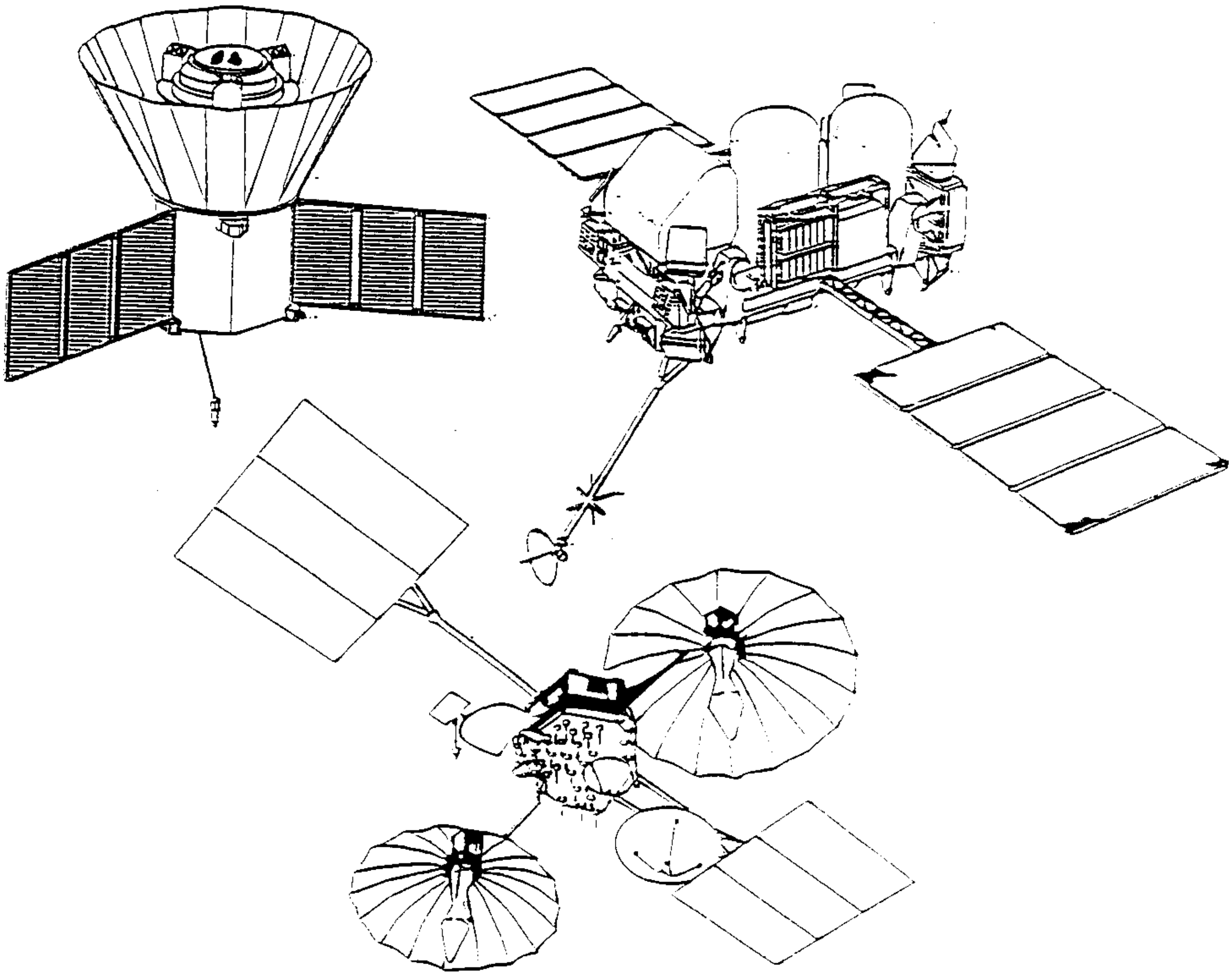
GODDARD SPACE FLIGHT CENTER

ENGINEERING DIRECTORATE

SPACECRAFT

DEPLOYABLE APPENDAGES



May 1992

Preface

The successful operation of mechanically deployed appendages for spacecraft managed by the Goddard Space Flight Center has always been one of the highest concerns associated with mission success. The on-orbit operation of these devices represents a single point catastrophic failure which in many cases could jeopardize the success of the entire mission. The development of such systems demands the technical respect of senior level engineers. Over the past 25 years, Goddard has developed the in-house expertise to design, analyze, fabricate, test, and successfully operate deployed appendage systems. This has been reflected in the outstanding success rate Goddard has enjoyed for almost every spacecraft we have launched. The knowledge which provides this expertise comes mainly from the time invested and experience of individuals who have developed appendage systems for Goddard programs over the years. One of the biggest threats, though, we have to our future success is retaining the savvy and expertise we've worked so hard to achieve. Through the efforts of Code Q and the Engineering Management Council (EMC), I believe this paper will serve as a contribution to the agency's knowledge capture program and will assist the NASA centers to maintain the highest success rate possible in the development of deployed appendage systems.

Thomas E. Huber
Director of Engineering

Acknowledgment

We are grateful to each of the authors and proof readers whose outstanding contributions form the basis of this paper. Jim Metzger, Ralph Mollerick, Peg Luce, John Young and Bill Leavey, each either a member or closely associated with the Mechanical Engineering Branch, have provided a valued look back at the deployed appendages associated with four of Goddard's most successful programs launched in the past 15 years. Rajeev Sharma, Tim Tyler, Claudia Woods, and John Sudey each of the Electromechanical Branch and Ed Devine of Swales and Associates Inc. (SAI) made significant contributions in the area of design philosophy for deployed appendages and mechanisms. Minh Phan and Bob Coladonato provided excellent contributions in the areas of pyrotechnic initiators and appendage verification testing, respectively. A special thanks goes to Rodger Farley of the Mechanical Engineering Branch who made significant contributions to both the historical section on the Cosmic Background Explorer (COBE) and the Design Philosophy section.

Steve Brodeur, Head of the Mechanical Engineering Branch, and Henry Price, Engineering Directorate Chief Engineer (retired), provided excellent technical editing support when it was really needed. Thanks also to Bob Bruff of SAI for his fine word processing and illustration support.

Finally, we deeply appreciate the financial support from Dan Mulville of the Technical Standards Division, Office of Safety and Mission Quality at NASA Headquarters.

Ken Hinkle

ACRONYMS

AC	Alternating Current
ACE	Attitude Control Electronics
CAD	Computer Aided Design
COBE	Cosmic Background Explorer
CRAF	Comet Rendevous Asteroid Flyby
CVCM	Collected Volatile Condensed Materials
DC	Direct Current
DE	Dynamics Explorer
DIT	Differential Inductance Transducer
EHD	ElastoHydroDynamic
EMF	ElectroMotive Force
EP	Extreme Pressure
ERBS	Earth Radiation Budget Satellite
ESSA	Electronic Switching Sperical Array
ETU	Engineering Test Unit
EVA	ExtraVehicular Activity
FMECA	Failure Modes Effect and Critical Analysis
GDA	Gimbal Drive Assembly
GDE	Generalized Differential Equation
GRO	Gamma Ray Observatory
GSFC	Goddard Space Flight Center
HGA	High Gain Antenna
HOP	High Output Paraffin
ICD	Interface Control Drawing
IM	Instrument Module
ISEE	International Sun-Earth Explorer
I&T	Integration and Test
IUS	Inertial Upper Stage
LED	Light Emitting Diode
MTM	Mechanical Test Model
NASA	National Aeronautics and Space Administration
NEI	Non-Explosive Initiator
NRL	Naval Research Laboratory
PFPE	PerFluroPolyalkylEthers
PWI	Plasma Wave Instrument
RCS	Reaction Control System

ACRONYMS
(continued)

RF	Radio Frequency
RMS	Remote Manipulator System
SA	Solar Array
SAA	Single Access Antenna
SAC	Single Access Compartment
SAI	Swales and Associates Incorporated
SARDJA	Solar Array Retention, Deployment and Jettison Assembly
S/C	SpaceCraft
SGL	Space Ground Link
SSPP	Solar Stellar Position Platform
TDRS	Tracking and Data Relay Satellite
TML	Total Mass Loss
TRMM	Tropical Rainfal Measuring Mission
UARS	Upper Atmosphere Research Satellite
UV	UltraViolet
XTE	X-ray Timing Experiment
ZEPS	Zenith Energetic Particle System

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1.0 INTRODUCTION [K. Hinkle/731]

The GSFC has had the prime responsibility for managing the development and on-orbit operation of communication and data gathering satellites for over 30 years. Each of these programs has been highly dependent on the successful operation of deployable appendages which are commonly used to properly position communication antennas and SA power panels. During the past 20 years, the GSFC has enjoyed a high success rate for performing this critical operation soon after each satellite achieved proper orbit. A major reason for this success has been the continuing technical contributions of the GSFC Engineering Directorate in the development of each deployment system. Over the years, the various disciplines within the Engineering Directorate have developed the capability to properly design, analyze and test every aspect of a deployment system such that a high degree of reliability for success can almost be guaranteed.

The purpose of this paper is to document the reasons for the outstanding success rate achieved by these GSFC managed programs. The paper will explore the design aspects, thoughts, experiences and rationale used by some of the leading experts within the Engineering Directorate who have developed such systems in the past and who are presently designing systems for future programs such as the X-ray Timing Experiment (XTE) S/C and the Tropical Rainfall Measuring Mission (TRMM) S/C. The paper also discusses the development of deployment systems engineered by several prime contractors which were managed by the GSFC Flight Projects Directorate. In these situations, the Engineering Directorate had a major role in the design, analysis, test and review of each system to assure successful operation.

The development of each deployment system for space flight applications is a lengthy and costly venture. Although the GSFC has suffered no complete failures of a deployment system in the past, we have not been without our share of minor hang-ups and glitches which have caused temporary, but serious, concerns. In several cases, the presence of an astronaut has averted catastrophic losses while in others, critical maneuvering synergized by intelligent real-time engineering corrected several mission threatening hang-ups. The lessons learned from each of these events, as well as those learned from the vast amount of deployment testing performed by the Engineering Directorate at the GSFC, will be discussed.

2.0 GODDARD SPACE FLIGHT CENTER (GSFC) DEPLOYED APPENDAGE HISTORY

The following section provides a general description of recent satellite programs managed by the GSFC. Special attention is given to the deployable subsystems.

2.1. DYNAMICS EXPLORER (DE) S/C [J. Metzger/731]

The DE payload was a 1981 Delta vehicle launch and consisted of two S/C, the DE-A and the DE-B joined by an 1809-A separation adapter. The 1929 lbs payload included a total of 375 lbs. distributed among 15 instruments, many of which had deployable features. Of particular interest is the DE-A S/C which was spin-stabilized and carried the PWI. The deployables for this instrument were four radial long wire antennas; two spin axis stem antennas; and a group consisting of a search coil, short electric antenna and a one square meter loop antenna. This group was mounted to a single plate and deployed to a radial distance of six meters by means of an Astromast boom. Due to its physical size, the loop antenna was folded and secured along the side of the S/C body to fit within the launch vehicle fairing. Consequently, successful deployment of the loop was required before extension of the Astromast could commence. A similar boom carrying a magnetometer was located on the opposite side of the S/C and provided a counter-balancing effect. These deployments were mission critical and appropriate analysis and testing was conducted to demonstrate design adequacy and reliability.

The design of these deployed appendages was conducted primarily with hand calculations, with some Computer Aided Design (CAD) simulations for verification. Much of the hardware design and methodology was based upon the then recent International Sun-Earth Explorer (ISEE) experiences with boom deployment and the new requirements involving Astromast features and a large rigid loop. In order to accommodate this device, the stowed position of the loop was compared to the deployed position and the intersection of the two planes represented the hinge axis for initial deployment of the loop. Due to a non-integral number of turns to deploy the Astromast, which rotates as it deploys, combined with a non-perpendicular alignment of the loop with the boom axis, the hinge axis was placed at a skew angle. A coil type deployment spring was included for the loop to ensure sufficient energy for both initial motion and lock-up under a wide range of S/C spin rates. This spring was sized using energy principles to also account for loop strain at lockup, flexing of electrical cables and friction where only dry-film lube was used. Lockup was achieved by bending a leaf spring which contained a cutout to capture a detent on the moving part of the hinge and also helped to reduce the shock loads. With deployment possibly occuring between 0 and 30 revolutions/minute, the worst case stresses were calculated and showed a margin of safety of 1.5 against yielding for the loop.

For hardware qualification, a series of tests were conducted using a relatively high fidelity Mechanical Test Model (MTM), which included most protoflight structure and mechanisms.

The structural dynamics test sequence consisted of a modal survey, several sine vibration runs, an acoustics test, pyro shock/separation and qualification tests for all the mechanisms. Thermal-vacuum testing was also conducted. The PWI loop was designed to be handled and installed in a 1-G environment, but the effect of gravity would distort the loop in the deployed configuration with the S/C spin axis in the normal upright position. Since the flight deployment would take place with a spinning S/C and a radial acceleration field which would be equivalent to slightly less than 1-G, it was decided to tilt the MTM on its side and test the loop downward with an approximately 1-G radial force field. This provided 20% margin over the maximum design spin rate for deployment of 30 revolutions/minute and eliminated the need for any special G-negation equipment. The testing was successfully completed with no significant problems reported and no design changes were needed.

Following launch of the S/C, both loop and Astromast were successfully deployed according to plan and no anomalies were reported. The key philosophy in this effort was to keep things as simple as possible and to draw upon design experience gained from previous programs.

2.1.1. DE Plasma Wave Instrument (PWI) Loop Antenna

Once the DE-A S/C was placed in orbit, the various instrument deployments took place. Among these was the PWI loop antenna release which occurred, nominally, at a spin rate of 25 revolutions/minute. The antenna was preloaded to enhance the initial motion and pyrotechnically released. The motion terminated against a hard stop and end-of-travel was indicated by the actuation of a microswitch. A latch prevented backlash. The nominal radial acceleration during deployment ranged from 0.5 G to 0.9 G, due to spin rate. The stub boom and loop antenna are shown in Figure 1 as they appeared prior to Astromast extension.

2.1.1.1. PWI Antenna Deployment Test Description

A Ransome table was used to place the S/C in the horizontal position with the stub boom oriented straight down so that the loop antenna would experience 1-G acceleration during its free fall deployment. Although deployment was expected to take place at a spin rate of 25 revolutions/minute, it was designed for at least 30 revolutions/minute. Justification for such a free fall deployment lies in the fact that the energy absorbed by the loop in this test was calculated to be 1.2 times the energy during a deployment at 30 revolutions/minute. This represents a 20% overtest for design qualification. In addition, a stress analysis on the loop design showed a margin of safety of 51% for the 1-G deployment.

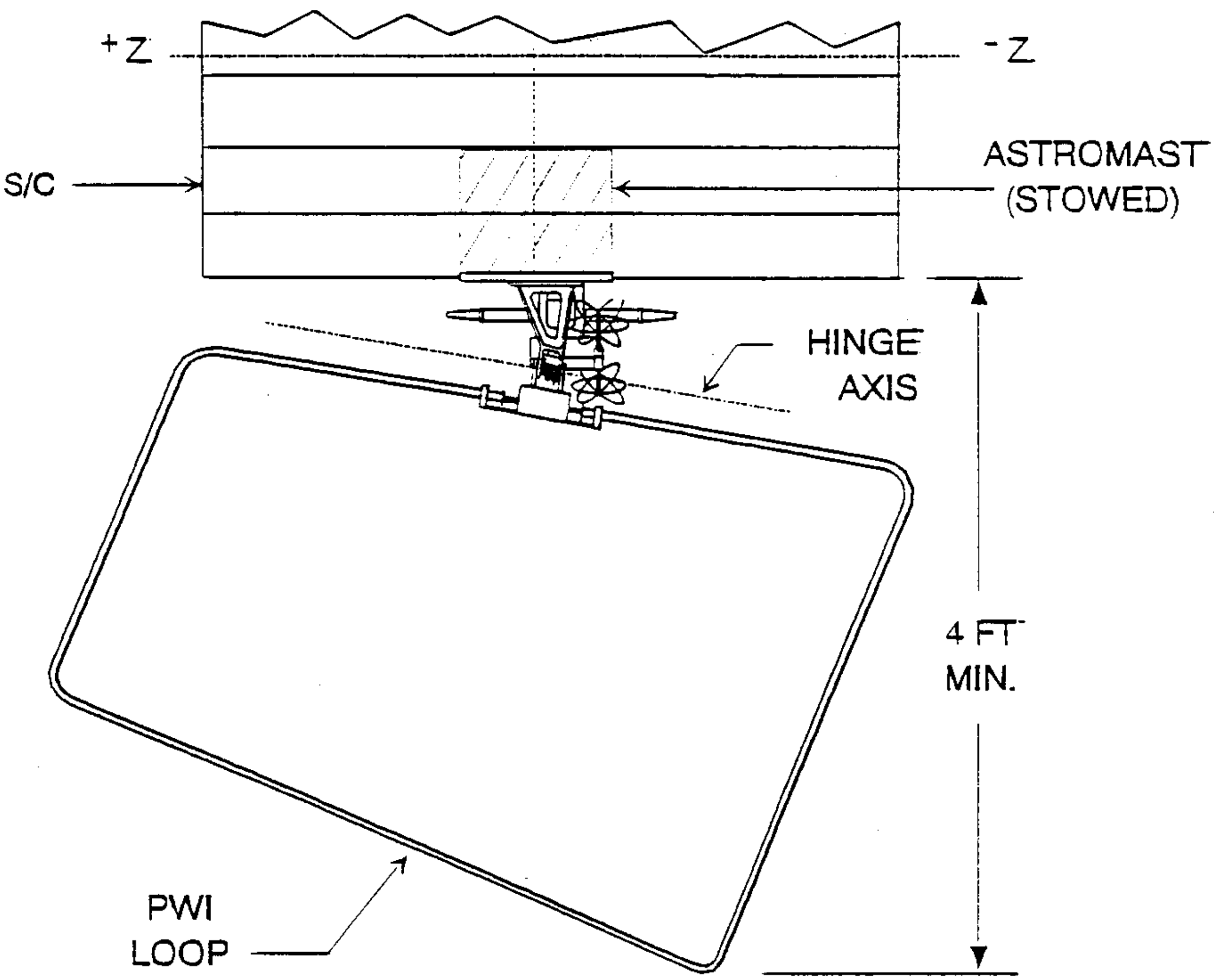


Figure 1. PWI Loop.

The deployment was initiated by firing one of two pyrotechnic dimple motors and the motion was recorded on high speed film. The test set up is shown in Figure 2.

2.1.1.2. PWI Antenna Deployment Test Results

Prior to installation, the weight of the PWI loop and preamp assembly was measured at 895 grams (1.97 lbs). The loop antenna was released and deployed in 543 msec with microswitch closure occurring approximately 30 msec prior to lock up. The predicted deployment time was 423 msec and the difference can be attributed to hinge friction, air drag on the loop and stiffness of the electrical cabling.

2.1.2. DE Astromast

Two diametrically opposed Astromasts were deployed nearly simultaneously on the DE-A S/C. The deployment began at a S/C spin rate of approximately 25 revolutions/minute. It took place in several steps, with a spin-up maneuver occurring at some intermediate position. Each Astromast achieved its full length of 20 feet at a S/C spin rate of 10 revolutions/minute. The centrifugal acceleration experienced by the tip mass, tending to extend the Astromast, ranged between 0.5 G and 0.8 G depending upon the spin rate and length of mast deployed. Final tip position and alignment were required to be within $+/-0.2^{\circ}$ of nominal. The Astromast was caged for launch and pyrotechnically released for deployment. Since the coiled elements of the mast contained more than enough energy to effect deployment, a motor driven lanyard was used to restrain and control the deployment and even retract it if desired.

2.1.2.1. DE Astromast Deployment Test Description

With the S/C in the horizontal position, the Astromast was deployed straight down with the force of gravity representing the centrifugal force from a spinning S/C and tending to aid the deployment.

The objective of this test was to complete qualification of the PWI loop and stub boom hardware by demonstrating compatibility with the Astromast and its unique deployment dynamics. Specific areas of interest included clearance between the PWI short electric antenna and S/C during initial Astromast motion.

The entire deployment was photographed at normal speed from two camera angles. The test set-up is shown in Figure 3.

