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ASTRIONICS LABORATORY

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SATURN-APOLLO PROGRAM - The primary objective of the Apollo project is a manned lunar landing (with safe return to earth) in this decade. Saturn will be the space vehicle that will launch the three Apollo astronauts on their historic voyage to the moon. The National Aeronautics and Space Administration (NASA) is developing three Saturn launch vehicle configurations (Fig. 1).

Saturn I, the first of the Apollo launch ehicles, is a two-stage vehicle that serves as a test bed for the evolving Saturn family. Its missions include the launching of unmanned Apollo command and service modules and the orbiting of large satellites. Saturn I can place a 10,000 kg (11 ton) payload in a low earth orbit.

Saturn IB, an improved Saturn I with a more powerful second stage (S-IVB), will be used for astronaut training with the Apollo spacecraft in earth orbit and for further vehicle and spacecraft development. The orbital capability of the Saturn IB is 15,000 kg (17 tons) in a 556 km (300 mile) orbit.

Saturn V is the vehicle with the rocket power to propel the Apollo spacecraft to the moon. The mission of the Saturn V launch vehicle then is to inject the Apollo spacecraft into a trajectory to the moon. The three-stage Saturn V will launch into earth orbit a 108,000 kg (120 ton) payload consisting of the third stage (S-IVB) and the Apollo spacecraft. After orbiting the earth, the S-IVB will be restarted, sending the 41,000 kg (45 ton) Apollo spacecraft to the moon for a manned lunar landing using the lunar orbital rendezvous technique. The Saturn V launch vehicle (Fig. 2) is being developed under the direction of NASA's Marshall Space Flight Center. It will be more than 110 m (360 ft) high, weigh 2.7 Mg (3,000 tons), and generate 33.5 x 10^6 N (7.5 million lb) of thrust. Saturn V will be composed of the S-IC, S-II, and S-IVB stages and the Instrument Unit.

The first stage of the Saturn V (Fig. 3), the S-IC, is under joint development by the Marshall Space Flight Center and the Boeing Company. This stage will use a cluster of five Rocketdyne F-1 engines. Each engine burns RP-1 and liquid oxygen to develop 6.7 x 10^6 N (1.5 million lb) for a total vehicle thrust of 33.5 x 10^6 N (7.5 million lb) during the

ABSTRACT

The actuation system for the Saturn V S-IC stage is described and compared to the Saturn I system. The use of mechanical feedback actuators that result in a significant increase in system reliability and the damping of load resonance is discussed. The unprecedented component sizes and system requirements are cited.

The components of the S-IC stage engine gimbal actuation system are similar to and a direct evolution of those used in the Saturn I vehicle system.

The major system components are the fluid supply, a filter manifold, lines, and the actuators. The system is the essence of simplicity. As a fluid power source, fuel is taken directly in a single path from the high pressure side of the engine turbopump. Flow from the actuators is returned to the fuel system rather than to a reservoir.

Because of the small number of components and the design concepts used, the reliability of this system is high.

approximate 150 second programed firing time. The S-IC stage is 42 m (138 ft) long and 10 m (33 ft) in diameter.

The second stage of the Saturn V, the S-II, is being developed by the Space and Information Systems Division of the North American Aviation Company. It has a cluster of five Rocketdyne J-2 engines, each burning liquid hydrogen and liquid oxygen to develop 0.9×10^6 N (200,000 lb) thrust for a total stage thrust of 4.5×10^6 N (1 million lb) during an approximate 600 second burning time. The S-II stage is 25 m (82 ft) long and 10 m (33 ft) in diameter.

The third stage of the Saturn V, the S-IVB, is being developed by Douglas Aircraft Company. This stage uses a single J-2 engine to develop 0.9×10^6 N (200,000 lb) thrust just long enough to inject itself and the spacecraft into orbit. Then, after all systems have been checked out, the S-IVB will be restarted to accelerate the vehicle to escape velocity and guide it into the desired lunar trajectory. The stage is 18 m (60 ft) long and 6.6 m (22 ft) in diameter.

The Instrument Unit (IU) is the Saturn's "brain." It contains the guidance, control, and sequencing equipment for the launch vehicle as well as the measuring, telemetry, and radio frequency equipment. The IU, approximately 1 m (3 ft) long and 6.6 m (22 ft) in diameter, mounts atop the S-IVB stage under the Apollo spacecraft. International Business Machines (IBM) is the prime contractor for development and fabrication of the IU.

SATURN V GUIDANCE AND FLIGHT CONTROL SYSTEM - The guidance and flight control system must orient the vehicle to the desired attitude as a function of time along a computed optimum trajectory. The system issues commands to the attitude control devices of each stage during its active operating time. As mentioned earlier, the vehicle guidance and control system is contained in the IU. The ST-124 inertial platform, the digital guidance computer and data adapter, the control computer, the rate gyros and control signal processor, the control accelerometers, and a switch selector, all operating as an integrated system and located in the IU (Fig. 4) are shared by all three stages of the Saturn V. Rate gyros and body-fixed control accelerometers will also be located on some of the stages. This integrated central system results in the advantages of minimum weight and maximum reliability.

In the vehicle control system (Fig. 5), attitude reference signals are obtained from the ST-124 platform, a three-gimbal device with unlimited freedom about two axes. Here vehicle attitude and acceleration are measured by the inertial platform and its accelerometer, converted to digital form by the data adapter, and processed by the digital guidance computer. The guidance computer determines control command signals and converts them to analog inputs for the control computer. The control computer shapes, mixes, and amplifies these signals, damping signals from the rate gyros,

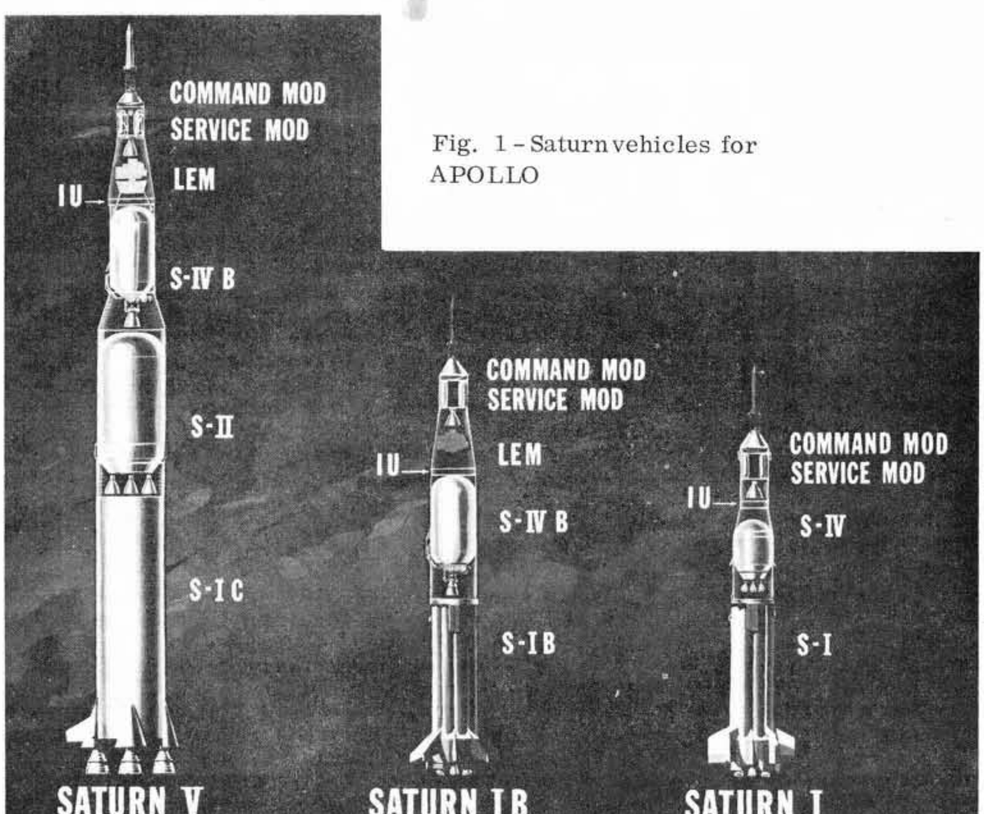

Fig. 1 - Saturn vehicles for APOLLO

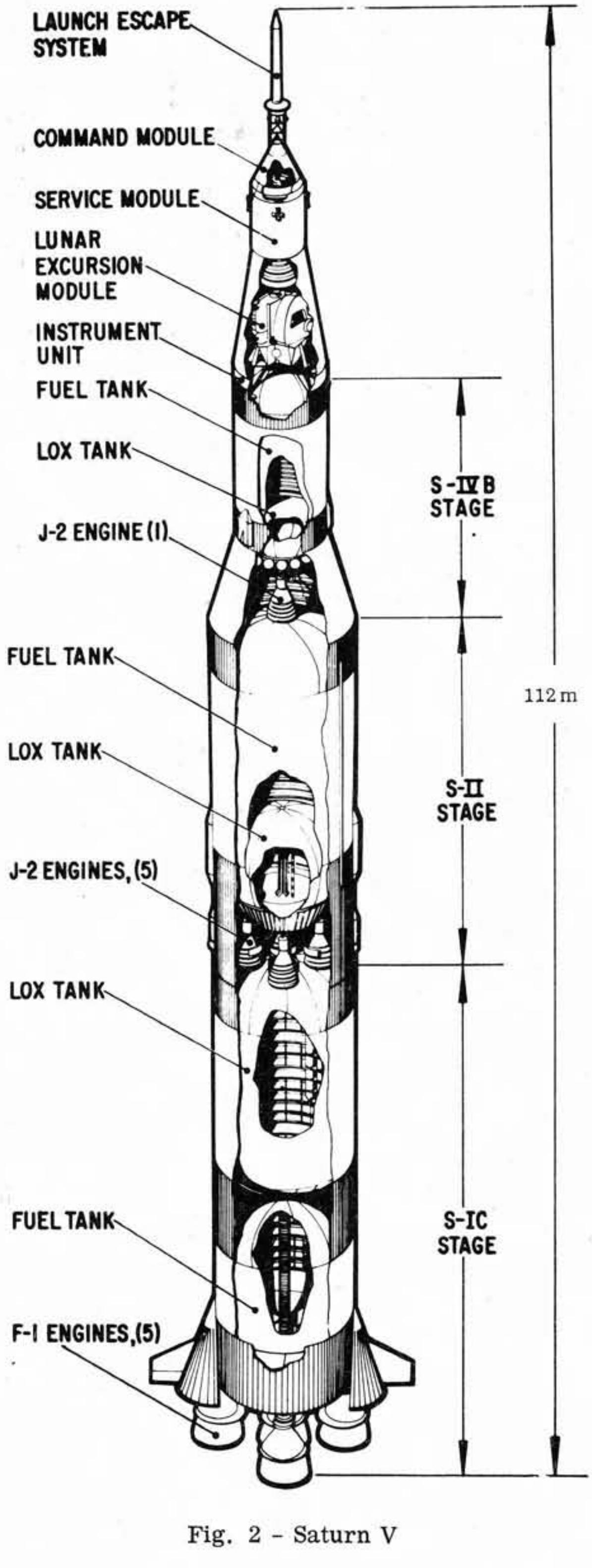

Fig. 2 - Saturn V

and control accelerometer information for angle-of-attack control. The vehicle guidance and control force requirements are obtained by making use of an engine gimbal system as attitude control device. During powered ght, the summed command signals direct the actuators of the respective stages to correct vehicle flight to conform to the desired trajectory. The gimbal control servoactuators

