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DRIVING AND LATCHING OF THE STARLAB POINTING MIRROR DOORS

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ABSTRACT

The Starlab Experiment, a major SDIO technology initiative, is an attached payload which will be delivered into Earth orbit aboard NASA's Space Shuttle in 1991. Starlab will generate and aim an eighty centimeter diameter laser beam into space through a large opening in the structure which houses the pointing mirror. Two doors, each somewhat larger than a desktop, cover the opening when the laser/optics system is non-operational. Latch Mechanism Assemblies hold the doors shut during liftoff/ascent and, again, during Orbiter reentry. Each door is powered by a Door Drive System during the many open/close cycles between various experiments. The design, testing and resultant failure modes of these mechanisms are the focus of this paper.

INTRODUCTION

Overall design and management of the Starlab Program is the responsibility of the Astronautics Division of Lockheed Missiles & Space Company, Inc. (LMSC). The U. S. Air Force's Space Systems Division (AFSSD) oversees all aspects of the program for the Strategic Defense Initiative Organization (SDIO). The Starlab Experiment physical equipment resides in two main entities: (1) the Module, a pressurized crew module similar in size and construction to Spacelab and mounted forward in the Orbiter cargo bay; (2) the Pallet, an unpressurized segment mounted on a standard ESA pallet and attached to the aft end of the Module. The Module's forward end is interconnected to the Orbiter cabin by a pressurized personnel access tunnel. Figure 1 shows the physical arrangement of major equipments.

The marker laser beam originates in the Module (which is manned during flight by astronauts) and is directed aft, through the main telescope mounted to the vertical optical bench of the Pallet (Figure 2). The beam is expanded to an 80 cm. diameter as it exits the telescope and is then aimed outside the Orbiter toward targets either in space or the Earth's atmosphere. The beam director, sometimes called the pointing flat, is a gimballed mirror which measures 1.47 meters across and is mounted to the horizontal optical bench in the extreme aft end of the Starlab Pallet. The pointing flat is shown in Figure 3.

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The Aft Protective Enclosure Assembly, simply referred to as the aft cover, is a large (8.63 ft. long, 7.26 ft. wide, 4.19 ft. high) aluminum honeycomb structure which completely encloses the Pallet's horizontal and vertical optical benches. The flight article is shown in Figure 4. Its sole function is to provide thermal and contamination protection for the telescope, pointing flat and other sensitive optical equipment located on the Pallet. In order for the marker laser beam (and other smaller ancillary beams) to safely exit through the aft cover opening without vignetting, each of two large doors is unlatched and rotated 90° to its full open position. This enables Starlab to optically acquire and track a variety of rapidly moving experimental targets with doors open, yet be fully protected from solar radiation with doors closed (Figure 5).

LATCHING OF THE DOORS

System Overview

The design of the Latch Mechanism Assembly was driven by two basic system considerations: (1) the device must be inherently reliable and (2) how many devices are needed to restrain both aft cover doors? The door latching system has a unique mission: restrain both doors, holding them in the closed position during the relatively high vibration levels of liftoff, ascent, and descent. Then, after Starlab is inserted into orbit and reaches a condition of operational readiness, the doors are unlatched. The latch mechanisms allow the doors to be rotated open and closed, when needed, by the Door Drive System. Reliability is the ability of the device to reach orbit intact, and then respond to the astronaut's "unlatch" (or "latch") command. Although both of these functions are important, the technical success of the mission is more dependent upon the unlatch function than it is on the relatch function. The reason for this is that the aft pallet cavity, with its key optical components, must be physically exposed to space and sky in order for the experiment to proceed.

One of the most intensely debated design issues was whether to have four door latch assemblies or two. Four latches equated to a fore and aft latch for each door. Just two latches required that one door overlap and restrain the other at the apex. This cap door would be the one which would be latched. The final decision was to choose the two-latch approach using the port door as the "capper". The main reason for the decision was to reduce the quantity of mechanism and moving parts to an absolute minimum, thereby maximizing system reliability.