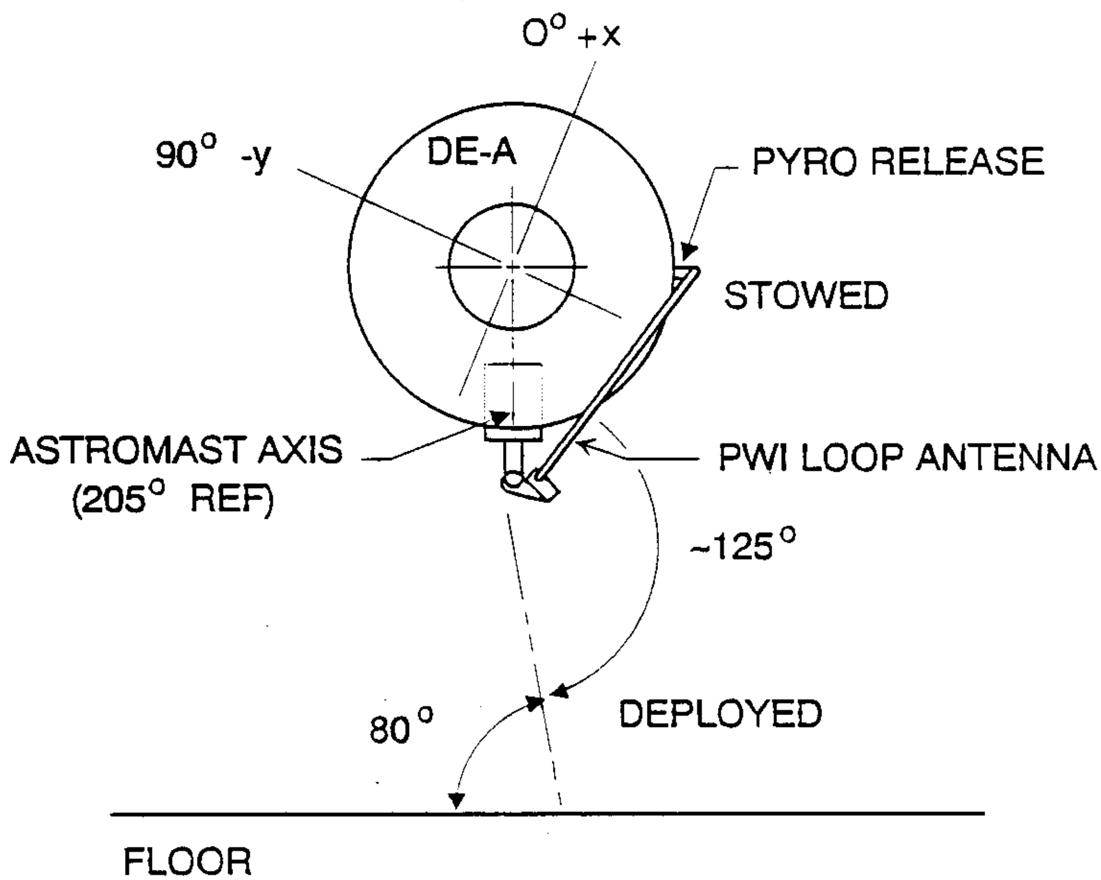


Figure 2. PWI Loop Deployment Test Configuration.

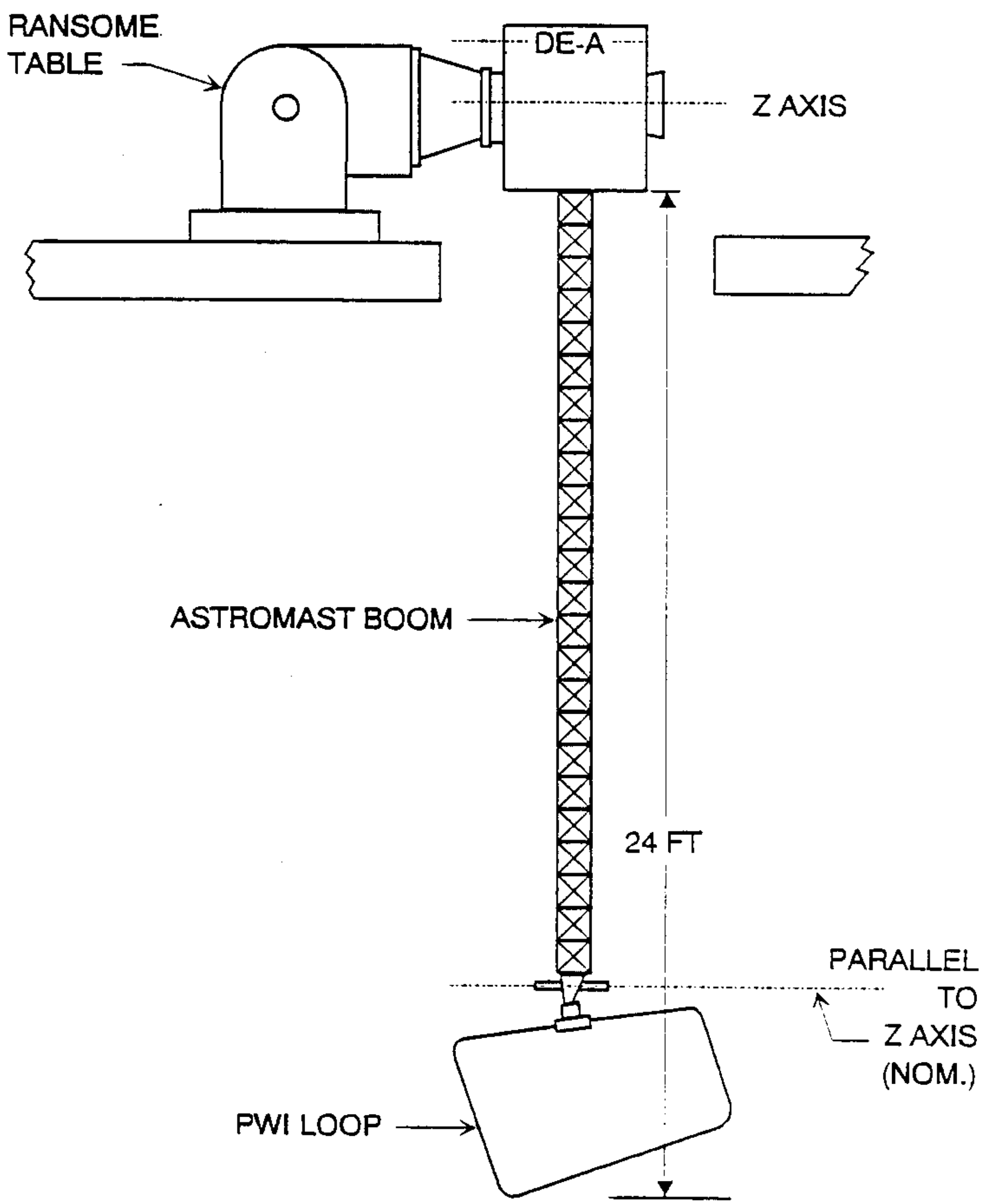


Figure 3. Astromast Deployment Configuration.

2.1.2.2. DE Astromast Deployment Test Results

Observation of the PWI instrument group (attached to the end of the Astromast) during each step of the deployment revealed no compatible problems. It was noted, however, that the final tip orientation was rotated approximately 1^{o} counterclockwise from its normal position. Although this was a crude measurement, it did warrant further investigation for the flight units to ensure proper final instrument alignment. It was also noted that the deployed-length potentiometer displayed voltage changes opposite to those expected from the given schematic and also warranted further investigation. One significant anomaly was encountered during this test when the end-of-travel shut off switch failed to actuate. The mast was observed to reach full extension but drive power did not terminate automatically. Power was controlled manually for an additional 12 sec in an unsuccessful attempt to produce switch actuation. A continuity check verified that the dual switch had remained closed. In order to avoid the loss of the lanyard, no further extensions were attempted. Retraction of the mast was accomplished without incident. Following this test, the Astromast was removed from the S/C and returned to the manufacturer for investigation of the switch problem.

2.2 EARTH RADIATION BUDGET SATELLITE (ERBS) [R. Mollerick/731]

The ERBS, launched in October of 1984, is an earth looking S/C with three scientific instruments (ERBS-S, ERBS-NS and SAGE-II) in a circular orbit of 600 km altitude and 57^{o} inclination. The 2250 kg (4960 lbs) shuttle payload was determinately mounted in the cargo bay using three longeron trunnions and one keel trunnion. The S/C is illustrated in Figure 4 and is composed of a base, instrument, and keel module.

There were two deployable subsystems on the ERBS S/C. These included an ESSA antenna and two SA's. Both systems use similar deployment drives and release mechanisms. The drive system, illustrated in Figure 5, uses redundant torsion springs and a strike arm that engages a precrushed honeycomb damper and lock system at the end of deployment travel. The release mechanism, illustrated in Figure 6, uses redundant Non-Explosive Initiators (NEI's) and is designed to release the appendage with either NEI.

The ESSA antenna is stowed between the SA's as illustrated in Figure 7, detail A. The antenna is supported by a boom that pivots through approximately 90^{o} to full deployment and engages a lock.

The SA's are stowed on each side of the keel module as illustrated in Figure 7. Each array is supported by a keel box beam and redundant torsion spring drive systems at each end.

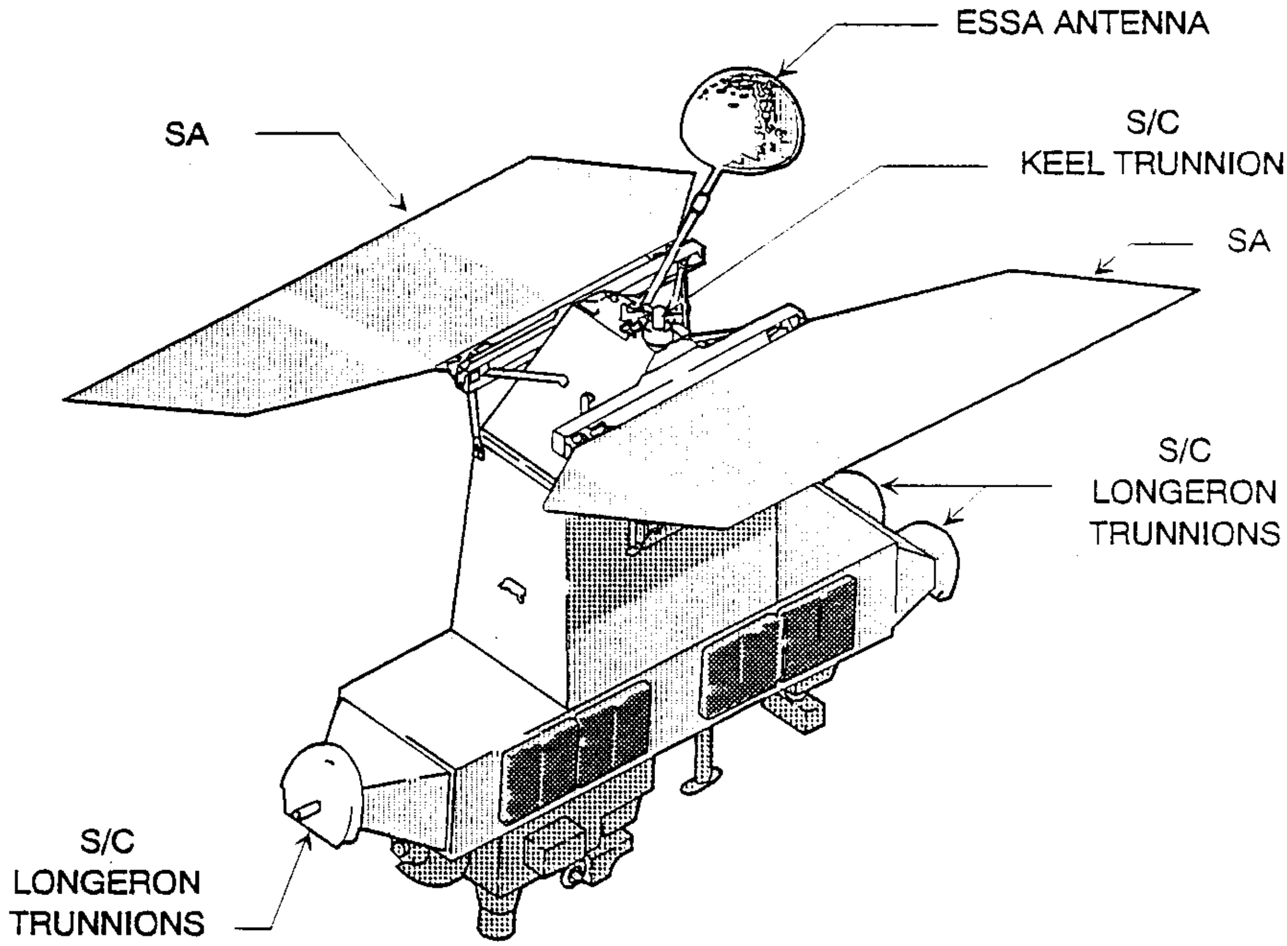


Figure 4. ERBS S/C Configuration (deployed).

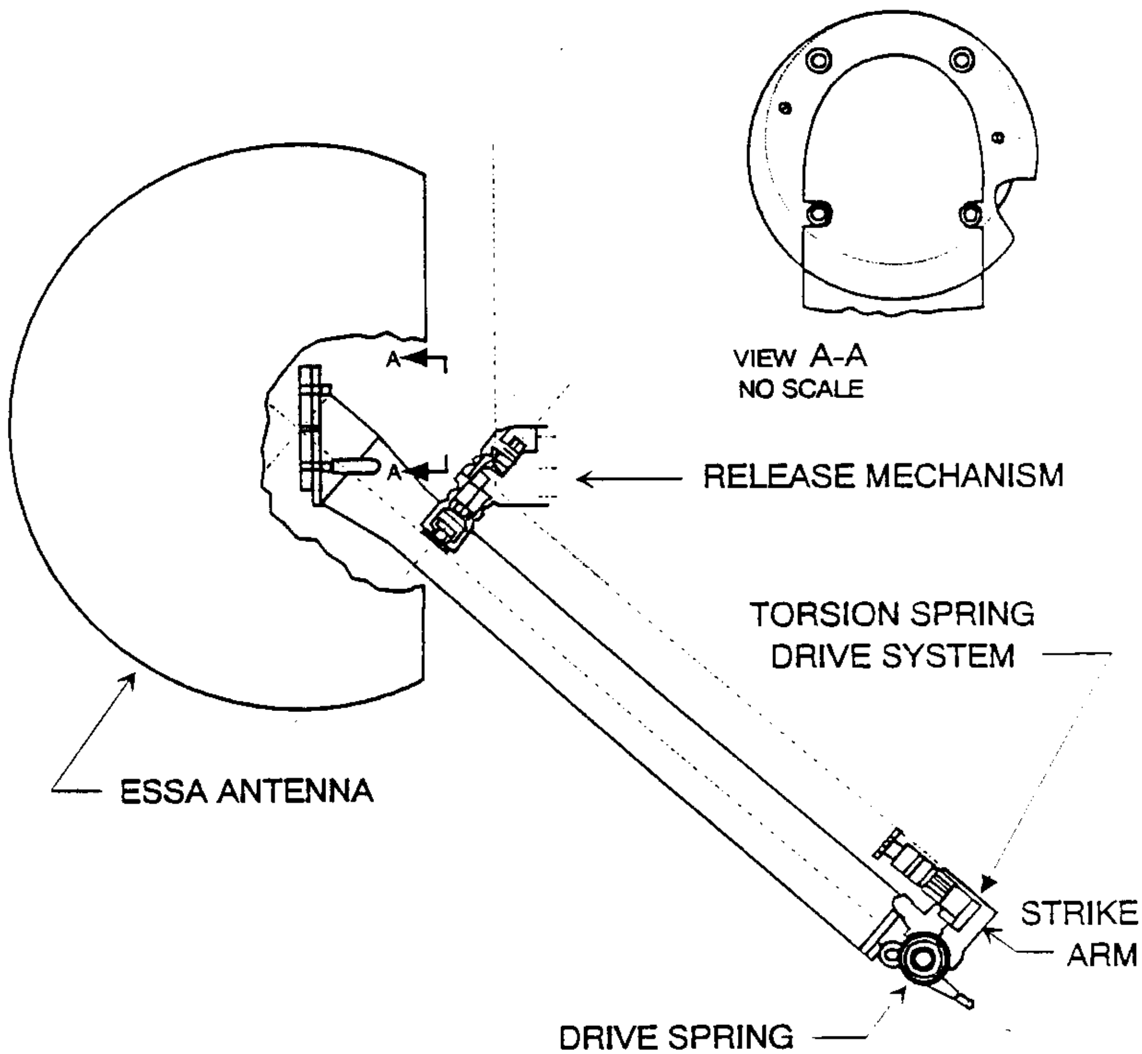


Figure 5. ERBS Release Mechanism and Torsion Spring Drive System (ESSA Antenna).

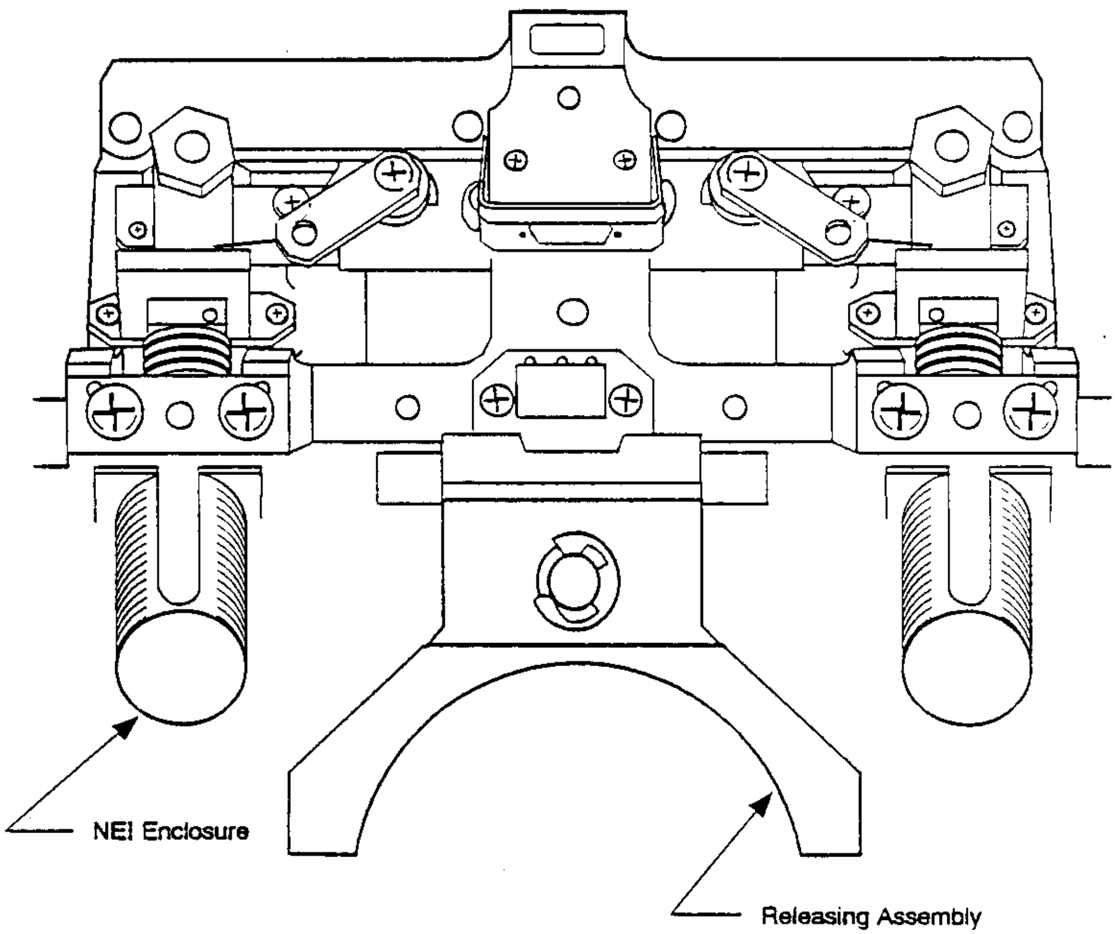


Figure 6. ERBS Release Mechanism.

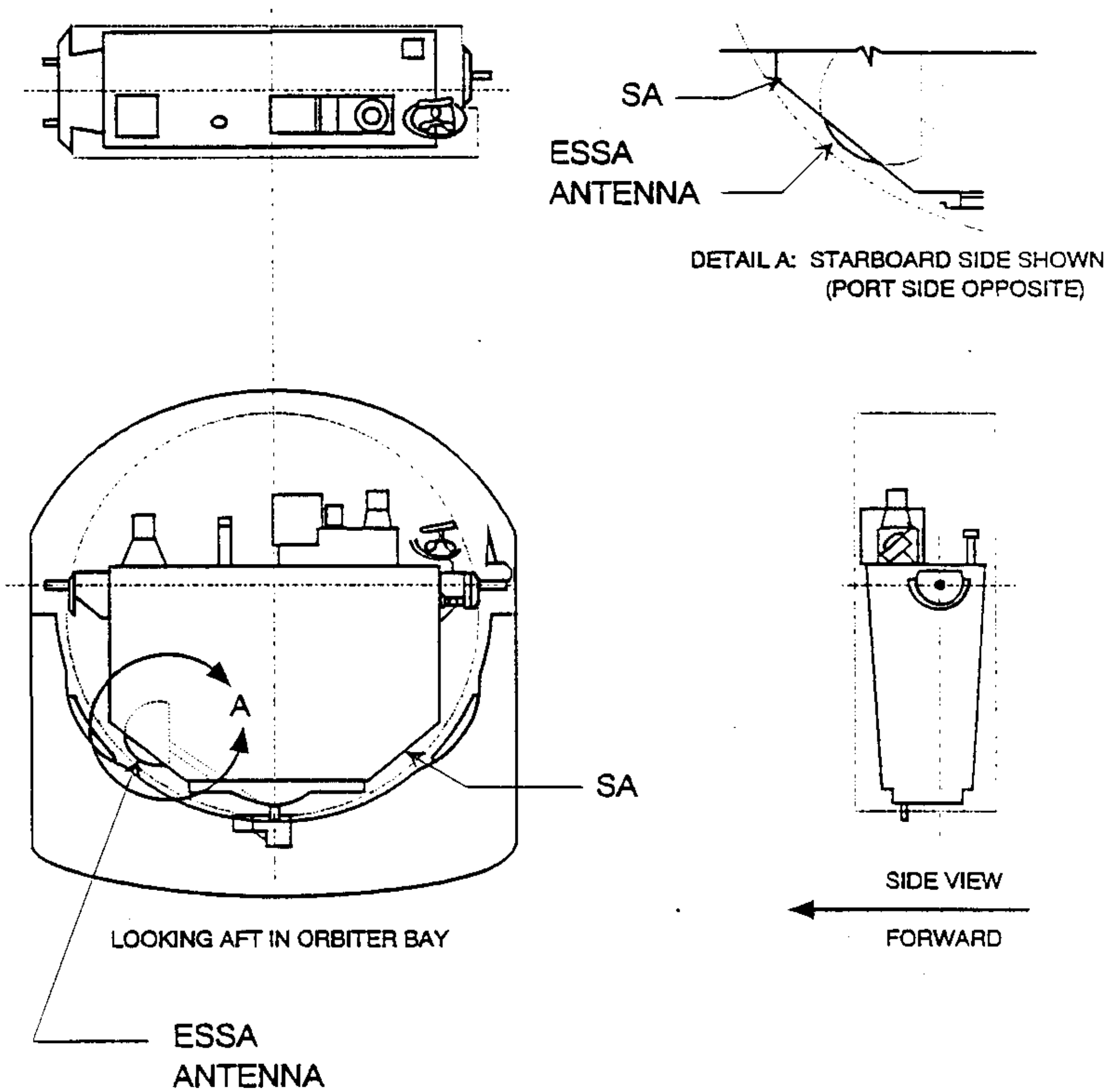


Figure 7. ERBS Launch Configuration.