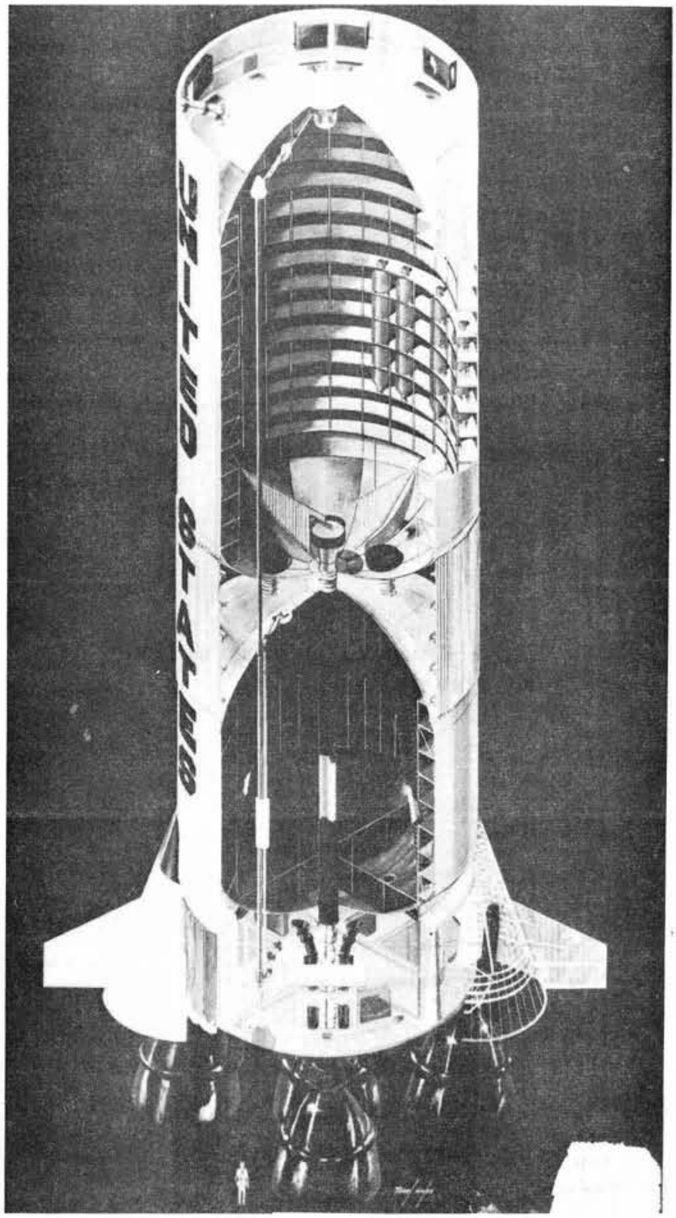

Fig. 3 - Saturn V S-IC stage

are all hydraulically powered. The arrangement of the servoactuators for the multiengine first and second stages and the single engine third stage is also shown in Figure 5. The redundant feature of the clustered engines in the first and second stages, with an independent hydraulic system for each gimbaled engine, is a major step toward achieving maximum reliability. With a failure of one engine or of the control of one engine, satisfactory vehicle control can still be maintained by the proper sizing of the engine swivel range. Having an independent hydraulic system on each gimbaled engine in contrast to a central system offers the following advantages:

1. The system can be qualified by single-engine tests.

2. One engine system can be checked out at a time.

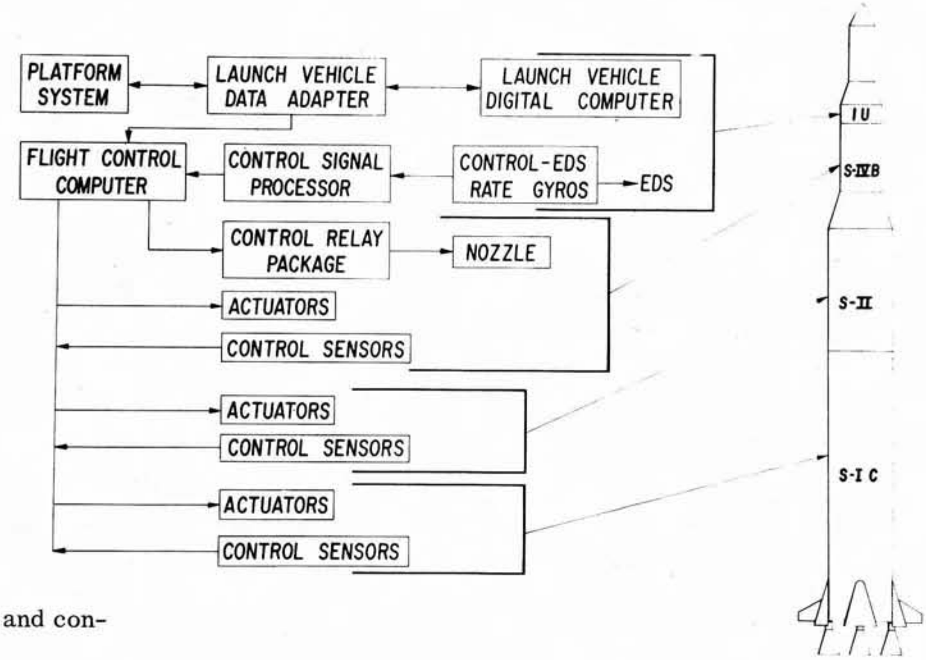

Fig. 4 - Saturn V guidance and control system

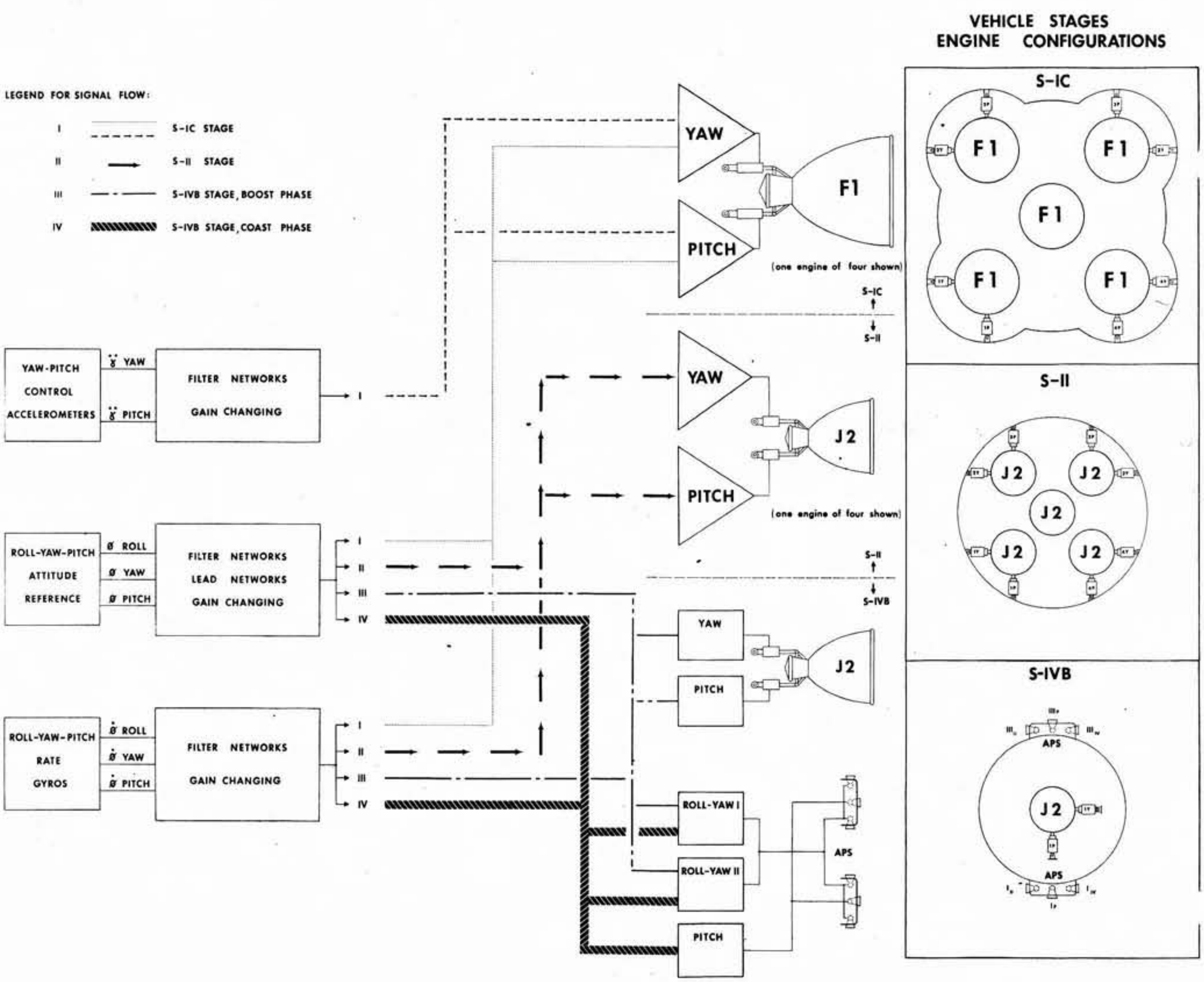

Fig. 5 - Saturn V control system block diagram

3. Post-test performance comparisons are easily accomplished and problem areas quickly located.

To further improve the reliability of the Saturn V vehicle, redundancy of the hydraulic gimbal system was considered. However, it was realized that the premature and careless use of redundant design could result in little or no improvement in reliability but could actually degrade reliability.

Saturn V will have redundant implementation in the control system electronics (control computer). Triple modular redundancy is used for those portions of the control computer that have long mission use time (i.e., more than 10 minutes). This specifically includes the electronics for the spatial attitude control system during S-IVB coast and for the gimbal control of the single engine of the S-IVB stage that is restarted after coast.

Limited redundant design of the hydraulic servovalve compatible with the S-IVB electronics is being investigated. Further implementation of redundant design is not feasible. The increased weight penalty of further implementation would result in too great a loss of payload. The complexity of the redundant system would also present additional checkout complexities. All the adverse side effects and trade-off penalties were considered. The redundant feature of the clustered engine stages and the other considerations mentioned did not justify the development of a more redundant hydraulic gimbal system.

To further improve the reliability of the servosystem, the Marshall Space Flight Center directed its efforts to optimizing the reliability of critical components. It was decided early in the development of Saturn V to utilize mechanical feedback in the thrust vector control servoactuators on all three stages (Fig. 6). The successful development and use of a mechanical feedback system by Martin-Denver on Titan prompted an evaluation of electrical versus mechanical feedback actuators. Studies showed that a significant improvement in reliability could be obtained with the use of mechanical feedback actuators. Their use in other programs verifies their dependability. With the mechanical feedback actuator, the number of electrical wires could be reduced (a change from 6 to 2) by eliminating the feedback transducer wiring. Since the actuator and the control computer are remote from each other in the vehicle, the number of solder joints and connections at each interface could be reduced approximately 67 percent. The feedback amplifiers and the summing network in the control computer could be eliminated. The potentiometer power supplies could also be eliminated. The potentiometer itself, which is sensitive to

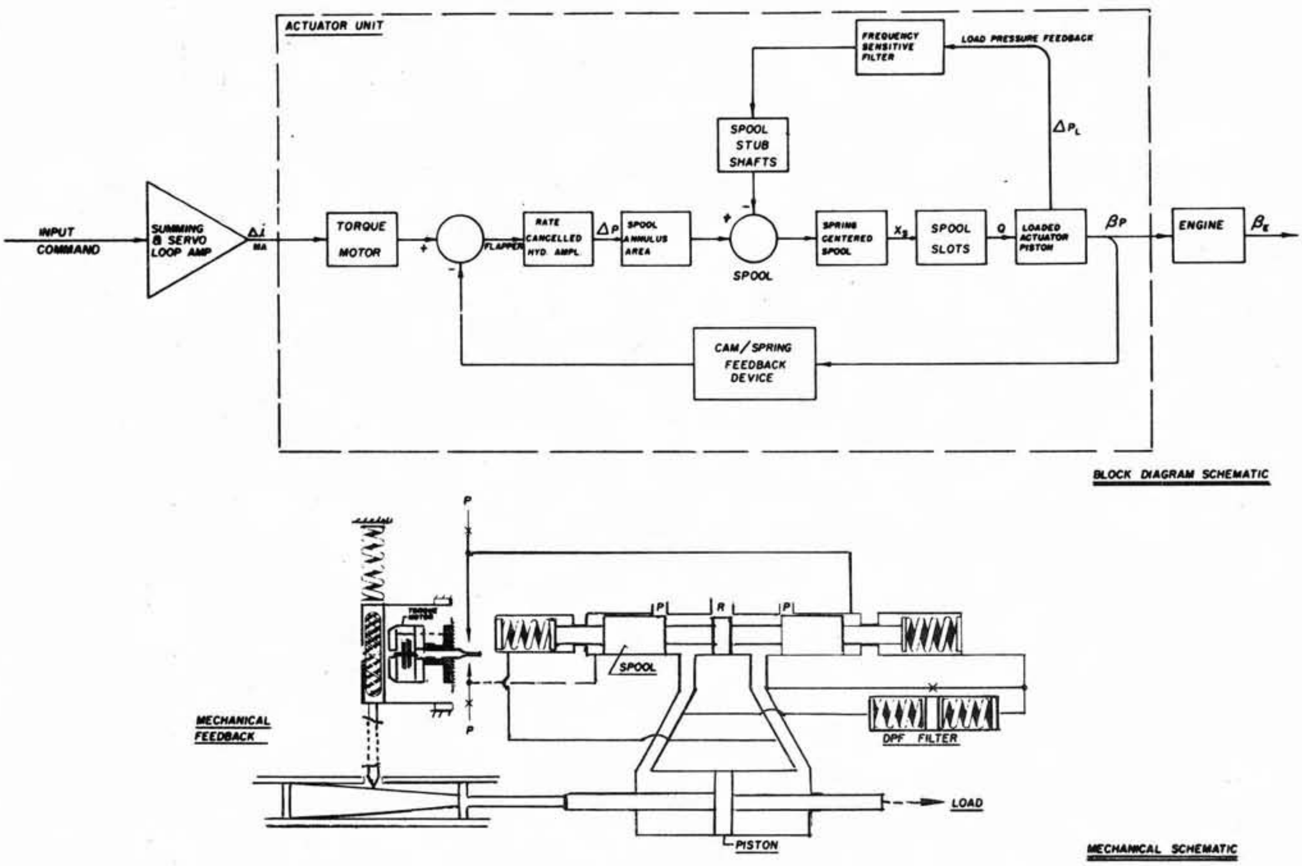

Fig. 6 - Mechanical feedback servoactuator

shock, vibration, wear, and moisture, could be eliminated. The mechanical feedback design also provides the fail-safe feature of actuator centering in the event of an electrical failure.

