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DESIGN AND DEVELOPMENT OF THE QUAD REDUNDANT SERVOACTUATOR FOR THE SPACE SHUTTLE SOLID ROCKET BOOSTER THRUST VECTOR CONTROL

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SUMMARY

The intent of this paper is to describe the design of the servoactuator used for thrust vector control of the Space Shuttle Solid Rocket Booster. To accomplish this, a description of the design and theory of operation is accompanied by highlights from the development and qualification test programs. Specific details are presented concerning major anomalies that occurred during the test programs and the corrective courses of action pursued.

INTRODUCTION

The National Aeronautics and Space Administration (NASA)/Marshall Space Flight Center (MSFC) in Huntsville, Alabama, has had responsibility for the design, development, flight qualification testing and procurement of thrust vector control (TVC) servoactuators for many large space vehicles. These applications included all stages of the Saturn I, IB, and V vehicles, Space Shuttle Main Engine (SSME) ground tests, and the Space Shuttle Solid Rocket Booster (SRB).

With the Space Shuttle came the first demand for a TVC servoactuator with some degree of redundancy. A three-channel version was developed for the SSME TVC ground testing. This design possessed servovalve bypass capability and a return to null mechanism. Out of this configuration evolved the final design for the SRB TVC servoactuator. This design basically encompasses four servovalve channels to create a fail-operate, fail-operate redundancy scheme in those components susceptible to common contamination and electrical failures.

Not only did the Space Shuttle create a demand for redundancy, it also demanded an actuator designed for a 20-mission lifetime. Furthermore, the vibration environment for each mission was to be more hostile than any of the single-mission Saturn series TVC servoactuators. Added to these requirements, the SRB TVC servoactuator must absorb splashdown loads and remain leakproof in the saltwater environment.

DESIGN AND PERFORMANCE CRITERIA

Objectives

Two SRB's are attached to the Space Shuttle external tank to provide primary thrust during the flight ascent phase. To provide TVC of the booster, two servoactuators are attached to the outboard side of each of the SRB's as shown in figure 1. These actuators are mounted in the "tilt" and "rock" planes which are at $\pi/4$ rads (45 deg) to the shuttle pitch axis. The basic objectives were to provide required thrust vector gimballing and rate capability against loads imposed by the SRB nozzle. Unlike the fixed pivot point gimbal bearing of Saturn vehicle engines, the SRB nozzle is attached to the Solid Rocket Motor (SRM) by a flex seal gimbal bearing which is protected by an ablative boot. This design not only introduced the complexity of locating the nozzle pivot point, but also imposed the high restraining torques characteristic of the flex seal and the protective boot. A last and very important primary objective was to provide the highest degree of redundancy attainable within the given restraints of weight, envelope, cost and scheduling.

Performance Requirements

Physical sizing of the actuator was dictated by the flexible bearing and boot nozzle restraining torque. These loads coupled with the required gimbal rates, nozzle/structure dynamics and installation geometry sized the main piston area and basic loop gain of the system. Computer simulations were used to size a dynamic pressure feedback (DPF) mechanism. From this mechanism, the load differential pressure is sensed and shaped to stabilize the first resonant mode of the gimballed nozzle mass and attaching compliances. Functional, environmental and dynamic requirements pertinent to the SRB TVC servoactuator design are summarized in Tables I, II and III.

DESCRIPTION AND OPERATION

General

The TVC servoactuator is a four channel proportional control device that operates normally after one or two channel failures. These failures originate from the drive electronics or within the servoactuator control channels. Figure 2 shows an assembly drawing of the TVC servoactuator, and figure 3 shows the redundant components and feedback. From these figures the major components can be located. These major components are: four servovalves (conventional torque motor/nozzle-flapper first stage with second stage spool); four differential pressure transducers; four isolation valves; four dynamic pressure feedback modules; mechanical position feedback mechanism; power stage spool; main piston; transient load relief mechanism; hydraulic supply switching and prefiltration valve (not shown) and the lock and manual bypass valve (not shown). The servovalves are arranged in a "V-4" configuration. Figure 4 shows a general servoactuator block diagram. Figure 5 contains the simplified linear mathematical block diagram with the four servovalves lumped into one channel with the power valve. The parameter list of table III together with figure 5 define the basic linear mathematical model of the servoactuator. Figures 2 through 5 can be used in conjunction with the following discussion to understand the basic system operation.

Normal Operation

Two pressure sources, primary (P1) and secondary (P2), are supplied from separate auxiliary power units (APU) to ports on the main actuator body. From these ports, the fluid enters a dual function, three-position hydraulic supply switching and prefiltration valve. Manual external positioning to the prefiltration mode allows flushing action for both P1 and P2 pressure sources while bypassing the critical valve components with potentially contaminated fluid. In the normal mode of operation, a compression spring and differential area allow source P1 (with both P1 and P2 present) or source P2 (with P2 only present) to pass on to the servoactuator. With normal supply source P2, supply switching will occur when P1 drops to $1.413 \times 10^7 \pm 1.034 \times 10^6$ N/m^2 (2050 ± 150 psi). Recovery of source P1 to $1.689 \times 10^7 \pm 1.034 \times 10^6$ N/m^2 (2450 ± 150 psi) will cause the valve to revert to the normal mode of operation.

From the supply switching valve, high pressure fluid is routed directly to the power valve and simultaneously through a replaceable 10 micron (nominal) filter to the redundant sections of the servoactuator (servovalves, valve pressure transducers, and isolation valves).

