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# HYDRAULIC MECHANISM TO LIMIT TORSIONAL LOADS BETWEEN THE IUS AND SPACE TRANSPORTATION SYSTEM ORBITER

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The Inertial Upper Stage (IUS) is a two-stage booster used by NASA and The Defense Department to insert payloads into geosynchronous orbit from low-Earth orbit. The hydraulic mechanism discussed in this paper was designed to perform a specific dynamic and static interface function within the Space Transportation System's Orbiter. This paper discusses requirements, configuration, and application of the hydraulic mechanism with emphasis on performance and methods of achieving zero external hydraulic leakage. The work was performed on Air Force Contract F04701-78-C-0040, Headquarters Space Division (AFSC).

## INTRODUCTION

The mechanisms discussed herein were designed to function in the natural environments of space and induced environments associated with orbiter boost, payload deployment, and reentry of the space shuttle Orbiter. The environments are severe and require design solutions unavailable in normal industrial applications.

The IUS interface with the Orbiter cargo bay is the IUS Airborne Support Equipment (ASE). The ASE structure consists of (1) an aft support frame that provides support for IUS X, Y, Z, Mx, and Mz loads, and (2) a forward support frame that provides support for IUS Y and Z loads during boost. A keel pin between the forward ASE frame and IUS carries the Y loads. The aft frame pivots during deployment to elevate the IUS to a position to clear the Orbiter cargo bay. The hydraulic mechanism is an integral part of the forward ASE frame. Figure 1 shows the Orbiter and its relationship to the forward and aft ASE frames.

During Orbiter boost to low-Earth orbit, the hydraulic load-leveler mechanism minimizes torsional loads applied to the IUS and absorbs part of the dynamic energy being transmitted to it. If attempts to deploy the payload are unsuccessful and an ASE Payload Retention Latch Actuator (PRLA) motor failure occurs during abort restow, the hydraulic mechanism can displace one actuator up and the opposite actuator down to restow the payload and maintain IUS to Orbiter alignment.

The major problem and drawback of using hydraulic systems for space applications is potential hydraulic oil leakage and contamination. Considerable test experience has established a design application, assembly technique, and screening test program that meet zero external leakage requirements.

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## DESIGN REQUIREMENTS

Natural environments and Orbiter-induced environments present design requirements for all mechanisms used in the Orbiter, such as:

a. Complete a minimum of 100 IUS shuttle flights over a period of approximately 10 years.
b. Operate in zero gravity.
c. Survive temperatures as low as -59 °C (-75 °F).
d. Have an operating temperature range of -23 to + 52 °C (-10 to +125 °F).
e. Be contained within the allowable dynamic envelope.
f. Use materials and finishes that will not outgas.

Specific hydraulic mechanism design requirements are:

a. Allow zero external leakage of hydraulic fluid.
b. Limit maximum operating pressure to 3000 lbf/in.$^2$.
c. Have an active life expectancy of 50,000 cycles without refurbishment.
d. Limit differential load due to friction to less than 450 lb.
e. Provide a damping coefficient of 225±50 lb-sec$^2$/in.$^2$, and limit load-leveler piston velocity to 14.0 in./sec and the maximum differential load between the two load levelers to 14,000 lb.

## HYDRAULIC LOAD-LEVELER MECHANISM

The load leveler mechanism on the forward ASE (fig. 2) is a closed loop hydraulic system that limits torsional loads applied to the IUS while providing determinant support in static and dynamic environments. The forward end of the IUS and spacecraft is supported and restrained by double-acting pistons within the two load-leveler actuators. The upper chambers of each actuator are plumbed together and the lower chambers are plumbed together. Any upward displacement of one actuator is matched by an equal but opposite deflection of the opposite actuator to maintain positive engagement of the ASE keel pin in the IUS socket. In the event the left hand and right hand PRLAs do not open simultaneous during deployment or close simultaneously during abort restow, mechanical stroke limiters on the load-leveler actuator rods limit the maximum possible height differential between the two PRLAs to approximately 1.25 in. A hydraulic accumulator, with a metal bellows type diaphragm, is connected to the upper chamber tubing to accommodate thermal expansion and contraction of the hydraulic fluid. The pressurant side of the accumulator is pressurized to 1080 lbf/in.$^2$ (nominal) with gaseous nitrogen.

