005 inch which is well within the criteria stated in section 2.1 of 0.0004 inch in 6 minutes at 3900 psi for a 2 inch test length.

Three specimens were exposed to 187°F and 95% relative humidity for 13 days. They were then tensile tested at room temperature. The average ultimate limit was 16,200 psi and the average proportional limit was 11,800 psi. These values compare favorably with the unexposed values above.

Based on the above tests, an ultimate of 11,500 psi and a proportional limit of 10,000 psi were used in the design.

2.2.2 Punch Shear Tests

Many punch shear tests were run. These were run in a two-fold effort to obtain not only an absolute value for punch shear but also to ascertain if a given molded plaque was molded properly from good quality resin. It was determined that a properly molded plaque approximately 1/8 inch thick should exhibit a punch shear in excess of 18,000 psi at room temperature. In addition to plaques, 1/8 inch thick sections cut from molded cylinder bases were tested with similar results. These values are listed in Table 5.

2.2.3 Phase I Cylinder Molding and Testing

One of the principal areas of concern in the feasibility of a plastic actuator is the integrity and strength of the hydraulic cylinders. These cylinders must not only withstand high burst pressures, but must be molded

TABLE 4

TENSILE TESTS OF POLYIMIDE/GLASS

Specimen Number	Pre-Conditioning	Test Temp.	Proportional Limit (psi)	Ultimate Limit (psi)	Creep(in)*
K1	Room Temp.	Room Temp.	11,000	16,800	0.0002
K2	Room Temp.	Room Temp.	10,400	12,700	0.0002
K3	Room Temp.	Room Temp.	11,300	15,200	0.0002
K4A	Room Temp.	Room Temp.	9,900	14,400	0.0002
K4	6 Min @ 270°F	270°F	12,600	12,600	0.0002
K5	6 Min @ 270°F	270°F	11,200	11,200	0.0002
K6	6 Min @ 270°F	270°F	9,800	13,500	0.0002
K7	13 Days @ 187°F 95% R.H.	Room Temp.	11,700	16,800	0.0002
K8	13 Days @ 187°F 95% R.H.	Room Temp.	11,200	12,900	0.0002
K9	13 Days @ 187°F 95% R.H.	Room Temp.	12,400	19,000	0.0002

* Creep = Plastic deformation in 6 min in a 2 inch test length at a tensile stress equivalent to 3900 psi

utilizing a plastic containing reinforcing fibers. These fibers tend to align themselves in the direction of mold closure (axially) so that the cylinder walls have virtually no fibers aligned circumferentially (which would give good hoop strength) and, hence, do not reinforce the walls optimally. This discussion is predicated on a mold which closes axially. Although a cylinder block could be molded by closure perpendicular to the cylinder axes, this is unfeasible because of molding difficulties (resin and fiber flow must occur circumferentially around the cylinder-forming plugs and then must knit perfectly on the downstream side of the plug).

To obtain preliminary knowledge regarding these areas of concern, a compression mold was designed and fabricated which formed cylinders as shown typically in Figure 2. Thermosetting plastic Kinel 5504 was used. The cylinder I.D. is 1/2 inch and the O.D. is 1 inch. The 1 inch diameter section is 3 1/2 inches long with a 2 1/2 inch diameter and a 3/4 inch thick base. Approximately 15 of these test cylinders were molded. The evolution of improvement in molding technique is shown in the figure. The cylinder at the left side of the photo was one of the first molded. Its fibrous, dry, and void-filled appearance is the result of improper heat-pressure cycling. The cylinder on the right is one of the last molded and exhibits good density and knit; it burst at a pressure of 7041 psi (2.6 x operating pressure of 2700).

After the proper molding temperatures and pressures were known, along with the proper mold filling technique, 12 cylinders were molded for the initial Phase I test program. It should be recognized that this Phase I mold and the process used were, of economic necessity, very unsophisticated (e.g., no mold heaters or thermocouples; raw unpreheated resin loaded into a 250°F mold). Porosity was suspected and therefore cylinder testing was directed at ascertaining basic mechanical strength information only.

2.2.3.1 Phase I Cylinder Burst Tests (Table 5.)

At the beginning of the contract, it was ascertained that proof and burst testing need only be accomplished at room temperature and not at 270°F. Since nominal system pressure is 2700 psi, burst test pressure was taken as 6750 psi for the cylinder. To avoid the expense and complexity of furnishing a hydraulic burst test setup with the attendant difficulties of sealing and bleeding each cylinder, a high compliance, white silicone rubber was poured into each cylinder to be burst tested and a steel dowel was used as a piston. An added advantage of using silicone was its characteristic of being retained in any fracture cracks or porosity voids so that they could be easily identified when the cylinders were sawed longitudinally after burst test. The piston was pushed into the cylinder with an Instron testing machine and the load at which the cylinder wall fractured was recorded.

Of the 12 cylinders tested, cylinders 4, 6, 11, and 12 were subjected to room temperature burst tests. Cylinders 6, 11, and 12 were first subjected to fatigue tests as called out in section 2.2.3.2. The average burst

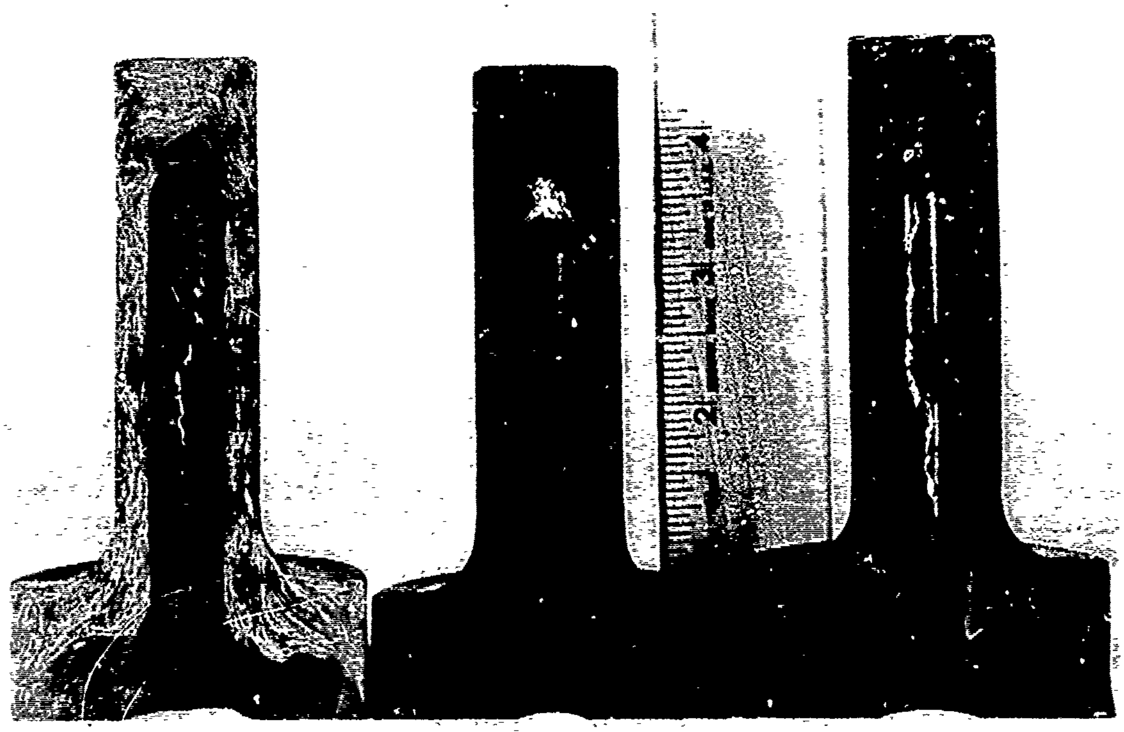


Figure 2. Phase I Typical Molded Cylinder Specimens

TABLE 5

PHASE I CYLINDER TESTING

Specimen Number	PRE-CONDITIONING Punch Shear (psi)	Specif. Gravity (GR/CM^3)	Fatigue Variable	No. of Cycles	Fatigue Load (lb)	Type of Corresponding Loads (2)	BURST Failing Load (lb) RT	Failing Load (lb) 270°F	Equiv. Stress (psi) (3)	Equiv. Press. (psi) (4)
1	20,150	1.86	No	One	-	-	-	820	6,970	4,180
2	23,380	1.87	Yes	23,000 (1)	590-670	Intermed.	-	-	-	-
3	18,150	1.85	Yes	1,100 (1)	740-800	Proof	-	-	-	-
4	18,130	1.87	No	One	None	-	950	-	8,080	4,840
5	18,470	1.86	Yes	180,000	460-530	Operating	-	1160	9,860	5,910
6	19,560	1.87	Yes	180,000	460-530	Operating	1360	-	11,560	6,920
11	27,730	1.87	Yes	180,000	460-530	Plus 100 Cycles 0 to 540	1030	-	8,759	5,255
12	29,280	1.89	Yes	180,000	460-530	Plus 100 Cycles 0 to 540	1330	-	11,735	7,041

Remark: No change in diameter after successful fatigue test. Rate of fatigue: Approximately 1800 cycles/min.

(1) Failed in fatigue

(2) Proof Pressure: 4070 psi
Oper. Pressure: 2700 psi

(3) Stress in material

(4) Pressure in actuator

pressure for these four specimens was 6014 psi. This corresponds to a maximum cylinder hoop tensile stress of 10,000 psi as calculated from Lame's equation for thick-walled cylinders and is based on a 0.25 inch wall The actual design employed a 0.30 inch wall providing a calculated hoop stress at a burst pressure of 6750 psi of 10,266 psi (see section A-6 of Appendix A). This, then, presented an obvious problem regarding the ability of the cylinders to meet the burst pressure requirement. The problem was mitigated by the following factors:

1. The ratio of cylinder O.D. to length is approximately twice as great in the test specimens as it was in the actual Phase II test actuators. This "slenderness" is particularly harmful to any effort to obtain a more random fiber orientation in the cylinder walls; the fibers tend to line up in an [illegible] direction giving minimum hoop strength. A lower slenderness ratio helps to obtain better fiber orientation.

2. The actual design is a block design; i.e., the cylinders are side by side with continuous material between. This enhances fiber orientation in the walls.

3. The control over resin preheat, mold heat and pressure, and process timing would be more precise in any production effort which should result in a more uniform and higher strength product.

At the conclusion of Phase I, the confidence for success with respect to meeting the burst pressure requirement was shared by the molding expert from the resin manufacturer who reviewed the design. The results of pressure tests on the molded aft case halves are compiled in section 5.3.

2.2.3.2 Phase I Cylinder Fatigue Tests

Table 5. shows that specimens 2, 3, 5, 6, 11, and 12 were subjected to fatigue tests. Specimen 3 was subjected to a sinusoidally varying load between 740 and 800 pounds which corresponds to cylinder pressures of 3776 psi and 4082 psi, respectively. A pressure of 4070 psi is proof pressure. (Phase I) A total of 1100 cycles were applied before rupture of the cylinder. Specimen 2 withstood 23,000 cycles before rupture between pressures of 3040 psi and 3418 psi which are intermediate pressures between proof and nominal system pressure of 2700 psi. Specimens 5, 6, 11, and 12 all withstood 180,000 cycles between 2348 psi and 2700 psi (2524 $\pm$ 176 psi) with no failure. In addition, specimens 11 and 12 withstood 100 cycles from zero psi to 2700 psi. Pressure cycles of $\pm$ 176 psi correspond to fin deflections around zero of $\pm$ 2.5°. This assumes an actuator torque relation such that:

$$T = 800 \cos (90 - \theta) \text{ in-lb}$$

(This formula gives T = 400 in-lb. at θ = 30° per the statement of work).

In summary, four specimens withstood 180,000 cycles of $\pm$ 2.5° fin deflection which, it is felt, simulates bench testing frequency response and flight. In addition, specimens 11 and 12 also were subjected to 100 "hard over" signals from 0 to 2700 psi which simulate possible extreme conditions during bench tests. It should also be noted that normal static cylinder pressures are approximately only one-half of system pressure or 1350 psi.

2.2.3.3 Phase I Cylinder Torsion Tests (See Figure 3.)

Cylinder specimens 7, 8, 9, and 10 do not appear in Table 5., but were used to determine the shear modulus, G, of Kinel 5504 for use in shaft design. Figure 3. shows that the cylinders were mounted to a base plate by means of four through bolts in the cylinder base. A hole at the tip of each cylinder accepted a dowel and lever bar which was pushed by an Instron testing machine to apply a known torque to the cylinder. Measurements of angular deflection between sections a known distance apart were made with a Martin Marietta fabricated optical telescope. G was then computed using measured values of moment of inertia, angular deflection, and shaft length.

Two specimens were measured at room temperature and two at 270°F. The computed values for shear modulus are:

Room temperature: $G_1 = 292,000$ psi

270°F: $G_2 = 174,000$ psi

Figure 4. contains graphs of the test results.

2.2.3.4 Phase I Cylinder Porosity Test

A test was run which investigated the porosity or permeability of the cylinder walls under pressure. A section of a cylinder was cut to form a pipe which was tapped with an MS 33649-6 tap at each end. This section was then pressurized at 2000 psi with MIL-H-5606 fluid for 2 hours. No seepage of fluid occurred through the walls. The section was then sawed in half longitudinally and the walls were examined under a binocular magnifier. No evidence of intrusion of fluid into the wall was observed. However, the pipe leaked slightly at the interface with the fittings. This was due to microcracks running axially which were caused by the tapping operation. In the actual design, although all major threads except one are designed as molded threads, the pressure inlet port is presently a tapped port. Special precautions, such as use of sharp tools with proper rake and use of the proper lubricant, were used in the fabrication of the test actuators.

2.2.4 Phase I Environmental Tests

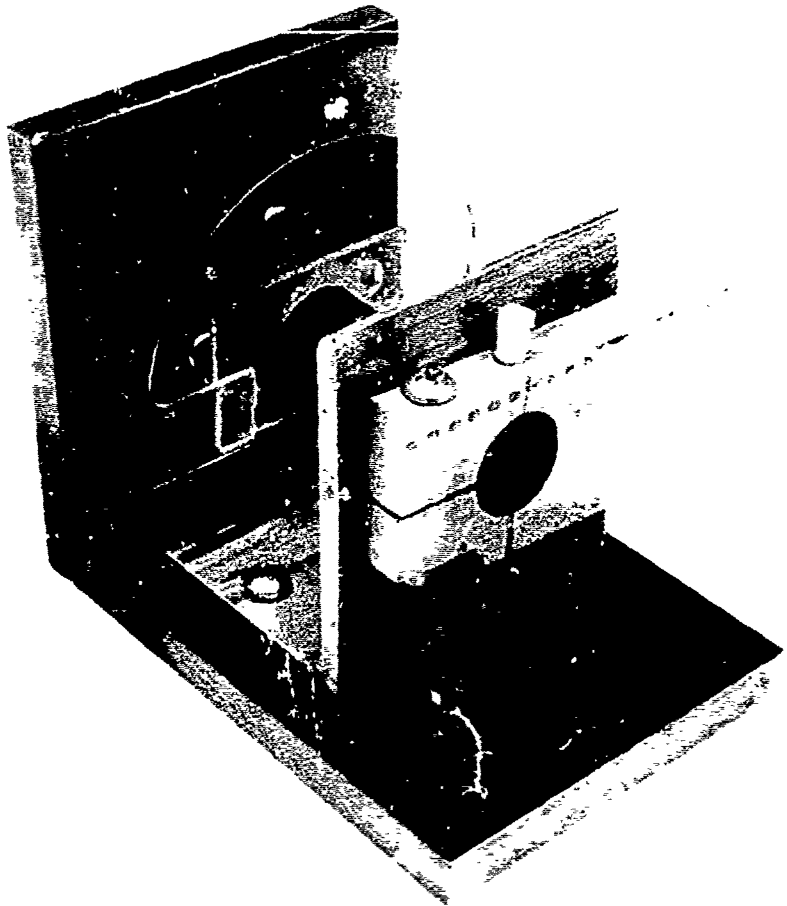


Figure 3. Torsion Test Fixture

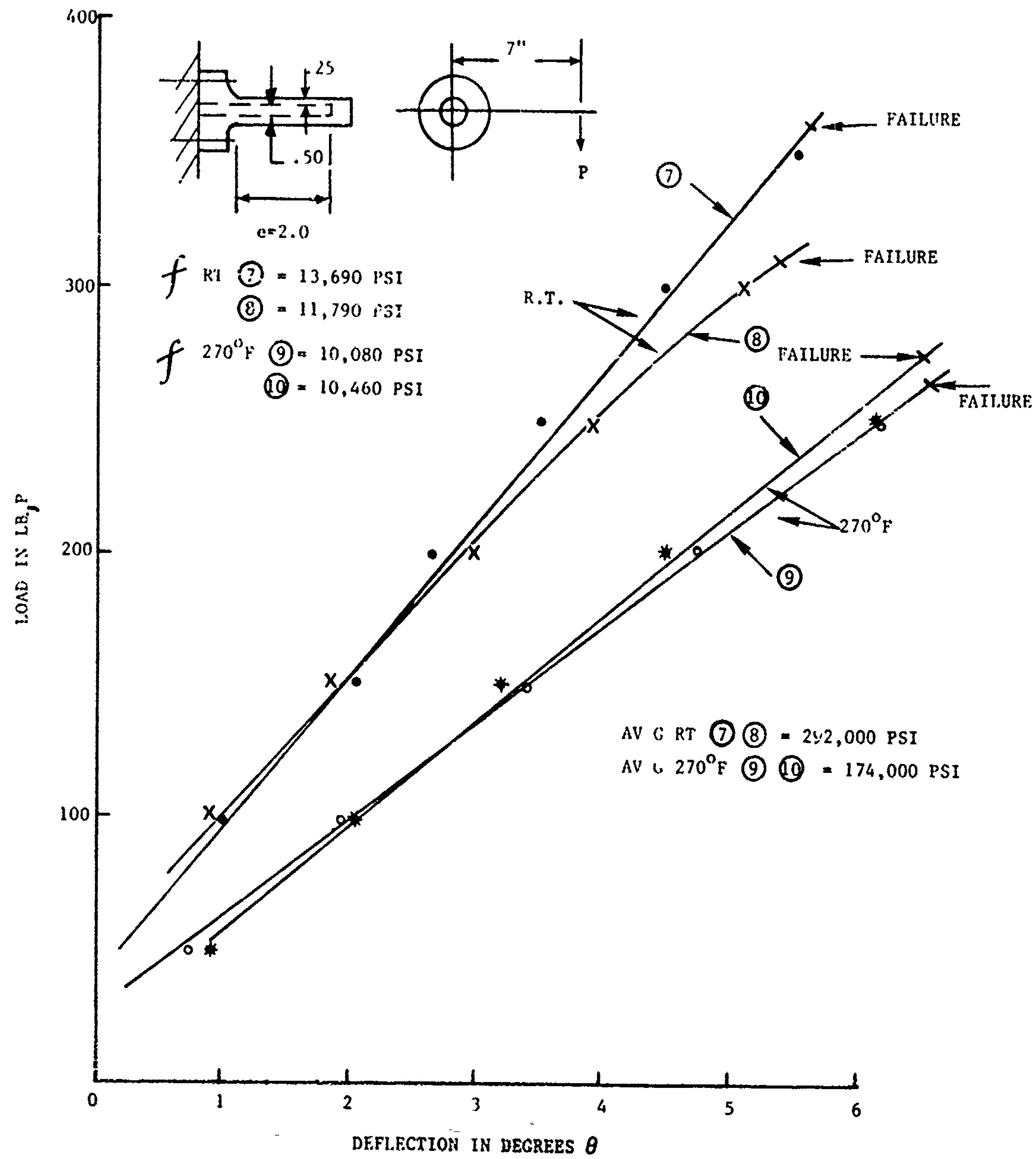


Figure 4. 5504 Polyimide Load Deflection Curve at R.T. and at 270°F in Torsion

2.2.4.1 Humidity and Water Exposure Tests

Section 2.2.1 contains a description of tensile tests on high relative humidity/high temperature exposure specimens. In addition, some data was made available from another program regarding dimensional and weight changes after either submersion in boiling water or exposure to various humidity conditions.

2.2.4.1.1 Boiling Water Submersion Test

A 3 x 9 x 1/8 inch plate molded from Kinel 5504 was cut into strips approximately 3 x 1/2 inch. Five of these strips were measured, submerged in boiling water for one week, and then remeasured. An average length increase of 0.0012 inch/inch, a width increase of 0.0017 inch/inch, and a 1.3% weight increase resulted. Translating this 0.0017 inch/inch figure to the 1/2 inch bore of the actuator cylinder results in an increase in its diameter of 0.00085 inch (after one week in boiling water), assuming this type of translation is valid. This is within the 0.001 swell allowance assumed in Section 2.1. It should also be noted that the five strips were cut from a sheet so that no gel coat remained along the cut edges. This could be conducive to wicking and, hence, absorption of water.

2.2.4.1.2 Humidity Swell Tests

A cylinder (as in Figure 2.) was measured as received, subjected to a 3% relative humidity, 160°F environment for one week, and then remeasured. The maximum change in any dimension was a shrinkage of 0.0006 inch/inch. A weight loss of 0.34% occurred. The specimen was then subjected to 100% R.H. at 135°F for one week and was remeasured. The maximum change in any dimension (from the as-received condition) was an increase of 0.001 inch/inch.

As an additional test, 10 specimens which were cut from a 3 x 9 x 1/8 inch plate (as in Section 2.2.4.1.1) were subjected to the same humidity and temperature conditions as the above cylinder. The maximum change recorded was an average increase above original dimensions of 0.0008 inch/inch measured after one week in 100% R.H. and 135°F. A weight decrease of 1.08% was measured after the initial week at 3% R.H.

None of the above tests resulted in excessive swell figures based on the 0.001 inch growth allowance for the 1/2 inch cylinder bore assumed in Section 2.1. Cut edges and wicking are the same as that described in Section 2.2.4.1.1.

Finally, a ring was machined from the base of a cylinder. Its dimensions were 2 1/2 inches O.D., 2 inches I.D., by 1/2 inch high. This ring was subjected to the same temperature/humidity cycles as the above cylinder. The maximum change in dimension was a shrinkage of 0.0087 inch/inch across the 2 1/2 inch dimension after exposure for one week to the 3% R.H. conditions. This very large and unacceptable shrinkage is attributed to the very large exposure of cut fibers occasioned by the machining operation, with attendant wicking and stress relieving. The experiment does emphasize a possible problem if large amounts of machining are performed on any part.

