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SATURN I ENGINE GIMBAL AND THRUST VECTOR CONTROL SYSTEMS

By

M. A. KALANGE and V. R. NEILAND

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By

M. A. KALANGE
Chief, Flight Control Section
Astrionics Laboratory

V. R. NEILAND
Chief, Fluid Design Unit
Propulsion & Vehicle Engineering Laboratory

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CONVERSION FACTORS

$\frac{kg}{ton}$	907.2
$\frac{kg\text{-}m^2}{slug\text{-}ft^2}$	1.3558
$\frac{kg}{\frac{lbs\text{-}sec^2}{in}}$	175.12
$\frac{kg}{lbs}$	0.45359
$\frac{m}{ft}$	0.3048
$\frac{cm}{in}$	2.54
$\frac{cm^2}{in^2}$	6.451
$\frac{cm^3}{in^3}$	16.387
$\frac{N}{lbs}$	4.4482
$\frac{kN/m^2}{psi}$	6.89
$\frac{N/m}{lbs/in}$	175.12
$\frac{MN}{ft\text{-}lbs}$	1.3558
$\frac{cm^3/s}{GPM}$	63.09
$\frac{Rad}{Deg}$	1.745×10^{-2}

SATURN I ENGINE GIMBAL AND THRUST VECTOR CONTROL SYSTEMS

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George C. Marshall Space Flight Center

ABSTRACT

The hydraulic systems for the two-stage Block II (improved) Saturn I vehicle are described with an evolution of their development.

The S-I stage is powered by a cluster of eight H-1 engines attached to the thrust frame of the aft end of the vehicle. A separate hydraulic system is employed to gimbal each of the four outboard engines which control the vehicle during first stage powered flight. Marshall Space Flight Center's design and development responsibilities were accomplished with significant support from the hydraulic industry. The design incorporates specific techniques resulting from the experience, procedures, and methods employed on previous programs. Particular emphasis is placed on the problems associated with contamination control, reduction of leaks, maintaining a "closed system" concept, and system dynamics.

The concepts utilized in the S-I system are being applied to the second stage of the Saturn I vehicle, the S-IV, which is under development by Douglas Aircraft Company. The S-IV vehicle is controlled by gimbaling six Pratt & Whitney RL-10 engines in response to control signals from the vehicle's instrument unit.

SATURN I

Saturn I (Fig. 1) is the first of the three Apollo launch vehicles. This two-stage vehicle can place 9980 kg of payload in low earth orbit. The Saturn vehicle is 57 m high and has a diameter of 6. 4 m. Its mission will be both manned and unmanned Apollo orbital tests, escape system qualifications, command and service module tests, and re-entry tests. In its early booster flights, all test objectives were achieved.

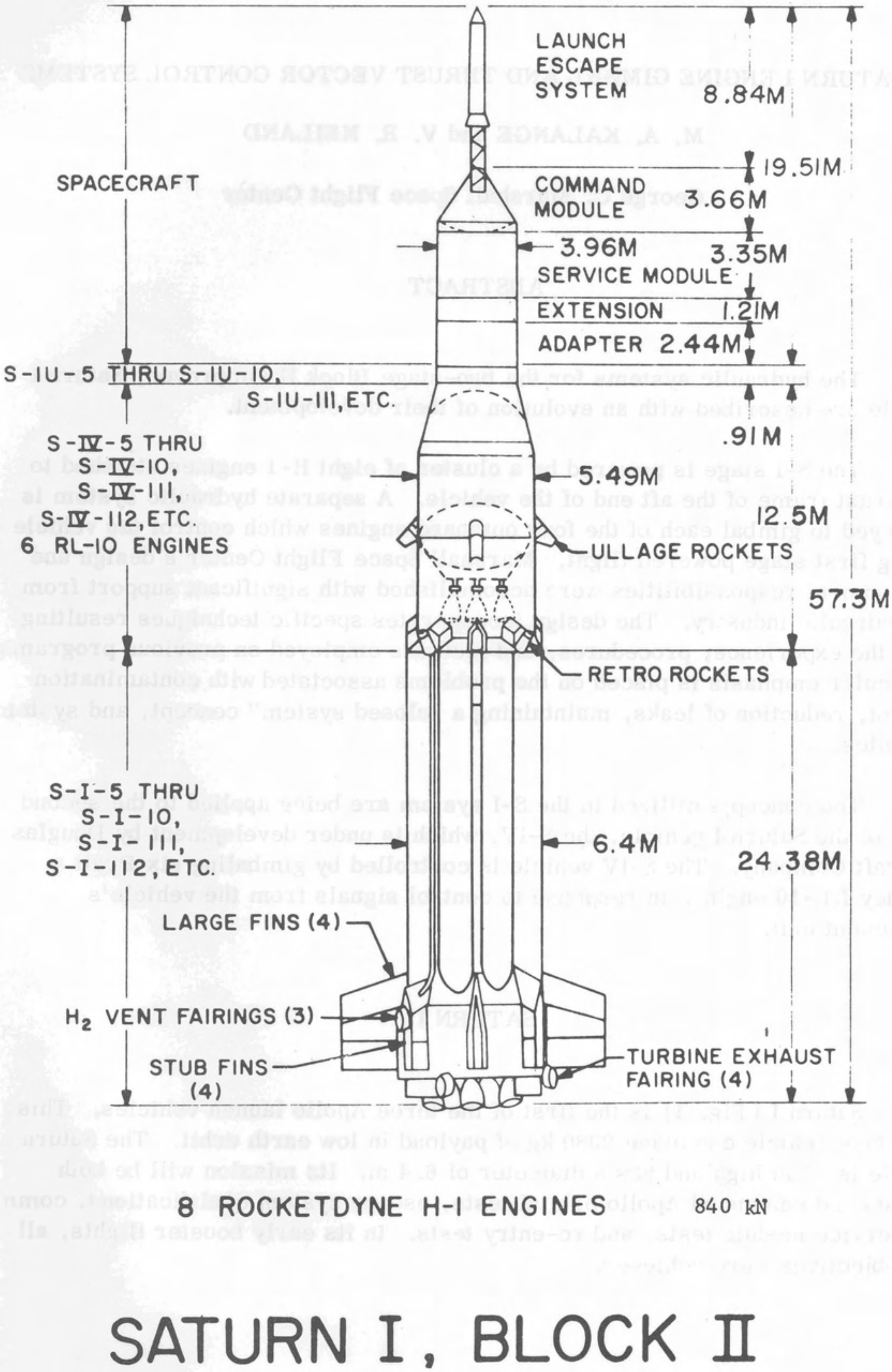


FIGURE 1. SATURN I VEHICLE

When the idea of Saturn was conceived, it was decided that the most reliable, expeditious, and economical way to build the space vehicle was to utilize, as much as possible, the components and systems that were available from the Juno II and Jupiter-C programs.

The first stage of Saturn I (Fig. 2), the S-I, was designed by Marshall Space Flight Center (MSFC) and used clusters of lengthened Jupiter-C and Juno II propellant tanks and improved Juno II engines. Each of the eight Rocketdyne H-1 engines burns RP-1 (kerosene) and liquid oxygen to develop 840 kN thrust for a total vehicle thrust of 6.7 MN during the 140-150 second programed firing time. The S-I stage stands 24.4 m high and has a diameter of 6.4 m.

The second stage of the Saturn I vehicle, the S-IV (Fig. 3), is being developed by the Douglas Aircraft Company. It has a cluster of six Pratt & Whitney RL-10 engines, each burning liquid hydrogen and liquid oxygen to develop 67 kN thrust for a total of 400 kN. Burning time for this stage is 460 seconds. The S-IV stage is 5.5 m in diameter and 12.5 m tall.

SATURN ATTITUDE CONTROL SYSTEM

An attitude control system is necessary to augment stabilization and to orient the Saturn vehicle on the required flight trajectory to complete its missions. Thrust vector control is used in Saturn I to maintain attitude control.

Control reference signals (Fig. 4) originate from a four-gimbal stabilized platform located in a section of the vehicle called the Instrument Unit. Here the reference signals are mixed with sensing signals in an analog control computer. Attitude rate information is obtained by differentiation in the control computer, but may be obtained from rate gyros in later vehicle configurations. Information to reduce lateral load torques caused by wind is obtained from body-fixed accelerometers. The signals from these sensors are shaped and mixed in the control computer in the proper gain and phase relationship and then routed to the actuation systems on the four outer engines (Fig. 5) which are gimbaled for control. To direct the engine thrust vector through the vehicle center of gravity, the outer engines are canted 0.1 rad to the longitudinal axis of the vehicle. The four inboard engines are fixed at a cant angle of 5.2×10^{-2} rad.