The hydraulic tubing connecting the lower chambers of the load-levelers has two orifices installed to provide a controlled damping coefficient of the total system. The damping coefficient requirement of 225 lb-sec$^2$/in.$^2$ nominal optimizes a trade off between load-leveler system travel and dynamic loads transmitted to the spacecraft. A reduction in damping coefficient reduces transmitted loads but exceeds total load-leveler piston travel available. Increasing the damping coefficient results in higher dynamic loads being transmitted to the spacecraft.

## HYDRAULIC MECHANISM DAMPING

Adjustment of the load-leveler mechanism damping coefficient and verification that resultant maximum differential loads and piston velocities were within required limits were demonstrated by test. A summary of the results is discussed herein.

The governing equation for summation of load-leveler system forces is

$$F_D = F_F \frac{|V|}{V} + CV|V| + MA$$

where

$F_D$ differential force, lb; 14,000 lb maximum

$F_F$ static friction force, lb; 450 lb maximum (acceptance test data show nominal friction force of 300 lb)

$V$ piston velocity, in./sec; 14.0 in./sec maximum

$C$ damping coefficient, lb-sec$^2$/in.$^2$; 225±50 lb-sec$^2$/in.$^2$

$A$ piston acceleration, in./sec$^2$

$M$ effective load leveler system mass, lb mass

$|\,|$ absolute value

The effective load-leveler mechanism mass was calculated by externally driving the load-leveler system at varying velocities and measuring the applied force and the acceleration of the driven piston relative to its housing. Measurements were taken at the time of maximum piston acceleration. The tests were conducted with no inline orifices to minimize damping. The results are summarized in table I. As shown, the average effective mass of the load-leveler mechanism was 10,980 lb mass; within 2% of the predicted value of 10,750 lb mass.

The load-leveler mechanism was tested with several orifices to establish the required damping coefficient. Test data for the production configuration are summarized in table II. The damping coefficient is calculated from measurements of the force applied to the driven piston and the velocity of the driven piston relative to the housing, calculated by differentiating the relative displacement curves. Again, the measurements were taken at maximum piston velocity. As shown, the average damping coefficient is 227 lb-sec$^2$/in.$^2$. The data scatter fall within the 225±50 lb-sec$^2$/in.$^2$ requirement.

The governing equation for the load leveler system forces becomes:

$$F_D = 300 \frac{|V|}{V} + 227\, V|V| + 10{,}980\, A$$

Figure 3 presents a comparison of the analytical prediction of system performance and actual test results. The analytical prediction was based on the assumption that the relative velocity between the piston and housing is sinusoidal and the maximum velocity amplitude is 5.8 in./sec (table II). As shown, predicted and test results are consistent, and maximum differential forces are approximately 10,000 lb.

Moreover, increasing the static friction force, $F_F$, and damping coefficient, C, to their maximum design values increases the maximum calculated value of $F_D$ less than 20%. The resultant maximum calculated value for the differential force satisfies the 14,000 lb maximum design requirement.

## ZERO EXTERNAL LEAKAGE

The ASE hydraulic load-leveler mechanism meets all system design requirements. The hydraulic mechanism was selected during the design phase because it provided the desired stiffness characteristics and load paths for a mechanical system, and met limited envelope requirements. In addition, flow limiter (orifices) were easily changed-out during system tests to obtain desired damping coefficients.

The major drawback of a hydraulic system for space application is external leakage. Small amounts of hydraulic oil discharged into a vacuum rapidly expand and are attracted to cold surrounding structures. A film of oil on critical hardware, such as star scanner, can potentially impact an entire mission.

The ASE hydraulic system was designed to limit the number of potential leakage paths. All tubing is welded, with seals (redundant) used only at tubing to load-leveler joints. The load leveler, shown in cross section in figure 4, includes redundant, primary and secondary, seals at all static and dynamic seal locations. In addition, storage cavities for primary seal leakage are used at all dynamic seal positions. Primary seal leakage of one drop per day at dynamic seals is allowed. Storage cavities are periodically drained through vent vent port plugs.

The O-rings used in the load leveler are fluorosilicone with a 70 to 80 durometer. Fluorosilicone was selected because of the extreme low-temperature survival requirements (-59 °C). The production configuration had nominal O-ring squeezes of 10% for dynamic seals and 15% for static seals consistent with standard O-ring design practices.

