DESIGN AND DEVELOPMENT
OF THE
REDUNDANT LAUNCHER STABILIZATION SYSTEM
FOR THE
ATLAS II LAUNCH VEHICLE

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ABSTRACT

The Launcher Stabilization System (LSS) is a pneumatic/hydraulic ground system used to support an Atlas launch vehicle prior to launch. This paper describes the redesign and development activity undertaken to achieve an LSS with increased load capability and a redundant hydraulic system for the Atlas II launch vehicle.

INTRODUCTION

General Dynamics started design of the original Atlas rocket in 1955. Since then, General Dynamics has refined and expanded the capabilities of this launch vehicle, which has proven itself to be a reliable and effective contributor to the United States space effort. However, without the support of the LSS, which is the subject of this paper, the Atlas would not have flown into history.

The Atlas vehicle has historically employed a controlled release launch system. During a launch sequence, the vehicle engines are ignited on the ground and verified for proper operation before the vehicle is released. The ground launcher system must support the vehicle in the prelaunch condition and release the vehicle at liftoff.

The Atlas ground launcher system (see Figure 1) consists of a large, welded steel structure that is firmly anchored to the launchpad along with pneumatic and hydraulic systems. The holddown and release (HD/R) system serves the primary role of supporting the vehicle prior to launch and releasing the vehicle when commanded. The vehicle is restrained from flight by two holddown pins mounted on the launcher structure acting in the Y-Y plane. The vehicle can freely rotate about this axis. At liftoff, the HD/R system retracts these pins and rotates the

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entire assembly away from the vehicle. The LSS is also part of the ground launcher system and provides an upward, balancing force called preload in the X-X plane to the outside structure of the vehicle to preclude it from toppling over. The LSS provides this force to the vehicle from the uppermost portion of the launcher structure called the A-frame.

The latest upgrade to the Atlas vehicle is the Atlas II. This new vehicle has the capability to launch heavier payloads but unfortunately not without an increase to vehicle weight and size. As a result, the previous design LSS could not provide a sufficient amount of preload nor counter the increased wind effects. Therefore, in order to retain the proven, controlled release method of launch, a redesign of the ground launcher system was necessary. The LSS was included as part of the overall ground launcher redesign. In addition, a design improvement to incorporate a redundant hydraulic system was implemented.

SYSTEM DESCRIPTION

The LSS is primarily required to support the vehicle from wind loads acting on the vehicle structure. It accomplishes this by sensing and adjusting the preload to apply more or less force on each side. The hydraulic system of the existing LSS did not use any pumps, valves, or complex control circuits and yet could precisely balance the vehicle in an upright position. However, if a leak were to develop, the LSS would malfunction and the vehicle would tilt unacceptably.

During ground checkout and prelaunch servicing, a mobile service tower (MST) is positioned to surround the vehicle on three sides and physically blocks any wind loads to the vehicle. With the MST in place, the LSS is backed up with adjustable shims that limit vehicle tilt and thereby provide an additional safety feature against tipping. Therefore, the time of critical need for the LSS is during the conditions of simulated and actual launches. In these situations, the MST is rolled away, the safety shims have been removed and the LSS must now counter the full effect of the wind loads. The response frequency of the LSS must react quicker than the vehicle tilt frequency.

During the launch sequence, there are several conditions that affect the LSS. They are thrust buildup (TBU), liftoff (LO), and thrust cutoff (TCO):

TBU = A condition in which the vehicle engines have been ignited but the release command has not been given, usually lasting about 2 seconds. There is a slight rise in vehicle position, resulting from the flexure of the vehicle and launcher mechanical structures, but the HD/R system is restraining the vehicle from flight. During TBU, the LSS must follow the rising vehicle and continue to exert a preload force to the vehicle while still maintaining balance.

LO = A condition following TBU, where the vehicle is released from the ground launcher system and allowed to lift off. As the vehicle rises off the launch pad, the LSS must follow the vehicle, and gradually reduce the preload to zero. The preload must be reduced before the vehicle physically separates from the LSS. This is necessary in order to avoid a sudden load transient being imparted to the vehicle or the payload (spacecraft).

TCO = A condition in which the launch has been aborted. This could occur at any time during the launch sequence, up to and including after the engines are ignited and the TBU condition exists. A TCO will result in a sudden, downward movement of the vehicle from a TBU condition. During a TCO, the LSS must follow the vehicle and stroke downward. The preload being applied to the vehicle should not change, and the LSS must still maintain vehicle balance against wind effects. After the initial downward movement, a series of diminishing rebound oscillations must be accommodated by the LSS.

GOALS/REQUIREMENTS

The primary goal of the LSS redesign was to develop a new system that could support the Atlas II vehicle. This support was necessary from the time of vehicle erection until liftoff and all intervening conditions, such as TBU, LO, and TCO. If possible, this new LSS was to be designed with growth capability in order to accommodate heavier payloads or launch vehicle weight increases as the Atlas program developed.

The secondary goal was to incorporate the redundancy feature into the hydraulic system of the new LSS. The redundancy feature must

provide a means to support the vehicle in the event of a hydraulic system failure, making the LSS single-failure-tolerant.

