N91-24610

POINTING/ROLL MECHANISM FOR THE ULTRAVIOLET CORONAGRAPH SPECTROMETER

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ABSTRACT

A pointing/roll mechanism for the Ultraviolet Coronagraph Spectrometer (UVCS) is presented along with a description of the mechanism control algorithm. The mechanism, operating in space, will position the 2.1 meter long, 0.7 meter diameter UVCS instrument in pitch and yaw, within a 54 arc-minute half angle cone, and will also allow it to rotate ±179.75 degrees. After considerable design effort, an optimum mechanical solution was achieved, which meets all scientific requirements as well as weight, volume, and power budgets. Evolution of the mechanism is presented along with the design status.

INTRODUCTION

Background

The Ultraviolet Coronagraph Spectrometer (UVCS) is a scientific instrument that will be used to observe the solar corona aboard the Solar and Heliospheric Observatory (SOHO) spacecraft. SOHO is a mission to study the sun, sponsored by European Space Agency (ESA) along with National Aeronautics and Space Administration (NASA). SOHO will be launched into a halo orbit about the L1 Lagrangian point where a gravitational null exists in the sun-earth system. L1 is an ideal location for continuous and unobstructed observation of the sun. During the two-year mission, the UVCS will aid in the study of such phenomena as solar wind acceleration in the extended solar corona. Figure 1 shows the UVCS instrument located on the SOHO spacecraft.

Scientific Goals

The UVCS instrument will provide ultraviolet spectroscopic measurements of the entire solar coronal region from near the solar horizon out to 10 solar radii (Ro). The telescope, pointed at the sun-center, receives coronal radiation from the primary field of view (FOV) as shown in Figure 2. Internal mirrors focus a thin strip of this radiation onto entrance slits of a spectrometer. A mechanism tilts the mirrors to scan coronal images across the entrance slits. By rotating the telescope, different radial regions of the corona can be viewed. The extended FOV can be accessed by offset pointing the telescope axis to observe direct solar radiation.

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UVCS Operation

Once in orbit, the SOHO spacecraft will closely maintain its longitudinal axis pointed at the sun-center. This provides a stable platform from which the UVCS instrument can perform coronal or solar observations. Prior to a typical observation, the UVCS telescope will be rolled about its longitudinal axis followed by a pointing adjustment in pitch and yaw (see Fig. 3., which shows the pitch, yaw and roll axes). The pointing adjustment will be necessary to re-align the telescope to required accuracy.

UVCS System

The UVCS system consists of the telescope and the forward and rear support assemblies (see Fig. 3). The telescope is a long tubular structure consisting of a seven-sided graphite fiber reinforced epoxy (GFRE) trussed framework to which is attached a cylindrical end-cap. The aperture cover/door mechanism is mounted to the end-cap while all the other optical components (mirror mechanism, sunlight trap and spectrometer) are housed inside the the telescope (see Fig 4).

The forward and rear support assemblies provide the pointing and roll capability to the UVCS instrument and are presented in detail in this paper.

UVCS POINTING/ROLL REQUIREMENTS

During operation, the UVCS is required to pitch and yaw within a pointing envelope defined by a 54 arc-minute half angle cone. The pointing accuracy throughout this envelope is 60 arc-seconds with the exception of the central 32 arc-minute cone, where, the instrument must be capable of fine pointing corrections with a precision of 2 arc-seconds. The mechanism is also required to rotate the telescope ±179.75 degrees with an accuracy of 6 arc-minutes.

The mechanism, which cannot exceed 20 kg mass, must be designed to operate 60,000 pointing cycles and 23,000 rotation cycles, providing a factor of safety of 4 over the actual mission requirements. To insure reliability, redundant position sensors and motor windings are required.

UVCS POINTING/ROLL MECHANISM DESCRIPTION

Pointing Mechanism

The pointing portion of the mechanism consists of three primary subassemblies: the pointing ring, aft support assembly and the pointing roller assemblies. A titanium pointing ring is attached to the GFRE telescope structure by filling the radial gap between the ring and the structure with polyurethane bonding material. The gap is sized to minimize thermally induced stresses due to the coefficient of thermal expansion (CTE) difference between the ring and the essentially zero CTE structure. The ring surfaces which contact the pointing and retention rollers are hard coated and dry film lubricated to minimize wear.

The aft structure assembly (see Fig. 5) provides telescope support and two axis rotational freedom. The structure incorporates four GFRE tubular legs which mount between the spacecraft and a GFRE roll drive interface plate. Each leg end has a spherical bearing at the mounting interface. The legs are oriented such that they form perpendicular 4-bar linkages. The geometry of the 4-bar link systems is such that they have coincident instantaneous centers of rotation, emulating the functions of a two axis gimbal. They are also positioned such that the telescope mounting is stiff in the radial directions. Axial stiffness is tailored to meet minimum operational requirements and to minimize stresses caused by thermal mismatch between the telescope structure and the aluminum spacecraft mounting plate when the telescope is launch locked.

The two pointing roller assemblies each consist of a linear translation stage, a crowned pointing roller, a spring loaded retention roller and a roller support (see Fig. 6). Pointing is accomplished by independently positioning the pointing rollers using the linear stages causing the telescope to pivot on the aft support. Essentially, moving the rollers towards or away from each other for pitch and together in the same direction to adjust yaw.