Following delivery of all production units, numerous leakage problems occurred during ground storage at ambient conditions. Intensive review by BAC* Engineering, Customer Representatives, and Parker Seal** design engineers, in addition to an extensive test program, identified five key elements of the design, which, if controlled carefully, result in zero external leakage.

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TABLE I. - TEST DATA USED TO DETERMINE EFFECTIVE LOAD-LEVELER MECHANISM MASS

| Forcing function | Force at V = 0, kips | Relative acceleration*, g's | Effective mass, LMB x $10^3$ (M = F/A) |
|---|---|---|---|
| 2 kip at 4 Hz | 1.55 | 0.15 | 10.3 |
| 4 kip at 4 Hz | 3.63 | .39 | 9.3 |
| 6 kip at 4 Hz | 5.30 | .51 | 10.4 |
| 8 kip at 4 Hz | 6.9 | .6 | 11.5 |
| 10 kip at 4 Hz | 9.1 | .79 | 11.5 |
| 12 kip at 4 Hz | 10.25 | .90 | 11.4 |
| 14 kip at 4 Hz | 11.5 | .95 | 12.1 |
| 6 kip at 6 Hz | 5.5 | .46 | 12.0 |
| 8 kip at 6 Hz | 7.3 | .68 | 10.7 |
| 10 kip at 6 Hz | 9.0 | .83 | 10.8 |
| 4 kip at 7 Hz | 3.5 | .35 | 10.0 |
| 8 kip at 7 Hz | 7.0 | .60 | 11.7 |

*Relative acceleration of load-leveler piston to housing. Results (based on geometry and fluid mass considerations): (1) $M_{AVG}$ = 10,980 lb mass; (2) Predicted $M_{AVG}$ = 10,759 lb mass.

TABLE II. - TEST DATA USED TO DETERMINE DAMPING COEFFICIENT OF LOAD-LEVELER MECHANISM

| Forcing function | Force at maximum velocity, F, kips | $V_{MAX}$*, in./sec | Damping coefficient, lb(in./sec)$^2$ |
|---|---|---|---|
| 2 kip at 4 Hz | 1.2 | 2.0 | 225 |
| 4 kip at 4 Hz | 2.3 | 3.2 | 195 |
| 6 kip at 4 Hz | 4.5 | 4.3 | 227 |
| 8 kip at 4 Hz | 6.2 | 5.2 | 218 |
| 10 kip at 4 Hz | 8.6 | 5.8 | 247 |
| 2 kip at 1 Hz | 1.7 | 2.4 | 243 |
| 4 kip at 1 Hz | 3.7 | 3.8 | 235 |

*$V_{MAX}$ obtained by differentiating deflection data. Results:
(1) $C = (F - F_F)/V_{MAX}^2 F_F$ = 300 lb**; (2) $C_{AVG}$ = 227 lb-sec$^2$/in.$^2$;
(3) Tolerance $C = (1 + 0.20)C_{AVG}$.
**Average of friction force measured for each ASE at ambient (launch) conditions.

