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APPENDAGE DEPLOYMENT MECHANISM  
FOR THE HUBBLE SPACE TELESCOPE PROGRAM

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ABSTRACT

This paper presents the key requirements, a design overview, development testing (qualification levels), and two problems and their solutions resolved during the mechanism development testing phase. The mechanism described herein has demonstrated its capability to deploy/restow two large Hubble Space Telescope deployable appendages in a varying but controlled manner.

INTRODUCTION

As part of the NASA Hubble Space Telescope (ST), a Shuttle-launched spacecraft, three appendages are deployed/stowed by an electromechanical hinge mechanism: two antennas and an aperture door. One use of the mechanism is for the deployment/stowing of the two high gain antennas (HGA) used to transmit scientific data to the ground station. The second use is to open/close the large aperture door (AD) located at the end of the telescope aperture, which must close to preclude the sun from shining in on the telescope primary mirror. Figure 1 shows the deployed telescope. The deployment mechanisms are shown in Figure 2.

DESIGN REQUIREMENTS

The deployment mechanism, a limited-angle hinge device, was designed to deploy/stow two spacecraft appendages with the following specific requirements:

- o The deployment mechanism must be capable of surviving five Shuttle launches, four returns to Earth, and eight in-orbit docking operations with appendages stowed and locked.
- o The deployment mechanism must be redundant so that no single component failure will jeopardize recovery of the spacecraft or result in hazards to personnel.
- o The deployable appendages must be capable of manual operation and jettison by an Extra Vehicular Activity (EVA) crewmember to ensure deployment, restowage, and safe return of the ST spacecraft to Earth.
- o The aperture door deployment mechanism must be capable of closing within one minute after a command from the sun sensor, precluding irreversible damage to the telescope optics if the spacecraft points the aperture toward the sun.

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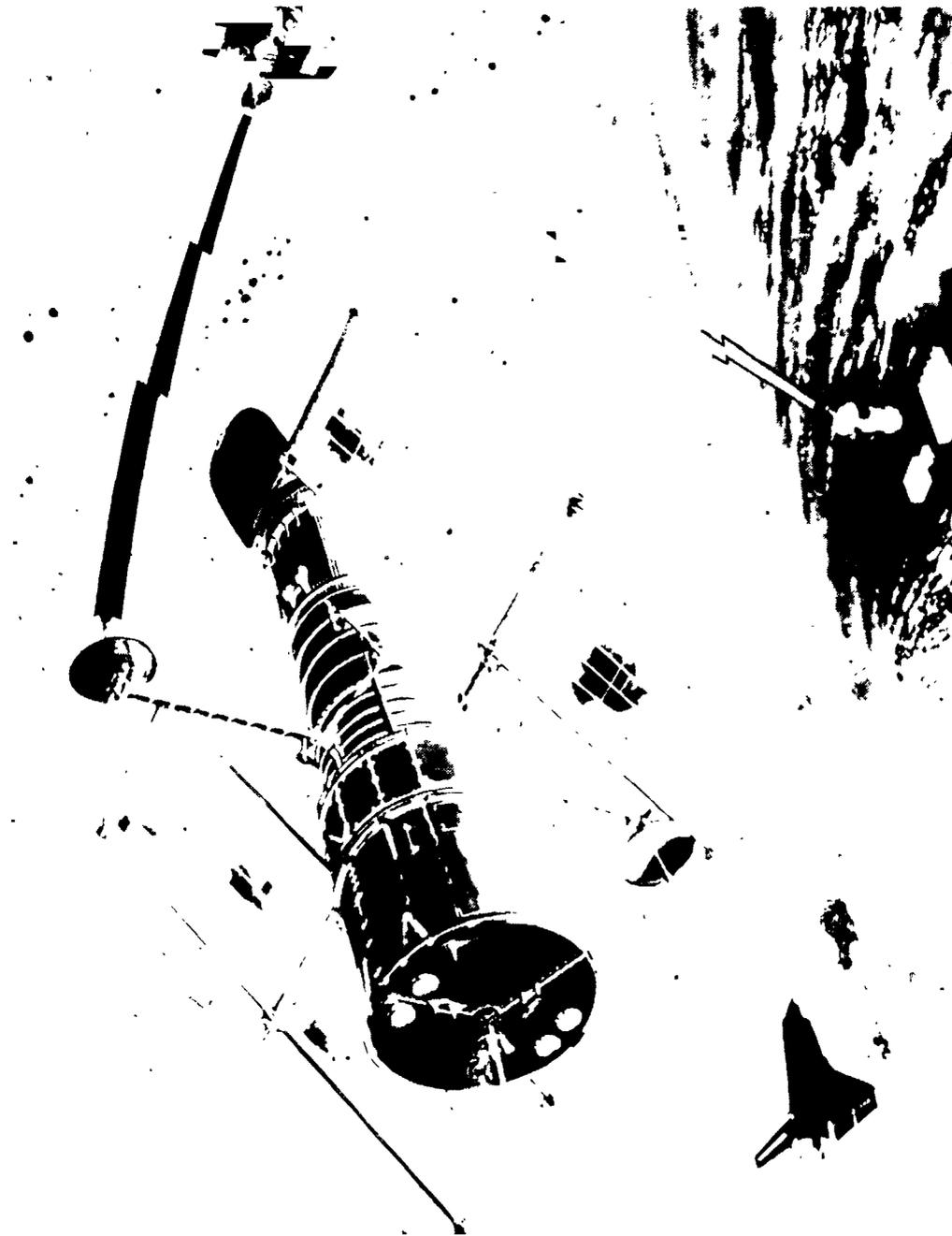


Figure 1 On-Orbit ST Deployed and Transmitting to Ground Station.