Design Requirements and Alternatives

The Latch Mechanism Assembly detail design is the result of conformance to the following requirements:

1. Military Specification MIL-A-83577B (USAF) 01 Feb 1988. Assemblies, Moving Mechanical, for Space and Launch Vehicles, General Specification For.
2. Main actuating devices shall be space-proven hardware.
3. Main actuating devices shall be dual redundant, i.e., one device shall serve as the primary actuator while the other device shall serve as the secondary or backup actuator.
4. Assembly shall have long narrow footprint due to the scarcity of aft cover surface area on which to mount it.
5. Mechanism shall be non-backdriveable with electrical power off. When mechanism is locked with power off it will remain locked during worst case vibration/shock environment. When open it will remain open.
6. Mechanism shall be fully functional in a hard vacuum at +154°F and at -24°F. It shall be capable of withstanding 7.4 g(rms) between 20 Hz and 2000 Hz for 1 minute on each of major axes at lab ambient temperature and pressure.
7. The door pin holding (main load path) components such as the hook, pivot pin, bushings and baseplate shall be able to handle maximum door pin loads during the high acceleration stages (liftoff, ascent and insertion) without undergoing plastic deformation.
8. The design shall be kept "spartan" in its simplicity through the elimination of any extraneous part or feature.

The WHY of Our Final Design Decisions

Details of the Latch Mechanism Assembly final design are illustrated in Figure 6. The comparisons below indicate some of the major design alternatives and demonstrate that adherence to the foregoing list of design requirements led us to our final design decisions.

- Linear Actuators versus Linear Solenoids. The actuator had been previously flown and was space-qualified. We were unable to find a solenoid with the correct force-displacement characteristic which was certified for flight. (Design Requirement #2). Since the linear actuator uses the jackscrew principle it is non-backdriveable. The linear solenoid is readily backdriveable. (Design Requirement #5). The selected linear actuator is capable of functioning over a temperature range from -65°F to +150°F. (Design Requirement #6). The linear actuator which was finally selected is shown in Figure 7.

- Single versus Dual Actuators. There was no debate on this issue because of Design Requirement #3. The use of redundant key components is a widely accepted method for substantially increasing aerospace mechanism reliability.

- Serial versus Parallel Interconnection. Again, the Design Requirement, #4 in this case, made clear the need for latch assembly geometry to conform to the long slim aspect ratio. Parallel ganging of linear actuators would have resulted in an overall assembly width which exceeded the available mounting area.

- Single Center Link versus Four Bar Linkage. The initial design of the connection between the tail end of the primary actuator and the front ball joint of the secondary actuator employed a four-bar linkage. Kinematic studies of an alternative single-link design indicated that, in the event of primary actuator failure, hook rotational displacement and the movements of both actuator bodies caused by secondary actuator ram retraction, were all acceptable. In keeping with Design Requirement #8 (Simplicity), the single center link was chosen over the four-bar linkage.

- In-line Pivots versus Random Pivot location. Figure 6 shows that the following centers of rotation (CR) are in-line: hook CR, center link CR and secondary actuator CR. The benefits of this CR alignment are a lower profile baseplate with minimum lip height, and ease of machining the six pivot holes associated with these CRs. (Design Requirement 8).

- Flag Feature.
  Examination of Figures 6 and/or 8 show an appendage on the head of the hook referred to as the flag. This feature serves as a redundant visual indicator of latch mechanism hook status (open vs. closed). Local TV cameras mounted aft in the Orbiter cargo bay can be trained on either Latch Mechanism Assembly in order to visually verify hook status in lieu of depending only on LED lamps at the control panel. The flag is painted a glossy reflective yellow to aid identification.

Test Program and Results

Acceptance testing of the three Latch Mechanism Assemblies consisted of a six-step sequence.

1. Initial Measurements and Adjustments.
    a. Insulation resistance measurement of all wiring using a megohmeter.
    b. Electrical bonding check by measurement of ground path resistance.
    c. Adjustment of hook and center link positions by setting of extend microswitches on each linear actuator.

2. Functional Test at Lab Ambient Conditions.
    a. Retract and extend primary actuator without simulated door pin. Monitor voltage and current.
    b. Retract and extend secondary actuator. Monitor voltage and current.
    c. Repeat above procedure with simulated door pin. Figure 8 shows the latch assembly functional test setup.

3. Thermal/Vacuum Cycling
    a. Test chamber is evacuated to a hard vacuum.
    b. Temperature is cycled between -24°F and +154°F 8 times with a 2 hour soak at each extreme.
    c. Each linear actuator performs one retract/extend cycle after soaking at each temperature limit.

4. Functional Test at Lab Ambient Conditions.
    a. Same as Step 2, above.

5. Random Vibration.
    a. 7.4 g(rms) over a frequency range of 20 to 2000 Hz for one minute on each of 3 major axes.
    b. Each linear actuator performs one retract/extend cycle after vibration on each axis.