The pointing rollers are 52 mm diameter double row ball type cam follower bearings with a 500 mm radius crown. The linear translation stages are stepper motor driven crossed roller type with lapped leadscrew/nut assemblies for minimum backlash. The 2.5 mm lead pitch combined with a 1.8 degree/step motor provides .02 mm of linear travel per step. This corresponds to about 1.5 arc seconds of telescope motion per step. A lapped nut was selected over a ball nut because of a requirement that the mechanism not back drive when subjected to telescope weight during ground operations with motor power off. The lapped nut meets the requirement without the necessity of adding a gear reducer to the motor, which would increase size, weight and backlash. A linear potentiometer is incorporated into each stage to monitor position. The linearity of the potentiometer is sufficient to meet the 60 arc-second pointing accuracy requirement (sun sensors provide fine position feedback to meet the 2 arc-second accuracy requirement within the central portion of the pointing envelope). The crowned rollers permit the required telescope motion while minimizing contact stresses at the pointing ring interface. Each roller assembly includes a retention roller which rides on an inner surface of the pointing ring. These rollers are also crowned and are spring loaded to 49 N each. The rollers are mounted with a pivoting link system that insures the ring maintains contact with the pointing rollers during operation. The pointing rollers and the retention rollers are finished with a zirconium nitride coating for wear resistance.

Roll Mechanism

The roll mechanism (see Fig. 7) consists primarily of a pre-loaded duplex bearing pair, a resolver, a harmonic drive gear reducer, a stepper motor, a rotation shaft and a rotation housing. The rotation shaft and housing are silicon-carbide reinforced aluminum for high stiffness and to match CTE characteristics of the bearings and resolver. A size 11, 90 degree stepper motor drives the telescope through a gearhead and Harmonic Drive reducer which yields a 5000:1 overall gear reduction. This provides approximately 1 arc-minute of telescope rotation per step. The Harmonic Drive was selected for efficient packaging and high torsional stiffness. The resolver is a redundant pancake type with 2 arc-minute accuracy. The rotating shaft mounts to the telescope and the housing mounts to the interface plate on the aft support (see Figs 5 and 7). A mechanical stop is incorporated to limit rotation.

Launch Lock Configuration

The pointing mechanism is not capable of withstanding the severe vibrational loading caused during launch. Therefore, an auxiliary launch lock system is engaged. The telescope is lowered by the pointing mechanism until pins on the telescope engage specific locations on the forward support. The pointing rollers continue to move outward and are separated from the telescope pointing ring. Afterwards, pawls are deployed which hook around the pins locking the telescope in place. This system provides restraint in three axes and transfers loads directly to the spacecraft through the interfaces. To disengage the launch lock, the pawls are rotated, which allows the telescope be lifted by the pointing mechanism from the launch lock restraints.

EVOLUTION OF THE UVCS POINTING/ROLL MECHANISM

The telescope pointing/roll mechanism went through a number of evolutionary iterations and many different design approaches were considered. It is worthwhile to review the main development stages.

From the mechanical point of view, the UVCS telescope is essentially a 91 kg cylinder, 2.1 meters long and 0.7 meter in diameter, with the center of gravity located approximately 0.9 meters along the roll axis behind the front aperture. To be able to fit within the allowed volume and not obstruct the front aperture, the mechanism was constrained to occupy volume available to the rear of the telescope and some space between the telescope and the spacecraft mounting panel.

The first consideration was to place a unified pointing/roll mechanism in the available volume at the rear of the telescope. With the center of mass located near the front of the telescope, the mechanism was required to accurately position and stabilize the large cantilevered inertial load. Early analysis indicated that the mechanism could not meet pointing, stability and launch requirements within volume, weight and power restrictions.

It was decided next to consider placing a 2-axis gimbal at the aft end and mount a separate mechanism as far forward as possible to do the actual pointing. This maximizes pointing "leverage" and allows larger mechanism step size. The concept at the proposal stage included a single mechanism located up front with rollers that would pinch a ring mounted around the telescope and a 2-axis linear motion device that would move the telescope up and down and side to side for pointing. The pinch rollers would double as a traction drive system for rolling the telescope.

At the preliminary design phase, the pinch roller concept was investigated in detail. It was determined that there was not sufficient space to install a single mechanism which met the stiffness and pointing requirements. There was also a concern about possible slippage of the traction roll drive system. Another issue concerned the fact that the contact point between the rollers and the pointing ring moves along an arc during telescope pointing. This required some compliance in the mechanism to allow for angular and distance variations.

This led to a mechanism where the telescope would rest on a pair of crowned rollers, each attached to a linear translation stage. With this arrangement, pointing was accomplished by independently moving the rollers together or apart to point the telescope for pitch motion or in the same direction for yaw motion. Roll was accomplished via a cable drive system where a cable is wrapped radially around the telescope near the pointing ring and around an idler pulley and a motorized pulley. The pulleys were attached to each of the translation stages near the crowned pointing rollers. The cable also acted to hold the telescope down onto the rollers.

This design was determined to have contamination drawbacks due to the use of stranded cable which was required for its flexibility. There was a concern that the cable would be impossible to thoroughly clean and that particulates would continually work out of the braid during operation. It was also not desirable from a reliability standpoint. Redundant cables required a considerable amount of room and if a cable were to break, the loose end stood a good chance of getting tangled. The cable drive also was sensitive to the telescope angular motion.