The LSS was designed to meet the following requirements:

Nominal preload (total)	462,600 N (104,000 lb)
Nominal preload (per side)	231,300 N (52,000 lb)
Wind moment capability	542,400 Nm (4.8x10^6 in.-lb)
Allowable vehicle tilt	+/-0.00349 Rad max (+/-0.2 deg)
TBU vehicle rise rate	152 mm/sec (6 in./sec)
LO vehicle rise rate	280 mm/sec (11 in./sec)
TCO vehicle descent rate	216 mm/sec (8.5 in./sec)
Response frequency	4 Hz minimum

DESIGN SELECTION/OPERATION

The previous design of the LSS used a passive hydraulic system that was pressurized with gaseous nitrogen (GN$_2$) but did not have a redundancy feature (see Figure 2A). An increase of the component sizes in this configuration would only meet the primary goal of supporting the vehicle. Therefore, the first task was to determine a system configuration with a redundancy feature that would support the vehicle. A series of trade studies were performed to evaluate several system configurations (see Figures 2B, 2C, and 2D). Each system was evaluated for basic capability, redundancy, simplicity, and cost. After the configurations were analytically modeled through computer simulation and static analysis, the proposed schematic of Figure 2D was selected.

The detailed designs of the cylinder and compensator were originally proposed as identical components. A common, tandem cylinder was chosen to minimize design efforts and costs. A tandem cylinder would be installed at each A-frame and two tandem cylinders joined together at their shafts would constitute a compensator. This design was further enhanced by placing the compensator cylinders adjacent to each other with a tilt beam to couple the shafts together (see Figure 2D). This resulted in a reduced size envelope for the compensator. However, this design was eventually discarded due to one major flaw. The compensator required many mechanical joints to enable the cylinders to be interconnected. These joints were a source of free play to the system and would have effectively degraded the performance of the LSS to the

point where it could not properly respond to vehicle motions. In addition, the cylinders required many flexible hoses to connect the system together. These hoses would have expanded with pressure, which would also have reduced system performance as well as provide potential failure points. As a result, the configuration was refined to a tandem cylinder and four-piston, common shaft compensator design (see Figure 3).

The operation of this new system is simple but elegant. GN_2 is used to pressurize all four pistons of the compensator. This pneumatic force acts on these pistons, which are, in turn, hydraulically connected to the tandem cylinders. Two separate hydraulic circuits are routed to each cylinder and the resultant forces from each hydraulic circuit are then joined to apply the balancing force to the vehicle. When a wind gust acts on the vehicle structure, the LSS responds immediately by countering the vehicle force with an increase in the LSS hydraulic system pressure on that side. This pressure increase is possible as the compensator simply shifts the amount of "effective" pneumatic force to the piston/circuits that demand it. This transfer of force also results in a reduction in the preload on the opposite side—a subtle feature that aids the LSS balancing efforts by minimizing the preload on the side that is actually aiding the wind in trying to tip the vehicle.

The redundancy of the LSS is possible through the use of the two individual hydraulic circuits on each side of the vehicle. If one of the circuits develops a leak, the remaining circuit pressure will essentially double to maintain support of the vehicle. The increase in hydraulic pressure is possible as the compensator pneumatic force shifts to the remaining circuit.

DESIGN DEVELOPMENT

A preliminary analysis showed that the oil volumes of the cylinder and the compensator circuits had to be virtually identical in order to function properly. This also meant that the tubing lines connecting the cylinders and compensator had to be exactly the same length. Furthermore, any amount of leakage would result in a volumetric imbalance and was undesirable, as the system would start to shift to the remaining circuit. To this end, all tubing pressure connections were either welded or were of the lipseal design to ensure leak-free reliability. In addition, the seals within the cylinder and compensator

had to be virtually zero leakage in order to eliminate them as a source of problems. However, in the event of leakage, an indicating device was desired to show when a circuit had failed. Several elaborate indicator designs were evaluated, but after development testing it was determined that monitoring the system circuit pressures was sufficient to evaluate the health of the system. If one circuit loses oil, the pressure drops off, and the adjoining circuit has a pressure increase.

A prototype test setup was constructed and preliminary development testing was conducted. This mock-up LSS was tested to verify the preliminary design assumptions of flow and pressure effects. Testing proved that the response time for the LSS would be in the order of 25 Hz. This is significantly faster than the predicted vehicle frequency of 4 Hz. The response capability of the LSS is comparable to an automotive brake system, where response is virtually immediate.

Component sizing was performed based upon the system load requirements. The LSS was required to provide a total preload of 462,600 N (104,000 lb) of force to the vehicle, 231,300 N (52,000 lb) per side. In order to provide this amount of force, each individual hydraulic circuit had to provide 115,650 N (26,000 lb). However, should one circuit fail, the remaining circuit then had to provide the full 231,300 N (52,000 lb) of force to maintain vehicle support. For this reason, each circuit had to be sized to permit a full load to be supported on only one cylinder. In effect, the system would be functioning only at half capacity at all times, but capable of switching to full capacity whenever necessary.

Preliminary sizing of the circuit was based upon a 31,000-kPa (4,500-lb/in.^{2}) operating system during a failed circuit condition. At normal operating pressures, the individual circuit pressures would have been 15,500 kPa (2,250 lb/in.^{2}). In order to achieve 115,650 N (26,000 lb) of force, the piston area had to be 74.59 cm^2 (11.56 in.^{2}). This would have necessitated a 97.54-mm (3.84-in.) diameter piston with an estimated cylinder external housing diameter of 178 mm (7 in.). This size would have readily fit into the installation; unfortunately, these design values did not offer any provision for growth capability. Thus, it was apparent that the system could be sized much larger.

