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REDUNDANCY EMPLOYING MAJORITY VOTING FOR A SATURN SERVOACTUATOR

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ABSTRACT

The servoactuator was developed to improve the reliability of the Saturn S-IVB thrust vector control system by insuring continued system operation if single point failures occur.

The selection of the majority voting technique is discussed. Its simplicity is cited along with the advantages of minimum weight, size, and power consumption, and compatibility with existing control electronics.

Operational features, design mechanization, and the analysis of test results are covered.

I. INTRODUCTION

A redundant, majority voting servoactuator has been developed for the attitude control system of the Saturn S-IVB stage, (Figure 1) the stage that provides the velocity increment to place the Apollo spacecraft into a trajectory to the moon. During powered flight, the Saturn vehicle attitude control is obtained by gimballing the main propulsion engines of each respective stage. The engines are gimballed by hydraulic actuation systems where servoactuators convert electrical command signals and hydraulic power into mechanical output which controls the vehicle thrust vector.

The Saturn booster lower stages (Figure 2) have a clustered, multiengine arrangement which provides a redundancy effect of not only the propulsion system but the thrust vectoring system as well. With this arrangement, satisfactory vehicle control can be maintained even with a failure of one engine (two at some times of flight). Other methods of enhancing the reliability of the single engine S-IVB stages were necessary. The complex mission requirements of this stage as a part of both the uprated Saturn and Saturn V launch vehicles dictate maximum reliability of all flight critical systems to insure mission success. The original, nonredundant S-IVB thrust vector control servoactuators are adequate in all operational performance aspects, but they are also a single point failure source. The need to eliminate single point failure possibilities in the thrust vectoring system resulted in a program to study various reliability improvement techniques. The already stringent quality control, inspection, and acceptance test procedures being employed were considered adequate. The only alternative was to consider a redundancy concept as the most likely reliability improvement approach.

Working under the constraints of a requirement that any redundant servoactuator would have to be mechanically and electrically interchangeable with the existing stage design, a preliminary feasibility study was begun. Various redundancy schemes were investigated and ultimately a majority voting servoactuator concept was chosen as the most feasible method to employ in reducing single point failures. Majority voting provided redundancy of the most critical servoactuator components without imposing weight and power consumption penalties on the system.

Two majority voting servoactuators were designed, fabricated, and subjected to extensive development testing. Actuator operating characteristics with various failures in the redundancy scheme were studied. The prototype actuators were integrated into a breadboard type stage hydraulic system, and further testing was performed. These initial tests resulted in several design changes to the servoactuator and to other hydraulic system components. These changes, discussed in detail in Section V, are being incorporated into the final flight configuration servoactuators.

II. FEASIBILITY STUDIES AND RELIABILITY CONSIDERATIONS

Feasibility studies were initiated to investigate the S-IVB thrust vectoring system requirements and the existing hardware design to determine if any system changes could be made to improve the overall system reliability and performance. A study program of hydraulic control system optimization techniques indicated that from an overall mission requirements standpoint, no changes in the S-IVB stage control system were required. Reliability considerations, however, made it highly desirable that single point failure sources be reduced.

Studies which considered overall stage requirements including the gimballing system indicated that some redundancy scheme could best satisfy the increased reliability requirements. The consideration of any redundancy concept involves considerable judgment in deciding between the actual gains made by going redundant against the losses from the added complexity of a redundant system. It is possible that in incorporating redundant components, failure detection devices, and switching devices, a decrease in reliability may result. Such an approach also leads to greater system weight and volume, and higher power consumption.

Since the S-IVB stage was already designed and flight qualified, it became necessary to consider parameters other than just the servoactuator. Any change in the actuator would have to be compatible with the existing control requirements, mechanical envelope dimensions, and control electronics. In order to keep the actuation system as simple as possible, the selected redundancy scheme should incorporate only those features necessary to eliminate possible single point failures. Dual hydraulic cylinders, pumps, accumulators, etc. would not increase the overall system reliability enough to compensate for the penalties imposed by greater system weight, power requirements, and cost.