At this point, a mechanism trade study was performed and it was decided to abandon the cable drive system. The crowned roller scheme, however, was determined to be a very good approach and it was decided to retain the pointing system, but incorporate a direct drive roll mechanism into the gimbal where it is immune to telescope angular motion. Retention rollers were added to the pointing mechanism to insure contact of the telescope ring with the pointing rollers.

The next major design change was driven by weight. A detailed weight analysis disclosed that the instrument was exceeding the overall mass allocation. It was recognized that a large percentage of the weight of the pointing/roll mechanism was the gimbal assembly. Launch loads and a 70 Hz axial stiffness requirement had driven up the weight of the gimbal and aft support structure to provide adequate stiffness. Clarification of the specification revealed that the 70 Hz resonant frequency was only a launch requirement. This allowed us to include a simple axial launch lock at the front support to meet the 70 Hz requirement and re-evaluate the telescope pivoting method in order to reduce weight.

The design, presented earlier, precipitated out of this evolutionary process.

MECHANISM CONTROL ALGORITHM

Successful operation of the UVCS pointing mechanism requires a simple control algorithm that will position the pointing rollers to proper locations, orienting the UVCS telescope to a desired angular orientation. A suitable algorithm, in the form of mathematical equations, relates θ and ϕ (pitch and yaw angles of the telescope) to Δ_1 and Δ_2 (positions of the pointing rollers). In order to develop such an algorithm, it was necessary to first mathematically describe the pointing geometry which yielded a set of simultaneous nonlinear equations. Since the complex equations were not suitable to be used directly, simplified polynomials were developed which approximate the solution to a desired accuracy throughout the pointing envelope.

Description of Geometry

The general configuration of the UVCS pointing system is shown in Figure 8. For a given orientation of the telescope (mathematically described by angles θ and ϕ), there is a unique position for each pointing roller. Also, the aft support legs assume specific orientations to accommodate the telescope. This logic is the basis for the analysis, since the mathematical description of the system must also be unique. Vector equations describe the contact point between the pointing rollers and the pointing ring as well as positions of the aft legs. Additional equations are included to constrain the system. The total number of equations developed is 28 with 28 unknowns.

Solution of Simultaneous Equations

From the preceding discussion, the description of the pointing geometry yielded a system of equations which are applicable to a particular orientation of the instrument. For a pointing envelope of ±54 arc-minute in both pitch and yaw, the system of equations was solved every 2 arc-minutes. The resulting data is represented as two surfaces, each describing the position of one roller versus telescope pitch and yaw.

Curve Fit of Pointing Surfaces

A curve fit approximation of the pointing surfaces was performed using polynomials in two variables. It was found that fourth power polynomials were required to achieve the desired 2 arc-second accuracy. The left pointing surface and corresponding polynomial is shown in Figure 9.

Design Optimization

In order to minimize the amount of sliding of the pointing roller on the roll ring, the pointing stages were placed on a angle of 6 degrees as shown in Figure 5.

This angle was mathematically optimized to account for relative motion between the pointing ring and the pointing rollers due to the arc motion of the telescope.

CONCLUSION

To date, the mechanical layout and analysis of the mechanism is complete as well as 95% of the mechanical drawings. Wear testing has been done to select finishes for the pointing ring and rollers and bearing lubrication analysis has been performed. Fabrication is under way on items for an engineering model and units to be life-tested. Computer simulation models have been developed to verify the motor sizes and control electronics, and also to determine the mechanisms' momentum and torque disturbance inputs to the spacecraft. These models include all of the mechanical characteristics of the system such as friction, spring rate and inertia. The models have been run at the minimum and maximum values (including safety margins) of the constituents to prove the mechanisms will operate correctly under any condition. Finite element modeling has been done on critical items such as the rotation shaft and the pointing ring. The pointing ring bond was also modeled to check for deflection of the ring during 1-g loading and thermally induced stresses of the bond material. The mechanism control algorithm is being implemented in the system software.

ACKNOWLEDGMENTS

UVCS is a scientific experiment of the Smithsonian Astrophysical Observatory (SAO) with Dr. John L. Kohl as the Principal Investigator. The project is also in collaboration with a group of scientists and engineers from Italy and with participation from Switzerland, Norway and Germany as well as several American institutions. Ball Electro-Optics/Cryogenics Division will provide the telescope assembly and electronics unit, Aeritalia and Officine Galileo will provide the spectrometer assembly and Contraves Corporation will provide the diffraction gratings and cells used in the spectrometer.

The authors wish to thank SAO for permission to publish this paper and all the UVCS participants for their support.