Filtered fluid is supplied to the four servovalves, which are of the two-stage mechanical feedback type units. The first stage consists of a 0.226 Nm (2 in. lb) torque motor and a conventional four-leg orifice bridge. Two fixed orifices and a movable flapper positioned between two nozzles make up the orifice bridge. A closed center spool with pressure feedback to a stub area on either end of the spool make up the second stage. This pressure feedback reduces the servovalve pressure gain and minimizes force fight between valves as they drive the power valve spool. Second-stage spool position is mechanically fed back to the torque motor through a wire spring element extending from the flapper into a groove on the spool. The feedback torque is proportional to spool position.

Command input currents to the torque motors cause the flapper to rotate, creating an imbalance in the orifice bridge. This imbalance produces a net driving force on the second stage spool of each servovalve creating an output differential pressure. These outputs are force summed on the large power spool. Movement of this spool meters flow to the main piston and controls piston velocity. The power spool position is mechanically fed back to the torque motors through wire springs extending from the flappers to grooves on the power spool.

The load differential pressure is developed across the ports of the power valve spool as a function of actuator load and spool position. This differential pressure is fed back through four dynamic pressure feedback modules to a set of nozzles directed onto the flappers of each servovalve. These networks provide a frequency sensitive-load damping feedback signal. The system remains stiff against static loads and dissipates energy created by the resonant frequency of the nozzle and structure.

The main power piston is located on the actuator centerline. Two bearings, one in the actuator body and the other in the cylinder, guide the piston rod. The piston position is sensed mechanically by a "scissors-like" mechanism consisting mainly of two elements pinned together producing four ends. Two ends of the scissors assembly are spring loaded to ride an internal conical cam located within the piston rod. With one end pivoted from a rigid point on the actuator body, the fourth end becomes the output member. Motion of the piston causes the feedback assembly to open and close causing movement of the output member proportional to piston position. Through a linkage, the output of the scissors mechanism drives two spring loaded cages resulting in a negative feedback torque to the servovalve flappers. The result is piston position proportional to servovalve input.

A hydraulic lock valve located in the actuator body locks the piston in a fixed position in the absence of hydraulic pressure. When system pressure exceeds $4.14 \times 10^6\ N/m^2$ (600 psi), a spring loaded spool moves allowing fluid to pass from the power valve output ports through the lock valve to the piston.

To absorb the water impact loads during splashdown, a large transient load relief valve is located within the piston assembly. This valve senses the transient load pressure across the piston and opens to bypass flow through the piston when the load differential pressure transient exceeds $2.48 \times 10^7\ N/m^2$ (3600 psi). The valve has an integrating mechanism incorporated to prevent opening for static or low frequency loads. This device protects the actuator and the attaching structure/SRB nozzle from splashdown loads.

Also located within the actuator are two monitoring devices, the piston position and load pressure transducers. The scissors mechanism moves the free end of a cantilevered beam instrumented with strain gages to convert piston position into a proportional output. The load pressure transducer senses the differential pressure output of the power valve.

Redundancy Management

The primary purpose of redundancy within the SRB TVC servoactuator is to eliminate catastrophic actuator behavior resulting from in-line control component failures. Included are failures within the computer, control electronics and the servoactuator redundant components. Not included are failures of simplex servoactuator components for which redundancy was not practical. These include the power valve, main piston, mechanical position feedback mechanism and the transient load relief valve. Such components were therefore designed for optimum reliability. These simplex components utilize large driving forces and from past experience have a record of proven reliability. In the redundant control components, the servoactuator can tolerate two failures with no significant degradation in performance.

Actuator failures are generally the results of fluid contamination which causes restricted orifices and/or excessive spool stiction. To execute the redundant capabilities of the servoactuator, differential pressure transducers are used in conjunction with a remote computer, the Ascent Thrust Vector Controller (ATVC), to detect and isolate offending servovalves. The transducers sense the pressure differential across the output ports of each servovalve. Basically, the transducer is a spring centered piston with a Linear Variable Differential Transformer (LVDT) coil mounted concentric to the piston. The piston drives the LVDT probe to produce an output voltage proportional to the differential pressure.

External to the servoactuator, the ATVC monitors the signals from these transducers. Here, the "failure detection, isolation and recovery" (FDIR) logic determines the status of the individual servovalves. If the output pressure of a given servovalve exceeds $1.52 \times 10^7 \pm 1.38 \times 10^6$ N/m^2 (2200 $\pm$200 psi) a timer is started. Should the differential pressure remain above this level for 120 ms, that particular channel is isolated from the system by terminating the output of the servovalve. To accomplish isolation, a solenoid operated isolation valve (one per channel) internal to the servoactuator is utilized. Energizing the solenoid results in the application of system pressure to the end of a spring loaded spool. The spool is driven to a position which blocks the servovalve output pressure from one side of the servovalve and connects the other side to both ports on the power valve normally driven by the servovalve in question. Thus, control of the offending servovalve and force fight among valves are eliminated.