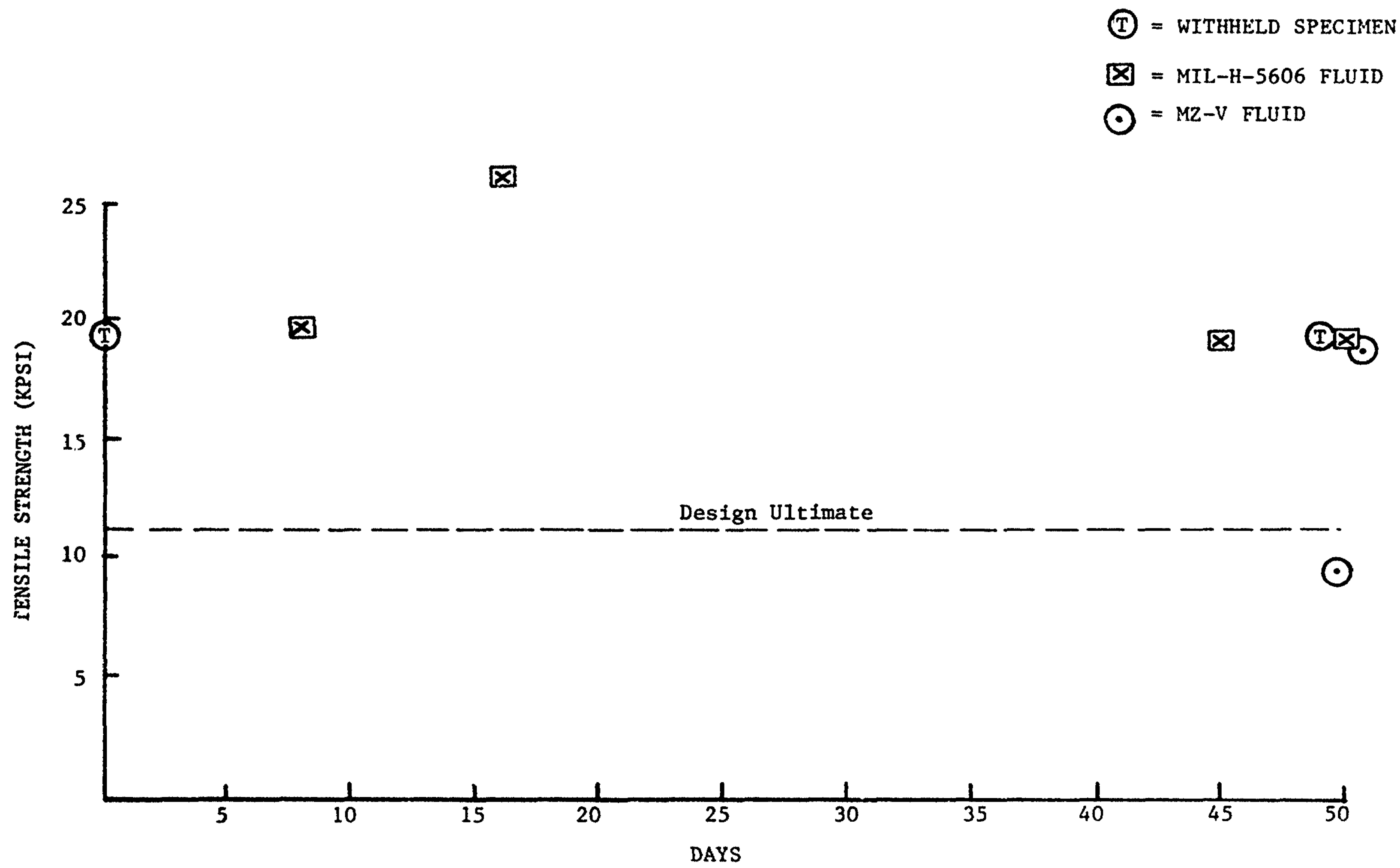


Figure 5. Hydraulic Fluid Immersion Test Placque #1

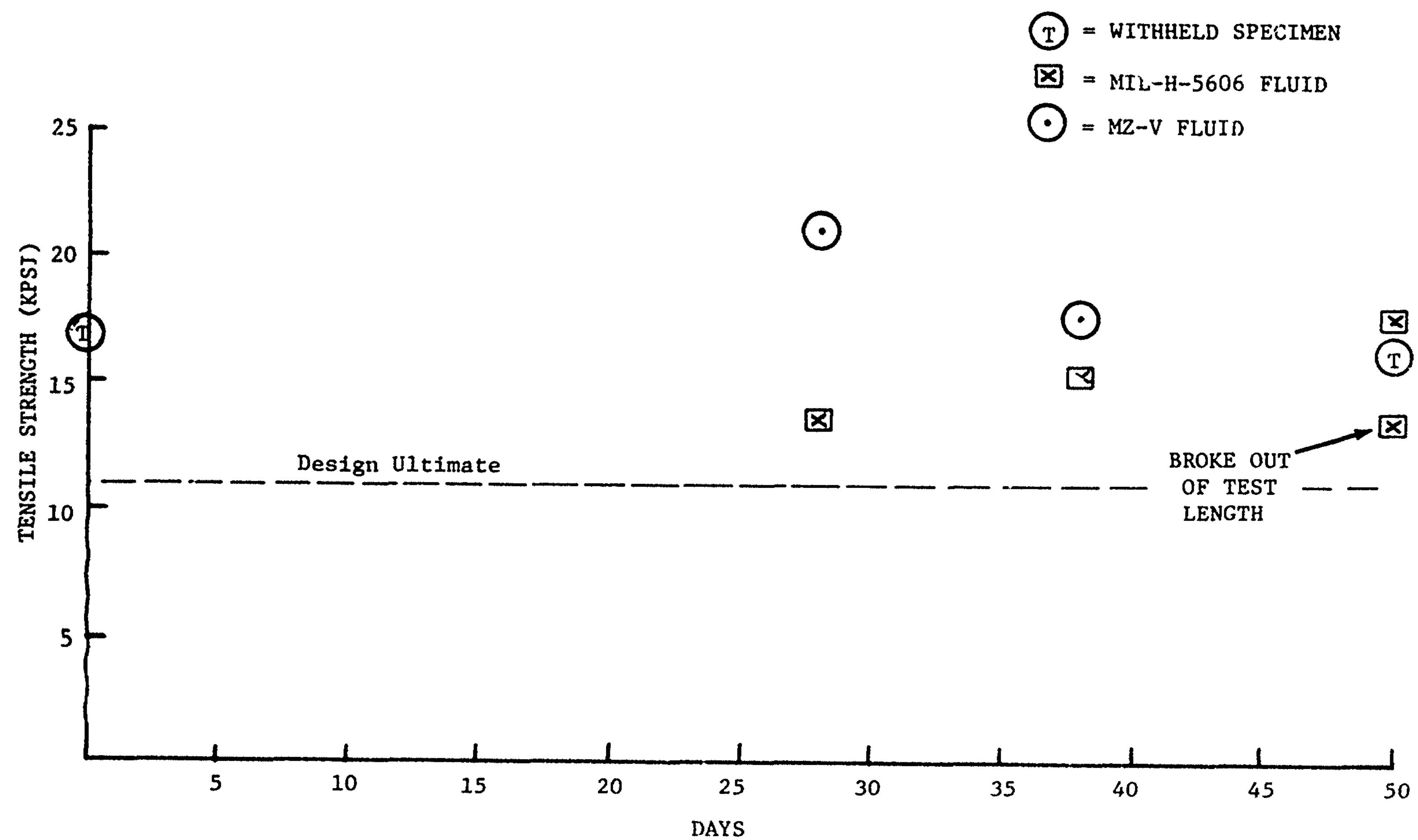


Figure 6. Hydraulic Fluid Immersion Test Placque #2

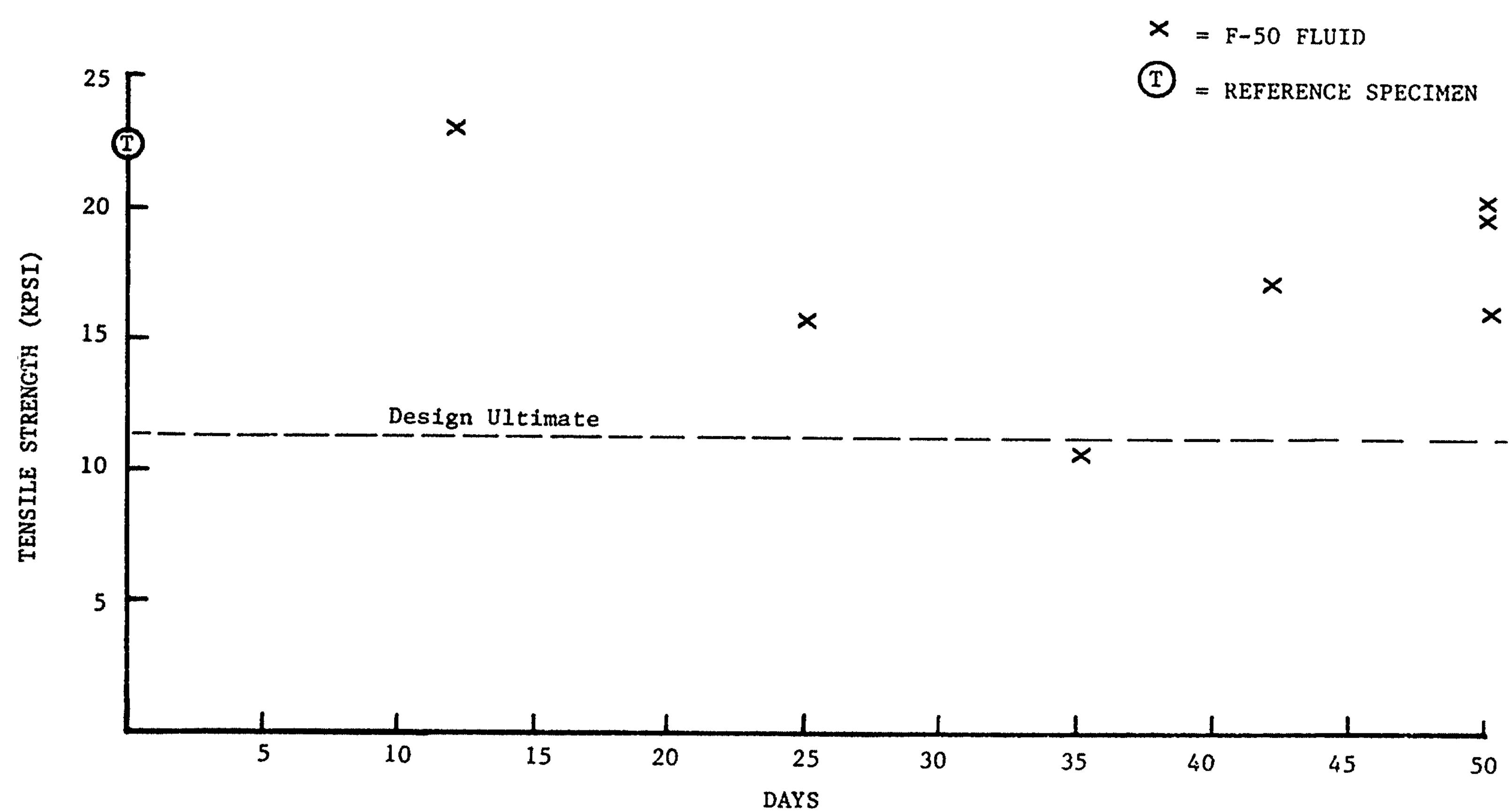


Figure 7. Hydraulic Fluid Immersion Test Placque #3

Rhodia personnel were contacted concerning the weight variations noted above. Their data indicate a weight increase of 0.9% after specimens were exposed to 250°F, 30 psi steam for 500 hours. Their conclusion was that the moisture was absorbed into the skin only and not deeply into the specimen. Specimen strength was not significantly affected.

2.2.4.2 Hydraulic Fluid Exposure Tests

To determine the effects of exposure of polyimide/glass to hydraulic fluid, three 1/8 x 3 x 9 inch plaques were molded. Test specimens approximately 4 inches long by 5/8 inch wide were then machined from these plaques and some were immersed in each of three different hydraulic fluids: MIL-H-5606 (petroleum base), M2-V (silicate ester base), and F-50 (silicone base).

Prior to immersion, the thickness, width, hardness, and weight of each specimen were recorded. Also, a non-immersed specimen from each plaque was tensile tested to serve as a reference. Two other specimens were withheld for testing at a later date to determine aging effects on non-immersed specimens.

2.2.4.2.1 Tensile Test Results

To compare tensile test values on a viable basis, tensile test values of all samples from a given plaque were plotted on graphs entitled "Plaque 1", "Plaque 2", or "Plaque 3". Stated differently, tensile test values are compared only between specimens that were fabricated from the same plaque to account for basic differences in plaque integrity. Figures 5. through 7. present these graphs.

Figures 5. and 6. show that the eight specimens soaked in MIL-H-5606 fluid showed no evidence of degradation in tensile strength during the 50 days of the test, and all exceeded the 11,500 psi ultimate used in the actuator design (Section 2.2.1). It was concluded that MIL-H-5606 had no effect on specimen tensile strength. (Note that one specimen, at 50 days, broke out of the test length, indicating an imperfection in the test specimen rather than a fluid effect.)

Four specimens were immersed in M2-V fluid. Three specimens exhibited tensile strengths equal to or higher than their corresponding reference specimens. The fourth broke at 9710 psi, which is grossly inferior, and indicates an imperfection in the molded part rather than any attack by the fluid. Although the number of samples immersed in M2-V was small, the tensile strength of polyimide did not appear to be adversely affected by the M2-V during Phase I.

Seven specimens were immersed in F-50. One sample was withheld as a non-soaked reference from this plaque. Figure 7., plaque 3, indicates that although a drop in strengths occurred between 25 and 42 days, no real downward trend existed at 50 days since two specimens broke at 20,000 psi.

At 35 days the specimen broke out of test length. Lack of a downward trend as time increases suggests no attack by the fluid; however, since 6 of the 7 specimens broke below the reference value, the possibility exists that the F-50 fluid affects the tensile strength. More extensive testing required to obtain conclusive results. (See Section 2.2.5 below.)

Both reference specimens withheld for 50 days from plaques 1 and 2 broke very close to their counterparts at the beginning of the test as shown below:

	Plaque 1	Plaque 2
0 days	19,657 psi	16,950 psi
50 days	19,730 psi	16,160 psi

2.2.4.2.2 Dimensional and Hardness Change Results (Phase I)

The maximum change in the 5/8 inch dimension (which approximates the 1/2 inch cylinder bore) noted for any specimen immersed in any fluid for up to 50 days was a 0.0003 inch decrease. Decreases and increases were recorded so that no trend was apparent. The variation in the data was within the ability of the observer to read a micrometer. The same comments apply to measurement of the 1/8 inch dimension. It was therefore concluded that none of the fluids significantly affect the polyimide/glass from a dimensional stability standpoint. The reference specimen withheld from plaque 1 grew 0.0009 inch in the 5/8 inch dimension. The reference specimen from plaque 2 did not grow significantly.

Similar results were obtained regarding hardness. A "Shore D" hardness tester was used before and after immersion. All readings were 93-94 before immersion and 93-94 after withdrawal from any fluid. No effect was observed.

2.2.4.2.3 Weight Changes (Phase I)

Figures 8. through 10. are plots of the percent weight increases versus days in fluid of the immersed specimens. All specimens were weighed before and after immersion on a Mettler H20T precision analytical balance which reads in increments of 0.01 milligram. All specimens showed increases in weight. The specimens were not washed with solvent to remove fluid before weighing but were first dried with paper towels to remove surface fluid. A typical weight increase for a fresh specimen immersed and dried, as above, is 0.034%.

Both F-50 (silicone) and M2-V (silicate ester) fluid soaked specimens show least weight gain (0.15% to 0.20%). Their weight increase versus time curves show minimal upward slopes. The 5606 specimens show gains up to 0.32% and the weight change versus time curves show definite upward slopes.

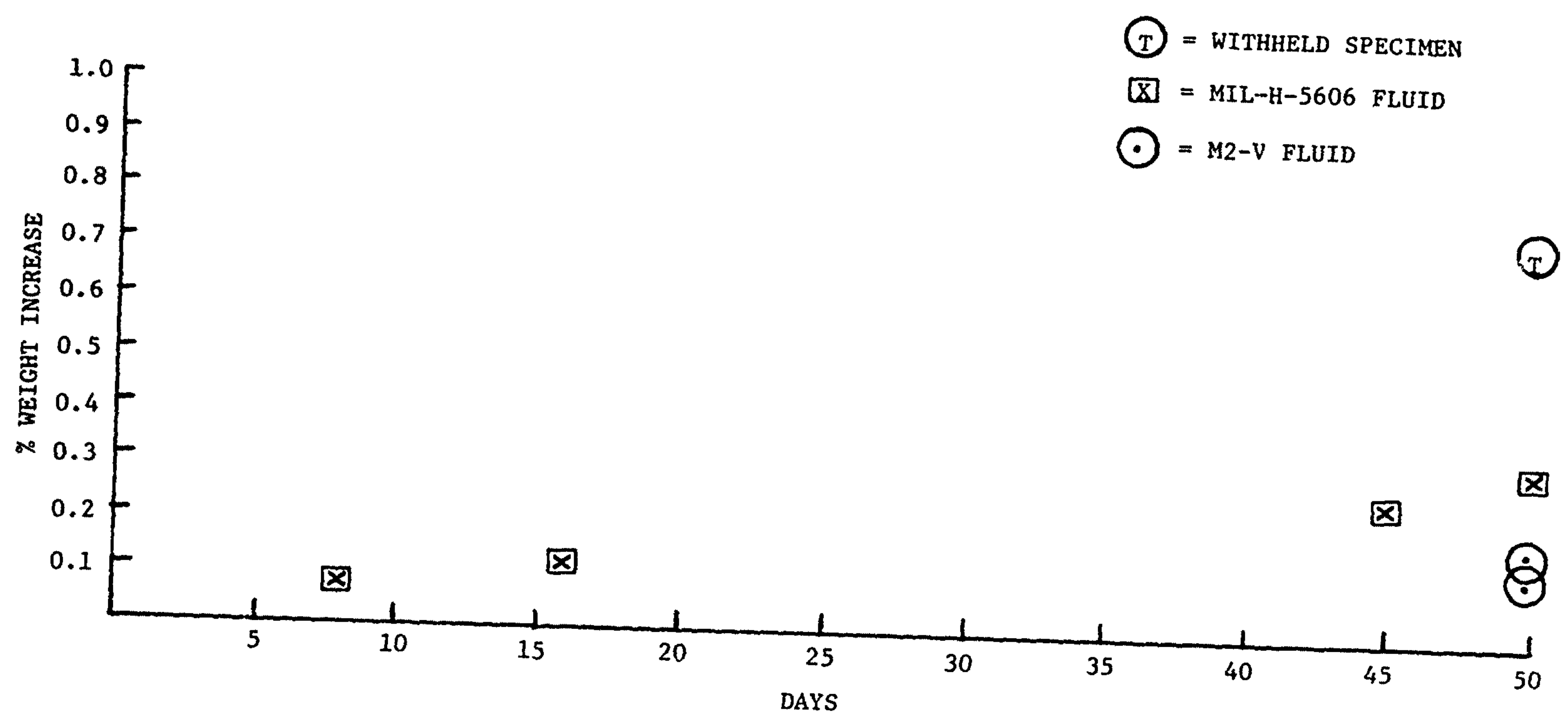


Figure 8. Hydraulic Fluid Immersion Test Placque #1

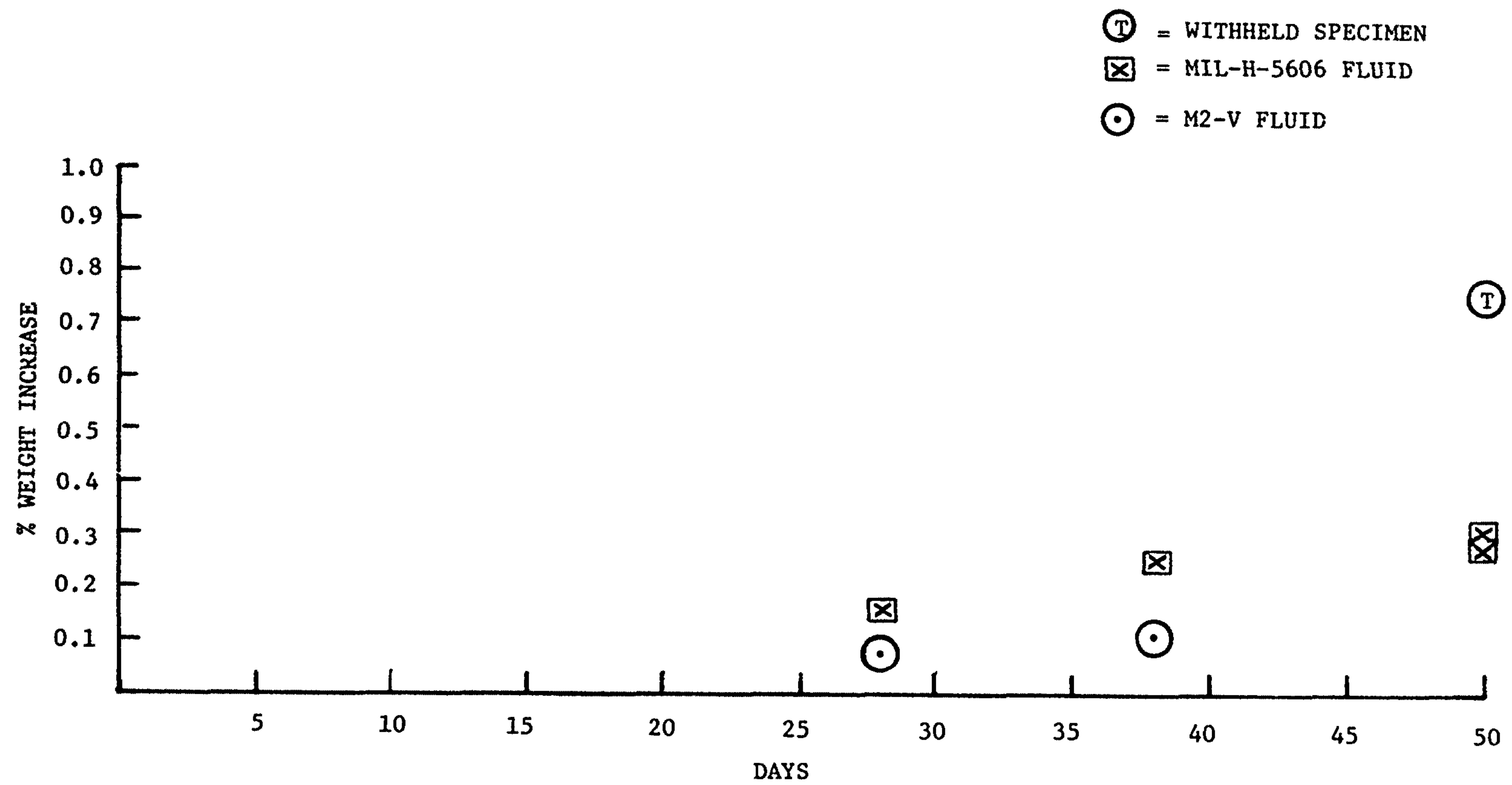


Figure 9. Hydraulic Fluid Immersion Test Placque #2

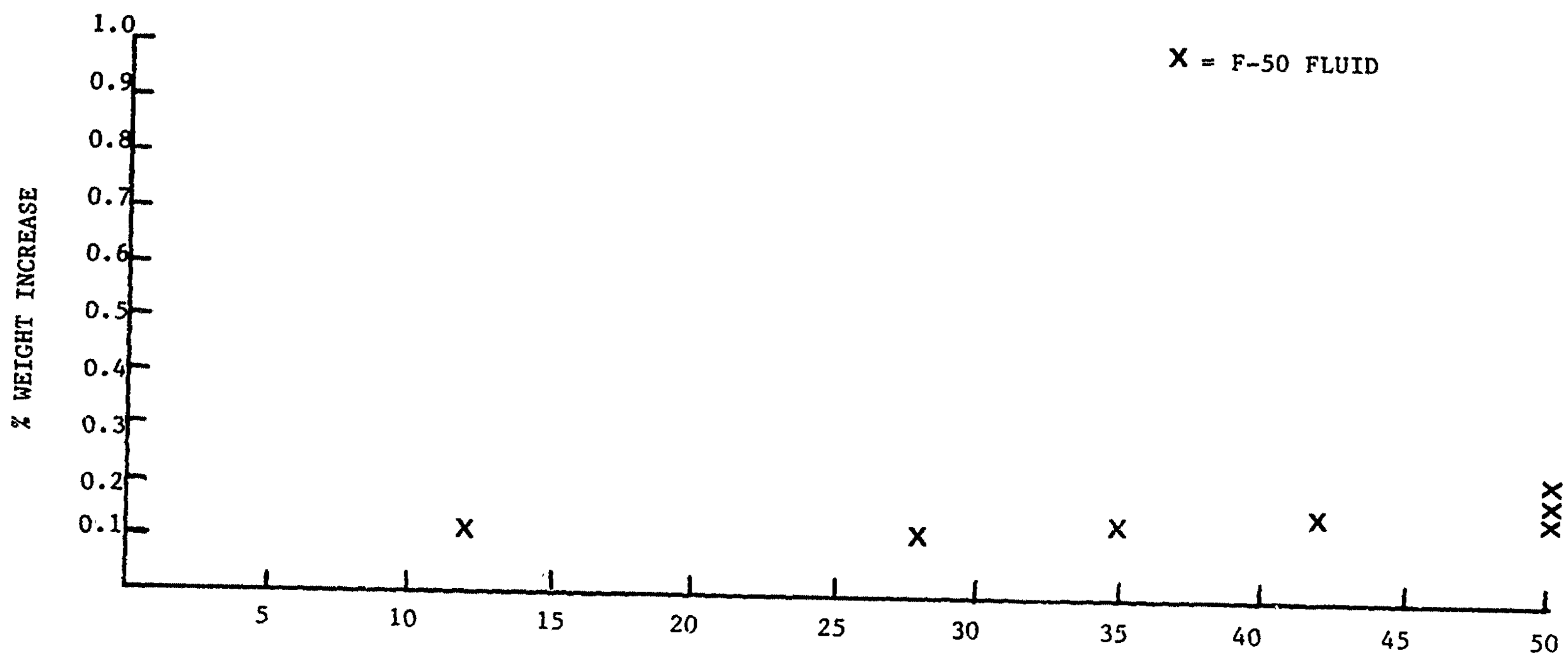


Figure 10. Hydraulic Fluid Immersion Test Placque #3

An interesting point was disclosed in that the two reference specimens that were withheld (50 days in a desk drawer) gained more than twice the weight increments (0.73 and 0.76%) demonstrated by the 5606 figure of 0.32%. These specimens, after being tested for tensile ultimates, were placed in a 150°F oven at 30 inches of vacuum for 2 hours. A weight loss of 0.20% in both specimens was observed indicating that the 0.75% (average) weight gain was due, at least in part, to absorption of moisture.

Weight gain is of little consequence since the polyimide/glass has demonstrated good dimensional stability, hardness retention, and strength retention when exposed to either hydraulic fluids or moisture (Section 2.2.4.2). From the hydraulic fluid exposure tests of Section 2.2.4.2, it would appear that the hydraulic fluid actually protects the plastic from atmospheric weight gain.

2.2.5 Phase II Environmental Testing

At the end of Phase I, tensile test specimens were subjected to various environments for a period of one year. At the end of this period, the specimens were withdrawn, measured, weighed and tensile tested.