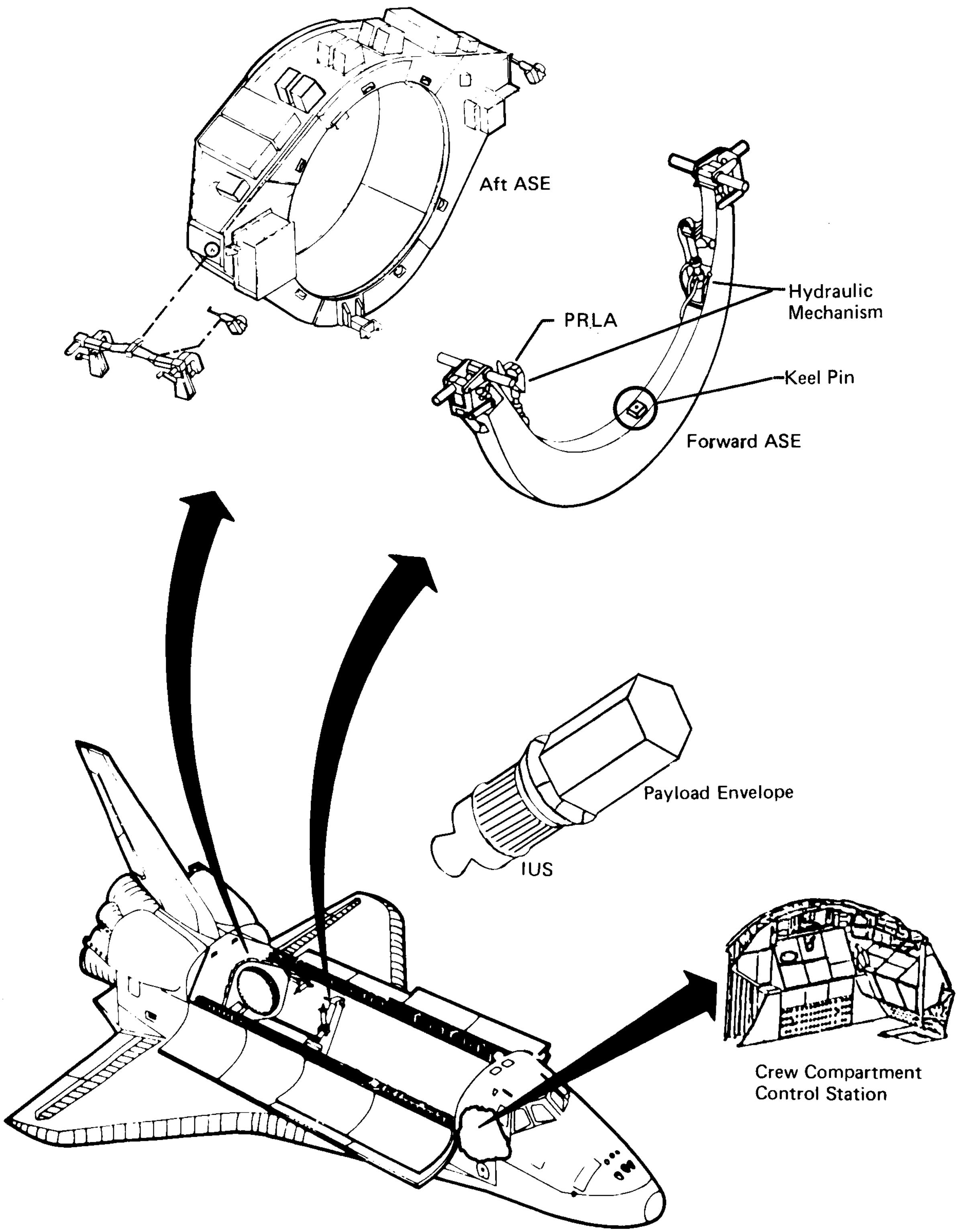



Figure 1. - IUS Airborne Support Equipment.