The final sizing of the system was based upon a minimum piston area of 298 cm^2 (46.18 in.^{2}), operating at a circuit pressure of 3,875

kPa (563 lb/in.^{2}). This would provide the necessary preload of 462,600 N (104,000 lb):

$$298 \text{ cm}^2 \times 3875 \text{ kPa} \times 2 \text{ circuits} = 231,300 \text{ N per side}$$
$$(46.18 \text{ in.}^2 \times 563 \text{ lb/in.}^2 \times 2 \text{ circuits} = 52,000 \text{ lb per side})$$

In the event that a hydraulic circuit should fail, the circuit pressure would double to 7,750 kPa (1,125 lb/in.^{2}). However, the system was purposely designed to handle an operating pressure of 7,750 kPa (1,125 lb/in.^{2}) and a failed circuit pressure of 15,500 kPa (2,250 lb/in.^{2}). Essentially, the system was designed to support twice the loads anticipated, thus satisfying the growth capability feature. The piston area required a large external housing diameter of 356 mm (14 in.). Concurrent with the LSS redesign effort, the A-frame was being redesigned to accommodate greater loads. Once the cylinder size was determined, the A-frame was designed to fit the new cylinder parameters. In order to expedite the installation of the LSS components, full-scale wooden mock-ups of the cylinders and compensators were fabricated.

The system was not complete without the tubing to connect the cylinders and compensator. The tube size was determined by the oil flow rate (during TBU, LO, and TCO) and the maximum system pressure anticipated. The highest flow rate for the vehicle movement was LO, at 280 mm/sec (11 in./sec). For a piston area of 298 cm^2 (46.18 in.^{2}), the equivalent flow rate of oil was calculated to be approximately 492 liters/min (130 gal/min). Furthermore, the maximum system pressure anticipated for the LSS was 31,000 kPa (4,500 lb/in.^{2}), based upon a failed circuit pressure of 15,500 kPa (2,250 lb/in.^{2}) and the wind moment of 542,400 Nm (4.8x10^6 in.-lb) added to it. In order to support these flow rate and pressure values, a line size of 31.75 mm (1.25 in.) diameter, with a wall thickness of 3.05 mm (0.120 in.) was selected. The final line lengths of each circuit were approximately 12,190 mm (480 in.) between the cylinders and the compensator.

PROBLEMS ENCOUNTERED

This new LSS was a marked departure from the previous design and required further analysis and evaluation to qualify the concept before the design could be finalized. During the course of this evaluation,

several problems surfaced and were resolved. Problems concerning heat expansion effects, TCO effects, handling difficulties, volumetric imbalance effects, and tilt effects were addressed.

The LSS would be subjected to the effects of solar heating and rocket engine blast. The heat from these sources would heat the oil and cause a volumetric imbalance resulting in a vehicle tilt. The launchpad installation results in half of the LSS being exposed to direct sunlight, while the other half is in the shadow of other structures. An oil temperature gradient of as much as 37.78°C (100°F) was anticipated between the two sides of the vehicle, and the resultant expansion of oil on one side would have resulted in a slight (and undesired) tipping of the vehicle. As the vehicle lifts off, the entire launcher is exposed to temperatures as high as 2,760°C (5,000°F) for approximately 12 seconds. In certain cases, direct rocket blast impingement could occur. These conditions would affect the operation of the LSS or its longevity. In both cases, the problem was solved by shielding the system from exposure, thus eliminating any heating problem. The selected method was an ablative, silicone coating. This room temperature vulcanizing (RTV) coating provides shade from the sun as well as an insulating barrier from rocket blast temperatures.

If a TCO were to occur, the LSS would stroke downward with the vehicle. This motion would displace system fluid from the cylinder to the compensator and compression of the GN_2 within the compensator would result. This compression would result in a significant pressure rise in the pneumatic chambers of the compensator and would rebound back into the hydraulic circuit and raise pressures uncontrollably. To alleviate this problem, storage bottles (pressure vessels) were connected to the pneumatic chambers of the compensator (see Figure 3). The GN_2 would flow into these bottles, thereby avoiding any undue pressure. An additional benefit of these storage bottles was to provide an additional supply of GN_2 should the primary supply be shut off or fail.

The operation of the LSS requires preload drop-off prior to vehicle separation. A flow-rate analysis showed a need for an orifice to control the oil flow during a LO condition. At the same time, this orifice must not constrict the flow so that separation occurs during TBU. Further, the orifice would restrict oil flow in the event of a TCO and result in an over-pressurization of the hydraulic system. Therefore, the orifice must

"free flow" during the TCO event. In order to achieve this, an orifice check valve was added to each hydraulic circuit. The orifice was sized to control the flow of oil for LO (yet not permit vehicle separation during TBU) and the free-flowing feature of the check valve would allow TCO to occur without incurring the unwanted pressure rise.

During fabrication of the LSS components, the size and weight of the units made machining or even simply moving items very difficult. A fully assembled cylinder weighs about 2,225 N (500 lb) and a compensator weighs approximately 8,900 N (2,000 lb) (See Figures 4 and 5). Aside from the weight, the size and shape of the items also contributed to the difficulty. Specialized tooling was required to hold, lift, move, or even turn items. This was accomplished by using portable gantry-type cranes, special lifting adapters, and wheeled dollies.