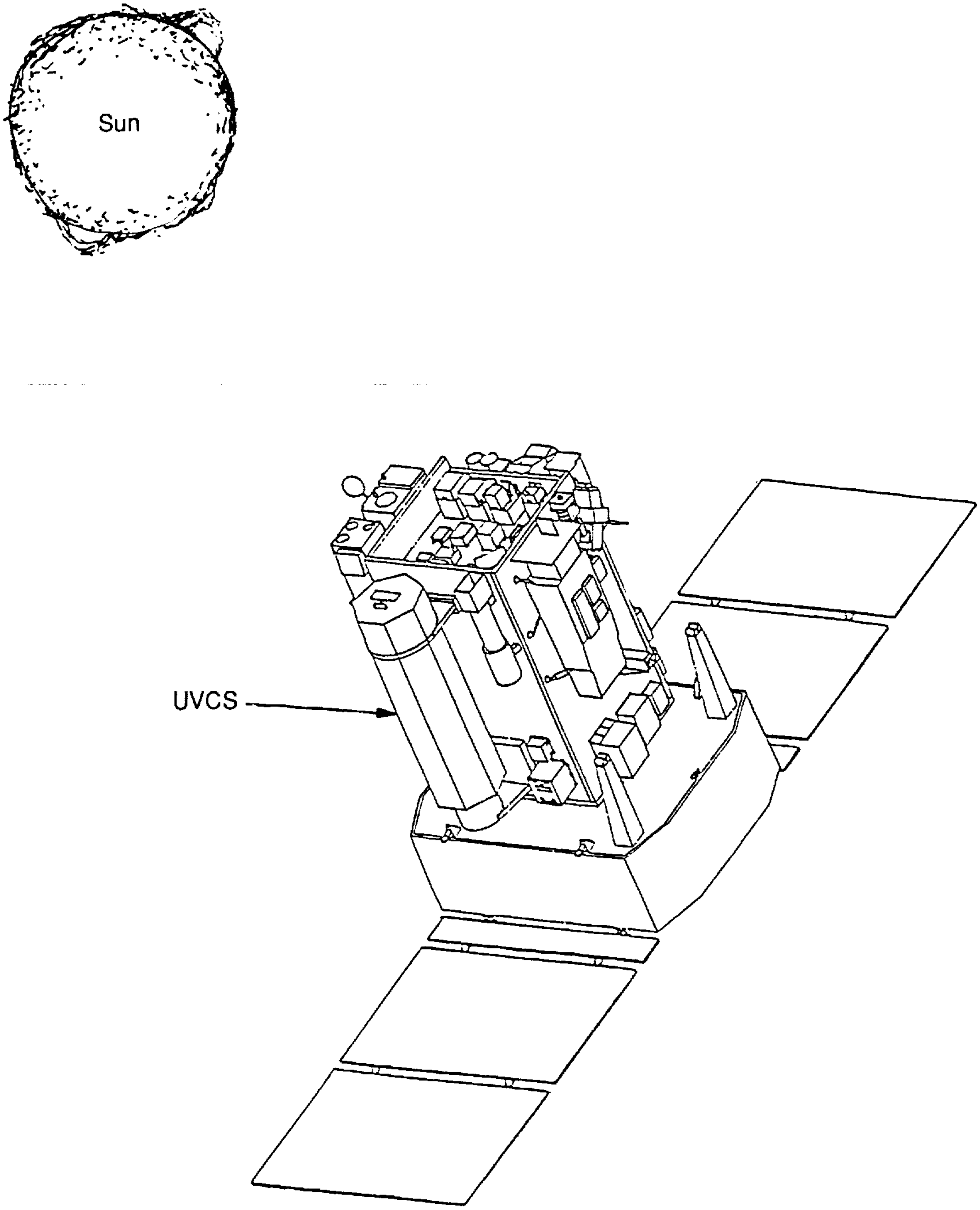


Figure 1. SOHO spacecraft with the UVCS instrument.

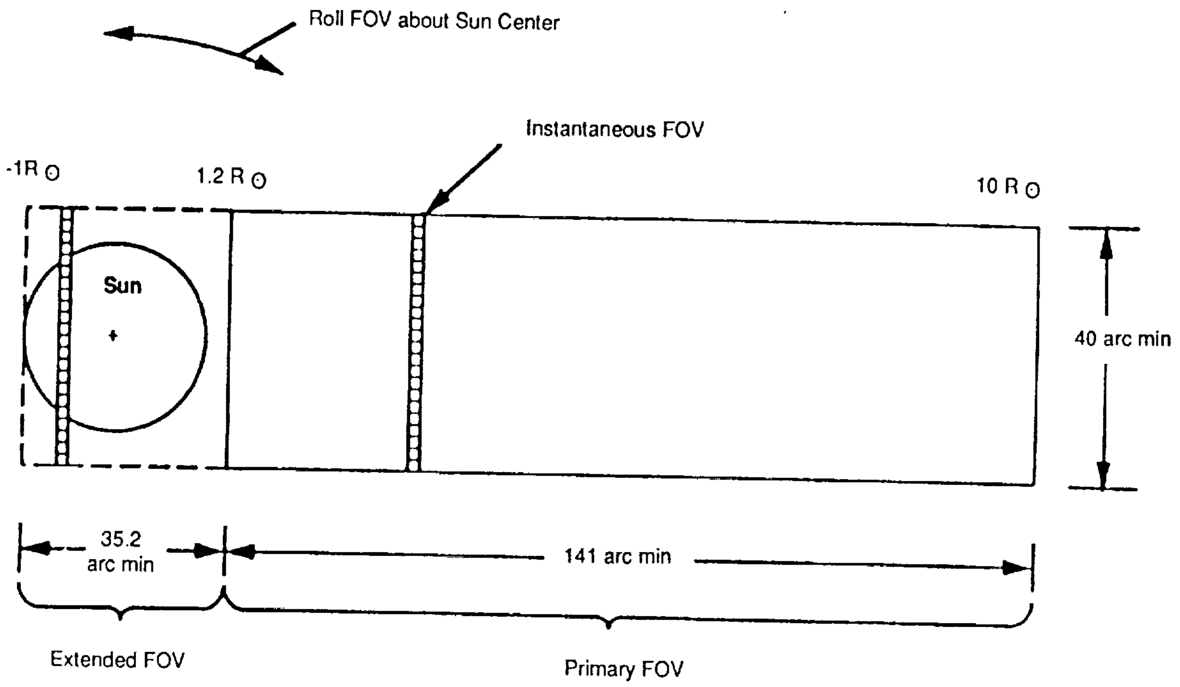


Figure 2. The UVCS instrument field of view (FOV).