Based on the above considerations and feasibility studies made by the stage contractors and actuator manufacturers, a decision was made to use mechanical feedback on all servoactuators on the Saturn V boost stages.

Another critical servosystem component that was improved in Saturn V was the servovalve. The electrical input power of the mechanical feedback actuator used in the S-IC, S-II, and S-IVB stages has been increased to:

1. Improve contamination tolerance
2. Generate higher spool driving forces
3. Improve the spool driving force gradient
4. Improve response performance (null shift and threshold).

To improve coil reliability, the coil wire size was increased from AWG 44 to AWG 38.

S-IC STAGE FLIGHT CONTROL SYSTEM

THRUST VECTOR CONTROL CONSIDERATIONS - The purpose of the S-IC stage flight control system is to orient the engine thrust vectors relative to the vehicle during powered flight and to stabilize and direct the vehicle along a computed optimum trajectory. The S-IC stage is unique in its tremendous size. The F-1 engines are almost an order of magnitude larger in thrust and weight than Saturn I booster H-1 engines. In choosing the thrust vector control system for the S-IC stage of the Saturn V vehicle, a feasibility study was made to analyze possible methods and determine the optimum system for the F-1 engine. Control by varying the thrust vector direction or magnitude of the main propulsion system thrust or of auxiliary nozzles was considered. Thrust vector direction can be controlled in a number of ways. One category is thrust vector positioning by gimbaling with hydraulic actuators either the nozzle, the chamber, or the entire engine, or by canting and rotating the nozzle. A second category is thrust vector deflection by "jet vanes," "jetavators," or paddles. A third category is "secondary injection." Extensive testing was conducted to ascertain the feasibility of this method of thrust vector control. The final selection of the gimbal engine concept was based on a comparison of system hardware weights and performance penalties and the consideration of simplicity, reliability, and the development problems of the various concepts. Although the entire mass of the thrust chamber and turbopump is gimbaled in the gimbaled engine concept, this method did not present any major development problems. The concept is well known, is simple, and utilizes a small number of components that are reliable and that are not pushing the state of the art. The system can be checked out repeatedly, actually gimbaling the entire engine either with a burning engine or with a cold engine.

HYDRAULIC SYSTEM CONSIDERATIONS - During powered flight, the hydraulic power supply for the gimbal systems of the Saturn I stages is provided in a closed recirculating system (Fig. 7) by a hydraulic pump driven from an accessory pad on the engine turbine and by an accumulator reservoir that provides for peak loads and dampens surges and pump ripple. An auxiliary pump driven by an electric motor is provided for checkout during prelaunch operations. A variation of this conventional scheme is being used in the Saturn V second and third stages with hydraulic system pressures of approximately 2500 N/cm^2 (3500 psi).

When the F-1 engine was designed, no provision was made for an accessory power pad. As a fluid power source, MIL-F 25576B fuel (RP-1) at 1250 N/cm^2 (1800 psi) is taken directly from the engine turbopump, used in the thrust vector control gimbal actuators as hydraulic fluid, and returned to the engine system. This relatively low pressure open, or single pass, system is very simple; because of the small number of components, system reliability should be high. There are, however, disadvantages and therefore a closed high pressure system was also considered. In this system, the fuel from the engine turbine would flow to a turbine driving a centrifugal pump pumping MIL-H-5606 hydraulic fluid at 2800 N/cm^2 (4000 psi). With this system, t servovalve flow and the actuator piston area would be less than half that of the low pressure system. MIL-H-5606 hydraulic fluid has been

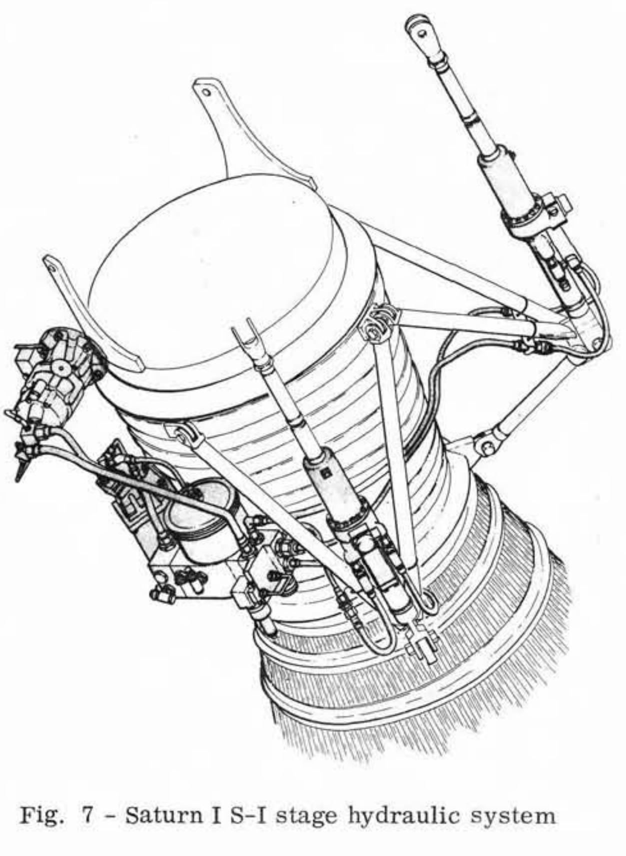

Fig. 7 - Saturn I S-I stage hydraulic system

used extensively whereas the use of the RP-1 in an actuating system, while not a new idea, has been used very little. In addition, the response of the lower pressure system would be limited. RP-1 also exhibited disadvantages in a hydraulic system. Its lubricity is poor, it is less viscous, it is more corrosive, and contamination control is poor. The fluid is also a safety hazard because it has a minimum flash point of 43°C (110°F). Fire codes require extensive provisions for its indoor handling and use.

However, because high pressure flexible hose and fittings were not commercially available and a turbine pumping system required a long development time, which would delay stage delivery schedules, and because of the simplicity of the low pressure direct fuel system, a decision was made to develop the low pressure direct fuel system. To overcome the disadvantages of using RP-1 as a hydraulic fluid, design considerations were made. Three-stage servovalves were necessary to obtain satisfactory response. To provide a high tolerance for contamination, servovalve fluid passages were designed to withstand higher levels of contamination than those usually found in conventional systems. To avoid the indoor use of the flammable RP-1, MIL-H-5606 is used at the manufacturers' plant and in the laboratory test. For some vehicle system tests, a substitute fluid compatible with RP-1 and meeting National Fire Code safety requirements was found. A commercially available jet fuel, RJ-1 (MIL-F-255580), very similar to RP-1 with respect to physical properties, can be procured with a minimum flash point of 93°C (200°F). As a seal, Viton A (MIL-R-25897) was found to be compatible with RP-1, RJ-1, and MIL-H-5606 fluids individually or in any sequence and/or combination. The problem of corrosivity was minimized by selection of materials and rigid control of the water content in either RP-1 or RJ-1.

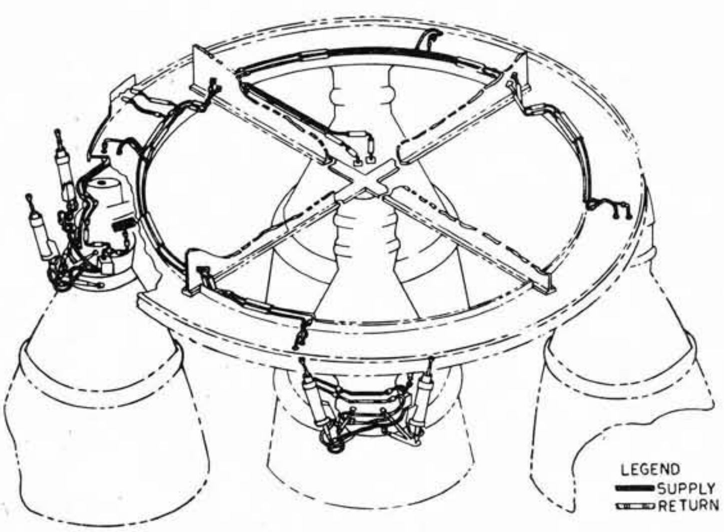

Fig. 8 - S-IC hydraulic system

The fact that the direct fuel system is a single-pass low-pressure system also presents a filtering problem. The low pressure requires greater flow for a specific gimbal rate. A single pass filter must do the same cleaning job in one pass that the filter in a closed recirculating system does in many passes.

S-IC HYDRAULIC SYSTEM

The S-IC hydraulic system (Fig. 8) consists of a fluid power supply system, the engine gimbal actuators of the thrust vector control system, and the F-1 engine controls. In the S-IC, the thrust vectoring system controls the direction of the stage thrust by gimbaling each outboard engine about its pivot point so that the thrust of the engine is positioned for proper vehicle control. An inflight hydraulic supply of 950,000 cm^3/s (15,000 gpm) is available from each F-1 engine turbopump. The relatively small flow required for the thrust vector control and engine control systems results in a well regulated pressure source. Because of virtually unlimited supply, the accumulator-reservoir, usually required in conventional systems, is not necessary.

FLUID POWER SUPPLY - The fluid power supply system consists of the engine turbopump pressure sources, a ground power source, a stage mounted distribution system, and the engine mounted distribution systems. For ground checkout and calibration of the thrust vector control system, the distribution system delivers fluid under pressure from the ground supply to the filter manifold assemblies of the four outboard engines where the fluid is filtered, delivered to the servoactuators, and returned to the distribution system. For operation of the thrust vector control system during static engine test firing and during flight, the distribution system distributes fluid from the F-1 engine turbopumps. The distribution system also supplies fluid power from the ground source to all five engines for control of the engine starting sequence and from the engine turbopumps for engine valve control during flight and engine static test firing. Each engine has an independent hydraulic system (Fig. 9) obtaining and returning fluid from either the ground supply or the high pressure feed duct of that engine.

During ground checkout of the thrust vector control system, the F-1 engines are not operating; therefore, fluid pressure is supplied from a ground power source. The ground power supplies RJ-1 fuel (MIL-F-25558B) to the stage umbilical connection. From the stage umbilical connection, the fluid passes through the stage-mounted system to the five F-1 engines. In the four outboard engines, the fluid passes through the distribution system to the engine starting control valves and to the ground supply port of the engine-mounted filter manifold. The filter manifold has two inlet ports with check valves and two outlet ports to the servoactuators.

One of the inlet ports connects to the ground supply line, and the other connects to the flight supply line from the engine turbopump. When the engine is not running, the check valve in the flight supply port closes and the ground-supplied fluid passes through the filter element to the servoactuators. Return fluid from the actuators passes through the return lines and the electrically operated two-way ground checkout valve to the stage mounted system and then returns to the reservoir of the ground supply source. During ground checkout, the two-way hydraulic checkout valve that is designed to route return fluid, either to the ground source or to the low-pressure side of the engine turbopump, is in the ground return position.