The results of these tests are shown in Tables 5. through 10. It can be seen that the samples which were immersed in MIL-H-5606 and F-50 silicone fluid in capped containers were virtually unaffected. Specimens in uncapped MIL-H-5606 containers were also unaffected except for weight gain (apparently due to moisture). Specimens immersed in M2-V silicate ester fluid exhibited some evidence of attack since they experienced a weight loss. Specimens which were left on a roof for one year exhibited severe attack on the side which faced the sun (all gel coat was removed) but little effect on the side which faced away from the sun.

2.3 <u>Phase I Analysis and Design</u>

All detailed calculation summaries for this section are located in the Appendices. Figures 11. through 13. depict the actuator.

2.3.1 Sizing and Optimizing AR (See Section A-1 of Appendix A)

The area (A) of the piston times the radius (R) of the rocker arm generally characterize the torque generation capability ($T = PAR\cos\theta$) of the actuator. Designs often are predicated on this basis. However, in this design, since the rocker arm rolls over the end of the laterally restrained piston, a slightly different approach was used which estimates the effect of friction due to side loading of this piston. It was assumed that 407 in-lb of torque were required at 31.5° fin deflection with a system reservoir pressure of 150 psi. It was further assumed that the pressurized O-ring friction would amount to 15% of the basic driving pressure (a rule of thumb derived from a previous similar actuator design). The 407 in-lb and 150 psi figures are taken directly from the statement of work. The 31.5° figure is taken from a statement of work requirement

TABLE 6.

IMMERSION IN MIL-H-5606 FOR ONE YEAR (CAPPED)

Specimen Number	Dimensional Change (%)	Weight Change (%)	Shore D Hardness Change	Tensile Strength (psi)
1	Nil	+ 0.09	Nil	11,538
2	Nil	+ 0.10	Nil	12,368
3	Nil	+ 0.08	Nil	15,000
5	Nil	+ 0.02	Nil	16,142
7	Nil	+ 0.08	Nil	12,702
8	Nil	+ 0.025	Nil	14,750
9	Nil	+ 0.025	Nil	15,405

NOTE: Specimen #4 was withheld in a desk drawer for one year. Dimensional and hardness changes were nil. Weight increased 0.43%. Tensile tested at 12,307 psi.

Specimen #6 was tested immediately before immersion of the remaining specimens. Tensile tested at 15,420 psi.

TABLE 7.

IMMERSION IN MIL-H-5606 FOR ONE YEAR (UNCAPPED)

Speciman Number	Dimensional Change(%)	Weight Change(%)	Shore D Hardness Change	Tensile Strength(psi)
1	Nil	+ 0.33	Nil	19,546
3	Nil	+ 0.33	Nil	16,854
4	Nil	+ 0.35	Nil	20,141
5	Nil	+ 0.37	Nil	18,808
6	Nil	+ 0.41	Nil	16,301
8	Nil	+ 0.37	Nil	25,958

NOTE: Specimen #2 was withheld and tested before immersion of the remaining samples. Tensile tested at 22,775.

Specimen #7 was withheld in a desk drawer for one year. Weight gain was 0.39%. Dimensional and hardness changes were nil. Tensile tested at 22,304 psi after one year.

TABLE 8.

IMMERSION IN F-50 SILICONE FLUID FOR ONE YEAR (CAPPED)

Specimen Number	Dimensional Change (%)	Weight Change (%)	Shore D Hardness Change	Tensile Strength (psi)
1	Nil	+ 0.14	Nil	14,637
3	Nil	+ 0.15	Nil	20,542
4	Nil	+ 0.14	Nil	15,670
5	Nil	+ 0.15	Nil	18,241
6	Nil	+ 0.08	Nil	17,429
8	Nil	+ 0.13	Nil	12,941
9	Nil	+ 0.06	Nil	17,904

NOTE: Specimen #2 was tested before immersion of the other specimens. Tensile tested at 15,230 psi.

Specimen #7 was withheld in a desk drawer for one year. Dimensional and hardness changes were nil. Weight change was + 0.38%. Tensile tested at 22,658 psi.

TABLE 9.

IMMERSION IN M2-V FLUID FOR ONE YEAR (CAPPED)

Specimen Number	Dimensional Change (%)	Weight Change (%)	Shore D Hardness Change	Tensile Strength (psi)
2	- 0.17	- 0.36	Nil	24,706
3	Nil	- 0.50	Nil	17,819
4	- 0.12	- 0.34	Nil	21,420
5	- 0.12	- 0.46	Nil	20,953
7	Nil	- 0.38	Nil	16,762
8	- 0.10	- 0.37	Nil	18,092
9	- 0.12	- 0.38	Nil	20,211

NOTE: Specimen #1 was withheld in a desk drawer for one year. Dimensional and hardness changes were nil. Weight increased + 0.40%. Tensile tested at 25,162 psi.

Specimen #6 was tested prior to immersion of the other samples. Tensile tested at 16,920 psi.

TABLE 10.

EXPOSURE TO THE ELEMENTS FOR ONE YEAR ON A ROOF

Specimen Number	Dimensional Change(%)	Weight Change(%)	Shore D Hardness Change	Tensile Strength(psi)
1	Nil	- 0.43	Nil	12,658
2	Nil	- 0.32	Nil	22,143
3	Nil	- 0.34	Nil	20,152
4	Nil	- 0.42	Nil	17,903
5	Nil	- 0.44	Nil	22,656
6	Nil	- 0.39	Nil	15,758

NOTE: Specimen #7 was tested prior to exposure of the other samples. Tensile tested at 18,615 psi.

Hardness reduction of 1 to 3 Shore D points was noted on the side which was exposed to the sun.

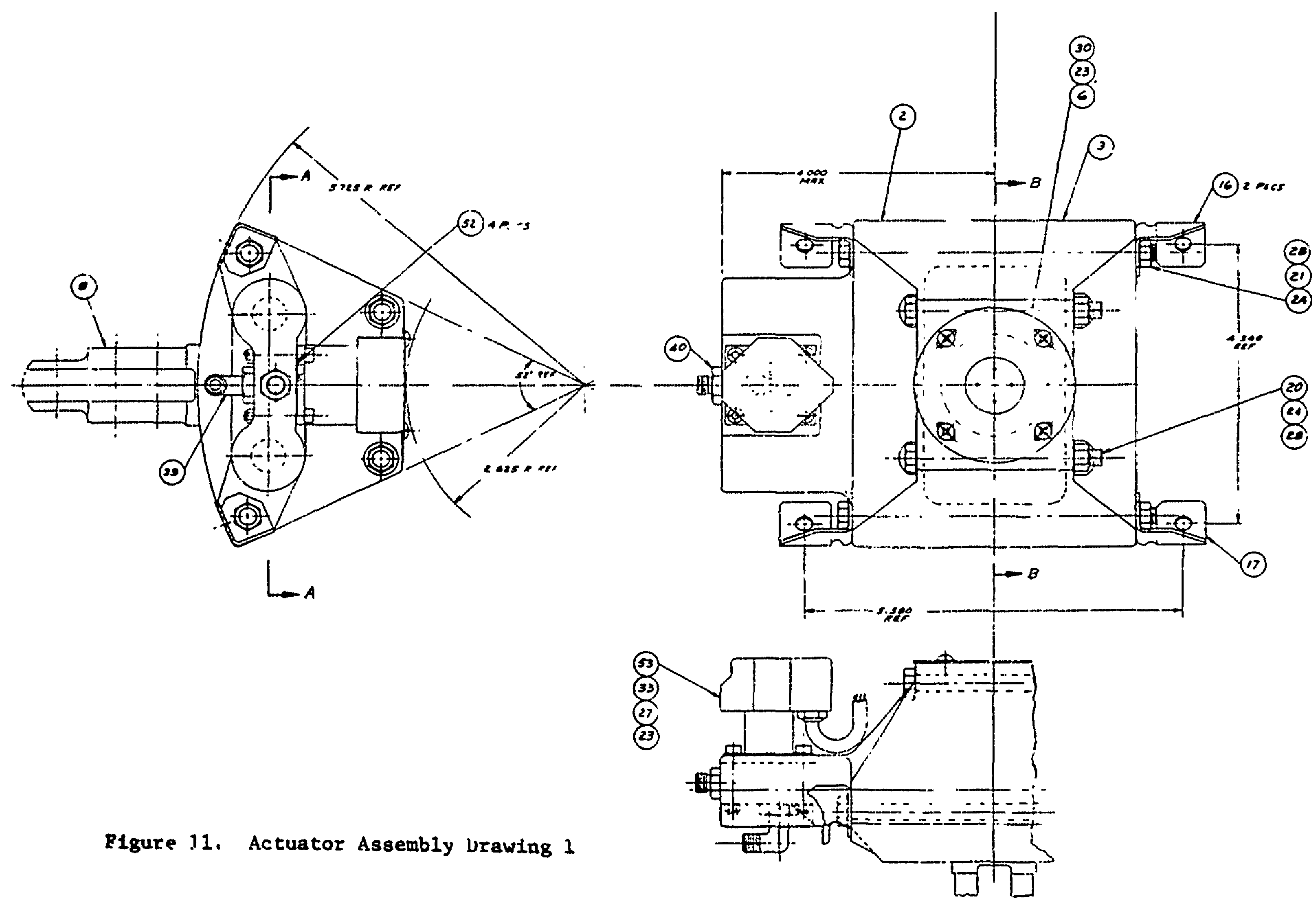


Figure 11. Actuator Assembly Drawing 1

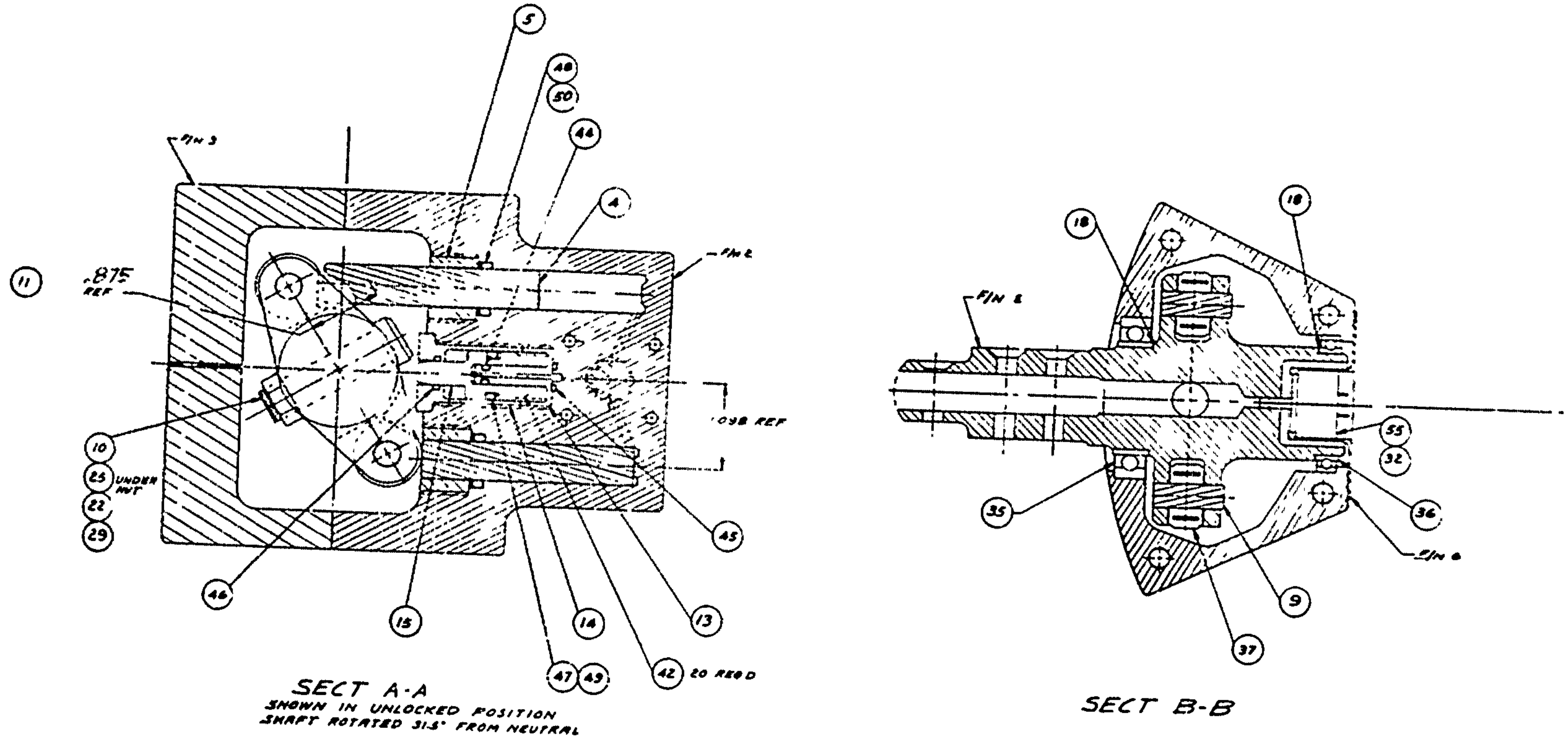


Figure 12. Actuator Assembly Drawing 2

	57				
	56				
1	55	78ESAT502	POTENTIOMETER		NEW ENGLAND INSTRUMENT
	54				
1	53	MODEL [illegible] OR EQUIV	SERVO VALVE		MOOG
4	52	080-04273-78	O-RING		MOOG
	51				
2	50	B-112	BACK-UP RING		
1	49	B-111	BACK-UP RING		
2	48	2-112	O-RING		
1	47	2-111			
1	46	2-011			
1	45	2-010		PARKER SEAL CO	
1	44	2-006	O-RING		
	43				
20	42	80500-018	BELLEVILLE SPRING		ASSOCIATED SPRING CO
	41				
1	40	2-F50X-S	UNION		PARKER-HANNIFIN
1	39	2-C50X-S	ELBOW		PARKER-HANNIFIN
	38				
2	37	SNBC713YJ	BEARING		TORRINGTON
1	36	B542DD	BEARING		
1	35	KP21B	BEARING		FAFNIR
	34				
4	33	NAS1352-06-16P	SCR, SOCKET HD		
1	32	NAS1081-06D6	SET SCREW		
	31				
4	30	MS35206-226	SCREW		
1	29	MS24665-300	PIN, COTTER		
4	28	MS21042-4	NUT		
4	27	MS21042-06	NUT		
	26				
1	25	AN960-716L	WASHER		
8	24	AN960-416	WASHER		
8	23	AN960-6	WASHER		
1	22	AN320-7	NUT		
2	21	AN4-45A	BOLT		
2	20	AN4-26A	BOLT		
	19				
1	18	60500313-006	WASHER, LAMINATED		
2	17	-005	BRKT		
2	16	-004	BRKT		
1	15	-003	LOCK PIN		
1	14	-002	LOCK BODY		
1	13	60500313-001	LOCK STEM		
	12				
1	11	60500311-004	WASHER, LAMINATED		
1	10	-003	LOCK BOLT		
2	9	-002	PIN, ROLLER		
1	8	60500311-001	SHAFT		
	7				
1	6	60500309-005	POT RETAINER		
2	5	60500309-004	PISTON GUIDE		
2	4	60500309-003	PISTON		
1	3	MODIFIED 60500309-001	HOUSING, CAP HALF		
1	2	60500309-001	HOUSING, CYL HALF		
	1				

Figure 13. Actuator Parts List

for at least $\pm$ 30° rotation capability but no more than $\pm$ 33°. (An additional 1.5° was added to the 30° figure for shaft wrap-up and manufacturing tolerance.)

Using the basic assumptions above together with measurements from the detailed drawings, the system pressure necessary to generate the required torque against reservoir pressure was determined (2206 psi). Next, a calculation was made to determine the axial friction force (127.5 psi when converted to pressure) on the piston due to the couple generated by the rocker arm roller being offset from the centerline of the piston. This friction pressure when added to the above system pressure gives a basic driving pressure. An additional 15% is then added to account for O-ring friction, thus giving the final system pressure of 2700 psi. (The test of unit #3 of 6.2.1 indicated a full pressure value of 2600 psi, with zero backpressure, using plastic pistons. Steel pistons reduced this pressure to approximately 2000 psi.)

Section A-2 of Appendix A is a study that was performed at the outset of the program which optimized the bore of the actuator at 1/2 inch.

2.3.2 Valve Sizing (See Section A-3 of Appendix A)

The statement of work requires a no load velocity of 350°/second. This corresponds to a maximum valve flow of 1.42 cu in per second. It was assumed that this velocity must be met at -65°F. At this temperature, the calculated pressure drop in the actuator is 620 psi. Since the reservoir backpressure is 150 psi and system pressure is 2700 psi, this leaves 1930 psi available for valve drop. The no-load valve rating would then be 1.87 cu in per second plus zero, minus 10% at a valve drop of 2700 psi. This would result in a no load, no backpressure velocity at room temperature of 461°/sec.

2.3.3 Piston Side Load and Rocker Arm Friction Analysis (See Section A-4 of Appendix A)

Section 2.3.1 discussed a roller being used on the rocker arm tip. If this roller were replaced with a rocker-arm-fixed contact device, this device must slide at least over a portion of the piston stroke because the point of contact between the piston and the device does not remain on the line of centers of rotation of the rocker arm and piston system. Although this sliding effect can be minimized as the piston reaches its outer stroke limit, some sliding motion must occur in other portions of the stroke. An estimate of this sliding force across the top of the piston at full torque is 397 lb based on a sliding surface interface coefficient of friction of 0.78. Although MoS_2 lubricant would lower this coefficient, it is not known whether or not MoS_2 would remain functional after many cycles of operation. This sliding friction force must be reacted to the cylinder walls. These reactions, in turn, generate axial forces on the piston. The calculations in Section A-4 of Appendix A show that a total axial friction force of 266 lb could be developed which would require 1330 psi additional pressure to overcome. The cylinder wall and piston wear under these conditions were not estimated.

The roller concept was instituted; however, if materials with sufficiently low coefficients of friction can be found, a non-roller type contact device can easily be tried.

2.3.4 Piston/Piston Guide Material Selection (See Section A-5 of Appendix A)

Kinel 5511 polyimide/25% graphite compound was initially selected for the piston material based on an advertised low coefficient of friction (0.10 to 0.25) and a calculated strength sufficient to withstand roller forces without using a steel button on the piston top. (Difficulties with the molding press, however, precluded the use of this material. See Section 5.2.6. The pistons for the test actuators were machined from steel.)

The existing piston guide design was selected for easy O-ring groove fabrication. Also, the guide bore can be molded very accurately (Kinel 5504) so that the piston has sufficient support. The piston guides used in the test actuators were machined from Kinel 5504 to save mold fabrication costs.

The O-ring is placed in the bore rather than on the piston, primarily because the bore may have glass fibers protruding through the walls which could damage the ring. The Kinel 5511 pistons described above have no glass fibers.

2.3.5 Piston Stops

The inlet ports which carry fluid in and out of the cylinder are designed so that the piston is (at least partially) snubbed as it bottoms. The cylinder port is essentially progressively closed off (at high piston velocities) as the piston bottoms, which is intended to decelerate the piston gradually. Calculations indicate that if a constant deceleration could be achieved over the diameter of this hole, the force necessary to effect this deceleration with the existing inertia would be on the order of 8 pounds. By far the greatest load would occur through the servo itself which would react to apply 540 pounds at zero velocity at the stop. The stop itself should be capable of withstanding 9400 pounds.

2.3.6 Shaft Lock Design (See Sections A-7 through A-10 of Appendix A)

Initial design efforts placed the fin locking device on the forward side of the actuator such that the lock pin protruded into the fin. Although pin shearing forces could be kept minimized in this design, the following compelling factors suggested a different approach:

1 The lock pin hydraulic cylinder is difficult to mold since resin flow within the mold requires circumferential flow around a cylinder-forming plug such that a weld or re-knit is formed upon mold closure. This weld line is a potential point of weakness in a highly pressurized cylinder.

2 The plug referred to in 1 must withstand considerable force perpendicular to its axis upon mold closure so that plug bending, and hence, cylinder deformation could occur.

3 With the pin locking cylinder located on the forward side of the actuator, some means must be designed to supply it with high pressure hydraulic fluid even though all other hydraulic requirements are present in the aft section of the actuator.

In view of these factors, the lock pin was placed between the cylinder bores, locking into the shaft. The pin and mechanism were designed to restrain a torque load of 500 in-lb and to be disengaged under that load with application of 85% of system pressure. The pin actuating cylinder can apply 413 lb of withdrawal force at 2700 psi pressure, assuming a 5/8 inch diameter piston. Appendix A contains a detailed calculation summary involving hydraulic forces, friction forces, O-ring forces, spring forces, and pin rejection forces due to pin taper. Also included are spring design and stress calculations on all critical parts.

2.3.7 Potentiometer Mount Design (See Section A-11 of Appendix A)

The potentiometer mount is a hat shaped section into which the pot is bonded. The pot shaft is attached by a setscrew to the actuator shaft. The pot mount is attached to the actuator body with screws which protrude through slotted holes in the hat brim so that the pot can be rotated before lock-down to center the shaft.

The axial end play in the actuator shaft can be shimmed to a maximum of 0.007 inch and, since the pot shaft is rigidly attached to the actuator shaft, the pot mount must absorb this play. The maximum axial force transmitted to the pot shaft is designed to be 11 lb, causing a maximum stress in the holder of 6160 psi which is acceptable for Kinel 5504, the material which would be used in production. (The test actuators used mounts which were machined from Teflon.) The pot manufacturer has been contacted regarding allowable pot shaft axial loads. The two relevant components inside the pot that must accept this load are a ball bearing and a snap ring. The snap ring is rated at 40 lb while Martin Marietta Aerospace design manuals rate the bearing axial thrust limit at 22 lb. The shear stress in the bonding material which bonds the pot to the holder is 8 psi.

2.3.8 Bearing Loads and Case Bolting Design (See Section A-12 of Appendix A)

Shaft loads are transmitted to the actuator case by two ball bearings, each capable of taking thrust loads. The outboard bearing is a KP21B Fafnir standard precision control bearing with a radial load limit of 9840 lb and a thrust limit load of 4400 lb. The inner bearing is a B542DD Fafnir standard precision control bearing with a radial load limit of 5950 lb and a thrust limit load of 2700 lb. Both bearings are sealed and were selected primarily because of size and cost. Since both bearings can take thrust loads, no shaft retainer is necessary with the split case design. This, together with insertion of the pot into the shaft end, allowed maximum spread between the bearings so that case stresses could be minimized. Prefabricated plastic journal bearings capable of taking both radial and thrust loads were found to cost more than the ball bearings called out above.

The actual bearing loads are much lower than the bearing ratings. The outboard maximum load is 915 lb and the inboard maximum is 782 lb based on a fin load which imparts 400 in-lb of torque load and a 1600 in-lb bending load to the shaft at a fin deflection angle of 31.5°.

2.3.9 Shaft Design and Stiffness (See Table 11.)

Two shaft designs were pursued, a cast aluminum shaft and a composite Kinel 5504/steel shaft. The aluminum shaft cost $15 more, so the composite shaft design was adopted as the production shaft. (For the test actuators, the shafts were machined from 17-4 steel to save mold costs. Steel was used rather than aluminum because steel more closely matches the stiffness of the composite shaft.)

Ideally, a shaft molded completely of Kinel 5504 was desired but not feasible because both the shaft design and loads were specified. The stress developed by the combined 1600 in-lb bending moment and the 400 in-lb torsional moment was too great at the fin attachment bolt hole nearest the missile skin (Table 11., item 3). To strengthen the shaft in this area and to enhance shaft torsional stiffness, a 4130 normalized steel sleeve was designed which could be machined to size and then inserted into the mold prior to mold closure. Several holes in the steel allow plastic to form pins and knit firmly to the steel, thus transmitting bending and torsional loads. The temperature coefficient of expansion of the steel is 6.3×10^{-6} in/in/°F, and that of Kinel 5504 is 7.8×10^{-6} which will result in the plastic rocker arm shrinking onto the steel shaft as the combination cool from a molding temperature of 450°F. This shrinkage stress will be approximately 1700 psi in the plastic when cooled from 450°F to -65°F. (Experience gained in molding actuator cases increases confidence in the moldability of the composite shaft.)

The angular stiffness of the steel sleeve when calculated from the center of the rocker arm to the inboard edge of the fin slot is 1,916,000 in-lb/radian or 0.00003 deg/in-lb. This stiffness is quite adequate to insure the required closed loop stiffness.

2.3.10 Case Stiffness

A detailed stiffness analysis of the actuator case was not made. However, a simplified "order of magnitude" computer analysis was made which assumed a hollow trapezoidal case with 1/4 inch walls, cantilevered at the large base end (Figure 14.). A load of 538 lb at the location of the inboard bearing in the direction of least stiffness gave a deflection of 0.0015 inch at that point. (538 lb corresponds to the maximum load expected in this direction.) The bearings are grossly understressed (see Section 2.3.8) and, according to the bearing manufacturer's information, have approximately 0.001 inch clearance between balls and races when not pressed into steel. Since these bearings are seated in plastic instead of steel, a deflection of 0.0015 inch can be easily accommodated by the plastic bearing housings.