The arrays rotate through 65^{o} and 115^{o}, respectively, at full deployment and engage the respective locks.

2.2.1 ERBS Deployed Appendage Test Objectives and Approach

The test objectives for the deployables were to demonstrate release and deployment under g-negation and to characterize appendage dynamics during hot and cold temperature extremes. Testing of each appendage was configured with flight release and drive mechanisms and with mass simulated arrays and antenna. Rotation of each appendage was around a vertically oriented hinge-line to minimize gravity effects. Release and drive mechanisms were instrumented to monitor all motion. Temperature extremes were achieved, although poorly controlled, with heaters for the hot case, and liquid nitrogen vapors for the cold case. The tests were performed by the contractor in a hi-bay facility with makeshift wrap-around plastic bags for the mechanisms to help achieve temperature goals. The bags allowed freedom of motion for full appendage deployment.

2.2.2 Deployed Appendage Test Results

Thermal vacuum test results indicated that conditions from room temperature to hot were easily achieved with repeatable data. The steady state cold case was difficult to achieve because of icing on the components and poor temperature control due to the makeshift test setup. The predicted cold case test temperatures of -49^{o} and -55^{o} C for the sun and anti-sun sides of the arrays, respectively, were not achieved under the described conditions. The coldest obtainable test temperature was -33^{o} C. The on-orbit monitored temperature in the center of the array box beam (sun side) was -44^{o} C.

2.2.3 ERBS Deployed Appendage Lessons Learned

The major lessons learned with the ERBS deployed appendages can be summarized in the following five areas.

1. Insist on the contractor providing sufficient data to verify satisfactory compliance of the test objectives. In this case, makeshift test setup conditions were inadequate to control or achieve temperature extremes.

2. Convenience, speed and cost savings are not always the best criteria for test purposes. Certainly the hi-bay turned out to be a convenient facility for the test setup. Quick and inexpensive techniques for temperature control appeared workable. The fact that a suitable chamber was not available in time for the test made all of the above choices

desirable. In retrospect, a humidity chamber of the right size and with high and low temperature control was necessary.

3. Insist on testing under realistic thermal conditions. Deployable appendages and associated mechanisms are some of the most difficult subsystems to accurately model and test because of numerous thermally conductive and non-conductive interfaces, when using, for example, rolling and point contact surfaces of bearings. Predicting bulk temperatures went well, although temperature gradients for the arrays were neither addressed nor tested. Temperature gradients between sun illuminated and shadowed structure are likely to contribute adversely during deployment.

4. Molybdenum disulfide (MoS_2), while a good dry-film lubricant for space applications, can also be a nemesis for failure. Case in point: each of the deployment drive mechanisms have bearings lubricated with MoS_2. It is known from the literature that MoS_2 has an affinity for moisture. It is not unreasonable to expect that under conditions of cold temperatures (-44^{o} C) and the balling-up phenomenon of MoS_2, moisture molecules could create frozen balls in the path of the rolling elements to impede available driving torque. In fact, this is the major contributor believed to have prevented the SA from initially deploying while the S/C was attached to the orbiter Remote Manipulator System (RMS). Fortunately with man in the loop, we were able to rotate the SA hinge-line into the sun and monitor the deployment drive unit temperature with a box beam thermistor located close to the drive units. Deployment occurred when the temperature climbed through 0^{o} C.

5. Spring drives must use a torque ratio of four as a minimum. The deployment drive systems each had a torque ratio of less than three. However, each drive had redundant torsion springs and analysis indicated ample capacity for successful deployments. In addition, testing indicated only small deployment rate variations between room temperature and hot and cold extremes. In looking at the ERBS SA deployment, it is conceivable that a combination of cold temperatures, thermal gradients, MoS_2 lubricant and insufficient torque margins are likely to result in deployment problems.

2.3. TRACKING AND DATA RELAY SATELLITE (TDRS)

[P. Luce/731, John Young/SAI]

The National Aeronautics and Space Administration (NASA) currently has four TDRS's in geosynchronous orbit. Two more S/C are being built by TRW to provide the system with additional redundancy. This constellation of satellites has dramatically increased the data rates and accuracy of communication between S/C in near earth orbit (including the shuttle) and ground stations.

The TDRS's weigh nearly 5,000 lbs each, including 1,200 lbs of hydrazine propellant used as needed for station keeping. They contain the most complex set of deployable appendages ever launched by NASA. The S/C is shown in the stowed configuration in Figure 8 and in the deployed configuration in Figure 9.

Mounted to an Inertial Upper Stage (IUS), TDRS is launched on the shuttle. A spring-loaded ejection system deploys the TDRS/IUS stack and the first burn of the IUS booster takes the TDRS to geosynchronous orbit. The second and final burn circularizes this orbit.

2.3.1. TDRS Appendage Deployment Sequence

Upon reaching geosynchronous orbit, and while still attached to the IUS, the deployment of antennas and appendages is started. The deployment sequence is as follows.

1. Deploy SA's.
2. Deploy SGL boom and antenna.
3. Deploy omni/C-band boom.
4. Separate TDRS from the IUS second stage.
5. Release SAA booms.
6. Position SAA's.
7. Open SAA's.

The details associated with each of these deployments are as follows.

2.3.1.1. TDRS Solar Array (SA) Deployment Sequence

The SA's consist of six large panels that form a hexagonal enclosure around the S/C body in the stowed configuration. Structurally there are two separate SA assemblies. Each assembly consists of a center panel and two outer panels.

Snubber fittings at two levels hold the outer panels to the main S/C body along the two lines on opposite sides of the S/C where the outer panels from each assembly come together. These outer panels are released by the activation of bolt cutters. Bolt inertia causes the snubber fittings to be ejected which in turn allows a pair of rods to be withdrawn releasing the four outer panel tips. Pairs of torsion springs located at each of the four hinges connecting an outer panel to the center panel rotate each wing out 60^{o} forming two sets of flat array assemblies. Microswitches indicate when individual panel motion nears latch-up.

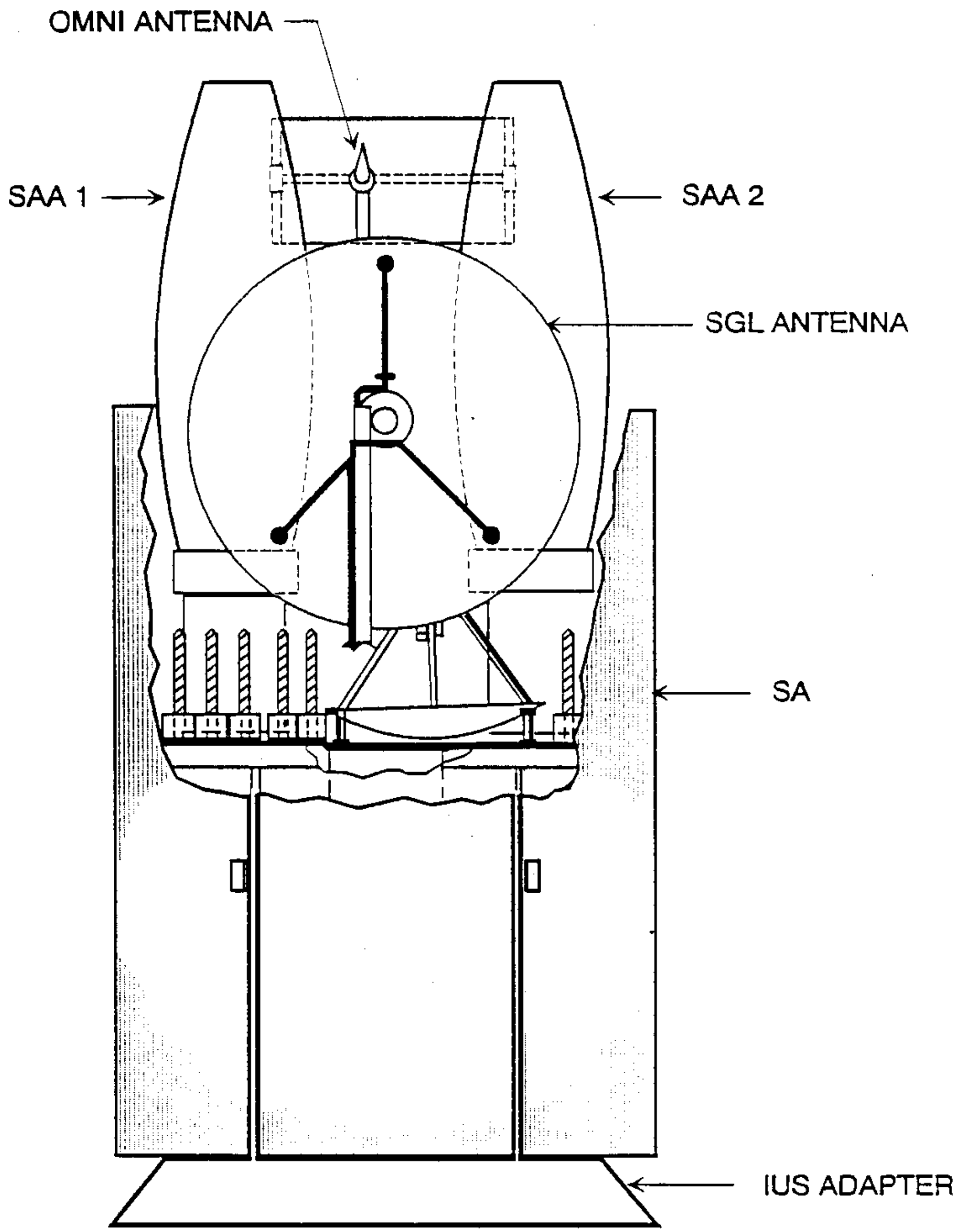


Figure 8. TDRS Stowed Configuration.

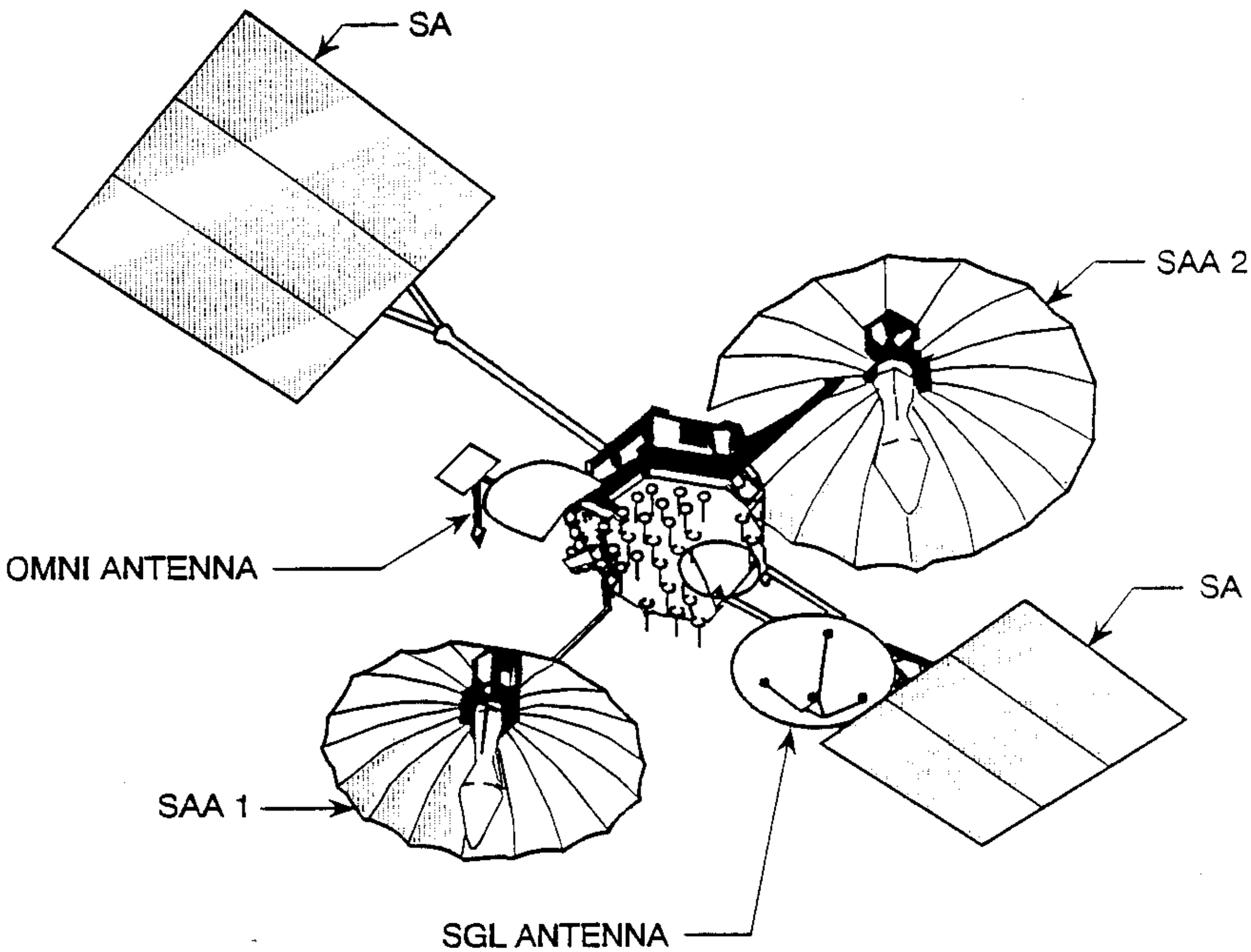


Figure 9. TDRS Deployed Configuration.

The two center panels are then released by activating bolt cutters at four locations on each panel. A kick-off spring aids initial center panel motion. This motion pulls a pin to release the SA boom from its S/C tie-down. The boom has an inboard hinge which deploys 90^{o} and an outboard hinge which deploys 180^{o}. Kinematic cables control the motion, ensuring that the inboard hinge latches first which minimizes load on this hinge. Compressed spring actuators in these cables provide cam controlled torque. A microswitch indicates when the boom inboard hinge is within 5^{o} of latching.

2.3.1.2. TDRS Space Ground Link (SGL) Antenna Deployment Sequence

The deployment of the cantilevered SGL antenna utilizes two motorized hinges, one inboard and one outboard, each protected in the stowed configuration with a pin-puller acting in a lug-clevis retention system. The inboard pin-puller is fired first to release the boom hinge. This hinge provides primary deployment of the appendages and "north-south" gimbal travel. Deployment is verified by a position potentiometer and motor shaft rotation indicator in the Gimbal Drive Assembly (GDA).

The outboard pin-puller is fired after appendage deployment. Its hinge allows "east-west" motion of the SGL antenna.

2.3.1.3. TDRS Omni/C-Band Deployment Sequence

Deployment of the cantilevered omni/C-band antenna boom utilizes an inboard spring-driven hinge that is protected in the stowed configuration with a pin-puller acting in a lug-clevis retention system. The pin-puller is fired to release the hinge. The appendage is driven out by five torsion springs acting in parallel about the hinge axis. The appendage is locked in the deployed position by a pawl and stop. Deployment is verified by redundant microswitches.

2.3.1.4. TDRS Single Access Antenna (SAA) Deployment Sequence

Each TDRS has two large umbrella-like SAA's that mount to an equipment compartment. The Single Access Compartment (SAC) mounts to the S/C both directly with three retaining pin-pullers in the lugs/clevises at the base of the SAC as well as via a long beryllium boom which has a duplex hinge at its S/C interface. The pin-pullers for SAC #1 are fired first. A kick-off spring and the two axis hinge act together to drive the compartment out of the clevises. The two axis hinge continues deployment of the appendage to the full first axis angle of 168^{o}. The second axis is then released by a staging mechanism and the appendage is rotated 68^{o} to line up with the S/C body axis.

2.3.1.5. Positioning TDRS SAA's

The pin-pullers that restrain the SAC's to the booms are fired. This restraint had protected the two GDA's that are located in the boom at right angles to each other near the SAC interface. These GDA's both position and steer the antenna. The SAC's are rotated a small amount (about 10^{o}) by the "north-south" GDA to uncage the restraint brackets. Each SAC is rotated 168^{o} by the "east-west" GDA to point the antenna to the earth.

2.3.1.6. Opening TDRS SAA's

Non-explosive initiators in each antenna are fired to release the 18 ribs that support the parabolic antenna mesh. A dual drive motor unfurls the mesh by driving a ball screw mechanism that displaces a carriage assembly which in turn causes the ribs to pivot about their base. A microswitch on the carriage assembly indicates full deployment.

2.3.2. TDRS Appendage Verification Testing

The following functional capabilities were verified by test

1. Strength and stiffness verified by load deflection testing

2. Margins of safety verified by structural analyses and tests

3. Minimum torque/angle criteria established based on individual and combined resistance torque measurements at worst case temperature cnditions

4. Maximum torque criteria set to ensure that deployment energy remains within safe strength/deformation limits under dynamic latch-up conditions

2.3.3. TDRS On-Orbit Deployment Anomalies

There have been three on-orbit deployment related anomalies, none catastrophic. In fact, none of the anomalies has interfered with normal S/C operations.

2.3.3.1. TDRS-A

The field of view for one of the SAA's was restricted. This was probably due to a pinched or snagged electrical cable which runs across one of the SAA gimbal joints. This restriction, however, was outside the range of normal operations.

2.3.3.2. TDRS-C

The SAA delayed deployment by nearly three hours when one of the compartment attachment lugs came into contact with the compartment kickoff spring mechanism. It freed itself without any action from the ground.

2.3.3.3. TDRS-D

One of the SAA drive motors stalled because the biax service loop harness became pinched between the boom and compartment. The motor was reversed to relieve the pinch and deployment proceeded nominally.

2.4. COBE [R. Farley/731]

2.4.1. SA Appendage Description

The COBE SA consisted of three wings, 120^{o} apart, each with three panels. The three inboard hinges had dampers which were heated with 10 watt strip heaters controlled by thermostats such that there would be minimal temperature variations between the three input dampers. This ensured that the major deployment geometry would be symmetrical. Each hinge line was supported by two hinges, each one with an Elgiloy torsional spring for redundancy, and spherical bearings to allow misalignments due to thermal distortions or manufacturing tolerances. This prevented jamming as well as providing a redundant torsional path for the hinge. The top hinge was connected to a potentiometer (position telemetry) through a U-shaped beryllium copper coupler which would allow large misalignments without affecting either the potentiometer or the deployment. The bottom hinge had the damper. These rotary viscous dampers were modified to use McGhan-Nusil CV7300 silicone fluid which has very stable viscosity characteristics and is a low outgasser. This was a vast improvement over the typical DC 93500 resin which has been used to a great extent in these dampers. The new fluid was relatively inexpensive and has infinite shelf life as opposed to the old fluid which was opposite on both counts. The DC 93500 also was extremely viscosity sensitive with regard to temperature.

2.4.2. SA Latch/Release Mechanisms

This system used the high reliability of pin pullers rather than bolt cutters, producing virtually no particulate contamination upon firing and retained all of the hardware after firing so as to not pollute space with debris which would affect the sensitive instruments on COBE. There were two different types of release mechanisms used on the SA's. The strategy was to structurally decouple the SA panels from the S/C and, thus, prevent them from experiencing launch loads which would otherwise pass through the primary structure. This was accomplished by having each SA wing determinately supported. Each panel was connected at the top and bottom along the centerline with a release mechanism. The upper release mechanism had a conical connecting surface which reacted loads in three translational and two rotational degrees of freedom. The lower mechanism was a V-guide connection which allowed relative motion in the vertical direction, since this was the source of greatest deflection from the structure and therefore the largest potential contributor of "second hand" loading into the panels. Whereas the top restraint required 450 lbs of preload, the lower restraint had essentially none, since relative vertical motion was a necessity. However, they were tightened until there was approximately a 0.003 inch gap in the V-guides to prevent rattling yet preserve its vertical motion capability.

2.4.3. COBE SA Testing/Issues

Hinge line assemblies were tested in a thermal vacuum chamber for independent characterizations and later air pad deployments on the flight S/C or Engineering Test Unit (ETU) tested the system performance. A few problems were encountered at the launch site during final testing. The first was that one of the dampers had to be replaced due to an air bubble in it for reasons that are complicated, but a replacement damper was on hand and the wing was redeployed on the air pad table. Second, during one of the last deployment tests a pin puller shaft fractured and rebounded back into the unfired position. That particular pin had two retaining holes drilled into it where there should have been only one, thereby critically weakening the pin. The SA release mechanism was designed to be redundant in that only one pin puller needed to actuate; however, this failure had the possibility of being a repeated manufacturing problem and so all of the X-rays of each pin puller on the S/C were intensely scrutinized. It was determined that one pin puller was suspicious, so it was removed and replaced with a screened one.