To start the F-1 engines, RJ-1 fuel (MIL-F-25558B) from the ground power source is delivered to the five engines as described in the preceding paragraph. The fluid, after the engine start signal is given, passes through the four-way valve on the engine, powers the opening of the main fuel and LOX valves, and controls the position of the hypergol injection and ignition monitoring valve. During the engine start cycle, the two-way ground checkout valve in the return system is in the engine return position so the return fluid from this operation is all directed to the low-pressure side of the engine turbopump. As the engine turbopump builds up to its rated pressure, the check valve in the ground supply port of the filter manifold closes and all fluid entering the filter manifold and the servoactuators is RP-1 (MIL-R-25576B). A similar condition exists at the inlets to the four-way valve in the engine start system, and the entire stage-mounted fluid power system is out of the functional loop.

During flight operation and static test firing, the servoactuators are supplied with RP-1 fuel from the engine turbopumps. The fuel passes through the filter manifold to the servoactuators and back to the low pressure side of the engine turbopump.

FILTER MANIFOLD ASSEMBLY - Hydraulic Research and Manufacturing Company has designed and developed the filter manifold assembly for the S-IC gimbal system (Fig. 10). The assembly filters fluids fed to the thrust vector control actuators in the gimbal system. It is a lightweight filtering unit containing a filter, two check valves, and a manual sampling valve. The filter element is a full flow, non-bypass, tubular unit. The filter element material is 18-8 stainless steel wire cloth pleated with a layer of stainless steel wire backup mesh on the inside and outside to provide maximum strength. The backup mesh prevents the pleats from squeezing together under pressure and prevents damage or tearing of the filter material. Fluid enters

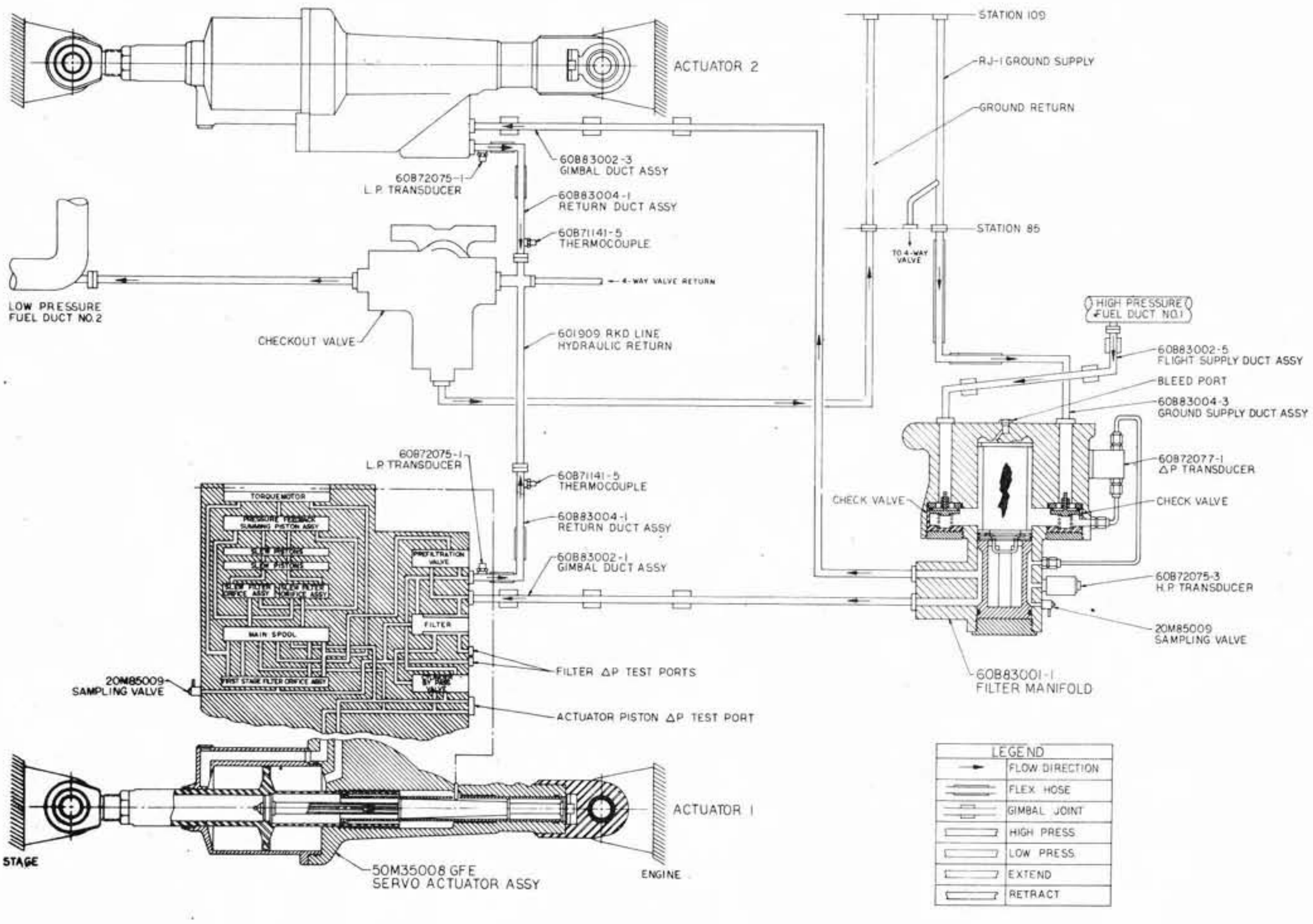

Fig. 9 - S-IC hydraulic system schematic

the housing surrounding the element, flows from the outside to the inside of the filter element, deposits any contaminants, and exits through the manifold to the servoactuator. The filter element is easily removed and can be cleaned for re-use.

The assembly provides one ground inlet, one flight inlet, and two outlet flanged ports or primary service connections. In addition, two MC 240-2 pressure sensing ports are furnished on a pad for externally-mounted transducers measuring inlet and outlet differential pressures. A larger MC 240-8 pressure sensing port to measure pressure downstream of the filter is incorporated for direct mounting of a pressure transducer. The check valves provided in both inlet ports prevent reverse flow through the ground or flight supply when the opposite mode of operation is being used. The sampling valve is installed on the side of the manifold assembly to sample the fluids on the downstream side of the filter element. A bleed plug is located on top of the filter case to vent air from the body upon initial startup. An exploded view of the assembly is shown in Figure 11. Table 1 lists the filter manifold assembly design requirements.

INTERCONNECTING DUCTING - The ducting interconnecting the filter manifold and actuators with the supply and return includes flexible sections to allow for thermal expansion, misalignment, and actuator movement. The gimbal duct assembly bellows are 718 stainless steel; the ducts are 321 stainless steel. Temperature and pressure transducers are located in each of the actuator return lines. All lines of the stage mounted system are connected with bolted flanges welded to the lines and sealed with gasket-type face seals incorporating an elastomeric sealing element bonded into a cavity in the metal seal plates. The lines incorporate flexible elements to accommodate thrust-ring dynamic deflections and manufacturing tolerances.

INSULATION - The engine-mounted fluid power system that is exposed below the heat shield operates in a thermal environment of more than 1038°C (1900° F) during flight. Therefore, thermal insulation is provided for all components of this system. The insulation limits the surface temperature of all the engine-mounted fluid power system components and their attachment bracketry to 190°C (275° F). There are openings in the insulation for flight and static test instrumentation. These openings have patch-type insulating covers that are secured in place for missions not requiring instrumentation. The insulation is readily removable and is replaced for maintenance and inspection of the system components.

ACTUATOR DESIGN CONSIDERATION

ACCEPTED DESIGN - The Marshall Space Flight Center hydraulic gimbal actuation design criteria have evolved from many actuator programs both at NASA and throughout the industry. The S-IC stage servoactuator assembly, as with the actuators on all stages of Saturn, consists of a linear, double-acting hydraulic actuator controlled by an electrohydraulic flow control servovalve. To improve system performance, facilitate checkout, and guarantee a high vehicle

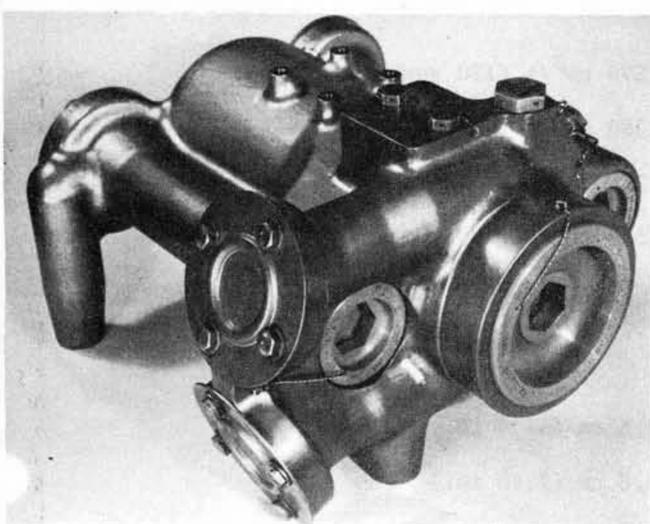

Fig. 10 - Filter manifold assembly

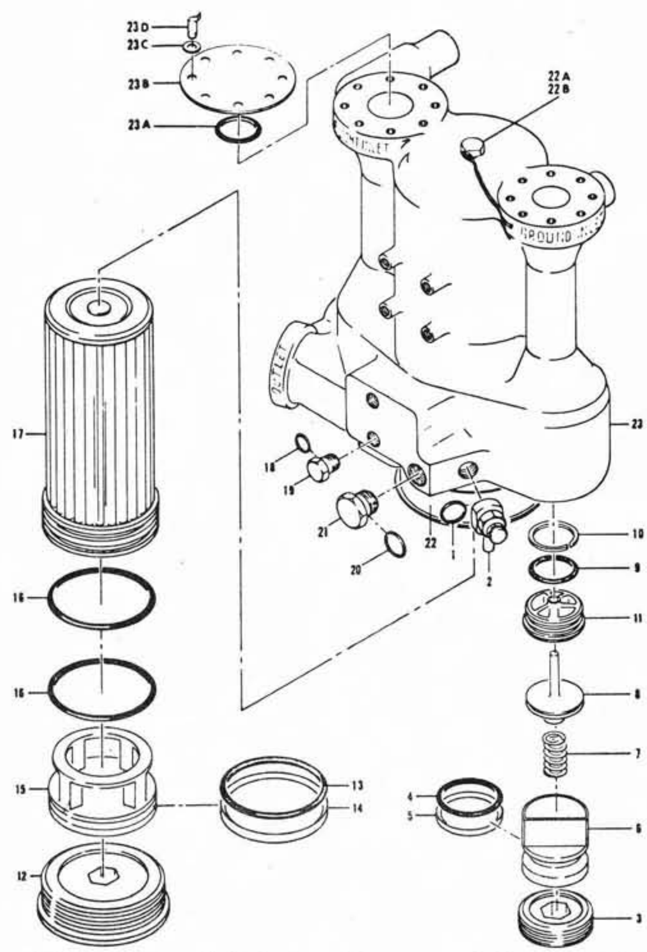

Fig. 11 - Exploded view filter manifold assembly

reliability, several accessory components are provided. There is a manually operated prefiltration bypass valve to permit system filtration without passing the fluid through the actuator servovalve or filter. To assure maximum reliability, a full flow filter and sampling valve are incorporated in each actuator. There is also a piston bypass valve so that the piston may be moved manually for calibration and test when the system is not pressurized. A removable midstroke lock capable of full actuation force is used for engine holding during ground handling operations and for providing a midstroke position reference. A vernier scale calibrated in engine deflection angle is also provided. On Saturn I, it was found necessary to provide some method of load damping to stabilize the spring mass system consisting of the engine-inertia/actuator/stage compliance combination. In addition, on all Saturn V actuators, position is fed back mechanically to the servovalve torque motor.

DESIGN REQUIREMENTS - The characteristics of each part of the thrust vector control system influence the servoactuator design and must be considered. The relatively low pressure fuel from the engine turbopumps and the high actuation forces dictated a large servoactuator piston area and a new three stage servovalve design. The engine-stage structure characteristics in Table II must be considered in the design of the servoactuator. The engine will see maximum gimbal torques at the time the vehicle encounters maximum dynamic pressure. A dynamic load design value of 323,000 N (72,000 lb) has been calculated as a requirement from these gimbal torques. The actuator system is required to gimbal the outboard engines through an angle of ±0.09 rad (±5.166°) at a maximum rate of 0.0875 rad/s (5°/s) at this design load.