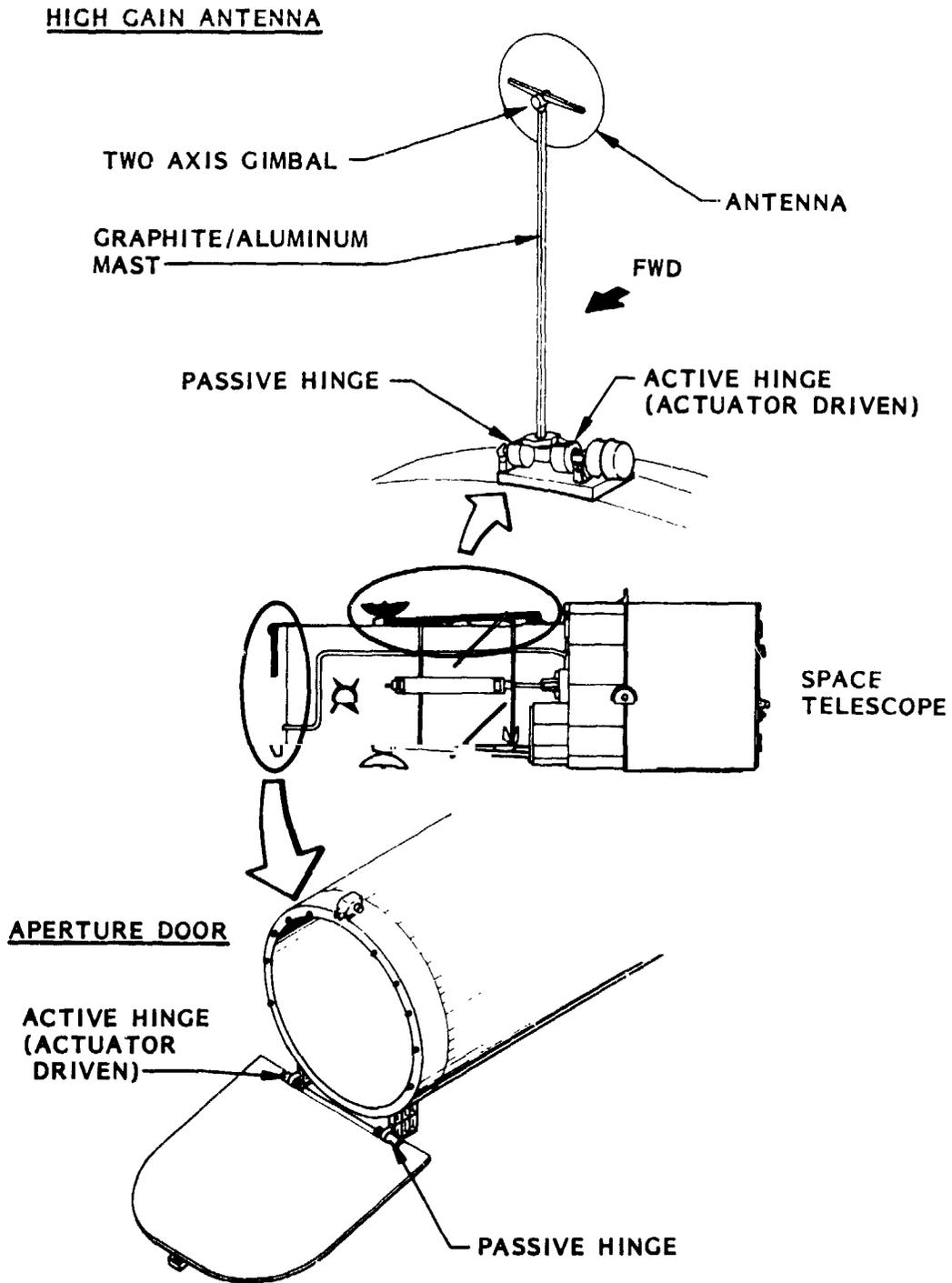


Figure 2 Stowed ST, Close-up of Two Deployed Appendages.

- o The deployment mechanism must operate in an Earth-orbiting pressure environment and at heater-controlled temperatures from  $-62.2^{\circ}\text{C}$  ( $-80^{\circ}\text{F}$ ) to  $+43.3^{\circ}\text{C}$  ( $+110^{\circ}\text{F}$ ).
- o The hinge mechanism deployed stiffness must meet a first mode natural frequency of less than 1.0 Hz.

#### MECHANISM DESCRIPTION

The deployment mechanism consists of an active hinge (dc rotary drive actuator powered) and a passive support hinge. Detailed design features may be identified in Figs. 3 and 4.

The large-diameter, angular-contact, output-support bearings and the self-aligning spherical bearing are identical in each active and passive hinge assembly. The self-aligning feature permits some angular misalignment between the active and passive assemblies since, in the aperture door hinge installation, they are approximately 147.3 cm (58 in.) apart.

The active hinge mechanism, Fig. 3, is driven by a Schaeffer Magnetics, Inc., three-phase, six-state, "wye" connected,  $1.5^{\circ}$  permanent magnet dc stepper motor driving through a 200:1 Harmonic Drive speed reducer. The bidirectional rotary actuator drives a torsion rod via a splined shaft which is keyed to the domed end cover. The torsion rod permits the actuator, with its inherent powered detent torque, to move larger inertias than if direct coupled. This is the point where the AD or HGA is attached. The spring-loaded friction plugs of Delrin AF rub constantly against the end cover to provide damping during movement of the mechanism. Just prior to the fully deployed position, the adjustable intermediate stops located in the domed cover contact the intermediate torsion tube and the torsion tube, both compliant members, although with different spring rates, which reduce the end-of-travel dynamic loads and contribute to the overall deployed mechanism stiffness; the intermediate torsion tube also has coulomb damping in the form of annular Delrin AF friction segments.

A manual wrenching point to rotate the active hinge is provided at the end of the torsion rod opposite the rotary drive actuator. The wrenching point accepts the standard EVA hexagonal 10.9-mm (7/16-in.) socket.

As noted in the mechanical schematic, Fig. 5, the rotary drive actuator drives into an adjustable stop just after preloading the torsion rod and the other two compliant members at the deployed position.

If an appendage is manually deployed by a crew member, the active hinge must be manually preloaded by imparting a small angle of travel to the output shaft of the rotary drive actuator; the crew member does this by inserting the standard socket with a long extension attached through a hole in the rotary drive actuator adapter housing and levering the actuator into its stop (Fig. 6).