6. Final Functional Test at Voltage Limits.
    a. Same as Step 2, above, with nominal voltage (28 VDC) applied.
    b. Same as Step 2, above, with maximum voltage (32 VDC) applied.
    c. Same as Step 2, above, with minimum voltage (24 VDC) applied.

Two of the three Latch Mechanism assemblies successfully passed all tests in the above test series. A third unit exhibited a retract microswitch failure during thermal/vacuum cycling at the cold temperature limit (-24°F). Microswitch failure analysis was underway but not complete at the time this paper was written. The plan is to replace the microswitch and completely repeat the six-step acceptance test sequence. Figure 9 shows the location of the failed microswitch.

DRIVING OF THE DOORS

Systems Overview

Starlab will orbit the Earth approximately every 90 minutes. During the total mission Starlab will complete approximately 112 Earth orbits with an experiment planned during each of up to 100 orbits. This plan will require 100 open/close cycles of each aft cover door. If either door were to stay in the closed position, or even partially closed, the experiment objectives would be placed in considerable jeopardy. Manual intervention by an astronaut would necessitate an extra vehicular activity (EVA). However, no EVAs are currently planned. For these reasons the reliability of the mechanisms which latch and drive the doors is a primary design goal.

In contrast to the latch mechanisms, both of which must unlatch only once, the Door Drive Systems, one to open/close the port door and one for the starboard door, MUST WORK during 100 full cycles (maximum). Couple that fact with another: the Door Drive Systems are much more complex in terms of function and parts count, thus system reliability is at greater risk. Examination of Figure 10, a Cadam layout of the Port Door Drive System, underscores the point made above regarding the complexity and parts count of the Door Drive System. The Dual Drive Actuator (DDA) is the prime mover whose output is transmitted to the single-pass spur gearset through a detent-type clutch. The spur gear is pinned to the main drive shaft which is coupled at each end to a moving hinge shaft via Oldham couplings. The Door Drive System can be perceived as being composed of a DDA/clutch assembly containing most of the super-precision components (tolerances in the ten-thousandths of an inch) and a driveline made up of larger precision parts (tolerances in the thousandths or coarser). A full section view of the DDA is shown in Figure 11. This drawing further demonstrates the complexity of the Door Drive System.

Design Requirements and Alternatives

The Door Drive System detail design is the result of conformance to the following requirements. Note that Requirements #1 through #4, below, are identical to Latch Mechanism Assembly Requirements #1, #2, #3 and #6.

1. Military Specification MIL-A-83577B (USAF) 01 Feb 1988. Often referred to as the MMA Spec (Moving Mechanical Assemblies).
2. Main actuating devices shall be space-proven hardware.
3. Main actuating devices shall be dual redundant, i.e., DDA (Dual Drive Actuator) System 1 shall serve as the primary torque transmission path while System 2 shall serve as the secondary or backup

torque path. Each DDA "System" includes its own dedicated brushless DC motor.

4. Mechanism and driveline (entire Door Drive System) shall be fully functional in a hard vacuum at +154°F and at -24°F. They shall be capable of withstanding 7.4 g(rms) between 20 Hz and 2000 Hz for 1 minute on each of 3 major axes at lab ambient temperature and pressure.

5. Total elapsed time for opening both doors from full closed to full open position shall not exceed 3.5 minutes. Total elapsed time for closing doors shall not exceed 3.5 minutes.

6. The DDA shall exhibit sufficient torque margin to drive a simulated door assembly cyclically through a full 90° arc with hinge line vertical in ground tests. Thrust loads due to door weight are sustained at the forward roller bearing of each hinge shaft assembly. Torque resistance at each of these bearings is greater during this ground test than it will be on orbit because door weight is zero on orbit.

7. Door hinge/shaft assemblies shall be of robust construction capable of carrying combined shear and torsional loads from each door assembly during the high acceleration stages (liftoff, ascent and insertion) without undergoing plastic deformation.

8. Limit switches, a pair mounted at each of the 4 hinge assemblies, shall stop rotation (Auto Mode only) at the full open and full closed positions and switch LEDs on the control panel to signal door status to the astronauts.

9. The design, though not "spartan", shall be kept as simple as possible.

The WHY of Our Final Decisions

The final design layout of the Door Drive System is shown in Figure 10. Details of the Dual Drive Actuator appear in Figure 11. Adherence to the design requirements list (above) was a major factor in arriving at the design decisions described below.