The arrays deployed as planned, with the only anomaly being the minor inconsistent behavior of a microswitch.

2.5. GAMMA RAY OBSERVATORY (GRO) [K. Hinkle/731]

The GRO was launched on 4/4/91 aboard the space shuttle Atlantis from the Kennedy Space Center in Florida. The observatory had two SA wings weighing approximately 500 lbs each and one HGA boom assembly weighing about 525 lbs. The array wings and their stowing latches were identical in construction and the -Y system (wing 1) could be mechanically interchanged with the +Y system (wing 2). The two wings were electrically unique in terms of their motor drive direction, microswitch designations and coarse sun sensor designations. The deployed GRO is shown in Figure 10.

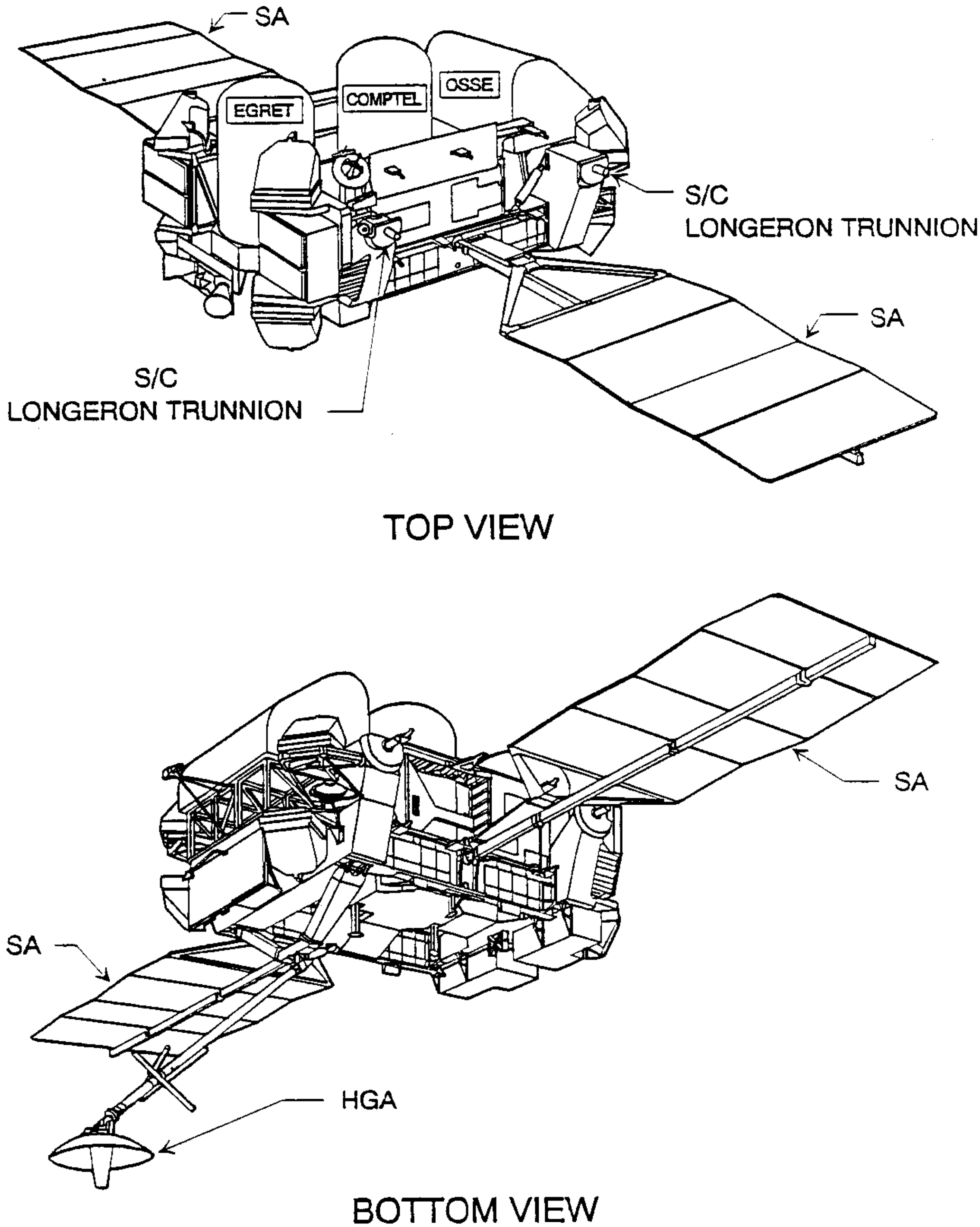


Figure 10. GRO Deployed Configuration.

The HGA boom and both SA wings were planned to be deployed while the GRO was being deployed from the orbiter and still attached to the orbiter RMS. The design of each appendage incorporated redundant mechanisms which allowed for astronaut back-up assistance in case of a mechanism jam or malfunction.

All unlatch and deployment operations were performed with stepper motors which were electrically redundant. These deployment and unlatch actuators were attached to drive linkage mechanisms which offered varying degrees of torque amplification during operation. The actuators were built by Honeywell to TRW specifications and provided variable stepping rates from 7.8 to 250 pulses/sec.

2.5.1 GRO SA Appendages

Each of the GRO SA wings were comprised of the following major components.

1. Four active panels
2. Two inactive or control panels
3. Four boom segments
4. A motor driven release system
5. A motor driven deployment system
6. A drive to orient each array

Each panel was connected to the boom at three locations with a combination hinge/pivot joint.

2.5.1.1 GRO SA Release System

Each array was retained during launch by six latches. An actuator opened all six latches for release. Each latch was driven by a bellcrank which provided two inches of travel to pull the panels into the latched position. The toggle action from the bellcrank provided a high mechanical advantage to develop the necessary preload. A torque tube connected the upper and lower latches and adjustable drag links connected the three upper and lower latches. The upper latches hooked over the panel hinges and pre-loaded them against a shimmed stop on the latch housing. The centerline of the plane of the latches was in line with the hinge centerline so that deflections of the panels during launch did not induce high load in the latches.

2.5.1.2 GRO SA Deployment Mechanisms

All array panels and boom segments were considered components of the deployment system. In addition there were three links, pivoting on the boom segments, which supplied deployment forces to the boom in the early stages of deployment. The actuator supplied the deployment energy. A plate attached to the output face of the actuator drove a linkage, which was similar to the one for the latch systems. The actuator/linkage also powered the drive shaft which also served as a boom hinge pin.

2.5.1.3 GRO SA Panel Structure

Each of the outboard panels was composed of a honeycomb substrate bolted to a supporting frame. The substrate was a sandwich of 0.010 inch aluminum face sheets bonded to a low density aluminum honeycomb core. Threaded inserts were potted into the substrate for attachment of the frame structure. A 2 mil Kapton film was bonded to the substrate front surface to provided electrical insulation for the application of solar cells. Feed-through holes provide access to the back side wiring harness.

2.5.1.4 GRO SA Boom Structures

The boom structure consisted of four subassemblies joined end to end which were accordion folded to the stowed position for launch. Each subassembly was composed of an extruded aluminum rectangular tube and had machined hinge fittings located at each end of the tube. All fittings were aluminum and were riveted to the extrusion.

A special preload device was added to each boom hinge (except the outboard) to prevent the freeplay or deadband motion normal to the array surface. Initial development tests indicated a large (6 inch peak-to-peak) motion due to play in the hinge-to-hinge pin interface which was required for thermal differential expansion. The preload device consisted of a fixed contoured cam which engages a spring loaded roller over the last 15^{o} of boom motion. Compression of the roller against the cam preloads each hinge to a level that will withstand on-orbit Reaction Control System (RCS) induced accelerations without gapping.

2.5.2 GRO High Gain Antenna (HGA) Appendage

The HGA boom supported the antenna reflector dish and the biaxial drive for pointing the antenna. It also supported the magnetic torquers for dumping momentum associated with the attitude control system. The boom was retained in the stowed position by four latches, two near the magnetic torquer bars and two at the antenna reflector. A single motor-drive latch actuator operated all four latches. The deployment actuator, at the base hinge of the boom which interfaces with the S/C keel structure, deployed the boom to the proper position to achieve the required on-orbit field of view.

2.5.2.1 GRO HGA Support/Release System

The support/release system consisted of four latches housed inside support fittings whose upper ends attached directly to the platform structure. The latches were opened by a single motor drive actuator which was supported at the outside edge of the platform to provide access for manual operation.

The latches engaged pins in the slide mechanism mounted to the boom. The slide mechanisms allowed axial motion of the boom relative to the supports due to deflections during launch and changes in temperature of the boom. Each aft slide mechanism consisted of a T-shaped bolt mounted to a fitting on the boom and the slider which supported the latch pin. The cylindrical head on the bolts slides in a hole in the slider. The forward sliders are similar except each has two bolts. These bolts are mounted to the horseshoe shaped antenna frame.

The boom loads are reacted by the latches and support fittings. Each of the forward and aft supports carry the loads through shimmed fittings which mate with the slides. Loads along the boom are reacted at the keel hinge.

2.5.2.2 HGA Boom Structure

The main boom element consists of a 6.0 inch diameter by 0.125 inch wall aluminum tube, with the rotating hinge on the keel end and a flanged fitting on the outboard end.

2.5.2.3 HGA Hinge/Deployment Mechanism

The hinge mechanism consisted of a stationary hinge half and a rotating hinge half joined with two 0.750 inch diameter hinge pins. Toggle links are used to drive the rotating hinge half and preload it against adjustable stops on the stationary hinge half in the deployed position. The driving link is powered by a motor driven actuator attached to one side of the stationary hinge.

Normal deployment of the HGA boom was accomplished by the actuator driving the toggle linkage until stop pads, an integral part of the linkage, engage, preventing further rotation. This stop provides +0.06/-0.02 inch overcenter travel of the linkage so that the high compressive preload prevents the hinge from unlocking. ExtraVehicular Activity (EVA) deployment was accomplished by backing out a captive pin to disengage the actuator and then using a wrench on the hex end of the drive shaft to drive the toggle links over center and preload the hinge in the deployed position.

2.5.3 GRO Appendage Key Design Requirements Versus Capabilities

1. Requirement: Satisfy fracture control requirements of NHB1700.7.

Capability: Fracture control analysis and inspections, as appropriate, were accomplished on all hardware.

2. Requirement: Design each deployable to withstand the induced launch, landing and on-orbit loads. Prevent gaping under design limit load at launch and landing and prevent freeplay under RCS loads on-orbit. Consider thermal, acoustic and relative displacements between the platform and stowed deployable in the design.

Capability: All elements of the deployable structure were designed to meet the minimum launch, landing, and on-orbit load environments. In some cases, the design was based on even higher loads used early in the program. The designs were tested to 1.25 times their design limit loads.

3. Requirement: Design flight structure to factors of safety of 1.5 to yield and 1.9 to ultimate.

Capability: All deployable structures were designed to the required factors of safety. All structures had positive margins of safety that were verified by test and analysis.

4. Requirement: Design flight structures to the following stiffness requirements, as represented by their first mode frequencies:

1. SA stowed: >20 Hz goal
2. SA deployed: >0.30 Hz
3. HGA boom stowed: >20 Hz goal
4. HGA boom deployed: > 1.2 Hz

Capability: The stowed SA and HGA boom resonant frequencies were confirmed by low level, base drive, sine sweep vibration testing. Deployed frequencies were confirmed by static load tests.

5. Requirement: Maintain clearances, fields of view and alignments per Interface Control Drawing (ICD) for all deployable structures (SA and HGA).

Capability: The deployable structures met their static envelope, dynamic envelope, internal clearances, shadow limitations and alignment requirements as demonstrated by analysis and test. In some cases, the clearances during both deployment and operation were less than one inch. These tests were performed with thermal blankets in place.

6. Requirement: Design deployables and their deployment mechanisms with adequate capability of withstanding the stall torque of their drive motors at full deployment and lockup position.

Capability: Both SA wings and the +Y latch were designed to the higher stall torque of the uprated -2DLA motor (2,500 in-lbs). All others were designed for the -1DLA motor (1,540 in-lbs). Capability to reach stall torque at ambient conditions was demonstrated by tests.

7. Requirement: The deployables shall be capable of being stopped at any point in the deployment cycle and restarted with a torque ratio >3.0 at ambient conditions and >2.5 at worst case conditions.

Capability: All deployable structures achieved the required margins as demonstrated by tests on flight and development hardware. Only development hardware was exposed to worst case conditions of cold temperature.

8. Requirements: The deployables shall have positive latching in their final position.

Capability: The design of all deployment mechanisms provided for an over center condition at full deployment to prevent restowing during on-orbit operations.

2.5.4 GRO Appendage Testing

The overall test program for deployables was quite extensive. Tests performed on both the SA and HGA boom hinges and single latches were used to understand the effects of thermal environments and evaluate the resistive torque effects of cable bundle wraps. Later tests subjected flight like deployment hardware of one full SA wing and the HGA boom assembly (including latches) to deployment, vibration and static load tests. The flight hardware was then subjected to a range of tests including acoustics, thermal vacuum and deployments.

2.5.5 GRO Appendage Special Issues

2.5.5.1. Clearances During Deployment and Operations

There were several areas of critical clearances presented during an observatory pre-test review which were later resolved. The clearances of the flight SA jettison fitting and deployment motor housing were measured relative to the platform. All clearances were found to be acceptable. Also, the clearance between the open SA upper latches and the deploying panel hinges were increased to 0.30 inch minimum by shimming and readjusting the latch timing.

2.5.5.2. HGA Initial Motion

Concerns of clearance during initial motion of the HGA boom were resolved by adding a guide to the boom.

2.5.5.3. SA Initial Motion Test

An initial motion test of the SA's on the observatory were considered. After careful deliberation, it was decided to forgo such a test because of risk to the array and test set-up costs. Instead, all mating surfaces and guides were examined to ensure the insulation was secure and not subject to snags. The fact that the array is guided during the first several feet of deployment was a predominate factor in not conducting separate initial motion tests.

2.5.6. GRO On-orbit HGA Deployment Anomaly

The only anomaly encountered with deployment of the GRO appendages was associated with the HGA. Prior to observatory release from the orbiter RMS, the HGA did not deploy when it was initially commanded. An astronaut EVA was required to physically shake the antenna to initiate the deployment sequence. Subsequent investigations corroborated with on-orbit photographs taken by the astronaut crew indicated that a portion of the antenna release mechanism (close to the antenna dish) was caught by a piece of thermal insulation blanket. This occurred because of large relative motion between the antenna and its support structure which allowed an exposed bolt to be caught by the neighboring thermal blankets. There were three areas identified which are believed to have caused the problem.

1. Several bolts adjacent to the thermal blankets were installed with the longer nut end of the bolts protruding in close proximity to the blankets. Although this was needed to circumvent a design flaw with the antenna support mechanism, interference with thermal blankets was not envisioned.

2. Structural vibration testing of the assembled HGA did not include thermal blankets. For practical reasons, these tests also did not include a simulation of the S/C relative stiffness between the HGA support points. It is this low stiffness which allowed large relative motions between the dish end of the antenna and the S/C. The only observatory level test with thermal blankets installed was an acoustic test. Acoustic tests generally do not produce large S/C loads or motion which would have been needed to uncover the problem prior to launch.

3. The technician and quality assurance personnel who performed the final stowing and inspection of the HGA, respectively, did not foresee the potential problem of large relative displacements which would have caused the bolts to catch the thermal blankets and prevent the antenna from deploying.

2.6. UPPER ATMOSPHERIC RESEARCH SATELLITE (UARS) [W. Leavy/SAI]

The UARS contained three deployable appendages as described below and shown in Figure 11.

2.6.1. UARS Solar Array Retention, Deployment and Jettison Assembly (SARDJA)

The SARDJA was used to restrain and control the deployment of the SA during nominal operations and jettison of the SA during contingency operations. The SARDJA contained redundant springs in the SA panel and strut hinges for deployment, redundant stepper motors to regulate the rate of deployment and redundant initiators in the separation nuts for jettison release.

The six panels of the SA were restrained in the stowed configuration by four retention bolts (with pyro activated separation nuts) attached to the primary structure of the Instrument Module (IM). SA deployment was initiated by firing the separation nuts in pairs. When the second set of separation nuts was fired the SA expanded out five to eight feet because of stored energy in the retaining bolts, hinge springs, and a synchronization (sync) cable. The SA was then completely deployed by commanding the deployment motor to complete the deployment.

Figure 12 details the mechanical overview of the SARDJA release mechanism.

2.6.2. UARS Solar Stellar Position Platform (SSPP) Instrument and HGA Gimbal Retention Subsystems

The SSPP and HGA used identical two-axis gimbal mechanisms. Both mechanisms incorporated similar retention systems to prevent inadvertent release of the subsystems.

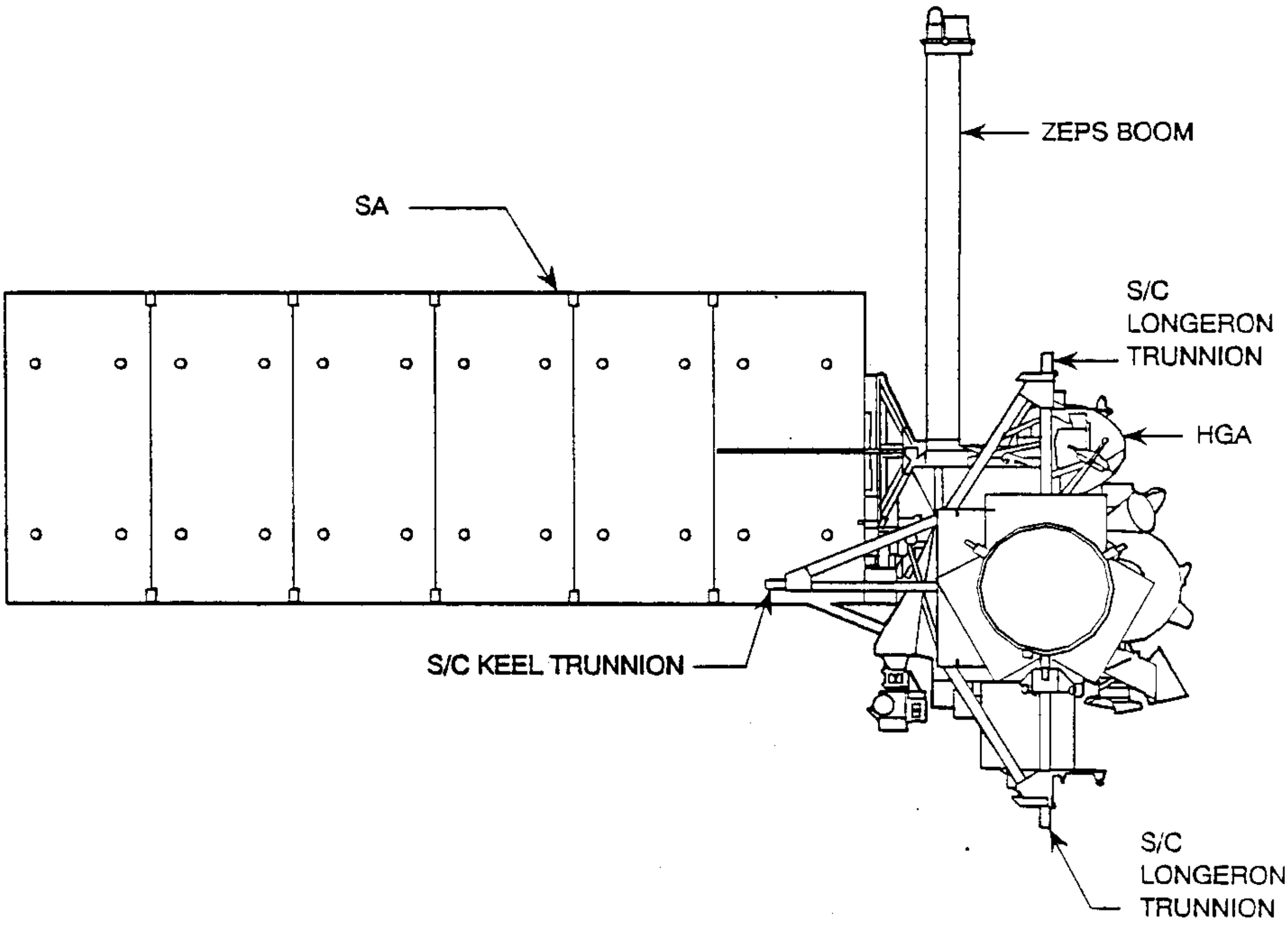


Figure 11. UARS SARDJA.