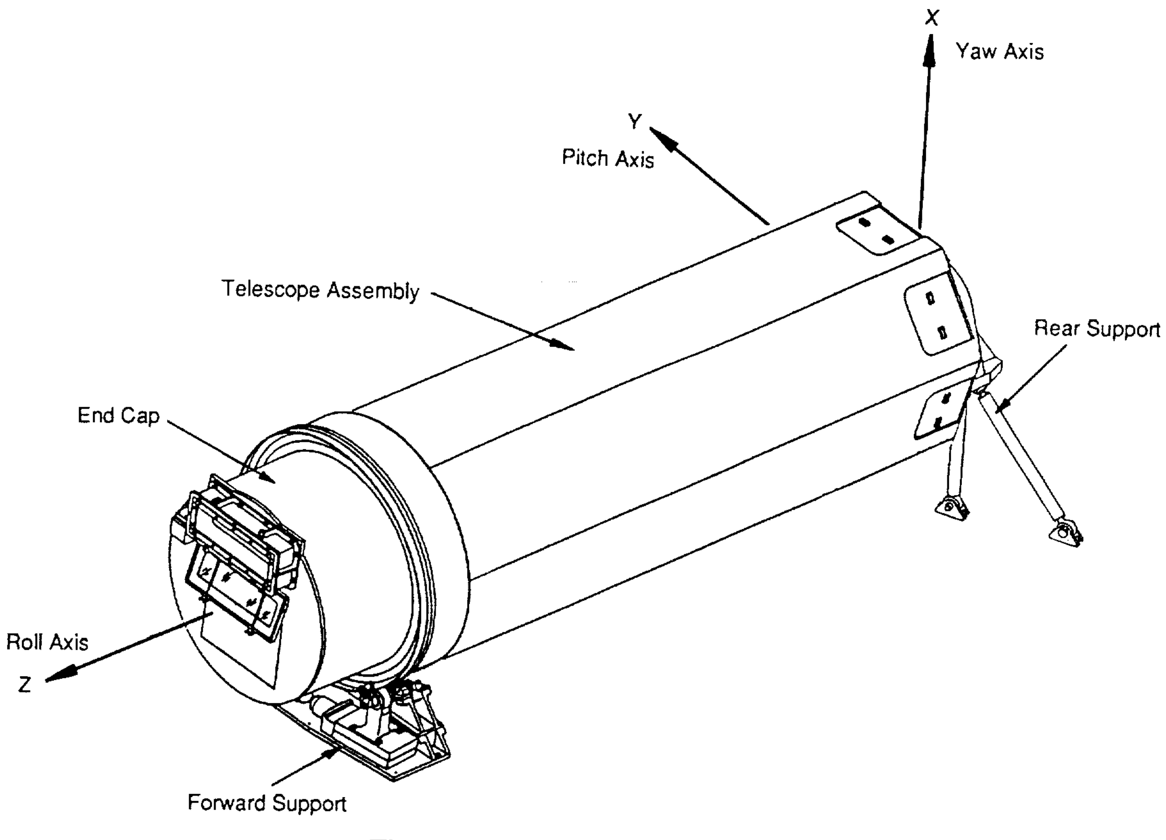


Figure 3. The UVCS instrument .