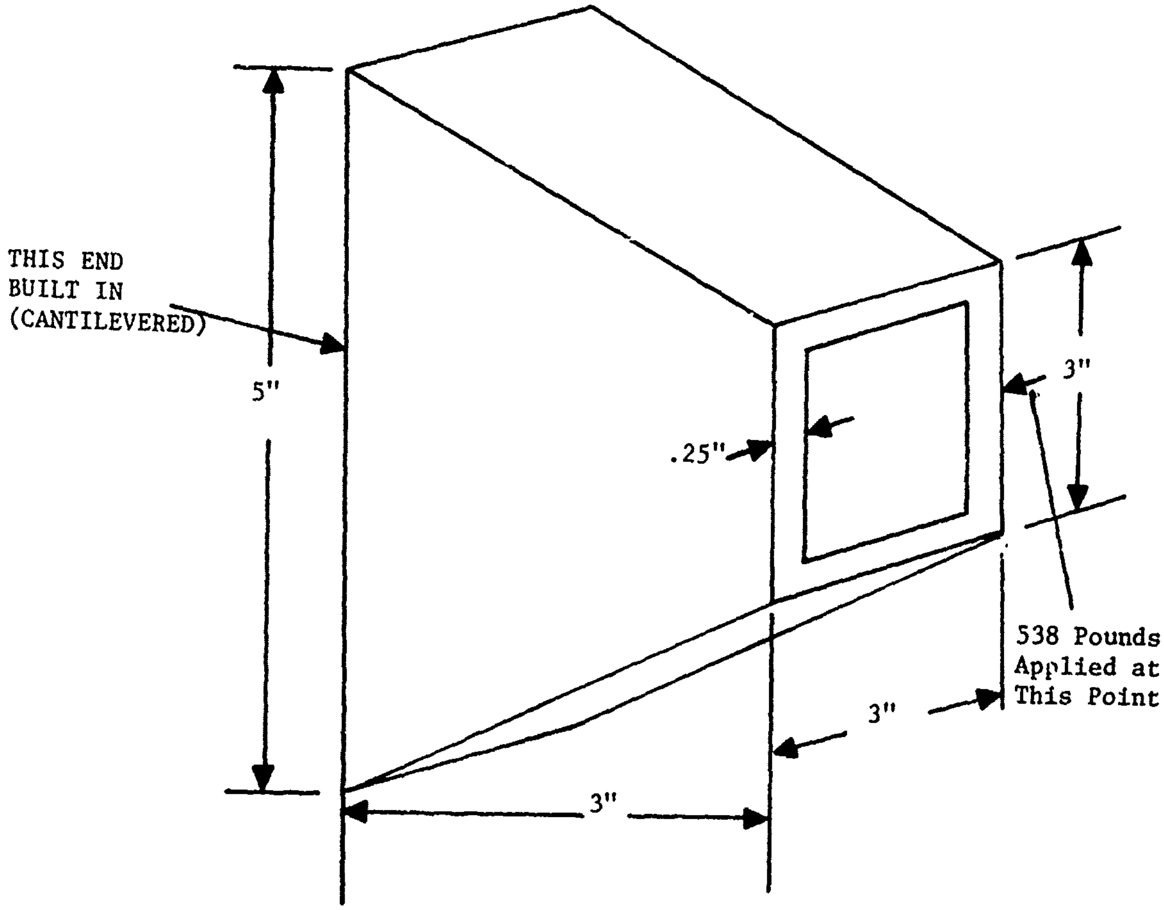


Figure 14. Approximate Case Representation for Stiffness Study

2.3.11 Stress Analysis Results

No detailed computer analysis of stresses was made. However, all areas (20) deemed important were checked by stress personnel using conservative techniques. The calculated stresses together with the allowables are summarized in Table 11.

Items 2, 3, and 11 of Table 11. show that the all-plastic shaft and the stress inside the cylinders at burst pressure are problem areas. The all-plastic shaft concept was dropped in deference to the steel sleeve shaft. The cylinder stress problem is treated in Section 2.2.3.1. All other calculated stresses are within demonstrated limits.

2.3.12 Major Procured Parts

The servo valve for the production actuators is a low cost (approximately $200.00) version of the Moog Model 30 series valve. Performance criteria are to remain essentially the same except that the low cost servo valve is wet coil and has a service life of approximately 25 hours. At least two other manufacturers make valves which will fit the available envelope. (For the test actuators, the low cost valves were not available in time so Moog Model 30 valves were used.)

Also called out is a low cost potentiometer ($16.00) made by New England Electronics (see Section 3.1). Two other manufacturers make pots which will fit the envelope (at additional cost).

2.3.13 Weight

The weight of the actuator is 5.15 pounds with a steel shaft. The steel shaft weighs 1.64 lbs. The composite shaft would weigh 0.70 lbs giving a production actuator weight of 4.2 lbs. It is expected, however, that in a production design, some weight savings could be realized since a detailed computer stress and stiffness analysis could be made. Overly conservative parts such as the forward case half might then be trimmed to save weight. The major contributors to weight are the case (1.44 lb), shaft (0.70 lb), servo valve (0.50 lb) and shaft bearings (0.21 lb).

2.4 <u>Phase I Martin Marietta Review/Problem Areas</u>

During Phase I the design was subjected to an in-house review during which experienced engineering personnel critiqued the design. The following suggestions were considered:

<u>1</u> "Locktite" bearing mounting compound could simplify bearing retention and piston guide plug assembly. (This was found to be unnecessary during Phase II.)

<u>2</u> A vent path for the fin lock plunger was provided by a drilled hole in the spring retaining plug.

TABLE 11

ACTUATOR STRESS POINTS CHECKED

	Area Checked	Calculated Stress	Allowable
1.	Box section at 1/4" wall midway between bearings. 400 in-lb torsion, 1600 in-lb bending	Combined stress is 541 psi	At 270°F F_{tu} = 11,500 psi F_{ty} = 10,000 psi
2.	Solid shaft with 400 in-lb torsion and 1600 in-lb bending perpendicular to fin at root of 1.25 in section	Critical stress is 12,100 psi tension	At 270°F F_{tu} = 11,500 F_{ty} = 10,000 } Plastic F_{tu} = 35,000 F_{ty} = 27,000 } Cast Aluminum
3.	Solid shaft as in 2. above but at bolt hole nearest missile skin	Critical stress is 14,400 psi at C.L. of bolt hole	Same as 2. above
4.	Steel sleeve shaft (assumes steel takes all critical loads)	112,884 psi bearing stress in bolt hole in shaft (only near-critical point in steel)	4130 normalized F_{by} = 129,000 psi
5.	Plastic rocker arm at burst pressure of 6750 psi	10,700 psi in bending. 9320 psi in bearing at roller pins	At 80°F F_{tu} = 11,500 psi F_{cu} = 32,500 psi (published figure)
6.	Shear stress in plastic pins in plastic rocker arm. Resisting bending moment from steel into plastic (See 2.3.9).	Shear stress is 5890 psi, double shear	Test value for double shear is 18,000 psi
7.	Tension load in inner bolts due to assumed load of 400 in-lb. torsion and 1600 in-lb bending plus piston force at operating system pressure	Load in bolt is <632 lbs	Tension capacity of MIL-S-7742 bolt is 1837 lbs

TABLE 11 (Cont'd)

	Area Checked	Calculated Stress	Allowable
7a.	Steady state compressive stress under inner and outer bolt head due to bolt torquing	Compressive stress is 1700 psi. Bolt tension load is 250 psi	Compression stress ultimate is F_{cu} = 32,500 (published)
8.	Tension load in outer bolts due to condition 7. above	Load in bolt ~244 lbs	Capability same as 7. above
9.	Shear in bolts under attachment lugs	Very low estimate; less than 150 lbs due to curved surfaces at interface between missile skin and actuator	MIL-S-7742 bolts single shear load = 3190 lbs
10.	Tension stress in mounting brackets	Maximum stress in steel brackets is 21,000 psi	4130 normalized F_{tu} = 95,000 psi F_{ty} = 75,000 psi
11.	Calculated inner hoop stress in cylinders at burst pressure	Tension stress is 10,266 psi	Average burst pressure demonstrated by test cylinders is 10,000 at 80°F. Tensile tests show F_{tu} = 11,500 psi
12.	Stress in cylinder walls when piston bottoms at full velocity	Shear stress in end plug walls approximates 1146 psi	Allowable shear stress by test is 18,000 psi
13.	Stress in plastic due to valve bolt force	Compressive stress is 933 psi	F_{cu} = 32,500 psi (published)
14.	Bearing stress of roller on top of piston at proof pressure (See A-5)	Calculated bearing stress is 24,059 psi	Brinell bearing hardness of piston plastic is 43,000 psi (Advertised)
15.	Maximum stress in pot. holder (See A-11)	σ_t = 6160 psi	At 270°F F_{ty} = 10,000 psi

TABLE 11 (Cont'd)

LOCK PIN STRESSES

	Area Checked	Calculated Stress	Allowable
1.	Lock pin shear at small dia. at instant of disengage of 500 in-lb load (See A-7B)	Shear stress is 10,660 psi	440F steel S_{sy} = 157,000 psi
2.	Bearing stress in lock pin cylinder (-012) (at lip). Conditions of 1. above (See A-7C)	Bearing stress is 66,880 psi	303 stainless 1/4 hard S_{BY} = 125,000 psi
3.	Bearing stress in plastic at outer lip of pin cylinder (See A-10)	Bearing stress is 6700 psi	S_{cu} = 32,500 psi for plastic (published figure)
4.	Pressurization hoop stress in pin cylinder at burst pressure (See A-10)	Hoop tension stress is 43,339 psi	303 stainless 1/4 hard S_{ty} = 67,000 psi
5.	Pressurization stress in plastic due to 4. above (See A-10)	Hoop tension stress is 3178 psi	Plastic at 270°F F_{tu} = 11,500 psi F_{ty} = 10,000 psi

The following items of concern were noted during this Phase I critique:

1 Heat generation due to operation of the system (complete parameters are not known) could cause temperatures to rise higher than 270°F. Such things as actuator integrity and hydraulic fluid capability should be determined. (The actuator performed satisfactorily during the hot tests of Phase II.)

2 Fiber orientation within the plastic affects such parameters as strengths and coefficient of thermal expansion.

3 More material testing should be run regarding, for instance, effects of high and low temperature, long term exposure to hydraulic fluids, exposure to moisture and ultraviolet light, long term creep, and bonding agent integrity.

4 More knowledge should be gained regarding moldability, especially in the area of the shaft since some welding or re-knitting of plastic material is required to bond the plastic to the steel. (The molding experience gained during Phase II increases confidence in the moldability of the composite shaft since void-free parts were obtained which accurately molded such things as screw threads and O-ring seats.)

5 Permeability and porosity of the plastic at high pressures and temperatures should be further investigated. Initial testing in this regard revealed no problems, however, an actual unit with all drilled passages would provide the only fully acceptable proof. An allied problem relative to thread tapping is discussed in Section 2.2.3.4. (No problems were experienced during Phase II with drilling or thread tapping. The exact mechanics of seepage encountered during Phase II should be investigated.)

6 The potentiometer mounting scheme requires pot shaft thrust carrying capability which has never been tested. Although analytically feasible, problems could develop which will not surface until the final design is tried. A slip fit joint from actuator shaft to pot shaft can easily be designed at increased cost. (No problems were encountered during Phase II with the potentiometer mount.)

SECTION 3

ELECTRONICS

3.1 Low-Cost Potentiometer

One of the basic considerations in selecting a potentiometer is cost. In this case the factor which affects cost most strongly is the power dissipation required at 270°F. That is, several vendors make pots that will meet the originally proposed continuous power dissipation requirement of 0.45 watts at 270°F, however, they cost approximately $40 to $50. New England Electronics manufactures a pot in the $15 range which can be operated at 0.3 watts at 270°F predicated on a duty cycle of 5 minutes power on and 5 minutes power off for 25 hours. The 2000 ohm resistance value in the proposal was changed to 5000 ohms which lowers the power rating to 0.18 watts. This pot has a linearity of ± 1% over ± 33° of travel.

These pots are designed to meet MIL-R-12934 standards which include 50g shock and 15g vibration environments. Although they have not been tested to MIL-STD-810B, it is presumed that the 45g shock and 9.36g vibration requirements called out in MIL-STD-810B can be met. Similar presumptions are made regarding the other environmental specifications. If these specifications cannot be met, several other vendor's units can be substituted directly (at increased cost).

3.2 Electronic Circuitry

The electronics configuration is shown in Figure 15. The initial design requirements were:

1 The electronics shall deliver a maximum current of ± 15 milliamperes (ma) to the servovalve parallel coil combination whose resistance is 250 ohm at 70°F and whose inductance is 2.2 henries. The electronics shall be capable of supplying the maximum current (± 15 milliamperes) over a frequency range of 0 to 100 Hz.

2 The ratio of servo valve coil current to vane deflection shall be 11.25 ma/degree.

These requirements were initially implemented; however, in an effort to improve frequency response, the amplifier gain was raised 30%. The loop was stable under all conditions so this gain was used.

The control actuator electronics are of a current feedback configuration which utilizes a high gain amplifier to sum the feedback potentiometer voltage (e_F), the input voltage (e_I), and the current sense signal.

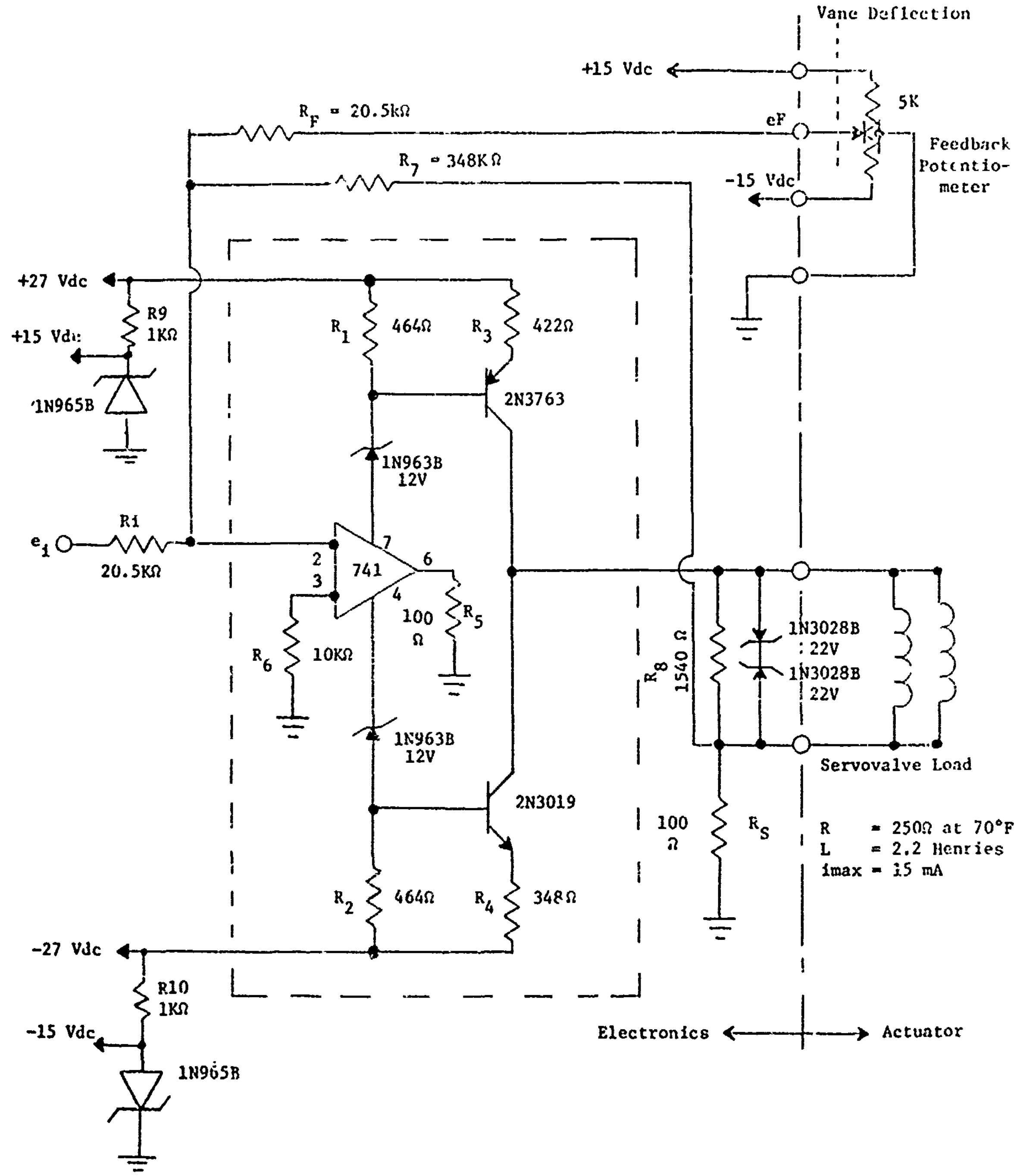


Figure 15. Electronics Schematic

The latter is the voltage developed across the sense resistor (R_S). It is proportional to the servovalve load current. Consequently, the output of the amplifier provides a load current to the servovalve coils which is proportional to the difference of the voltages e_F and e_i. For the configuration of Figure 15., the feedback potentiometer scale factor is 0.08 Vdc/degree, the amplifier gain to a feedback signal being 170 mA/Vdc. The maximum source impedance of the potentiometer is 0.63K, the loading error being 2%. This error could be reduced by rescaling the resistor network or can be eliminated by the addition of a non-inverting unity gain amplifier but is not considered important in view of the lack of a linearity limit in the statement of work.

The maximum voltage output of the amplifier is governed by the requirement that the amplifier deliver 15 ma into the servo load at 100 Hz. The amplifier load can be characterized by a 2.2 henry inductance in series with a 490 ohm resistor, the latter consisting of a 100 ohm sense resistor in series with the coil resistance at 270°F. The required maximum amplifier output is ± 22 volts. Short circuit protection is achieved by resistors R_3 and R_4 as well as the inherent short circuit capability of the 741 operational amplifier. The zener diodes across the servo valve coils protects the amplifier from the voltage surges due to the coil inductance. Although shaping networks are not required in this configuration, there is growth potential for the inclusion of such networks in the control actuator electronics.

The feedback potentiometer is a 7/8 inch diameter, servo mount, single turn, center tapped unit, the size being governed by packaging considerations. The center tap is grounded to set the electrical null to within ± 0.1 degree of the output shaft center position independent of excitation voltage variations. The use of a film potentiometer minimized resolution errors, the resolution being less than ± 0.003°. The temperature requirement is met by a full travel (340°) conductive plastic potentiometer of 5K total resistance with 30 Vdc (+15 and -15 Vdc) across the unit (see Section 3.1). The output scale factor is 0.08 Vdc/degree and the maximum source resistance as seen by the electronics is 0.63K.

The pertinent final loop gain information follows:

Amplifier Gain	= 170 ma/volt
Feedback Gain	= 0.08 volt/degree
Actuator Displacement	= 0.22 cu.in./radian
Servo Valve Gain at 1900 psi drop	= 0.095 cu.in./sec/ma.

SECTION 4

COST

Representatives from the Martin Marietta Engineering Prototype Lab, Manufacturing Engineering, Financial Estimating, Procurement, and R&D Engineering established a coordinated effort to arrive at a reasonable estimate for the cost of the actuator. Those costs (based on 2000 units) are presented below in the order called out in 4.2.4 of the "Stateme of Work." See Appendix C for a detailed breakdown.

The total price of $1,303,447 represents the production costs (recurring and non recurring) for 2,000 actuator units. ($651/unit) The time phasing for this price is June 1974. Martin Marietta Plant Wide Labor rates (June 1974), as published in Section 1.12.3.1, page 2 of 2, of the Martin Marietta Estimating Guidebook, were utilized in addition to specific rates where applicable. Overhead and burden rates used are those negotiated with DCASO on 18 October 1973 for CY 1974.

All rates were applied to detailed bottom-up division estimates and to a detailed procurement bill of materials. The bill of materials used was priced with applicable Normal Production Allowance and bulk material factors, and consisted of vendor quotations equal to 98% of the bill of material value.

4.1 Tooling (Design, fabrication and tryout of preform and finish molds, including materials)

Fabrication of detail tooling and test fixtures, including materials.

Price $107,855.00

4.2 Manufacturing Materials

Includes all raw materials and purchased parts required for the plastic actuator without control valving electronics. Includes 20% rejection rate of all plastic parts.

Price $792,655.00

4.3 Fabrication

Includes tool maintenance, manufacturing, production control subassembly inspection and shipping. Includes 20% rejection rate of all plastic parts.

Price $342,541.00

4.4 Test - Engineering Design and Qualification Tests

Price $12,825.00

4.5 Production Test

Piece part testing and final assembly, quality inspection. Includes 100% test of all actuators at proof pressure.

Price $47,471.00.

4.6 Comparative Cost Estimate

The cost of a production metal actuator of the size and complexity of the subject plastic actuator was estimated based on an actuator presently in production at Martin Marietta Aerospace. Table 12. presents the results of this estimate.

TABLE 12.

ACTUATOR UNIT COSTS (QUANTITY - 2000)

	MMA Production Metal Actuator	MMA Production Plastic Actuator	AFFDL Phase I Plastic Actuator
Tooling, fab & misc procurement	$ 869	$ 471	$ 424
Servovalve	337	337	200
Cost with profit (no testing)	1206	808	624
Production & qual testing allowance	27	27	27
Total cost	$ 1233	$ 835	$ 651

SECTION 5

FABRICATION, PHASE II

5.1 Mold Description (See Figure 16.)

Figures 16., 17., 18., and 19. are pictures of the finished mold. By prior agreement in the interest of economy, only one mold was fabricated. The critical part of the actuator is the aft case half which contains all hydraulic portions of the complete actuator. Molding of the composite shaft, potentiometer retainer and piston guides was not attempted but is considered feasible based on the success of molding the aft case. The aft case mold also contains cavities for molding pistons; however, problems with the pressure control on the molding press precluded molding the pistons (see Section 5.2.6). The press used was a Dake 150 ton hydraulic press with heated platens (55 inches of useable daylight).

The complete mold consists of the following parts:

1 Cavity

2 Punch

3 Inserts

4 Ejection mechanism

5 Loading zone

6 Mold bases and closure guide pins

7 Cal-rod heaters and thermocouples.

5.1.1 Cavity

The cavity is the female negative of the aft portion of the actuator case (see Figure 17.). This cavity was formed by first machining two graphite electrodes (see Figure 19.) which have the identical exterior shape as the aft case. These two electrodes (roughing and final) were used to machine the cavity by the EDM process (Electron Discharge Machining). The cavity block is made from de-gassed 4130 mold steel.