DEVELOPMENT AND QUALIFICATION TEST PROGRAM

Development testing of the SRB TVC servoactuator was accomplished jointly by the servoactuator vendor and MSFC. To verify the design approach, a two-phase development test program was devised and conducted at the facilities of the vendor. Phase one consisted of component and subassembly testing to verify performance, life and reliability. The components and subassemblies undergoing the vendor conducted development tests were: 4 servovalves; 8 DPF modules; 12 servovalve differential pressure transducers; hydraulic supply switching valve; hydraulic lock valve; 4 solenoid isolation valve assemblies; transient load relief valve; and the power valve assembly. These elements were subjected to the life, environmental, functional and performance criteria of MSFC document 16A03000 (SRB TVC Electro-Hydraulic Servoactuator Design and Procurement Specification). Performance was verified by determining pressure gains, feedback loop gains, stability, friction levels, linearity, null shifts, pressure switching levels and leakage. Phase two verified the performance of a complete servoactuator assembly, the development test unit. Although representative of the deliverable actuator's functional design, the development test unit fabrication was not restricted to production unit tooling, methods or NASA quality control. Two objectives were met with the development test unit. First, the design approach was proven to meet the functional and performance requirements of 16A03000; and second, an acceptance test procedure (Moog report MR A-2237) applicable to production hardware was developed.

The first production unit of flight configuration was the engineering test unit (S/N 001). This servoactuator was delivered to MSFC and was subjected to a "prequalification" test program. The tests centered around the vibration environment imposed by 16A03000. Previously, no complete SRB servoactuator had been subjected to these vibration requirements (see Table II). The engineering test unit was periodically removed to the dynamic inertia simulator and load-flow test bench at MSFC to verify that performance criteria were maintained as specified. The purpose of this test program was to detect and modify any deficient elements and thus minimize problems with the formal qualification test program to follow.

Two servoactuators (S/N's 005 and 008) were designated to undergo the formal qualification test program as specified in MSFC's "SRB TVC In-House Qualification Test Procedure" (MSFC document MTCP-CC-SRB-529). At present these tests simulating 20 missions are approximately 5 percent complete.

Another effort, which began prior to and runs parallel with the these tests, verifies performance of the servoactuator with a hot fired APU. These tests are being conducted at the MSFC verification test stand. Initially, these tests were conducted without an active ATVC to verify compatibility of the APU and the servoactuator. Currently, these tests are utilizing an active ATVC, which integrates the APU and servoactuator with the failure detection, isolation and recovery logic in the ATVC. The objective of these tests is to verify the total system performance.

At Thiokol's Wasatch Division, the SRB motor was static test fired in the horizontal position. Four development SRM's (DM1 through DM4) and three qualification SRM's (QM1 through QM3) were to be tested. All test SRM's have been fired as of mid February 1980. Two servoactuators (S/N's 006 and 007) were used for DM3 through QM3. SRB servoactuators were not available for the DM1 and DM2 firings. A test plan for each development motor static test was jointly agreed upon by the SRB vendor and MSFC. Each test plan featured unique and extensive test duty cycles for both the tilt and rock actuators. Various amplitudes of sine wave, ramp and step commands were input to the servoactuators. A frequency response for each static firing test was conducted on the tilt servoactuator, spanning the frequency range from 0.2 to 20 Hz at an amplitude of ±6 percent of total stroke. The commands were delivered to the servoactuator through the ATVC, however, the FDIR logic was not active. These static firing tests enabled testing of the servoactuator with the flight type mating structures. Thus, for the first time the first mode resonance of the flight type SRB nozzle and attaching structure was determined. For the final SRB Static Test (QM-3), a small hydraulic accumulator was installed in the Rock Servoactuator fluid supply line. The frequency response normally conducted on the tilt system was conducted on the Rock Servoactuator. The effects of the accumulator on the pump and hydraulic line modes observed on prior tests were investigated.

DEVELOPMENT AND QUALIFICATION PROBLEMS

General

In general, the SRB TVC servoactuator development and qualification programs have been relatively trouble free. The prior development of the SSME TVC ground test servoactuator by the same vendor and MSFC are mainly responsible. Although the SSME ground test servoactuator used three control channels, many of the components shared common designs.

Most problems were of a routine nature and required minimal efforts to analyze and implement corrective action. Such problem areas include: excessive spool friction; improper frequency response; meeting the required linearity and gain criteria; adjustment of feedback gains; and external and internal leakage and seal problems. These problems are of a relatively trivial nature and will not be dealt with. The intent of this section is to address the three most significant anomalies encountered during the development and qualification programs. These three problem areas were deemed "most significant" because of the level of concern and effort devoted to the problems' natures, analyses and solutions.

End of Stroke Instability

A major problem was detected during testing of the development test unit. A considerable vendor and NASA effort was expended to determine the problem root cause and recommended course of action.

Simply stated, a chattering or oscillation at a frequency of 45 Hz was observed as the piston was moved into or out of the stroke extremities. The oscillations occurred during operational modes only when the piston came within 5.1×10^{-4} m (0.02 in.) of the end of stroke (extend or retract direction) and physically bottomed on the cylinder end. The resultant response of the servoactuator output differential pressure was a 45 Hz square wave with amplitude varying from zero to full system pressure. Superimposed upon this basic wave was a higher frequency (1000 Hz) resulting from the line dynamics of several oil supplying passages. Driving the piston harder into the stops caused the oscillation to cease. The oscillation occurred also in a nonoperational mode when the actuator was pressurized with the mid-stroke locks installed. This lock is an external shipping and handling device that mechanically holds the piston in the null position. Since the mid-stroke locks are always removed for operation, this condition posed no problem.