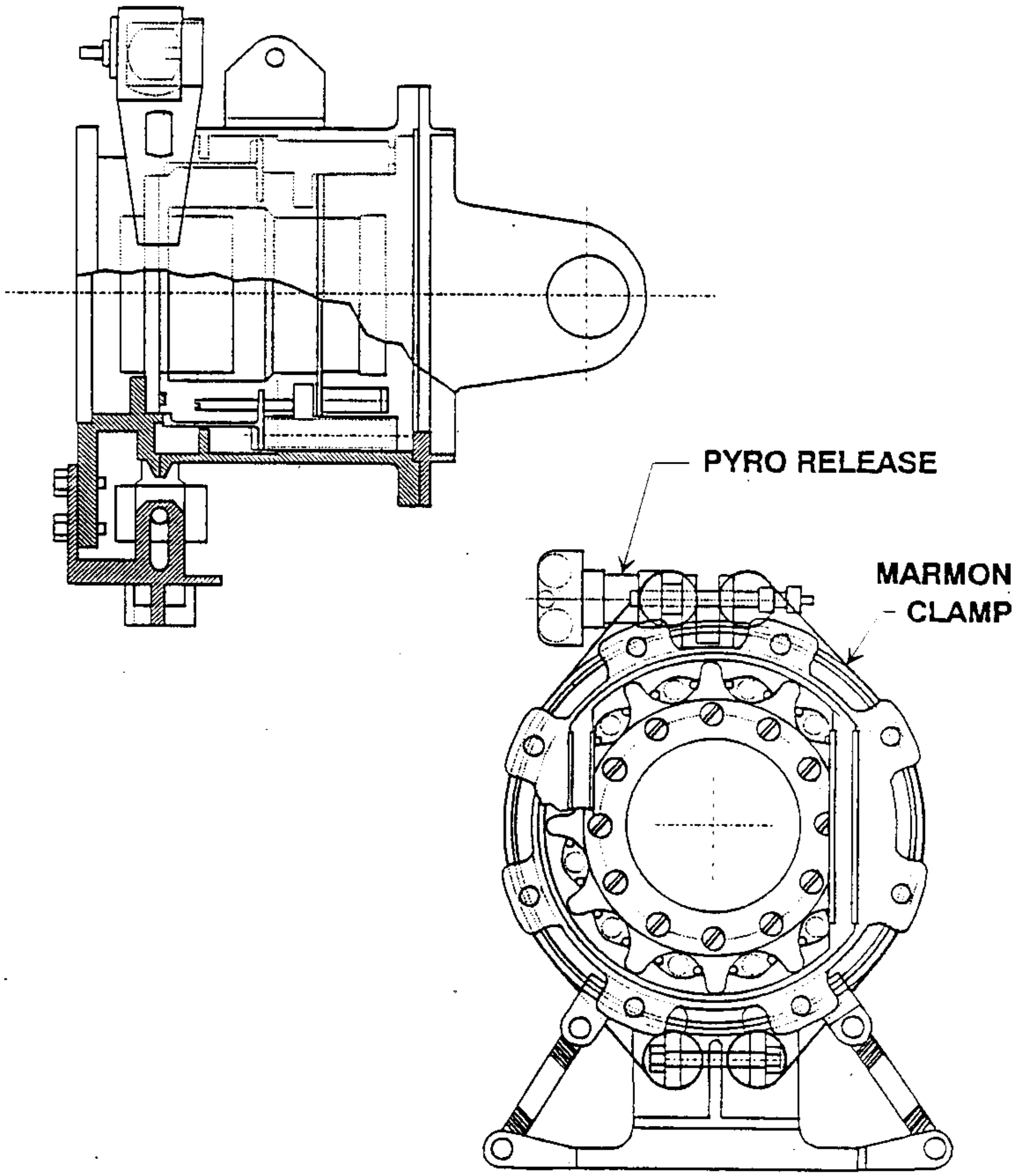


Figure 12. UARS SARDJA Release Mechanism.

The SSPP was restrained at two points between the gimbal interface and the front face of the SSPP and at two points between the platform structure and the gimbal support structure. Gimbal lockout for the HGA was provided at three points between the antenna extension boom structure and the gimbal support adapter. Each retention point consisted of a cup/sphere assembly held together by an overcenter in-line linkage operated by DC motors. The drive motors were operated in series for redundancy and incorporated a specified detent torque to retain the proper linkage position during all mission phases.

2.6.2.1 Design Problem

The switches used on all mechanisms were Honeywell 10HM30-5RELPGM miniature hermetically sealed units. The auto stow function operated the two gimbal motors to their respective "ready to latch" position, where actuation of the switches caused shut off of power to the motor. One set of switches was actuated by a rotary cam with its travel being at 90° to the switch axis, the other by a linear plunger which moved parallel to the switch axis. A problem occurred with the design when attempting to adjust it on the ground under 1-G conditions. In the orientation at the time, gravity effects caused the plunger to deflect away from the switch, resulting in no motor cutoff at the desired position. The switch was limited in its overtravel capability, so adjustments to make it work properly on the ground would cause destruction of the switch on-orbit (in 0-G conditions).

2.6.2.2 Design Solution

The solution was to redesign the switch activation device and make it similar to the cam operated switches. Instead of a rotary cam, however, the actuation was done by using the linear motion of a rod which was stepped with two diameters (Figure 13). This arrangement was not critical to linear motion. Overtravel was limited by the two diameters. This allowed for both ground testing and on-orbit operation.

2.6.3. UARS Deployable Zenith Energetic Particle System (ZEPS) Boom

The ZEPS boom was a truss assembly consisting of three 13 foot long glass/epoxy longerons, triangular battens and diagonal stiffeners between battens. The boom was stored in the instrument canister by coiling the longerons into interlacing helices. The ZEPS instrument mounting plate was attached to the end of the longerons and was latched to the canister until deployment. Boom extension was initiated by activating the latch and the deployment mechanism motor. Deployment was effected by the stored energy in the longerons and regulation was provided by redundant DC motors that metered out a lanyard connected to the instrument mounting plate. The boom could be restowed by reversing the direction of the

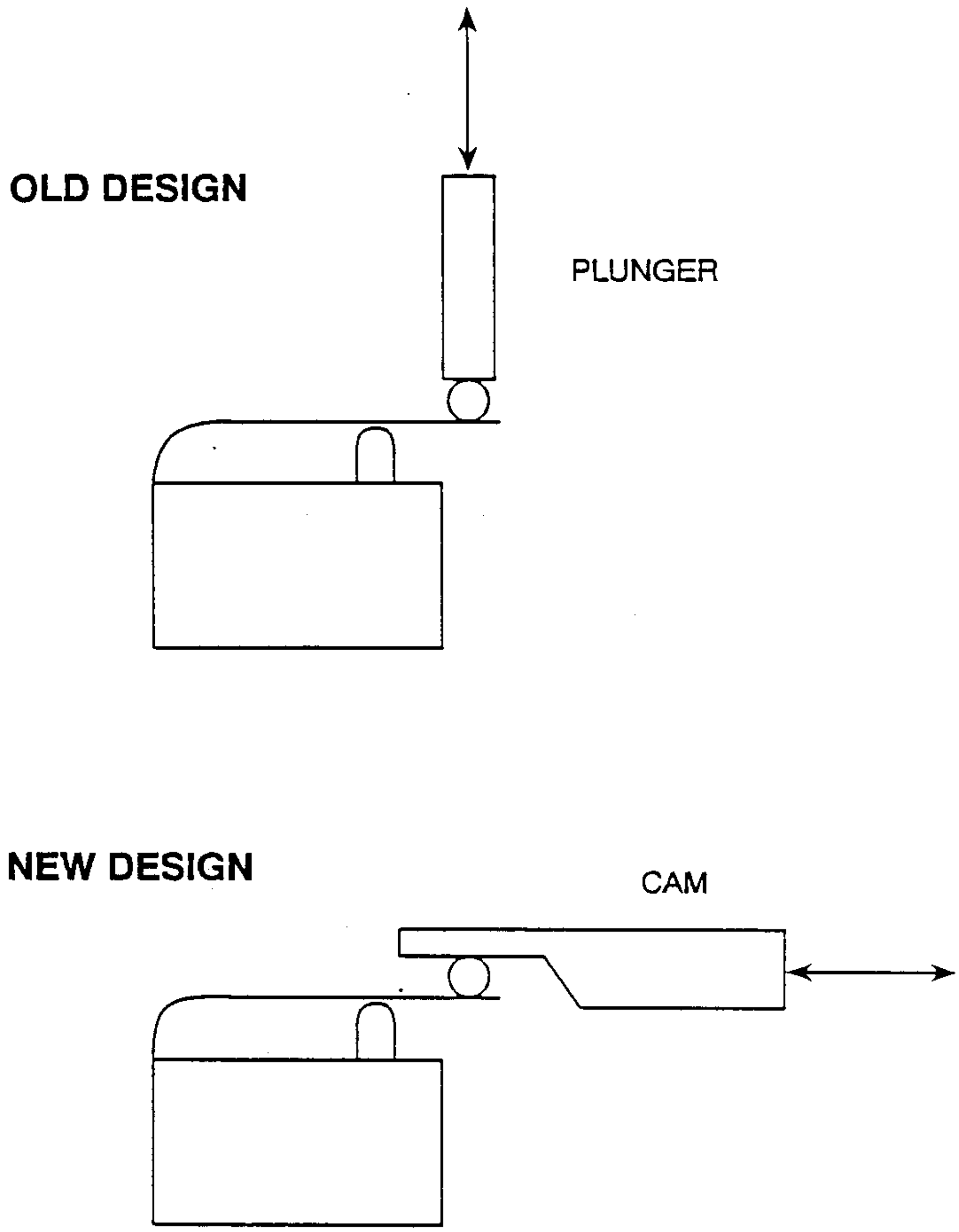


Figure 13. UARS Switch Actuation Redesign Solution.

motor. The retention system of the ZEPS boom prevented boom deployment until a safe separation distance had been reached by the orbiter.

3.0 <u>DEPLOYED APPENDAGE DESIGN PHILOSOPHY</u>

3.1. <u>GENERAL OBJECTIVES</u> [R. Farley/731, E. Devine/SAI]

The design of a S/C deployed appendage is a systems engineering task driven by many critical and often conflicting requirements.

If there is anything in general that can be said about designing S/C deployable systems it is that one should design hardware that can be analyzed, tested, easily built and handled in the most system harmonious and failure tolerant manner possible. Considerations include incorporating the entire life cycle loading and wear spectrum which includes protoflight testing, Integration and Test (I&T), shipping, handling, launch and on-orbit operations. The engineer must also at the outset design with clearances between moving parts which consider thermal expansions and static and dynamic deflections both direct in nature or induced by S/C interface motions. This is to avoid the "Velcro effect" between adjacent parts. Deployment systems have in addition to the mechanical elements, connectors, wires, thermal blankets, thermal coatings, terminal boards, diode boards, piggyback sensors, bonding straps and sharp edges (which should be avoided). It cannot be stressed enough that the designer needs to consider these details from the very beginning. A classic example is that of the back side of a SA. When the panels are stowed, a situation can present itself where connectors are folded on top of other connectors, terminal/diode boards are folded over flat wire bundles or panel-to-panel jumpers fold around hinges. This leads to the very real possibility of snagging, which leads to the general rule that such components should be staggered.

Appendage failures seem to come in four categories.

1. Nonredundancy of the motion producing elements
2. Too little torque margin
3. Snagging
4. Stiction

A sometimes confusing issue is the difference between torque margin and torque ratio and which values are minimally acceptable for GSFC designed subsystems. The minimum torque required to overcome friction and accelerate the device which is to be deployed is defined in terms of torque ratio and torque margin. Torque ratio (T_r) is defined as:

$$T_r = T_{available} / T_{max}$$

where: $T_{available}$ is the maximum torque which can be provided by the actuator device.

T_{max} is the maximum resistive torque (friction, etc.) that will prevent rigid body deployment motion of the device which is to be deployed.

The minimum required T_r for GSFC devices is 4.0.

Torque margin (T_m) is defined as:

$$T_m = T_r - 1$$

The minimum required T_m for GSFC devices is, therefore, 3.0.

Some designs are produced with too little torque margin. All friction sources were not accounted for, special thermal cases were ignored or the design was too sensitive to assembly technique. Good design practice should eliminate designs of too little margin. Panel hinges should be made with spherical bearings with axial clearance making binding impossible. Dampers to dissipate the potential energy of deployment springs should be used to reduce the kinetic energy at impact. That way a very large torque margin can be used without concern for high impact loads. In any case, with all springs working, the torque to friction ratio shall be at least be 4:1 (torque margin of 3) with all parts functioning in the worst case friction.

There should always be a back up source for deployment torque and it should have at least a 4:1 torque ratio over the worst case friction. The deployment energy can either be in the form of redundant springs or redundant windings in a motor. The release mechanisms should also have redundant actuators with separate electronic circuits.

Snagging of wires or thermal blankets is an insidious failure because many tests can be run and the problem may never appear until it is too late. This happens because many of these items are not used in deployment tests, the tests do not really reflect the actual flight conditions or snagging points do not become obvious until flight integration, after flight integration, or after vibration testing. Once again, good practice should eliminate this by accounting for their effects from the start.

Avoiding stiction means using the proper lubricants, proper angles and kick off springs. Moly-disulfide solid lube has a tendency to absorb water which can later freeze and jam hinges or V/cone guides. These types of lubricants should be avoided with preference given to hard-slippery coatings such as Tiodize V. In addition, cone supports should not have angles more acute than 30^{o} to avoid locking. If there is still some residual stiction, the deployment system must account for it by the judicious use of kick-off springs.

The proper characterization of a wire harness bending stiffness is extremely critical when the harness must traverse a rotating joint and the harness can reach low temperatures. The bending stiffness of electrical harnesses rise sharply with decreasing temperatures, thus providing additional resistive joint torque which must be overcome by the joint actuator. The maximum stiffness of the flight harness must be determined by (thermal) testing and properly accounted for in the design of the joint and the sizing of the joint actuator.

Designing a system that is easy to assemble, is testable and analyzable is a very desirable goal. This means designing a system that is not overly constrained (statically determinant, more or less). This allows for S/C "breathing", thermal distortions and imprecise assembly. Shear ties and other alignment critical elements should all be on easily adjustable fixtures. Hinges that are separated on the same structure should have self-aligning bearings. Making the system testable and analyzable usually means sequencing the deployment into successive subdeployments of one or two degrees of freedom motions. Many times this is necessary to keep the deployable elements from hitting each other, although this does increase the number of "mouse traps" that have to work, and so it becomes a program trade-off.

Telemetry has its place but should not be necessary for proper operation of the deployment. It is useful during testing to characterize and monitor the components. It is also very useful when there is an operational failure in orbit. Sensors should be considered for initial motion, for intermediate positions and for latch lock indication. SA articulation motors should also include output shaft position, a null reference indicator, a motor rotor "strobe" and a motor current indicator.

3.2 FRICTION AND LUBRICATION [E. Devine/SAI]

Surfaces that must separate or move relative to one another must be designed to preclude the possibility of metal to metal adhesion which causes increased friction and, in extreme cases, galling. To minimize friction problems the following guidelines are followed.

1. Maximum utilization of rolling surfaces, as opposed sliding motion, should be used.

2. Lubrication or separation of all moving surfaces either by a suitable aerospace grease or dry lubricant coating should be used. No exceptions are allowed, even for lightly loaded "friction compatible" surfaces.

3. On hard mating surfaces where hard coatings are used (such as Type III anodizing on aluminum) loads must be kept below the bearing yield strength of the substrate metal (e.g. 60 ksi for 6061-T6 aluminum).

4. Smooth and polished mating surfaces are preferred.

5. Dissimilar material mating surfaces should have low mutual solid solubility, or at least one of the two should have a heavy dissimilar coating (e.g. nitride, carbide or oxide).

6. Caging devices should be designed to positively preclude relative motion between clamped surfaces when subjected to shipment or launch vibration. Any separation (gaping) under launch loads is undesirable. Small amplitude oscillatory motion between mating surfaces can damage lubricating films and, in the extreme, result in fretting (adhesion) of the surfaces.

Two alternatives are available for lubrication - "wet" lube with a low vapor pressure aerospace grease and "dry" lube by means of bonded or sputtered MoS_2 coatings.

The wet lube is generally preferred because the lubricating film is self healing and frictional behavior is more consistent and predictable. The grease with the most heritage is the Bray 600 series, a synthetic fluorinated oil thickened with micron sized Teflon powder. The grease has extremely low outgassing (TML $< 0.1\%$ and CVCM $< 0.05\%$ for the standard 125^{o} C - 24 hour test) and concerns relative to contamination are negligible for virtually all S/C applications. The wet lube usable temperature range is -80^{o} to $+200^{o}$ C.

For extreme low temperatures, cryogenic applications and other special circumstances, MoS_2 coatings are suitable. For most consistent performance, and lowest possible outgassing, these films should be applied in-situ by the ion sputtering process. Epoxy and polyimide bonded films can be successfully employed with proper application and burnishing to remove excess material.

3.3 LONG DEPLOYABLE APPENDAGES [R. Sharma/716]

3.3.1. Astromast Type Booms

Two basic types of deployable structures currently exist. Lanyard deployment types deploy external to the canister housing and are restrained by a lanyard that deploys the boom through a motor gear box arrangement or a viscous damper. The motor unwinds the lanyard spool and this, in turn, deploys the boom structure. This type of boom does not achieve maximum stiffness until full deployment because of an existing transitional region that bridges the fully deployed sections and the stowed boom sections. Generally lanyard deployed types of booms are not easily retracted but can be designed to do so. Initial release, deployed length, and end-of-travel telemetry is typically required for boom deployment monitoring.

The canister type Astromast is deployed within the canister housing and has maximum stiffness as it deploys from the housing. This type of boom is rigid throughout its deployment and is retractable. The deployment mechanism can be similar to the design indicated above. Generally both designs mentioned require rigid midpoint supports for attaching instrument payloads.

Considerable attention must be given to the thermal design for these types of booms. Depending on the stability of the S/C (spin stabilized, three axis stabilized, etc.) and spin rate, the space thermal environment will cause mechanical bending due to temperature gradients that may exist on the boom. In addition, thermal coating requirements generally conflict with the conductivity requirements of the boom. Full-length S/C level deployments must be made to determine and avoid potential thermal blanket interference.

Astromast type deployable structure design requirements are specified below.

1. Bending stiffness
2. Torsional stiffness
3. Bending strength
4. Critical buckling torque
5. Linear weight per unit mass
6. Packaging ratio
7. Mast diameter
8. Bay lengths
9. Longeron crossection
10. Batten diameter
11. Diagonal diameter
12. Stowed and deployed frequencies
13. Random and quasistatic loads
14. Thermal environment

Presently, the ISTP, GGS, WIND and POLAR S/C's are using six and 12 meter long lanyard deployed booms for deploying magnetometer instruments.

3.3.2. Articulated Booms

Articulated booms are rigid boom assemblies usually composed of tube or rod segments used for deploying magnetometer instruments from S/C. Sequencing of a two segment boom is preferred over a three segment boom for simplicity and S/C volume constraints. These booms are not retractable. They are deployed by a constant force spring hinge assembly. Generally, better alignment between magnetometers and a higher deployed natural frequency can be maintained with these booms compared to the Astromast type deployable structures. The same design requirements are needed of these booms as of the Astromast type deployable structures indicated previously. Generally, this type of boom is not inherently sensitive to deflections caused by thermal distortion because of the boom's rigidity. Proper selection of thermal blankets and coatings will help minimize temperature effects.

3.3.3. Tubular Booms

Tubular booms are thin wall circular crossection antenna elements. Various crossection tubular booms exist. They range from 0.5 inch diameter to about 1.12 inches in diameter. They are mainly used as antennas for electric field experiments and instruments. The structural properties of the boom element are primarily determined by the boom diameter, thickness, material properties, number of tapes and whether the edges of the tapes are overlapped or joined to each other. Some crossectional types are as follows.

3.3.3.1. Overlap Design

Booms consisting of a single tape with overlap edges are the simplest construction and are used when low torsional rigidity is sufficient for the S/C mission. Two tape spools that form the boom by overlapping one another as the tapes are deployed is another variation of the overlap boom type.

3.3.3.2. Interlocked Design

Booms with interlocked, also called "zippered", edges consisting of one or two tapes have a circular crossection and high torsional rigidity, which is required for maintaining the angular position of payloads located at the tip of the boom or deployment along the spin axis of a rotating S/C. The welded-edge booms have the highest torsional rigidity and generally have an elliptical crossection.

3.3.3.3. Thermal Design

Several techniques have been employed to minimize boom deformations due to the thermal environment in space which is the dominant cause of boom distortions. Highly reflective coatings on the external boom surface have been used to reduce the absorbed thermal energy and thereby diminish the temperature gradient on the perimeter of the boom. All booms presently used have highly reflective coatings even though this method cannot, by itself, completely eliminate thermal deformations. Perforated booms have been developed to permit the solar radiation to strike the inner surface of the boom on the side facing away from the sun to equalize the boom temperature. The pattern and size of the perforations and the coatings on the inner and outer surfaces are selected to balance the thermal input for all boom orientations. Theoretically, this method can completely eliminate thermal bending but some residual bending is inevitable due to tolerances and degradation of thermal surface properties in the space environment.

3.3.3.4. Extension Mechanisms

Booms have been extended in orbit by motorized mechanisms or have been self-extended by the spring energy stored in the tape material. The self-extending units require power only to uncage the boom. The motorized booms require power for the deployment motor, length indicators, end-of-travel limit switches and temperature sensors.

3.4 ACTUATORS [R. Sharma/716, A. Tyler/716, R. Farley/731]

3.4.1. Springs

Good engineering design practice dictate that redundant push-off springs be incorporated for initial release of all S/C deployable appendages. The release of deployable appendages for space flight use is always critical for mission success. Generally, rate controlled constant force spring driven hinges are used for deployable appendage actuation. Typically, these appendages are released from the S/C by an explosive bolt/cable cutter, pin puller or separation nut type devices that upon electrical command initiates the firing of a pyrotechnic actuator which, in turn, releases the deployable appendage from the S/C. Having an additional redundant kick-off spring assembly at the appendage- S/C interface will facilitate a more positive initial release of the deployable device.