Table I - Filter Manifold Design Requirements

Filtration Rating	15 micron absolute 5 micron nominal
Fluid Media	RP-1 Fuel Per Specification MIL-R-25576; RJ-1 Fuel Per Specification MIL-F-25558; or Hydraulic Fluid Per Specification MIL-H-5606
Fluid Medium Temperature Range	+1.7° to +37.8°C (+35° to +100°F)
Pressure Data:	
Operating	1,515 N/cm^2 (2200 psi)
Proof	3,030 N/cm^2 (4400 psi)
Burst	6,060 N/cm^2 (8800 psi)
Flow Rates:	
Flight Inlet Port	7,570 cm^3/s (120 gpm)
Ground Inlet Port	6,250 cm^3/s (99 gpm)
Outlet Ports	5,240 cm^3/s (83 gpm)
Through Filter Element	7,570 cm^3/s (180 gpm)
Ambient Temperature Range	-17.8° to +135°C (0° to +275°F)
Overall Dimensions:	
Length	28.8 cm (11.4 in.)
Width	18.8 cm (7.40 in.)
Height	30.7 cm (12.10 in.)
Weight (Approx)	13.15 kg (29 lb)

Table II - Engine-Stage Structure Characteristics

Engine Inertia	38,500 kg-m^2	(28,500 slug ft^2)
Actuator Bell Crank Radius	1.62 m	(63.76 in.)
Reflected Load at Actuator	14,800 kg	(84.13 lb-sec^2 in.)
Combined Structural Spring Rate	43,700 kN/m	(250,000 lb/in.)
Load Resonance	8 Hz	(8 cps)
Load Damping	0.05	
Bias Load (All loads except inertia and viscous friction)	0-36,300 kg	(0-80,000 lb)

Table III - S-IC Actuator Requirements

Input Power, Maximum	.25 watts	
Input Current	±50 mA	
Stroke	29.2 cm	(11.48 in.)
Area	367 cm^2	(57 in.^{2})
Force Output	510,000 N at 1380 N/cm^2	(114,000 lb max. at 2000 psi)
Torque Output	810,000 Nm	(600,000 ft lb)
Actuator Mass	145 kg	(320 lb dry)
Actuator Length	1.52 m	(60 in.at null)
No Load Velocity	380 cm/s	(15 in./sec max.)
Operating Pressure	1380 N/cm^2	(2000 psi)
Proof Pressure	2270 N/cm^2	(3300 psi)
Burst Pressure	4130 N/cm^2	(6000 psi)
Engine Actuation Response with 5% Input	<25° @ 1 Hz (cps) <3 db max. amplitude ratio	
Threshold	<1/2%	
Hysteresis	<3%	
Life	>200,000 cycles	
Acceleration	12 g	
Vibration	0.44 g^2/Hz (cps) 20-2000 Hz(cps)	
Shock	100 g	
Acoustic Noise	202 db	
Fluid Operating Temperature Range	4°C to 49°C	(40° to 120°F)
Ambient Temperature	<93°C	(200°F)

Table II - Engine-Stage Structure Characteristics

Engine Inertia	38,500 kg-m^2	(28,500 slug ft^2)
Actuator Bell Crank Radius	1.62 m	(63.76 in.)
Reflected Load at Actuator	14,800 kg	(84.13 lb-sec^2 in.)
Combined Structural Spring Rate	43,700 kN/m	(250,000 lb/in.)
Load Resonance	8 Hz	(8 cps)
Load Damping	0.05	
Bias Load (All loads except inertia and viscous friction)	0-36,300 kg	(0-80,000 lb)

Table III - S-IC Actuator Requirements

Input Power, Maximum	.25 watts	
Input Current	±50 mA	
Stroke	29.2 cm	(11.48 in.)
Area	367 ~~[illegible]~~ cm^2	(57 $in.^2$)
Force Output	5 ~~4~~10,000 N at 1380 N/cm^2	(114,000 lb max. at 2000 psi)
Torque Output	810,000 Nm	(600,000 ft lb)
Actuator Mass	145 kg	(320 lb dry)
Actuator Length	1.52 m	(60 in at null)
No Load Velocity	380 cm/s	(15 in./sec max.)
Operating Pressure	1380 N/cm^2	(2000 psi)
Proof Pressure	2270 N/cm^2	(3300 psi)
Burst Pressure	4130 N/cm^2	(6000 psi)
Engine Actuation Response with 5% Input	<25° @ 1 Hz (cps) <3 db max. amplitude ratio	
Threshold	<1/2%	
Hysteresis	<3%	
Life	>200,000 cycles	
Acceleration	12 g	
Vibration	0.44 g^2/Hz (cps)	20-2000 Hz (cps)
Shock	100 g	
Acoustic Noise	202 db	
Fluid Operating Temperature Range	4°C to 49°C	(40° to 120°F)
Ambient Temperature	<93°C	(200°F)

The system was designed to meet the additional design requirements in Table III.

MECHANICAL FEEDBACK VS ELECTRICAL FEEDBACK
Increased attention to overall Saturn V vehicle reliability required closer looks at individual subsystem reliability. Other than employing redundant techniques in the engine-positioning hydraulic systems, the use of mechanical feedback servoactuators appeared to offer the most in reliability improvements and was compatible with vehicle schedules. A slight deterioration in system linearity, resolution, and flexibility for gain changes that might be required for changing engine load dynamics, and a need for improved linearity and tolerance control in the servoamplifier were the main disadvantages. The ability of the servoactuator to null with loss of signal path continuity from the servoamplifier, the elimination of the electrical feedback element and its associated wiring, and the integration of the servoactuator functions by enclosing the engine load controlling loop in one assembly were the predominant advantages. At the same time mechanical feedback yields an integrated package that lends itself to the use of more advanced fluid amplifier techniques that that can improve the reliability of an already extremely reliable device.

In a basic count, 326 wires amounting to 22 to 45 kg (50 to 100 lb) and 744 solder or crimped joints were eliminated from the control loop in the total Saturn V boost vehicle by changing to mechanical feedback servoactuators.

LOAD DAMPING - Large-engine vehicles have inherent performance problems created by engine inertia and compliance of gimbal and actuator support structure. The resulting resonance presents severe stability problems for the position servoactuator.

The inertia of the F-1 engine, coupled with the compliance of the stage structure, creates a severe resonance at about 8 Hz (cps). The spring mass load on the actuator results in about 20 db of gain at its resonant frequency. The servovalve and actuator system must be designed to obtain approximately 20 db of attenuation at this frequency. In the actuator, a hydromechanical filter must be designed to sense load pressure and regulate valve flow to limit power input. The limiting is greatest at load resonance.

To meet the overall system requirements, the positional open loop gain is adjusted to obtain the proper phase lag at the control and guidance loop frequencies. The pressure open loop gain is adjusted to give the correct amount of damping to the engine position.

MOOG SERVOACTUATOR DESIGN

Moog Servocontrols, Inc., is supplying electrohydraulic servoactuators that will be used for positioning the Saturn V S-IC stage F-1 engine. As is true with most other components for this enormous vehicle, the servoactuators are larger than similar hardware produced previously. A comparison of the Saturn I, S-I stage, and the Saturn V, S-IC stage, actuators are shown in Figure 12. There is more to these servoactuators than sheer size, however. Several innovations were developed, and these are perhaps more significant than the size of the actuator itself.

PHYSICAL DESCRIPTION AND CONSTRUCTION - The actuator housing is basically a two-piece design with a forged aluminum tailstock/body member and a forged steel cylinder/front bearing member (Fig. 13). The two housing members are jointed at a piloted flange by 17 high-strength bolts. The double-ended piston is of single-piece construction from a steel forging. Both rod-end and tailstock bearings are roll staked to bearing holders that are threaded to the piston body.

The body is made from high-strength aluminum forging alloy 7079-T62, which was selected for the hardness and physical properties obtainable in large section forgings. Because this alloy has an unfavorable history of stress corrosion cracking, processing sequence precautions were taken to minimize the possibility of this problem. Alloy 7075-T73, a high-strength aluminum alloy with less susceptibility to stress corrosion, was also considered for the body forging; however, because of extreme section thickness, which caused limited heat-treat penetration, this material could not be properly heat treated to the T73 condition.

The body houses most of the accessory components, including the cylinder bypass valve and supply line flushing valve (both manually operated), a replaceable full-flow filter element (20 micron nominal, 45 micron absolute), two capacitance pistons for the pressure feedback, various test and bleed ports, and the electrical connector for the servovalve. The servovalve mounts to a vertical face of the actuatory body and is covered with an O-ring-sealed cast aluminum cover. Removal of an access plate in the cover permits final adjustment of the centered position of the servoactuator.

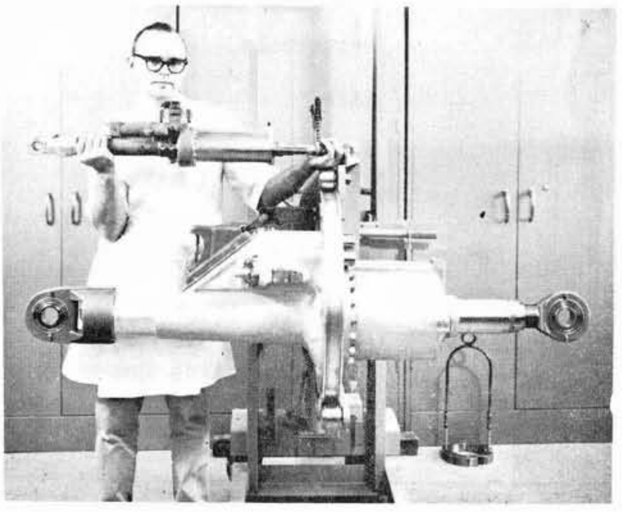

Fig. 12 - S-I and S-IC actuators

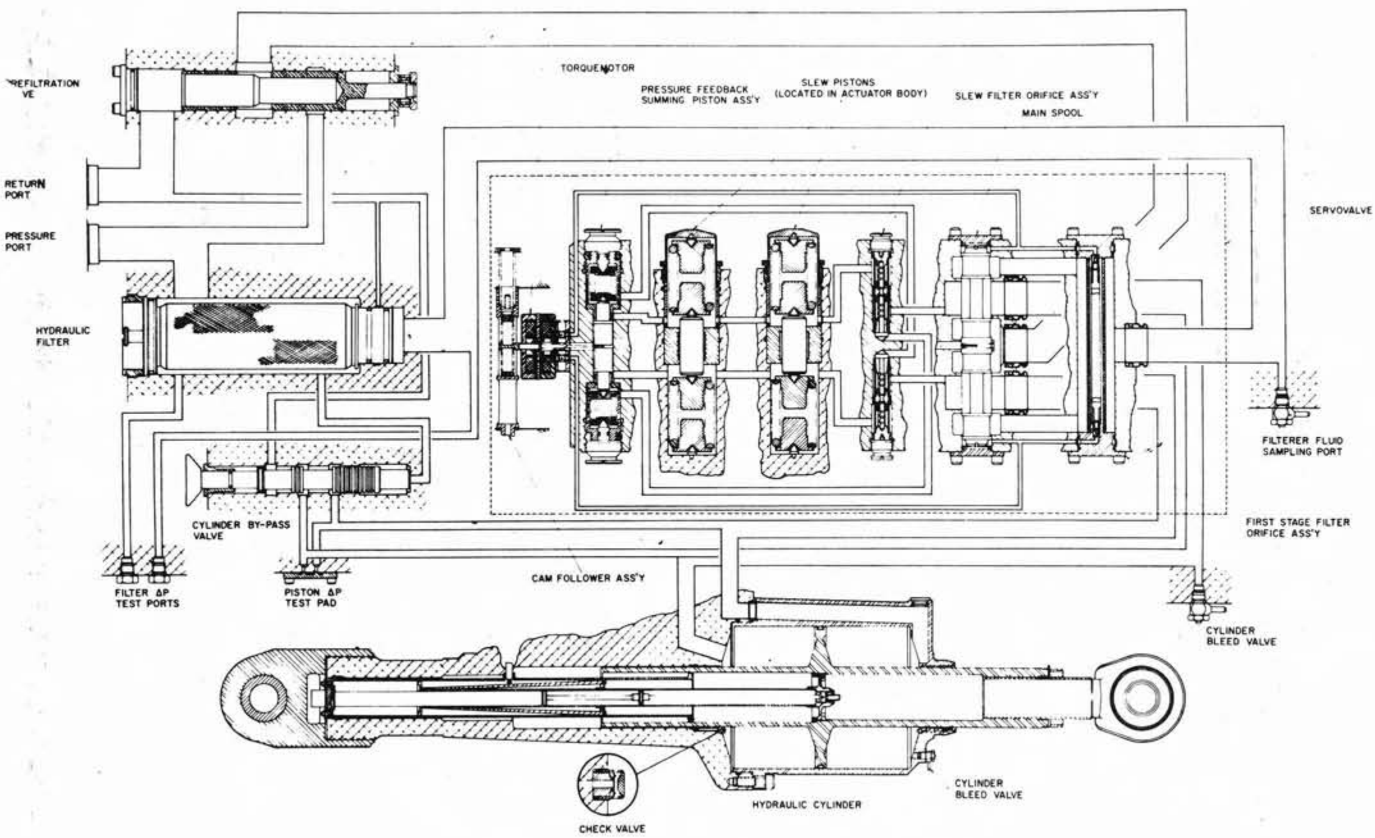

Fig. 13 - S-IC schematic moog servocontrols

The steel cylinder is forged AISI 4240, hardened and tempered to RC 34-38. The cylinder walls are approximately 0.8 cm (5/16 in.) thick and are designed for 4120 N/cm^2 (6000 psi) burst pressure. The cylinder surface is hard chrome plated and ground to a 0.2 micron finish. Both rod and piston head dynamic seals are capped with Teflon rings, which reduce running friction and improve seal life. The piston-head Teflon cap seal is radially grooved to prevent pressure loading of the seal, which could allow flow bypass. The piston bearing areas at the two rod ends are lined with Rulon A to prevent metal-to-metal contact caused by high loads normal to the piston rod.