The development test unit was instrumented with pressure transducers in all areas where pressure transients could be produced by the oscillations. High pressure spikes were detected at the ends of several long, small passageways terminating with small restrictions. High frequency spikes were recorded in excess of 6.895×10^7 N/m^2 (10,000 psi). To eliminate these spikes, orifices were installed in the lines of the four servovalves and main-cylinder differential pressure transducers.

An inherent high loop gain around the dynamic pressure feedback loop at the end of stroke was determined to be the root cause of the oscillations. The gain of this loop includes the actuator-oil stiffness as well as the dynamics of the DPF mechanism, servovalves and power valve. Figure 5 shows the simplified linear block diagram of the nominal servoactuator with all four servovalves lumped into one channel. Table III defines the associated parameters. When operating normally, the actuator piston acts as a near ideal integrater, converting power valve flow into piston position. When the piston physically bottoms against the cylinder end, the error signal from the piston position feedback is driven to zero. Under these conditions, figure 6 becomes the representative block diagram of the system. The differential pressure output of the piston then becomes a function directly of the fluid bulk modulus (B_o) and indirectly of the line volume (V_o) from the power valve to the actuator cylinder. The quantity "B_o/V_o" = 7.384×10^{12} N/m^2/m^3 (17,550 psi/in.^{3}) resulting in an open loop gain of 493 sec^{-1} for figure 6. This gain is more than sufficient to drive the loop unstable.

Several factors contribute to this instability. These are the high flow gain of the total valve assembly, relatively high DPF loop gain and the frequency response of the three stage valve configuration. At the high gain encountered at the end of stroke, the servovalve dynamics are not adequate to maintain stability. This problem was never encountered on the Saturn vehicles' servoactuators. This is attributed to more common use of two stage servovalves and lower DPF gains on the Saturn vehicles.

Attempts were made to introduce damping by adding piston bypass flow at the end of stroke. The large flow quantity required, combined with the complexity and reduced overall reliability, made this approach unacceptable. Because of the modification difficulties encountered, tests were conducted at MSFC to establish the effects of the oscillation on the SRB and associated structure. These tests verified no detrimental effects were imposed by the oscillations upon the SRB, the attaching structure and the actuator itself. At the vendor's facility the servoactuator was cycled in and out of the ends of stroke for two hours with no detectable change in performance.

Because of the discrete occurrence of the oscillation within the last 5.1×10^{-4} m (0.02 in.) of the end of stroke and the benign detrimental effects, a redesign of the servoactuator was not justified. A modification to the flight ATVC will limit current commands to the servovalves such that the servoactuator will not be driven into the end of stroke.

Position Feedback Cam Failure

During prequalification vibration testing on the engineering test unit (S/N 001) at MSFC, a failure of the piston position feedback cam occurred. When the failure occurred, the servoactuator was undergoing the boost random vibration input to the rod (nozzle) end of the longitudinal axis. The cam fractured around the total periphery at the small end of the conical section where the cone blends into the end flange. A cross section of the cam is shown in figure 2. A new cam was instrumented with strain gages and accelerometers and installed into the servoactuator.

Using the data collected in subsequent tests, a fatigue failure analysis was performed by MSFC personnel. A finite element model was utilized and showed a maximum stress of 3.158×10 N/m^2 (4580 psi) on the inner radius of the cam. The stresses were shown to be low away from the radius. The failed cam was made from 6061 aluminum which was hard anodized. The S-N diagrams for this material showed that the fatigue strength was reduced approximately 40 percent by anodizing. Test data showed the cam/support to have a high Q resonant frequency of 1400 Hz. This mode, coupled with the reduction in the fatigue characteristics of the material, was shown to result in a fatigue failure of the cam.

An analysis by MSFC's Materials and Processes Laboratory was made of the failed part. Examination of the fracture surface showed that contact between fracture halves occurred during longitudinal oscillating applied stress. A metallurgical cross section of the cam fracture showed that initiation probably occurred in the hard anodize at the inner radius machined transition point between the cam cone and end. The fracture then propogated through the parent metal. Susceptibility of this material and configuration to fatigue induced failure was further exemplified by the presence of multiple longitudinal cracks in the hard anodize. It was recommended that the material be changed to a high strength material such as A-286 stainless steel.

The cam was redesigned using A-286 stainless steel with other modifications. Retaining guides were added to prevent the small section from significant movment should a failure occur. Secondary retention was also implemented to hold the cam in the piston. The new cam design was then tested to the vibration criteria with no problems encountered.

Servovalve Differential Pressure Drift

The SRB static test firings at Thiokol's Wasatch Division facility identifidd a servovalve differential pressure drift phenomenon not observed on previous tests. The problem was detected on the DM-3 static test article, the first SRB development motor firing with flight configuration TVC servoactuators installed. The output null differential pressures of all servovalves drifted with time, starting at near zero and increasing to the $4.1 \times 10^6 - 5.5 \times 10^6$ N/m^2 (600-800 psi) range toward the end of the 120 sec duration firings. Referring to the layout of figure 3, valves "A" and "C" drifted with a positive polarity while valves "B" and "D" drifted with a negative polarity.

After observance of this phenomenon, review of previous tests from the verification test stand at MSFC revealed similar behavior. A number of special tests were conducted on MSFC's inertia simulator and flow bench to attempt reproduction of the pressure drift. However, these tests, using the facility hydraulic supply source with a large reservoir, exhibited no pressure drift with time. After review of the various tests, it became apparent that every test conducted with the APU supplying hydraulic fluid produced similar servovalve pressure drifts.