End-of-travel telemetry (TLM) switches are provided to indicate when the deployment mechanism is near either the stowed or deployed position. In addition, the aperture door mechanism includes a rotary potentiometer to provide full mechanism travel-position readout.

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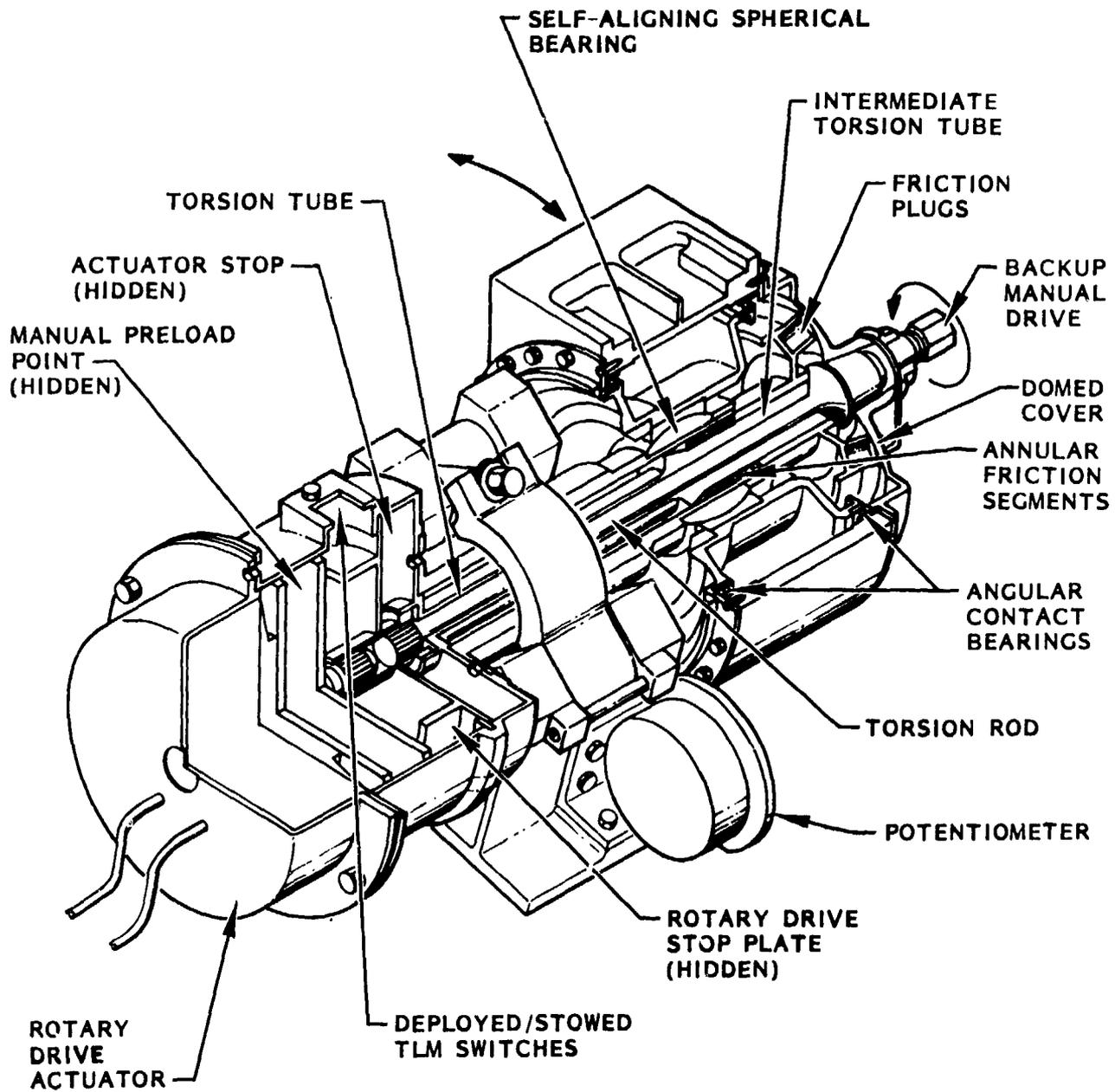


Figure 3. Active Hinge Assembly.