The most critical component of any hydraulic servoactuator is the servovalve. Typical high performance servovalves require extremely close fitting spools and sleeves, and small orifices in the nozzle-flapper area. This makes the servovalve susceptible to contamination present in the operating fluid. In addition, the servovalve first stage has an electrical torque motor which is susceptible to open or short circuits. A system that would provide redundancy of these components would offer significant gains in system reliability without undue penalties of overall system complexity.

The original nonredundant S-IVB stage servoactuator has a position mechanical feedback arrangement whereby the need for an electrical feedback transducer is eliminated. Mechanical feedback was a significant advancement in hydraulic servoactuator technology because it eliminated the requirement for the electrical feedback transducer (an inherently failure prone component), the electrical power supply and associated wiring. Mechanical feedback also eliminated the need for a feedback amplifier and summing network in the control electronics system. Because of the improvements resulting from a mechanical feedback actuator, this design feature was retained in the redundant majority voting servoactuator configuration.

III. SERVOACTUATOR OPERATION AND DESIGN

The majority voting actuator uses three active torque motors driven by three separate amplifiers as shown in Figure 3. Voting is performed in the servoactuator only. The torque motors receive feedback from the valve spool and actuator piston. The output pressures of the three torque motor driven hydraulic amplifiers are summed at the end of the valve spool. The actuator piston behaves as the majority of the hydraulic amplifiers dictate. A majority decision exists when two hydraulic amplifiers are in agreement.

Each of the three hydraulic amplifier assemblies is as nearly alike as possible (Figure 4) to minimize the differences of each assembly in overall performance characteristics. Each torque motor has a balanced T-shaped armature-flapper mounted for pivotal motion on a flexure tube. A polarizing magnetic flux circuit is formed by upper and lower pole pieces supported by two permanent magnets. The motor armature is positioned in the magnetic flux circuit by the flexure tube. The flexure tube also acts as a seal between the electrical and hydraulic sections of the servovalve. For increased reliability, two torque motor coils are wired in parallel and positioned about the armature. Majority operation is possible because the mechanical feedback torque motor armatures are essentially free beams sensitive to force balance. A failure of any redundant element (loss of input, loss of feedback, hardover armature flapper, change of forward-loop gain, etc.) would result in a change in the output of that respective hydraulic amplifier and result in a change in spool end pressure and a change in valve spool position. The change in spool position is fed back to the torque motor flappers and in turn the two properly functioning hydraulic amplifiers equalize the spool end pressure generated by the faulty hydraulic amplifiers. By making the feedback and hydraulic amplifier gains sufficiently high, only a very small spool displacement is required to correct the effect of the failure.

Working within the specific design and performance criteria for the S-IVB stage, two prototype servoactuators (Figure 5) were fabricated and subjected to an extensive development test program. The test program included hardware evaluation at both a component and system level. A schematic of the redundant actuator is shown in Figure 6. Each servoactuator incorporated the following features, whose function is described.

A. Servovalve

The three torque motor mechanical feedback majority voting servovalve (Figure 6) is the key element of the selected redundancy concept. The three first stage assemblies consist of a dual coil d.c. torque motor, flexure mounted flapper, first stage nozzles, filter orifice assembly, feedback cantilever spring, load damping nozzles, and load damping nozzle filters. The first stage assemblies control fluid flow to the servovalve,the actuator, piston, and cylinder. The load damping or dynamic pressure feedback (DPF) is a method of damping resonant loads in order to minimize their effect on control system performance. This will be discussed in more detail later.

Of the three first stage hydraulic amplifiers, two used a balance armature feedback concept. For these two assemblies, springs are located at each end of the motor armature where the ends protrude through the air gaps. One spring is preadjusted to a fixed position between the armature and servovalve body. The other spring is fitted between the armature and the position feedback cam by a cantilever beam cam follower assembly. The third hydraulic amplifier differs in that it uses a flapper extension to hold the flapper and armature assembly in a balanced position. This design arrangement provides a simpler design of the mechanical feedback mechanism. The flapper of each first stage amplifier is rigidly attached to the midpoint of the armature. Each flapper extends through a flexure tube and passes between two first stage nozzles and two dynamic pressure feedback (DPF) nozzles to provide two sets of variable orifices between the nozzle tips and the flapper. The amplifier passages from the first stage nozzles terminate in common chambers at the ends of the second stage spool. The DPF nozzle passages are joined in a common bushing groove and are ported through the servovalve body by the DPF pistons. Each torque motor is linked mechanically to the feedback wire to provide mechanical feedback force during servovalve operation.