5.1.2 Punch

The punch is shown in Figure 18. It consists of a large steel block which was machined to fit closely (within 0.020 inches all around) inside the loading zone cavity shown in Figure 17. To the base of this block is mounted a case hardened steel foot plate one inch thick which fits the

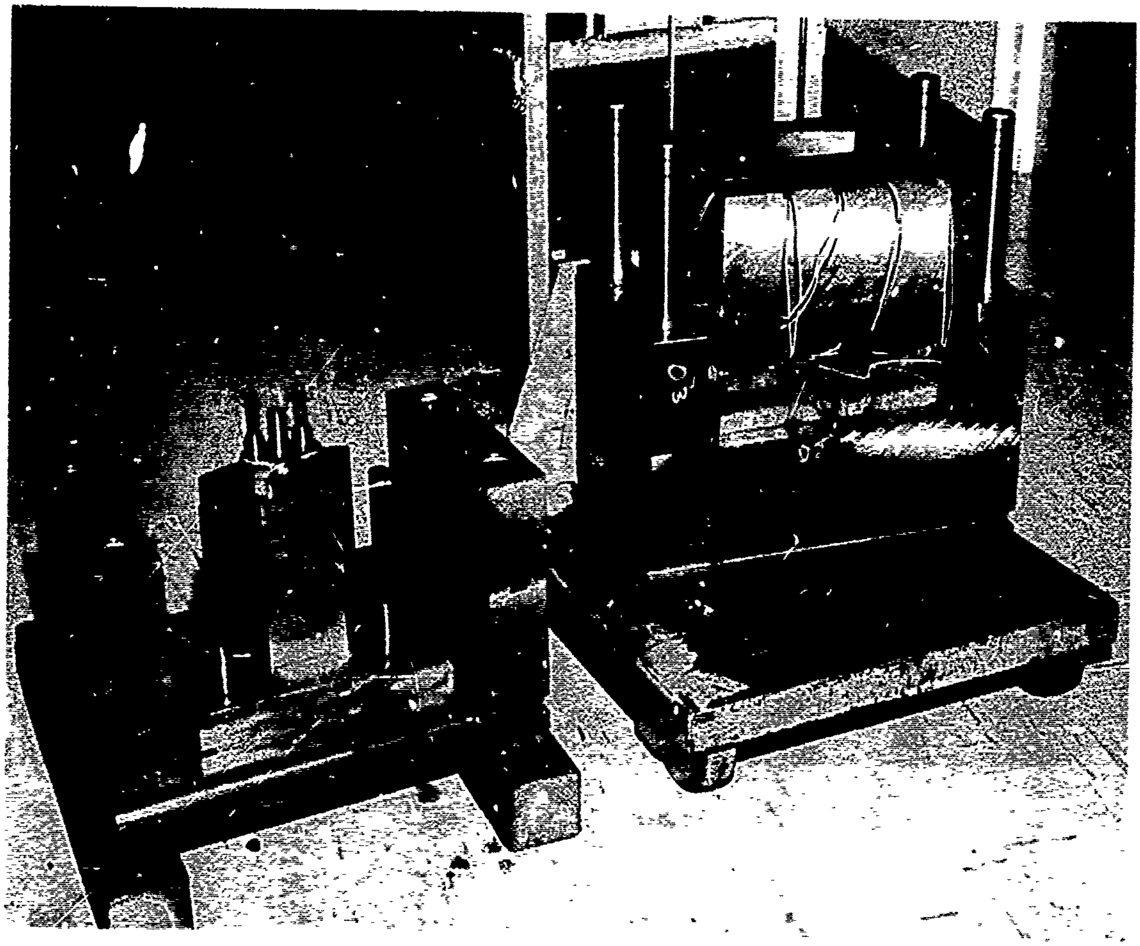


Figure 16. Mold Parts

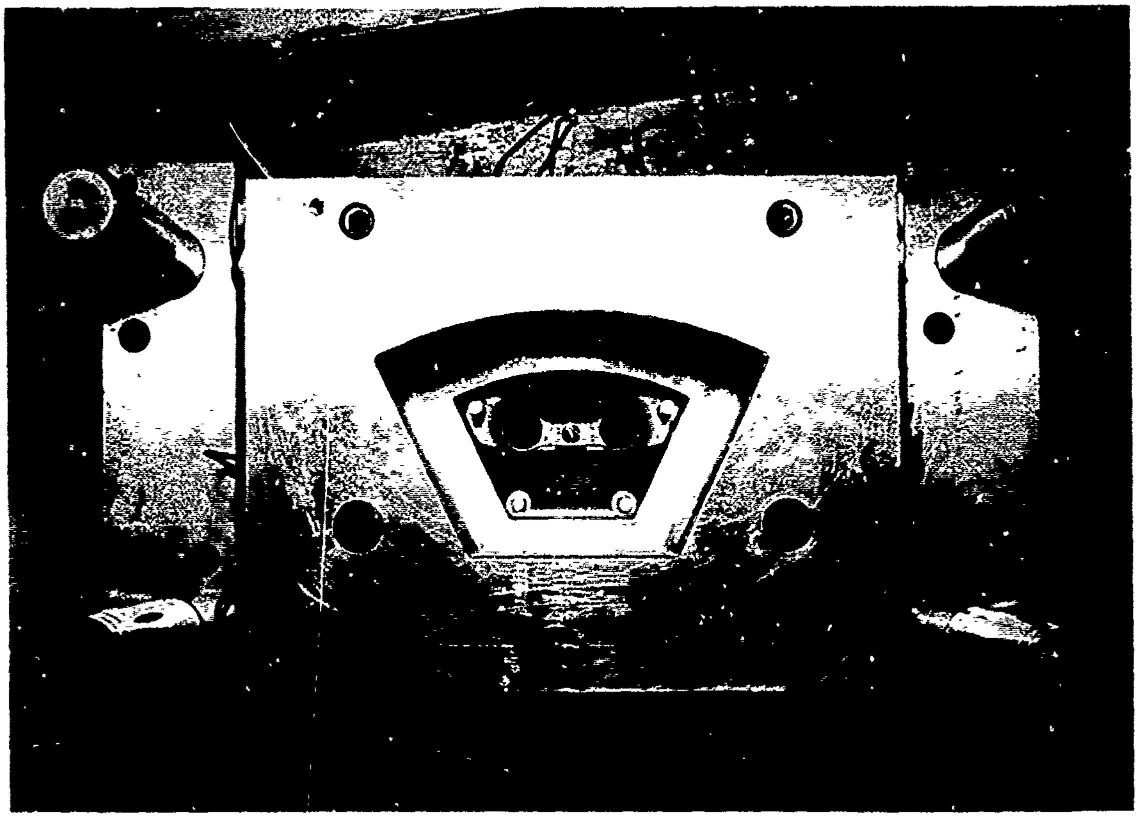


Figure 17. Cavity

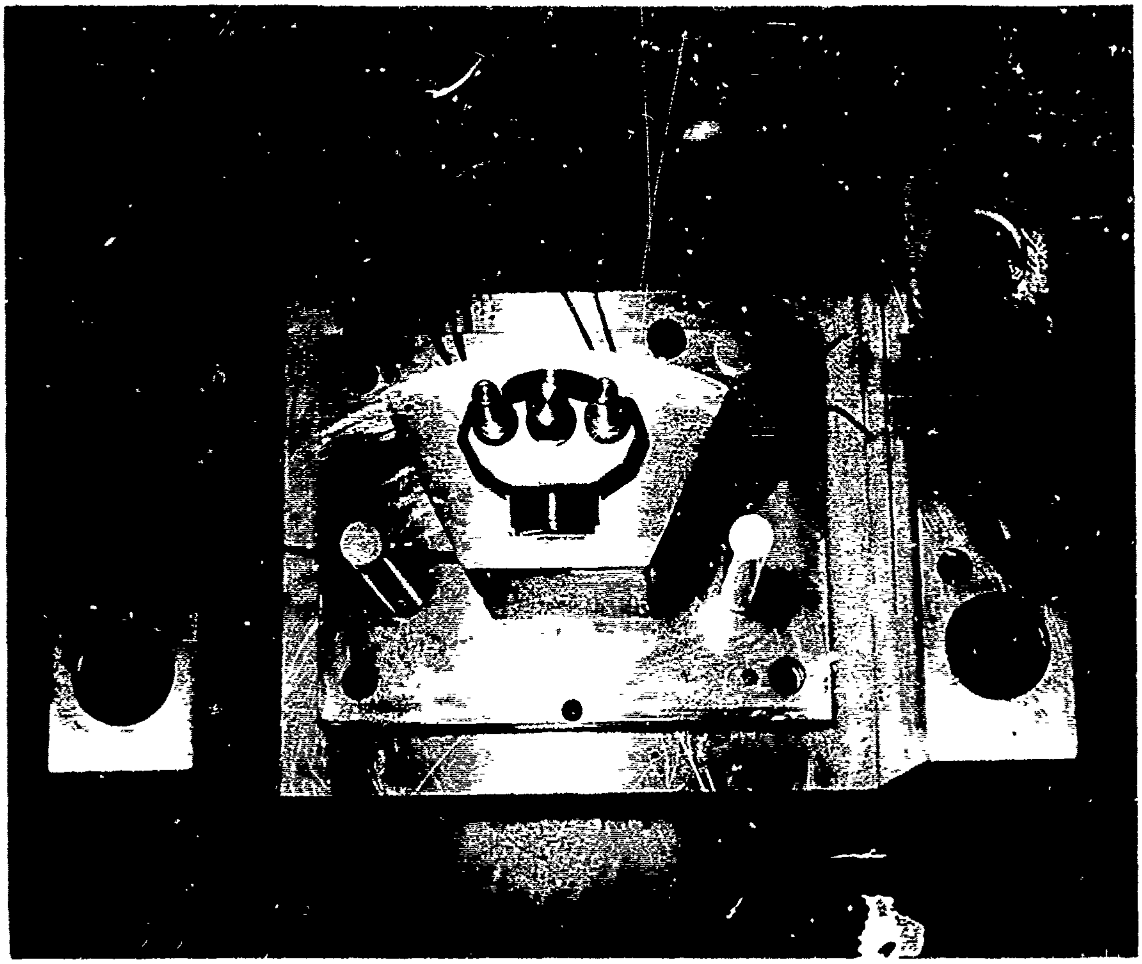


Figure 18. Punch

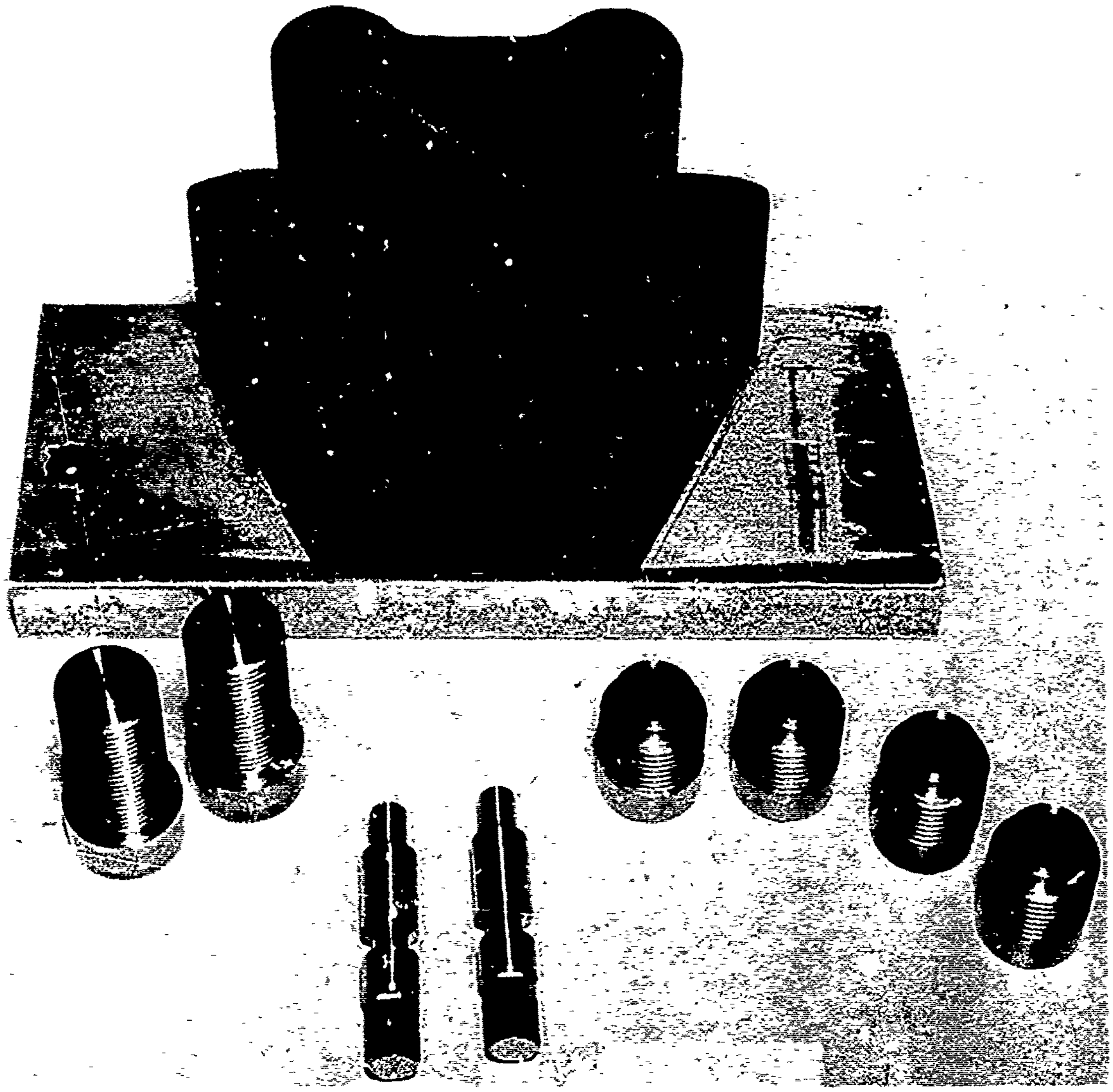


Figure 19. Mold Inserts and EDM Electrode

loading zon cavity wit 0.005 inch tolerances around its periphery. To the base of the foot pl e are mounted the steel pins (removable) which form the cylinder cavit s and the machined block which forms the inside the aft case cavity and aft bearing grooves. All dimensions were made identical to the d sign nts. No draft was machined on the cylinder pins.

5.1.3 Inserts (See Figure 19.)

In order to form the O-ring seats and w threads associated w the piston guides a in lock housing, thr serts were machined at slip over the pins from the m . In addition, a fourth insert is inserted into the bottom of the cavity which forms the return port threads and O-ring seat. This fourth insert, when inserted in the cavity, is captivated there by a pronged fork which is inserted into the side of the cavity so that after the part is molded, the punch can be withdrawn first while the molded part is held in the cavity. The pronged fork is then withdrawn releasing the molded part so that it can be ejected by the ejection mechanism.

5.1.4 Ejection Mechanism

Two 7/8 inch chrome plated movable steel pins protrude through the bottom of the cavity directly under the cylinders. These ejection pins rest on a steel bar which passes under the cavity so that as the bar is raised, the pins push the molded aft case part out of its cavity.

5.1.5 Loading Zone

The loading zone is best shown in Figure 17. It is mounted on the top of the cavity a cepts the bulk resin or preforms prior to closing the punch. The cross s al area of the loading zone cavity is twice the projected area of t cavity beneath it. This was done in order to acco the large amount of bulk resin w h must be heated and compressed into the cavity below. The resulting mo. par , then, has a thin "flash" which conforms to the loading zone ar

5.1.6 Mold Bases and Closure Guide Pins

The cavity is mounted on a lower mold base which raises the cavity approximately 5 inches off the lower press platen. This clearance allows space for a stout ejection bar to be passed under the cavity. The punch is mounted to an upper mold base. Both bases have grooves to which tiedown clamps are fastened for mounting to the platens. The closure guide pins can be seen in Figure 16. These pins assure alignment of the punch/loading zone/cavity combination as the punch closes.

5.1.7 Cal-Rod Heaters and Thermocouples

In order to heat the cavity, punch, and loading zone, both upper and lower platens were heated (up to 600°F) by built-in platen heaters with adjustable controls at the press control console. In addition, 24 cal-rod heaters were installed throughout the mold which were controlled by a set of five 110 volt Variacs. Four iron-constantan thermocouples sensed temperature at the following locations:

1 On the case hardened foot plate of the punch.

2 At the front of the split line between cavity and loading zone.

3 At the rear of the split line between cavity and loading zone.

4 At the bottom of the cavity.

5.2 Process Description

A total of 7 basic processes were tried. In addition, several post curing cycles and molding pressures were tried which are called out separately in Table 14. Typical process sheets which were used by the molding personnel are included in the Appendix.

5.2.1 Raw Resin, Slow Process

The first successful parts were made by this process. It is similar to the process by which the cylinders of Section 2 were made during Phase I except for pre-heat.

The resin was loaded into an 8 x 8 x 1 1/2 inch Teflon box with a trap door bottom. The box, with resin, was inserted into a LaRose dielectric heater for 8 minutes at which time the resin temperature reached 250°F. The box was then slid onto a carrying board and was transferred to a loading funnel which had been placed into the opened allowing the preheated resin to fall into a 250°F mold. The press was then closed and the temperature was taken as rapidly as possible to 450°F (approximately one hour elapsed time). The part was then ejected and subjected to post cure.

5.2.2 Raw Resin, Fast Process

This process used the same preheating cycle as in 5.2.1, however, the resin was loaded into a mold which was at a temperature of 450°F. The part remained in the mold at 450°F for approximately 15 minutes for cure, then ejected. Post cure followed. The strongest part made was by this process.

5.2.3 Pelletized Preform, Moderate Temperature Process

Pelletized preforms, 1 1/2 inches in diameter by 1 3/4 inches high were made by subjecting 30 grams of resin to 4500 psi pre-cure at 230°F for 75 seconds. These pellets were heated in a 250°F oven for 15 minutes and then placed in the dielectric preheater for 4 minutes and 20 seconds. The pellets were then compressed in a 330°F mold and were held for 5 minutes at this temperature. The mold temperature was then increased to 400°F as quickly as possible and held there for 10 minutes. At this point one of the following cures was instituted:

1 Punch was withdrawn approximately one inch and heat was cut. When temperature reached 330°F, part was ejected, inserts were removed, and part placed in oven for post cure and cooling.

2 Same as 1 above except part was cooled in mold, then ejected and post cured.

The most void-free cases were made by this process.

5.2.4 Raw Resin, Moderate Temperature Process

This process was the same as 5.2.2, paragraph 2, except that the mold was at 400°F.

5.2.5 Short Fiber Resin, Pelletized

This process was tried one time only. Pellets were preformed using 1/8 inch glass fiber filled polyimide resin made by Fiberite Corporation. These pellets were processed as in 5.2.3, paragraph 2, above. Pressure tests were unsatisfactory so that no additional parts were made by this process.

5.2.6 Piston Molding Process

The loading zone block and cavity block described in 5.1.1 and 5.1.5 also contain the necessary cavities for molding two pistons, with punch rods attached to the upper mold base. Two 30 gram pelletized preforms of Kinel 5511 were cut into quarters and three quarter segments were loaded into each of the two cavities after preheat. The preheat consisted of 70 seconds in a dielectric oven such that pellet temperature reached 250°F.

The pistons could not be successfully molded in this mold due to the following reasons:

1 The necessary molding pressure of 4500 psi was not attainable since the least pressure attainable on the one inch diameter punch pins was approximately 6700 psi. This occurred because the mechanical feature on the press which is intended to counterbalance the weight of the press ram (5 tons) was not operative.

2 In addition to 1, the rate of application of the weight of the ram (ram speed) was indeterminate since the ram descent rate is a function of load resistance and hydraulic fluid leakage rate within the press.

As a result of the above, the pressure versus time relationships were unknown and uncontrollable. The attempted molding operation resulted in fragmented pistons. No further attempts at molding the pistons were made with this mold.

5.2.7 Forward Case Molding Process

The cylinder and pin lock forming pins referred to in 5.1.2 were removed from the punch plate and several parts were molded which did not, therefore, contain cylinders or lock cavities. Three of these parts were made and cut and polished to form forward case halves thus eliminating the need to fabricate another expensive (and unnecessary) mold.

The process used to form these parts was identical to 5.2.4 except that an additional 60 grams of resin were used.

Several additional forward case halves were molded which were cut and machined to form pistons. In all instances, Kinel 5504 resin was used.

5.3 Inspection and Test Results

5.3.1 Inspection

It should be noted that the pins in the mold cavity which formed the four case bolt holes had to be shortened so that these holes were not completely molded into the case halves. This was caused by experimentation with various molding processes in which insufficiently preheated resin was loaded into the mold and the press was quickly closed without allowing proper time for resin liquifaction. In so doing large lateral forces were built up against the sides of the pins, causing them to bend. They were therefore shortened.

Table 13. presents the results of physical inspection of three aft cases and three forward cases as molded. Figure 20. explains the dimensions referred to in the table. It can be seen that internal case dimensions of the molded parts such as the cylinder bores, bearing groove diameters and bearing spacing dimensions are very repeatable between the three units tested although all dimensions showed shrinkage in the order of 0.001 inch/inch. The dates that the parts were molded are shown. No dimensional change as a function of time is apparent. External dimensions are somewhat less repeatable probably because plastic shrinkage can occur more readily away from the mold steel in the case of external dimensions.

The flatness of the molded servo valve mounting base was measured on three aft cases. Total indicator readings were all within $\pm$ 0.001 inch. The aft-most portion of the mounting base was comparatively rough in the as-molded condition. This was caused by an intentional slightly rough surface in the mold at this point. Mold surface roughness is intended to help hold the molded part in the mold when the punch is withdrawn. In this particular area, however, a highly polished surface would have been more appropriate and would have saved a short polishing touch-up operation on the molded part.

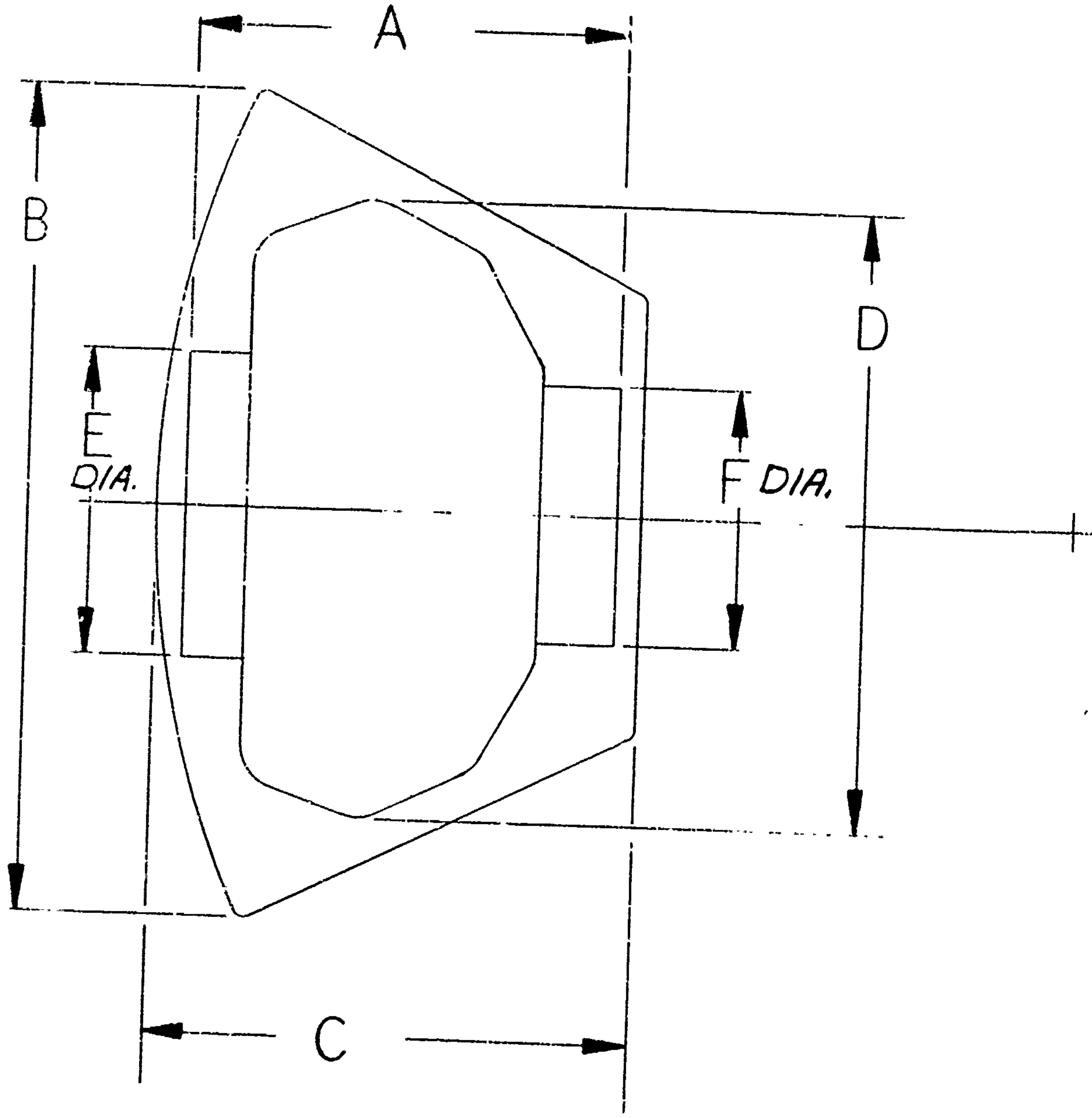


Figure 20. Inspection Dimensions

TABLE 13.

AFT CASE INSPECTION DATA

Measured Dimension	Molded Part #1 3/18-1	Molded Part #2 5/21-1	Molded Part #3 7/9	Dimension of Steel in Mold
Cylinder A	0.5005	0.5015	0.5017	0.5025
Cylinder B	0.5015	0.5025	0.5015	0.5025
Front Bearing (E) Diameter	2.064	2.064	2.065	2.066
Rear Bearing Diameter (F)	1.748	1.749	1.750	1.751
Distance (A) between Bearings	2.727	2.726	2.726	2.728
Max Outside Case Dimensions (B)	4.989	4.984	4.983	In Mold. Not Accessible
Case Outside Breadth (C,	3.053	3.049	3.049	In Mold. Not Accessible
Max Case Inside Width (D)	3.701	3.698	3.702	3.705
Forward Case				
Max Outside Case Dimension (B)	4.984	4.983	4.984	
Case Outside Breadth (C)	3.048	3.048	3.050	

```
                              Pressure
                             Capability
                               (psig)
                                    |
                                    | X
                               5000 |
                               4800 |
                                    | X
                               4600 |
                                    | ⊠
                               4400 |
                                    | ⊠
                               4200 |
                                    | ⊠
                               4000 |
                                    | ⊠X
                               3800 |
                                    | X
                               3600 |
                                    | X
                                  - |
                                    | XXXX
 Design                           - |
 Proof                              | ⊠XX
 Pressure                  →   3000 |
                                    | ⊠⊠XXX
                               2800 |
                                    | ⊠XXX
                               2600 |
                                    | ⊠XXXXXXX
                               2400 |
                                    | ⊠XX
                               2200 |
 Design                             | ⊠
 Operating                 →   2000 |
 Pressure                           | X
                               1800 |
                                    | X
                               1600 |
                                    | X
                               1400 |
```

X Denotes all parts made ⊠ Denotes parts made with preformed pellets

Figure 21. Distribution Curve for 65% Glass Filled Polyimide Hydraulic Actuators for WPAFB

From Table 13. it was concluded that the critical internal dimensions of the part can be repetitively molded in production so that no machining is required to form acceptable cylinder bores, bearing grooves and screw threads. The less critical external dimensions of the actuator are also acceptable but not as repeatable from molded part to molded part. This subject is further discussed in 5.5.