- Rotary versus Linear Door Motion for Opening and Closing. Although the first concept called for linear door motion, it later became clear that rotating the doors was the only viable approach. The only spatial clearance issue with rotating doors was: when the doors are rotated full open, do their tips clear the big closed doors of the Orbiter cargo bay? Kinematic studies done in scale on Cadam indicated several inches of clearance assuming worst case tolerances throughout. By comparison, the angled top faces of the aft cover precluded linear doors since, when sliding open, they would have collided with a variety of equipments mounted near the outside perimeter of the aft cover. A more easily debated reason for rotary instead of linear was our strong preference for a completely rotational door system allowing the use of a variety of inexpensive yet high quality off-the-shelf rotary-type bearings and the avoidance of long guide-rails, linear bearings and routing of aircraft cable most likely required in the linear motion approach. Rotary door motion more readily conformed to Design Requirement #9.

- Selection of the Dual Drive Actuator (DDA).
  Without question, the DDA is the cornerstone of the Door Drive System design. Therefore, the choosing of available dual drives was a key hardware selection decision. The field was immediately narrowed by Design Requirement #2 requiring space-proven hardware. The DDA finally selected was developed some years ago at Caltech's Jet Propulsion Laboratory and, with various design nuances, was the actuator of choice in a number of space-related programs. It has passed several full-blown Flight Qualification Test Series and had flown once, functioning successfully in the space environment. The DDA has many attractive features such as: dual redundancy (Design Requirement #3), huge torque multiplication in a small package, rugged yet lightweight and prequalified for space use on prior programs. Of course, all of these fine features do not come free. The fabrication, finishing and inspection of many intricate parts, plus the complicated assembly procedure followed by a series of environmental acceptance tests result in a very costly final assembly.

- Spur Gears versus Worm & Wheel in the Driveline.
  Trade studies of the torque/speed characteristics of various methods for linking the DDA output to the main driveline boiled down to two approaches: spur gears versus worm/wheel. Parameters which differentiate the two types of gearing are ratios, parasitic torque and backdriveability. Worm ratios are generally higher per gear pass which meant lower driveline speed and longer door cycle times. The mesh efficiency of the spur involute profile is hard to beat and results in lower torque resistance. A close watch has been kept on all parasitic torques because of limited available torque output from the DDA. Worm non-backdriveability, sometimes a very useful feature, was perceived as a disadvantage for the case of door mass and acceleration perturbations causing high stress and potential wear at the worm and wheel contact spot. This was a non-problem for the spur gears which when backdriven to torques higher than 100 lb-in., cause the detent clutch to ratchet, thus limiting tooth stress. Spur gears were selected as the best method for meeting Design Requirement #5.

- Tapered roller bearings versus Deep Groove Radials for the hinges. Although deep groove radial bearings are miserly consumers of torque, the deciding feature for the hinges was bearing thrust capacity; measured in thousands of pounds for tapered rollers and tens of pounds for radials. If a door assembly weighs 30 lb. at 1 g it will "weigh" well over 200 lb. during liftoff/ascent. And this is not a steady load but a pounding load due to superimposed random shock input. High Hertzian contact stress in radial bearings can result in brinelling (raceway indentation) because the ball to race contact area is essentially a point contact. Tapered rollers make line contact with their raceways thus reducing contact stress. The price to be paid for the superior thrust capacity of tapered roller bearings is their inherently greater torque resistance which, for this application, is about 2 lb-in. per bearing. With 4 roller bearings per driveline, their total torque resistance will be approximately 8 lb-in., a figure which is high enough to merit attention but low enough to be acceptable. These bearings will perform in accordance with Design Requirement #6.

- Independent Hinge Shafts versus A Single Driveline Shaft.
  The initial concept for shafting to drive the doors was a single .500 inch diameter stainless steel shaft driven near the center of its length by the spur gear and pinned to a moving hinge at each end. The major flaw in this concept was differential thermal expansion/contraction between the aluminum honeycomb structure to which the fixed hinges are attached and the one-piece stainless steel shaft. Differential expansion had the potential to cause severe binding between the fixed and moving hinges which, in turn, could result in higher than acceptable torque resistance. The 3-shaft idea, i.e., a main drive shaft driving 2 independent hinge shafts through couplers which act like expansion joints, was selected because it disallows axial force buildup as a function of differential thermal expansion or contraction. Reference Design Requirement #7.