B. Second Stage Bushing and Spool Assembly

The second stage bushing and spool is physically larger than a normal electro-hydraulic servovalve. This large physical size provides comparatively high spool driving forces and there by reduces the valves susceptibility to contamination. The bushing itself is a flange mounted laminar fit into a dead end bore.

C. Position Feedback Mechanism

The actuator position feedback mechanism consists of a tapered cylindrical cam, three cam follower assemblies, and three position feedback springs. The cam is attached to the servoactuator piston rod and any movement of the rod is transmitted through a cantilever beam cam follower assembly and position feedback wire back to the first stage hydraulic amplifier assembly. Restraining devices are used to minimize deadband in the moving mechanical joints. This position feedback mechanism provides triple mechanical feedback redundancy of servoactuator piston position.

D. Dynamic Pressure Feedback (DPF) Network

The servovalve incorporates a DPF network using two nozzles and two springs centered capacitance pistons to provide damping of resonant loads. A pair of DPF nozzles are located directly above each pair of first stage nozzles and are hydraulically connected to opposite sides of the spring centered pistons. These pistons provide redundant operation and are located in the servoactuator body. The piston stubs are exposed to the cylinder pressure lines and are hydraulically coupled to the DPF nozzles to form the frequency sensitive DPF network.

The DPF arrangement is identical to that used on the original S-IVB stage servoactuator. By balancing of spool end pressures, nozzle flow, and torques throughout the servovalve, load damping is accomplished. Two spring centered capacitance pistons provide redundant operation in the event one DPF piston fails. The system is designed to dampen resonant loads well below the resonant frequency.

E. Electrical Position Transducer

The S-IVB stage majority voting actuator utilizes a strain gage position indicating transducer. The strain gages are mounted on the cam follower cantilever beams of the mechanical feedback mechanism. The free end of the cantilever beam, in moving, is stressed linearly and in turn is read as strain by the wire type strain gage. Signal conditioning equipment is provided to amplify the output up to a level compatible with existing measuring and telemetry equipment. The use of strain gages also eliminates using the potentiometer shaft as a feedback cam driving mechanism.

F. Accessory Components

In addition to the major subcomponents of the majority voting servoactuator, other items are provided to protect the assembly during operation or to facilitate actuator and stage checkout. Briefly, these items are:

1. Hydraulic Fluid Filter - A non-bypass type filter is located in the main actuator housing. It is rated at 505 cm^3/sec. (8 gpm) with a 17 N/cm^2 (25 psi) pressure drop. It is a 5 micron nominal, 15 micron absolute filter fabricated from a stainless steel wire cloth material. In addition, smaller wire mesh type filter orifices are provided for the servovalve first stage hydraulic amplifier assemblies and for the DPF network.

2. Prefiltration Bypass Valve - The prefiltration valve assembly allows fluid circulation through the actuator inlet port and directly through to the outlet to facilitate complete hydraulic system flushing.

3. Cylinder Bypass Valve - The cylinder bypass valve, when operated, allows manual movement of the piston rod by interconnecting both sides of the cylinder. The bypass valve also connects the cylinder lines to a transducer pad providing for a differential pressure transducer to monitor actuator cylinder line pressure.

4. Midstroke Lock - A simple, removable midstroke lock is provided to mechanically lock the actuator piston rod at a mechanical null position. The lock is designed to withstand full actuator force output without damage to either the lock or actuator.

5. Piston Position Indicator - A mechanical indicating device is included to aid in actuator calibration and checkout at field sites. The indicator is calibrated in degrees of engine displacement and when used with the vernier provided, is capable of 0.04 degree readout accuracy.