3.4.2. Deployment Drive Motors

Brushless DC, permanent magnet DC and brushless DC stepper motors should be the motors of choice for application in the harsh environment of space.

The design output torque of the motor relative to the load torque required should be large. Recommended design torque margin of any flight motor-drive system is three. The design torque margins should be verified during flight acceptance testing.

The inherent design of these motors allows high reliability in any space flight application if the proper workmanship (materials, soldering, heat sinking, assembly and testing) practices are followed. Because of friction, mechanical misalignment, unpredictable wear on components and the unpredictability of lubricant outgassing in mechanism design, the initial design torque margins should be maintained at the recommended levels. These motors have been used for various programs within Code 700 including the Astro-1 BBXRT cover mechanisms, GRO EGRET instrument, COBE FIRAS mirror transport mechanism, San Marco antenna deployment mechanism and DE antenna deployment mechanisms. Additionally, planned in-house mechanisms work to use brushless motors in designs include the following.

1. ISTP/WAVES antenna
2. CASSINI/CIRS
3. MODIS-T engineering model scan and tilt assemblies
4. AXAF/XRS gate valve-filter wheel

Brush type motors should not be used for space flight applications for the following reasons. The carbon brushes generally used in the design of brush motors wear extremely fast when subjected to mechanical loads in a vacuum environment. This wear produce considerable debris which penetrates the motor shaft bearings and substantially increase bearing drag torque while corrupting the bearing lubricant and life. Additionally, the accumulated debris electrically shorts adjacent commutator bars of the rotor assembly and the motor exhibits a sharp increase in the running current. This leads to motor failure and a possible contamination source for scientific instruments. This process occurs very rapidly in an atmospheric environment where air (which acts as a lubricant) is present. Recent motor failures with the MAPS/MDRA and ISTP/GGS lanyard deployed boom mechanisms are only two of the numerous examples that can be given.

3.4.2.1. Stepper Motors

Stepper motors are essentially brushless DC motors with special construction and properties which are capable of converting digital signals into fixed increments of shaft rotation. This is generally achieved by magnetic alignment of "teeth" on the stationary and rotating parts of the motor. This makes it a useful and reasonably accurate device for positioning a load without the need for direct feedback systems.

The only error induced by the motor is its final positioning accuracy which is usually within 5% of the last step taken, depending on the Coulomb friction in the system. This error is noncumulative, and with a reduction drive the final position of the output shaft is within extremely small tolerances. Because the motor is an incremental device, it can be easily integrated into a microprocessor based control system.

The load (inertia, friction, damping) can affect many motor parameters, such as motor resonance, instability, overshoot, ringout time and pull-in rate. Stepper motors are characterized as lightly damped spring-mass systems where the motor has several torsional resonant frequencies that can reduce the available torque or go unstable if driven at a resonant frequency. There are three general types of stepper motors:

1. Variable reluctance
2. Permanent magnet
3. Hybrid

3.4.2.2. <u>Stepper versus Direct Current (DC) Torque Motors</u>

The advantages of a stepper over a DC torque motor is that the control of a stepper is very simple compared to the DC torque type. Open loop control of a stepper is simple to implement and the motor also has the advantage of an internal magnetic detent to hold the last position without dissipating power. This detent gives the system high torsional stiffness with a reduction drive, whereas the direct drive DC torque motor must be held in position by the control system at all times and has low torsional stiffness. In all fairness, a stepper motor can be commutated with Hall effect sensors to simulate the behavior of a DC torque motor and still have a detent torque, although the ripple torque will be higher than a motor designed to minimize ripple torque and with a reduction drive should have high torsional stiffness. In this configuration, the motor will require a closed loop control system and velocity sensor, as would a straight DC torque motor.

Simpler, smaller, lower cost electronics utilizing open loop and microprocessor control are the key advantages.

Even though the motions are small, they are still impulsive in nature creating a set of frequencies that need to be avoided to minimize S/C jitter and motor resonance. Those are the key disadvantages.

3.4.2.3. <u>Rotary Actuators</u>

These devices can be broken down into the following three main subparts.

1. Motor
2. Reduction gearing
3. Output shaft support

The motor can of course be either a DC torque motor or a stepper motor. As previously mentioned, a magnetic detent can be provided for either kind. The straight DC torque motor should be of the brushless-permanent magnet style and will require a position and velocity sensor.

The reduction gearing of choice is the harmonic drive, which has three components.

1. Fixed circular spine
2. Flex spine (which engages the circular spine when deformed)
3. Elliptical wave generator (which deforms the flex spine)

The last major component is the output shaft support. This is typically a pair of back to back duplex bearings, preloaded for a desired stiffness and life span. The back to back configuration is desired over the face to face because of the former's high moment carrying ability when used as a single pair. The size of the duplex pair is chosen with regard to the static and dynamic loads of launch and deployment impact.

Other optional components are added for specific applications such as limit switches, travel hard stops, incremental and absolute position encoders, tachometers and slip rings or cable wraps.

As an example, the proposed design for the TRMM SA drive actuator will use a three phase permanent magnet type stepper motor with a step size of 1.5^{o} through a harmonic drive reduction of 200:1. This will result in an output step size of 0.0075^{o}. This baseline actuator has a starting rate of 300 steps/sec when an inertia of 350 slug-ft^2 is rigidly attached, which is approximately six times greater than the TRMM array inertia. This means that no steps are lost in the initial transient acceleration. In fact, when there is no load inertia attached, the starting rate is about 1000 steps/sec. This remarkable performance can be explained for two reasons. The first is that the step size is very small at the output shaft and the second is the compliance of the harmonic drive. The rotor can get ahead of the load an equivalent of three to four steps during initial acceleration due to the fact that the harmonic drive is elastically winding up a displacement equal to three to four steps. This wind up will unload when the output load reaches a constant speed.

In the undesirable situation where the actuator accidently runs into a hard stop, the behavior will be as follows. After driving the output into the hard stop, the motor rotor continues to turn which elastically winds up the harmonic drive. This reaches a point where from switching from one energized state to the next, there will be less motor torque than the elastic torque it was reacting a moment ago. The motor torque may even be zero or negative for that combination of state and rotor position. The end result is that the spring energy in the harmonic drive catapults the rotor backwards maybe three steps as it unloads. The motor then picks up the pulse signals as if it were starting from a standstill and drives into the hard stop repeating the same series of events over and over. Preventing the actuator from running into a hard stop is the design goal and is achieved through the proper use of end-of-travel limit switches.

Typically, a Type 5 Schaeffer Magnetics rotary actuator will have internal losses due to harmonic drive generator friction of 5 in-oz (input end) and a torque efficiency of 75%. The flex spine has a torsional stiffness of 100,000 in-lbs/rad and the transverse stiffness provided by the duplex bearing has been measured at 500,000 in-lbs/rad. The motor rotor/wave generator inertia is 0.854 lb-in^2. The duplex bearing pair is preloaded to 100 lbs resulting in a drag torque of 1.0 in-lb on the output end. The peak motor torque produced by energizing the coils is on the order of 80 in-oz and the detent torque can be as high as 33% of the motor torque (27 in-oz).

3.4.2.4. Motor Torque

A DC torque motor is commutated (coordinated phase switching) for maximum torque/minimum ripple, where the motor torque is only dependent on the current in the windings (or rotor speed and applied voltage). The stepper motor is uncommutated and the torque then depends not only on the current in the windings, but also on the position of the shaft with respect to those fixed windings. The motor torque can then look saw- toothed during multi-step operation and the exact wave form will depend on the load (friction, inertia, viscous effects), step rate and intrinsic motor characteristics. This makes a closed loop control system based on assumptions regarding motor/speed characteristics very difficult to implement.

Initial motion acceleration of the output shaft can be estimated using the system inertia and the static torque per state curve, assuming that the phase switching is more or less coordinated since the detent keeps the rotor in the proper initial orientation. The static torque curve can then yield an average torque value and be used in the equation:

$$\text{Angular acceleration} = T_{avg}/J_{sys}$$

where: T_{avg} = average torque value

J_{sys} = system reflected inertia

This equation does not take into account the reduction drive or load flexibility.

The reaction torque of the motor self compensates to the load regardless of the motor current or speed. It does this due to the inefficiency of not being commutated. That is to say that the signal can either lead or lag the rotor which results in a lower torque output than if it were perfectly commutated. With no load on the output shaft, the rotor can essentially glide ahead of the signal by almost one step. This produces no net torque even though the power dissipation remains the same (but goes into heat instead of mechanical work). This is not to say that the stepper motor behaves smoothly as does an Alternating Current (AC) synchronous motor. Quite the contrary, the variable torque is the result of a saw-tooth static torque curve where the positive motor torque is equal to the load plus the negative motor torque component. Virtually smooth operation can be achieved with small step angles and a reduction drive system, if structural and motor resonant frequencies plus associated jitter frequencies are avoided.

3.4.2.5 Motor Reduction Gearing

The reflected inertia through the reduction gearing that the motor "sees" is diminished by the square of the reduction ratio, n, where:

$$J_{reflected} = J_{load}/n^2$$

This makes the reduction gearing a powerful buffer to isolate the motor from inertial load effects. In the case of TRMM with a 200:1 reduction, the output shaft load inertia will decrease by a factor of 1/40,000. Even with this large reduction, the TRMM SA reflected inertia is still 10 times larger than the motor shaft (rotor + wave generator) inertia and therefore has great influence in the motor performance.

The TRMM rotary actuator uses a harmonic drive reduction gear which has the advantages of zero backlash and simplicity but does have some disadvantages. The main disadvantage is the spectral content of the output which is contaminated with low level sinusoidal variations at 2, 4 and 9 times the wave generator revolution period (which is usually the same as the motor shaft speed). A one per revolution sinusoidal variation can exist as well if care is not given to the concentricity between the wave generator and the flex spine. Careful selection of stepping rates is required to minimize S/C jitter where the need is warranted.

Other harmonic drive issues are with respect to trade-offs between stiffness/preload and gear life/backlash.

3.4.2.6. Motor Control Options

The basic classical difference between open and closed loop control, as applied to stepper motors, is that open loop counts the number of step commands sent without any knowledge that the physical steps were actually taken, where closed loop counts the number of steps executed with some type of incremental encoder on the input shaft. Care must be given to the design of the incremental encoder on the input shaft so that motor ringing movements are not mistaken as steps.

Variations include the use of absolute position encoders on the output shaft and quasi closed loop systems. In such a scheme, a position is driven to by the open loop method; the final absolute encoder position is compared to the desired position. If there is an error, the number of steps to correct is calculated and then executed open loop fashion. Such an error would be considered anomalous and should be flagged in the telemetry. It could indicate a failure in the encoder, the drive electronics, the Attitude Control Electronics (ACE) box, a power supply interruption, a single event upset in the electronics or a mechanical failure of the unit.

Another variation simply drives the motor at a desired frequency without even counting the step commands. The absolute position encoder is then used as a switch to turn off the motor when the array is within a certain range or prepare to stop by commanding a lower step rate before turning off. Again, the operation should be passively monitored with the position encoder during the open loop time with telemetry flags if the execution is deemed anomalous.

If the array needs to be slewed at a rate that would cause significant jitter, a technique used is to drive the motor with a randomly generated range of frequencies to smear out the spectral output and thus lower structural frequency responses. However, the integral of that band of random frequencies should equal an equivalent desired frequency in order to arrive within a given amount of time.

3.4.2.7. High Output Paraffin (HOP) Actuators

A HOP actuator is used to initiate the release of deployable appendages such as SA's, booms, protective covers and uncage launch restraints for mechanisms.

HOP actuator advantages are as follows.

1. HOP's are small in size, generate a large force, produce a long gentle stroke, are non-magnetic and are reusable.

2. Pyrotechnic actuators generate shock, are not resetable and raise safety concerns.

3. Electric motors are large, heavy and costly.

4. Since HOP's are resetable, a cost savings is incurred during extensive ground testing.

HOP actuator disadvantages are its requirements for large currents and relatively long actuation times depending on ambient temperatures. Additionally, the HOP actuator thermal environment must be controlled to prevent self-actuations.

Known HOP heritage is as follows.

1. Naval Research Laboratory (NRL) UltraViolet (UV) plume experiment launched 2/90 on the LACE/RME satellite operated dozens of times during ground testing and on-orbit.

2. BASD cooling system flown on Delta Star (Delta 183) SDIO satellite launched 5/89 operated once per orbit.

3. NRL High Resolution Solar Telescope launched on a Black Bart research rocket.

Other NASA programs planning to use HOP actuators are the Comet Rendezvous Asteroid Flyby (CRAF) mission; the CASSINI mission; the Earth Observing Systems; the International Solar Terrestrial Physics Program's WIND, POLAR and GEOTAIL S/C and instruments and the Mars Observer program.

3.4.2.8. Actuator Position Encoders

Position encoders fall under two broad categories: incremental and absolute. The following is a short list of the various types of encoders. All have problems of one sort or another involving complexity, cost, size, resolution or drift/calibration.

1. Potentiometers
2. Optical encoders
3. Wave form resolvers
4. Differential Inductance Transformers (DIT's)
5. Light Emitting Diode (LED) strobes

3.4.2.9. Actuator Electronic Drivers

The job of the drive electronics is to take an input drive step rate and turn that into the proper sequence of coil energizing which will produce the desired motion at the same step rate. There are two basic types.

1. Constant voltage driver with current limiting
2. Constant current driver with pulse width modulation

By far the most common for rotary actuators is the constant voltage type with a current limiter.

3.4.2.10. Typical S/C Actuator Application/Operation

The motor drive profile has three basic operational modes.

1. Tracking
2. Slewing
3. Hold

Assuming that the operational modes are achieved using an open loop method, the following could be a typical scenario. The ACE would read the position pot and determine the present array angle. Then, the S/C computer calculates the position to move into (by the ephemeris). The ACE sends a specific clock frequency and direction line to the drive electronics which in turn drives the motor. During the operation, the ACE monitors the progress with the absolute position encoder. When the array is within switching range, the encoder acts as the switch to proceed to the next operational mode in the drive profile when the calculated angle is sensed. If anomalies are detected, they should be flagged in the telemetry.

Since the motor torque adjusts to the load torque at any motor speed, there will be an initial transient torque to accelerate the array followed by a torque just large enough to overcome the internal and external friction load as the array comes up to a constant speed. When the motor is finally commanded to stop, a transient deceleration torque will disturb the S/C at that time.

The drive frequencies should be software selectable to adapt to any unforeseen contingencies that may occur in orbit, with a ground commandable discrete pulse capability. The software should also have the capacity to be reprogrammed from the ground.

The slew and tracking modes should be driven at specific frequencies that minimize S/C jitter response and optimize smoothness of the motor by avoiding the resonant frequencies of both the array and motor. Since the tracking speed is given, the motor manufacturer will optimize the motor design to smoothly operate at that speed.

3.4.2.11. Motor/System Dynamic Simulation

A simple model can be made of the motor/reduction gear/S/C/load combination. Two complications arise; first is obtaining an accurate representation of the harmonic drive (stiffness, damping and output angle variations) and second is the Coulomb friction. The harmonic drive output angle is a series summation of sine functions where the coefficients need to be determined or estimated. Estimations can be made with the manufacturer's design data and test data from other sources. The torsional stiffness in a Type 5 actuator has been measured to be approximately 100,000 in-lbs/rad. The Coulomb friction also needs to be measured or estimated. This has contributions from the harmonic drive and the output duplex bearing pair. But with regard to the equations, the friction is always in an opposite direction to the relative velocity direction and thus the sign of the friction force in the coupled Generalized Differential Equations (GDE's) changes . This means that the solution must be monitored for changes in relative velocity direction. When they occur, stop the solution and change the sign of the appropriate terms in the equations, and then restart the time domain integration.

Time domain integration schemes such as 4th order Runge-Kutta or predictor-corrector methods are used to solve the coupled GDE's.

3.4.3. Gimbal Drive Assemblies

Similar motor torque design criteria apply to gimbal drive assemblies. The design torque margins should be verified during flight acceptance testing.

Mechanical alignment between respective gimbal axes joints is a concern. Placing the payload's center of gravity at a desired location relative to the gimbal's rotational axis is critical in many applications.

Torque margins for flight operations, pointing capability and repeatability of the system should be measured and verified.

3.4.4. Actuator Issues and Concerns

The primary concern for actuators is S/C jitter induced from the stepper motor's incremental motions. This motion has spectral contents of the drive frequency and the motor resonant frequency and the harmonic drive contribution of 2, 4 and 9 times the input shaft period. This is why typically the motor is required to run only in a narrow band of frequencies that have been tested to confirm minimum jitter. Also, resonant stepping rates of the motor need to be avoided. These rates correspond to the motor natural frequency, f_n, and any subharmonic frequency; $f_n/2$, $f_n/3$, etc., although the prime concern is the first harmonic, f_n. If the system is very lightly damped, then the subharmonics can be of equal concern.

3.5 MECHANISMS [R. Sharma/716, C. Woods/716, R. Farley/731]

3.5.1. General Philosophy

Generally, when considering mechanism design for space flight applications, the following criteria must be defined in order to achieve a design that meets the required performance. These criteria can then be further broken down for specific applications as the design process matures. These criteria are identified below.

1. Size
2. Weight
3. Power
4. Temperature limits
5. Launch locks (manual versus remote recage)
6. Life
7. Redundancy
8. Alignment
9. Zero G testing
10. Direct drive versus geared systems
11. Position accuracy and repeatability
12. S/C disturbances
13. Launch loads

3.5.2. Gears

Definition of gear loads (tooth-to-tooth), geometry, allowable backlash, vacuum compatible lubricants, gear-to-gear alignment, etc., is necessary before gear type and material can be selected.

Gear testing must be conducted with sufficient instrumentation to verify gear input and output speeds, wear between mating surfaces, torques and performance.

3.5.3. Hinges

The torque generated by the hinge assembly should be greater than the torque required to deploy the appendage in a 1-G and a zero G environment by a minimum factor of 4, overcome all initial release constraints, overcome all on-orbit S/C induced loading during deployment such as centrifugal and coriolis forces on the appendage, and overcome frictional effects within the hinge assembly that may reduce the hinge torque.

The recommended hinge assembly testing is as follows.

1. Individual leaf spring output torque should be measured prior to integration into a hinge assembly.

2. Hinge assembly output torque should be measured.

3. Fluid damper damping rate, deployment time over temperature for a full range of motion and the O-ring/damper housing interfaces should be tested and checked for fluid leaks.

4. Clearances between fluid damper vanes and damper housing should be measured for noninterference.

5 Zero G testing in a 1-G environment should be thoroughly thought out and implemented. The use of gravity off-loading devices, air tables, cables and pulley systems, bungee cords and water tables should be investigated.

ISTP, GGS, WIND and POLAR S/C mechanisms with similar hinge assemblies are being tested in the above manner.

3.5.4. Latches

Mechanical latches should be designed to lock the deployable appendage in the fully stowed position during launch and the fully deployed positions after on-orbit deployment.

Mechanical latches must withstand worst case launch loads and worst case reaction loads seen at the latch interface during S/C maneuvers. Temperature effects on latch loads must also be considered.

Sufficient strain gauge, torque transducer and accelerometer data should be measured to verify the latch loads, strength and performance during testing.

3.5.5. Bearings

The design of the bearing/lubrication system should be taken into account in the early stages of designing a mechanism. Decisions concerning, for example, load capacity, lubrication and contamination requirements, and precision requirements should be integrated with the design of the overall mechanism.

Bearings are long lead-time items, which usually are very expensive, especially when ordered in small quantities. The United States bearing industry is still in decline, so this state of affairs is just going to get worse. Most bearings shown in vendor catalogs are not "off-the- shelf" items, especially if the system needs are even slightly different than the standard item. Lead times of one year to receive flight quality bearings (not including the procurement cycle) are not unusual. Sometimes it is difficult to even get a vendor to bid on a job. These issues must also be taken into account at the beginning of a project.

3.5.5.1. Hertzian Contact Stress

Analysis should be done for worst case maximum Hertzian contact stress. For applications in which low torque noise/high positioning accuracy is critical, the peak stress in any contact should be kept below 508,000 psi. for stainless steel bearings (440C steel). In less critical applications, the peak contact stress should be kept below 580,000 psi. for stainless steel bearings (440C steel).

3.5.5.2. Torque Tube or Thin Section Bearings

These types of bearings are usually limited in axial load by the height of the shoulder, i.e. increasing the axial load will cause the balls to ride over he shoulder of the bearing long before the maximum contact stress is reached. Also, for high contact angle bearings, this "over-the-shoulder" phenomenon may occur for radial loads as well at stresses less than the maximum allowable.

3.5.5.3. Bearing Launch Load Estimation

The rule of thumb for estimating launch loads at the beginning of a design is to multiply the estimated weight of the rotating portion by the estimated gravity load value and by a Q (amplification) factor. Q factors have been measured as high as 25, but at the beginning of a design we usually estimate around 10. A Q factor of at least 5 should be used. In general, the more detailed/correct the vibration load analysis is, the better the design will be and the less rule of thumb safety margin is needed.

3.5.5.4. General Bearing Mechanical Design

The following guidelines should be followed in the design of space flight bearing applications.

1. Experience has shown that what you order is not always what you get from a bearing vendor in the way of contact angles, raceway curvatures and other geometrical features. Depending upon how critical the application, and how much margin has been left in the system, it may be warranted to measure these parameters in-house. It is also a good idea to buy several times more bearings than are needed, so that there is a pool to choose from.