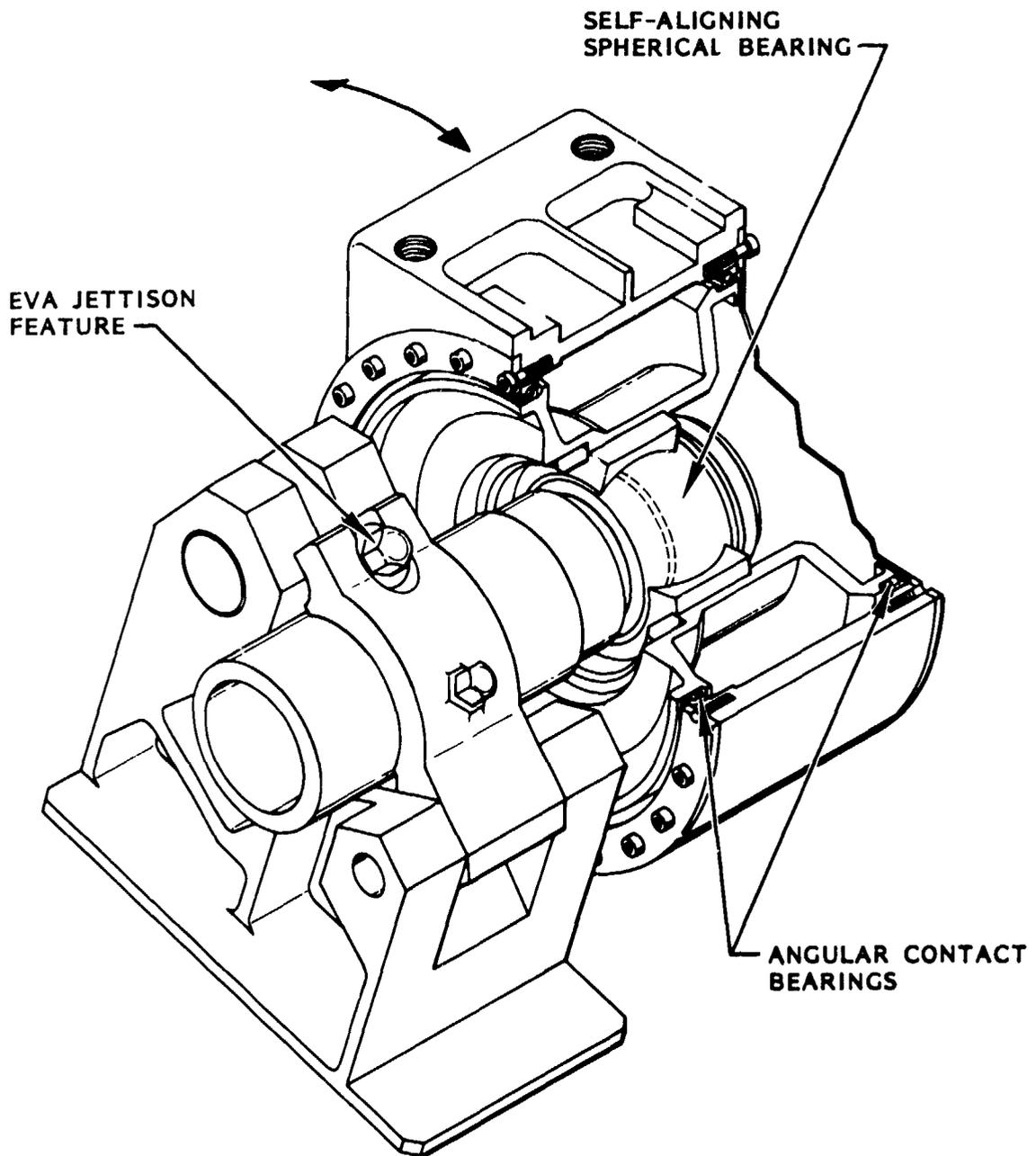


Figure 4. Passive Hinge Assembly.

The self-aligning bearing not only provides mechanism misalignment correction capability, but also acts as a redundant rotational bearing to the two angular contact bearings. Note that the angular contact bearing inner races are supported on a "zee" section spring diaphragm support. This provides some compliance between the inner and outer components as the mechanism goes to orbital temperatures where different material coefficients of thermal expansion (contraction) could cause additional bearing loads. The zee shape deflects sufficiently to maintain only a slight increase in bearing preload at the coldest orbital temperatures.

Bray Oil Co. wet lubricants, Brayco Micronic 815Z oil as well as their Braycote 3L-38RP (Braycote 601) grease are used to lubricate vital components within the mechanism, including the rotary drive actuator as well as the potentiometer. The large angular contact bearings are lubricated with a solvent-thinned solution of the Braycote 3L-38RP grease (grease plating), then vacuum-baked, leaving a thin, evenly distributed film of the lubricant.

The passive hinge, Fig. 4, uses the same rotational bearing support, except for one additional feature: the 15-5 PH stainless steel stub shaft is permitted to slide along the axis of rotation of the inner race of the stainless steel spherical bearing. A groove is cut into the shaft, providing a reservoir for Braycote 3L-38RP grease. The sliding feature is important for the AD use of the mechanism since the active and passive hinges are approximately 147.3 cm (58 in.) apart and small temperature gradients between the large aluminum honeycomb door and the 3.05-m (10-ft)-diameter spacecraft structure change the width between the two hinge assemblies. The slip feature then accommodates this dimensional change.

Both active and passive hinge assemblies carry radial loads via the bearings; however, only the active hinge resists thrust bearing loads (parallel to hinge line), and they are carried out through a conical pin installed in the hinge support bracket and seated in the outer support tube of the mechanism.

#### HINGE MECHANISM OPERATION

The rotary drive actuator is a current-controlled device which also has its step rate (pulses per second), rotational angular travel (total steps), and direction of travel controlled. For the two deployment mechanisms described here, the actuator is driven at two rates, either 30 pulses per second (0.23 deg/sec), or 300 pulses per second (2.25 deg/sec), or a sequential combination of the two. The total travel angle in either use is less than one revolution; 92.7° to deploy/stow the HGA or 105.3° to open/close the AD. The electronic control unit commands the travel angle to the stepper motor via a predetermined number of steps which includes approximately 1000 more steps than are actually needed to reach the commanded position. This assures more than sufficient steps for any contingency.

The deployment sequence is initiated by driving the actuator at the desired stepping rate, which relieves a slight stowed preload in the torsion rod. With the preload relieved, the actuator begins winding up the torsion rod, which