The drift characteristics were analyzed and verified by test and analysis to be a function of the time rate of change of hydraulic fluid temperature. The rate of fluid temperature change is directly related to the power dissipation in the hydraulic fluid and the total hydraulic fluid volume. The power dissipation in the hydraulic fluid is a function of the commanded duty cycle. The fluid volume in the SRB TVC system is approximately 26.5 liters (7 gal). Since the fluid is in direct contact with the power valve body, increases in the fluid temperature causes the aluminum power valve body to expand at a different rate than the cage assembly and steel power spool. The difference in expansion of the power valve body with respect to the cage assembly causes an error torque on the flapper assembly. This error torque results in a pressure output from each of the servovalves. The expansion causes valves A and C to move in the direction opposite to valves B and D. Thus the output differential pressures of valves A and C drift with a polarity opposite to that of valves B and D. The summation of these differential pressures was always zero (within the power spool friction force levels) at any given time. Therefore, the power spool and main actuator piston positions were unaffected by the pressure drift.

Since these differential pressure outputs are sensed and utilized by the ATVC for failure detection and isolation, the impact of the pressure drift phenomenon was investigated. It was confirmed by the ATVC vendor that the magnitudes and rates of the pressure drifts observed were well within design tolerances. The development motor static firing tests and extensive testing at MSFC showed the pressure drifts to be very predictable with no detrimental effects on system performance. Also, the servoactuator test duty cycles were much more severe than those expected in flight. Therefore, a redesign was not deemed necessary.

CONCLUDING REMARKS

As of January 1, 1980, the design and development of the SRB TVC servo-actuator is complete. The engineering test unit has undergone a 20-mission flight vibration test program at MSFC with miniminal redesign. A number of verification tests at MSFC with hot-fired APU's have been successfully completed. Four SRB's equipped with flight type servoactuators have been static test fired with no failures. The final qualification test motor (QM-3) was test fired in mid February 1980. With the formal 20-mission qualification test program presently underway, it appears that the present configuration will be the final flight version of the SRB TVC servoactuator.

TABLE I. DESIGN AND PERFORMANCE REQUIREMENTS

Operating fluid	MIL-H-83282
System pressure, N/m^2 (psi)	$2.068 \times 10^7 - 2.241 \times 10^7$ (3000-3250)
Burst pressure, N/m^2 (psi)	5.602×10^7 (8125)
Proof pressure, N/m^2 (psi)	3.361×10^7 (4875)
Weight, N (lb)	1.446×10^3 (325)
Length at null, m (in.)	1.346 (53)
Rated input signal, A (mA)	$\pm 5 \times 10^{-2}$ ($\pm$50)
Stroke, m (in.)	$\pm 1.626 \times 10^{-1}$ ($\pm$6.4)
Piston effective area, m^2 (in.^{2})	2.085×10^{-2} (32.32)
Moment arm, m (in.)	1.819 (71.6)
Output force, maximum, N (lb)	467,000 (105,000)
Rated load, N (lb)	2.818×10^5 (63,360)
Rate at rated load, m/s (in./sec)	0.151 - 0.212 (5.95-8.33)
Frequency response:	
Bandpass, Hz	2.5 - 3.5
Phase lag at 1 Hz, rad (deg)	0.349 (20)
Load 1st mode resonant frequency, Hz	13.8
Internal leakage, m^3/s (gpm)	1.893×10^{-4} (3)
Intersystem leakage, m^3/s (gpm)	2.145×10^{-6} (0.034)
Null shift, A (mA)	2.37×10^{-3} (2.37)
Hysteresis, A (mA)	1.15×10^{-3} (1.15)
Threshold, A (mA)	5×10^{-4} (0.5)
Null bias, m (in.)	1.397×10^{-3} (0.055)
Pressure gain:	
Servovalve, $N/m^2/A$ (psi/mA)	$3.447 \times 10^9 \pm 8.619 \times 10^8$ (500 $\pm$ 125)
Power valve, $N/m^2/A$ (psi/mA)	3.448×10^{10} (5000)
Temperature operating range, K (°F)	266.5 to 338.7 (20 to 150)
Water pressure, N/m^2 (psi)	4.895×10^5 (71)
Water entry pressure, N/m^2 (psi)	1.069×10^6 (155)

TABLE II. DESIGN AND TEST VIBRATION REQUIREMENTS

Criteria	Input Source	Axis	Level	Total exposure time, sec/axis
Vehicle dynamics	Nozzle	Radial Tangential Longitudinal	3.7 g peak 3.7 g peak 2.4 g peak	120
	Aft skirt	Radial Tangential Longitudinal	3.7 g peak 3.7 g peak 1.0 g peak	150
Flight random	Nozzle	All	26.5 g rms	2640
Liftoff random	Aft skirt	Radial Tangential Longitudinal	5.0 g rms 6.3 g rms 6.3 g rms	250
Boost random	Aft skirt	Radial Tangential Longitudinal	6.1 g rms 9.2 g rms 9.2 g rms	880
Re-entry random	Nozzle	Radial Tangential Longitudinal	14.7 g rms 14.4 g rms 14.4 g rms	660
	Aft skirt	Radial Tangential Longitudinal	11.2 g rms 12.7 g rms 12.7 g rms	660