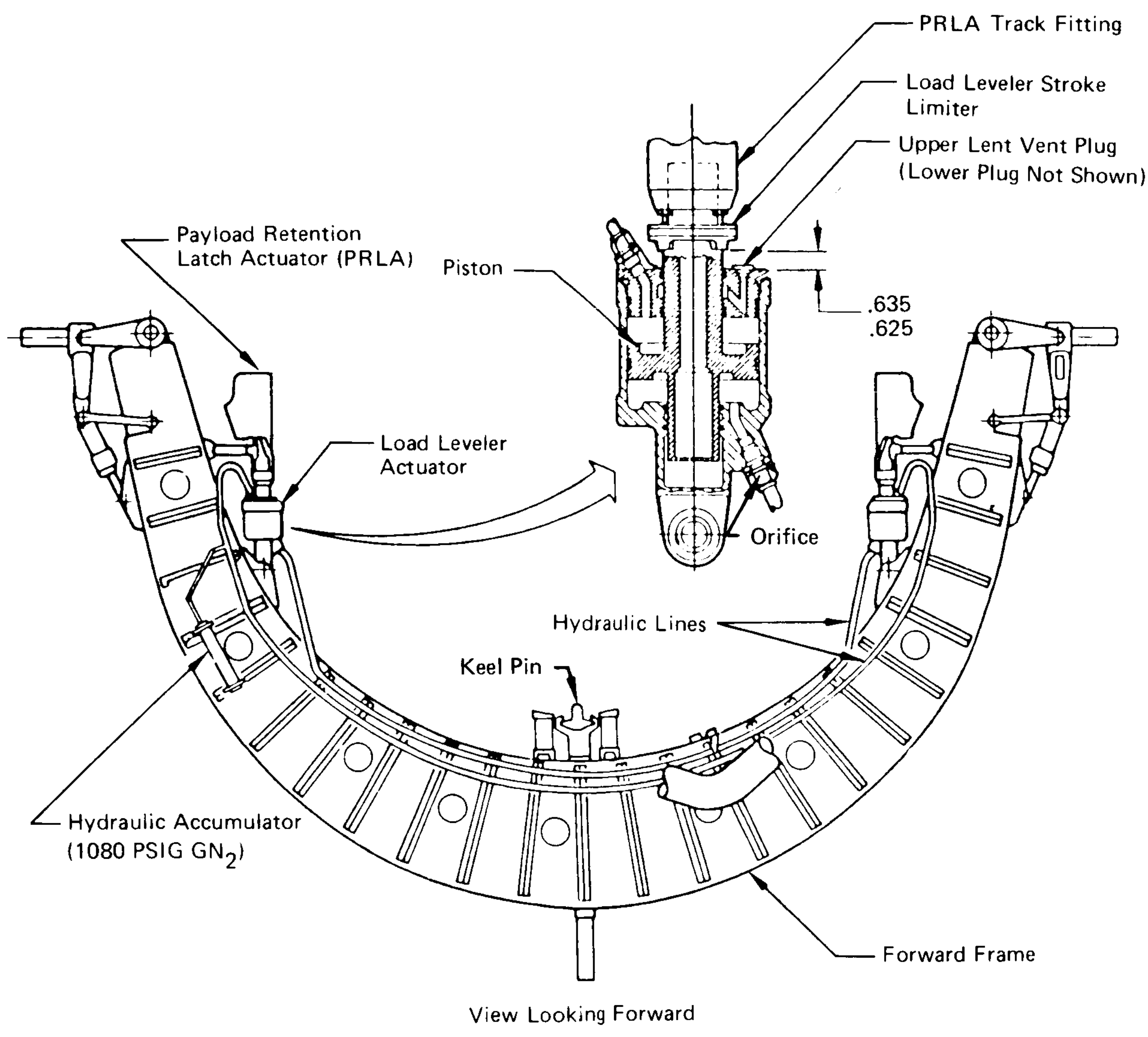



Figure 2. - Forward ASE load-leveler system.