- Redundant Door Control Electronics.
  From the very beginning of our dialogue regarding the door control system, Systems Engineering insisted upon a redundant approach. That is, door opening and closing would be done semiautomatically, which means an "open" or "close" command would be manually initiated with the remainder of the sequence being automatic. If, for any reason, the semi-automatic system should fail, the doors could still be opened and closed by the astronauts' switching to "Manual" operation. This redundant systems approach requires the use of a pair of microswitches at each of 4 hinge assemblies, one switch for Auto mode and one for Manual mode. Figure 12 indicates the method employed for switch mounting and actuation. Note that only one switch is seen because the pair is stacked side-by-side with one hiding the other from view. This scheme is in consonance with Design Requirement #8.

Test Program Plans

Acceptance testing of the three Dual Drive Actuators will consist of a six-step sequence almost identical to that performed on the Latch Mechanism Assemblies.

1. Initial Measurements and Adjustments.
2. Functional Test at Lab Ambient Conditions.
3. Thermal/Vacuum Cycling. Test DDA at Combined Limits of Temperature and Voltage.
4. Functional Test at Lab Ambient Conditions.
5. Random Vibration.
6. Final Functional Test at Voltage Limits.

Note: The sole deviation of the DDA test sequence from that of the latch mechanisms is Step 3. The DDAs are run with full load at each high temperature plateau and each of 3 voltage levels. Also the DDAs are run with full load at each low temperature plateau and each of 3 voltage levels.

Two additional major tests are planned: (1) Driveline Torque Margin Verification wherein a simulated port door is rotated full open and full closed at the high voltage limit then at the low voltage limit. Test measurements will include times required for full opening and full closing in addition to DDA motor temperatures. Present plans call for this to be done at lab ambient temperature and pressure. Figure 13 illustrates the driveline mounted to special test equipment. (2) Full-Up Aft Cover Thermal/Vacuum Tests will be done in a large environmental test chamber. The full-up aft cover assembly (flight hardware) will be fastened to a large handling dolly and suspended, with thrust axis and hinge drivelines vertical, inside the thermal/vacuum test chamber. The chamber will be pumped down to a hard vacuum and then temperature cycled between +154°F and -24°F for at least 8 full cycles. At each temperature plateau the real flight door system will, for the first time, be exposed to combined worst case temperature and voltage in a vacuum. It is this door cycle test of the full-up aft cover assembly that will finally prove or disprove the flight readiness of the aerospace mechanisms which do the driving and latching of the Starlab pointing mirror doors.

LESSONS LEARNED AND CONCLUSIONS

Several valuable technical lessons were learned during the engineering development of these moving mechanical assemblies. This information came to us from two sources: (1) our own design, fabrication, assembly and testing efforts and (2) other projects working on similar devices, such as the DDA.

o Failed Microswitch on Starlab Latch Mechanism Assembly.
The function of this sub-subminiature microswitch, one of 3 used in each of 2 linear actuators per latch assembly, is to interrupt

current to the actuator motor when the ram reaches full retract position. The microswitch failed to function with the temperature at or near -24°F. The linear actuator is rated to perform down to -65°F which is substantially colder (41°F) than the failure temperature.

The first part of the analysis on the faulty actuator was to verify that the circuit was open across the retract microswitch by performing a continuity check. The results indicated that the circuit was open, when in fact it should have been closed due to the fact that the actuator was in the full mechanical retract position. This verified the failure encountered during the initial thermal cycle test. At this point, the actuator case was removed to investigate the reason for the failure. During removal of this case, the microswitch contacts closed.

The actual activation of the microswitch is accomplished by a cam, traveling on a linear path, pushing up on a triggering device which, in turn, compresses the microswitch plunger. It was determined that the triggering device was not adjusted properly and therefore the microswitch plunger could not be compressed a sufficient distance to cause activation at a cold temperature. The trigger was then adjusted so as to compress the plunger when the cam made contact. After the rework, the unit was tested for 20 cycles at both -24°F and +145°F with no failures. This verified that the failure was due to the trigger. This was typical of an "infant mortality" type of failure of a tiny mechanism composed of intricate precision parts. One question which arises, is this failure mode impending on the latch mechanism assemblies which have already successfully passed these same acceptance tests?

- DDA Motor Stalled at Low Temperature/Low Voltage.
  This acceptance test failure (not Starlab) points to the wisdom of thoroughly testing a flight assembly in an environment which faithfully simulates real operating conditions. In this case the motor stall occurred at -40°F and was caused by a seized universal joint. The U-joint ball was defective due to a design error allowing too large a tolerance on ball diameter. The replacement ball was a better fit with the mating part and the DDA performed well in the repeat test.