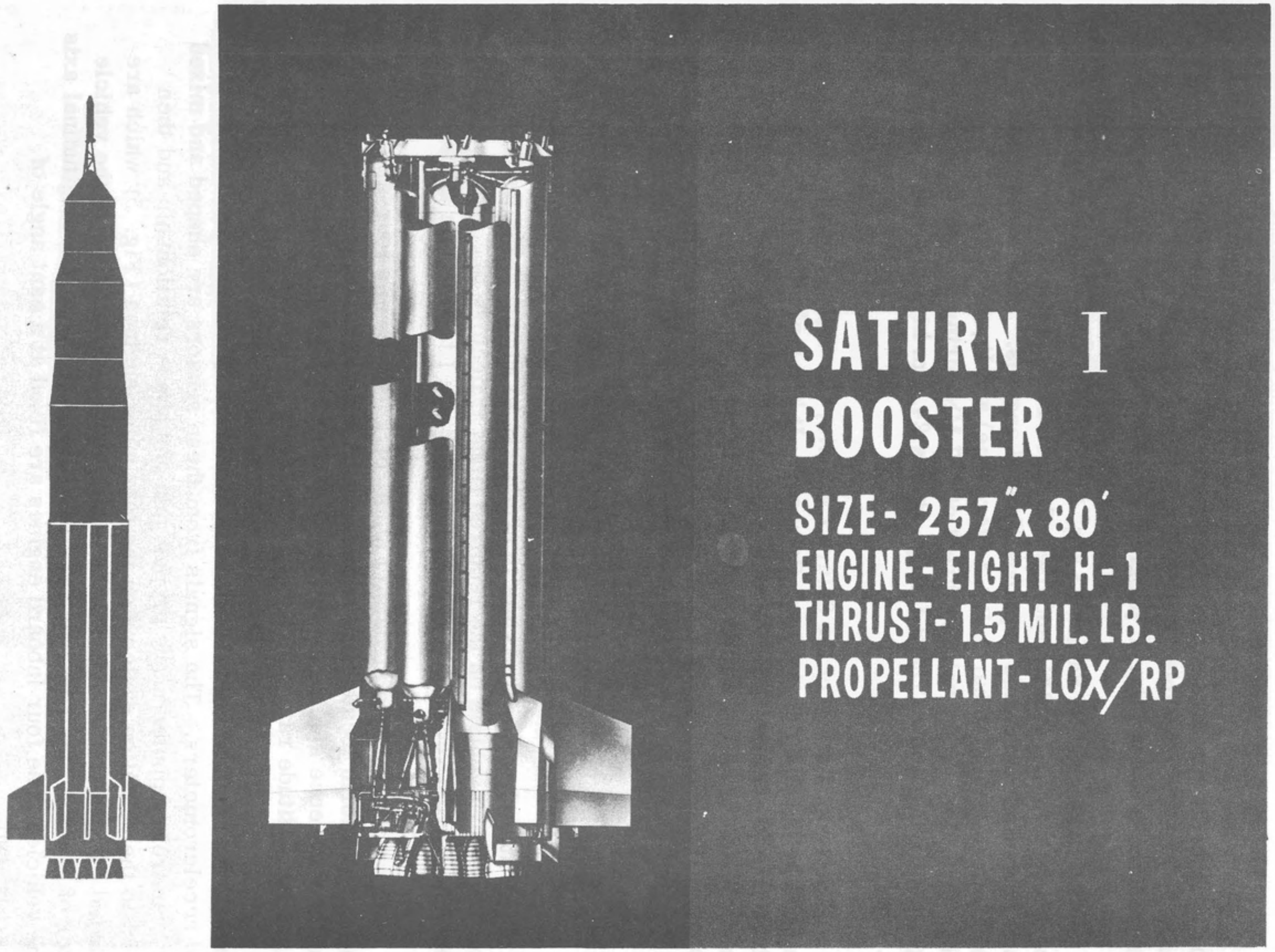


FIGURE 2. SATURN S-I BOOSTER

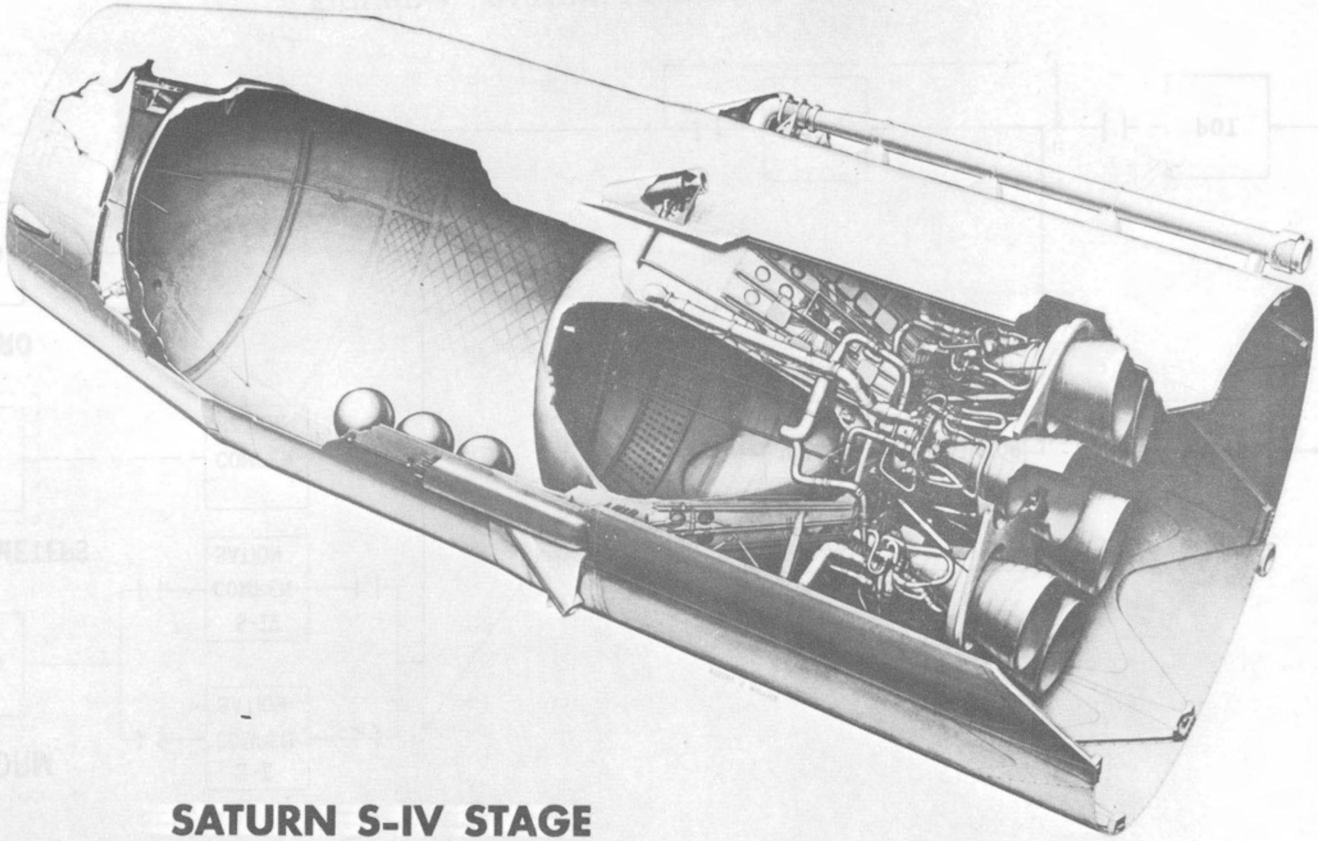


FIGURE 3.   SATURN S-IV STAGE

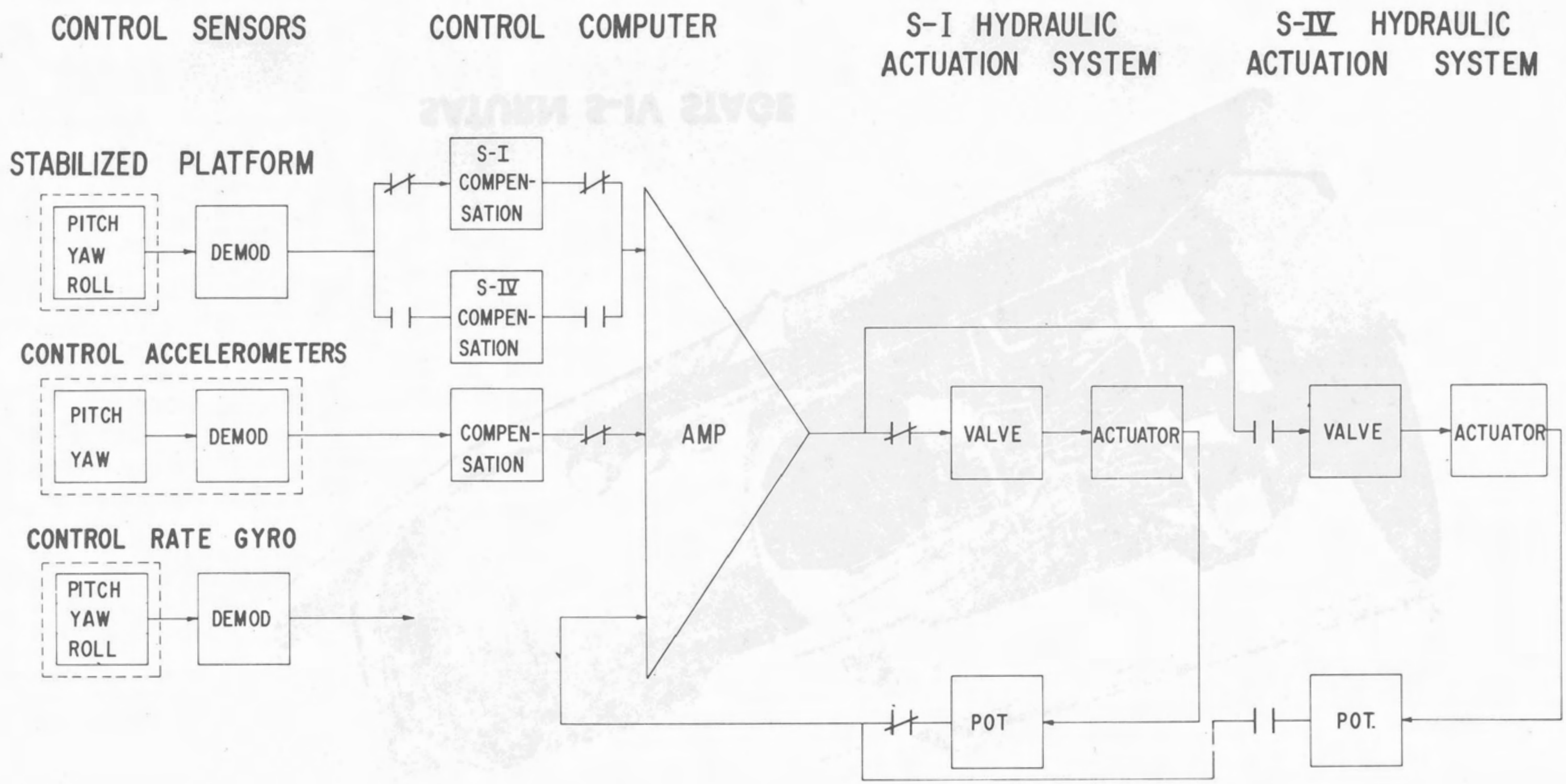


FIGURE 4. SATURN I CONTROL SYSTEM

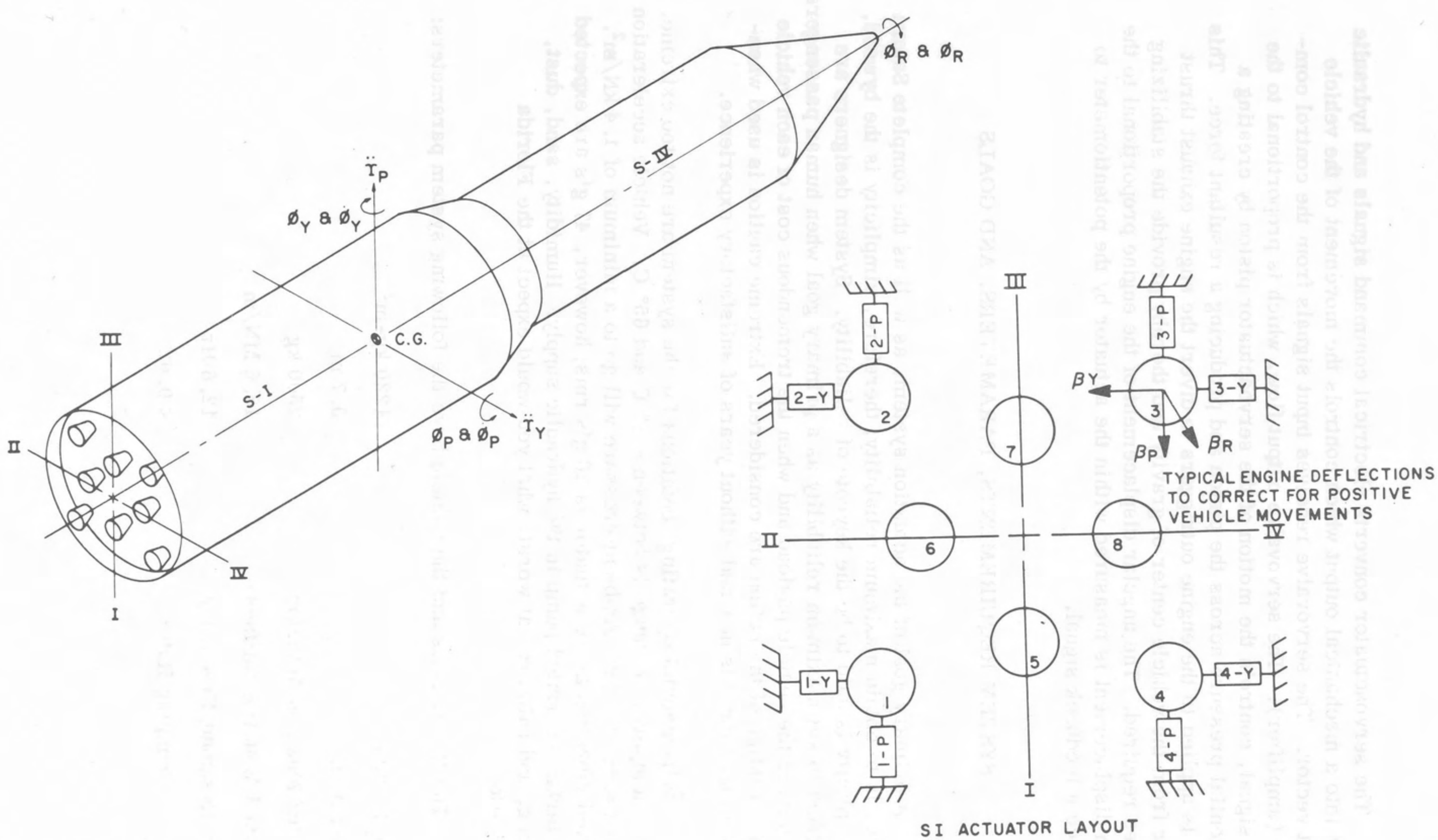


FIGURE 5. S-I ENGINE CLUSTER

The servoactuator converts electrical command signals and hydraulic power into a mechanical output which controls the movement of the vehicle thrust vector. The servovalve receives input signals from the control computer (amplifier). The servovalve output flow, which is proportional to the input signal, controls the motion of the servoactuator piston by creating a differential pressure across the piston and producing a resultant force. This force is applied to the engine outriggers to divert the engine exhaust thrust vector from the vehicle center of gravity and thereby provide the stabilizing torques required. The angular displacement of the engine proportional to the piston displacement is measured within the actuator by the potentiometer to produce a feedback signal.

SYSTEM REQUIREMENTS, PARAMETERS, AND GOALS

A primary goal of the actuation system, as well as the complete Saturn vehicle, is to obtain maximum reliability; therefore, simplicity is the byword, for simplicity is said to be the keynote of reliability. System designers are compelled to set maximum reliability as a primary goal when human passengers are a part of the vehicle payload and when the tremendous cost of each vehicle and the prestige of the nation are considered. Extreme caution is used whenever a requirement is new and without years of satisfactory experience.

Environmental operating conditions for the system are not too extreme. Ambient temperature range is between -18° C and 65° C. Vehicle acceleration does not exceed 6 g's. Ambient pressure will go to a minimum of 1.4 kN/m^2. Specified vibration at the actuator is 26 g's rms; however, 47 g's are expected at the turbine-mounted pump in the hydraulic supply. Humidity, sand, dust, salt, fog, and rain are, at worst, what you would expect at the Florida launch site.

The H-1 engines and the vehicle have the following system parameters:

Engine Inertia	1220 $kg\text{-}m^2$
Moment Arm	0.7 m
Reflected Mass on Actuator	2500 kg
Combined Mounting Stiffness	15.6 MN/m
Engine Resonant Frequency	12.6 Hz
Estimated Damping Ratio	< 0.02

Maximum gimbal torques are encountered at the time of maximum dynamic pressure. These estimated loads are itemized as follows:

Source	Torque in m-N
Longitudinal Acceleration	5700
Offset Thrust	5800
Gimbal Bearing Friction	5000
Lateral Translational Acceleration	1200
Vehicle Rotational Acceleration	1800
Engine Angular Acceleration	2000
Propellant Suction Lines	2700
Change in Turbine Angular Momentum	100
Change in Momentum of Exhaust Gas Flow with Nozzle Rotation	700
Heat Shield Curtains	14 400

A dynamic load design value of 45 000 N actuator force is calculated as a requirement from these gimbal torques. The actuation system is required to gimbal the outboard engines through an angle of ± 0. 14 rad at a maximum rate of 0. 29 rad/s at this design load. These forces, excursions, and rates, necessary to maintain adequate vehicle stability, required the selection of a hydraulic system as the S-1 stage actuation system. The system was designed to meet the following requirements:

Stall Force at 21 MN/m^2	67 kN
Rated Force	45 kN
Velocity at Rated Force	20. 4 cm/s 0. 29 rad/s
Displacement	± 96 mm ± 0. 14 rad
Position Loop Gain (s^{-1})	20
Pressure Loop Gain (s^{-1})	20
Engine Frequency Response	< 0. 35 rad at 1 Hz < ± 1 db to 6 Hz 1 db max. AR

HYDRAULIC SYSTEM

The Juno II hydraulic system was developed as an "open" hydraulic system; i.e., the system obtained its hydraulic fluid power for checkout and vehicle launch sequencing from a ground system which was disconnected at lift-off. The connecting and disconnecting of the ground system, often in a poor, dusty environment, was a source of contamination. Possibly, some operating personnel were not even aware that most aerosphace hydraulic systems must maintain cleaner fluid than the water from a drinking faucet.