TABLE III. SRB TVC SERVOACTUATOR PARAMETERS

Symbol	Description	Units	Value
I_e	Command input current	A (mA)	±0.05 (±50)
K_{TM}	Torque motor gain	N-m (in.-lb/mA)	4.519 (0.04)
K_V	Valve flow gain	m^3/s/N-m (Cis/in.-lb)	0.212 (1460)
ζ	Valve parameters		0.7
ω	Valve parameters	s^{-1}	1571
ζ_0	Valve parameters	-	0.471
ω_0	Valve parameters	s^{-1}	910.1
ζ_1	Value parameters	-	0.4
ω_1	Value parameters	s^{-1}	1189.1
ζ_2	Valve parameters	-	0.502
ω_2	Valve parameters	s^{-1}	2880.1
A	Actuator piston area	m^2 (in^2.)	2.085×10^{-2} (32.32)
K_T	Total system compliance	N/m (lb/in.)	3.003×10^7 (171,500)
K_L	Load compliance	N/m (lb/in.)	3.387×10^7 (193,400)
J	Gimballed moment of inertia	N-m/s^2 (lb-in.-sec^2)	1.491×10^4 (1.32×10^5)
B	Nozzle viscous friction	N-m/s (in.-lb-sec)	6.56×10^4 (580,590)
K_b	Nozzle restraining torque gain	N-m/rad (in.-lb/rad)	3.39×10^6 (30×10^6)
d	Moment arm	m (in.)	1.819 (71.6)
H	Position feedback gain	N-m/m (in.-lb/in)	1.392 (0.313)
T_p	DPF time constant	s	0.125
K_p	DPF gain	N-m/pascal (in.-lb/psi)	2.524×10^{-9} (1.54×10^{-4})
B_o	Effective bulk modulus	N/m^2 (psi)	1.034×10^9 (150,000)
V_o	Effective end volume	m^3 ($in.^3$)	1.4×10^{-4} (8.5)
Q_v	Power valve flow	m^3/s (Cis)	$\pm 5.57 \times 10^{-3}$ (±340)

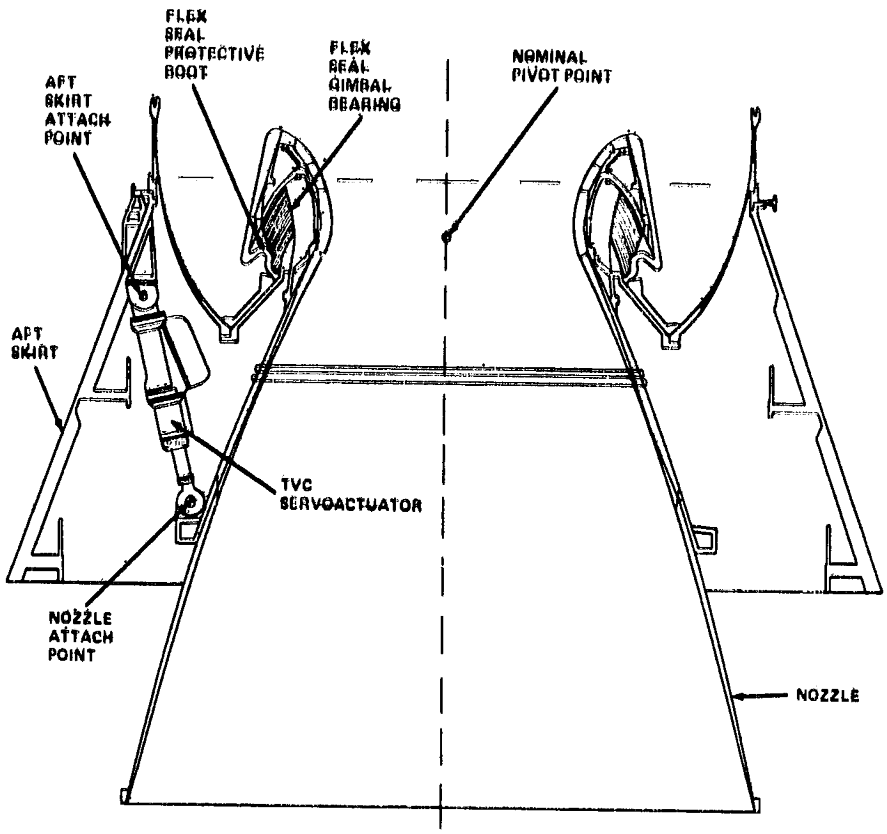


Figure 1.- SRB TVC servoactuator/nozzle installation.