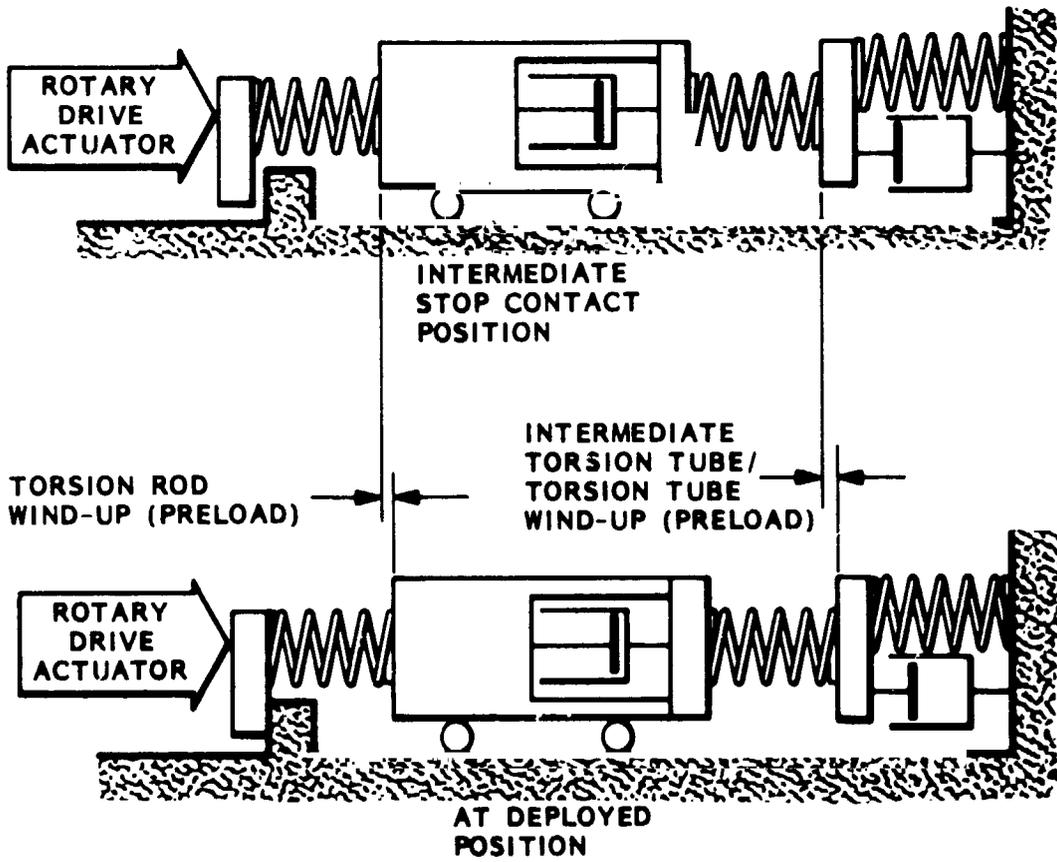
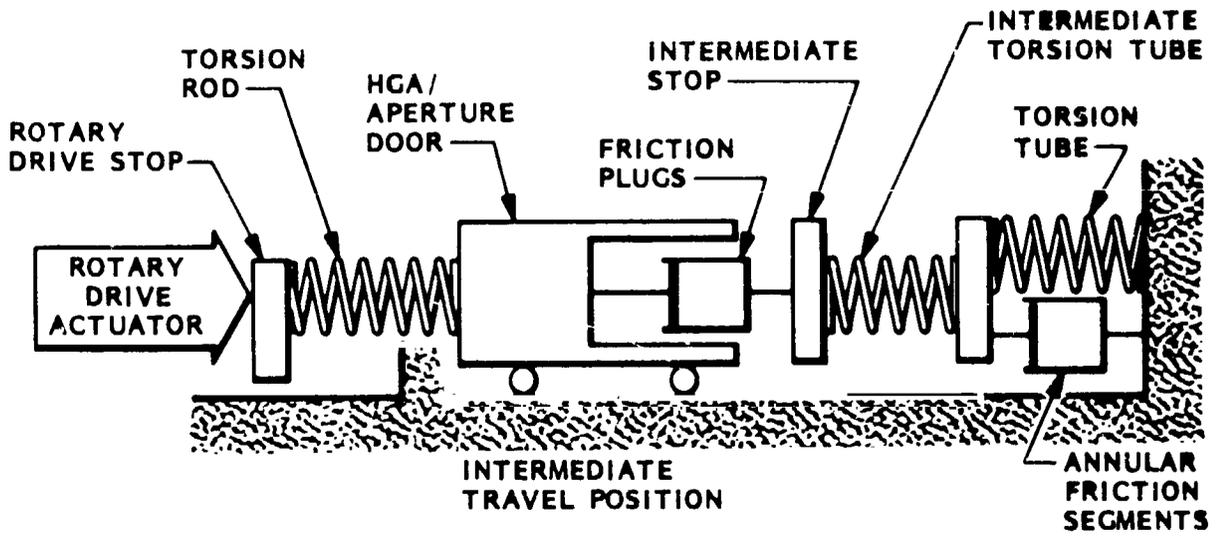


Figure 5. Mechanical Schematic.

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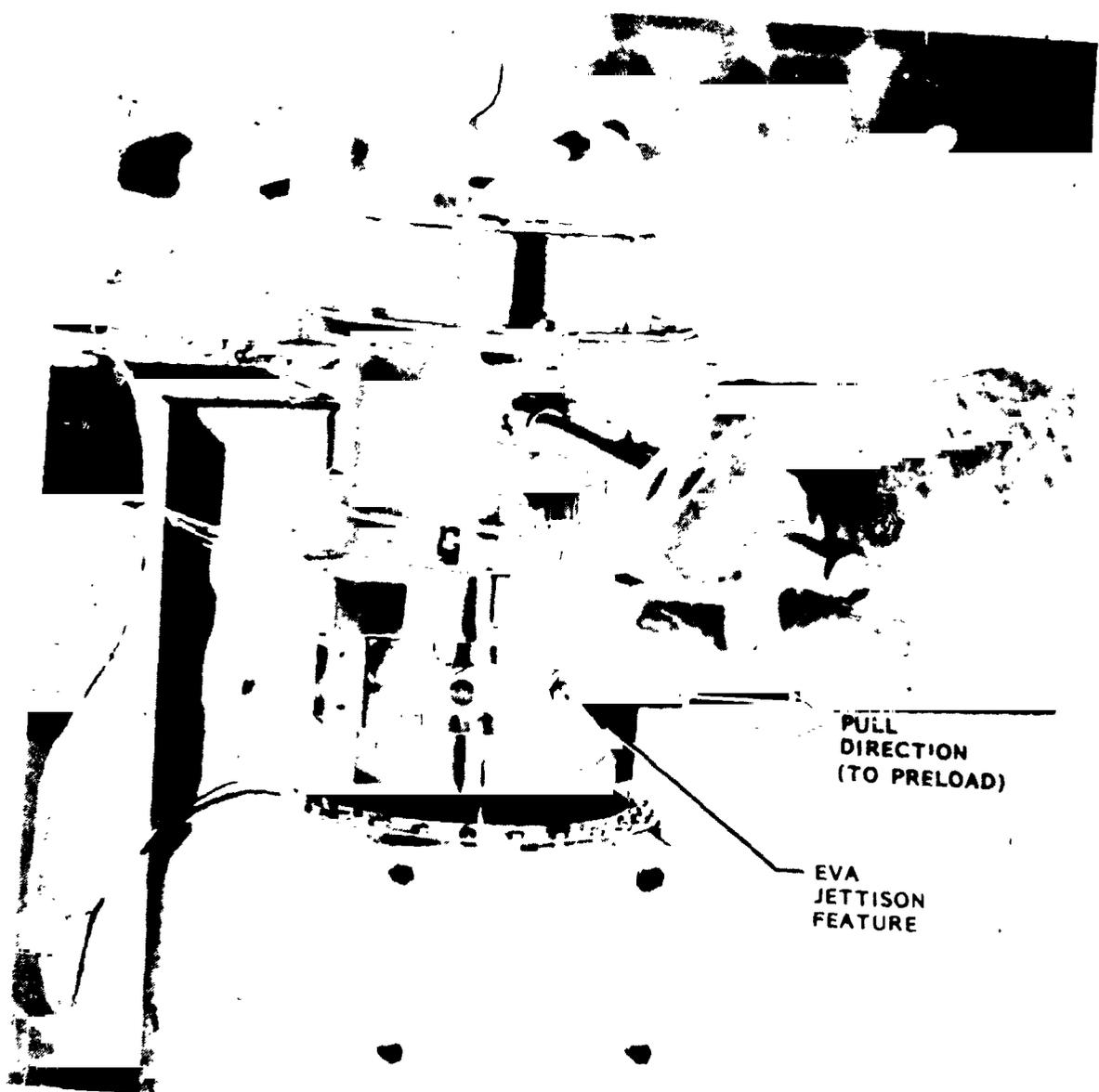