The major problems that needed improvement involved the ground equipment, leakage, and contamination. Possibly, the greatest problem area encountered with Juno II was the uncertain reliability of the ground system from the standpoint of contamination generation and accidental disconnection. Ground system lines were particularly vulnerable to accidents. During early system development, there were many failures of the quick disconnects associated with the ground equipment which resulted in failure of missile components. Human error also caused many malfunctions in the system. It became apparent that a self-contained packaged hydraulic system was necessary; however design freeze prevented the correction of these problems.

When it became apparent that Saturn would consist of multiple gimbaled engines, the following system possibilities were considered:

1. Four independent systems with either one or four ground systems
2. One system for the four engines with one ground system
3. Two independent systems with on-board checkout capability
4. Four independent systems with on-board checkout capability.

The last approach was selected since it eliminated the ground system, consolidated system components, and thereby improved the leakage and contamination aspects.

Having a separate system on each outboard engine offered the following advantages:

1. Great number of single engine tests
2. One system may be checked out at a time
3. Malfunction and replacement can be isolated to one system
4. Performance comparisons are easily made and problem areas quickly located.

The experience gained from the Juno II program was applied to the Saturn system to circumvent the contamination problem. The Saturn "closed" system incorporates specific techniques and includes concepts derived during Juno II hydraulic system development. A "closed" hydraulic system (Fig. 6) is assembled on each of the four outer engines. The system is filled, bled, purged until a satisfactory cleanliness level is obtained, and checked out prior to installation of the engines in the vehicle. Once the system is filled and bled, utilizing a ground hydraulic servicer, the system remains in a "closed" status.

The hydraulic system's major assemblies (Fig. 7) consist of a 950 cm^3/s main pump, a 190 cm^3/s auxiliary motor-pump, and an accumulator-reservoir manifold used as the hydraulic supply; all are connected by tubing and flex hoses to the actuators. Minor components such as the filter, filter ΔP indicator, check valves, low and high pressure relief valves, pressure transducer, thermal switch and probe, bleed valves, and filling couplings are contained within the manifold assembly. Gimbaling can be conducted both during engine firing or nonfiring operations without the use of an external hydraulic pressurizing source. For flight, fluid flow required by the actuators is provided from the variable delivery main pump mounted on the turbine-driven engine accessory pad. During checkout, actuator requirements are provided by an accumulator and the auxiliary motor-pump which receives electric power from a ground source. With this system, a fluid contamination count not exceeding the following is consistently being maintained:

100 ML	ARP-598 (SAE)	SAMPLE
Microns		Particles
10 - 25		2150
26 - 50		530
51 - 100		60
Over 100 and Fibers		10

Considerable time (Fig. 8) was required to obtain this level for the first Saturn vehicle, SA-1; now, only a relatively short purging and bleeding time is required. This level corresponds approximately to the SAE-ASTM level No. 2. Satisfactory operation of the hydraulic system for over 50 hours has been demonstrated at the SAE-ASTM level No. 4.

After the vehicle is checked out, it is captive fired for full flight duration. During the firing, a test gimbal program is performed. The system response