6. Bleed Ports, etc. - Various bleed ports, test ports, and oil sample ports are located to provide proper bleeding of the actuator and to allow easy installation of measuring transducer for checkout and flight telemetry.

IV. TEST PROGRAM DETAILS

A. Initial testing of the S-IVB majority voting actuator consisted of performing acceptance tests on each unit to the current S-IVB servoactuator acceptance test criteria. Additional tests were derived to properly test the majority voting portion of the servoactuators. Both prototype actuators were operated in a normal mode with all three torque motors active. All performance parameters were within the S-IVB actuator requirements with the exception of total internal leakage and mechanical stroke.

B. In order to meet the external envelope dimensions of the stage, the actuator stroke was reduced. The higher internal leakage resulted from having three hydraulic amplifier assemblies instead of one as in the nonredundant actuators. These differences did not represent design or performance discrepancies, only required deviations in order to fabricate the two prototype actuators. The effect of these deviations will be discussed in Section V.

C. Additional testing was performed to illustrate operation of the actuator with various failures induced in the actuator. Examples of these failures include plugged nozzles, plugged inlets, broken flexure sleeve, broken DPF springs, electrical opens, hardovers, and null offsets. This test program was designed to provide meaningful data on overall actual differences in actuator performance with various failure modes. The test series provided data on the servovalve, the actuator, and the two subassemblies together under both loaded and no-load conditions. A summation of the testing performed and the results on servoactuator performance is shown in Table 1.

To further demonstrate the overall performance and reliability gains of the majority vote actuator, high temperature and vibration environmental tests were performed. Again, no major design discrepancies were detected. The servovalve did show sensitivity to certain sinusoidal vibration inputs, however, this can be corrected by minor redesign of the servovalve first stage. This vibration sensitivity has been experienced on other actuator designs and has been corrected with very minor changes such as mechanical stops to limit flapper travel and variation of the support spring rates.

One known characteristic of the majority vote configuration was accurately demonstrated by the test program. The majority voting actuator operating in a normal mode essentially duplicates the nonredundant actuator in performance. Certain failure modes such as one open torque motor or one hard over torque motor will affect performance. With one motor hardover there will be approximately a two percent shift in the normal output position and the normal gain position will be reduced by one-third. Although overall actuator performance is affected by a torque motor failure, the system will compensate and will still provide adequate control forces to complete a flight mission. A torque motor failure in a nonredundant actuator would result in loss of control and compromise or abort the flight mission. Figures 7, 8, and 9 are typical performance curves obtained during this testing.

A failure in the load compensating, or DPF, network was also simulated. The worst case failure in the DPF section was simulated by removing a piston cap seal and "O"ring from one of the DPF pistons. A constant cylinder differential pressure of ±400 psid was maintained for this test. The break frequency of the DPF network increased from 0.8 Hz to 2.0 Hz which would alter the load damping characteristics of the actuator. However, a reduction of this magnitude is too samll to affect the control system. Other failures in the DPF network have no more influence over overall performance than a seal failure, therefore, it can be concluded that adequate DPF action is maintained with one failed DPF network. In a nonredundant configuration, complete load damping would have been lost.

Additional test data was obtained primarily to insure overall hydraulic system compatibility of the majority vote actuator configuration in the S-IVB stage. Various system pressures were tested, and attempts were made to optimize the overall system with a minimum number of actual hardware changes. System thermal characteristics were studied for compatibility with the existing stage design.

The conclusion reached from this extensive test program was that the majority vote actuator could be used on the S-IVB stage without extensive redesign of the flight stage or existing ground support equipment. In a normal no failure mode the actuator was adequate for the control forces required. In a single failure mode of any of the critical servovalve components, actuator performance was compromised slightly, but proper vehicle attitude control could be adequately maintained to successfully complete a flight mission.