2. For high precision and long life applications, a 200 hour run-in test of the bearings should be done at operating speeds, followed by visual inspection and torque tracing (in a very clean environment). Inherent problems with particular bearings can be detected in this way. Draper Labs has established a procedure for measuring bearing resistive (wear) torque over time. This procedure works well for a general class of small instrument bearings. For the larger thin section bearings, bearing degradation data over time is unavailable.

3. Torque margins of 4 or 5 times measured values should be specified at the beginning of the project. System torque measurements and worst case end of life torque analyses should be done as early in the program as possible to maintain minimum torque margins of 3.0.

3.5.5.5 Bearings/Lubrication

Because of the severe environment of space, the lubricant of choice since the 70's has been PerFluoroPolyalkylEthers (PFPE's), known as synthetic perfluoronated fluids. In fact, the unique environmental advantages demand their use. However, there are special problems with these compounds which deserve our attention. The typical oils used in bearings have been Bray 815Z (Burma-Castrol) or the Krytox 143xx series (DuPont). Typical greases used in gear trains have been Bray 600, 601, 602, 3L-38 and 3L-38RP. All these are examples of PFPE's.

Lubrication between surfaces can be described as behaving in two manners: one in which the surfaces have metal to metal contact (boundary lubrication) and the other is when there is no contact, separated by a thin film of oil. This second case, where one surface "hydroplanes" over the other is called ElastoHydroDynamic (EHD) lubrication. In most space mechanisms, boundary lubrication is what normally occurs, as it requires high speed to establish EHD lubrication. Because of this, severe problems with perfluoronated fluids have been observed. Although PFPE's are chemically inert to almost everything, they are subject to catalytic attack by metal fluoride (Lewis) acids. These acids are formed under severe mechanical stress (as happens in boundary lubrication conditions) where a few molecules of oil are broken and react with metal on the gear or bearing to form metal fluorides. This catalyst decomposes the oil which eventually starves the bearing. The end product of the polymerized oil is referred to as "brown sugar." Common susceptible metals are titanium (most reactive of all), aluminum, any alloy of iron and even gold.

The reason that Krytox 143 is lately being preferred over Bray 815Z (even though the Bray oil has superior physical properties) is that the molecular structure of Krytox 143 is more branched making it slightly less susceptible to catalytic attack. Switching to the Krytox is not a solution, as it too will eventually break down.

In order to prevent the destructive chemistry, the surfaces in contact need to be passivated in some manner. Ceramic hard coatings like TiN or TiC will eliminate the catalytic reaction. Conventional nitride hard coatings are also effective. In the case of ball bearings, replacing the stainless steel 440C balls with ceramic $Si3N_4$ balls eliminates the breakdown. Also, to prevent oil starvation, it is always good practice to use a porous ball retainer which functions like a reservoir of oil and dilutes any breakdown products.

Using bearings for small rocking motion applications has its problems. Even with a porous retainer, there is no fresh supply of oil to replenish the contacting surfaces with small oscillatory motions. Torque can skyrocket as either the oil breaks down or the bearing starves. In a test conducted at Hughes, Bray 815Z had half the initial torque of Apiezon C (with an Extreme Pressure (EP) additive) in a 4^{o} gimbal bearing test. But after $7x10^4$ cycles, the Bray turned to "brown sugar" and the bearing torque quickly increased by a factor of 10. The Apiezon C with EP additive maintained its original friction out to the conclusion of the test ($8x10^6$ cycles). Apiezon C is a natural hydrocarbon and therefore soluble with EP additives, whereas the Bray 815Z (or any PFPE fluid) is insoluble with just about everything, making EP additives not yet possible. If the surfaces have been passivated in some manner, the bearing life could have been extended since the breakdown chemistry would be prevented. It is impossible to predict how much longer the bearing would last if surface passivation were used since bearing starvation is a key problem due to the small motions and the fact that EP additives are not possible with PFPE's. The exact magnitude of the benefit of EP additives in Apiezon C in this rocking bearing case is also unclear, as there were no test cases run with plain Apiezon C.

The following guidelines should be followed for all space flight bearing applications.

1. All work with precision instrument bearings should be done in at least a Class 10,000 cleanroom on a Class 100 flow bench.

2. The nature of the application, e.g. high/low speed, sensitive optics and number of cycles, needs to be considered in the choice of lubricant and in the design of labyrinth paths/shields

3. Lubricants received from vendors should be examined for unacceptable contamination.

3.6 PYROTECHNIC INITIATORS [M. Phan/731]

Pyrotechnic actuated devices have been successfully used to secure S/C deployable systems during launch and then release them into the final orbit operational configurations for various S/C built by the GSFC. The DE S/C used cable cutters to release its boom. The GOES S/C also used cable cutters to release the SA's. The TDRS S/C used bolt cutters to release the SA's and pin pullers to release antenna booms. The EUVE S/C used bolt cutters to release the contamination aperture covers of its instruments. The SHEAL used pyro pin pullers in the brake mechanism to secure the BBXRT telescope during launch and landing. The COBE was probably the S/C that utilized the largest number of pyrotechnic actuated devices. The COBE cryogenic tank used two band clamp separator devices to secure the tank cover during launch and then to separate the clamp band to eject the tank cover once the COBE S/C was in space. The COBE also used two pin pullers for the deployable Radio Frequency (RF) thermal shield system, two pin pullers for the deployable OMNI antenna system, and 24 pin pullers for the three wing SA deployable system. The new TRMM and XTE S/C's are also planning to use pyrotechnic pin pullers for their SA's and HGA deployment systems.

There are various types of pyrotechnic actuated devices and each device has its own best applications. All pyrotechnic actuated devices come with various sizes, shapes and single or redundant pressure cartridge (pyro) ports. Pyro actuated devices can be customized to fit the required application. However, the new customized devices are required to go through a development test and qualification test program. The customized engineering effort and the development and qualification test program can raise the cost of a device lot up to 40% and extend the delivery time a minimum of an extra two months. The normal delivery time for an existing design pyrotechnic actuated device is about six months.

Pyrotechnic actuated devices have several advantages over the paraffin actuators. The two main advantages of the pyro actuators are the high load carrying capability and the short and predictable actuation time. The pyrotechnic actuated devices have the capability of carrying high shear loads (pin puller) and high tension loads (tension rod, separation bolt, nut separator). The devices themselves are not the load carrying members but are used in high load transfer applications such as cable and bolt cutters. The actuation time of pyro devices is very short and very predictable. The normal actuation time of a pyrotechnic actuated device is less than 10 milliseconds. The normal actuation time for a NiTiNol bushing is unpredictable and between 18 and 30 seconds. The normal actuation time for a HOP actuator is measured in minutes. When more than one release mechanism is used to stow a SA panel (one release mechanism at the top of the panel and one at the bottom of the panel), and if both release mechanisms are not simultaneously released within a fraction of a second, then the SA may be twisted due to uneven loading which may cause the jamming of other release mechanisms.

The main disadvantage of pyrotechnic actuators is that a high shock level is produced when a cable/bolt cutter blade or pin puller is activated. Therefore, they cannot be used near any shock sensitive instrument.

3.6.1. Explosive Bolt Cutters

Bolt and steel cutters are used to cut away steel bolts, rods and metallic or non-metallic cable which is designed to restrain solar wings, antenna booms or instrument aperture doors. Cable cutters and bolt cutters are not load carrying devices. The inertia force of SA panels, antennas, booms and aperture door covers is transferred back to the S/C primary structure through the cable, bolt or rod during launch. The cutter will not see any inertia force because it is not in the load path. Bolt cutters are used in applications where the bolt or rod is designed to take both tension and shear loads and where kinematic mounting of a deployable system is not required. Because of the flexibility of the cable, the cable cutter is a good candidate for the release mechanism where it is required to have one or two degrees of freedom (kinematic mount). One drawback of the cable/bolt cutter is the tendency to eject fragments of the cable, bolt or blade during the cutting process. The ejected fragments may cause the contamination of S/C instruments or the jamming of the release mechanism of the deployable system. The cutters are normally used in pairs that are located side by side for mechanical redundancy. When a pair of cutters are used for redundancy, the cutter that is located further away from the separation plane or boom should be designated as the primary cutter and must be activated first. The cutter that is located closer to the separation plane or boom should be designated as the redundant cutter and must not be activated at the same time or before the actuation of the primary cutter. The release mechanism will lose its redundancy feature if the selection of the primary and redundant cutter is reversed. The reason for losing the redundancy feature is that if the cutter closer to the separation plane is actuated first, and the blade fails to completely cut the cable or bolt, then it will jam the cable/bolt even though the redundant cutter further away from the separation plane cuts the cable/bolt completely after that. When functionally testing the cable/bolt cutter assembly, it must be demonstrated that the cable/bolt cutter is capable of completely cutting a cable or bolt that has a zero preload. This is the worst case, because it is more difficult to cut a zero preloaded cable or bolt than a highly preloaded cable or bolt.

3.6.2. Pin-Pullers

Pyrotechnic actuated pin pullers normally are used to control the release mechanism for instant detachment of structural fittings, spring loaded mechanisms, packages, rods or cable connector assemblies. A pin puller is a shear load carrying device. A pin puller does not support any tension load. The pin puller is best suited to applications where pyrotechnic actuated devices are required to be in the load path. The inertia force of the SA panels, antennas, booms, aperture door covers, etc., will be reacted through the release mechanism - through the pin puller in the shear direction across the pin and the housing to the S/C primary structure. Pin pullers are normally used in pairs for redundancy. Unlike the cable/bolt cutter, it is not necessary to designate which pin puller is the primary and which one is the redundant because the functional failure of either pin puller will not cause the jamming of the deployable system with a proper release mechanism design. The other advantage of the pin puller over the cable/bolt cutter is that the pin puller does not eject any fragments during or after functioning of the device. The GSFC presently favors the use of pin pullers for S/C SA's, antenna booms and thermal shield deployable systems over other pyro actuators because they are load carrying devices, have high load carrying capability, do not eject any particles when actuated and make it easy to design a redundant release mechanism.

3.6.3 Other Pyrotechnic Devices

The pyrotechnic clamp separator is designed to separate or release a ring or circular type clamp used for covers, supply plumbing, umbilicals, tanks and other equipment clamps as fastening or retaining members.

Pyrotechnic tension rod separators and large separation bolts are the tension load carrying devices. Large separation bolts have tension load capability in excess of 500,000 lbs. They have diameters from 1.0 to 5.0 inches and lengths up to 25 inches. Separation bolts and rods are used in applications to separate tension rods and cables and to release tanks, rods and power stages. They are used wherever rod or cable jettison is designed into a system.

Bolt catchers are designed to retain tension bolts after separation to prevent structural damage. Separation nuts, with and without redundant power pressure cartridges, are designed to be used in separation or release of structure or components such as multi-stage vehicles, nose cones and capsules, ejection seats, launching pads and sleds and release of booster rockets, tanks and other jettisonable equipment. Separation bolts and nuts have very high load carrying capability. However, these two devices are the mechanical single point failure in a release mechanism.

3.6.4. Pyrotechnic Initiator Lessons Learned

Pyrotechnic actuated devices have approximately a 99.9% reliability. The GSFC has experienced total success of pyro functions in space on recent S/C's (COBE, SHEAL, DE, etc.). However, we have learned a few lessons regarding pyrotechnic actuated devices.

During one of the SA deployment tests in 1989, a pin puller in a release mechanism was actuated, the pin was retracted inside its housing to release the SA but then rebounded back out of the housing. The normal function of a pin puller is to retract and stay flush inside the pin puller housing. The malfunction of this pin puller did not stop the deployment of the SA because of the mechanical redundancy of the release mechanism. The rebound of the pin outside the housing forced us to investigate our pin puller design. The result of this investigation showed that an extra shear pin hole was accidentally drilled at 120^{o} away from the original shear pin hole on the pin puller. The extra shear pin hole reduced the buckling strength of the pin. Therefore, when the pin puller was actuated, the pin was retracted and came to an instantaneous stop inside the housing at the end of travel; the impact force caused the pin to break in buckling and the top portion of the pin protruded back to the outside of the pin puller housing. The extra shear pin hole was not detected at the component inspection level because the pin (piston) drawing only required measuring the hole location and hole diameter but did not require the counting of the number of shear pin holes. The extra shear pin hole was not detected at the pin puller assembly level inspection by using X-ray either. Up to that time, pyro actuated devices were only required to be X-ray'd on one side view of the actuator assembly. Therefore, the extra shear pin hole was not shown in this particular X- ray view. After this incident, we recommended that all pyro actuated device assemblies have two X-ray pictures taken - one on the side view and one turned 90^{o}.

Pyrotechnic pin pullers have millisecond actuation times; therefore, they normally are subjected to very high impact forces (impulse forces). From past design experience in pyro actuators, we learned that the material impact strength is usually a more critical variable than the ability of the pin puller to withstand the high inertia load of a deployable system, especially if the device is required to operate in a cold environment. Material impact strength drastically drops when the temperature drops. It is recommended that when the inertia load of the deployable system is low, a more ductile material be selected over a brittle material for the actuator housing and piston. It is also recommended that for the pin puller lot acceptance test, at least one test be to subject the pin puller to zero inertia load in the shear direction and in a cold operating temperature actuate the pin puller with 125% explosive powder material. The zero external shear load allows the pin (piston) to retract at a higher velocity which, in turn, yields a higher impact force. The material impact strength is also lower at the lower temperature. This is the worst case test for impact strength.

Actuators are actuated by the high pressure and high temperature gas which is produced during the pyro reaction. The pyro reaction is initiated by an electrical current in the pyro bridgewires. Some pressure cartridge (pyro) designs have a single bridgewire and some pyro designs have double or redundant bridgewires for electrical redundancy. Single bridgewire pyros normally use a non-hygroscopic initiation powder material. However, the double bridgewire pyros must use a hygroscopic initiation material to satisfy the insulation resistance requirement between bridgewires and the housing case and between bridgewire to bridgewire. The hygroscopic initiation powder has a tendency to absorb moisture which will cause the malfunction of a pressure cartridge. If redundant bridgewires are required, then an initiation powder bake-out procedure and pressure cartridge loading procedure must be carefully reviewed and applied by the pyrotechnic engineer as well as the deployables engineer. If electrical redundancy is not required, the non-hygroscopic initiation powder is recommended. The other option for electrical redundancy is to use two pressure cartridges per actuator. The drawback for this double pressure cartridges design is that it requires a larger envelope to accommodate the two pressure cartridge configuration. The total weight of an actuator and pressure cartridge assembly will also increase. If weight and space are critical, this double pyro actuator configuration is not recommended.

The actuator is actuated by a high pressure and high temperature gas. The pressure of the gas will be reduced to a lower level (but not to a zero pressure) as the actuator cools off to the surrounding environment temperature. Due to the differential pressure, some gas molecules will leak out of the actuator assembly during the actuation, and some over the period of time after the actuation, because most pyrotechnic actuators have only a single Viton O-ring. If the S/C requires an extremely high level of cleanliness, then a hybrid sealing system must be designed for the actuator assembly. Hybrid sealing system design consists of a Viton O-ring and silicon O-ring combination that are located side by side in the leakage path. The Viton O-ring is capable of withstanding high temperature and high pressure gas but it allows the gas molecules to leak through when the actuator assembly gets cold. The silicon O-ring cannot withstand the high temperature and high pressure gas but it works very well for cold temperature gas. By locating the Viton O-ring on the side that it is going to be contacting hot gas and the silicon O-ring on the outside of the leakage path, we have a hybrid O-ring sealing system to prevent leakage in either hot or cold environment.

Figures 14 through 17 show various pyrotechnic devices. Subassembly sketches of pin puller, cable cutter and bolt cutter applications are included.

3.7 INITIAL MOTION DEVICES [E. Devine/SAI]

In cases where tiedown points are located at some distance from the deployment hinge

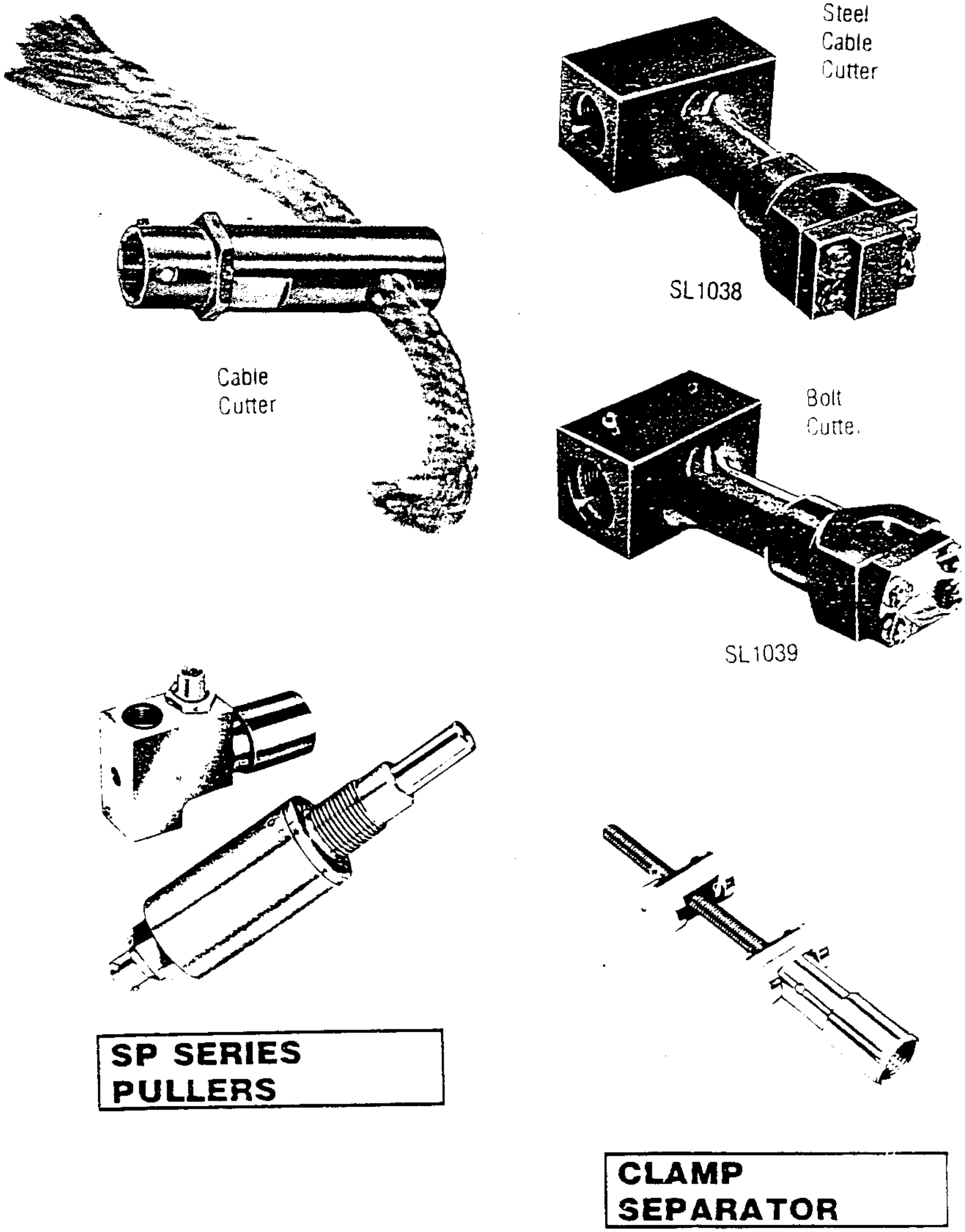
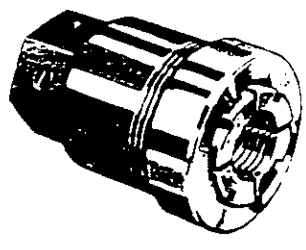
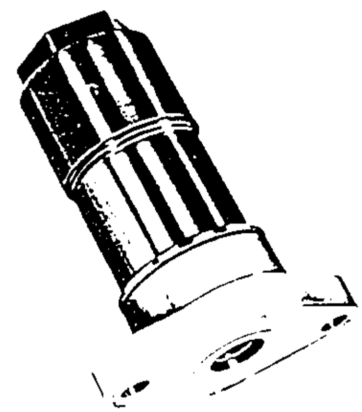


Figure 14. Pyrotechnic Actuated Devices.