The LSS compensator piston assembly consists of four individual pistons stacked together (see Figure 5). They are held together by a nut and bolt assembly on a central shaft. The entire assembly had to be mechanically preloaded to ensure there was no free play that would degrade system performance. Unfortunately, there was no way to physically apply the load and still torque the nut. The problem was solved by initially assembling the pistons, then pneumatically pressurizing the entire assembly. The nut was tightened, and when the pneumatic pressure was removed, the assembly was automatically preloaded and the four individual pistons acted as a single unit.

Volumetric imbalance of the LSS hydraulic system was evaluated. The imbalance could result from improper fill and bleed, where excess air is not removed from the system or if the chambers do not contain the same amount of oil, as in the case of a leak. During system testing, the LSS performance was evaluated with a fully bled system and with calibrated, increasing amounts of air (of as much as 3% of the total system) within the system. Test results indicated the volumetric imbalance with air in the system did not affect the system operation significantly. The reason for this was twofold. First, the air volume within the hydraulic system would compress when the system was pressurized, resulting in a very small volume, thus minimizing its effect to vertical displacement. Second, the relatively large piston area requires a large imbalance to cause any significant vertical displacement. Therefore, a minor imbalance can be readily accommodated. The vertical displacement is directly related to vehicle tilt, which is automatically minimized by the small effect of these imbalances. In all cases, the

+/-0.00349 Rad (+/-0.2 deg) of vehicle tilt was not exceeded.

During testing, a method of fill and bleed was employed that consisted of flowing pressurized oil through the LSS. After filling and bleeding, due to the length and complexity of bends and connections, there was no way to determine when the fill and bleed was complete and all the air had been removed. System-level tests did not indicate any presence of air within the system due to the previously mentioned large piston area. It was therefore necessary to accurately determine the amount of air within each circuit, and a method of verification was developed. The method employed was to connect a small test cylinder to the hydraulic system, fill the entire system with oil, and pressurize the opposite side of the piston within the small cylinder (see Figure 3). In this way, the air within the hydraulic system would compress and the test cylinder's piston would stroke the same amount as the air would compress. This method was tested and proved to be a very simple yet accurate means of checking the condition of the system.

LESSONS LEARNED

Design/Analysis Aspects
1. Discretely separate analysis activity was conducted to cross-check the overall design. One method was through computer simulation modeling, and the other was static analysis of the system operation.
2. Computer simulation modeling was performed to predict the performance of various system configurations without having to build physical prototypes of these systems to test their performance.
3. A single-component design was not the best option, despite the relative simplicity and cost savings, as the free play in the mechanical joints and expansion of the hoses results in system response degradation.
4. Maximization of performance was obtained through deliberate oversizing of the components while staying within fixed interface requirements.
5. Simple effects, such as solar heating, can have a dramatic effect on system operation and should not be overlooked.
6. System-level evaluation of the entire design must be conducted to preclude such problems as the TCO pneumatic system pressure rise and handling difficulties.

Test/Installation Aspects

1. System performance could be monitored through the use of pressure gauges rather than elaborate indicating devices.

2. The effects of volumetric imbalance and temperature effects were minimized by the large piston area, which resulted in a minimal vertical displacement.

3. After deciding on a system configuration, building a prototype test setup was beneficial in that crucial design assumptions of flow and pressure effects could be laboratory-tested to verify computer predictions.

4. An auxiliary cylinder can be used to verify the quality of a hydraulic system fill and bleed.

5. Fabrication of a full-scale wooden mock-up ensured a perfect fit of the components prior to the deliverable hardware arrival at the site installation.

CONCLUSION

The redesign project of the LSS was successfully completed in a very short period of time. The LSS was conceived in January 1989. The design/fabrication contract was awarded in June 1989, and fully fabricated and tested hardware was delivered by August 1990. In short, a program that should have taken two to three years to design and develop was conducted successfully in just over a year and a half.

At this time, four LSS units have been fabricated. Two units have been acceptance tested and delivered to Cape Canaveral Launch Complex 36A and 36B to support upcoming Atlas II launches. A third system is intended as a spare unit and the remaining system is currently undergoing extensive qualification testing.

