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TOPEX HIGH-GAIN ANTENNA SYSTEM DEPLOYMENT ACTUATOR MECHANISM

Stephen R. Jones *

ABSTRACT

A deployment actuator mechanism has been developed to drive a two-axis gimbal assembly and a high-gain antenna to a deployed and locked position on the Jet Propulsion Laboratory Ocean Topography Experiment (TOPEX) satellite. The Deployment Actuator Mechanism requirements, design, test, and associated problems and their solutions are discussed in this paper.

INTRODUCTION

A High-Gain Antenna System (HGAS) was developed, under contract to Fairchild Space Company, to provide a data link between the TOPEX satellite and the Tracking and Data Relay Satellite System (TDRSS). Figure 1 illustrates the deployed HGAS mounted on the TOPEX satellite. The major components of the HGAS include a Two-Axis Gimbal (TAG) assembly, a High-Gain Antenna (HGA), a cradle assembly, and a Deployment Actuator Mechanism with a 1.4-meter long, 0.15-meter diameter, thin-wall aluminum mast. A layout of the stowed HGAS is presented in Figure 2.

Due to constraints within the ARIANE launch vehicle shroud, the HGAS will be launched in a stowed configuration on the outside of the satellite instrument module and will deploy once the satellite is in orbit. During launch, the TAG assembly and the HGA are supported by the cradle assembly, using three pyrotechnically actuated separation bolts. Once in orbit, the separation bolts are fired, and the Deployment Actuator Mechanism drives the TAG assembly and the HGA to a fixed 90-degree position with respect to the spacecraft. As the deployment nears completion, a pair of lock pins engage to secure and align the system.

* Honeywell Satellite Systems Operations, Glendale, Arizona

DESIGN REQUIREMENTS

The Deployment Actuator Mechanism is required to deploy one time in orbit. Manual retraction capability is provided for ground-based testing. The system is required to utilize redundant drive motors, bearings, lock assemblies, and lock and position telemetries. The lock assembly must provide for proper alignment of the gimbals and antenna following deployment and must provide the stiffness necessary to meet the deployed first-mode frequency requirement. Full deployment must be complete in less than 1800 seconds following pyrotechnic release of antenna and gimbals from the cradle assembly. As a final requirement, the deployment actuator mechanism must be thermally isolated from the spacecraft on which it is mounted.

MECHANICAL DESIGN

The Deployment Actuator Mechanism is a passive device that is driven by a pair of multileaf negator springs and is rate limited by a viscous fluid rotary damper. A pair of spring-driven, wedge-shaped lock pins align and secure the system as deployment is completed. The device utilizes a redundant bearing design featuring a lined spherical bearing and a dry lubricated journal bearing to provide two dissimilar rotating surfaces. A two-channel ganged potentiometer provides positional telemetry during deployment, while a pair of microswitches indicate lock position. The Deployment Actuator Mechanism is presented in Figure 3, and a layout of the device is shown in Figure 4. Design requirements and performance capabilities are presented in Table 1.

A passive system was traded off against an active system during the conceptual phase of the Deployment Actuator Mechanism design. Because the system is required to deploy once in orbit, and retraction capability is not required, a passive design offered advantages of less weight, higher reliability, and no power consumption. A redundant electromechanical actuator would be considerably more complex than a passive design, requiring redundant electric motors, gear trains, and clutches. Also, the drive electronics required to operate an electromechanical actuator could not be readily accommodated by the existing gimbal drive electronics boxes. Finally, initial weight estimates predicted that an active system would weigh approximately 65 percent more than a passive system.

A number of spring configurations were studied as a means of storing the energy required for deployment, including helical extension springs, helical torsion springs, and constant torque multileaf negator springs. In terms of package size and weight, the multileaf negator spring was the most attractive option. Additionally, this design has flight history in deployment applications, originating as a clock spring design on the Mariner Mars Spacecraft in 1971 and later modified to the present multileaf configuration for use on the Viking spacecraft in 1975 (Ref. 1). The spring layout is presented in Figure 5. The Deployment Mechanism utilizes a pair of 3.6-newton-meter six-leaf springs, each capable of deploying the system.

Crushable honeycomb, a viscous fluid rotary damper, and an eddy current damper were investigated as a means of controlling the rate of deployment and minimizing impact loads during lock-pin engagement. Crushable honeycomb was eliminated from consideration due to concerns regarding torque margin, repeatability, and ease of test. The eddy current rotary damper was given serious study; however, due to its complexity and lack of flight history, it was eliminated from consideration. A viscous fluid rotary damper was selected based on prior flight history, reliability, and cost.

The selected viscous fluid rotary damper is a small, light, limited-rotation device that produces approximately 452-newton-meter/(radian/second) damping at room temperature. Similar devices have flight history on commercial satellites as well as the Goddard COBE satellite. Damping is produced through viscous shear of 30,000-centistoke McGahn Nusil CV7300 silicon fluid between a rotating vane shaft and a stationary housing. A bypass valve provides adjustability of the damping rate by allowing fluid to be pumped from the high-pressure side of the vane shaft to the low-pressure side. The damper uses a reservoir with a flexible diaphragm to compensate for fluid volume changes with temperature. The reservoir interfaces with the damping chamber through a set of check ball valves that isolate the high-pressure side of the vane shaft from the reservoir during damper operation. A cross section is shown in Figure 6.