The piston is first hammer forged from 4340 steel. The center is then hollowed out by gun drilling, and then the piston is hardened to RC 30-34. The piston rods are hard chrome plated and ground to a 0.4 micron finish. The finish dimensions of the piston rods are 10.15 cm (3.998 in.) O.D. with a nominal 0.7925 cm (0.312 in.) wall thickness. The relatively large rod diameter was necessary to gain column strength to withstand a 22,300 N (5000 lb) side load on the actuator while operating.

The mechanical feedback cam, cam guide, and telemetry potentiometer (supplied by Markite) are located inside the aft piston rod. The cam and cam guide are machined from 440C stainless steel. Figure 14 shows the servovalve and cam guide assembly. The cam is fitted to the guide (which has a nominal 6.03 cm (2.374 in.) diameter) with a diametral clearance of 7.5 to 25 micron (0.0003 to 0.0010 in.). The finish on the inside of the cam guide is held to 0.2 micron throughout a length of nearly 63.5 cm (25 in.). To avoid unnecessary weight, the housing has a thin wall, which tends to warp during heat treatment. The final machining therefore removes almost 370 microns of stock. The best technique found for this final machining is honing, which unfortunately is a time consuming task.

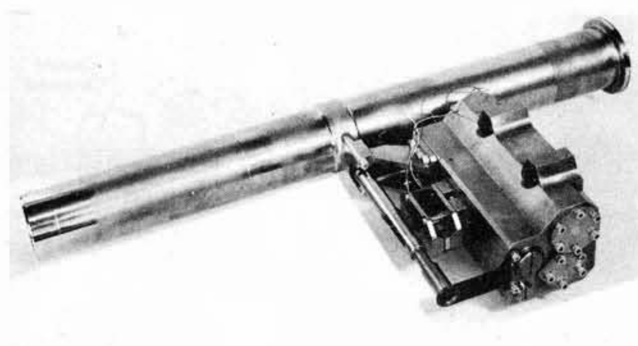

Fig. 14 - Servovalve cam guide assembly

The servovalve body is forged 17-4 PH stainless steel. The bores for both the third-stage valve spool and for the pressure-feedback summing piston are nitrided for hardness. No bushing or sleeve is used around the third-stage spool. Rather, full-circumference annular cuts are provided in the servovalve body to form the various flow-metering edges. This type of construction eliminates the need for a number of O-ring seals as normally used with a sleeved-spool construction.

Both the rod-end and tail-bearing holders are machined from steel forgings. The bearings have a 5.07873 cm (1.9995 in.) bore.

SERVOVALVE DESIGN - For the servovalve, the first approach was to use a large two-stage mechanical feedback design. This approach was extremely simple and permitted the use of very large nozzles and orifices; however, the dynamic response of a large two-stage servovalve is limited. This response was marginal for the S-IC actuation requirements; consequently, a more conventional three-stage design was used (Fig. 15).

The 2.54 cm (1 in.) diameter third-stage spool has a ±.279 cm (±0.110 in.) stroke to give a rated flow capacity of approximately 8400 cm^3/s at 1380 N/cm^2 (200 gpm at 2000 psi).

The servovalve second stage is a conventional spring-centered, four-way spool. All flow to the first and second stages is filtered by a separate 40-micron stainless-steel-mesh filter element.

A torque motor with a 0.23 Nm (2 in-lb) output drives the servovalve first stage. A high rated current input (50 mA) permits the use of a relatively small number of turns of large (AWG 38) coil wire in the torque motor. The torque motor has two identical coils wired in parallel for additional coil reliability.

The three-stage servovalve is a mechanical feedback design. The feedback is provided by a cantilever spring fixed to the end of the flapper. The movable end of the cantilever engages a slot in the center of the third-stage valve spool.

MECHANICAL FEEDBACK DESIGN - Even though experience existed in the design of mechanical feedback mechanisms, and mechanical feedback servoactuators for engine positioning had been supplied on other booster programs, it was necessary to evaluate various designs for this application since scaling-up of smaller designs might not have been satisfactory. The most attractive scheme was a conically tapered cam located on the centerline of the actuator. This approach had been used extensively by Moog in previous servoactuators and, upon reexamination for the S-IC, was again chosen considering overall performance and manufacturing techniques. The design offers freedom from backlash, thermal shifts, and vibration effects. It is not subject to distortion from actuator loading or piston rotation and cannot be damaged by field handling.

Several problems were involved in adapting this technique to the S-IC servoactuator. These include: (1) the additional center-to-center length needed to accommodate the cam housing caused a packaging problem, and (2) th

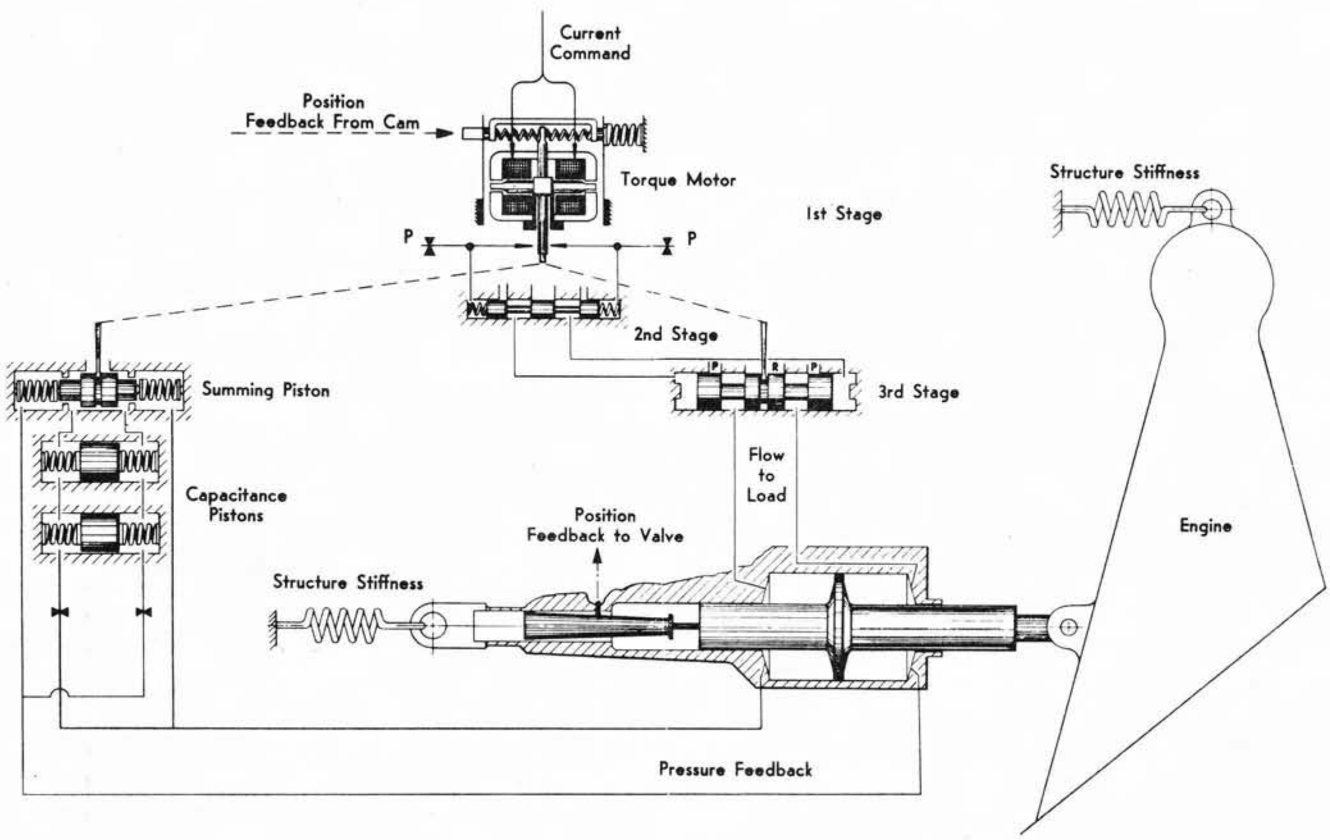

Fig. 15 - S-IC servoactuator - pictorial schematic

machining and finishing of the cam and housing imposed a manufacturing problem. The packaging problem was solved by a concentric telescoping design, in which the telemetry potentiometer fits inside the cam, and the cam housing fits inside the inner position rod. This design allows a 29.2 cm (11-1/2 in.) total stroke with a nominal 1.52 m (5 ft) overall servoactuator length.

LOAD DAMPING DESIGN - Much experience also existed in the techniques for damping engine resonance; however, several alternate schemes were investigated. This investigation led to an improvement over previously used designs. The S-IC servoactuator requirements dictated that load damping by proportional pressure feedback was not desired, since this caused the servoactuator to move in the presence of a static load. This position error could be large with high static loads, such as caused by a slight offset of the engine thrust vector. Under the same static load, the vehicle structure deflects to cause an additional position error. The magnitude of the structural deflection error is about half that of a servoactuator with proportional pressure feedback.

Moog uses a dynamic pressure feedback (DPF) servovalve configuration to eliminate the static position error caused by actuator pressure feedback. However, even with this configuration, the structural deflection errors are still present. The new techniques developed for the Saturn S-IC eliminate errors caused by pressure feedback and those caused by deflection of the vehicle structure. Functionally, this is achieved by combining conventional pressure feedback (which has a negative polarity) used for damping with positive pressure feedback used for static load error washout (SLEW). The positive pressure feedback includes a frequency sensitive network to attenuate the feedback at higher frequencies. This allows the negative pressure feedback to predominate at the load resonant frequency as needed for damping. Both pressure feedbacks are applied to a pressure summing piston which moves according to the difference in feedback pressures. The hydraulic components for SLEW are arranged so that clogging of the orifices or stoppage of a capacitance piston will not result in loss of engine control.

Physically, SLEW pressure feedback is applied in the servovalve by a novel twin feedback wire arrangement. One cantilever feedback wire relates third-stage valve spool motion to torque on the flapper in the normal manner. A second feedback wire is held by the same flapper, and its movable end is positioned by the pressure summing piston.

ASSEMBLY AND TEST - Handling and test equipment for the large servoactuator imposed special problems because of the actuator size and the high output flow required. The solutions to these problems have worked well, as shown in Figure 16, which is a photograph of a servoactuator during a phase of manufacture.

Assembly of the servoactuator starts with the main body, which is held by a mounting ring. The ring can be picked up at three points by a roller arrangement carried by a portable hydraulic lift. The rollers permit rotation of the ring and servoactuator about their centerline. The lift has provisions for rotating the entire ring, so the servoactuator can be turned 360 degrees throughout the horizontal and vertial positions.