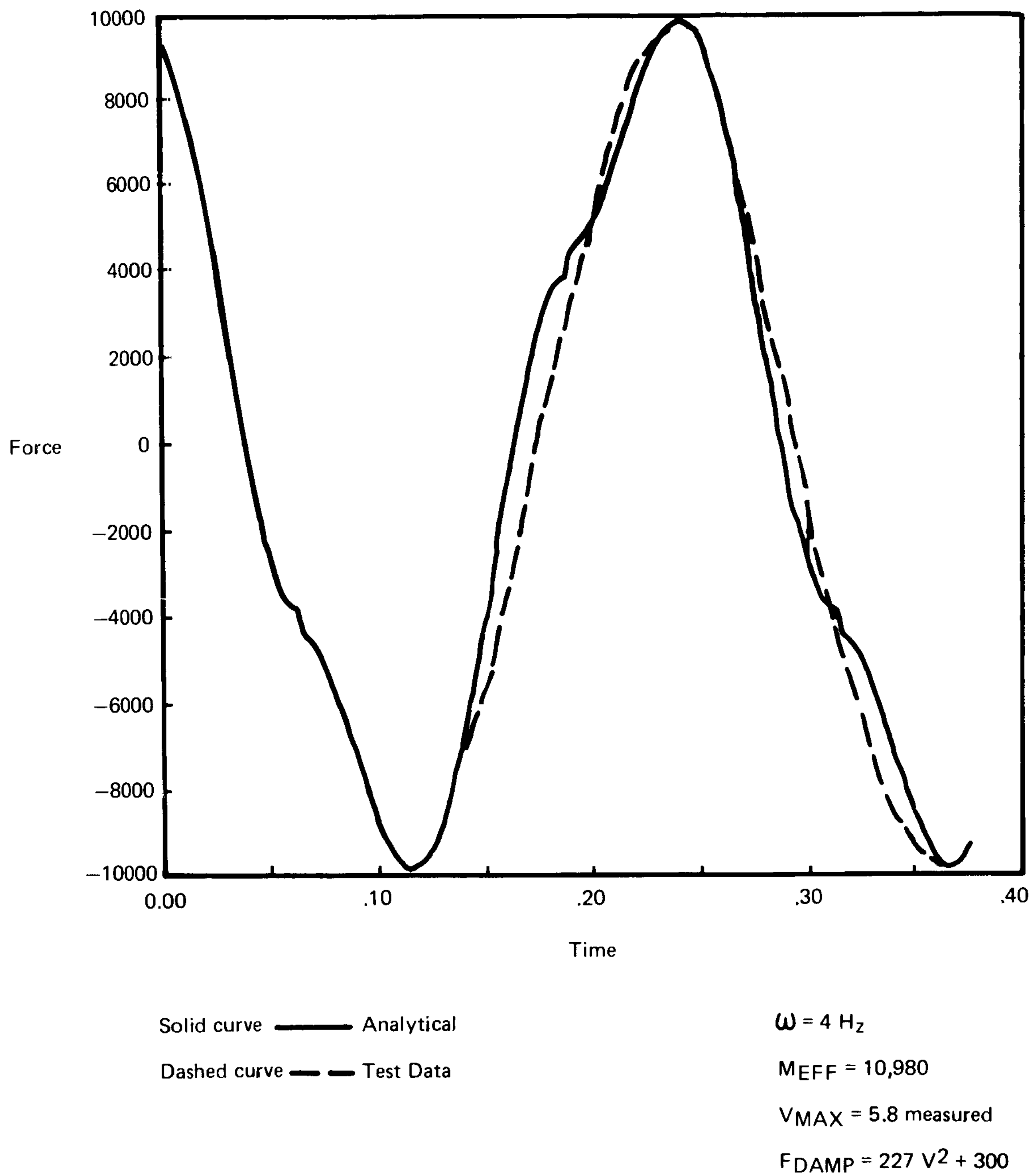
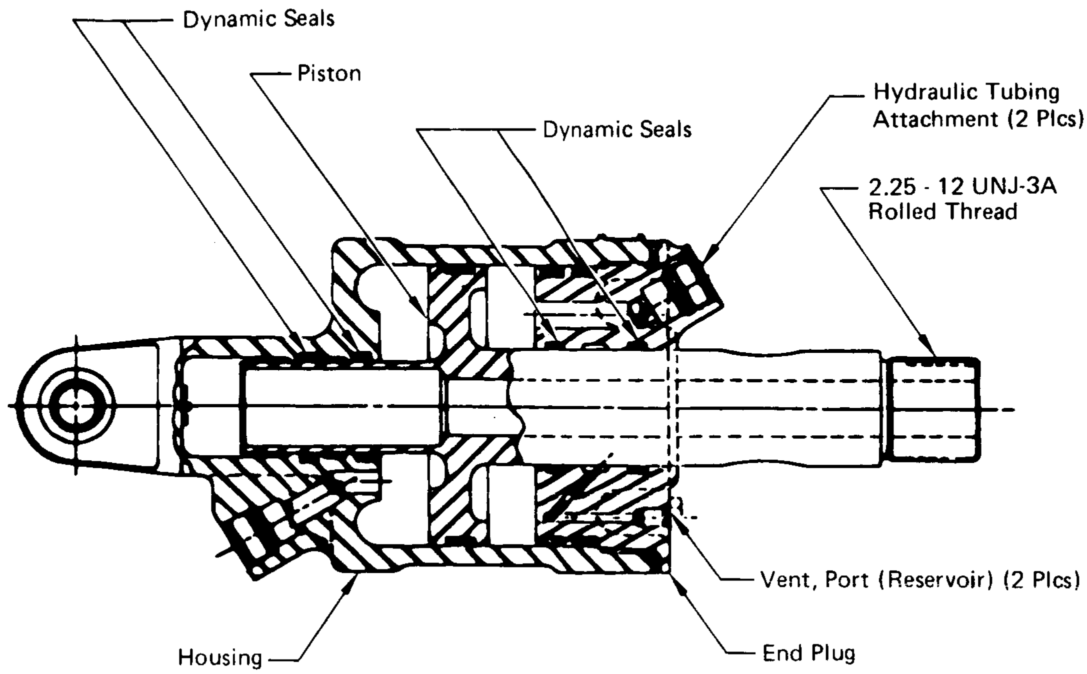



Figure 3. - Comparison of analytical and test piston loads.




Bleed-Fill Valve 2 Places (Not Shown)
Located 90° From HYD. PORTS.

Seals Per MIL-R-25988/1 Dynamic
Seals Per MIL-R-25988/4 Static

Figure 4. - Load leveler actuator.