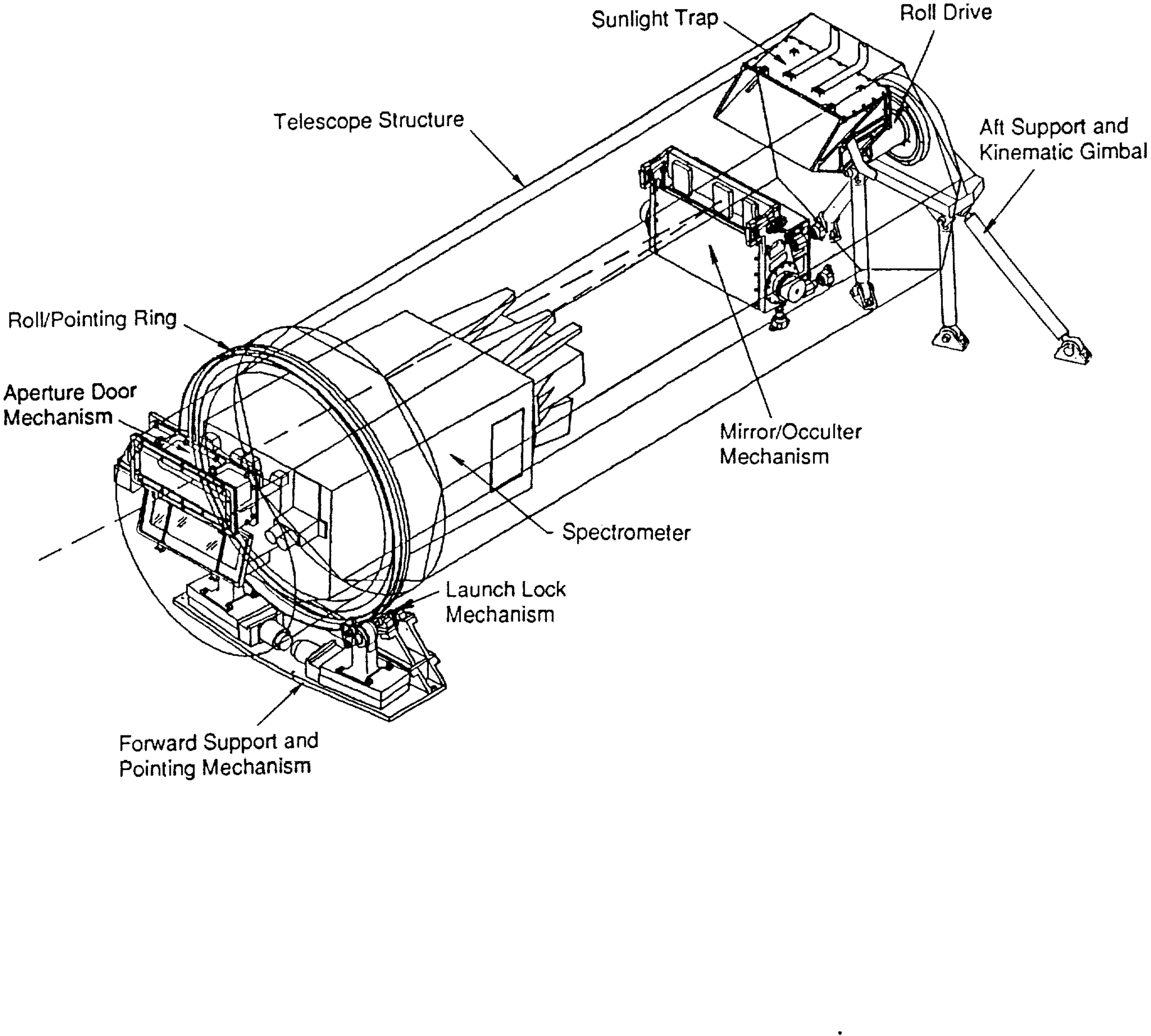


Figure 4. UVCS instrument system overview.

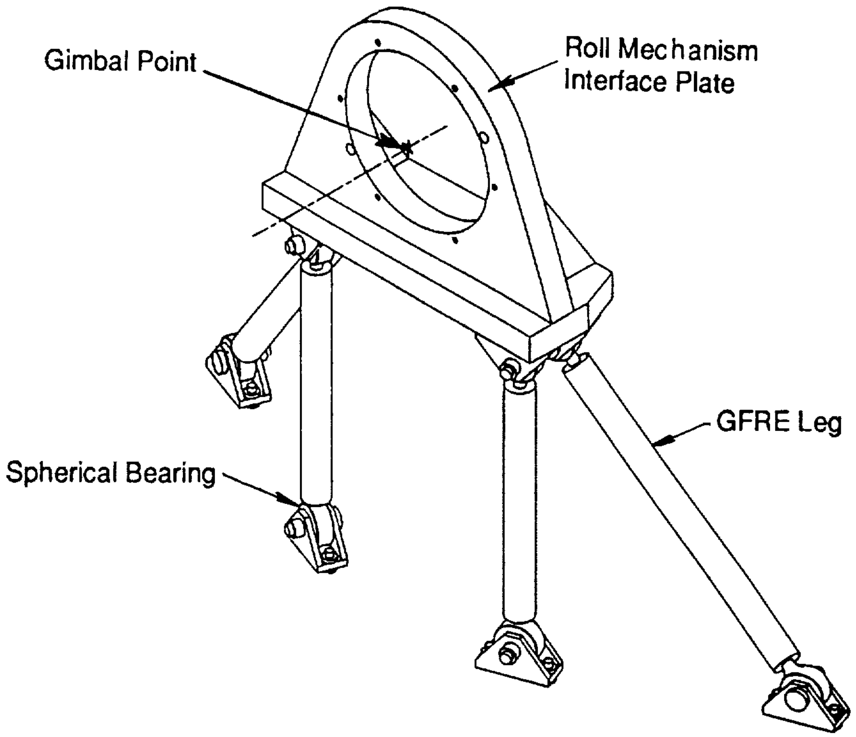


Figure 5. Aft support and kinematic gimbal.