The bearing design must be capable of carrying launch and thermal-induced loads and must provide for a single, limited-rotation, low-speed deployment once in orbit. A number of redundant bearing configurations were considered, including a three-ring bearing design, a tapered roller bearing with a journal bearing interface, and a spherical bearing with a journal bearing interface. A spherical bearing with a Teflon®-impregnated Nomex liner in conjunction with a dry lubricated journal bearing at the shaft-to-inner-race interface proved adequate, given the requirements. A layout of this configuration is presented in Figure 7. The spherical bearings are manufactured to MIL-B-81820 and feature a low no-load breakaway torque, a low coefficient of friction, tight radial clearance, and a high static load carrying capability. The dry lubricant meets the requirements of MIL-L-81329 and consists of molybdenum disulfide and graphite in a sodium silicate bonding agent.

The lock assembly must align and lock the Deployment Actuator Mechanism into the deployed position. The lock assembly consists of a pair of wedge-shaped, spring-loaded pins, which preload adjustment screws against a hard stop, as shown in Figure 8. The adjustment screws allow for precise alignment of the deployed system, and the preload provided by the wedge-shaped pins results in zero backlash, allowing the system to meet the deployed first-mode requirements. A tool allows the pins to be retracted for restowing during test.

The thermal design was approached from a number of directions. The thermal isolation requirement was met by installing NEMA FR4 epoxy laminate thermal isolating pads on each side of titanium mounting bolts at the spacecraft interface. The damper utilizes a heated cover and is thermally isolated from the surrounding structure to ensure that it is maintained above its minimum operating temperature. Additional thermal control measures consist of gold plating and thermal blankets.

Positional telemetry is provided during deployment by a redundant, single-shaft, wire-wound potentiometer that interfaces with the Deployment Actuator Mechanism shaft. A pair of microswitches indicate when the system is fully deployed and lock pins are engaged.

TEST

An engineering Deployment Actuator Mechanism was assembled for development testing, including characterization of spring and damper operation, drag torque measurements, alignment repeatability measurements, three-axis vibration, and hot and cold thermal-vacuum deployments. The engineering Deployment Actuator Mechanism was then integrated with other engineering components for system-level development testing, including three-axis vibration, stowed, and deployed first-mode frequency measurements and deployment testing in which thermal-induced loads were simulated and pyrotechnic separation bolts were fired.

Assembly of the flight Deployment Actuator Mechanism began with characterization of spring and damper operation. The springs and damper were tested using a fixture that utilizes a pulley and weights to produce an input torque. Leaf spring torque was found to be within 3 percent of the 3.6-newton-meter analytical prediction. The damper was found to produce damping rates of 1356 and 169 newton-meters/(radian/second) at temperatures of -20 and 75 degrees C, respectively. Damping was found to vary with input torque; however, because the spring torque is constant, this is not a concern in this application.

Room-temperature drag torque measurements were made without the damper and springs installed. Representative cables were installed across the joint to simulate the resistance torque that the TAG, coaxial, and pyrotechnic cables would produce. Although the dry lubricated journal bearings and the Teflon-lined spherical bearings each resulted in an average drag torque of 0.7 newton-meter, the journal bearings proved to be the primary rotating surface. With 7.0 newton-meters of available spring torque, the torque margin is 10.0. Based on engineering test data, the drag torque can be expected to increase by approximately 0.2 newton-meter at -20 degrees C, resulting in a torque margin of 7.8. The Deployment Actuator Mechanism drag torque was considerably less than the 1.7 newton-meters originally budgeted, resulting in a torque margin greater than the design requirement of 4.0.

Although alignment measurements were not made with the flight hardware, alignment repeatability measurements were made on the engineering hardware by using optical cubes and a theodolite. In five consecutive deployment tests, alignment was found to be repeatable to 0.02 degree, considerably better than the 0.05-degree requirement.

The Deployment Actuator Mechanism was subjected to 9.8g's rms random vibration as a workmanship test. Random vibration of the flight hardware was trouble free.

Deployment time was measured as a function of temperature in a thermal-vacuum chamber. The mechanism was deployed horizontally to minimize gravity effects. A short length of mast replaced the 1.4-meter flight mast. It was not necessary to simulate the inertia of the rotating load because the relatively high damping rate of the system in relation to the rotational inertia makes the rate of deployment essentially independent of inertia.

The time required for deployment varied between 45 seconds at 58 degrees C and 238 seconds at -8 degrees C. Room-temperature deployment time was approximately 85 seconds. The cold-temperature deployment time fell easily within the 1800-second requirement. Based on acceptable impact loads, the minimum acceptable deployment time is 45 seconds. Although the hot deployment did not violate the 45-second minimum deployment time, it did deploy somewhat faster than expected, due primarily to the low drag torque of the flight deployment actuator.