NON-CAPTIVE SEPARATION NUTS



SH200 BOLT CATCHER

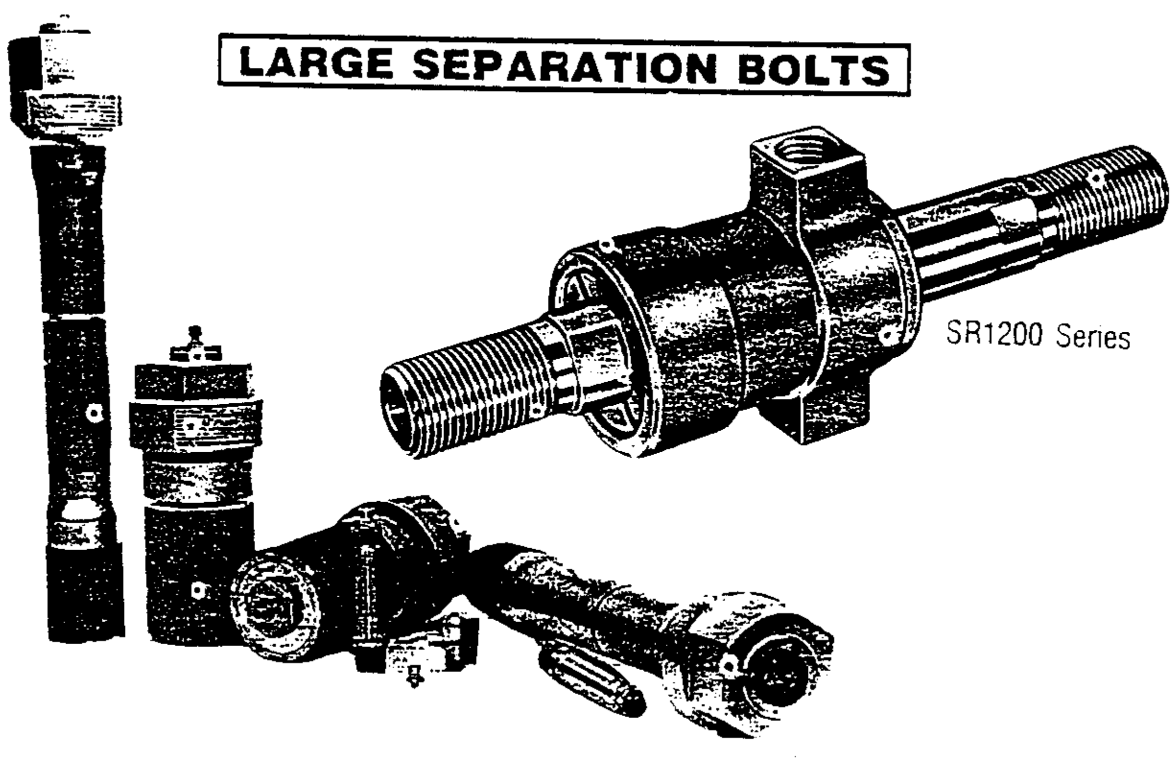


Figure 14. Pyrotechnic Actuated Devices - continued.

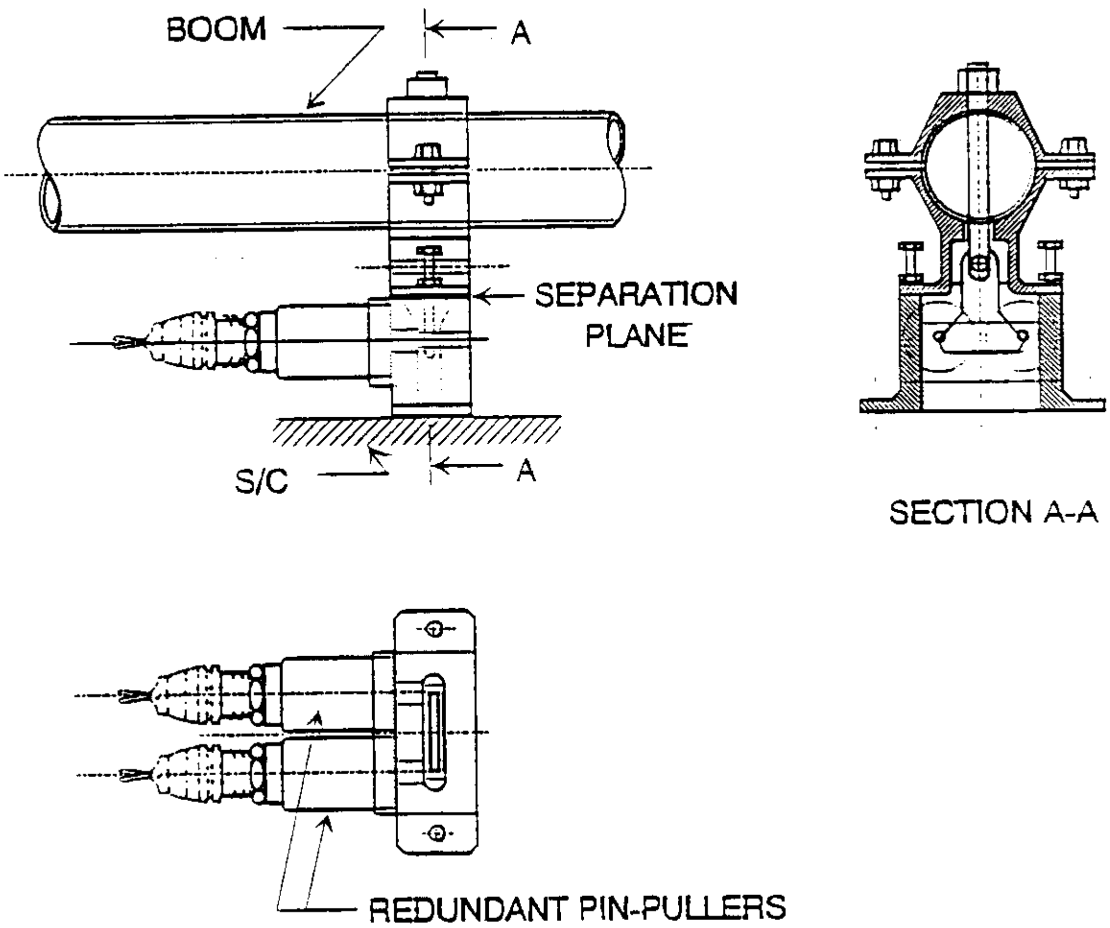


Figure 15. Pin Puller Application.

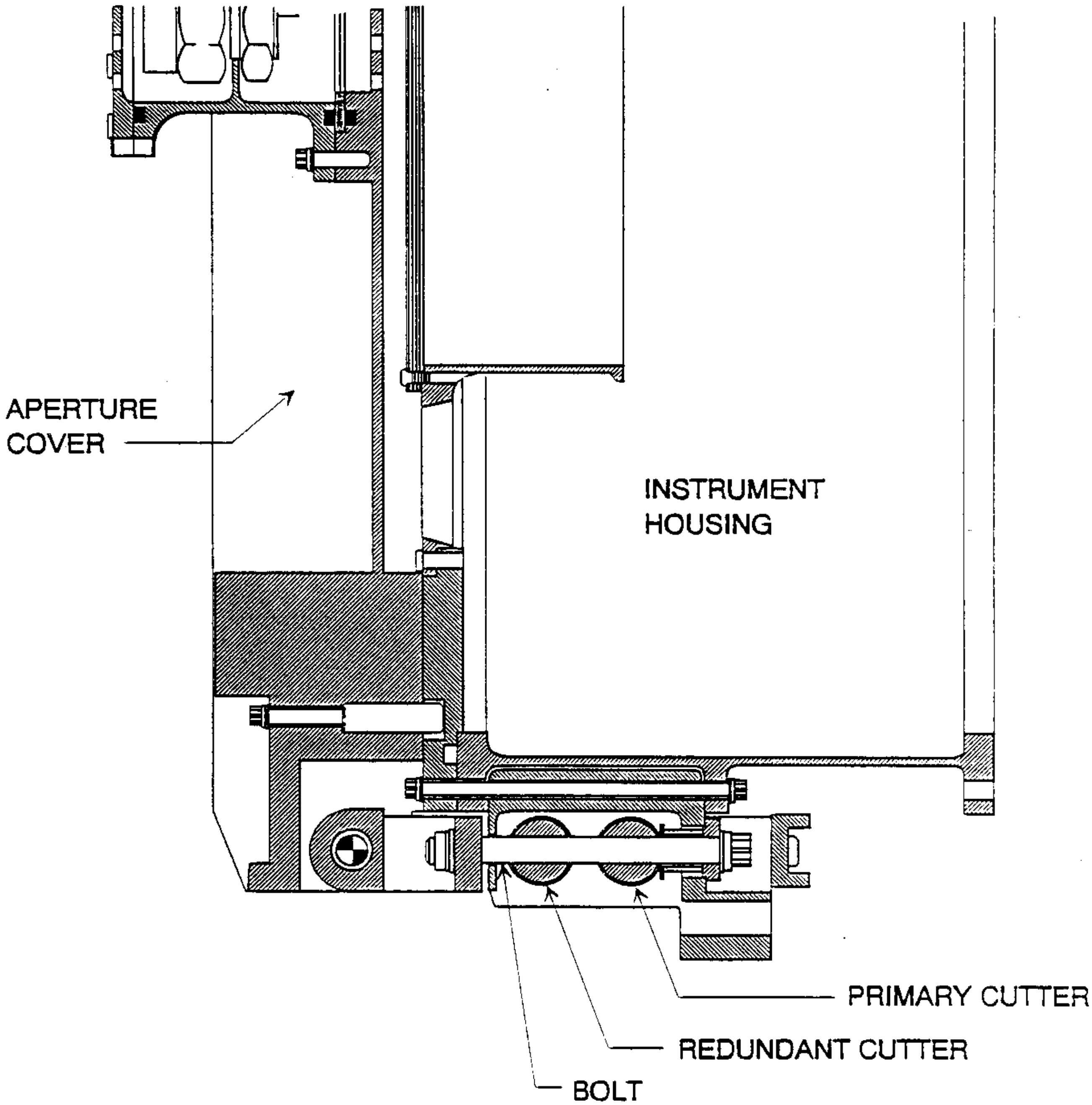


Figure 16. Bolt Cutter Application.

or actuator (for example boom mounted packages) one should consider adding a kick-off spring or forcer located at, or adjacent to, the tiedown point. This simple device can prevent a small hangup force at the tiedown point from stalling the actuator due to the large disadvantageous moment arm. A further advantage accrues since the kick-off spring stroke need be only enough to assure separation from the restraint fixture. Thus, a minimum of excess kinetic energy is imparted to the deploying system contrasted to what would result from increasing the hinge torque enough to overcome the hangup.

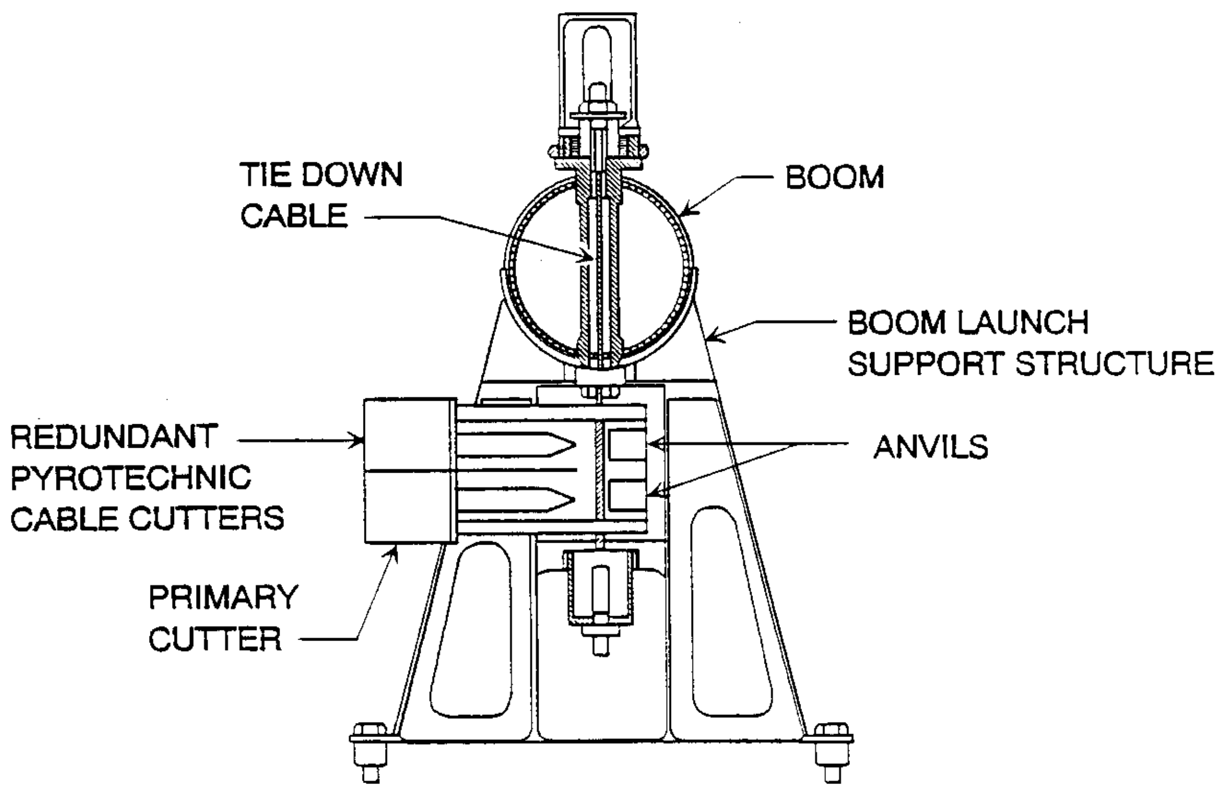


Figure 17. Cable Cutter Application.

4.0 DEPLOYED APPENDAGE VERIFICATION TEST PROGRAM [R. Coladonato/750]

4.1. PHILOSOPHY

At the GSFC, the basic philosophy is to establish confidence prior to launch that all deployables will meet mission objectives and requirements satisfactorily and without degradation. Confidence is obtained by performing a series of interdependent tests and analyses for the flight hardware. Tests and analyses are performed at the component, subsystem and system level to the extent practicable and deemed appropriate.

4.1.1. Analyses

Analyses are performed in order to take into consideration adverse buildup of tolerances, thermal distortions and mechanical misalignments, as well as the effects of static and dynamic displacements induced by particular mission events. Also a kinematic analysis of all mechanical operations is conducted to ensure satisfactory performance and adequate margins under worst-case conditions, to ensure that satisfactory clearances exist for both the stowed and operational configurations as well as during any mechanical operation, and to ensure that all mechanical elements are capable of withstanding worst-case loads that may be encountered.

4.1.2. Testing

Tests are performed in order to qualify all parts of the deployable hardware and to demonstrate that all aspects of the deployment operation will function properly. The configuration for the test is such that the actual flight conditions are duplicated as closely as possible to include representations for things that could affect the operation, such as thermal blankets and wiring harnesses. Mechanical function testing for nominal conditions, which is defined as the most probable conditions to be expected during normal flight, is done to establish proper functioning. Other mechanical function tests to demonstrate margins beyond the nominal conditions are conducted in order to account for variations in parameters such as temperature, friction, spring forces and stiffness.

4.2. TYPES OF TESTING

Testing begins at the component level and progresses to the subsystem and system level. Environmental testing for strength verification, vibroacoustics, shock and thermal vacuum is conducted at all levels of assembly when necessary. The choice of what environmental tests to do at what level of assembly is variable and project dependent. Vibroacoustic and shock testing is usually done at the system level in addition to any other environmental testing done at other levels of assembly.

Mechanical function testing is sometimes performed at the component level, and is always done at the subsystem level or system level. Testing at hot and cold temperature extremes is done at all levels of assembly. In some cases it is not feasible or practical to perform a system level deployment test at extreme temperatures. For these cases, a program is developed which will provide adequate confidence through the accumulation of test and performance information from testing at temperature extremes which is done at the component and subsystem levels of assembly.

Some payloads have more than one deployable item. If the design is such that the deployment sequence is critical, then deployment testing which duplicates the actual deployment sequence is performed. Otherwise, separate deployment of individual items is generally done. Also, some payloads may be spinning at an appreciable rate at the time of deployment for one or more items. If this is true, then the deployment test is conducted using either a spinning payload when possible or equivalent acceleration (centrifugal) environment.

Many deployables are released from the stowed configuration through the use of pyrotechnic devices. At the subsystem or system level, a comprehensive mechanical function test consists of a fully automated deployment from release to lock-in using the actual release devices. Regardless of the level of assembly used to performed the comprehensive test, a limited mechanical function test is done on the flight payload. The limited function test typically consists of a manual demonstration of proper release and initial motion. Sometimes a walk-out is done to exercise the hardware throughout the full motion. The limited function test also serves as a last chance to verify that there are no as-built interferences from such things as thermal blankets and wiring harnesses.

The GSFC designs and builds a wide range of payloads using deployables. Deployable items such as SA's, antenna booms, thermal shields, other booms and antennas themselves are part of GSFC deployables. Payloads ranging from a few hundred pounds to several thousand pounds are accommodated. The smaller payloads may have deployables that are able to be self-supporting in a 1-G field, whereas the larger payloads typically have deployables that have to be G-negated during deployment testing. The complexity of the deployment test is related to the size of the deployable and the need for G-negation. Testing is greatly simplified if G-negation is not required.

Deployment tests that require G-negation are performed at the GSFC using one of several different techniques. The most common types of G-negation systems employed at the GSFC are the air pad or bearing system, the bungee cord system and the counterweight/pulley system. All three have proved to be successful. However, the counterweight/pulley system is the least desirable because of the potential for problems with the system due to inherent friction.

Different types of G-negation systems have been used by others in the aerospace industry. Systems that take advantage of buoyancy, such as helium filled balloons and surface water tanks, have also been used successfully. Although the buoyancy type of test has not been done at the GSFC, it could be implemented. For deployment testing of large items, facilities that have lots of floor space and room height are very desirable, and the GSFC has such facilities.

As with all types of structural testing, there are many ways to accomplish the objective. At the GSFC, the project has a great deal of latitude in determining the particular verification program that will satisfy the requirement. The project must devise an acceptable technical program that fits within the programmatic constraints of cost and schedule.

5.0 CONCLUSION [K. Hinkle/731, J. Sudey/716]

The engineering design, analysis, and testing of flight deployed appendages are critical for the mission success of almost every S/C developed by the GSFC. During this development period, other issues aside from engineering often determine design and testing objectives. At the beginning of a project, an inadequate understanding of the functional operation of the deployable appendage or mechanism can result in poor schedule and cost planning. During the testing phase of the program, funding problems and difficult testing requirements often cause analyses to be performed in lieu of testing of actual flight hardware. The primary goal is to design flight hardware which can be performance tested during and after each environmental test with the hardware in the closest configuration to flight as possible. Testing is the best confidence source we have to ensure flight hardware will successfully operate on-orbit. Therefore, we should fly only what we test, and the resulting schedule and test associated with testing should keep pace during the last stages of appendage development.

In addition, almost every deployment device related to S/C on-orbit configuration change is a mission catastrophic single point failure if it does not function properly. The following are some ground rules from lessons learned for designing such devices.

1. All deployed appendage programs must have ETU's.

2. All flight units and ETU's must be tested to determine deployment margins.

3. Analyses must be verified by judicious hardware testing programs.

4. There must be adequate life testing early in the program.

5. There must be redundant backup systems in all critical areas.

6. Worst case analyses and Failure Modes Effects and Critical Analyses (FMECA's) must be performed and verified by actual hardware testing. Conditions that must be considered include worst case friction, misalignment and excessive preload caused by temperature gradients and/or manufacturing tolerance buildup.

7. All devices should be designed to be as simple as possible to do an adequate job.

8. Consider the effects of mounting system redundancy and structure induced input forces not only on the devices but also on the internal components of the devices.

9. Look for all possible hostile environmental effects and design to minimize their impact. Pay particular attention to vacuum, thermal control and zero G effects that are not always intuitive to the designer.

10. Select devices which are individually tested and reusable rather than devices that are statistically qualified to a pass/fail criterion.

11. Use the largest possible margin of operation in all devices consistent with consideration of undesirable effects on the surrounding hardware. These undesirable effects include large forces developed by end-of-travel latch-up and shock from pyrotechnic device firing.

12. Make installations such that the devices can be verified for proper installation. Knowledge of preloads, position of parts, status of switches or other electrical interfaces should be known or testable.

Very often the above ground rules are compromised as part of a cost savings exercise. In most cases, failures result (often late in a program) that cost the program much more than if the above rules were followed in the first place. These rules have evolved from much experience and any program that considers violating them should be fully aware of the extremely serious risks being taken.

6.0 DEFINITIONS

1. Base speed. The range of stepping rates at which the motor and load can start from a standstill and remain in synchronization without having to use the ramping technique (from zero to the pull-in rate).

2. High speed. The range of stepping rates where the motor can run in synchronism to the input frequency of the controller, but cannot start/stop/reverse without resorting to ramping (from pull- in to pull-out rate).

3. Pull-in rate (or starting rate). The maximum demanded stepping rate to which the motor can respond to without loss of steps (synchronism) when initially at rest. This is, in general, much lower than the pull-out rate.

4. Pull-out rate. The motor speed (step rate) at which the maximum available motor torque (pull-out torque) equals the load torque. Above this rate the motor will lose steps, synchronism or possibly stop altogether.

5. Pull-out torque (dynamic torque). The maximum motor torque available at a given speed given by a speed versus torque curve. An inertial load has little effect on the steady state dynamic torque. On the contrary, friction must be overcome to move the load.

6. Ramping. The method of gradually increasing or decreasing motor speed to achieve high speed operation without losing synchronism. The time taken to ramp up to speed is the acceleration time and the time to ramp down is the deceleration time.

7. Static torque. Given by a torque versus rotor position with one state energized with a constant current.

8. Step rate. The rate at which the motor states/phases are excited resulting in a step per clock signal.

9. Step size. The incremental motion of the rotor shaft due to the excitation of a motor state/phase.

10. Stopping rate. The maximum stepping rate which can suddenly be switched off without the motor overshooting the last energized step.

11. <u>Torque ratio</u> (T_r) The relative size of the desired torque to produce rotation about an axis with regard to the minimum torque to produce that rotation, i.e.:

$$T_r = T_{available} / T_{max}$$

where: $T_{available}$ is the maximum torque which can be provided by the actuator device.

T_{max} is the maximum resistive torque (friction, etc.) that will prevent rigid body deployment motion of the device which is to be deployed.

A minimum value for T_r is typically four.

12. <u>Torque margin</u> (T_m) The extra amount of torque allowed to produce rotation about an axis beyond what is needed at a minimum. For appendage deployment this is typically three.

$$T_m = T_r - 1$$