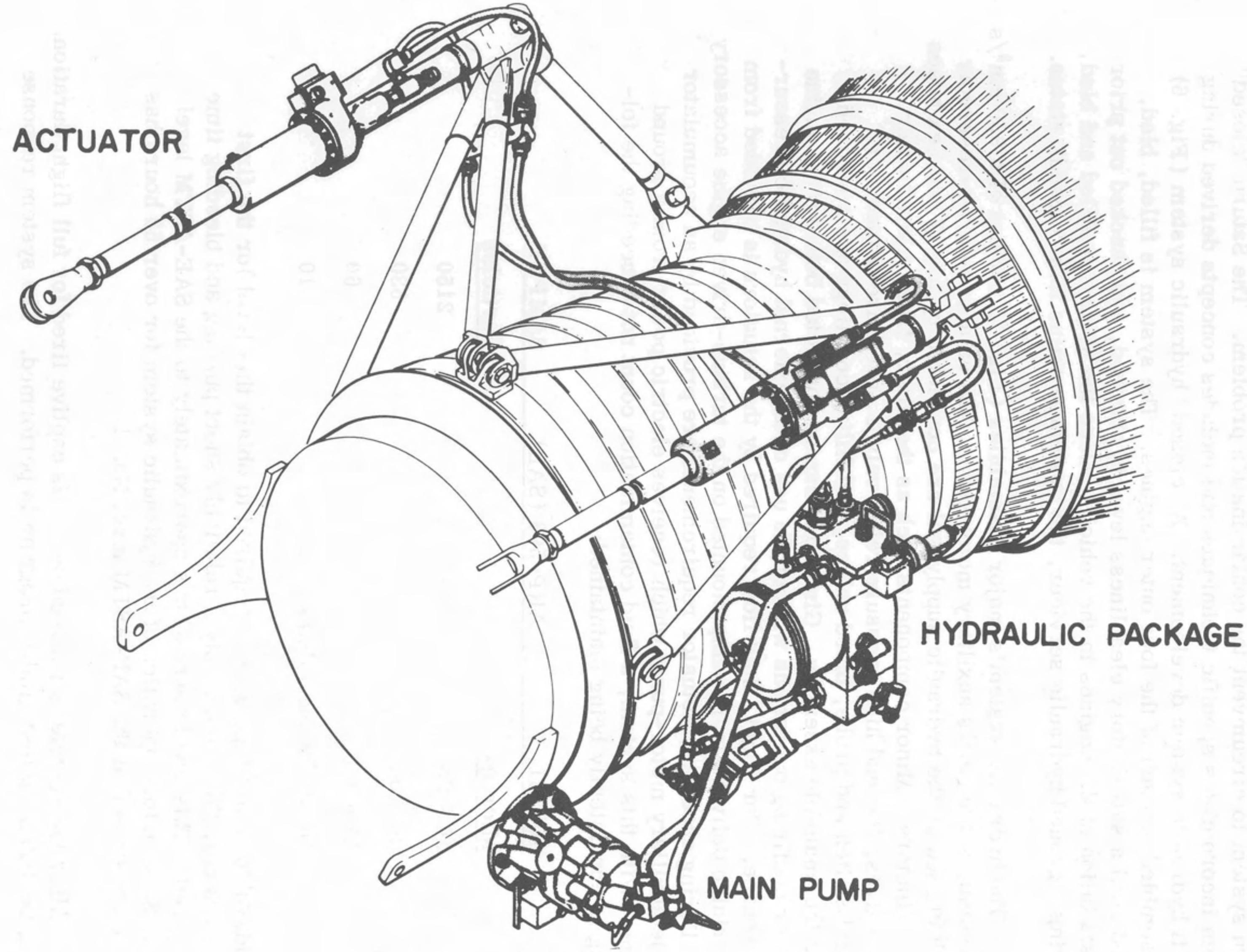


FIGURE 6. H-1 ENGINE AND HYDRAULIC SYSTEM

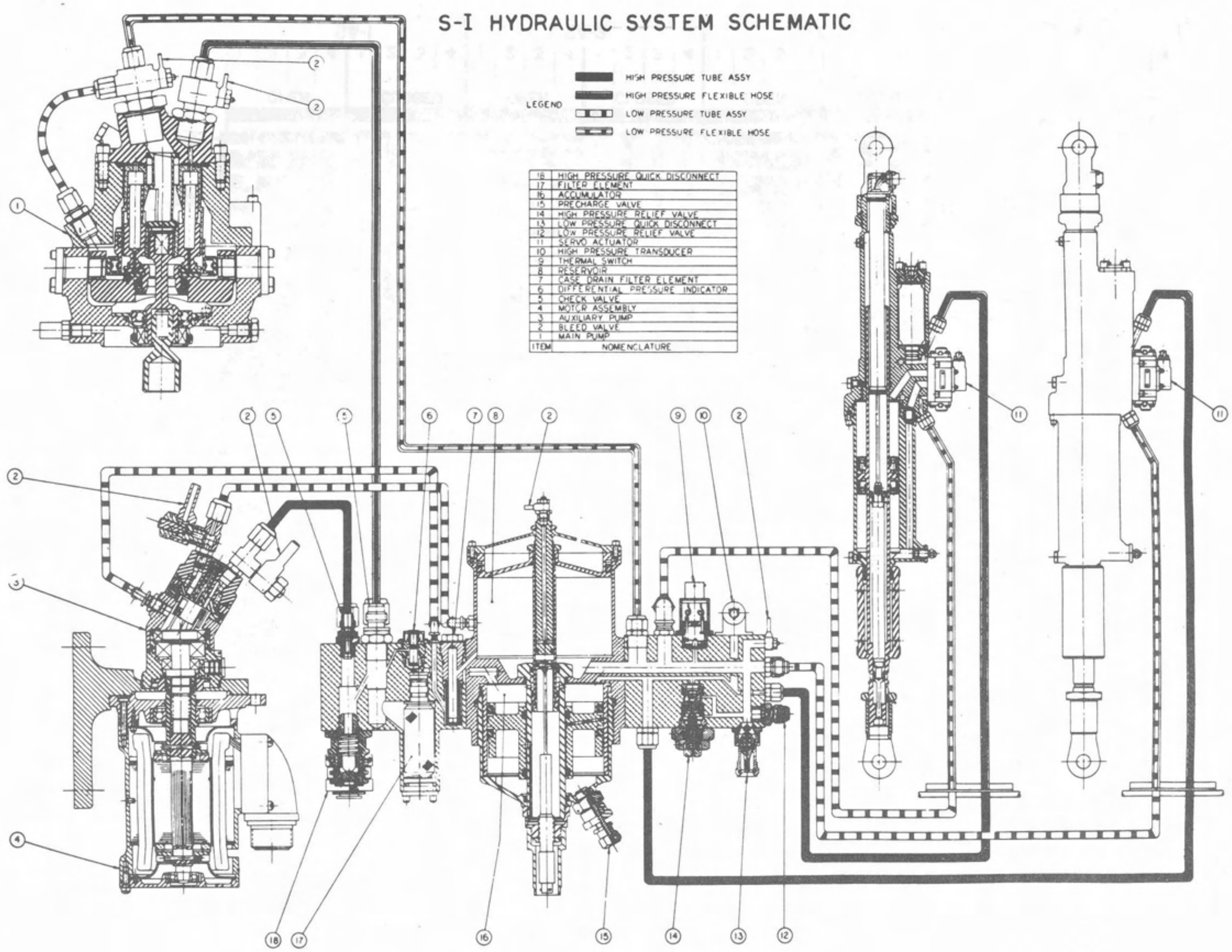


FIGURE 7. S-I HYDRAULIC SYSTEM SCHEMATIC

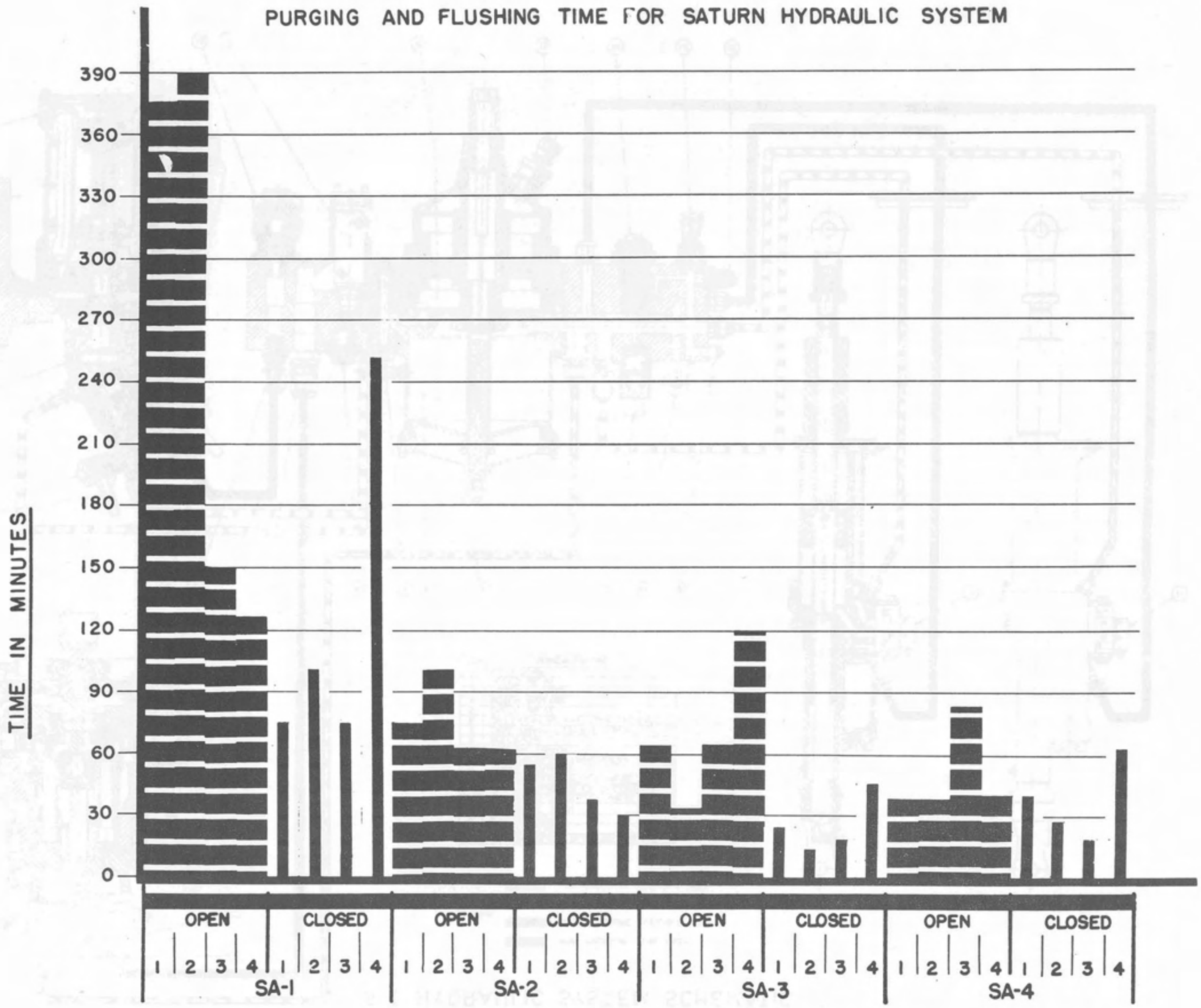


FIGURE 8. PURGING AND FLUSHING TIME FOR HYDRAULIC SYSTEM

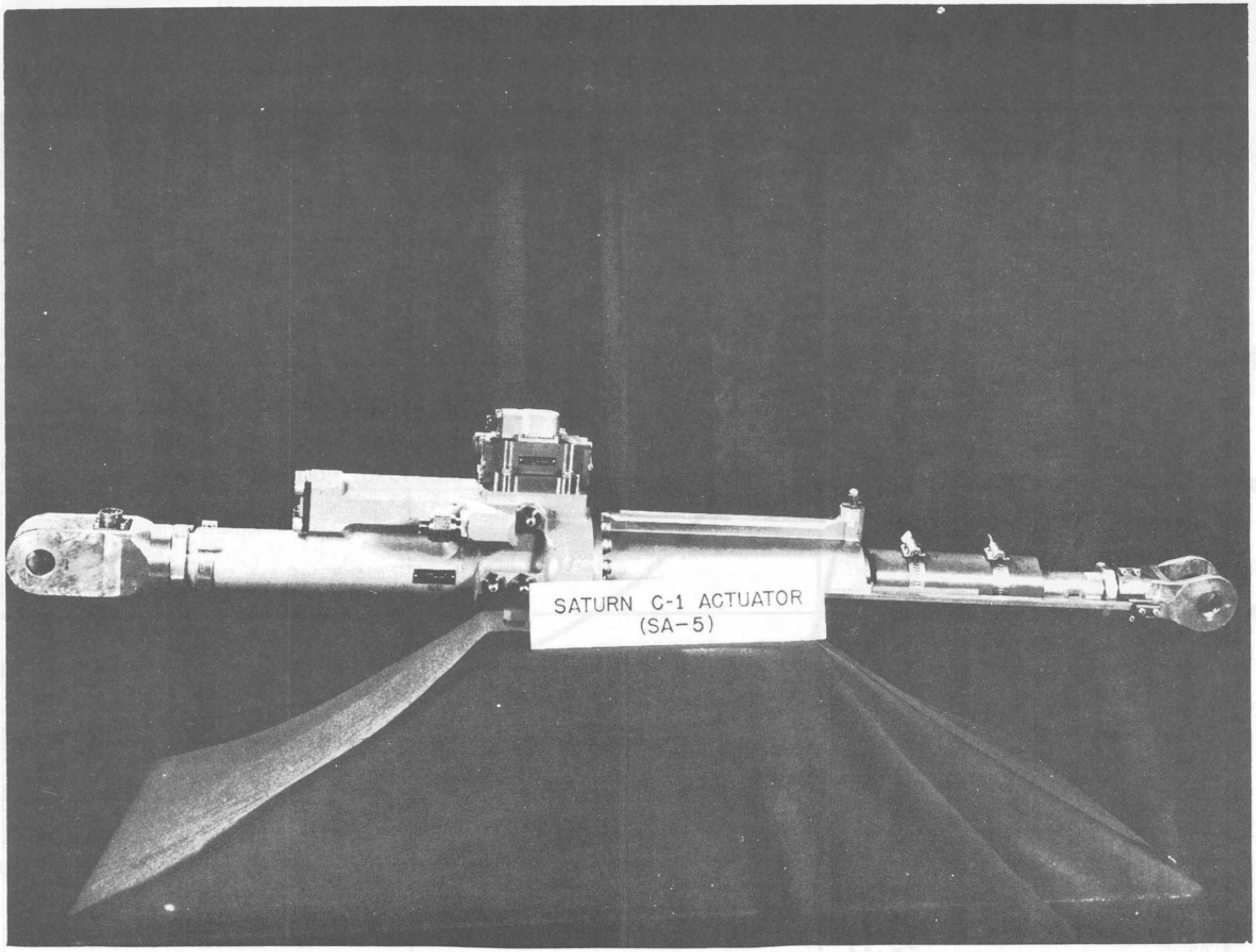


FIGURE 9. S-I ACTUATOR

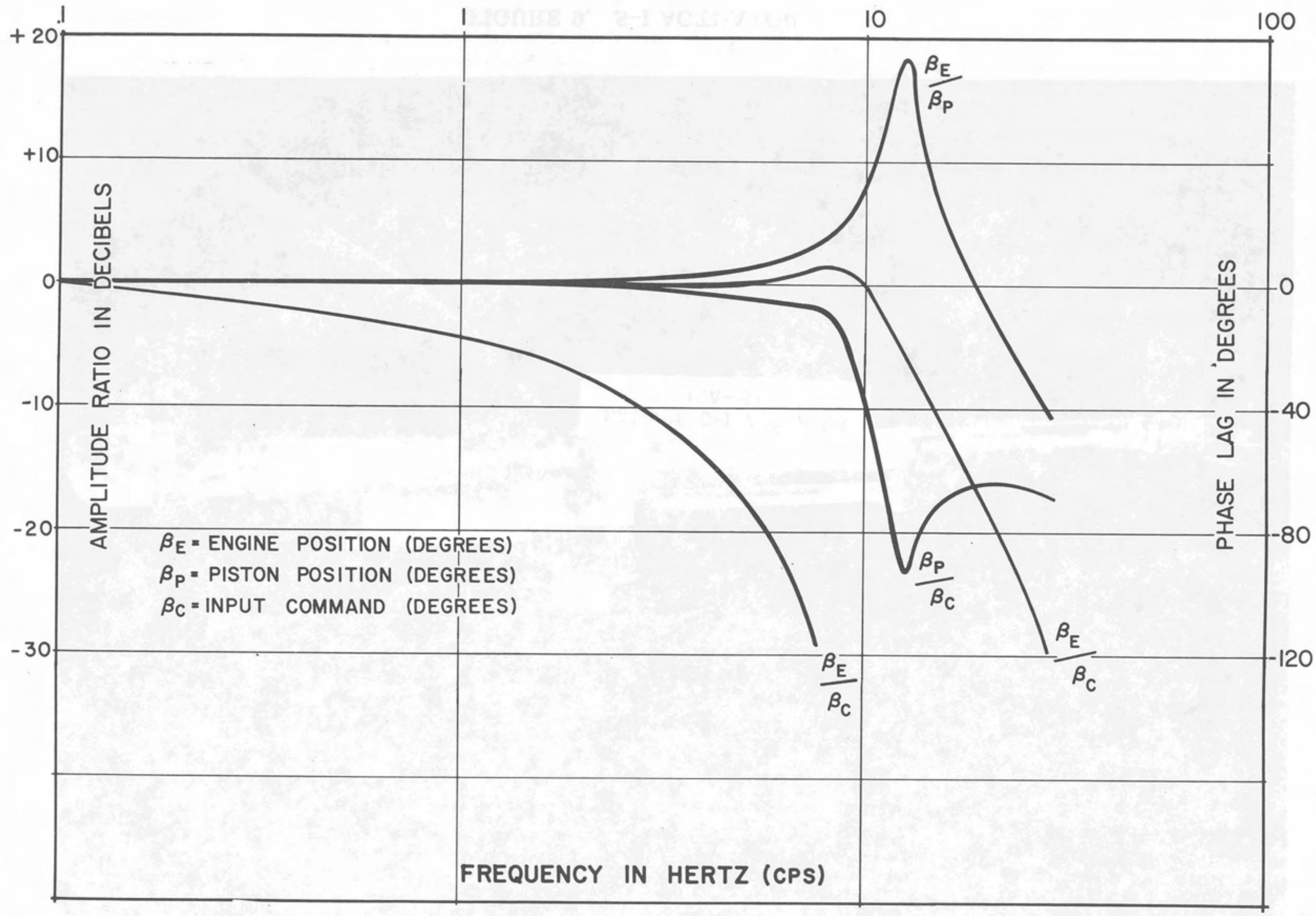


FIGURE 10. GIMBAL SYSTEM RESPONSE

is checked from 1 Hz to 20 Hz at an engine amplitude of ± 0.0087 rad. Step functions of ± 0.035 rad are exercised to observe damping characteristics.

As in flight, the system performance is monitored by a group of measurements on each system. Hydraulic system pressure, reservoir pressure, hydraulic fluid temperature, accumulator level, actuator position, actuator piston differential pressure, and actuator vibration are measured. The S-I stage hydraulic system requirements are not extreme; by state-of-the-art standards, the system is relatively simple. By military specification, it may be classed as a common 21 MN/m^2 (3000 psi) type II system.

S-I SERVOACTUATOR

Actuator Development

Moog Servocontrols, Inc. was given the task of developing the actuators used to position the S-I stage engines. These actuators (Fig. 9) are self-contained, integrated units incorporating a double-acting, equal area hydraulic cylinder; an electrohydraulic servovalve; a piston position feedback potentiometer; a cylinder bypass valve; a hydraulic line filter; a flushing valve; and various bleed and test provisions. The initial Saturn actuator design used in early development tests included a flow control servovalve. Similar flow control servovalves had been used previously in the Juno II program. However, the first static test firings of the S-I stage disclosed system dynamic complications caused by an interaction of engine inertia and structural compliance and the resulting small natural damping forces. This resulted in an engine resonance (β_E/β_p) at the system natural frequency as shown in Figure 10.

The flow control valve caused the actuator to act as a stiff link (both statically and dynamically) between the engine and vehicle structure. To obtain additional damping in the control system, the flow control servovalve was replaced with a pressure feedback (PQ) servovalve. The PQ servovalve acts as a notch filter (β_p/β_c) at the engine resonance frequency. In the valve the load differential pressure, as well as the electrical command signal input, is used to control the output flow. This velocity force relationship is equivalent to damping and provides a large stabilizing effect (β_E/β_c) on the controlled load.