PROBLEMS RESOLVED

Post vibration inspection of the engineering Deployment Actuator Mechanism revealed that the dry lubricant in the journal bearing was flaking, resulting in a measurable increase in drag torque. A number of possible causes were investigated including improper application and moisture contamination. Although the sodium silicate bonding agent used in the dry lubricant is impervious to many solvents, it can be dissolved by moisture. Improper application could include surface contamination, inadequate curing, or excess thickness. After considerable testing, the problem was found to be excessive lubricant thickness. The solution involved applying the lubricant to a single bearing surface, rather than both bearing surfaces, and burnishing the lubricant to reduce thickness. This solution resolved the problem and resulted in decreased drag torque during normal operation.

It was during thermal-vacuum testing of the flight hardware that the most serious problems arose. These problems involved the potentiometer and viscous damper.

During thermal-vacuum deployment testing, noise appeared on one channel of the redundant potentiometer. A failure analysis investigation revealed contamination on the windings. A review of the materials and processes used in the manufacture of the part showed that the contamination is not an inherent problem and is probably an isolated incident. Five qualification parts are being tested for signs of contamination. In addition, the spare flight part is being screened to establish flight worthiness.

The most serious problem is related to thermal-vacuum operation of the viscous fluid rotary damper and is characterized by a region of undamped travel immediately after deployment has been initiated, followed by normal operation throughout the remainder of the travel. The phenomenon is random in nature and has occurred at hot and cold temperatures. The maximum undamped travel was 15 degrees, resulting in unacceptably high impact loads in the damper input shaft as damper operation returned to normal. Because a potentiometer was not installed to provide positional telemetry during engineering model thermal-vacuum testing, the problem did not become apparent until acceptance testing of the flight unit.

A review of the fill procedure revealed inadequacies that could result in air or a void within the damper. Although air was found in an engineering damper, there was no sign of air in the flight damper. A void could not be ruled out, however, so the damper was emptied and refilled using a fill fixture in which the device was evacuated and then filled with degassed fluid under pressure.

A second area of concern was the method used to compensate for changes in the damping fluid volume over temperature. The damper uses a reservoir with a flexible diaphragm to compensate for fluid volume changes. The reservoir interfaces with the damping chamber through a set of check ball valves that isolate the high-pressure side of the vane shaft from the reservoir during damper operation. In previous space applications, the external side of the diaphragm was pressurized at 1 atmosphere with a sealed cover on the rear of the damper. Positive pressure on the diaphragm appears to be critical to the operation of the device. Due to concerns that pressurization of the damper would increase the likelihood of a fluid leak developing at the shaft seal, the damper used on the Deployment Actuator Mechanism had a vented cover. The vented cover was eliminated in favor of a sealed cover.

Subsequent testing has shown that the anomaly still exists. Thermal-vacuum test results are presented in Table 2. The average free travel was 3.8 degrees with a maximum of 10.1 degrees and a minimum of 1.8 degrees.

Because operation of the damper is not well understood, and a solution to the problem is not forthcoming, design changes are being implemented to replace the original damper with a customer-furnished damper. The customer-furnished damper was developed for use on the TOPEX solar array drive mechanism and has successfully completed extensive qualification and acceptance testing. Operation of the customer-furnished damper is similar to operation of the original damper. Additional testing following integration of the customer-furnished damper is expected to show that the Deployment Actuator Mechanism meets or exceeds all design requirements, with performance similar to that shown in Table 1.

CONCLUSION

A Deployment Actuator Mechanism has been developed and is presently undergoing flight acceptance testing. The most challenging problem encountered during the development and test of the Deployment Actuator Mechanism was a viscous rotary damper anomaly. Although virtually identical dampers have flight heritage, the device proved unsuitable in this application, and design changes are being implemented to replace the device with a customer-furnished damper. With the exception of the damper anomaly, development has been essentially trouble free. The Deployment Actuator Mechanism is expected to meet or exceed all design requirements following integration of the customer-furnished damper. Although developed specifically for the TOPEX satellite, the Deployment Actuator Mechanism is a versatile design and may be adapted to a variety of other deployment applications.

REFERENCE

(1) "Viscous Rotary Vane Actuator/Damper," Jack D. Harper, 10th Aerospace Mechanisms Symposium, April 22-23, JPL Technical Memorandum 33-777, pp. 198-207.

TABLE 1. REQUIREMENTS AND CAPABILITIES

Parameter	Requirement	Capability
Mass (kg)	<18.1	12.6
Footprint (m x m)	0.40 x 0.29	0.40 x 0.29
Operating Temp (°C)		
Maximum	38.0	58.0
Minimum	11.5	-8.5
Non-operating Temp (°C)		
Maximum	63.0	83.0
Minimum	-7.5	-27.5
Deployment Time (s)		
58°C	=>45	45
-8.5°C	<1800	300
Spring Torque (N-m)	6.8	7.0
Drag Torque (N-m)	<1.7	0.9
Torque Margin	>4.0	7.8
Damping Rate (N-m/rad/s)		
58°C	>147	169 @ 75°C
-8.5°C	<4859	1356 @ -20°C
Alignment (deg)		
Initial	0.05	0.05
Repeatability	0.05	0.02
Deployed Stiffness (Hz)	>5.0	7.2