Figure 6. Manually Preloading Mechanism.

soon overcomes bearing viscous friction and coulomb friction from the Delrin plugs and moves the outer bearing housing, which is connected to the appendage. During deployment, only viscous friction provides damping while coulomb damping is only effective when the mechanism is at or near its end-of-travel when motion direction may reverse. This occurs during the AD speed change and at intermediate stop contact.

Approximately 3 deg before the final open position in the AD deployment sequence only, the control electronics changes the actuator step rate from 2.25 deg/sec to 0.23 deg/sec. Motion continues until less than 1 deg from end-of-travel, when the stops on the moving domed cover contact the intermediate torsion rod and the torsion tube. The rotary drive actuator continues to drive, increasing the wind-up (load) in the torsion rod, intermediate torsion tube and the torsion tube until the adjustable actuator stop is contacted. The actuator stalls against the stop for the final 1000 steps.

This preload, amounting to 9.04 N·m (80 in.-lbs), ensures that the appendage stays in the deployed position despite small spacecraft maneuvers. The deployed position is depicted in the mechanical schematic, Fig. 5, which shows the appendage ultimately positioned between the small angular wind-up of the torsion rod and torsion tube. Unpowered magnetic detent of the rotary drive actuator provides greater than 22.6 N·m (200 in.-lbs) of resistive torque to maintain the preloaded position. The detent torque is developed by the permanent magnet 1.5 deg stepper motors within the actuator, and the torque is amplified by the 200:1 Harmonic Drive gearhead.

The stowing operation is the reverse of deployment, the primary difference being the absence of the domed cover stops, thus only the torsion rod winds up as the securing latches are contacted.

#### APPENDAGE SYSTEM OPERATION

##### Aperture Door

To protect the optics, the door must close within 60 seconds, it is moved at the higher step rate of 300 pulses per second (2.25 deg/sec). The sequence is to drive the 1164-N·m-sec<sup>2</sup> (97 slug-ft<sup>2</sup>) door inertia through 102 deg of its 105.3 deg of total travel in approximately 47 secs, at which time the step rate is reduced to 30 pulses per second (0.23 deg/sec). At the point of speed change, the door has effectively blocked direct sunlight from shining down the optical aperture and may then close at the slower rate, reducing the end-of-travel impact dynamics. For system convenience, the AD is opened using the same operation used for closing the door - this means that only the direction of travel of the rotary drive actuator has to be changed.

### High Gain Antenna

This appendage consists of a 132-cm (52-in.)-diameter parabolic reflector which is pointed at a relay satellite by a two-axis gimbal subsystem. Thus in turn is attached to a 358-cm (141-in.)-long mast/rf waveguide support. The mast is then bolted directly to an adapter connecting both the active and passive hinge mechanisms, which are close to each other (Fig. 2). With the exception of the angle of travel, these are the identical components used in the AD deployment mechanism. Here the mechanism is deploying 2532 N·m-sec<sup>2</sup> (211 slug-ft<sup>2</sup>) through an angle of 92.7 deg of total travel.

Because the two gimbals are low-output torque devices and are subject to disturbing torques (i.e., coming into or out of the stow position), they are powered to hold their zero position during appendage deployment or restowage. The slowest deploy/restow rate of 30 pulses per second (0.23 deg/sec) is used for the full travel of this mechanism, requiring over 7 min to deploy or restow the appendage.

### DEVELOPMENT TESTING

#### Scope

All in-house development testing is now complete. Thermal testing consisted of testing to qualification levels of at least 16.6°C (30°F) lower and higher than worst case predicted orbital temperatures. To validate the structural analysis, static and dynamic proof load tests were performed to 140% of design limit loads.

It should be noted that the bidirectional rotary drive actuator was the subject of thorough prototype, development and flight acceptance testing as part of the purchase subcontract. The testing by the supplier Schaeffer Magnetics, Inc., included thermal vacuum testing by driving an inertia simulator, at all step rates, through compliant elements simulating the torsion rod spring rate used in the mechanism. Although the actuator is thermostatically heater-controlled to operate at -34.4°C (-30°F) or above, it was operated successfully to as low as -73.3°C (-100°F).

The wire-wound potentiometer was also tested by its supplier, Spectrol Electronics Corp, to test levels well beyond its expected orbital requirements.

The in-house testing was conducted in three phases:

#### Phase I - Mechanism Characterization

- o Thermal vacuum testing of lubricated angular contact bearings and coulomb friction elements.
- o Thermal testing of the self-lubricated (low friction linear) spherical bearing and determination of parasitic torque characteristics of the electrical cable and rf coax. The cables are required to bend as the HGA mechanism rotates.