The pressure feedback servovalve has been used on the first four Block I Saturn launchings with a 25.8 cm^2 piston area actuator. The Block II Saturn carries the live S-IV stage with the Apollo modules. Preliminary load analysis indicated that this vehicle configuration would require a greater load control capability from the booster actuators. The Block I actuator (piston area) was redesigned to meet this requirement. The Block II design has an area of 32.3 cm^2, a stroke of ± 9.6 cm, and a velocity capability under a 45 kN load of 20.4 cm/s. With this larger area, a servovalve with increased flow capacity was needed. This flow was beyond the capability of the existing pressure feedback servovalve. The high servovalve flow and the system dynamic considerations dictated a new servovalve development. It was decided that the new servovalve should incorporate other improvements in servovalve technology which had taken place in recent years; namely, (1) the use of internal mechanical feedback from the power spool to the torque motor; (2) the use of a torque motor with increased electrical power input (144 mW vs. 64 mW in Juno II and Block I Saturn); and (3) the use of a frequency sensitive filter in the pressure feedback path, which permits pressure feedback only in the range of engine resonance. This dynamic pressure feedback (DPF) improves the static accuracy of the system as the actuator retains the stiffness of a flow control valve at low frequencies; with static actuator loads, such as from an offset of the engine thrust vector from the gimbal point, the position deflection of the actuator is negligible.

Actuator Physical Description

The servoactuator (Fig. 11) has a basic two-piece housing construction consisting of (1) a forged aluminum tailstock and body and (2) a forged steel cylinder with front rod bearings. The piston is a symmetrical, double-ended design. The aluminum body forms the rear bearing for the piston rod. It also contains the majority of actuator accessories, including bores for the flushing and bypass valves, supply and return fittings, filter cavity, bleed and test ports, and a mounting maniford for the servovalve. The use of aluminum for housing most of the accessory components, together with steel for the cylinder and piston members, results in a compact design having adequate strength, maximum weight of 15 kg, and good life and wear characteristics. The aluminum tailstock forging is 2014 with anodize for corrosion protection. The cylinder and piston are forged from 4140 steel and hardened to about Rc 35. Both the piston and cylinder bore are hard chrome-plated. The external surfaces of the cylinder are cadmium-plated for corrosion protection. Alignment of the rod bearings and cylinder bore is established by piloting of the cylinder into the aluminum body. The two housing members are joined by a flange and ring of attachment bolts. The mounting of the actuators is accomplished by clevis fittings at each end; these fittings mate with spherical bearings contained on the engine and in the vehicle structure. Both end fittings are adjustable and contain locks to permit setting of the nomial actuator overall length at the time of installation in the vehicle.