ACKNOWLEDGEMENTS

ATLAS VEHICLE

PIVOT POINT

WIND LOADS

A-FRAME

LSS COMPENSATOR

LSS CYLINDER

MOBILE SERVICE TOWER

HOLD DOWN/RELEASE SYSTEM
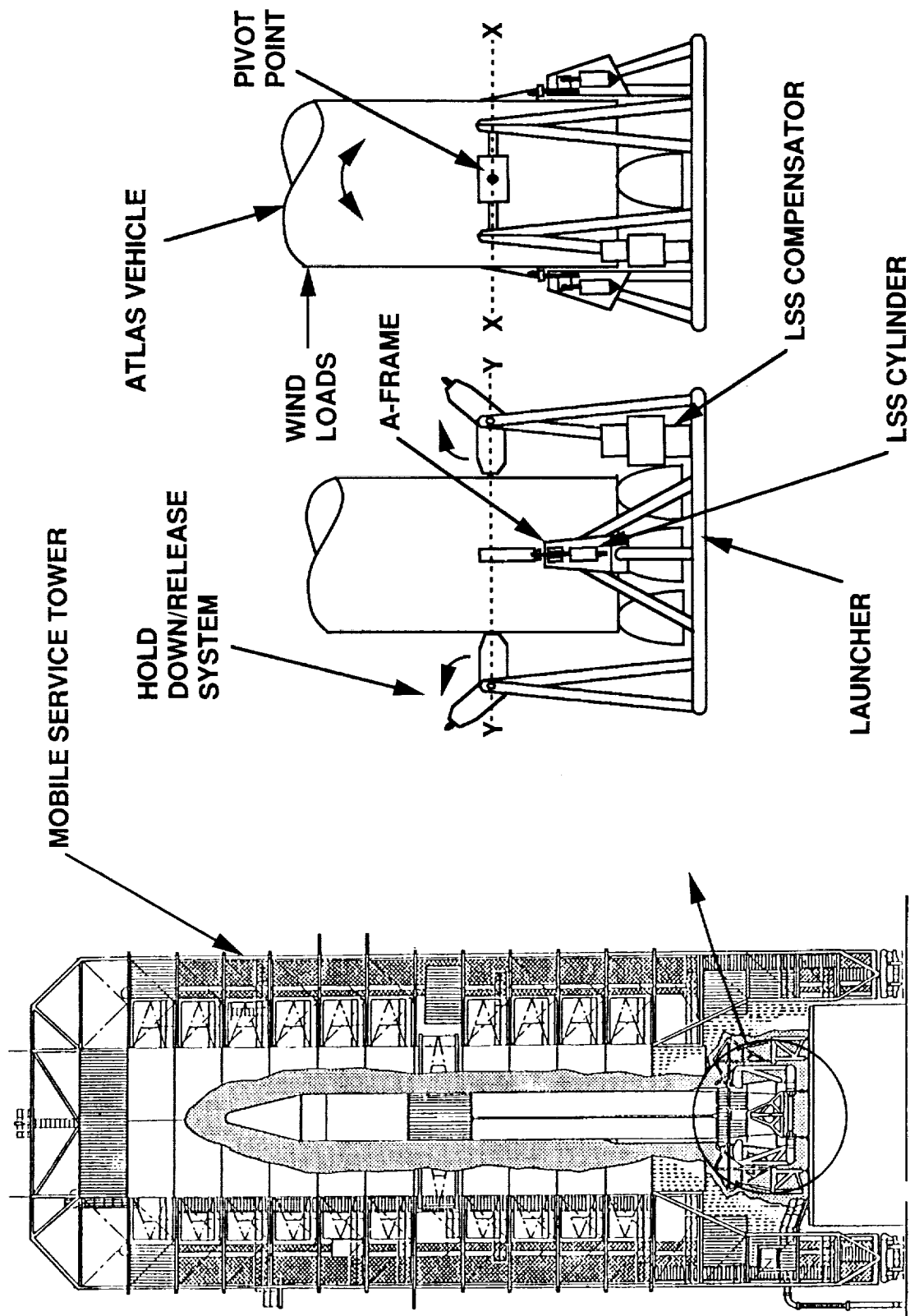
LAUNCHER

Figure 1. Atlas Ground Launcher System

ORIFICE

STABILIZATION
CYLINDERS

TEMPERATURE
COMPENSATORS

GN₂
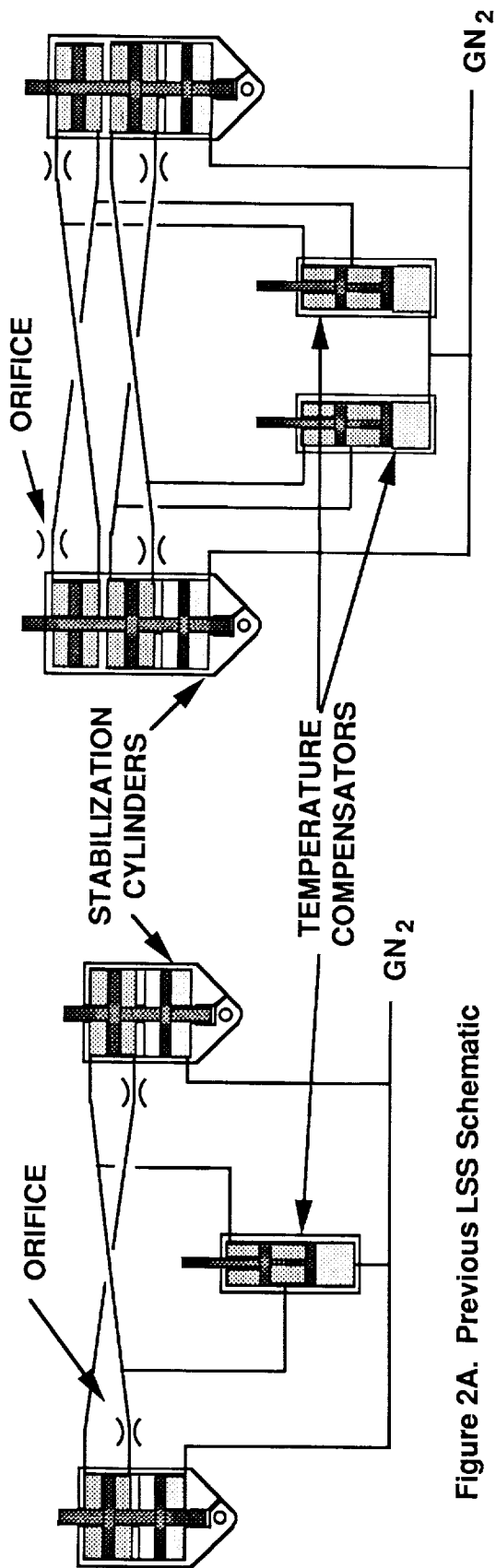