TABLE 2. THERMAL-VACUUM DEPLOYMENT TEST RESULTS

Test	Temperature (°C)	Deployment Time (s)	Average Damping (N-m/rad/s)	Undamped Travel (deg)
1	29.8	87	376	1.9
2	-8.8	220	943	2.2
3	-7.5	218	956	4.2
4	59.0	45	210	6.4
5	-6.8	214	917	1.9
6	-8.4	238	1022	1.8
7	58.3	48	1965	3.9
8	22.4	108	4166	2.1
9	58.5	46	1997	10.1

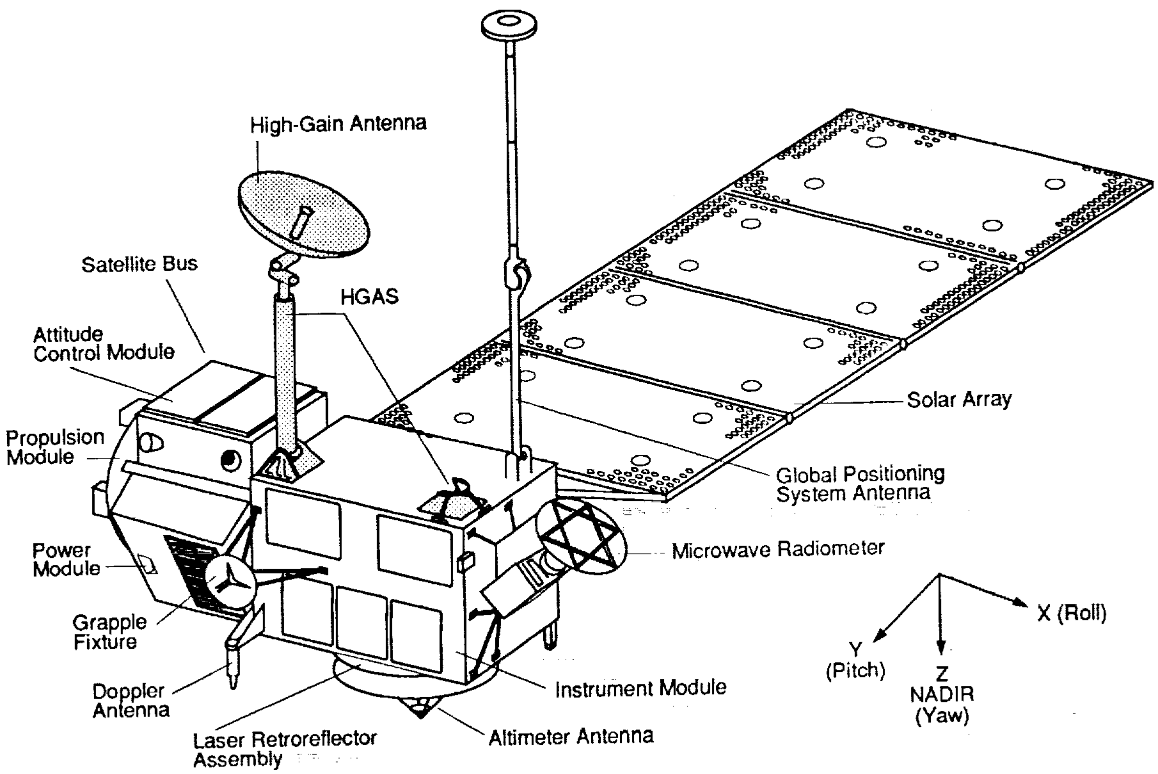

Figure 1. TOPEX Spacecraft/HGAS

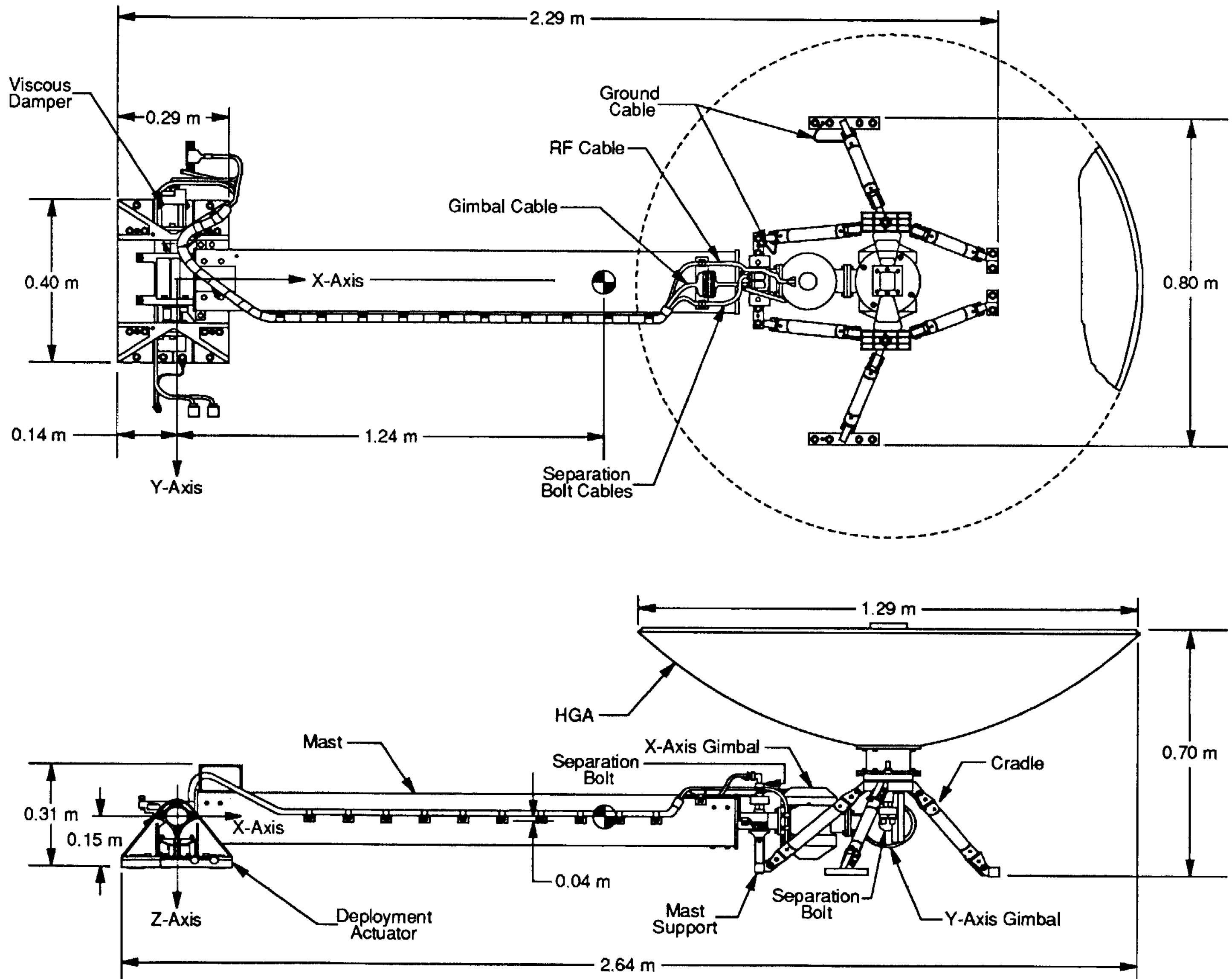

Figure 2. TOPEX HGAS Layout

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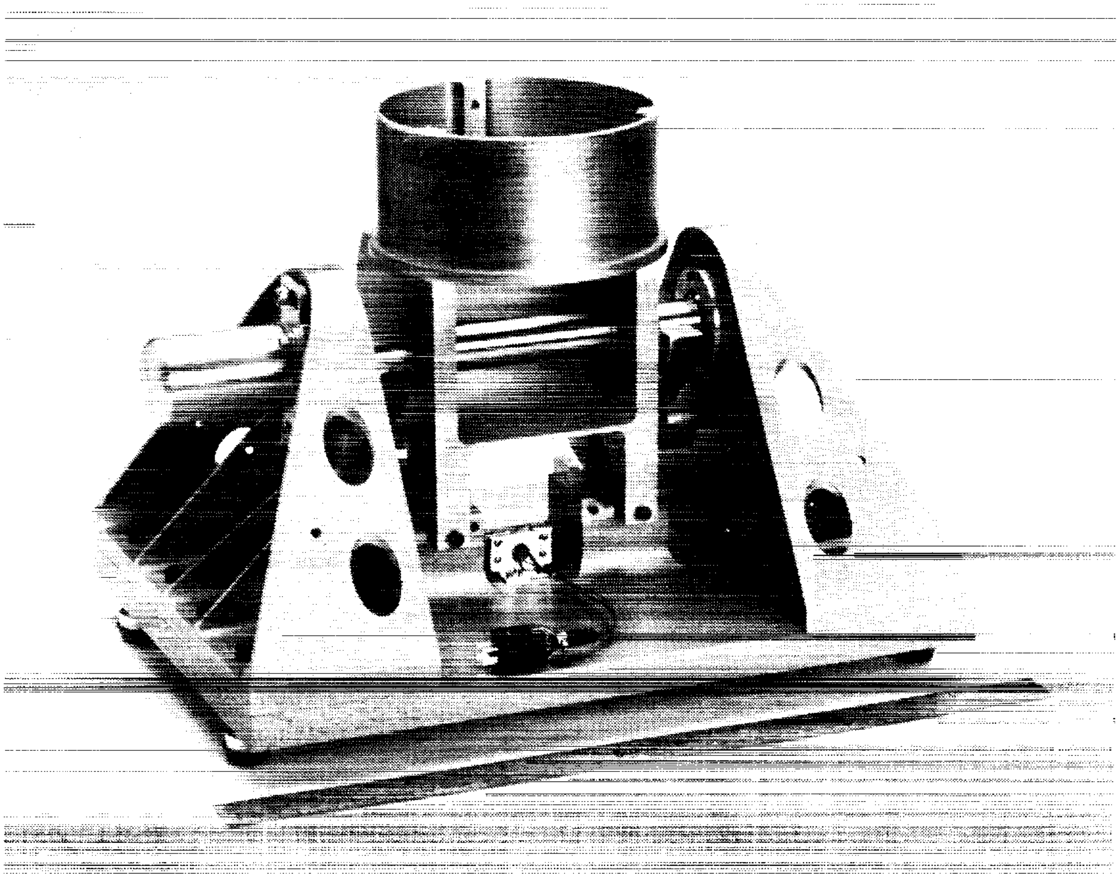