## Phase II - Hinge Mechanism Operation

- o Thermal testing of the mechanism with an inertia simulator driven through full travel at the two deploy/stow rates, 2.25 deg/sec and 0.23 deg/sec.
- o Static proof load testing (1.4 times design limit load) simultaneously applied normal and parallel to the mechanism rotation axis.
- o Dynamic inertia proof load testing (1.4 times the design limit load).
- o Random vibration of the mechanism in three axes for 1 min in each axis.
- o Deployed mechanism natural frequency verification.

## Phase III - Subsystem Testing

- o Subsystem operation at ambient conditions - approaching and leaving securing latches and preloading at the deployed position.
- o Flightcrew member interface verification; manual operation of mechanism, including appendage jettison features.
- o Compatibility verification of mechanism and flight breadboard electronics.
- o Evaluation, using flight two-axis gimbals, of the solution to a line-of-sight jitter problem with the pointing and control of the HGA.
- o Demonstrate at least 77 mechanism operations to verify the life requirement.

## TESTING RESULTS - PROBLEMS IDENTIFIED

### Phase I - Mechanism Characterization

Key testing in this phase involved nonpowered motion study of the principal mechanism elements at vacuum and cold temperatures. The friction damping elements used to decrease oscillations during speed changes or change of direction proved to be within 5-10% of the desired damping range.

Problem 1: The spacecraft's thermal control system was based on biasing temperatures toward the colder regimes to reduce thermal gradients. Because of this, initial predicted AD mechanism temperatures could be well below -87.2°C (-125°F). This proved to be a problem for the grease plating of Braycote 3L-38RP on the angular contact bearings. The solidification temperature of this lubricant is near -84.4°C (-120°F). The resulting "stiffness" of the lubricant caused unacceptably high bearing torques even though the mechanism would operate to as low as -107°C (-160°F). Because the Braycote lubricant has low outgassing characteristics (a particularly attractive feature for an

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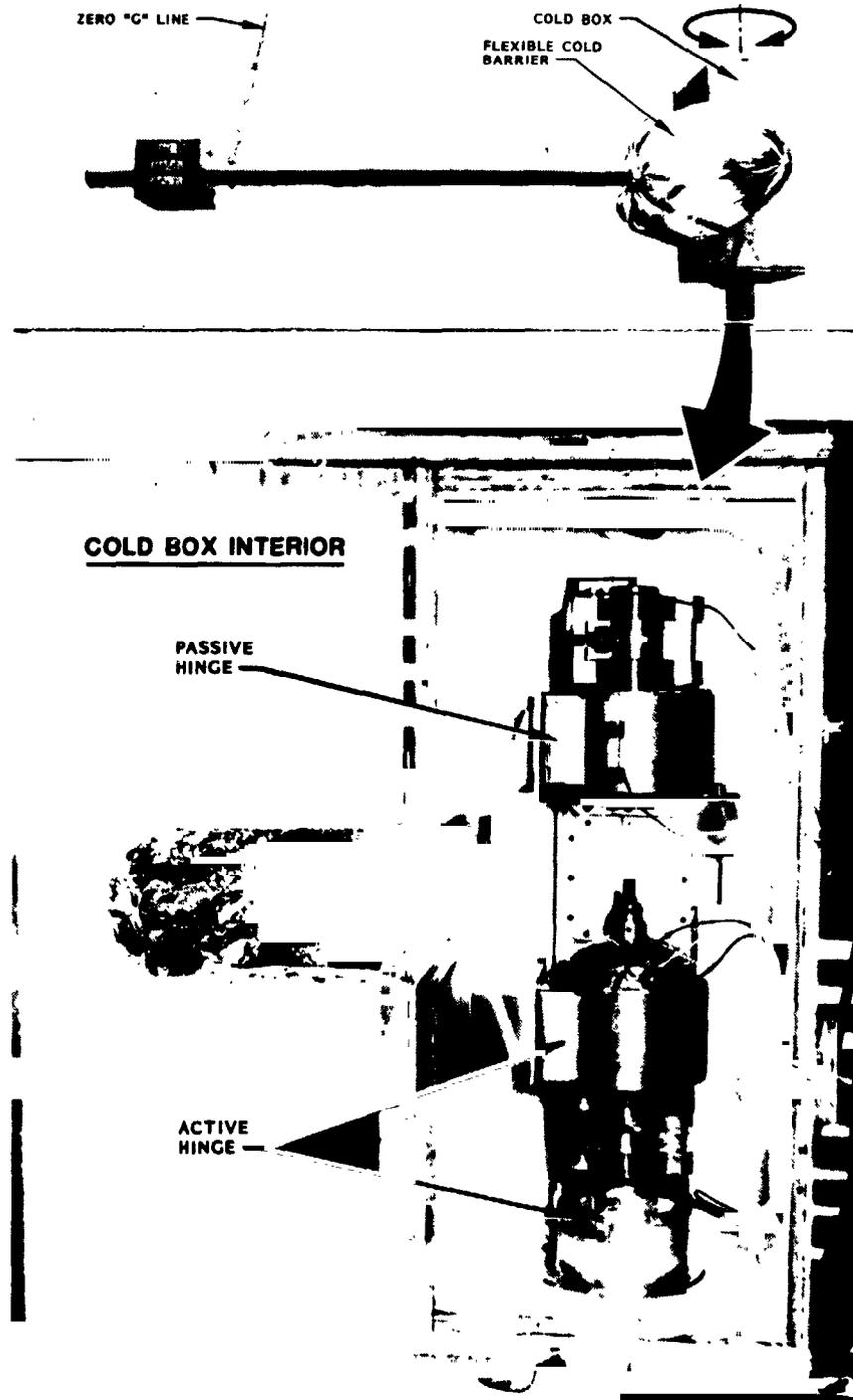


Figure 7. Hinge Operation with Cold Box.

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Figure 8. Vibration Test Setup.