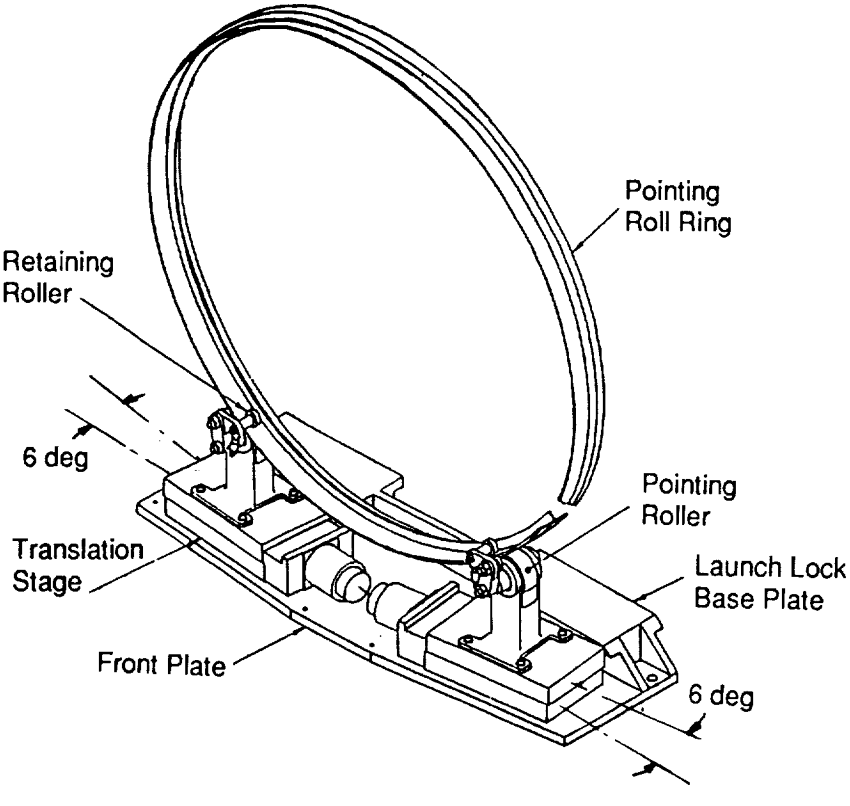


Figure 6. Pointing mechanism.

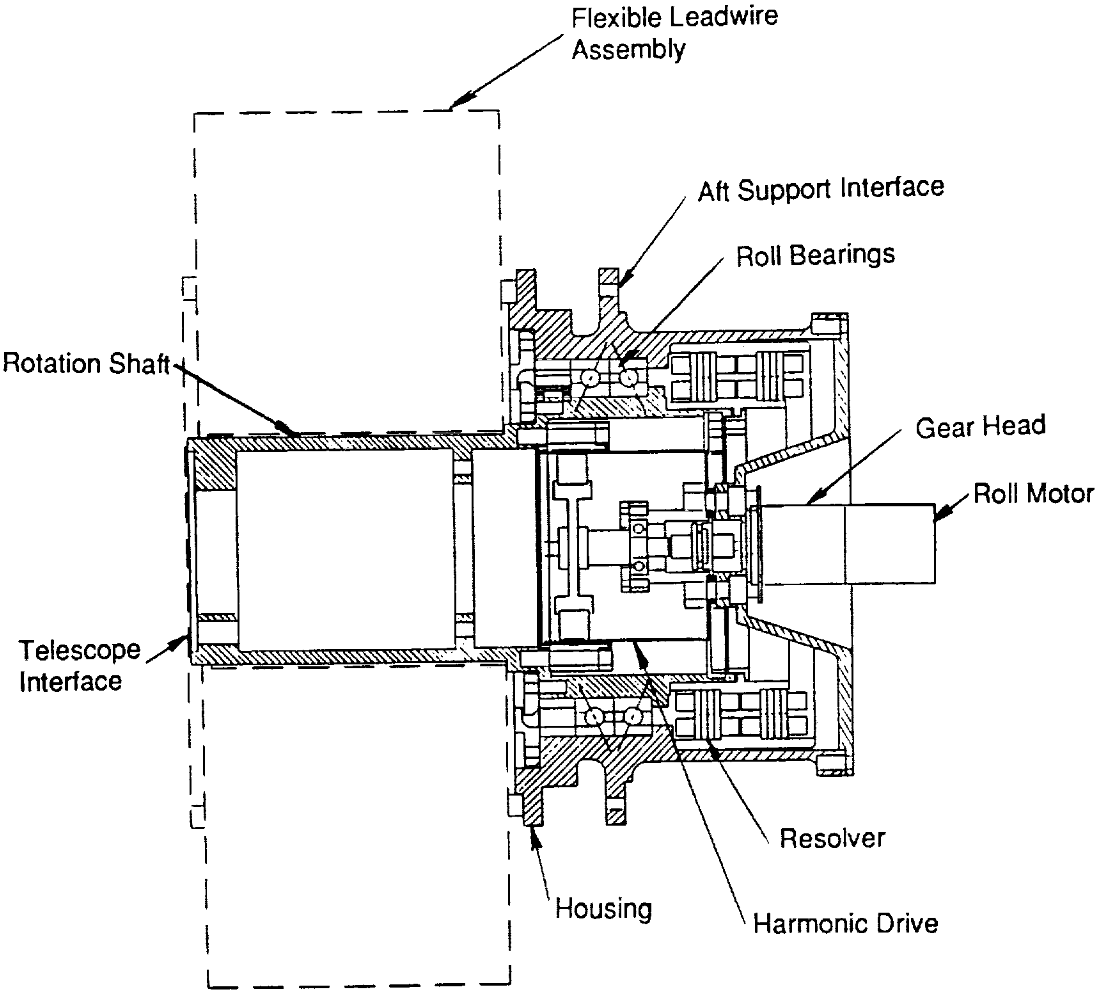


Figure 7. Aft roll drive.