V. REQUIRED DESIGN CHANGES

Concurrent with the majority voting servoactuator program, the complete stage hydraulic system was analyzed to determine those parameters which might be changed to gain in overall reliability and to interface with the majority voting actuator. Various minor design changes in lines, brackets, etc. were made to facilitate incorporation of the new actuator configuration. All changes made to the stage or to the actuator resulted in improvements of control system performance and/or reliability. Table 3 is a summary comparing the nonredundant actuator with the majority voting actuator design. The significant design changes are discussed briefly below.

A. Control Computer Changes

Since the control electronics was already triple redundant, the only changes to the control computer were to remove the comparator and switching circuits, which sensed an amplifier failure and provided switching to a good channel. This change was necessary since the selection, or voting, is now done by the actuator. This configuration effectively reduces the complexity of the S-IVB control electronics.

B. Stage Hydraulic System Pressure

The S-IVB stage hydraulic system was originally designed for an operating pressure of 2517 N/cm^2 (3650 psi) and the thrust vector control actuator sized accordingly. With experienced gained through flight testing it became evident that the in-flight loads, and therefore, the demands on the actuator were considerably less than the actuator design capability. In considering all aspects of the hydraulic system, it was decided that the system operating pressure could be reduced to 1827 N/cm^2 (2650 psi). The effect of this system pressure reduction would be a reduction in actuator internal leakage, a reduction in the overall stresses on the various hydraulic system components, and a lower probability of external leakage of joints and fittings. The actuator is still capable of providing the required control forces with this reduced system pressure and with no increase in actuator piston area.

C. Servoactuator Stroke

The prototype majority voting actuator was limited to a ±6 degree stroke, in comparison to a ±7 degree stroke in the original non-redundant actuator. This stroke reduction in the prototype actuator was necessary to conform to the stage mounting dimensions. In determining the final production configuration, repackaging of the three torque motors and some internal dimensional changes resulted in restoring the ±7 degree gimbal angle. The ±6 degree stroke would have been adequate for control purposes, but, the additional stroke is an added safety margin.

D. Internal Leakage

A three torque motor servovalve, with three hydraulic amplifiers has a higher leakage rate than an actuator with a single first stage. Changes were made in the on-board auxiliary hydraulic system to provide the increased flow capability needed for proper auxiliary hydraulic system operation.

VI. RELIABILITY IMPROVEMENT ASSESSMENT

Historically, most servovalve/servoactuator failures ($\approx$80 percent) result from a failure in the first stage of a servovalve because of its highly critical requirements. By incorporating redundancy in this first stage, it is evident that certain gains in reliability are achieved. Since there is, as yet, no in-flight data on the majority vote servo-actuator and no production quantities on which to base an analysis, the reliability assessments made thus far are based on experience with similar actuator and hydraulic system designs. The comparative numbers derived are based on the probability of known failures, their cause, and their end result. By considering not only the redesigned actuator but all other resultant system changes as well, the overall criticality improvement is approximately 20 to 1. The majority voting actuator itself is approximately 16 times less susceptible to functional failure than the original actuator design.

Table 3 is a comparison chart showing the improvement resulting from incorporation of the majority voting servoactuator. This table illustrates that the most significant improvement was in the servoactuator itself, and this combined with the other system changes significantly decreased system criticality. In no instance is a component less reliable than the original design.

VII. SUMMARY

In recognition of the need to increase the overall thrust vectoring system reliability of the S-IVB stage, a study program was initiated to select the optimum improvement method. Working within existing requirements of the operational flight stage and the need to implement any changes with minimum impact on existing flight schedules, a majority voting actuator concept was chosen. This concept provides redundancy in the critical first stage area of a servovalve, yet holds to a minimum size, weight, and power consumption. Majority voting requires no failure detection devices or switching arrangements, therefore, overall system complexity for a redundant system is minimized.

A detailed study program was conducted to demonstrate the feasibility of a majority vote actuator, the overall design suitability, and the resultant gains in system reliability. Results to date indicate that overall system reliability has been significantly improved by the majority vote actuator. Plans to incorporate this actuator concept on flight stages have been initiated, and the stage contractor now has production quantities of hardware on order.

Current planning for space exploration beyond the lunar landing program uses the S-IVB stage for a variety of missions. These missions will also reap the benefits of this control system improvement.

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