Both bench testing and loaded actuator velocity tests are performed with the actuator mounted vertically in test stands. These minimize the floor area needed for testing the servoactuator. Actuator dynamic response testing is performed on a pendulum load simulator shown in Figure 17. The weight of the pendulum is 4080 kg (9000 lb), which is approximately half that of the F-1 engine. However, the moment arm for attachment of the actuator has been adjusted to provide an exact representation of engine inertia. The pendulum configuration matches the actual engine c.g. location, so only the pendulous frequency of the scaled-down load differs from that of the actual engine. This change is small, being 0.26 Hz (cps) for the actual engine and 0.36 Hz (cps) for the pendulum simulator. (The resonant frequency of the simulated inertia and springs is exactly equal to that of the engine.)

For convenience and safety, all production testing is performed with MIL-H-5606 hydraulic oil rather than RP-1 (kerosene). Most servoactuator setup and testing requires only normal test branch flow capacity. The servovalves are tested for static flow performance on a high flow test stand. This test stand combines the outputs of four pumps for a maximum capacity of

Fig. 16 - Servoactuator in mounting ring

4000 m^3/s at 2070 N/cm^2 (95 gpm at 3000 psi). Accumulators are attached at the servoactuator supply line to provide transient full flow capacity. This is needed during velocity and dynamic response testing. The data during high flow testing are recorded for later reduction.

HYDRAULIC RESEARCH SERVOACTUATOR DESIGN

The Hydraulic Research and Manufacturing Company has designed and developed a linear-output-displacement hydraulically-powered servo-actuator for positioning the gimbaled F-1 engines on the S-IC stage of Saturn V to control vehicle attitude. The unit is a closed loop servo position system.

Integrated into the servo housing as shown in Figure 18 are additional accessory valving and hardware which facilitate testing, installation, flushing, rigging, ground servicing, handling, and maintaining system cleanliness.

PHYSICAL DESCRIPTION AND CONSTRUCTION- The linear actuator has been designed with reliability as the primary consideration. The piston rod is fabricated from 4340 chrome-molybdenum steel which is hard chrome plated. This material was developed to retain the excellent strength and heat treatability of the more common 4130 while improving its machineability and stability after heat treatment because of its more homogenous grain structure

Based on extensive tests on actuator designs with regard to minium leakage, minimum contamination generation and wear, and maximum

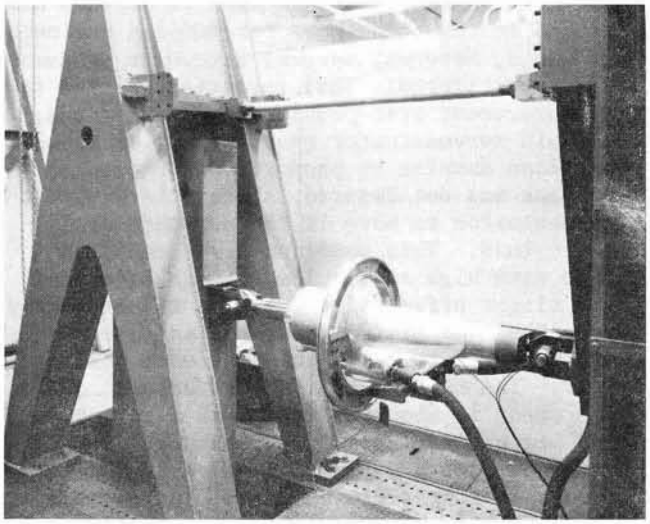

Fig. 17 - Pendulum load simulator

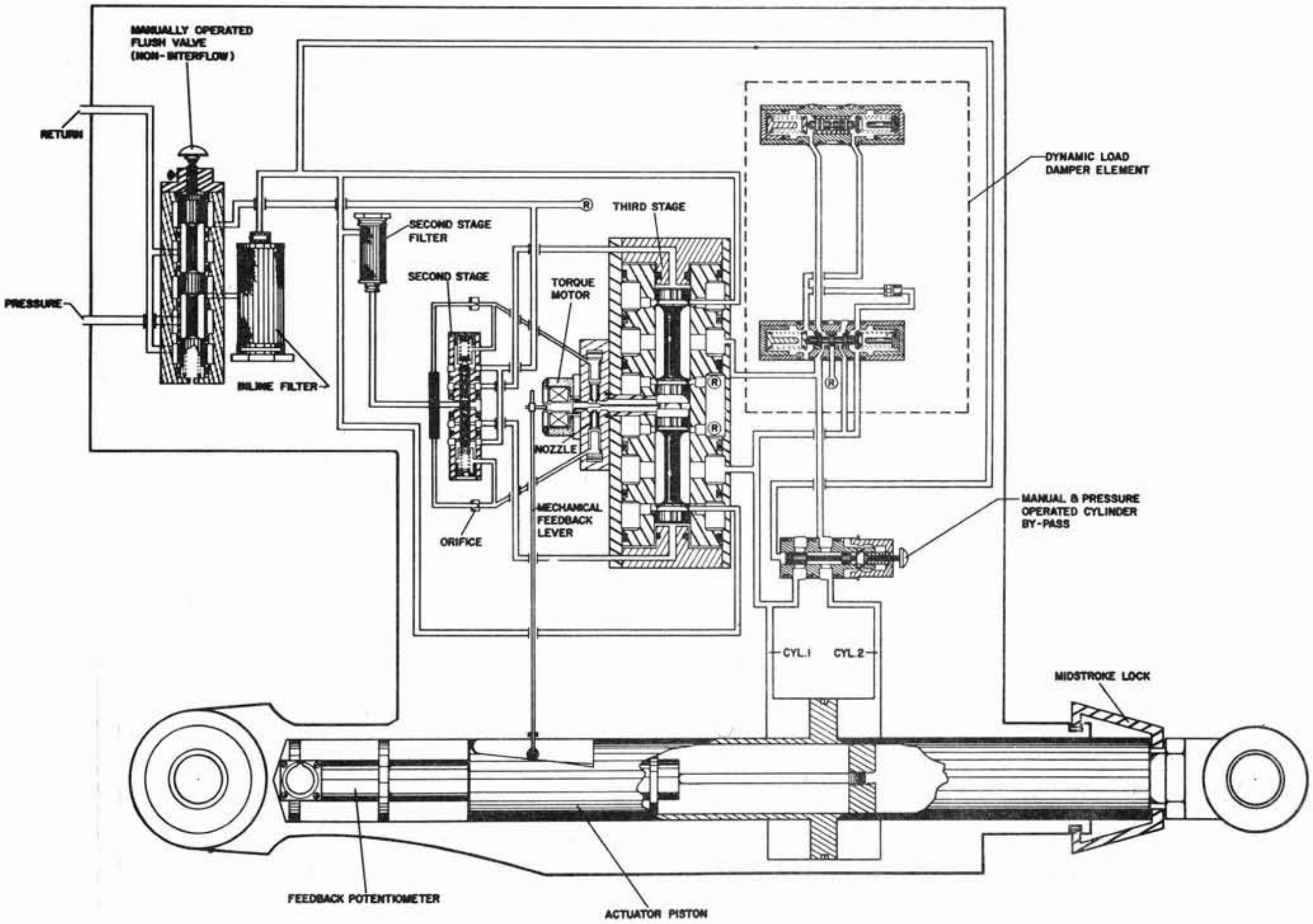

Fig. 18 - S-IC schematic hydraulic research

seal life, actuator piston rods are held to approximately 0.10 micron finish.

The piston seal consists of two step-cut Teflon rings installed with the step cuts 180 degrees from each other and held in position by a locating pin. Initial sealing and low pressure sealing are achieved by the use of a flat stainless steel wave-spring strip installed between the inside diameter of the rings and the piston-head-groove diameter. The seals are pressure energized at operating pressures. The use of Shamban seals on many similar actuators, with excellent reliability records, makes this type of seal most advantageous for this application.

To minimize external leakage, dual-channel piston rod seals are used in this design. The area between the seals is vented to return, which allows the external or oil-to-air seal to operate at return line pressure differential. The redundancy of the seals allows added reliability through the use of a protected restrictor in the vent line. In the event of a failure of either seal, the restrictor would allow only a limited amount of oil to pass either overboard or from cylinder to return. As the restrictor size can be very small to allow normal inner seal leakage to vent to return, its maximum flow rate from return overboard in the event of failure of the outer seal would be approximately 0.135 cm^3/s (0.5 $in.^3/min$). In the event of an inner seal failure, the small restrictor would be effective in limiting the cylinder port return flow to less than 2.73 cm^3/s (10 $in.^3/min$) and thereby extending the control period a sufficient amount to achieve the flight objective. A Teflon scraper is used to protect external seals from damage by particles adhering to the shaft surface and thus aids in the achievement of maximum reliability.

The actuator housing and accessory body are fabricated from 2014-T6 aluminum alloy forging. The forging is done with two dies, a rough (blocker) die and a finishing die. Profiling and machining are held to a minimum on the external configuration to make the best possible use of increased strength provided by forging flow lines and forged crystalline structure of the material. The 136 kg (300 lb) actuator is shown in Figure 19.

SERVOVALVE DESIGN - A three stage servovalve was selected for this application based on two considerations: (1) electrical power required to develop the maximum flow and (2) high flow forces on the power stage. Available electrical power to the torque motor is 0.25 watts. This power is only available at full stroke of the actuator piston. Only eleven percent of this power (.028 watts) is available to obtain the maximum flow required when the torque motor is sized so that the servo loop achieves the dynamic and static performance requirements. A three stage servovalve is needed for the high power amplification from .028 watts to 11,900 cm^3/s (193 gpm).

These high power stage flows generate significant flow forces which affect servovalve performance. These effects are minimized by a three stage configuration as the driving force to the power stage is much higher than that of a two stage valve.

The existing torque motor design has a proven MTBF of over 65,000 hours as part of a primary flight control system. This design has been scaled up by a factor of 1.5 to provide adequate margins of torque to assure dynamic response and reliability within the required limits.

The increased size of the torque motor allows the large coil to be constructed with AWG 33 wire. This compares with conventional coils which are wound with wire as small as AWG 44. The advantage of the larger wire lies chiefly in the increased reliability of the junction of the coil wire and lead wire. Faulty junctions at this point are a major cause of failure in conventional coils.

Null shifts and nonlinearities in the torque motor affect piston position directly as the torque motor is outside the position loop. Experience has proven that there is an improved resistance to environmental changes when magnetic path air gaps in the torque motor are kept large in comparison with normal armature excursions. The air gaps corresponding to the required torque motor output are approximately 40 percent greater than those of the smaller design. Linearity is within 5 percent up to rated signal.

The hydraulic amplifier is a conventional double nozzle-flapper configuration. The nozzles and orifices used are identical to those used in present Hydraulic Research production valves. The nozzle material is 440 stainless steel at a hardness of 58-60 RC. The orifice material is naval brass. The largest feasible nozzle spacing using available torque motor

Fig. 19 - Model 300300 electrohydraulic mechanical feedback servoactuator

power was selected. This nozzle spacing will allow greater acceptance of contamination. This advantage will permit the use of filters of higher micron ratings. The net result will be that filter life will be greatly increased even with a badly contaminated system.

The second stage is a spring-centered, three-land spool 0.906 cm (0.357 in.) in diameter. The configuration chosen is identical to the power stage used in production servovalves. The sleeve seals are conventional O rings. The spool and sleeve materials are 440 stainless steel, heat treated to 58-60 RC hardness.

The third stage spool design was dictated by flow output requirements and dynamic considerations. The combination of 3.7 cm (1.5 in.) diameter and 0.178 cm (0.070 in.) stroke is used to give the required flow slot area. This spool is of the three-land type, with the outer two lands being used as the pressure lands. The sleeve contains the O-ring grooves for fluid sealing. A portion of the annular flow passage is in the sleeve and a part is a matching groove in the body. This is a weight saving device because it allows making the seal sleeve small in diameter. The spool and sleeve material is 440 stainless steel, heat treated to 58-60 RC hardness.

MECHANICAL FEEDBACK DESIGN - The feedback stroke divider reduces and translates the rectilinear portion of the actuator piston to rotary motion at the feedback arm. Motion of the fe back arm torques the armature to null the ser actuator position loop.

The mechanical feedback device was selected after a study of many feedback devices built by Hydraulic Research as well as experience with a breadboard model of the design used. The various methods of mechanical feedback considered are shown in Figure 20. The helix version of the mechanical feedback device is used for the S-IC actuator because of its minimum envelope, simplicity, and minimum number of moving parts. One moving part, the rotary output cylinder, is mass balanced to minimize the effects of vibration and shock. This device has three major components: (1) the follower assembly which is driven directly by the actuator piston and is allowed to rotate, (2) the stationary shaft which is helix keyed and causes the follower assembly to rotate, and (3) the rotary output cylinder which has the feedback arm is rotated by the follower assembly.