Figure 2A. Previous LSS Schematic

ORIFICE

GN₂

Figure 2B. Proposed LSS Schematic

STABILIZATION
CYLINDERS

ORIFICE

HYDRAULIC
COMPENSATORS

GN₂

Figure 2C. Proposed LSS Schematic

TILT BEAM

GN₂
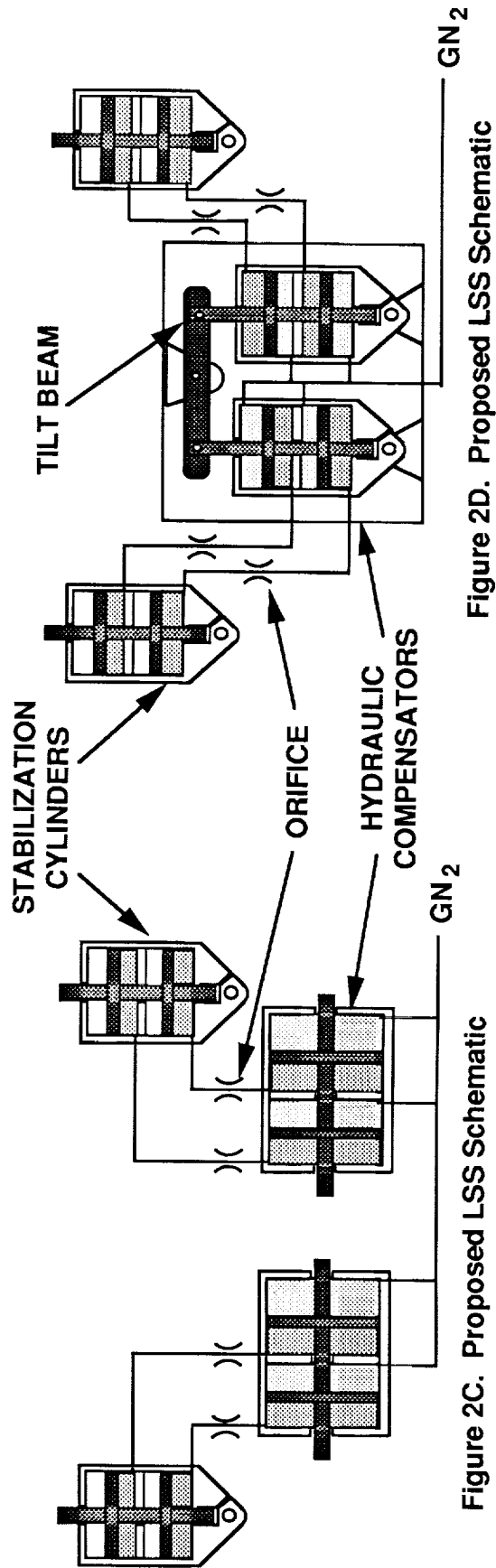
Figure 2D. Proposed LSS Schematic

Legend
= Hydraulic fluid volume