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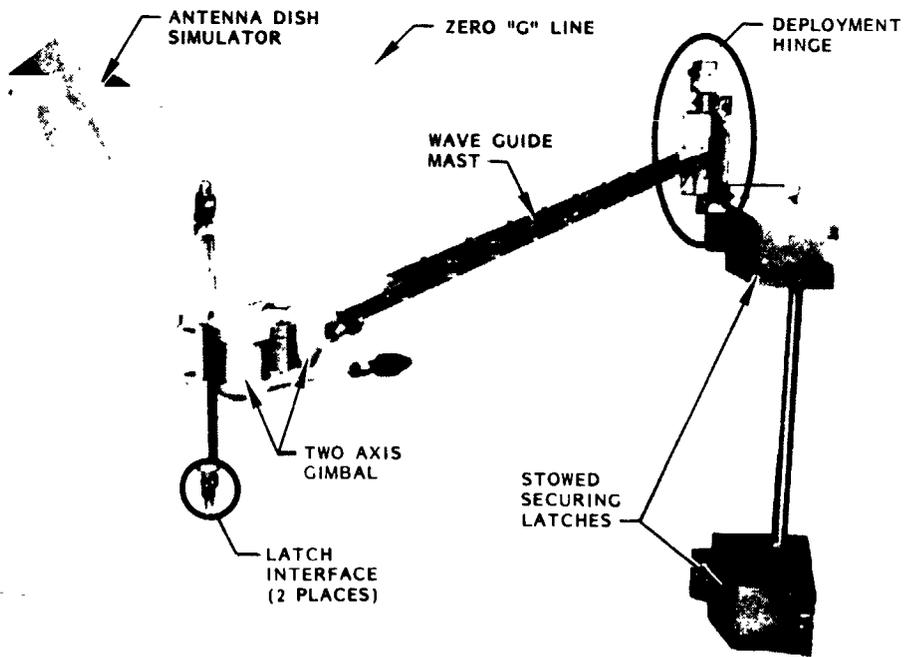


Figure 9. HGA Appendage Test Setup.

optical system) and because of a desire to avoid a lengthy lubrication development program, heaters capable of keeping the mechanism at a temperature of  $-62.2^{\circ}\text{C}$  ( $-80^{\circ}\text{F}$ ) or above were installed to solve the problem. The addition of heaters to both the active and passive hinge mechanisms for the AD assured consistent operation for this important appendage. The HGA mechanism is not fitted with heaters (other than the one on the rotary drive actuator) since predicted cold operation is  $-47.2^{\circ}\text{C}$  ( $-53^{\circ}\text{F}$ ) or above.

Thermal testing of the self-aligning spherical bearing was needed since (1) it is the redundant rotational feature to the angular contact bearings and (2) no cold-test data were available from the supplier. The cold-testing results were acceptable and the bearing torque contribution was factored into the overall mechanism torque margin.

Likewise, a cold test of the HGA power and TLM data cable, routed across the mechanism along with the rf coax from the mast/waveguide, showed acceptable parasitic torque contributions. This series of tests sensed the torque contribution of the cable routing, as a function of temperature, over the full angular travel range of the mechanism.

#### Phase II - Hinge Mechanism Operation

This series of tests involved mechanism operation over the full range of travel at hot and cold temperatures and with the test specimen connected to inertia simulators representing both the HGA and the AD. The mechanism rotational axis was mounted vertically and the additional weight was supported by a zero "G" line pivoting in line with the mechanism rotational centerline. A cold box was constructed around the mechanism (Fig. 7); the rotary drive actuator was kept outside the test environment so the mechanism operation contributors could be identified directly.

This test phase permitted verification of the analytical model as well as providing a test bed for a two-axis, static-proof test and a deploy-direction, dynamic-proof test. These tests at ambient temperature were preceded and followed by functional operation and no anomalies were identified.

The deployed and preloaded natural-mode frequency of the mechanism was verified during this test phase to be 0.875 Hz, below the 1.0-Hz maximum.

The deployed or open position of the mechanism was also verified during this test phase and was shown to be repeatable within 0.033 deg, well within the AD position accuracy needed of  $\pm 0.25$  deg.

The final phase of this test series centered on a three-axis vibration test using the mechanism configured for the HGA usage (Fig. 8). The mechanism was connected via a short tube to a pivoted mass simulating one of the two-axis

gimbals. The assembly was then secured by flight-like development latches. The antenna dish was also represented by a mass simulator. The rotary drive actuator was monitored during the test to sense unacceptable mechanism motion and none was detected during all three orientations of the vibration test. An ambient functional test, performed after each axis was tested, revealed no damage to the mechanism.

### Phase III - Subsystem Testing

This test phase addressed, in particular, the HGA configuration and its deployment and restow. Figure 9 shows this setup which included the preproduction mast/waveguide, a foam-core simulation of the antenna dish, and the pair of securing latches. The configuration was initially fitted with nonpowered gimbal simulators; these were later replaced with flight gimbals for the last segment of this test phase.

Problem 2: The first portion of this test sequence was plagued by a problem that was finally diagnosed as "lost motion", which resulted in variable performance at the stow position. As the mechanism drives into the stow position, it must be positioned correctly for the latches to "capture" and secure the appendage. This involves also tripping four telemetry switches which indicate to the ground station when the latches may be closed, a separate operation. Although always within latch "capture" range, the inconsistent performance needed correcting.

"Play" in the keyway tying the torsion rod to the domed aluminum cover was determined to be the cause, and once corrected by small shims in the keyway, consistent and reliable results were demonstrated. A redesign, however, was initiated to effect a long-term solution to compression yielding of the aluminum portion of the cover interface brought on by numerous manual deployments and stowings. The solution selected was to provide a conical or tapered fit-up of the 15-5 Ph stainless steel torsion rod to the aluminum domed housing. The tight friction fit in conjunction with the keyway assures a "no-lash" joint. (See Reference 1 for an in-depth discussion of this design feature.)

Because the Hubble Space Telescope is designed to be compatible with the Shuttle crew members, it was important to demonstrate and verify crew member interfaces, including manual deployment and restow of the appendages. Two crew member foot restraints are located to permit lifting or pulling on the long mast/waveguide for manual deployment or stowage. If a jettison of the appendage should be required, one crew member must first disconnect the wing tab electrical harness connectors, then back off the two bolts securing the jettison clamp-halves, while another crew member pushes the appendage away from the spacecraft. All of these interfaces were verified and were shown to function satisfactorily.