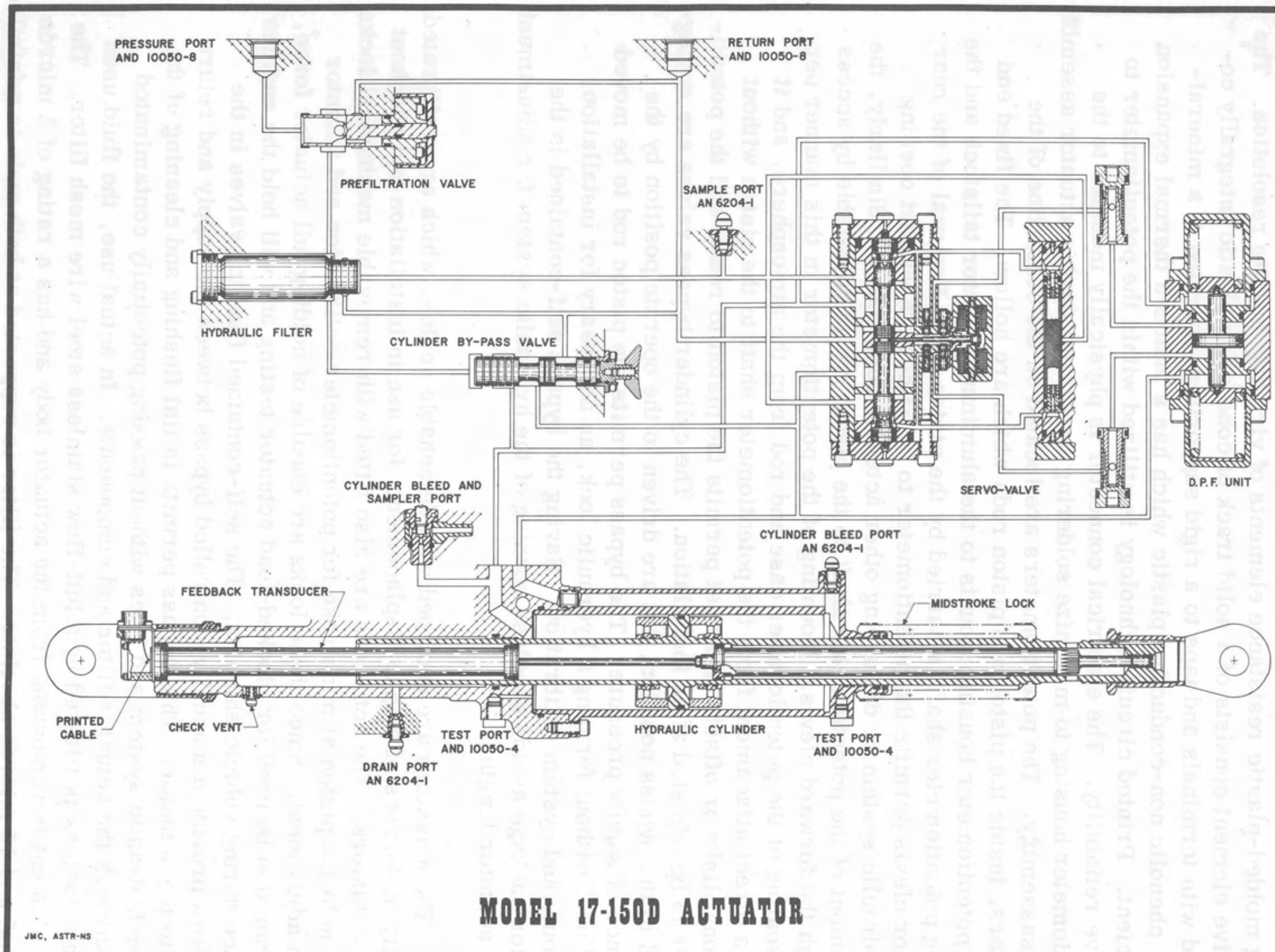


FIGURE 11. S-I ACTUATOR SCHEMATIC

The actuators contain a full-stroke, dual-element Markite potentiometer having molded-plastic resistance elements of virtually infinite resolution. The resistive element consists of a solid track of conductive plastic integrally co-molded with terminals and taps to a rigid supporting structure of a mineral-filled, phenolic non-conductive plastic which has a matched thermal expansion coefficient. Printed circuit technology is utilized within the potentiometer to improve reliability. The electrical connector is physically mounted to the potentiometer housing to minimize soldering operations during actuator assembly and disassembly. The potentiometers are located on the centerline of the actuators, inside the piston and piston rods which are hollow. The fixed end of the potentiometer housing mounts to the aluminum actuator tailstock and the moving potentiometer shaft is carried by the piston rod. Removal of the rear actuator clevis permits the potentiometer to be withdrawn without opening the hydraulic system or disturbing other actuator components. Similarly, the attachment of the potentiometer shaft to the piston rod is adjustable by access through the forward clevis. Mounting of the potentiometer in this manner permits sealing of the potentiometer case and rod from the atmosphere, and it gives a direct attachment from the potentiometer shaft to the piston without additional links or offset arms, yet permits the piston to rotate and the potentiometer to be adjusted after installation. The cylinder bypass valves are spring-loaded to the bypass position, but are driven to the operate position by the presence of supply pressure. The bypass permits the piston rod to be moved in and out, without forming a hydraulic lock, as necessary for installation, checkout, and system calibration. Having the bypass self-contained in the actuator package avoids needless opening of the hydraulic system for adjustment of the actuator length.

The actuators are equipped with detachable scales, which are calibrated directly in degrees of engine displacement, for use in installation and checkout of the actuators. The actuators are also fitted with removable mechanical locks which hold the piston at midstroke for potentiometer calibration and actuator length adjustment. Since these locks are capable of holding full actuator force, they can also be used for blocked-load actuator testing and will hold the engines in place during vehicle shipping. The self-contained flushing valves in the actuators provide a manually-controlled bypass between the supply and return lines to the actuator. This bypass permits initial flushing and cleaning of the vehicle hydraulic system and lines without passing potentially contaminated fluid through the actuator filter and components. In actual use, the fluid used by the actuators is filtered by a full-flow stainless steel wire mesh filter. The filter element is removable from the actuator body and has a rating of 5 micron nominal and 15 micron absolute. The filter is supported at both ends to minimize vibration effects. A double o-ring seal is used to prevent contaminated fluid from bypassing the filter when high pressure surges are present. The actuators are fitted with bleed valves, immediately upstream and downstream of the filter and servovalve, which can be used to obtain fluid samples for contamination analysis. The actuators are designed to withstand the full range of vehicle and engine loads and environments.

HYDRAULIC SUPPLY

The fluid flow required by the actuators is provided, on demand, by the variable delivery main pump during engine operation and by the auxiliary motor-pump assembly during "dry" gimbaling. The main pump is directly coupled and flange-mounted to the turbopump and is driven by multiple reduction gearing. The auxiliary pump is driven by an electric motor that receives power from an external alternating current source.

The high pressure hydraulic fluid from the main or auxiliary pump outlet enters the maniford through similar check valves. These check valves prevent reverse flow through the pumps during filling and purging of the system. The main pump check valve is closed during "dry" gimbaling by the auxiliary pump pressure and prevents flow to system return through the main pump. The auxiliary pump check valve is closed by the main pump pressure during engine firing to prevent flow through the auxiliary pump.

The unfiltered hydraulic fluid from the pump (main or auxiliary, depending on mode of operation) enters a filter cavity in the maniforld and surrounds the wire mesh portion of the filter element. The fluid passes through small openings in the element, depositing the contaminant in the wire mesh, and flows into the accumulator. A differential pressure indicator is ported to both sides of the filter element. By visually indicating pressure drop across the filter, it indicates when contamination is sufficient to warrant replacing the filter element.

The hydraulic fluid returning from the auxiliary pump through the case drain line is routed to a second filter cavity in the manifold. The fluid flows through the element, depositing the contamination in the wire mesh, and into the reservoir.

The accumulator acts as a secondary source of fluid power, supplying instantaneous actuator demand flow in excess of the operating pump's capacity. The accumulator also serves as a pressure surge dampener and pump ripple suppressor.

Fluid from the accumulator flows to the actuators, returns to the manifold, and is ported into the reservoir. The reservoir is bootstrapped to the accumulator, maintaining return line pressurization and preventing pump inlet cavitation. The reservoir also stores the operating accumulator fluid during non-operating periods and provides volume for fluid expansion caused by increases in temperature.

During pump operation, system overpressure is relieved by a high pressure relief valve located in the manifold. Excess pressure is relieved from the high pressure to the low pressure portions of the system. A low pressure relief valve, located in the manifold, is ported to the low pressure side of the system. This valve relieves excess return line pressure to atmosphere during filling and purging. It is capped for flight operations.

During filling and purging, high and low pressure quick disconnect couplings are used to connect the Saturn hydraulic servicer to the hydraulic system. The couplings are located on the manifold with the inlet coupling ported to the filter cavity.

Bleeder valves are used for removing air trapped in the system and for taking fluid samples for contamination analysis. The valves are located at the main and auxiliary pumps, reservoir, and manifold. Three components are employed in the hydraulic system to furnish performance information. The fluid level indicator, attached to the reservoir piston, measures reservoir fluid level during filling of the system and in flight. The pressure transducer is mounted separately on the manifold; its purpose is to monitor pressure changes in the accumulator. The thermal switch is mounted directly into the low pressure side of the system on the manifold; it indicates when the fluid temperature has reached a predetermined value.

Accumulator-Reservoir and Manifold Assembly Description

The accumulator-reservoir and manifold assembly (Fig. 7) developed by Cadillac Gage Company consists of a high pressure, double wall, piston-type accumulator; a low pressure, piston-type reservoir; and a manifold for external connections and measuring devices. The accumulator receives high pressure fluid from the pump and serves as a dampener for pump discharge pulsations. It also provides a source of high pressure fluid for sudden actuator demands. The reservoir stores return fluid from the actuators and provides fluid to the pump inlet. The reservoir fluid is pressurized by a bootstrap piston. High pressure fluid acts on a small bootstrap piston, which is 1/60 of the area of the reservoir piston, and causes the reservoir fluid to be pressurized to approximately 370 kN/m^2. The manifold contains filling and bleeding ports, main and auxiliary pump ports, and actuator outlet and inlet ports.

Accumulator-Reservoir

Characteristics

Pressure	Accumulator	Reservoir
	MN/m^2	kN/m^2
1. GN_2 Precharge	11	--
2. Operating	22	370
3. Proof	35	2100
Fluid Volume at 21° C	cm^3	cm^3
1. Operating	557	770
2. Active	524	1700
3. Maximum	623	2000
4. Active Filling	0	1230
Weight (accumulator-reservoir and manifold assembly)		
1. Dry	16 kg	
2. Wet	17. 5 kg	

Auxiliary Pump Description

The auxiliary pump (Fig. 7) supplied by Vickers, Inc., is a single stage, fixed-angle, variable delivery, pressure-compensated unit used to supply fluid for ground checkout. The pump is driven by a 400 Hz motor. It has a normal operating speed of 1100 rad/s and delivers 189 cm^3/s at a pressure of 20 MN/m^2.

Nine pistons are attached by piston rods to the drive plate at a fixed angle to the cylinder block axis. As the cylinder block and drive plate rotate together, the angle of the plate causes the pistons to reciprocate, resulting in a pumping action. Piston travel remains the same regardless of the output flow which is regulated by the position of the valve plate.

The position of the valve plate is controlled by a rotary actuator. A torsion spring holds this actuator in the maximum displacement position until the actuator is rotated by an increasing pressure input from the compensator valve.

When the pump outlet pressure is sufficient to overcome the compensator spring force, the spool is moved to allow control pressure to reach the rotary actuator. The actuator then rotates the valve plate, decreasing the pump output. If a system demand causes the pump outlet pressure to decrease, the compensator spring moves the spool, connecting the actuator to the case drain and causing the spring to rotate the valve plate to a position of greater pump output.

Main Pump Description

The main pump (Fig. 7) is a two-stage, cam-actuated variable displacement unit supplied by American Brake Shoe Company. A two-stage design was selected to permit the main pump to pressurize the system even if inlet pressure is lost.

Rotation of the cylinder barrel assembly moves the seven dual-diameter pistons along the variable angle cam plate. This movement causes the pistons to reciprocate within the cylinder block. The intake strokes of the pistons cause the hydraulic fluid to be drawn through the inlet port into a fixed pintle and into the primary stage of the pump. Here the discharge strokes of the large diameter portion of the piston pressurize the fluid to approximately 0.7 MN/m^2 and deliver it along the outside of the pintle to the second stage. The primary stage supplies more fluid than the second stage requires. The excess fluid goes through a relief valve into the inlet port of the pump. In the second stage, the fluid is pressurized to approximately 22 MN/m^2 by the smaller diameter of the pistons. The high pressure fluid is discharged as the cylinder block rotates and aligns each piston with the high pressure outlet port. The pump outlet (delivered with actuator demand) is regulated by a compensator that maintains a pressure output which varies inversely with the flow.

In operation, system output pressure is ported to the compensating valve. This valve contains a spool held in a closed position by an adjustable spring load. When the system pressure exceeds the spring load, the spool shifts, admitting outlet pressure into the stroking piston cylinder. The stroking piston rotates the trunnioned hanger, to which the cam plate is attached, decreasing the cam plate angle and reducing output flow. When the system pressure force on the spool drops below the spring load (because of flow demands greater than the existing pump output), the spring shifts the spool in the opposite direction, porting the stroking cylinder to the case. The stroking piston retracts and allows pumping loads to rotate the trunnioned hanger. This causes an increase in the cam angle, which in turn increases the output flow.