5.3.2 Process and Pressure Tests

Thirty-seven aft cases were molded. Not all were acceptable (principally in the early stages of molding) due to mold and/or personnel problems. Of the 37, 23 were subjected to pressure tests of the cylinder bores. This was accomplished by screwing a steel insert into the cylinder bore and pressurizing the bore with a hand pump while the unit was inside an explosion-proof box. No porting or mounting holds had, as yet, been drilled in the cases. The pressure at which leakage occurred was recorded. Most failures occurred as small pinhole leaks through the cylinder walls although, in a few cases, cracking of the cylinder wall or leakage from the pressurized cylinder into the adjacent fin lock cylinder bore occurred. In no case did any cylinder shatter or break into pieces. In many cases the failed parts were disected and viewed under magnifying binoculars to determine porosity.

Table 14 presents the results of this initial testing process with appropriate notes regarding the molding process used (see 5.2 above) and relevant variations to these processes.

Figure 21 shows the distribution curve of pressure capability of the molded cylinder case halves. Since the cylinders were pressure tested separately, each point represents one cylinder. It will be noted from Figure 21 that 90% of the parts made exceeded 2200 psig capability and that the use of preformed pellets improved the overall distribution of the pressure capability by about 400 psig. It can also be seen that several of the parts exceeded the final proof pressure requirement of 3000 psig. The distribution curve indicates that some further process development would be required before going into production on this particular part in order to increase the yield. Experience indicates that process development should include the use of pre-forms and a closely controlled temperature aging after molding.

5.4 Machining

After the initial three aft case halves which withstood 2700 psi in the bores without leaking had been selected (parts 5/22-1, 5/22-2, 5/23-1), it was necessary to drill the servo valve mounting and porting holes in the plastic. It was also necessary to tap the pressure inlet fitting hole. A steel drill jig block was fabricated which was indexed to the smaller shaft bearing groove. In this way all nine holes associated with mounting and porting the servo valve and feeding pressure to the fin lock could be drilled in a matter of minutes. It was necessary, however, to maintain the drills in sharp condition with a slight negative rake to avoid splintering

TABLE 14.

PROCESSES AND PRESSURE TEST RESULTS

Mold Date	Pressure Test Date	Pressure Capability (psi) Bore A	Bore B	Molding Process (Section Number)	Comments (Exceptions, Post Cures, Press Force)
3/18-1	3/20	2500	2700	6.2.1	Ejection pins up when loaded. Post cured 20 hrs @ 480°F. 66 tons.
3/18-2	4/17	4640	>5000	6.2.2	Ejection pins up when loaded. Post cured 20 hrs @ 212°F. 68 tons.
3/18-3	3/20	3300	2500	6.2.2	Ejection pins up when loaded. Post cured 20 hrs @ 480°F. 68 tons.
3/18-4	3/20	3150	2450	6.2.2	Ejection pins up when loaded. Post cured 20 hrs @ 480°F. 69 tons.
4/10-1	4/17	3000	2680	6.2.2	Ejection pins down when loaded. Post cured 24 hrs @ 480°F. 75 tons.
4/11-1	4/17	2980	3320	6.2.2	Ejection pins down when loaded. Post cured 24 hrs @ 480°F. 75 tons.
4/11-2	4/17	2460	2600	6.2.2	Ejection pins down when loaded. Post cured 24 hrs @ 480°F. 75 tons.
5/6-1	5/9	2400	2420	6.2.2	Pins up. Resin in vacuum 2 hrs. Post cured 20 hrs @ 290°F. 68 tons.
5/6-3	5/9	3320	3840	6.2.2	Pins down. Resin in vacuum 2 hrs. Post cured 20 hrs @ 290°F. 68 tons.
5/7-1	5/9	3340	No test	6.2.3*	Part removed from mold at 400°F. Post cured 16 hrs @ 390°F. 45 tons.
5/9-1	5/12	2850	4440	6.2.3	Part removed from mold at 400°F. Post cured 20 hrs @ 290°F. 47 tons.
5/9-2	5/12	2380	2740	6.2.3	Part removed from mold at 400°F. Post cured 20 hrs @ 290°F. 48 tons.
5/13-1	5/14	2000	1660	6.2.3	Part removed from mold at 400°F. Post cured 16 hrs @ 475°F. 45 tons.
					Fibcrite short-fiber resin used.
5/15-1	5/19	1600	3580	6.2.4	Loaded at 400°F for 25 minutes. No post cure. 68 tons.
5/15-2	5/19	2600	2240	6.2.4	Loaded at 400°F for 30 minutes. No post cure. 65 tons.
5/15-3	5/19	2860	3160	6.2.4	Loaded at 400°F for 30 minutes. No post cure. 65 tons.
5/16-1	5/19	2740	3640	6.2.4	Loaded at 400°F for 25 minutes. No post cure. 65 tons.
5/16-2	5/19	4020	4320	6.2.3	Cooled in mold. No post cure. 48 tons.
5/21-1	5/27	>2700	2150	6.2.3	Cooled in mold. No post cure. 48 tons.
5/22-1	5/27	>2700	>2700	6.2.3	Removed from mold at 330°F. No post cure. 48 tons.
5/22-2	5/27	>2700	>2700	6.2.3	Cooled in mold. No post cure. 48 tons.
5/23-1	5/27	>2700	>2700	6.2.3	Removed from mold at 330°F. No post cure. 48 tons.
5/23-2	5/27	2500	>2700	6.2.3	Cooled in mold. No post cure. 48 tons.

* All subsequent loadings made with pins down

and chipping. (This method was suggested in literature supplied by the resin manufacturer.) The tap used for the pressure inlet hole was also required to be sharp with a negative rake. Several holes were drilled and tapped in rejected case halves for practice before the selected case halves were drilled and tapped. The final drilling and tapping operations were uneventful.

The first pistons tried were machined from sections cut from molded forward cases. The lathe tools were kept sharp and negatively raked. No problems were encountered. They were finished using fine emery paper. Steel pistons were eventually used (see Section 6.2).

The piston guide sleeves were also machined from blocks cut from forward case halves. These sleeves required a 7/8 inch fine thread over their entire exterior surface with a close tolerance 1/2 inch hole through their centers. No problems occurred.

5.5 Assembly (See Figures 22, 23 and 24)

The following items worthy of note occurred during assembly of the actuator:

1 The actuator piston seals which are captivated in their groove by the piston guide sleeves were not being compressed sufficiently by the sleeves. This was due to the fact that the groove was dimensioned as a reciprocating seal only. Insufficient squeeze of the back-up ring/O-ring combination resulted in leakage past the seal and out the sleeve threads. A 0.035 inch step was machined on the bottom of the sleeve which applied squeeze to the seals. This extra squeeze aleviated the problem.

2 Some movement of the punch apparently occurs during molding which causes a mismatch of the external case surfaces.

However, the external misalignment of the two case halves does not deter from the very good fits and alignment of all internally molded features such as bearing grooves, threads, and cylinder bores.

3 The initial design called for Locktite bearing mount compound to be used for seating the bearings. This was found to be unnecessary since the mold shrinkage referred to in 5.3.1 above caused a small gap (approximately 0.008 inch) to exist between case halves so that the bearings were clamped securely as the case bolts were tightened.

Assembly of the feedback potentiometer presented no problems. Dexter Corporation 934 epoxy cement was used to bond the potentiometer to the teflon potentiometer retainer. Teflon was used instead of Kinel 5504 to save mold costs. No problems are predicted molding this part of 5504 in view of experience gained in molding aft and forward case halves. The teflon was treated with "Tetra-Etch" prior to bonding to the potentiometer.

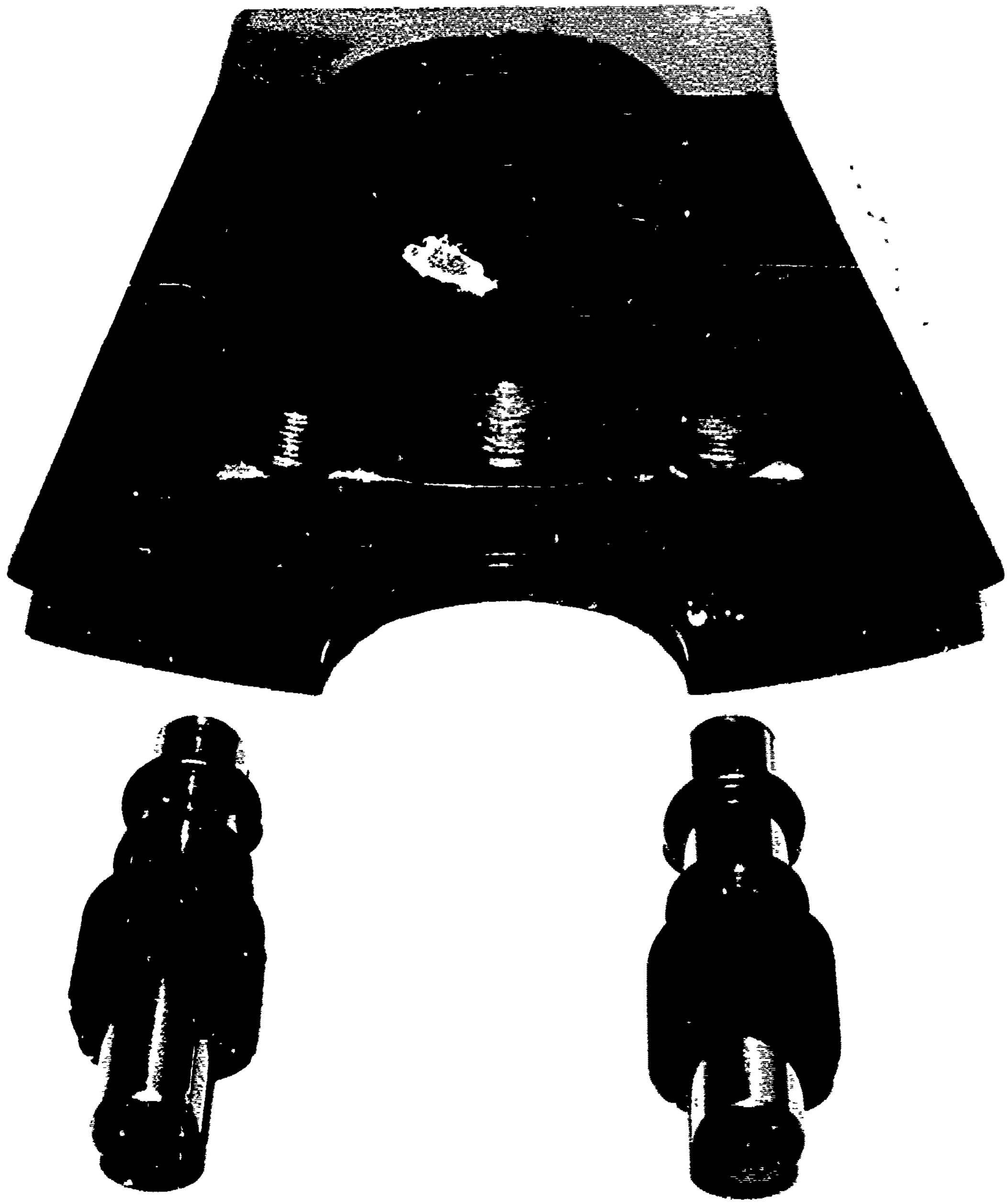


Figure 22. Inside View

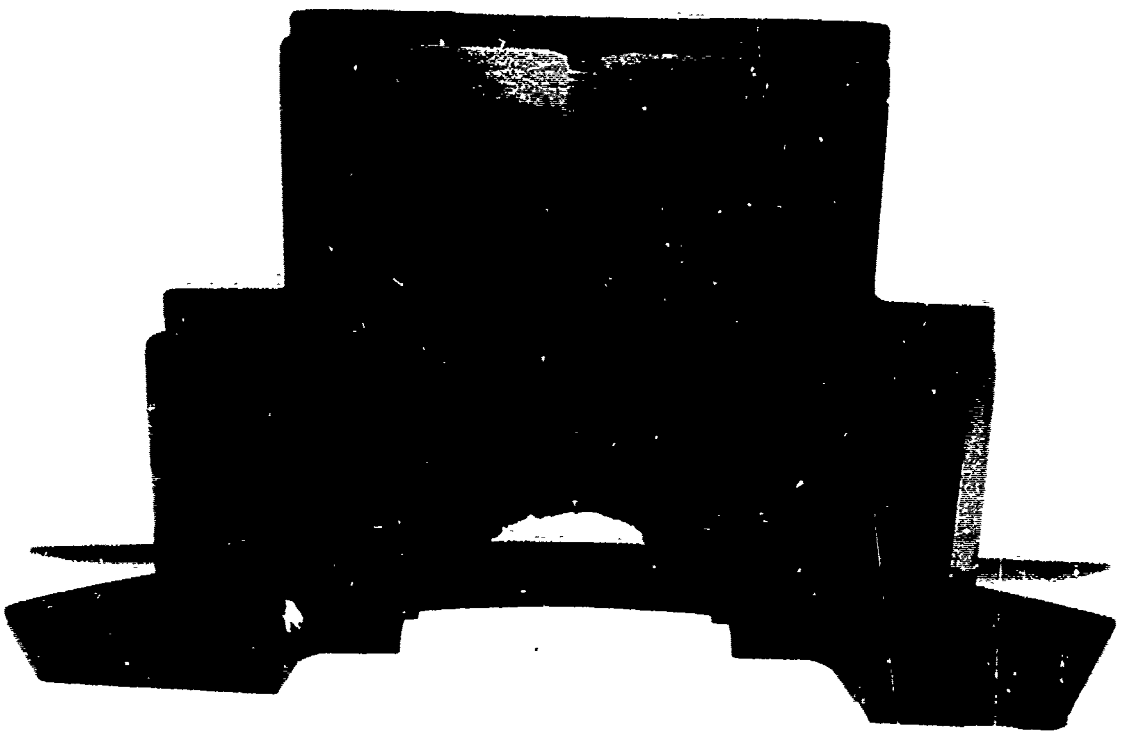


Figure 23. Cutaway View

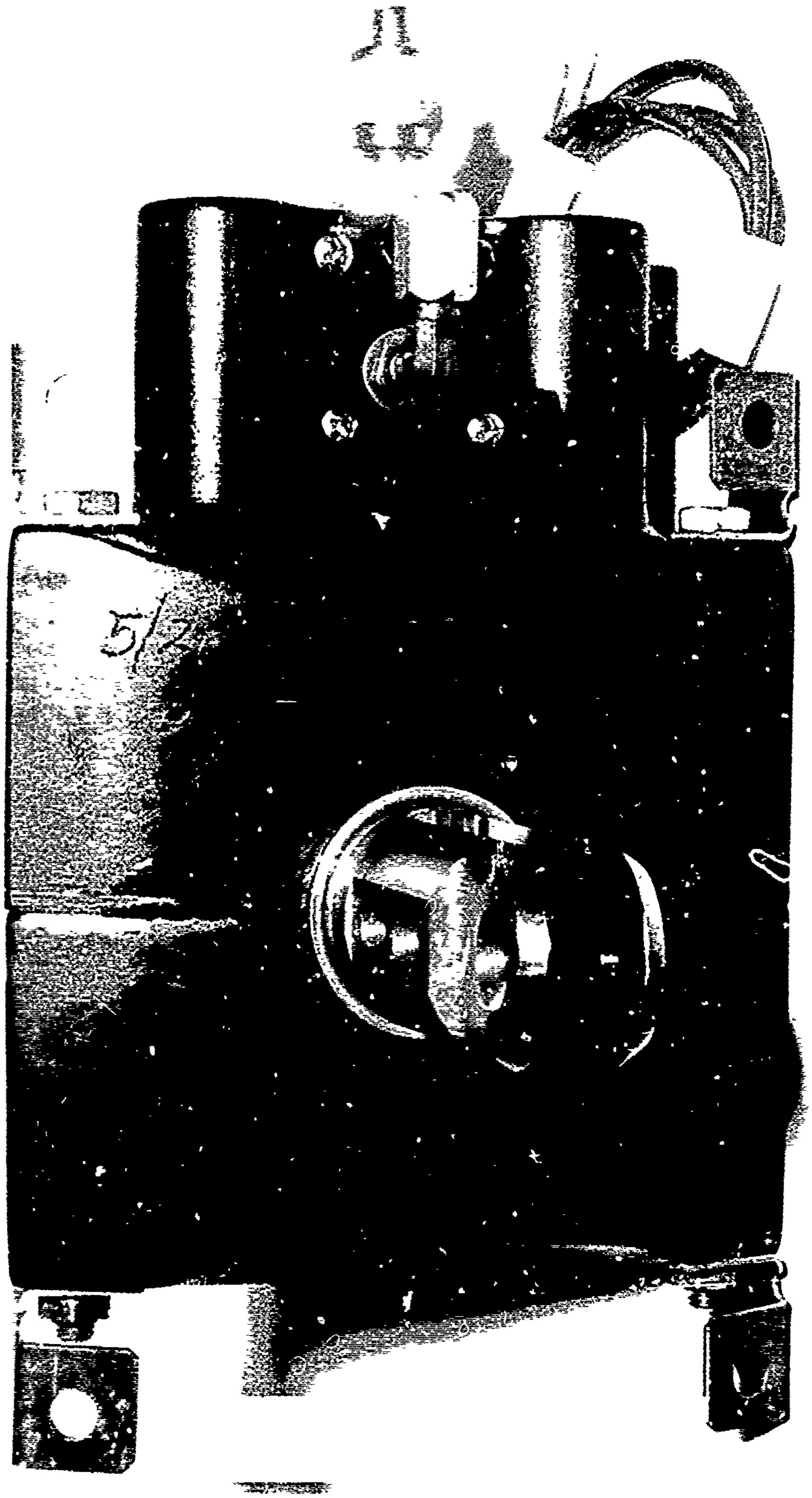


Figure 24. Complete Actuator

SECTION 6

PERFORMANCE TESTING

6.1 Test Plan

The test plan agreed upon follows:

PLASTIC ACTUATOR TEST PLAN

All bench tests will be performed with system pressure necessary to obtain 200 in-lb of torque and again at 400 in-lb torque. Units are to be installed in the test fixture, amplifier connected and electrical null set prior to testing.

Unit #1

Room Temperature, MIL-H-5606 fluid:

1 Load demonstration test: Demonstrate required system pressure to obtain 400 in-lb of torque at $\pm$ 30° shaft angle as measured by torque wrench.

2 Threshold and hysteresis test: Full dynamic load. Trace of hysteresis loop 0°, +30°, -30°, 0°. Record input signal, follow-up signal and valve signal.

3 Step test: Full velocity. Input steps of $\pm$ 15°. Record input and follow-up signals. Full dynamic load.

4 Frequency response: Full dynamic load. $\pm$ 1.5° input, 0 to 100 Hz. Record input and follow-up signals.

Temperature Shock Test:

Subject unit to two cycles of the following temperatures:

+ 185°F for 4 hours
- 65°F for 4 hours
+ 270°F for 7 minutes

Then run tests 1, 2, 3 and 4 above.

Temperature Tests with M2-V Fluid:

Run tests 1, 2, 3 and 4 above with +270°F fluid and air at room temperature.

Run tests 1, 2, 3 and 4 above with -45°F fluid and air at room temperature.

Unit #2

Room Temperature, MIL-H-5606 fluid:

Run tests 1, 2, 3 and 4 above. Then run tests 5, 6, 7 and 8 below.

5 Life test - 1/2 full static load. 24 cycles of the following at full system pressure:

2 1/2 minutes, $\pm$ 1°, 3 cps
2 minutes, $\pm$ 3°, 1 cps
1/2 minute, $\pm$ 15°, 1/2 cps
2 cycles at $\pm$ 30°

After completion of 24 cycles, bottom each piston 25 times.

6 Demonstrate capability to lift 100 lbs through angle of $\pm$ 30° on 16 inch bar.

7 At completion of above steps, install fin lock and load 16 inch bar with 100 lbs applied at an offset angle necessary to apply 500 in-lb of torque. Raise system pressure until lock pin disengages. Record system pressure.

8 Stop-to-stop cycling test: No load. 10 cycles, +30°, -30°, +30° steps.

Unit #3

Room Temperature, MIL-H-5606 fluid:

Run tests 1, 2, 3 and 4 above at 1/2 system pressure.

6.2 Testing Events

The testing events listed below are written in chronological order. That is, Unit #3 was the first unit subjected to test, then #2, then #1.

6.2.1 Unit #3 (Serial Number 5/22-2)

This unit was bolted into the test fixture and was connected to the servo amplifier after having its electrical null approximately aligned with its mechanical null using a volt-ohm meter. After power was applied, the null was set precisely using a bubble clinometer; the potentiometer was rotated until follow-up voltage read zero at zero degrees of shaft position. The potentiometer holder was then locked in position.

All initial testing was run with no fin lock parts installed (all fin lock parts were, however, made and available). All holes were drilled and/or tapped prior to mounting the actuator in the test fixture except for the small hole which feeds pressure to the fin lock mechanism.

Unit #3 had plastic pistons installed during initial tests.

Tests 1 through 3 of Section 6.1 were run at a system pressure of 1300 psi which caused the actuator to deliver 200 in-lb of torque as measured by a torque wrench clamped to the output shaft. (It was agreed that test #1 would not be performed using more than 1/2 load since it was decided that application of the full 1600 in-lb bending moment to the shaft was a one-time test only. The torque wrench was therefore used to determine torque capability at a given system pressure.)

The instrumentation used to record input, follow-up, and valve current signals was a C.E.C. 7-inch recording oscillograph with #319 galvanometers. A Hewlit Packard model 203A wave generator was used to generate square and sinusoidal waves.

Tests 1 through 3, run at 1/2 system pressure, were satisfactory for this input pressure and no leakage occurred. During preparations for the frequency response test, however, leakage occurred past one of the piston seals. The unit was disassembled and it was found that the piston O-rings had failed by rolling and/or spiraling. Friction between the machined plastic pistons (which contain fiberglass fibers) and the O-ring was too great to allow proper sliding. (This does not rule out plastic pistons however, since gel coated pistons would normally result from a molding process.) Steel pistons were made (steel slugs) and no further problems were encountered with O-ring spiraling. The frequency response was run with no leakage. With steel pistons, 200 in-lb of torque were delivered at 950 psi system pressure. Output shaft rotation was 31.0° CCW and 30.5° CW.

The fin lock was subsequently installed and the unit was re-tested per the test plan at 1/2 system pressure.

6.2.2 Unit #2 (Serial Number 5/22-1)

This unit had steel pistons installed prior to initiation of testing. The set-up procedure and instrumentation used for this unit were the same as described in 6.2.1 above.

Tests 1 through 4 were run as in 6.2.1 above at a system pressure of 950 psi which caused 200 in-lb of torque to be delivered as measured by a torque wrench. Output shaft total rotation was 31.0° CCW and 30.5° CW. Test 5 was then started at 1000 psi system pressure. The static load fixture was attached to the shaft and 50 lbs was hung from it at a distance of 16 inches from the outer actuator bearing. Two complete load cycles were run as called out in Section 6.1. The unit was then run 55 minutes continuously at $\pm$ 1° at 3 cps with the 50 lb load on the load bar. No leakage occurred.

At this point the system pressure was increased so that 400 in-lb of torque were delivered (measured with a torque wrench). This occurred at a system pressure of 1850 to 1900 psi. Tests 2 and 3 were then run at full system pressure. Leakage was noted past a piston O-ring during the step test. The unit was again disassembled. A slight amount of wear was detected in the plastic piston guide bores (approximately 0.001 inch diametral increase). New guides were installed with 0.036 inch steps at their base which increased O-ring squeeze. Upon reassembly, no leakage occurred. Torque output remained unchanged at 400 in-lb at 1850 psi.

At this point the life test was initiated at full system pressure and with 1/2 full load on the load bar. Twenty-four load cycles were performed in an essentially non-stop manner. No leakage occurred. The pistons were then bottomed so that full load pressure would be delivered to the piston bores. Twenty-five of the full pressure cycles were applied. Leakage occurred after the fifth cycle from one of the servo valve mounting screws. This leakage rate was approximately one drop every 3 seconds but only during the time that full load pressure was being applied to the "A" cylinder. No leakage occurred when the actuator was centered.

It was noted during the step test at full pressure that only 225° per second of shaft velocity was being attained. This was caused by line drop through two 4-inch lengths of 1/8 inch diameter tubing and through the 1/8 inch inlet and return fittings on the actuator. The input was increased to 3/16 inch tubing and a 3/16 x 1/8 elbow while the return fitting was changed to a 1/4 inch by 1/8 inch reducing fitting. A re-run of the step test gave a maximum velocity of 375° per second (spec requirement is 350° per second).

Unit #2 was subsequently subjected to tests 6, 7 and 8 of Section 6.1 after the fin lock was installed. A small amount of fluid leaked from the piston seals under full load at maximum shaft angle. Leakage from the cylinder walls and/or servo valve mounting bolt holes was slight. During test 8, the actuator was commanded into the piston stop at full velocity ten times. No increase in leakage was noted.