Figure 3. Deployment Actuator, Deployed Position

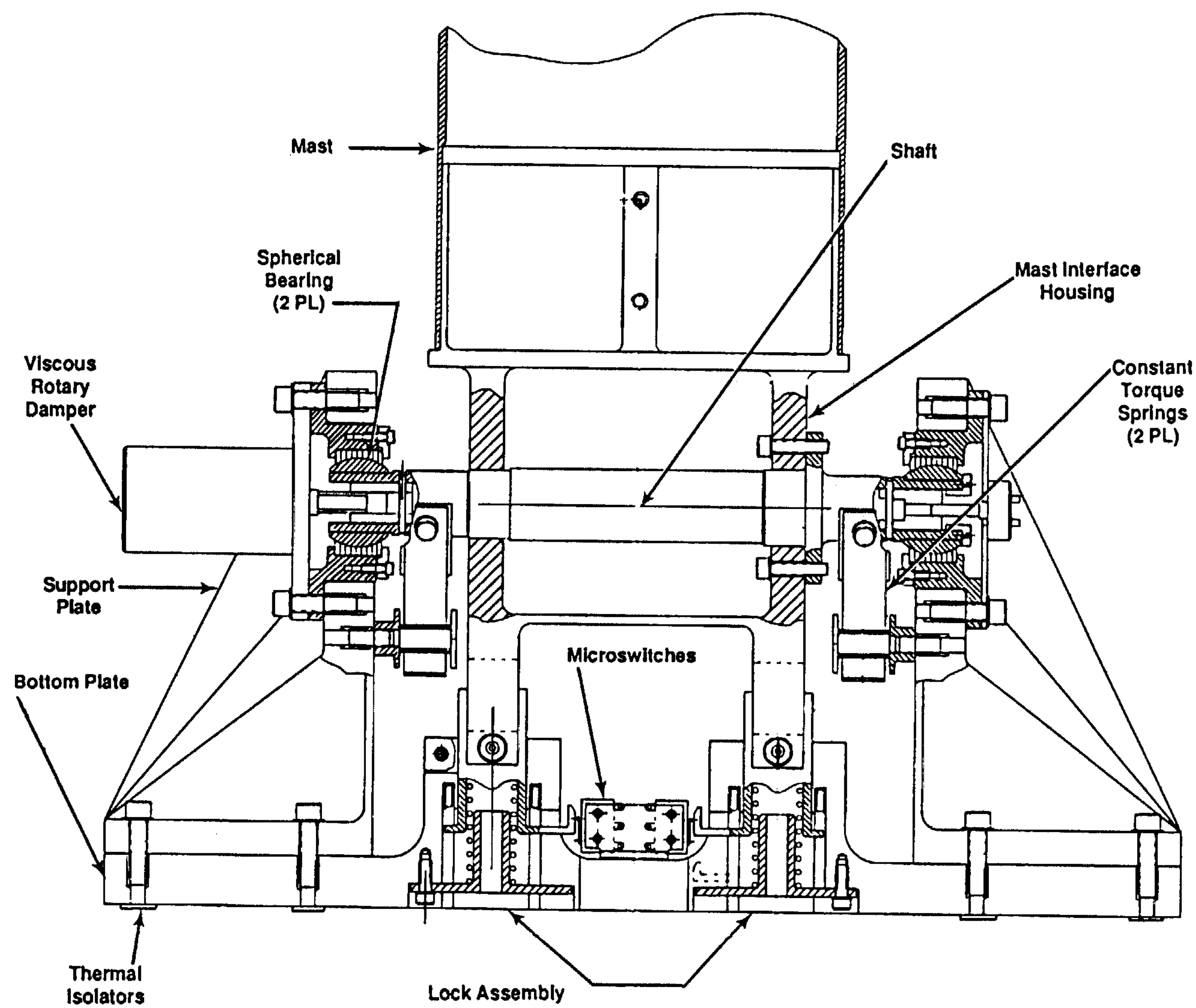

Figure 4. Deployment Actuator Layout