The engine position control loop is shown schematically in Figure 21. A positive command signal to the torque motor generates a clockwise torque that causes the second stage and third stage to be hydraulically driven to the right. Cylinder port C_1 is pressurized and the actuator body and engine are driven downward. The mechanical feedback mechanism senses the relati-

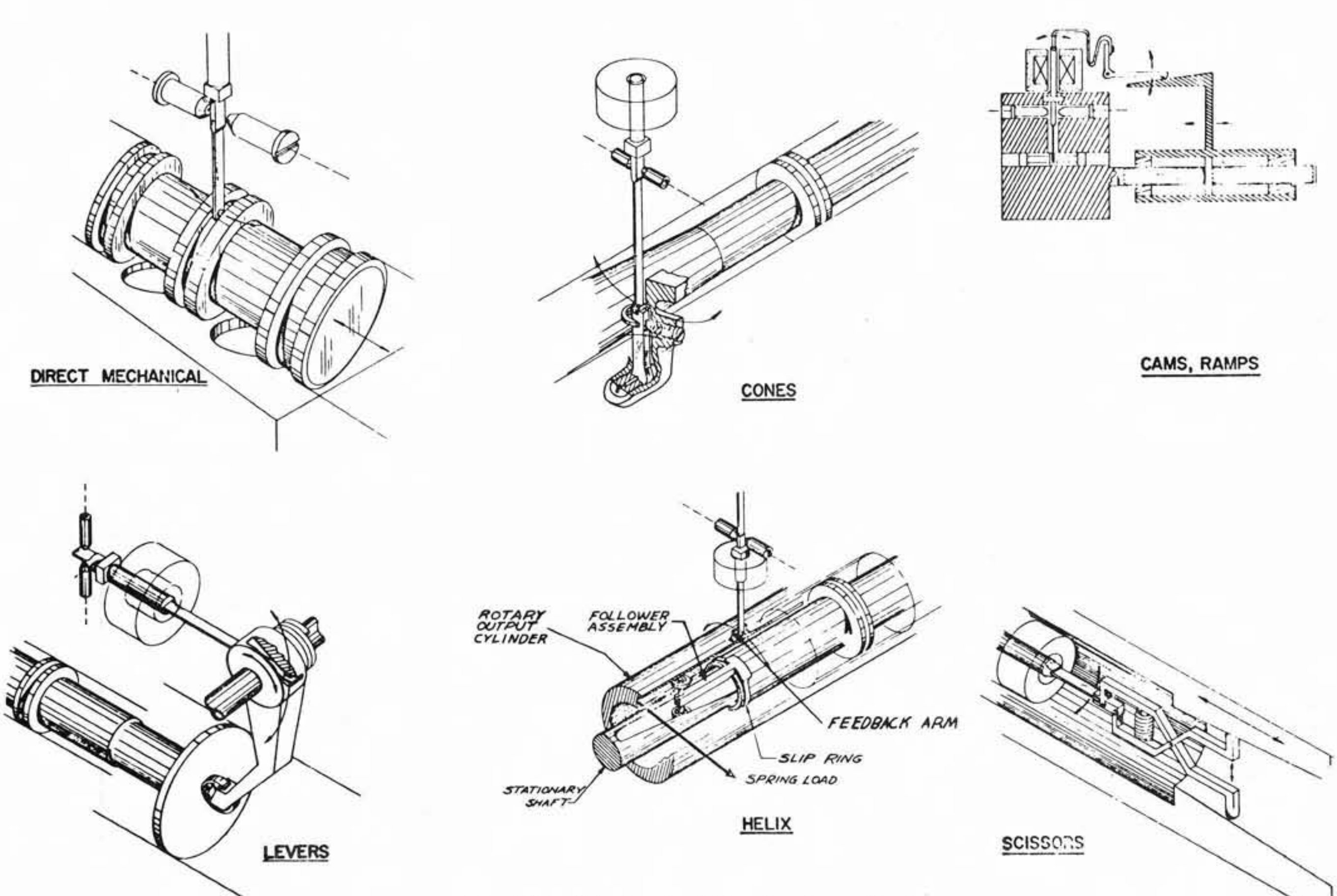

Fig. 20 - Mechanical feedback methods

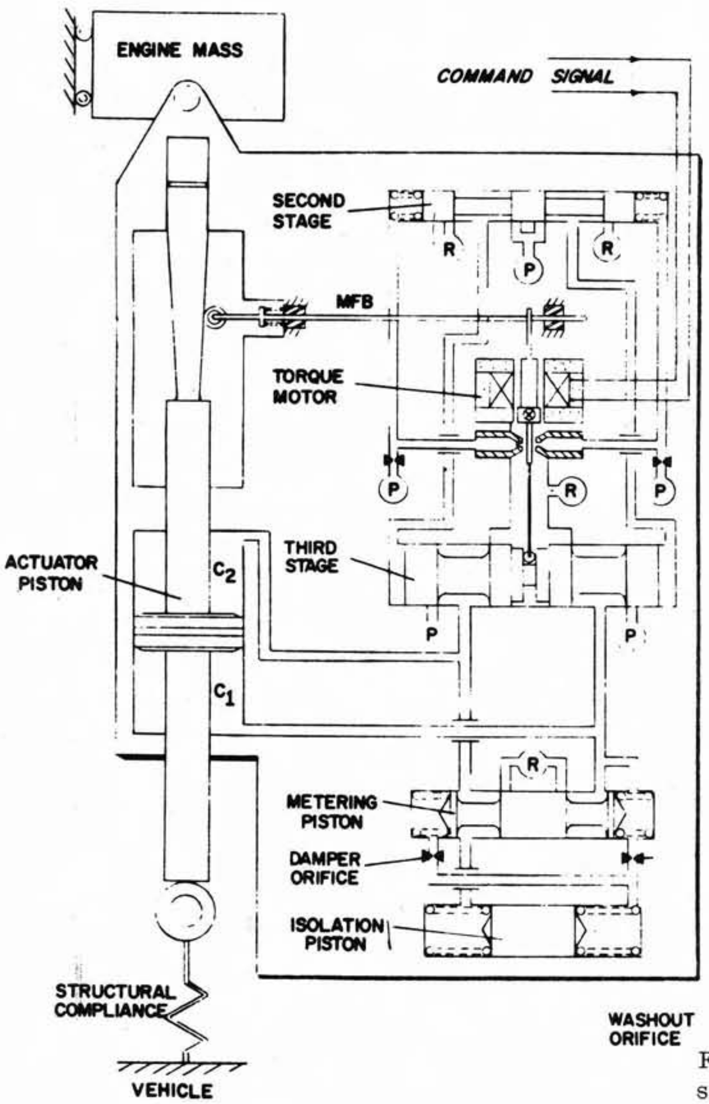

Fig. 21 - S-IC servoactuator control schematic

motion between piston and body and applies a counterclockwise torque to the torque motor armature. The actuator stops at the position where the feedback torque equals the command-signal-generated torque.

LOAD DAMPING DESIGN - The pistons and orifices shown in the lower portion of Figure 21 form a hydraulic shaping network designed to limit engine displacement near load resonant frequencies. This load resonance is caused by the engine mass being attached through the servoactuator and structural compliance to the vehicle. A pressure pulse in C1 will displace the metering piston to the left, venting C_1 to return, thereby limiting the instantaneous pressure buildup. The washout orifice and isolation piston allow the metering piston springs to center the metering piston for steady state pressure loads on the actuator. In this manner, output power of the actuator is limited at load resonance and static stiffness of the actuator is maintained. This assembly is called the dynamic load damper (DLD).

The DLD provides load damping without increasing the complexity of the servo flow control valve. Failure or partial failure of the DLD network will not affect the operation of the servovalve, thus affording an additional margin of reliability. The DLD network is comprised of two spring-centered spool-and-sleeve assemblies. Each assembly is complete in itself and does not depend on an operational reference to the housing, which eliminates potential problems caused by thermal expansion or shifts caused by acceleration. The material used in the spool-

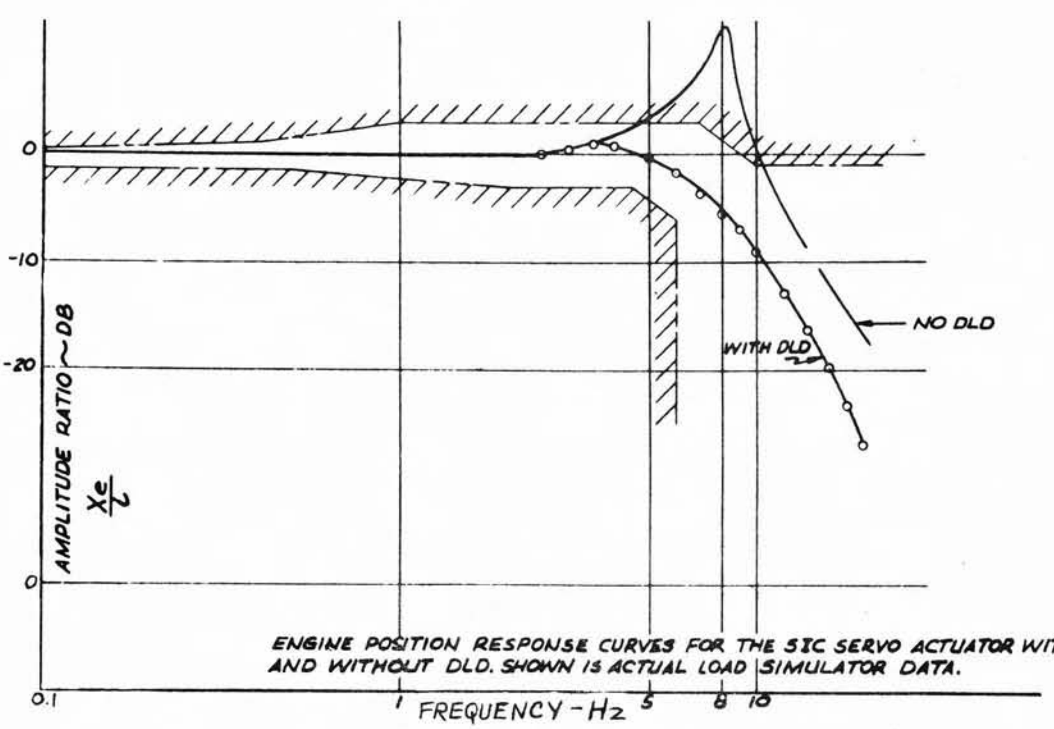

Fig. 22 - Engine position response curves

and-sleeve assembly of the DLD is hardened 440 stainless steel. Special lapping and construction techniques are used to assure the lowest possible frictional threshold in these components.

Engine position response curves, with and without the dynamic load damper (DLD), are shown in Figure 22.

CONCLUSION

The Saturn-Apollo program is an ambitious undertaking. The guidance and control systems use complex methods and have stringent accuracy demands. Special design, manufacturing, and test precautions are used to obtain the high reliability required for this mission. Redundancy is used where feasible. Thrust vector control systems have been optimized through studies, evaluations, and advancement of state-of-the-art system approaches and hardware design. The S-IC hydraulic system is greatly simplified by using the engine fuel as a fluid power source. Reliability has been improved and checkout greatly simplified by using mechanical feedback servoactuators.

The replacement of the potentiometer with the more reliable mechanical feedback mechanism and the improvement in the design of the servovalve have resulted in actuator reliability that exceeds that required of the actuator in the reliability budget.

To guarantee the quality and improve the reliability of all Saturn vehicle components, MSFC has implemented a strict surveillance and traceability program. The NASA Quality Control NPC 200 documents are the medium used to establish the quality programs that insure that the NASA space systems, launch vehicles, spacecraft, and the associated ground support equipment achieve the required reliability for a successful mission.

The NASA Reliability Program Provisions for Space System Contractors (NPC 250-1) was established as a guide for designing reliability into space systems. The NASA programs cannot tolerate an approach in which there is relatively high risk in first design attempts and where the necessary reliability is only achieved in later design. The NASA programs, with only small quantity production, must have the required reliability in the first as well as all later designs.

With the use of reliability prediction models, designers can reevaluate and determine where to improve their designs. These models should be used more to find and improve the low reliability areas of a design.

The many component tests, system checkouts, and vehicle simulated flight tests, and a critical review of the results obtained, have eliminated many malfunctions in flight. Every effort made to assure quality and improve reliability in the Saturn vehicle is an effort to insure a manned lunar landing in this decade.

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