OPERATIONAL PERFORMANCE

Telemetered data from past flights have indicated satisfactory operation. All temperature, level, pressure, vibration, and performance measurements remained within acceptable operating limits. Typical fluid temperature at ignition is 38° C but gradually decreases 6° C to 8° C during flight.

System and reservoir pressure and accumulator level have remained constant throughout flight, indicating that the average demand during flight is well within the pump capacity. Vibration levels of 2.5 g's rms have been measured on the actuator. Actuator excursions of 0.87 rad have been measured; rates of 0.22 rad/s and loads of 19 kN are typical.

During single engine testing with open tail, no difficulty was encountered with low temperatures. The standby time for a single engine firing was usually quite small. When the engines were mounted in the vehicle and partially shrouded, the hydraulic systems experienced low temperatures after propellant loading. The engine compartment was heated to maintain the hydraulic and engine system components at mild temperatures during engine starting. With the main pump mounted near the LOX suction duct, temperatures lower than -7° C were recorded. When the heating ducts were rerouted to provide a more direct coverage of the pump, the starting characteristics were improved and greatly reduced the starting surges.

Leakage is a chronic problem, and it appears that no absolute solution is immediately available. The conventional AM/MS fitting was initially employed but was far from adequate. Finally, this fitting evolved into the X-AN and subsequently into the MC. A marked reduction in the number of leaks was immediately noted. The MC fitting is a precision AN/MS that has been used in the modified form on SA-3 and SA-4.

On the Block II vehicles, the MC fitting will be used in much greater numbers. The leakage with MC fittings is barely perceptible when compared with AN fittings tested under the same environmental conditions. The MC fitting is not considered as the optimum answer to this problem; it is only an interim solution until a more refined and positive means of sealing can be found. MSFC, like everyone else in the industry, is looking for a connector that can attain the reliability and versatility necessary for space-age applications.

A major problem concerning the accumulator-reservoir assembly, during the early development period, was the internal leakage of nitrogen to the fluid side during an extended standby period. Initially the accumulator piston utilized a single o-ring to provide a dynamic seal between the nitrogen precharge and the high pressure supply fluid. During initial testing conducted at ambient temperatures, no leakage was noted; but leakage developed when these same units were placed on the static test vehicle and the propellants were loaded. This leakage was clearly indicated by the reservoir piston position potentiometer. The first flight was conducted using units with a single seal. Subsequently, a double seal was incorporated that did not completely prevent nitrogen leakage but vented it to the outside, keeping it out of the hydraulic fluid.

Although contamination was a major problem with Juno II, malfunctions from contaminations have been eliminated on Saturn with the "closed" system. To assure a system at a satisfactory clean level, detailed cheaning of all tubing, fittings, hoses, and components is required. Only clean and controlled hydraulic fluid is used. Cleaning and operating procedures are carefully adhered to and strictly enforced at all tests.

Early in the Juno II program, filters were a source of great concern in the hydraulic system, primarily because of the many unknown factors relative to servovalve failure; also the contamination sampling and analysis technique was not developed to a consistent and reliable method. Subsequent analysis of a number of filters by MSFC revealed a number of basic deficiencies.

Sintered bronze elements shed particles and some even fractured during operation.

Some of the early mesh elements were not properly supported and either generated contamination or permitted it to pass around the element. This latter reason was suspected as the cause of contamination bypassing the filter in the Juno II system and resulting in endless hours of purging and flushing. On some mesh elements that were exposed to vibration, pieces of the wire mesh were dislodged and found downstream of the element. Even though the mesh type filter did have some deficiencies, it did lend itself to improvement. The aerospace industry has almost universally accepted this type of element for flight applications. To improve the dirt holding capacity of the mesh type of filter, additional media is now being added to transform it into a "depth" type of filter.

S-IV HYDRAULIC SYSTEM

The S-IV stage is controlled in pitch, yaw, and roll by gimbaling its six engines (Fig. 12) through an excursion of $\pm 7 \times 10^{-2}$ rad at a maximum rate of 15.7×10^{-2} rad/s with a design load of 5.4 kN. Each engine also has an independent hydraulic system. The system (Fig. 13) is similar to the S-I stage system; however, a different component packaging concept is used. The S-IV system is unique in that the accumulator-reservoir and manifold assembly is packaged colinear with one actuator (Fig. 14) as an integrated unit and provides one of the attachment points for the actuator system. The other actuator is packaged colinear with a small reservoir. The hydraulic supply system includes a 31.5 cm^3/s variable delivery auxiliary motor-pump and a 63 cm^2/s fixed delivery main supply pump driven from an engine accessory pad as the RL-10 engine turbine.

The production model reservoir and the accumulator-reservoir and manifold assembly were designed and manufactured by Bertea Products. Moog Servocontrols designed and manufactured the servoactuator and packaged them with the accumulator-reservoir manifold and the reservoir. The accumulator-reservoir and manifold assembly incorporates the bootstrap reservoir, a filter, sequencing valves, relief valves, and the accumulator with pressure indication and position transducer for monitoring its performance.

As in the S-I system, the S-IV system actuators contain the DPF servovalve, a feedback potentiometer, a prefiltration bypass valve, a hydraulic filter, the hydraulic cylinder, a cylinder bypass valve, midstroke mechanical lock, a piston position indicator, fluid sampling valves, bleed and test ports, and adjustable rod end bearings. A piston differential pressure transducer is also provided to monitor the load.

The development of the Saturn I hydraulic systems has been truly a team effort. The knowledge gained from all previous efforts in this field by the research organizations and aerospace companies, as well as the component and subsystem manufacturers, contributed significantly in the excellent record Saturn has achieved.

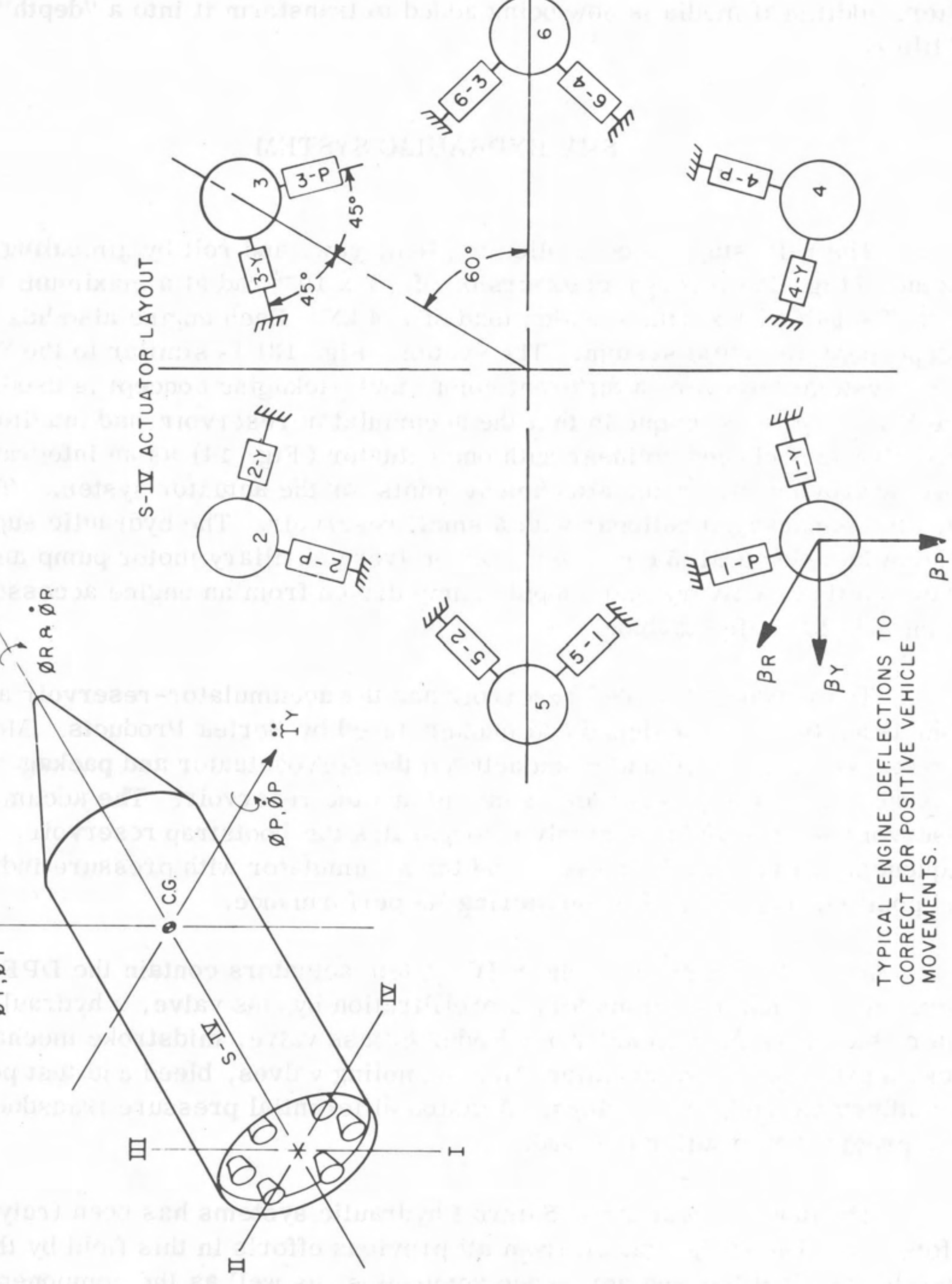
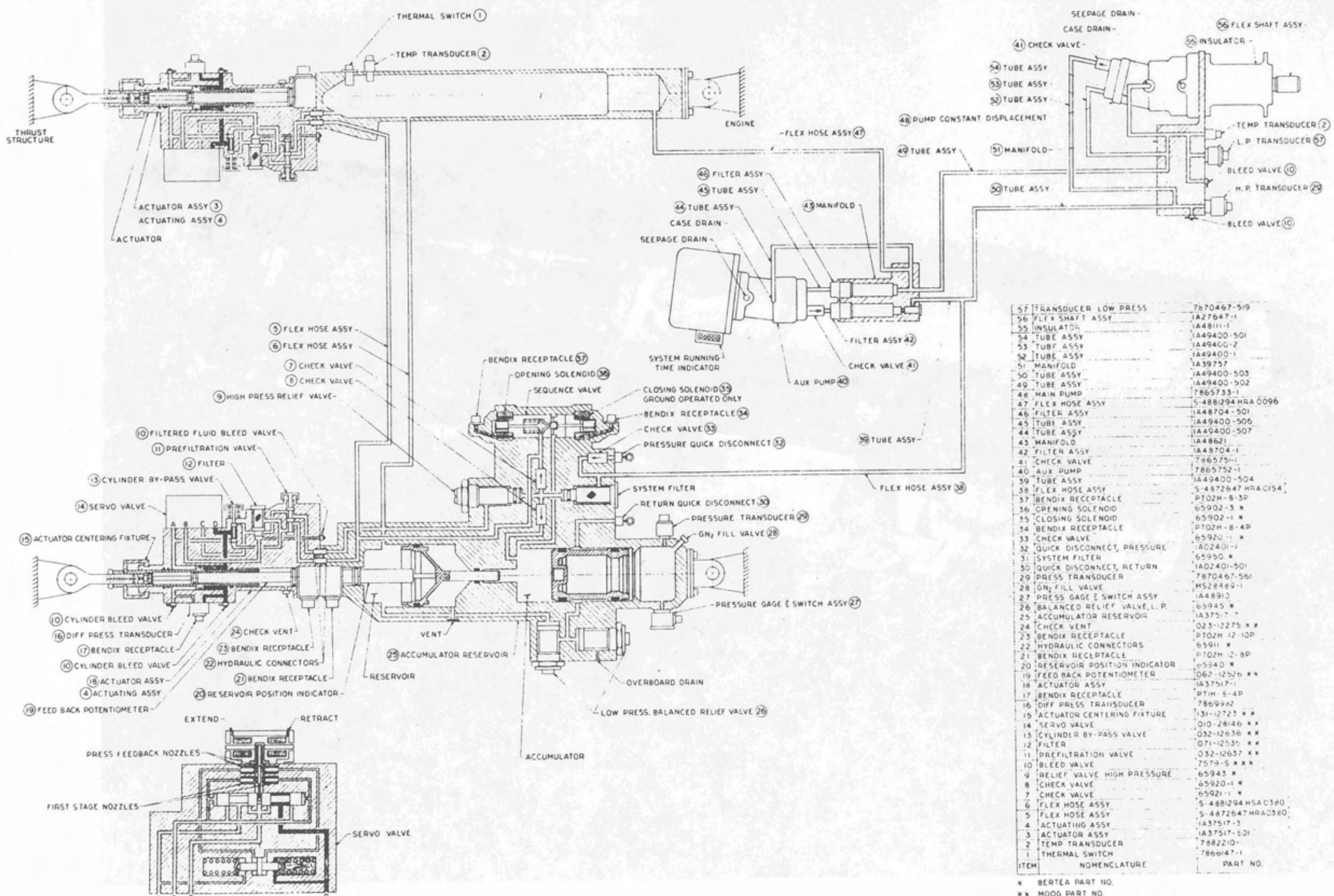