6.2.3 Unit #1 (Serial Number 5/23-1)

This unit was tested per procedure 6.1 at 1/2 system pressure. It was then tested with 270 F fluid. This fluid was blown down from a 400 cu.in. accumulator which was installed in an oven. A calibrated iron-constantan thermocouple with direct readout meter was installed in the fluid line close to the actuator (see Figure 25). The unit was first cycled intermittently for several minutes at 300 psi to assure that hot fluid filled the lines and actuator. No leakage occurred after completion of tests 1 through 4 at full pressure. The unit was then cycled into the piston stops at full system pressure. No leakage occurred.

The accumulator was then installed in a cold chamber so that fluid temperature could be taken to -45°F. A thermocouple bridge was used to determine temperature. The unit was cycled intermittently at 300 psi for several minutes for cool down. During test #1 of Section 6.1 the torque wrench read 400 in-lb at a system pressure of 1925 psi. At this point a slight leak was noted at the end of bore B. After one minute of cycling at 1950 psi, tests 2, 3 and 4 were run. The unit was then cycled into the piston stops at full system pressure several times during which the leakage

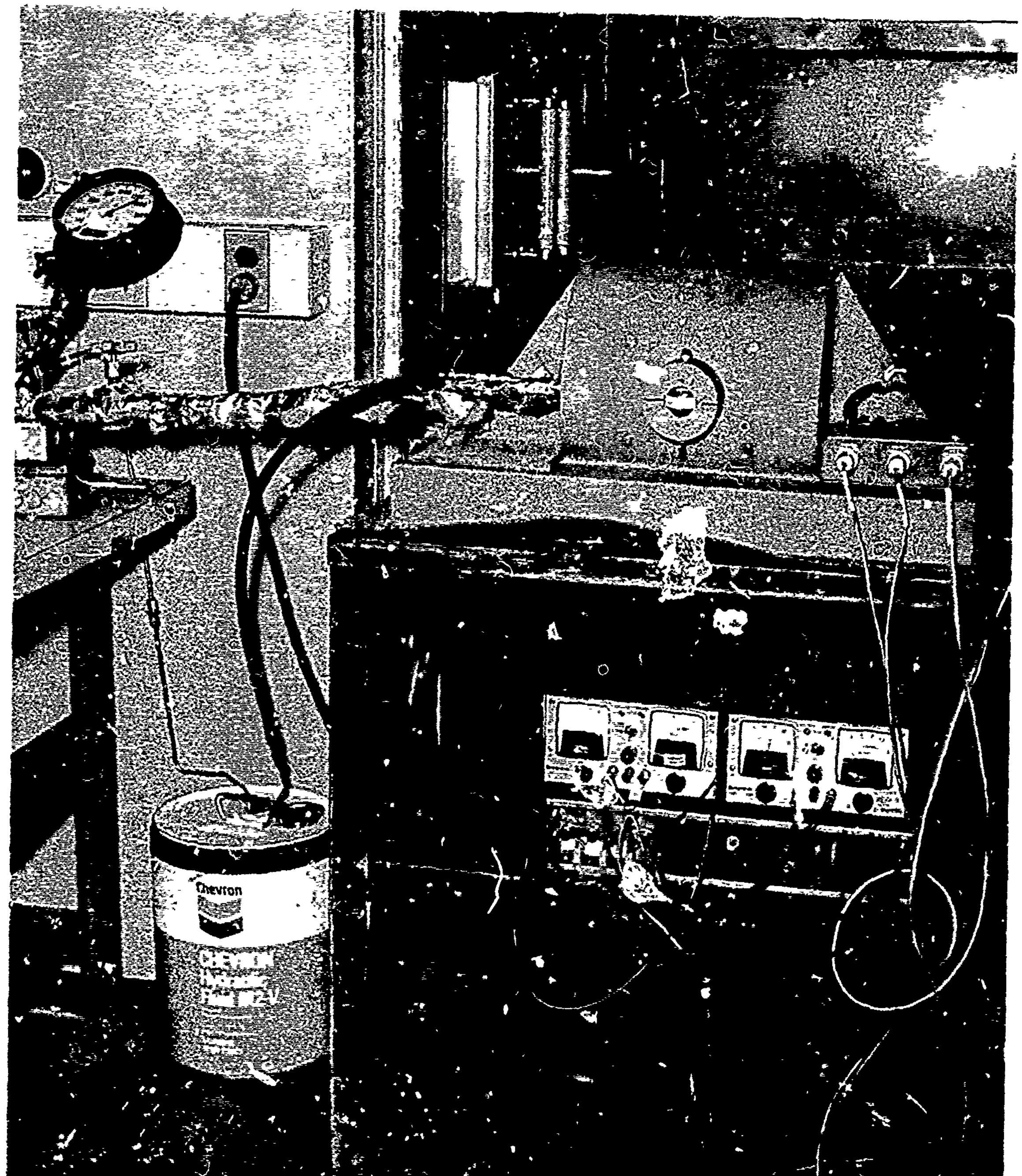


Figure 25. Hot Test Setup

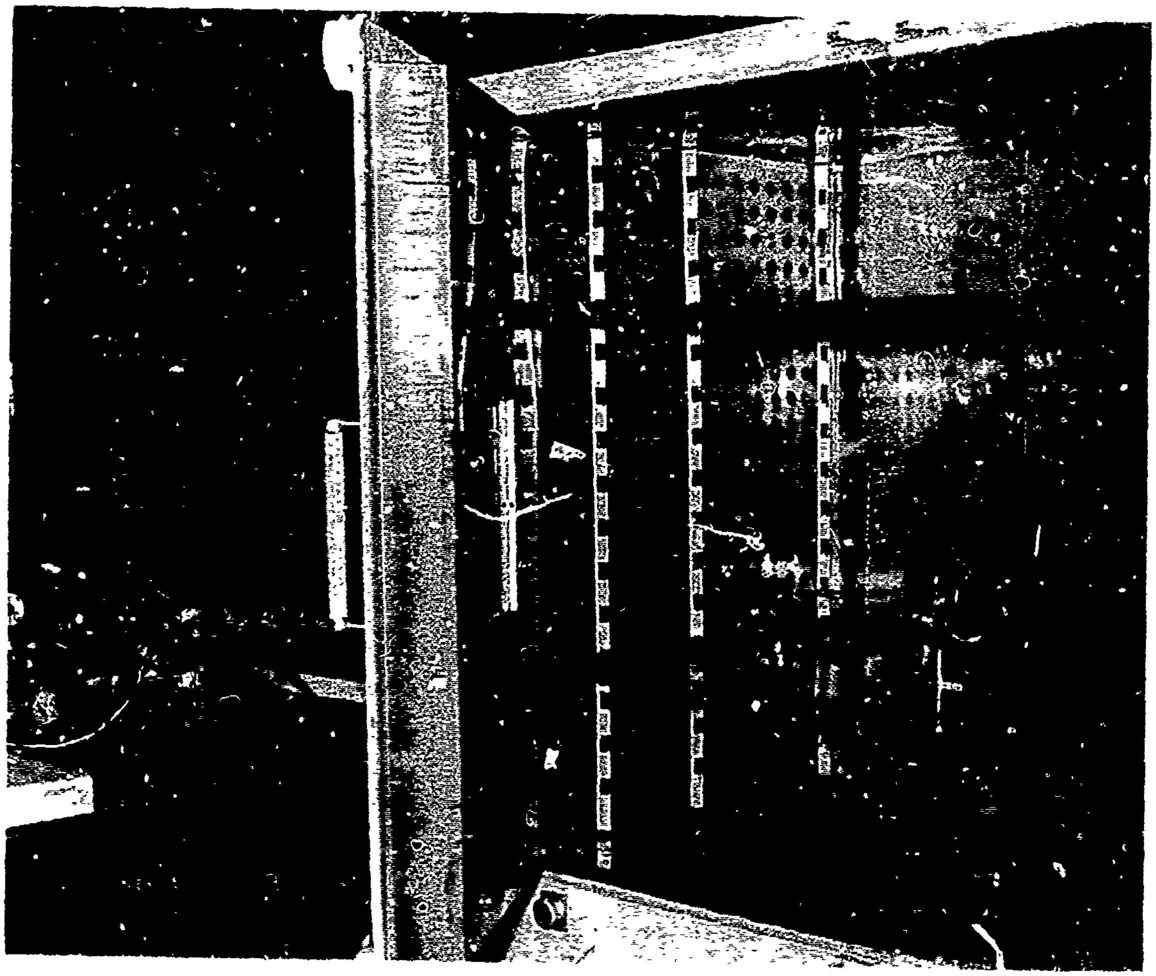


Figure 26. Hot Test Setup - Accumulator

rate was one drop per two seconds. The leakage rate at actuator null was one drop per five seconds. The unit was then subjected to the temperature shock tests of Section 6.1. Post temperature shock tests revealed little, if any, change in leakage or actuator characteristics.

The fin lock was then installed and the unit was subjected to full system pressure tests. Leakage out the end of bore B increased significantly. Eventually a fine spray eminated from the end of B bore.

6.3 <u>Test Results</u>

Table 15. is a summary of the actuator characteristics measured or demonstrated by the tests. Figures 27 and 28 are plots of the frequency responses of units #1 and #2. (Unit #3 was not tested at full pressure, per agreement.)

6.3.1 Deadzone

The deadzone figures listed in Table 15. were derived from frequency response runs in the following manner: Deadzone was assumed to be the quantity of input signal necessary to cause first motion of the output shaft. Therefore, the 1 Hz, $\pm$ 1.5° portion of the frequency response was inspected and the amount of servo valve current necessary to cause first motion was noted. Since the loop gain was 13.6 ma/deg (see Section 3.2), the deadzone was determined as the amount of equivalent degrees of follow-up input necessary to cause the exhibited current.

6.3.2 Hysteresis

Hysteresis was assumed to be the sum of the absolute values of input signal which occurred as the actuator was cycled between $\pm$ 30° measured at the points where the follow-up signal crossed zero.

6.3.3 No Load Velocity

No load velocity was measured with a 0.02 in-lb-sec^2 inertial load bolted to the shaft. This velocity was within the statement of work specified value of 350° per second on both units at all temperatures. At room temperature, unit #1 no load velocity averaged 440°/sec. Unit #2 averaged 371°/sec.

6.3.4 Frequency Response (See Figures 27 and 28)

Frequency responses were run with a 0.02 in-lb-sec^2 inertial load bolted to the shaft with an input amplitude of 3° peak-to-peak. The amplitude ratio curves obtained were within the statement of work requirements for both Unit #1 and Unit #2 for all temperatures. The phase lag encountered with Unit #2 (life test) was also within specification. However, Unit #1 was slightly out of the phase lag specifications at 270°F above 60 Hz and grossly out of specification at -45°F. This marginal phase

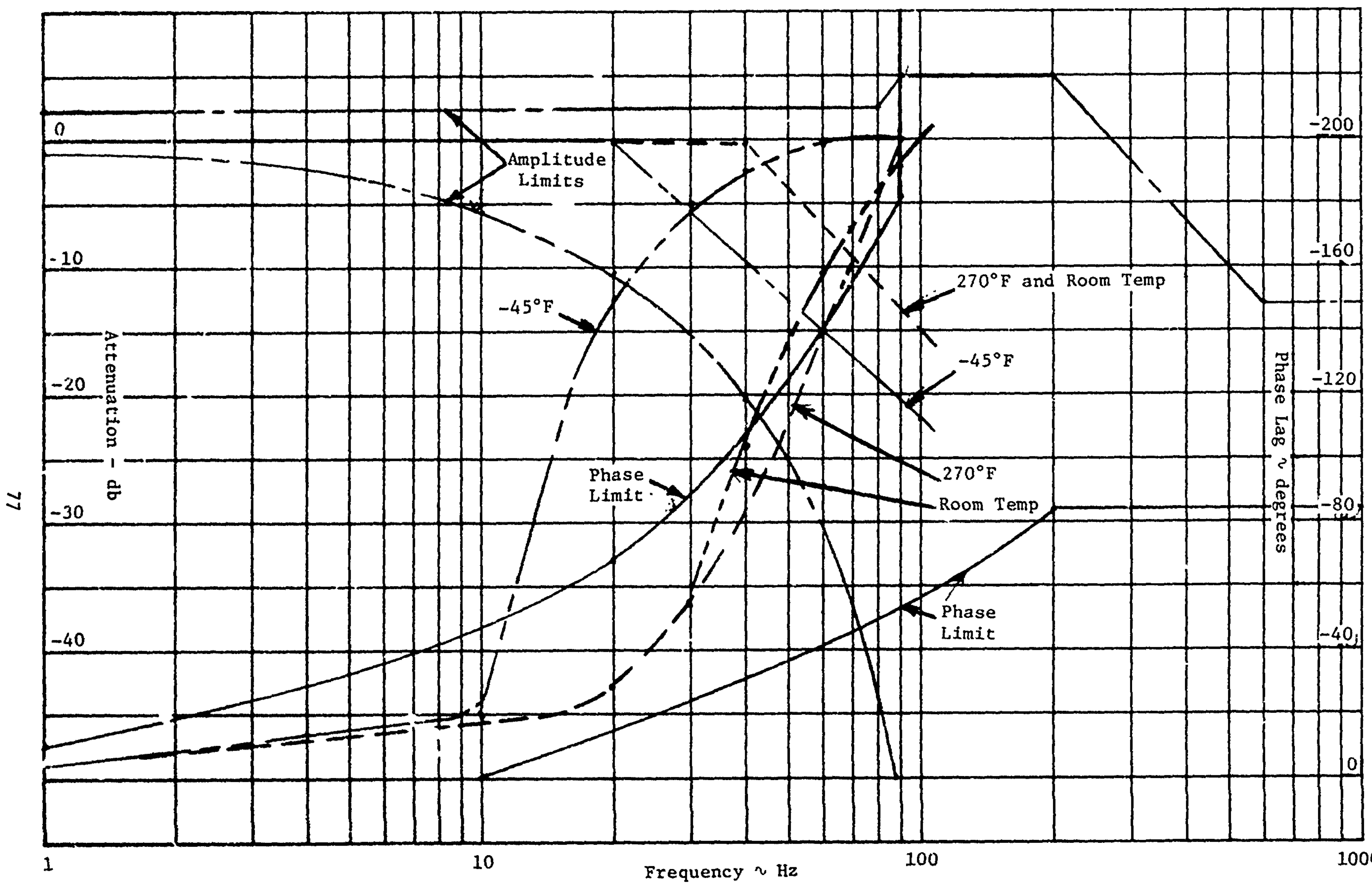


Figure 27. Frequency Response Unit #1

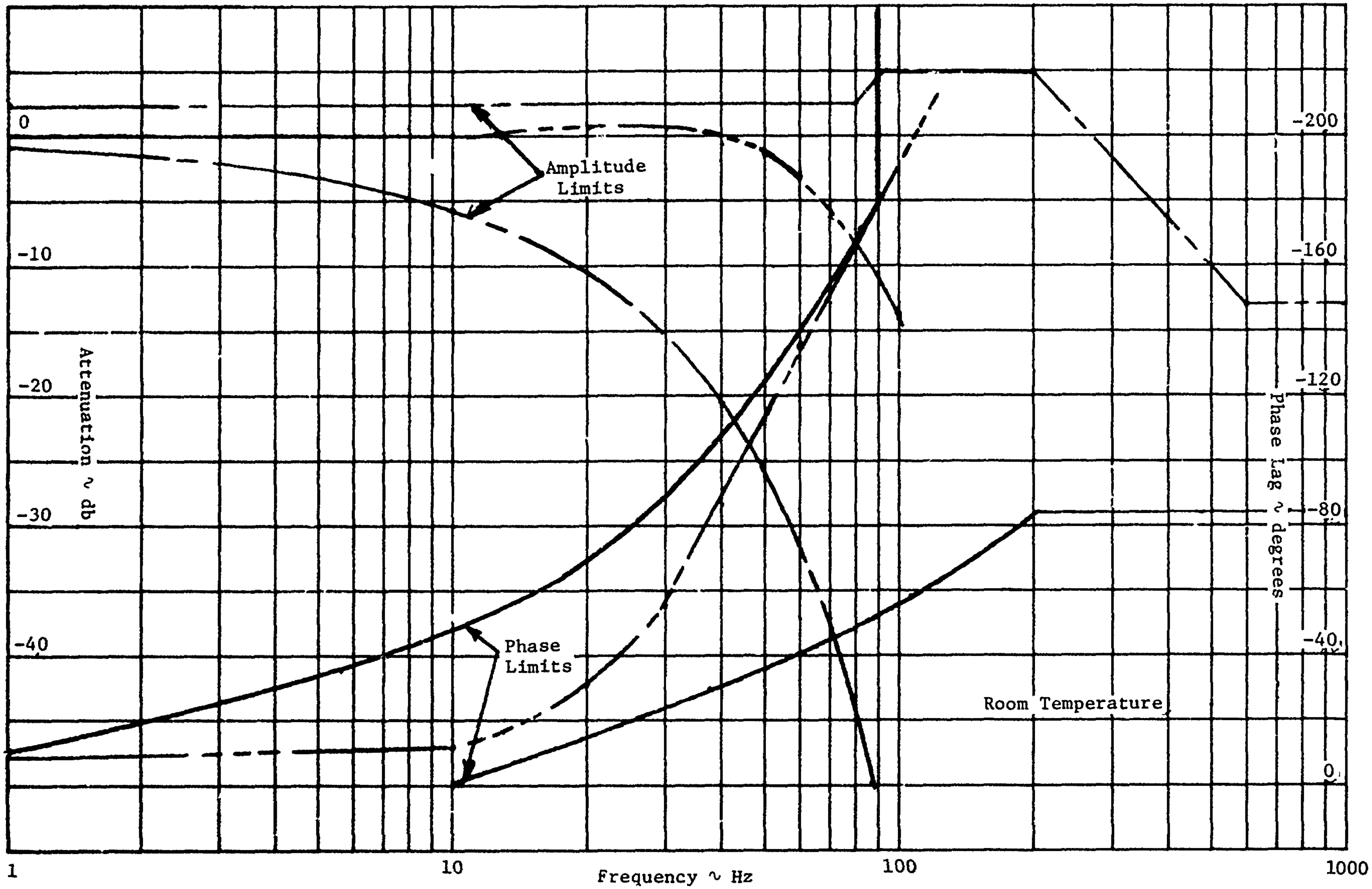


Figure 28. Frequency Response Unit #2

TABLE 15.

TEST RESULTS

Item	Unit #1, S/N 523-1	Unit #2, S/N 522-1	Unit #3, S/N 522-2	Statement of Work Requirement
1. Stall torque	400 in-lb @ 1925 psi	400 in-lb @ 1850 psi	200 in-lb @ 1000 psi (full torque not required of this unit)	4.1.1 400 to 415 in-lb
2. No load slew rate	> 350 deg/sec at all temperatures tested	> 350 deg/sec	Not tested at full pressure	4.1.2 350 deg/sec
3. Shaft rotation	31.0° CCW, 30.5° CW	31.0° CCW, 30.5° CW	31.0° CCW, 30.5° CW	4.1.3 > 30°, < 33°
4. System press.	1925 psig (torque wrench)	1850 psig (torque wrench), 2000 psig (load test)	950 psig (torque wrench at 1/2 load)	4.1.4 < 3200 psig
5. Freq. resp.	See Figure 27	See Figure 28	Not tested at full system pressure	4.1.5 See Figure 27 for requirements
6. Stiffness	Not tested	Not tested	Not tested	4.1.6 > 1500 in-lb/deg
7. Bending moment (shaft)	Not tested	> 1600 in-lb	Not tested	4.1.7 1600 in-lb
8a. Feedback pot travel	340°	340°	340°	4.1.8a. > 35°
b. Null width	Infinite resolution	Infinite resolution	Infinite resolution	4.1.8b. < 0.1°
c. Pot excitation	15 volts	15 volts	15 volts	4.1.8c. 15 volts or less
9a. Servo valve	Moog Model 30	Moog Model 30	Moog Model 30	4.1.9a. Off shelf item
b. Supply voltage	27 Vdc	27 Vdc	27 Vdc	4.1.9b. 27 Vdc

(Continued)

TABLE 15. (Cont'd)

Item	Unit #1, S/N 523-1	Unit #2, S/N 522-1	Unit #3, S/N 522-2	Statement of Work Requirement
10. Threshold	0.037° @ 80°F & 270°F 0.073° @ -45°F	0.037°	Not tested at full pressure	4.1.10 < 0.05°
11. Hysteresis	Nil @ 270°F, 2.8% @ 80°F, 3.2% @ -45°F	Nil at full load	Not tested at full pressure	4.1.11 < 3.5% rated signal
12. Lock pin disengage pressure	Not tested under load	900 psi @ 500 in-lb load	Not tested under load	4.1.12 < 1700 psi @ 500 in-lb torque
13. Envelope	In except for flash	In except for flash	In except for flash	4.1.13 See S.O.W.
14. Proof and burst pressures	Leaked at 2000 psi during cold test	Leaked at 1900 psi during life test	Not tested	4.1 1.5 x P_S proof 2.5 x P_S burst

lag performance is attributed to velocity limiting occasioned by intentionally designing the actuator cylinder port passages with small diameters. This problem could be alleviated in an actual flight prototype (after definite response requirements were determined) by adjusting one or all of three parameters - increasing servo valve flow rate, increasing system pressure or increasing cylinder port passage diameter.

One question that remained after testing was completed related to the fact that Unit #1 was out of the phase lag specification at room temperature while #2 was in specification even though Unit #1 exhibited a no-load velocity of 440°/sec while Unit #2 exhibited 371°/sec. A possible explanation for this discrepancy is that Unit #1 may not have been properly bled (of air) since this unit was subjected to a frequency response test almost immediately after installation in the test stand without a great deal of prior exercising.

SECTION 7

SUMMARY

The feasibility of reducing the costs of hydraulic actuators for tactical missiles was demonstrated by designing, fabricating and testing three hydraulic actuators capable of meeting the specification requirements of the actuators used on the Maverick missile.

The cost analysis showed that the unit cost of the complete servo-actuator in quantity of 2000 units would be reduced by 33%.

The plastic material selected for its high temperature creep and strength characteristics, compatibility with hydraulic oils, and resistance to moisture and aging was a 65% glass filled polyimide marketed under the trade name KINEL 5504. It is a compression moldable thermosetting plastic.

The actuator housing was designed so that it could be molded in two pieces with a split plane through the shaft centerline allowing the use of as-molded surfaces for the cylinders, O-ring seats, threaded attachments and bearing seats.

Molding processes and techniques were investigated by molding and testing-to-failure the critical part, the cylinder housing, which demonstrated the part could be made to withstand pressures well above the operating pressure.

Pressure and vent passages for the servovalve installation were drilled, demonstrating that the 65% glass filled polyimide was easily machinable.

Three complete servo-actuators were assembled and successfully tested demonstrating performance under specified environmental conditions.

SECTION 8

CONCLUSIONS AND RECOMMENDATIONS

8. Conclusions and Recommendations

Feasibility - From the effort performed under this contract and reported herein, it is concluded that the low cost plastic actuator is feasible; three units successfully passed performance, environmental and life tests.

Material - The best material currently available to meet the 270°F, 400 inch-pounds torque and other requirements is a 65% glass filled polyimide (trade name KINEL 5504) which is a compression moldable thermosetting material.

Cost Saving - A cost saving of 33% to 50% can be realized when fabricating a quantity of 2,000 units. The 33% cost saving stems primarily from a reduction in machining operations by using as-molded threads, cylinder bores, O-ring seats and bearing seats as demonstrated in the three test units. This cost saving can be increased to 50% by adoption of a low cost, short life servo valve available from Moog, Inc. instead of the standard valve used.

Molding Process - The molding process and procedures need further refinement to increase the yield of high strength parts. It is recommended that a further study of the molding process and procedures be made in which the best low cost molding process is developed by a thorough evaluation of the effects of: pre-heating, molding temperature and pressure, ejection temperature, cool down rates and post-curing.

Further Recommendations - Further cost savings can probably be effected by the use of injection molding for some of the detail parts which do not require high strength. It is recommended that a further study be performed, including selection of material, to determine the feasibility of a further cost reduction through the use of injection molding for some of the parts.

APPENDIX A

Phase I Design Calculations

A-1 Summary of System Pressure Computations

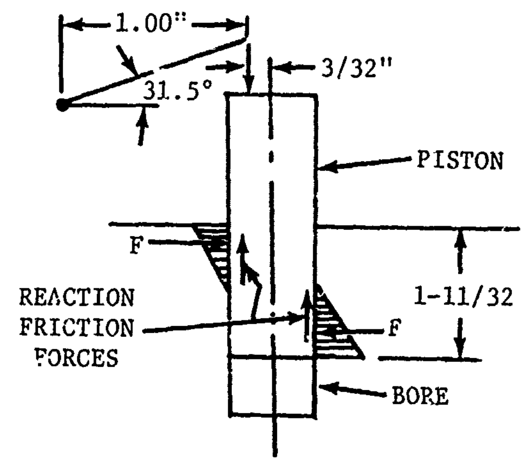


Figure A-1
Piston Loads

Assumptions:

1. 407 in.lbs. required torque at 31.5 degrees deflection
2. 150 psi reservoir pressure
3. O-ring friction (pressurized) is 15% of basic drive pressure
4. Dimensions scaled from print.