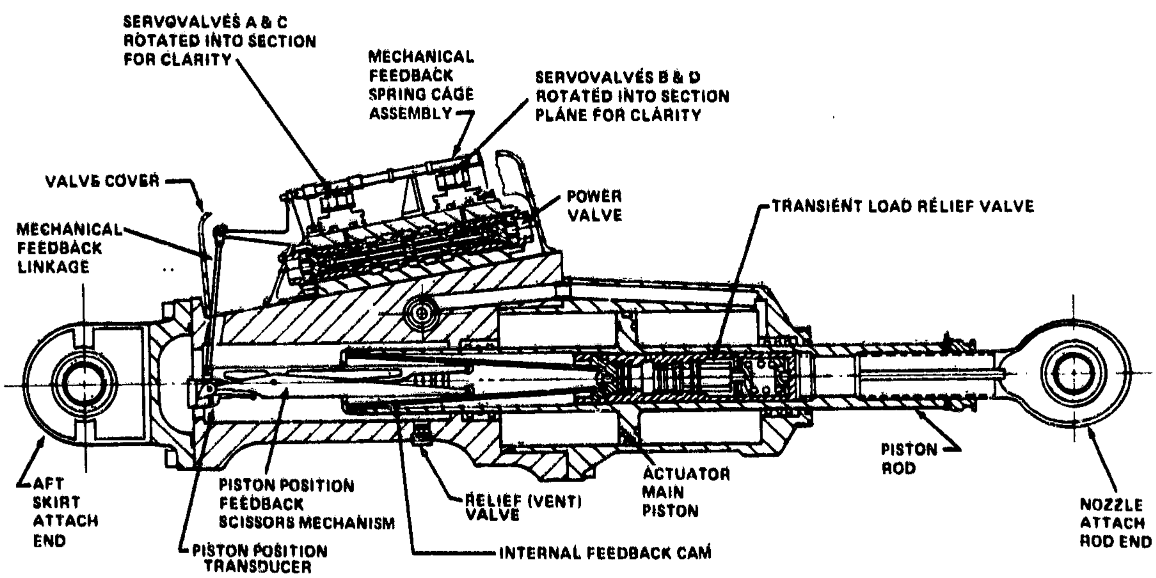


Figure 2.- SRB TVC servoactuator assembly.

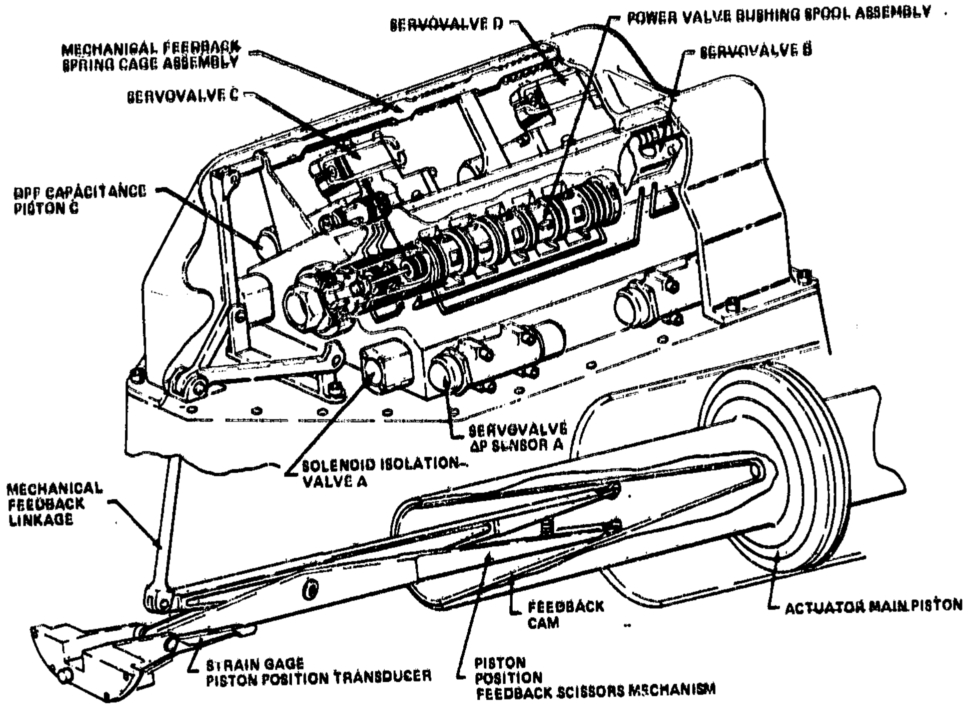


Figure 3.- SRB TVC servoactuator redundancy and feedback mechanisms.