= Pneumatic volume (GN₂)
= Unpressurized air

57

STABILIZATION CYLINDER

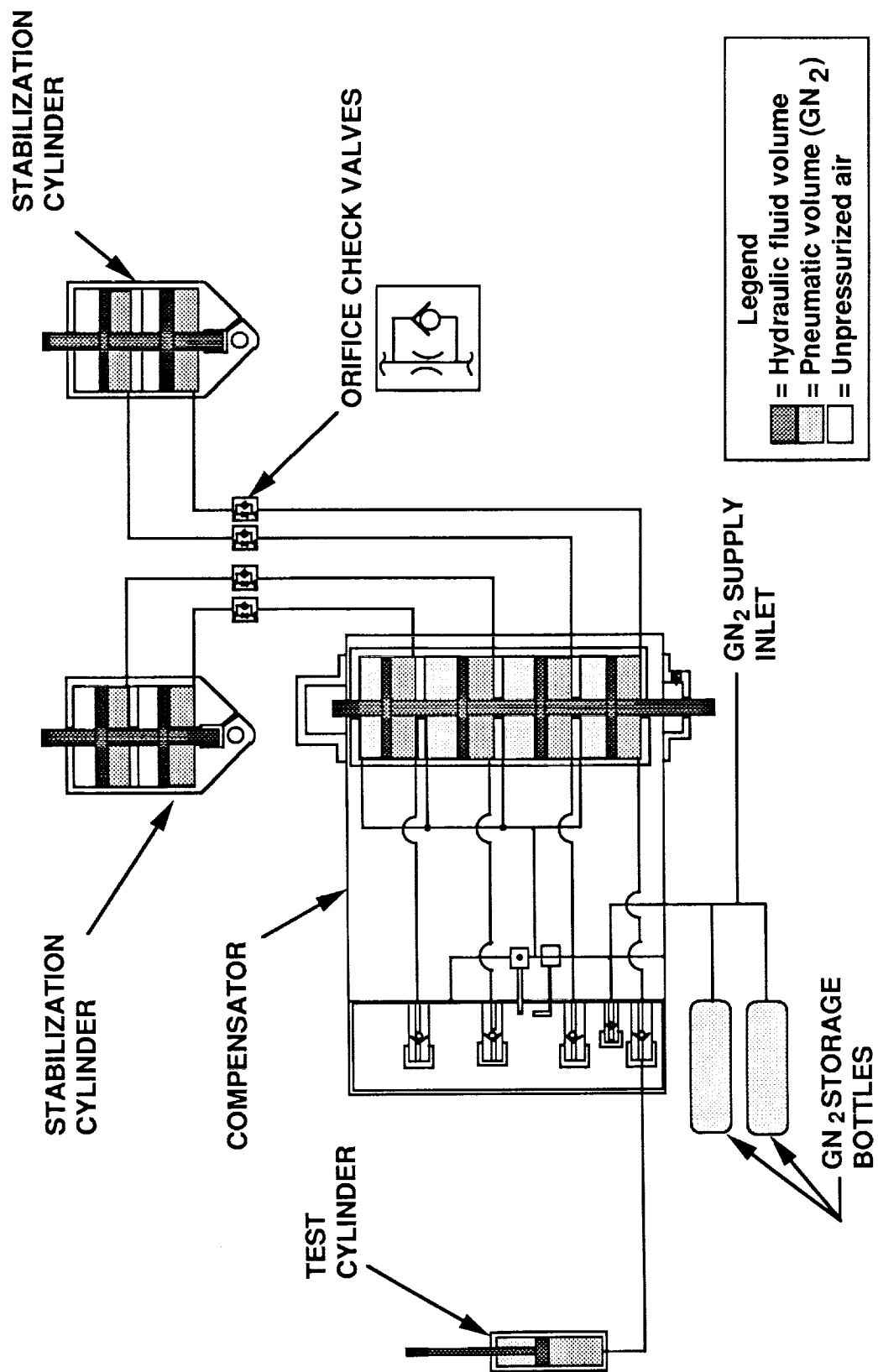ORIFICE CHECK VALVES

STABILIZATION CYLINDER

COMPENSATOR

TEST CYLINDER

GN$_2$ SUPPLY INLET

GN$_2$ STORAGE BOTTLES

Legend
■ = Hydraulic fluid volume
▨ = Pneumatic volume (GN$_2$)
□ = Unpressurized air