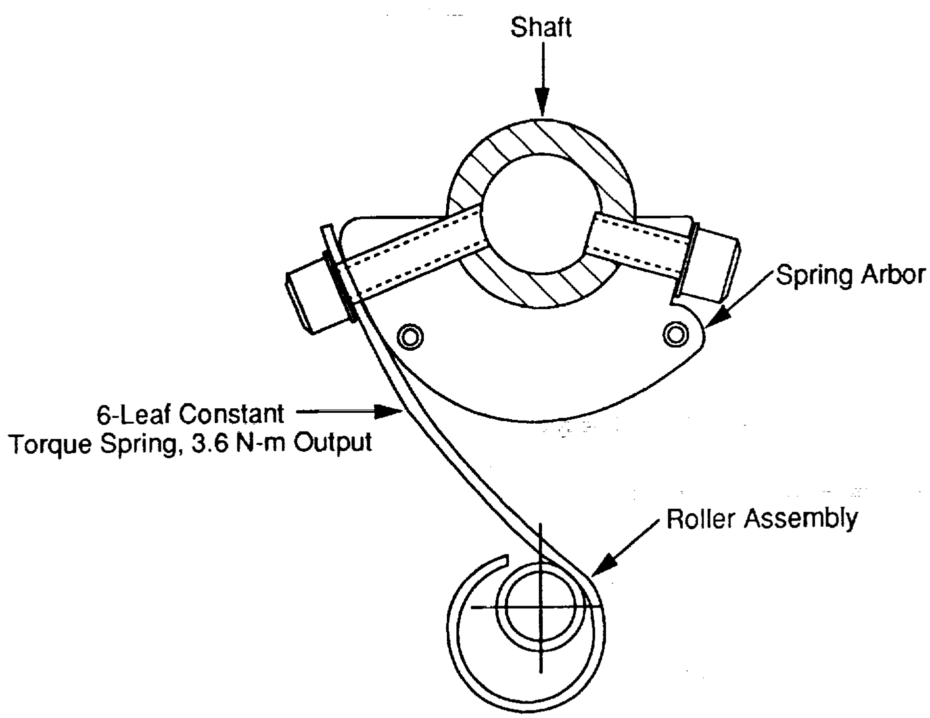

Figure 5. Constant Torque Spring Detail, Deployed Position

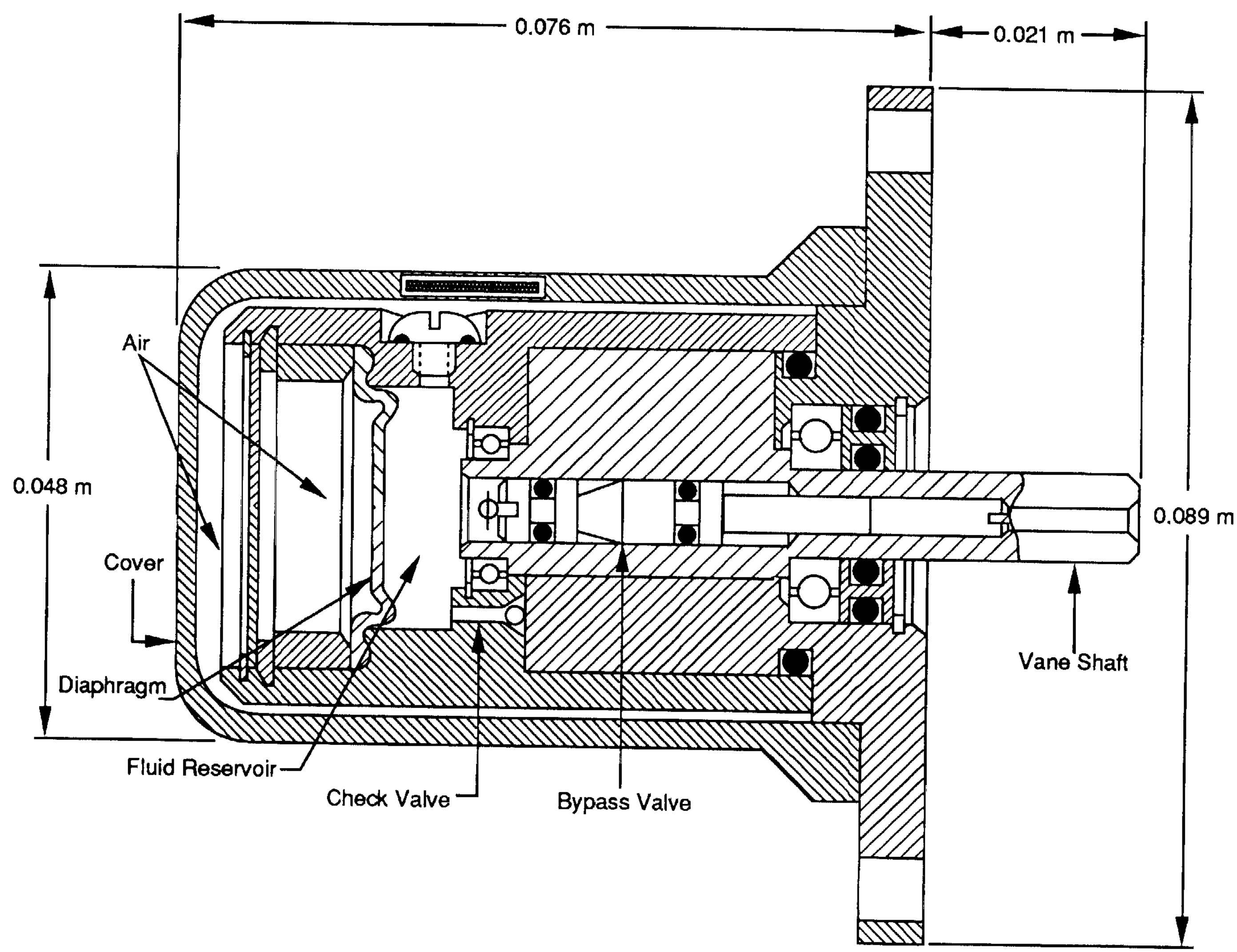

Figure 6. Rotary Damper Cross Section