FIGURE 12. S-IV ENGINE CLUSTER

S-IV HYDRAULIC SYSTEM SCHEMATIC (REDESIGNED)



ITEM	NOMENCLATURE	PART NO.
57	TRANSDUCER LOW PRESS	7870467-519
56	FLEX SHAFT ASSY	1A27647-1
55	INSULATOR	1A48111-1
54	TUBE ASSY	1A49400-501
53	TUBE ASSY	1A49400-2
52	TUBE ASSY	1A49400-1
51	MANIFOLD	1A39757
50	TUBE ASSY	1A49400-503
49	TUBE ASSY	1A49400-502
48	MAIN PUMP	7865733-1
47	FLEX HOSE ASSY	S-4881294 HRA 0096
46	FILTER ASSY	1A48704-501
45	TUBE ASSY	1A49400-506
44	TUBE ASSY	1A49400-507
43	MANIFOLD	1A48621
42	FILTER ASSY	1A48704-1
41	CHECK VALVE	7865751-1
40	AUX PUMP	7865752-1
39	TUBE ASSY	1A49400-504
38	FLEX HOSE ASSY	S-4872647 HRA C154
37	BENDIX RECEPTACLE	PT02H-8-3P
36	OPENING SOLENOID	65902-3 *
35	CLOSING SOLENOID	65902-1 *
34	BENDIX RECEPTACLE	PT02H-8-4P
33	CHECK VALVE	65920-1 *
32	QUICK DISCONNECT, PRESSURE	1A02401-1
31	SYSTEM FILTER	65950 *
30	QUICK DISCONNECT, RETURN	1A02401-501
29	PRESS TRANSDUCER	7870467-561
28	GN_2 FILL VALVE	MS28889-1
27	PRESS GAGE & SWITCH ASSY	1A48910
26	BALANCED RELIEF VALVE, L.P.	65945 *
25	ACCUMULATOR RESERVOIR	1A375[illegible]
24	CHECK VENT	023-12275 **
23	BENDIX RECEPTACLE	PT02H-12-10P
22	HYDRAULIC CONNECTORS	65911 *
21	BENDIX RECEPTACLE	PT02H-12-8P
20	RESERVOIR POSITION INDICATOR	65940 *
19	FEED BACK POTENTIOMETER	062-12526 **
18	ACTUATOR ASSY	1A37517-1
17	BENDIX RECEPTACLE	PT1H-8-4P
16	DIFF PRESS TRANSDUCER	7869982
15	ACTUATOR CENTERING FIXTURE	131-12723 **
14	SERVO VALVE	010-28146 **
13	CYLINDER BY-PASS VALVE	032-12636 **
12	FILTER	071-12535 **
11	PREFILTRATION VALVE	032-12637 **
10	BLEED VALVE	7579-5 ***
9	RELIEF VALVE HIGH PRESSURE	65943 *
8	CHECK VALVE	65920-1 *
7	CHECK VALVE	65921-1 *
6	FLEX HOSE ASSY	S-4881294 HSA 0380
5	FLEX HOSE ASSY	S-4872647 HRA 0380
4	ACTUATING ASSY	1A37517-3
3	ACTUATOR ASSY	1A37517-501
2	TEMP TRANSDUCER	7882210-1
1	THERMAL SWITCH	7860147-1

* BERTEA PART NO.
** MOOG PART NO.
*** FLUID REGULATOR PART NO.

FIGURE 13. S-IV HYDRAULIC SYSTEM

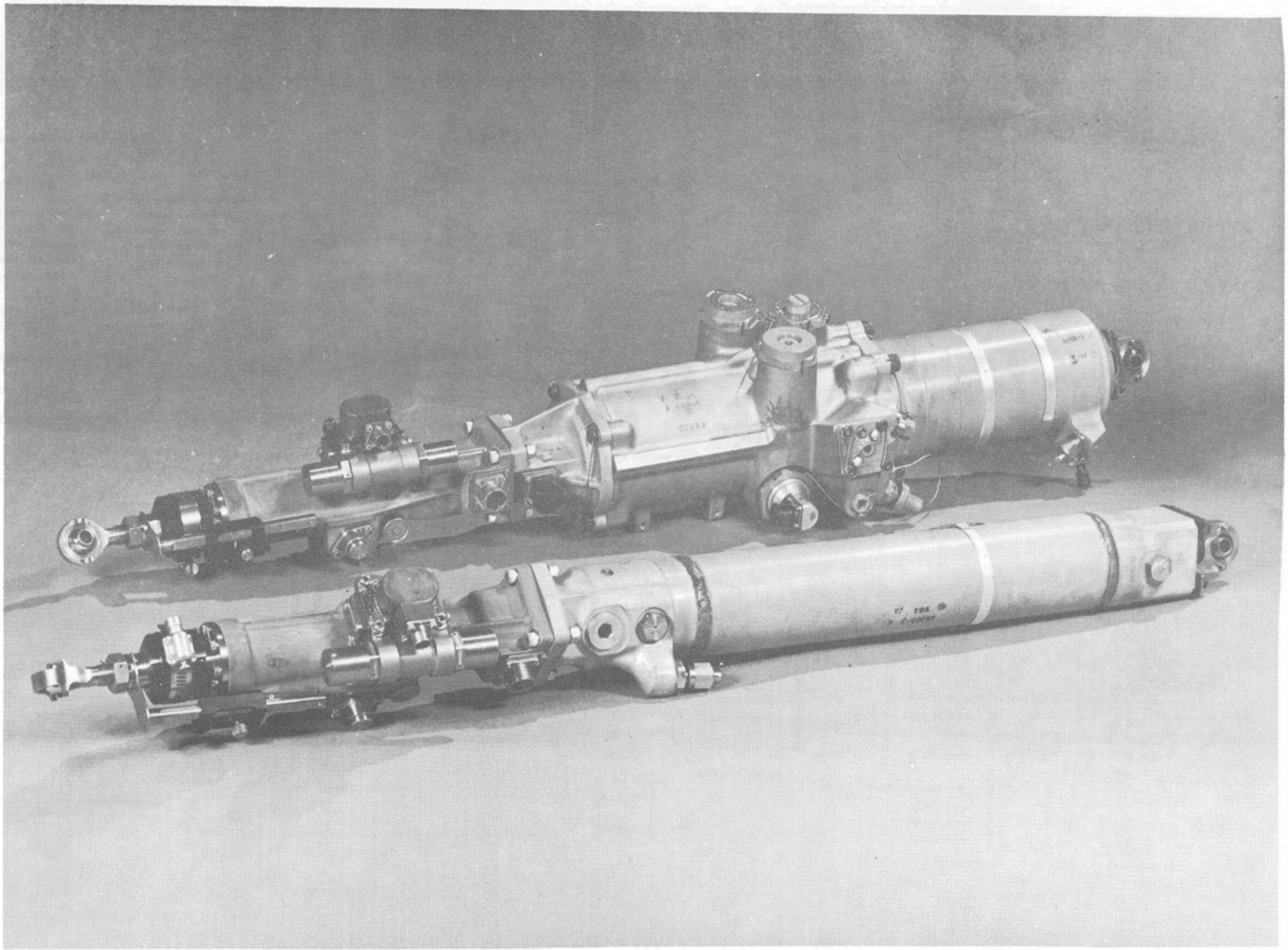


FIGURE 14. S-IV ACTUATORS

REFERENCES

1. Moore, F. B. and Brooks, Melvin, "Saturn Ascending Phase Guidance and Control Techniques," ARS Preprint 2458-62, Cleveland, Ohio, July 1962.

2. "Actuator, Thrust Vector Control, Specification for," June 17, 1963, George C. Marshall Space Flight Center.

3. Thayer, W. J., "Saturn S-I Servoactuators," unpublished paper.

4. Neiland, V. R., and Kalange, M. A., "Technical Advantages of Fluid Power," SAE No. 593A, October 8 - 12, 1962, Los Angeles, California.

5. Shoults, H. R., "Hydraulic System Description of the Saturn H-1 Rocket Engine (SA-5 Type)," October 25, 1962, George C. Marshall Space Flight Center, Internal Note No. 21.

6. Davis, Perry W., "Equations Describing the Engine Position Control System for the Saturn S-I Stage of the SA-5 Vehicle," March 19, 1963, George C. Marshall Space Flight Center, Internal Note M-ASTR-IN-63-7.

7. Scott, D. R., "Saturn SA-5 Navigation Specification, Accelerometer Angle of Attack Control," July 15, 1963, George C. Marshall Space Flight Center, Memorandum M-ASTR-N-496.

8. Koelle, H. H., <u>Handbook of Astronautical Engineering</u>, McGraw-Hill Book Company, New York, 1961, Section 14.137 and 14.163.

9. Noel, E. F. and Jones, C. B., "Technical Information Summary Concerning Saturn I Vehicle SA-5," George C. Marshall Space Flight Center, Memo No. M-ASTR-TSJ-75-63.

10. Hollinger, Richard H., "Operating Tests of the H-1 Engine Hydraulic System and Components Under Known Conditions of Fluid Contamination." Franklin Institute Report F-B1812-01, 1962.

11. Caudle, J. M. and Colbert, D. C., "Flight Control Computer for Saturn Space Vehicles," IEEE 1963 National Space Electronics Symposium, October 1 - 4, 1963, Miami Beach, Florida.

12. Riehl, W. A. and Hawkins, C. E., "Studies on Filters for Missile Hydraulic Systems," 21 June 1960, ABMA Report DSN-TR 1-60.

13. Smith, T. H., "Investigations Directed Toward Eliminating Leakage Problems on High Pressure Flared Tube Connections," January 18, 1962, George C. Marshall Space Flight Center, IN-P&VE-P-62-6.