Figure 3. New LSS schematic

PISTON SHAFT

CYLINDER UPPER ATTACHMENT POINT

PISTON

HYDRAULIC PORTS

HYDRAULIC BLEED PORTS

CYLINDER LOWER ATTACHMENT POINT
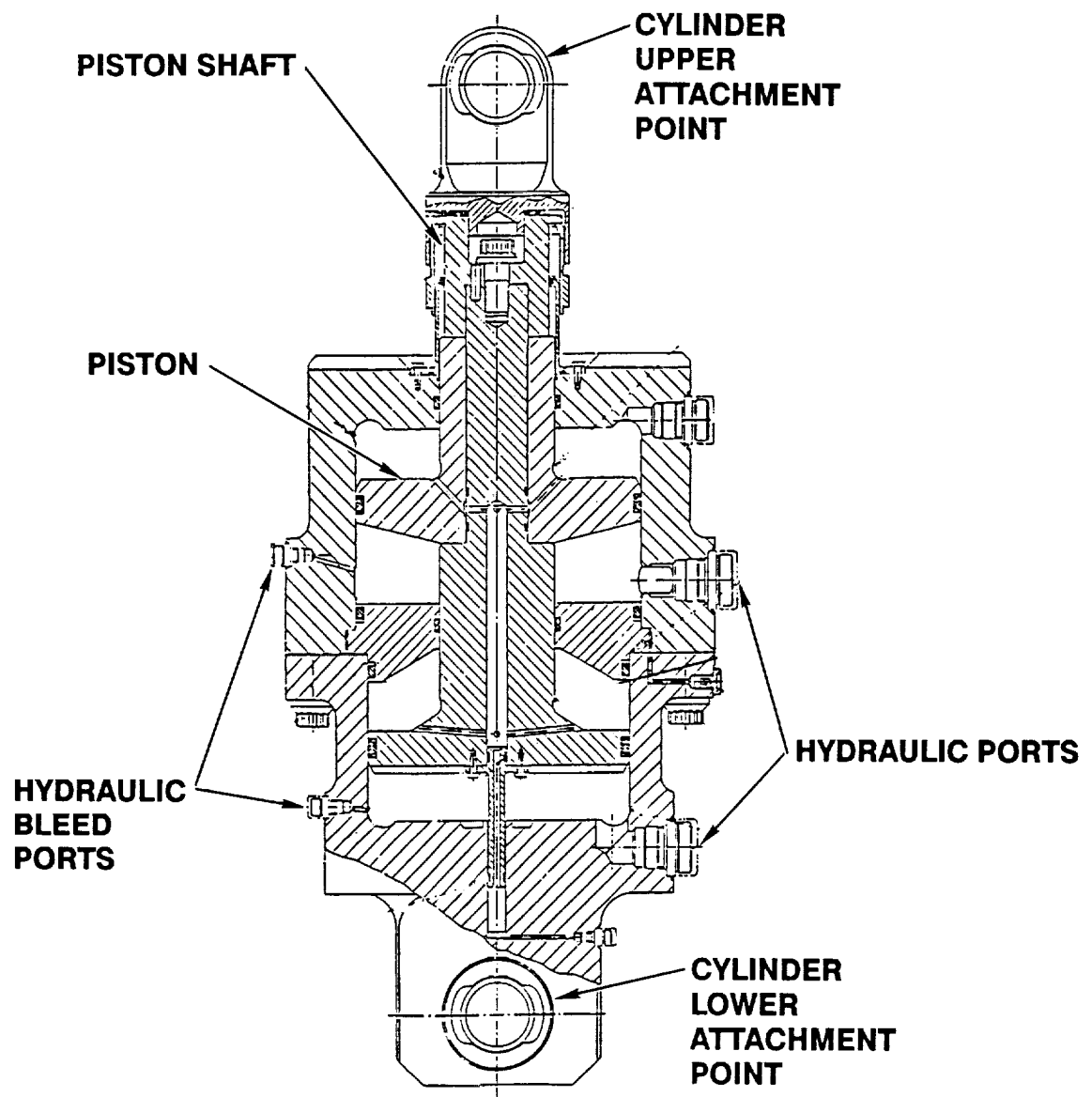
Figure 4. LSS cylinder assembly
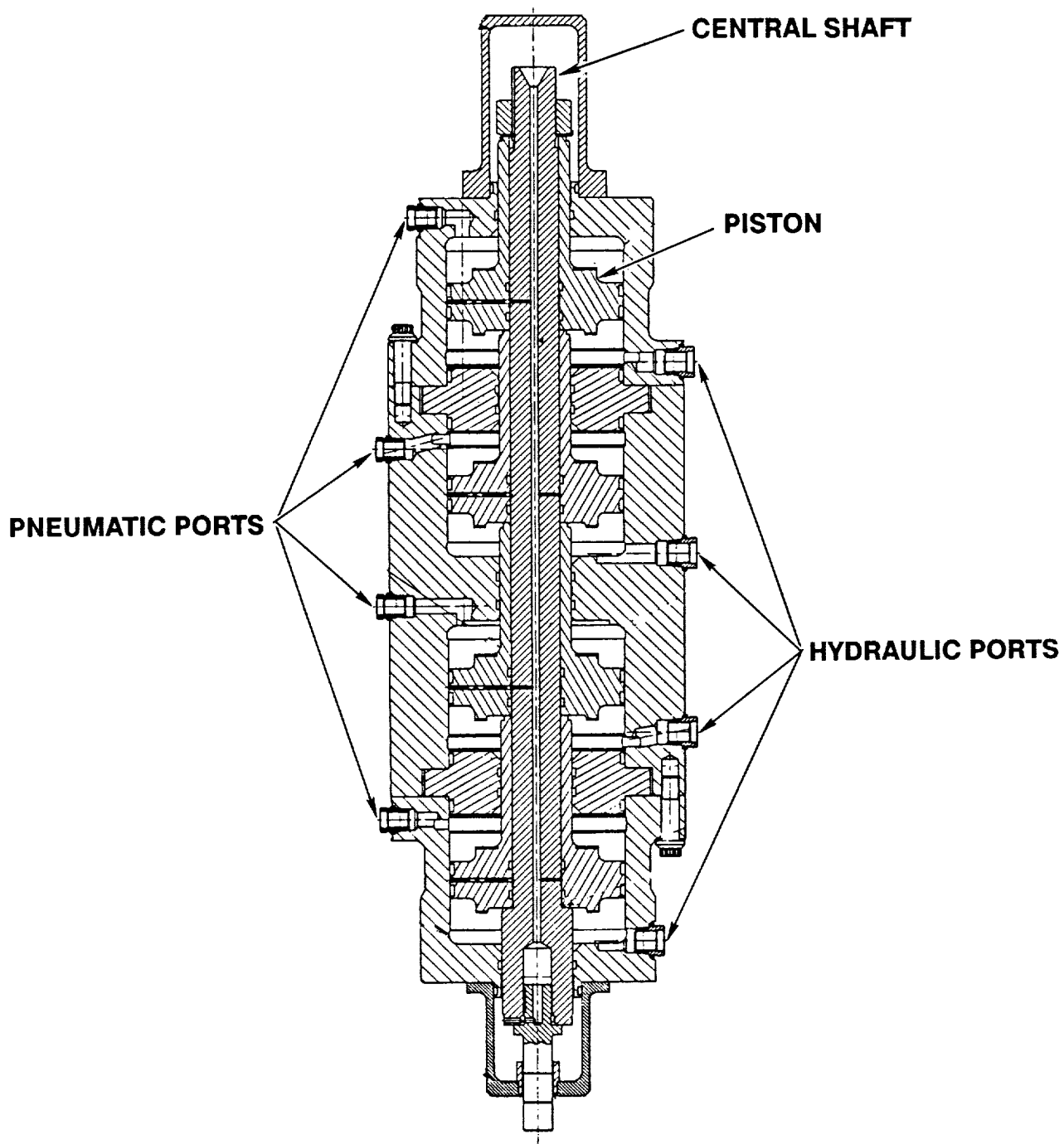
CENTRAL SHAFT

PISTON

PNEUMATIC PORTS

HYDRAULIC PORTS

Figure 5. LSS compensator assembly