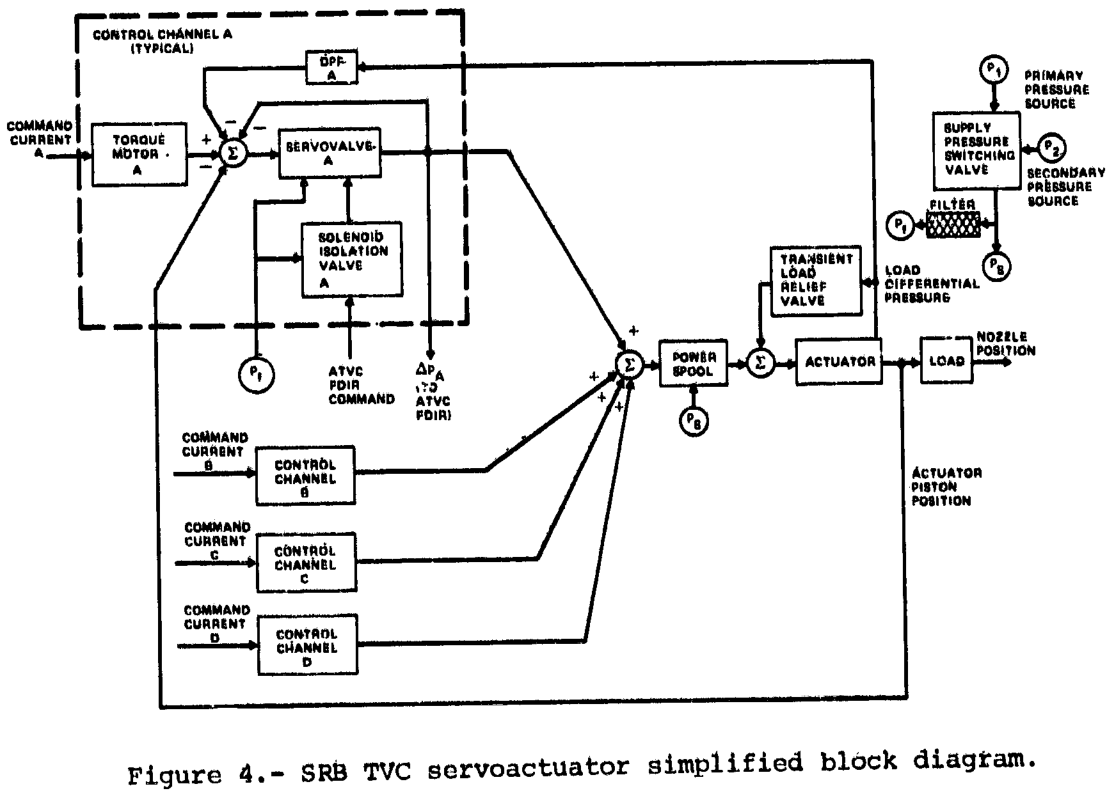
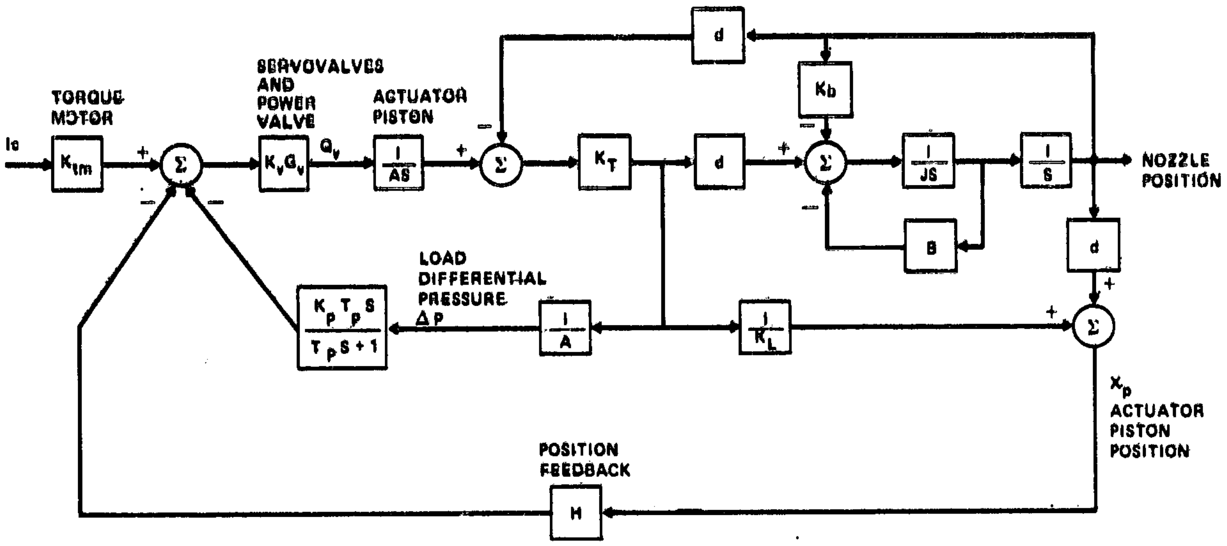


Figure 4.- SRB TVC servoactuator simplified block diagram.



$$G_V \equiv \frac{\{\omega,\zeta\}}{\{\omega_0,\zeta_0\}\{\omega_1,\zeta_1\}\{\omega_2,\zeta_2\}} \quad \text{WHERE} \quad \{\omega i,\zeta i\} \equiv \frac{s^2}{\omega i^2}\ \frac{2\zeta i}{\omega i}\ s + 1$$

Figure 5.- SRB TVC servoactuator linear mathematical block diagram.

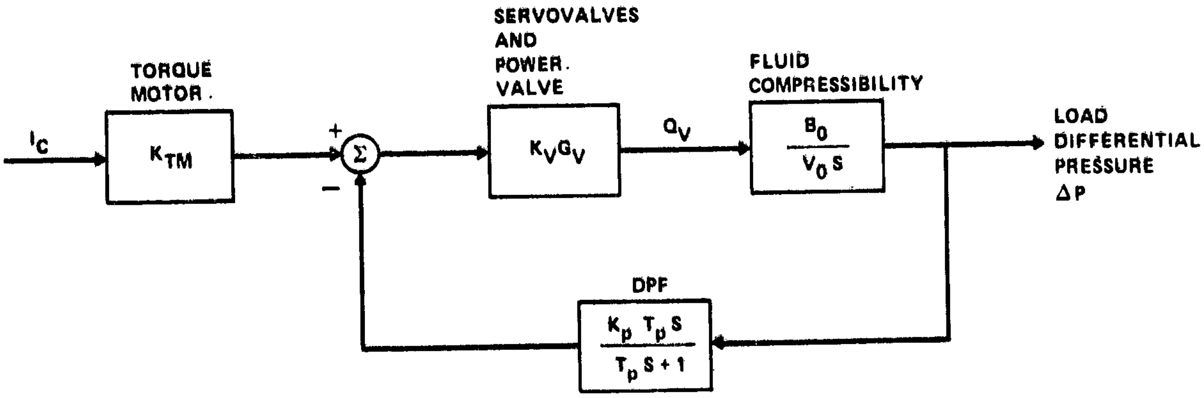


Figure 6.- End of stroke block diagram.