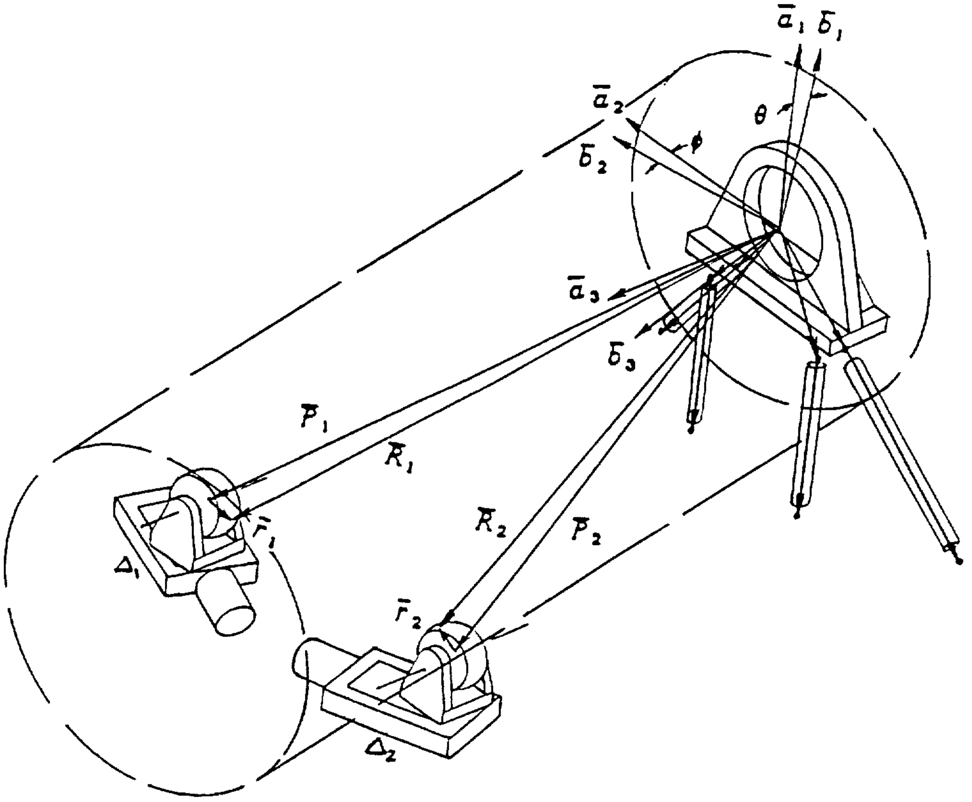
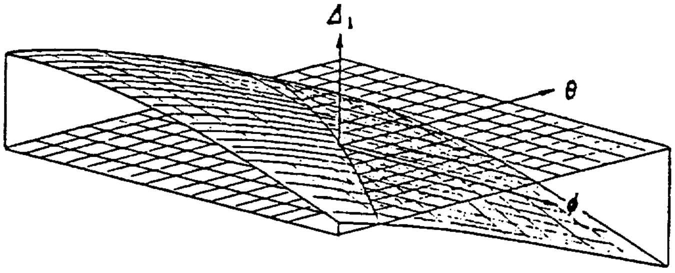


Figure 8. Vector description of the pointing geometry.



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\begin{array}{rrl}
\Delta_1 = & 1.777360\text{ e }{-04} & \\
 & -2.709935\text{ e }{+01} & \theta \\
 & -2.567364\text{ e }{+00} & \theta^2 \\
 & -3.682123\text{ e }{-01} & \theta^3 \\
 & -6.672202\text{ e }{-02} & \theta^4 \\
 & -1.898960\text{ e }{+01} & \phi \\
 & -6.390665\text{ e }{-02} & \phi\theta \\
 & -1.472490\text{ e }{-02} & \phi\theta^2 \\
 & -3.844192\text{ e }{-03} & \phi\theta^3 \\
 & -8.912779\text{ e }{-04} & \phi\theta^4 \\
 & -2.651744\text{ e }{-02} & \phi^2 \\
 & -4.691100\text{ e }{-03} & \phi^2\theta \\
 & +6.769133\text{ e }{-03} & \phi^2\theta^2 \\
 & -1.382030\text{ e }{-04} & \phi^2\theta^3 \\
 & -2.629500\text{ e }{-03} & \phi^2\theta^4 \\
 & -6.892810\text{ e }{-04} & \phi^3 \\
 & -4.318384\text{ e }{-05} & \phi^3\theta \\
 & -1.720272\text{ e }{-05} & \phi^3\theta^2 \\
 & -2.555102\text{ e }{-06} & \phi^3\theta^3 \\
 & -5.113322\text{ e }{-07} & \phi^3\theta^4 \\
 & +1.400646\text{ e }{-03} & \phi^4 \\
 & -4.843214\text{ e }{-07} & \phi^4\theta \\
 & -2.594668\text{ e }{-03} & \phi^4\theta^2 \\
 & +5.462916\text{ e }{-10} & \phi^4\theta^3 \\
 & +9.144991\text{ e }{-04} & \phi^4\theta^4
\end{array}
$$

Figure 9. Left pointing surface and corresponding pointing algorithm in the form of a fourth degree polynomial.