Basic System Pressure:

$A \times R \times (P_{S_1} - 150) = 407$

$R = 1.00''$ @ $31.5°$

$A = 0.196$ in.sq.

$P_{S_1} = 2205.5$ psi

Estimation of Wall Friction Force (Side Friction):

Imposed couple = $407 \times 3/32$

Reaction friction force = 25.5 lbs, or a pressure of 127.5 psi

Basic drive pressure = $2205.5 + 127.5 = 2343$ psi

System pressure = $P_S = 2343 \times 1.15 = 2700$ psi

A-2 Study to Optimize Bore

Assumptions:

1. System pressure is 2700 psi for 1/2 inch piston.
2. Next O-ring sizes are 9/16 or 7/16 inch.
3. Burst is 2.5 times system pressure or 6750 psi.

Larger Piston:

If piston increases to 9/16, area increases to 0.2485 in.sq. and pressure decreases to 6750 $(\frac{.1963}{.2485})$ = 5332 psi. For 1/2 inch piston, hoop stress is, with 0.25" wall:

$$S_t = 6750 \left(\frac{.5^2 + .25^2}{.5^2 - .25^2}\right) = 11{,}250 \text{ psi.}$$

To get same stress with 9/16 inch bore, wall radius is:

$$11{,}250 = 5332 \left(\frac{r^2 + (9/32)^2}{r^2 - (9/32)^2}\right), \ r = 0.470 \text{ inch.}$$

Thus, 0.030 inch could be shaved from the outer wall. However, this small advantage must be balanced against the problems associated with the larger piston. The larger piston requires a larger cutout of the rocker arm to accept the piston at full stroke. Also, the piston guides enlarge, lessening clearance between guide and fin lock housing. Concensus, leave bore at 1/2 inch.

Smaller Piston:

Now assume a 7/16 inch bore. Either the rocker arm (R) gets longer or pressure must increase. Trial layout says R can't increase because of envelope limit. Also, piston gets longer and increases beyond 50% extension beyond bore at full stroke.

If pressure is allowed to increase, it must increase by 23.5% because d = 7/16, A decreases by 23.5%. For same stress in wall r = 0.568 which is out of envelope since servovalve won't fit. Concensus, leave bore at 1/2 inch.

A-3 Valve Flow Rating

Assumptions:

1. At -65°F, MIL-H-5606 viscosity = 2800 cs.
2. Diameter of piston port holes is 0.090 in.
3. Length of piston port holes is 1.0 in.
4. Required flow for 350° per second is 1.42 in^3/sec.

Calculation:

Calculated pressure drop in one port in actuator ΔP_1 = 310 psi. Total drop ΔP_T = 620 psi. Available valve drop at -65°F:

$$2700 - 150 - 620 = 1930 \text{ psi at } 1.42 \text{ in}^3/\text{sec}.$$

Therefore, must order servovalve to give

$$Q = \frac{1.42}{.9}\sqrt{\frac{2700}{1930}} = 1.87 \text{ in}^3/\text{sec} \begin{matrix} +\ 0\% \\ -10\% \end{matrix}$$

at 2700 psi valve drop. This valve will give fin velocity of 461 deg/sec at 2700 psi valve drop at room temperature.

A-4 Side Forces on Piston, No Roller (Also see A-1)

Assumptions:

1. No rollers on rocker arm.

2. Steel cam on rocker arm and steel button on piston top.

   C_f = 0.78 (MoS_2 may rub off after many cycles of rubbing)

3. Assume Figure A-4 applies. Measurements from print.

4. Assume piston to bore C_f = 0.3.

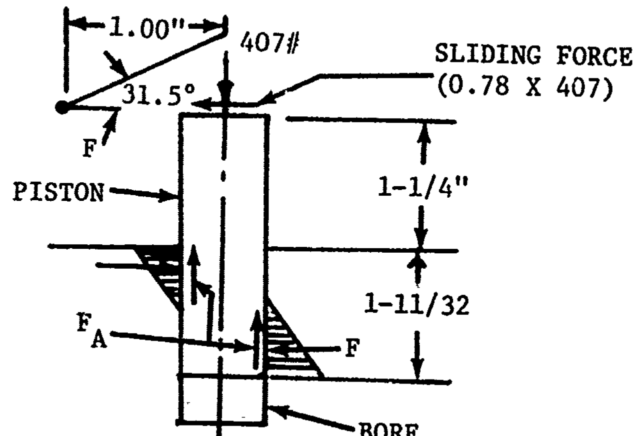


Figure A-4
Side Forces

Calculation Summary:

Moment on pistons due to rubbing of rocker arm on top = M_p; M_p = 397 in.lb.

Resultant force, F, in bore to counteract M_p; F = 443 lbs.

Resultant axial friction force, F_A, due to F ; F_A = 266 lbs.

Plus force from A-1 (due to offset of cam from C.L.):

F_{TOTAL} = 266 + 25.5 = 291.5 lbs. (axial force)

A-5 Roller Stress

On Piston:

Must not deform at proof pressure = 1.5 x 2700 = 4050 psi.

From Riechman/Timo:

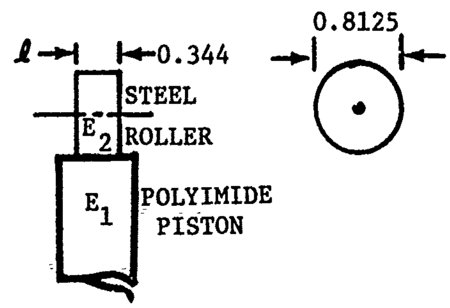


Figure A-5
Roller Forces

$$P' = \frac{P}{l} = \frac{4050 \times .196}{.344} = 2308$$

$$P = 0.59\sqrt{2P'\left(\frac{d_1 \times d_2}{d_1 + d_2}\right)\frac{1}{\frac{1}{E_1}+\frac{1}{E_2}}}$$

$d_1 = \infty$, $d_2 = .8125$

$E_1 = 4.5 \times 10^5$, $\frac{1}{E_1} = 0.000002222$ (Kinel 5505)

$E_2 = 30 \times 10^6$, $\frac{1}{E_2} = 0.00000003333$

$$P = 0.59\sqrt{2 \times 2308(0.8125)\frac{1}{.00000226}}$$

P = 24059 = 24059 psi

Kinel 5505 Brinell = 43,000 psi advertised

A-6

Cylinder Inner Hoop Stress: O.D. = 1.10, I.D. = 0.500

$$S = 2700 \times 2.5\left(\frac{(.55)^2 + (.25)^2}{(.55)^2 - (.25)^2}\right)$$

$$S = 6750\left(\frac{.3025 + .0625}{.3025 - .0625}\right)$$

$$S = 6750\frac{(.365)}{(.24)} = 10{,}266 \text{ psi}$$

(Avg. of 4 cylinders = 10,000 psi)

A-7 Fin Lock Pin Forces and Stress Summary

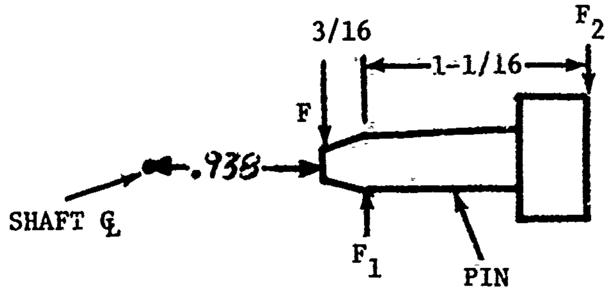


Figure A-7A
Pin Dimensions

Assumption:

Shaft torque is 500 in.lb.

Moment force = $F = \frac{500}{.938} = 533$ lbs

$F_1 = 627$ lbs

$F_2 = 94$ lbs

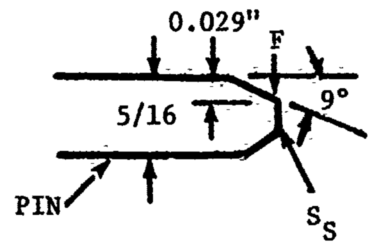


Figure A-7B. Pin Tip

Shear stress at F (Figure A-7B) across end of pin: $S_S = 10{,}660$ psi

(440F, Rockwell C57, $F_{sy} = 157{,}000$ psi)

A-7 Fin Lock Pin Forces and Stress Summary (Cont'd)

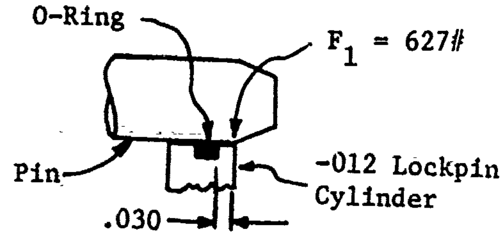


Figure A-7C. Pin Reaction

Bearing pressure at F_1 (Figure A-7C) on 0.030 inch section:

S_B = 66,880 psi (Stainless 303, F_{B_Y} = 125,000)

Actually, area behind O-ring takes some stress. Also, the 0.030 area will be made larger when detailed.

A-8 Bellville Spring Summary

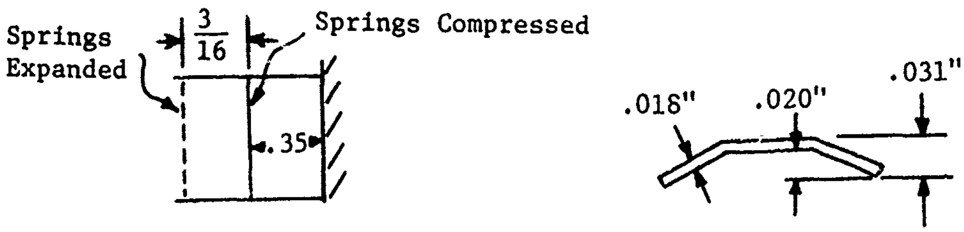


Figure A-8. Spring Dimensions

Diameter of springs: O.D. = 0.562, I.D. = 0.255 inch

Thickness of spring material: 0.018 ± 0.00025 inch

Free height = 0.031 inch

Available compression = 0.020 inch

Number of springs = 20 total in series, 22 permissible

Flat stack force = 67.2 lbs

Expanded stack force at pin extension limit = 42.8 lbs

A-9 Pin Retract and Staying Forces (Phase II)

Assumptions:

1. Torque load on shaft is 500 in.lb.
2. Hydraulic pressure available is 85% of 2000 psi.

Calculation Summary:

Spit-out force on pin due to 9° pin slope and 500 in.lb. load = 82 lbs.

Spring force from A-8:

Flat stack force = 67.2 lbs

Expanded stack force = 42.8 lbs

Pressurized O-ring force (estimate) = 80.5 lbs

Hydraulic force available at 85% system pressure = 300 lbs.

A-10 Pin Housing Stress Summary

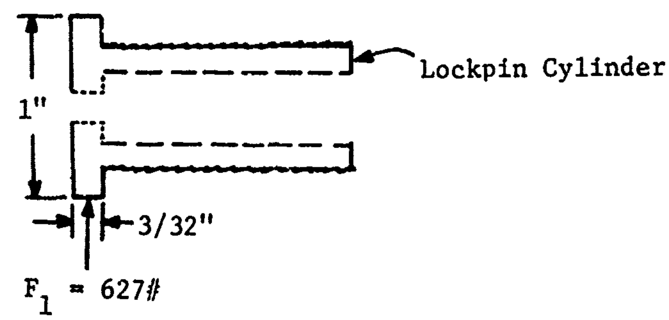


Figure A-10. Pin Housing

Bearing stress in polyimide assuming all pin reaction is taken out on 3/32 by 1 inch bearing area in Figure A-10:

S_b = 6700 psi

Allowable bearing stress from literature:

F_{b_u} = 32,500 psi

Pressurized Stress in Steel Bushing (-012)

Hoop stress at burst pressure:

S_t = 43,339 psi

Allowable in 303 stainless (yield)

F_{t_y} = 67,000 psi

Pressurized Stress in Plastic due to Expansion of Steel

S_t = 3178 psi

Allowable in plastic:

F_{t_y} = 10,000 psi

A-11 Potentiometer holder Force and Stress Summary

Force exerted on pot shaft due to axial rigidity of holder when shaft deflects 0.007 inch:

$P = 11.2$ lbs.

Max tensile stress in holder at this force level:

$S_t = 6160$ psi

Based on:

$$w = \frac{P}{16\pi D}\left[a^2 - b^2 - \frac{4a^2b^2}{a^2-b^2}\left(\ln\frac{a}{b}\right)^2\right]$$

where

$$D = \frac{Et^3}{12(1-\mu^2)}$$

$a = 1.00$ in.

$b = 7/16$ in.

$\mu = 0.22$

$w = 0.007$ in. deflection

$t = 0.030$ in.

$E = 2.2 \times 10^6$ psi

Material assumed is polyimide 5504.

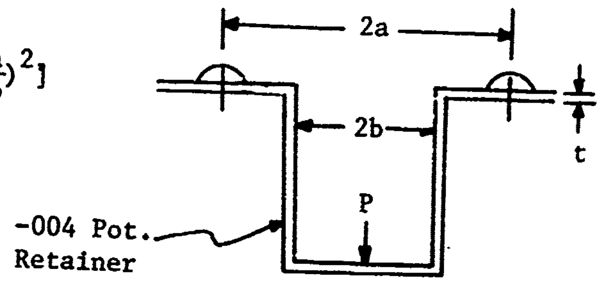
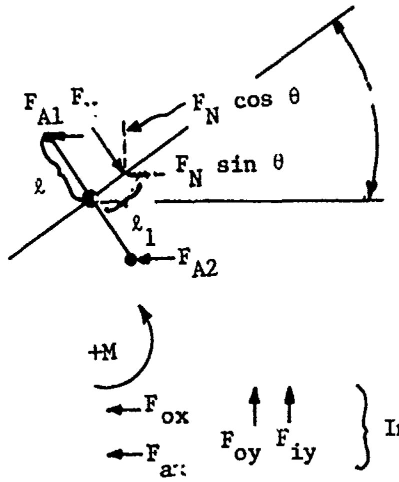


Figure A-11. Potentiometer Holder

A-12 Bearing Loads



Inner and Outer Bearing Reactions

Must Assume A Load and Load Point Diagram. This is From Proposal Where $F_N = 400\#$ From $F_N = 1600/Y = 400/X$

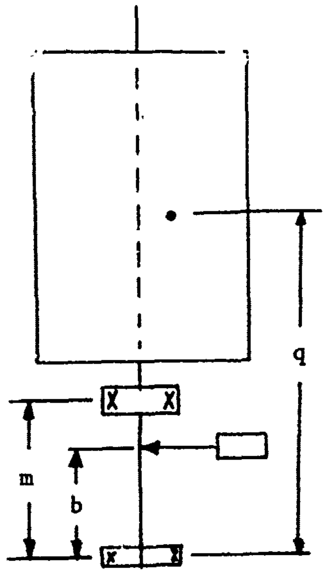


Figure A-12
Bearing Loads

Equations:

$\Sigma F_X = 0$

$F_N \sin\theta - F_{A1} - F_{A2} - F_{ox} - F_{ix} = 0$

$\Sigma F_y = 0$

$F_N \cos\theta = F_{oy} + F_{iy}$

$\Sigma M = 0$

$\ell_1 F_N = R\, F_{A1} \cos\theta - R\, F_{A2} \cos\theta$

$\ell_1 F_N = [R \cos\theta](F_{A1} - F_{A2})$

$\Sigma M_y = 0$

$qF_N \sin\theta - mF_{ox} - bF_{A1} - bF_{A2} = 0$

$\Sigma M_x = 0$

$qF_N \cos\theta = mF_{oy}$

From $\Sigma M = 0$

$$\ell_1 F_N = R(F_{A1} - F_{A2}) \cos\theta$$

Let $F_{A1} = (1620 + \Delta P)A$ and $F_{A2} = (1620 - \Delta P)A$

Where 1620 is a typical null pressure and ΔP is the deviation from null pressure caused by load.

Then: $P = \dfrac{\ell_1 F_N}{2AR \cos\theta}$ and

$$F_{A1} = (1620 + \frac{\ell F_N}{2AR \cos\theta})A, \quad F_{A2} = (1620 - \frac{\ell F_N}{2AR \cos\theta})A$$

A-12 Bearing Loads (Cont'd)

$\theta = 31.5°$

$F_N = 400$
$\cos\theta = 0.853$
$\sin\theta = 0.523$
$q = 6.406$
$m = 2.406$
$b = 1.686$
AR effective = 0.21
$A = 0.196$
$\ell = 1.0$

Results:

$F_M = +536$
$F_A = +98.7$
$F_{iy} = -567$
$F_{ox} = +112$
$F_{ix} = -538$
$F_{oy} = +908$
F_o resultant = 915# Max Outer Bearing Load
F_i resultant = 782# Max Inner Bearing Load

Radial Load Limits

$\left.\begin{array}{l}\text{Inner B542DD} = 5950\#\\ \text{Outer KP21B} = 9840\#\end{array}\right\}$ (Selected for Size)

Bearing Load at $\theta = 0°$ (Assuming no Aero Load, $F_N = 0$)
Piston Forces = 1620 x 0.196 x 2 = 640#

$$\frac{640 \times 1.686}{2.406} = F_{Outer} = 450\#$$

$$F_{Inner} = 190\#$$

APPENDIX B

PROCESS PLAN FOR MOLDING P/N 60500309
HOUSING, MOLDED PLASTIC ACTUATOR

1. To fabricate the part the following equipment, material and personnel are required:

   1.1 Equipment Required:

   Press: 150 ton EQ # 790017 (or equiv.)
   Dielectric Heater:
   Mold (with Cal-rod heaters):
   Variacs:
   Pyrometers:
   Temperature Recorder:
   Funnel:
   Teflon Box and Board:
   Asbestos Blankets:
   Asbestos Gloves:
   Pliers:
   Safety Bar:
   Scale:
   Process Sheet (for recording data)
   Clamp-on Ammeter

   1.2 Material Required: Rhodia 5504 - 60% glass filled polyimide (525 grams/part)

   1.3 Personnel Required: Press operator, dielectric heater operator, two helpers.

2. Pre-molding setup and checkout procedure:

   2.1 Mold Preparation:

   Setup mold in press and check alignment
   Secure mold in press
   Check adjustment and operation of eject mechanism
   Open mold
   Check installation of inserts and fork
   Adjust platen stroke for maximum speed
   Check thermocouple operation
   Check cleanliness
   Energize platen heaters

   2.2 Dielectric Heater Preparation:

   Connect air hose
   Energize breaker
   Turn on filament power

3. Molding procedure

3.1 Obtain process sheet and record date, time and names of press operator, dielectric heater operator and two helpers. NOTE: All data is to be recorded on process sheet.

3.2 Coat mold with release agent - record.

3.3 Set all variacs at 130; measure current in lines to Cal-rods - record.

3.4 Weigh charge and place in teflon box - record.

3.5 Set dielectric heater plates - record.

3.6 Place loaded teflon box in dielectric heater and set timer - record.

3.7 Stabilize platen temperatures - record.

3.8 Reset variacs to obtain initial mold temperature - record.

3.9 When dielectric heat cycle is within approximately 5 seconds of completion start the following sequence:

- a Press operator opens press to full open position
- b Helper no. 1 inserts safety bar and removes door from mold insulator
- c Helper no. 2 lays down asbestos blankets
- d Press operator inserts funnel
- e Dielectric heater operator inserts charge and pulls funnel
- f Helper no. 1 pulls safety bar
- g Press operator closes press and sets initial mold pressure and measures temperature - record

3.10 Set platen temperatures and variacs to obtain final mold temperature - record.

3.11 When B staging (liquification) is apparent from observing movement of the mold, increase press loading to the final mold pressure and observe time - record.

3.12 Observe final mold temperature and time - record.

3.13 Reduce punch temperature - record.

3.14 Open press 2 inches, remove T-bar and open press fully.

3.15 Install safety bar.

3.16 Remove part using pliers, observe appearance - record.

3.17 Repeat procedure starting at paragraph 3.1.

PROCESS SHEET FOR PLASTIC ACTUATOR

Date ____________ Time ____________

Press Operator ____________ __________

Dielectric Operator ____________________

Helpers ____________________

____

Mold Release Agent Used ____________________

Cal-Rod Current with Variac Set @ 130 ____________

Charge Weight (525) ________ grams ____________

Dielectric Heater Settings (18) __________ turns

(350) __________ seconds

Platen Temperatures (600°F) Upper ________ °F

Lower ________ °F

Variac Settings to Obtain Initial Mold Temperature: ________ ________

________ ________

Initial Mold pressure (15) ________ tons

Initial Mold Temperatures (240-250) ________ °F ________ °F

________ °F ________ °F

________ °F ________ °F

Variac Settings to Obtain Final Mold Temperatures ________ ________

________ ________

Final Mold Pressure (44) ______ tons, Time ______

Final Mold Temperatures (445-480) ________ °F ________ °F ________ °F

________ °F ________ °F ________ °F

Time: ______(Start) ______(Finish) (16 min)

Punch Temperature ______ °F @ ______ Time (Start)

______ °F @ ______ Finish (10 min)

Notes on Appearance of Part at Removal: ____________________

____

____

Additional Notes and Recommendations: ____________________

____

____

PRICE ANALYSIS

COST ELEMENT	COLUMN NO. 5 MANUFACTURING MATERIAL			COLUMN NO.			COLUMN NO. 6 SUMMARY		
		Units	DOLLARS	MAN HOURS OR MID POINT	RATE	DOLLARS	MAN HOURS OR MID POINT	RATE	DOLLARS
Servovalve	$200	2,000	400,000						
Potentiometer	$ 15	2,000	30,000			Tooling			107,855
Molding Compound	$11.25	2,000	22,500			Des. & Qual. Test			12,825
Roller Bearing	7.72	2,000	15,440			Manufacturing Mat'l			792,755
Control Bearing	2.49	2,000	4,980			Fabrication			342,541
Control Bearing	2.74	2,000	5,480			Production Testing			47,471
Misc. Nuts, Bolts, Pins,									
Washers, etc.	3.95	2,000	7,900			Total			1,303,447
Bolt, Lock Pin, Cyl. &Retain.	15.35	2,000	30,700						
Subtotal			517,000			Unit Price			651.72
Unidentifiable Materials	1.38%		7,135						
Subtotal			524,135						
N.P.A.	10%		52,414						
Subtotal			576,549						
G & A	25%		144,137						
Total Cost			720,686						
Profit			72,069						
Price			792,755						

APPENDIX C

PRICE ANALYSIS

COST ELEMENT	COLUMN NO. 3 PRODUCTION TEST			COLUMN NO.			COLUMN NO. 4 FABRICATION		
	MAN HOURS OR MID POINT	RATE	DOLLARS	MAN HOURS OR MID POINT	RATE	DOLLARS	MAN HOURS OR MID POINT	RATE	DOLLARS
PTO	1,433	6.23	8,928			Tooling	486	6.79	3,300
Quality	600	6.59	3,954			Factory	10,187	5.40	55,010
						PC/IE	2,037	5.77	11,753
Subtotal			12,882			Shipping	459	5.79	2,658
Manufacturing Overhead		168%	21,642			Quality	2,399	6.59	15,809
Subtotal			34,524			Subtotal			88,530
G & A		25^	8,631			Mfg. Overhead		168%	148,730
Total Cost			43,155			Tooling Mat'ls			1,298
Profit			4,316			Shipping Mat'ls			2,000
						Quality Travel			930
Price			47,471						
						Unident. Mat'l			244
						Subtotal			241,732
						G & A		25%	60,433
						Require.&Controls			9,236
						Total Cost			311,401
						Profit			31,140
						Price			342,541

PRICE ANALYSIS

COST ELEMENT	COLUMN NO. 1 TOOLING			COLUMN NO.			COLUMN NO. 2 ENGINEERING DESIGN TEST and QUALITY TEST		
	MAN HOURS OR MID POINT	RATE	DOLLARS	MAN HOURS OR MID POINT	RATE	DOLLARS	MAN HOURS OR MID POINT	RATE	DOLLARS
Molds - EPL.						Tech A	360	6.57	2,365
Design	320					Sr. Engr. (45)	160	8.25	1,320
Preform	680	6.68	15,999			St. Engr. (47)	40	10.00	400
Finish	1,235								
Tryout	160								
Subtotal			15,999			Subtotal			4,085
Engr. Overhead		100%	15,999			Engr. Overhead		100%	4,085
Test, Assembly, Metal Fab						Test Fixture Material			1,100
Fixtures - Tooling	1,665	6.79	11,305			Unident. Material			57
Subtotal			11,305			Subtotal			9,327
Mfg. Overhead		168	18,992			G & A		25%	2,332
Mold Materials			8,000			Total Cost			11,659
Tooling Materials			5,938			Profit			1,166
Unident. Material			1,058			Price			12,825
Subtotal			77,291						
G & A		25%	19,323						
Requirements & Controls			1,436						
Total Cost			98,050						
Profit			9,805						
Price			107,855						

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