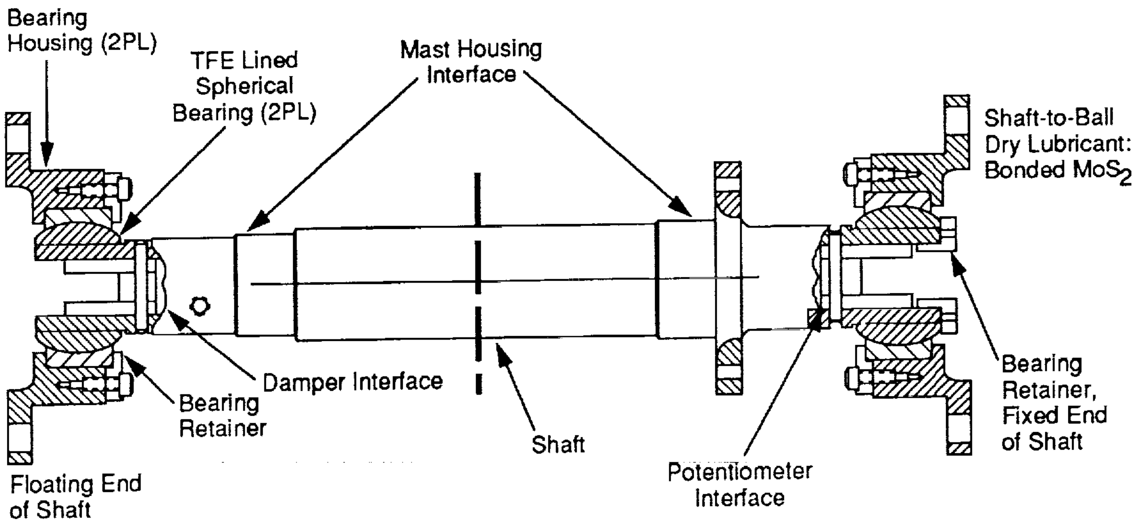

Figure 7. Bearing Detail

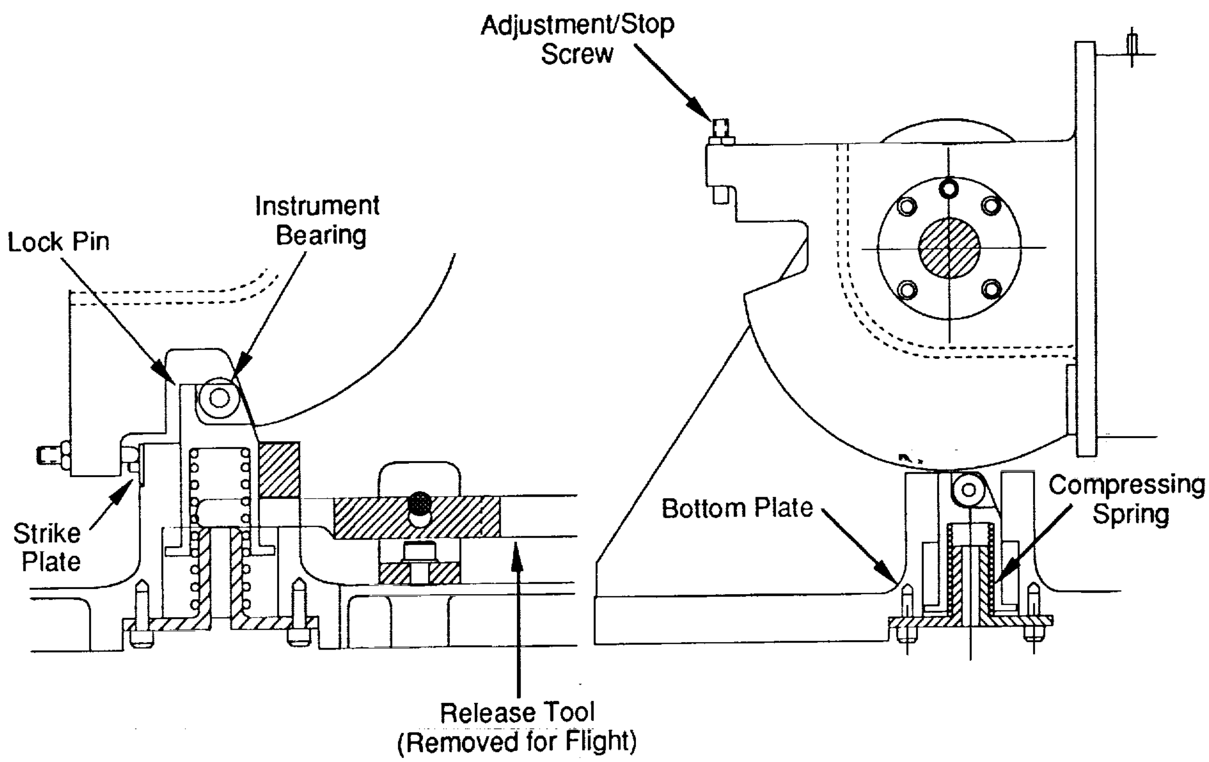

Figure 8. Lock Assembly Detail