- DDA Incapable of Producing Output Torque.
  This, also, was an acceptance test failure (not Starlab) wherein the DDA output shaft was unable to drive the load even though the motor was turning. Failure analysis indicated that a major subassembly in the transmission path, the harmonic drive, was experiencing gear disengagement and tooth skipping. This was the result of incorrect hardware selection, i.e., the commercial grade (loose gear fitup) harmonic drive was selected instead of the aerospace quality unit with tighter gear fitup. The replacement harmonic drive was installed and the DDA successfully passed the repeat acceptance test.

Some Lessons Learned

1. Always ground test flight hardware at the anticipated worst case combination of conditions. For example: test at lowest temperature combined with lowest pressure combined with lowest voltage combined with longest time duration.

2. It pays to pay attention to detail. The fine design details, such as the correct mounting and actuation schemes for microswitches, are too often thought of as mundane issues, deserving of only minimal attention by the design engineer. If <u>any</u> item has the potential for crippling the mission then it warrants engineering attention.

3. Selection of off-the-shelf precision components and devices is fraught with danger. The design engineer is at risk, usually because he assumes he understands what he needs to, about the workings of the device. Complex devices are frequently full of surprises regarding their functional limitations. Contact the vendor; he is the real expert on his particular device.

4. Study the successes and failures of devices used on other projects which are similar to what you are developing. Gathering the necessary information is an activity well worth the effort. Keep in mind that design engineering is very much a business of collecting, filtering and applying good technical information.

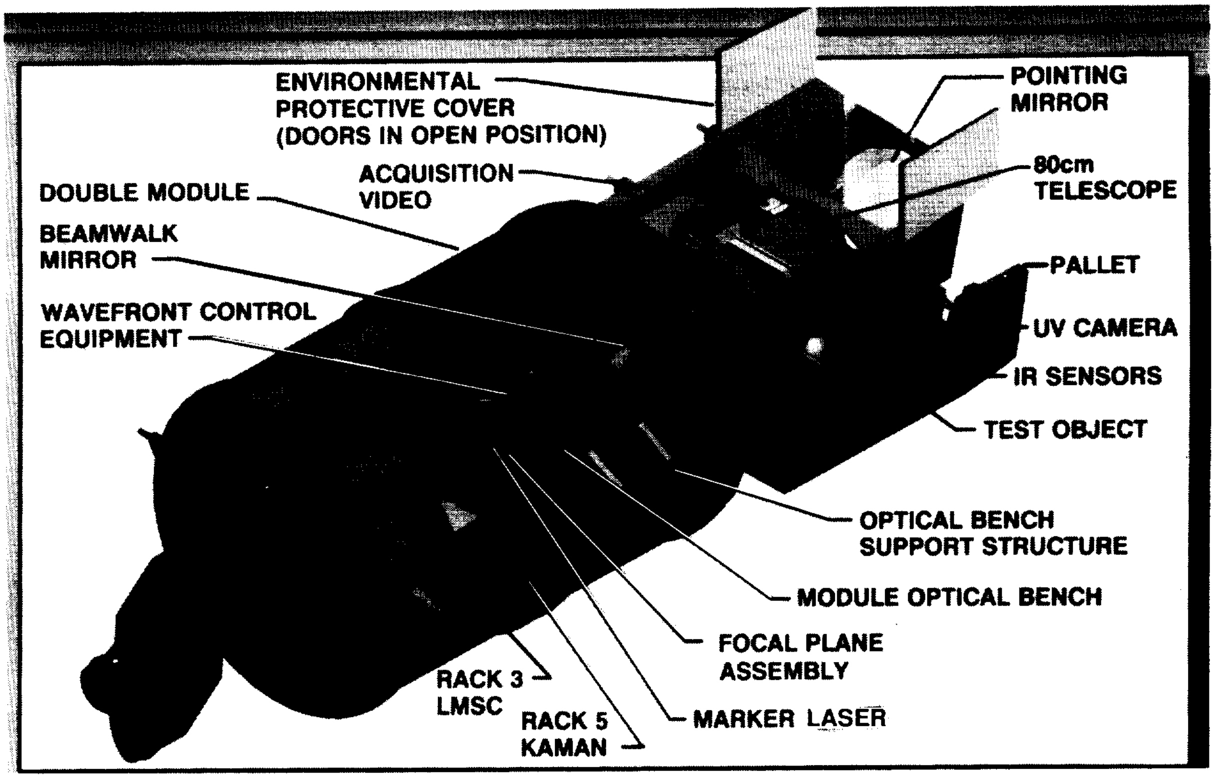

Figure 1. Starlab Experiment Hardware Layout

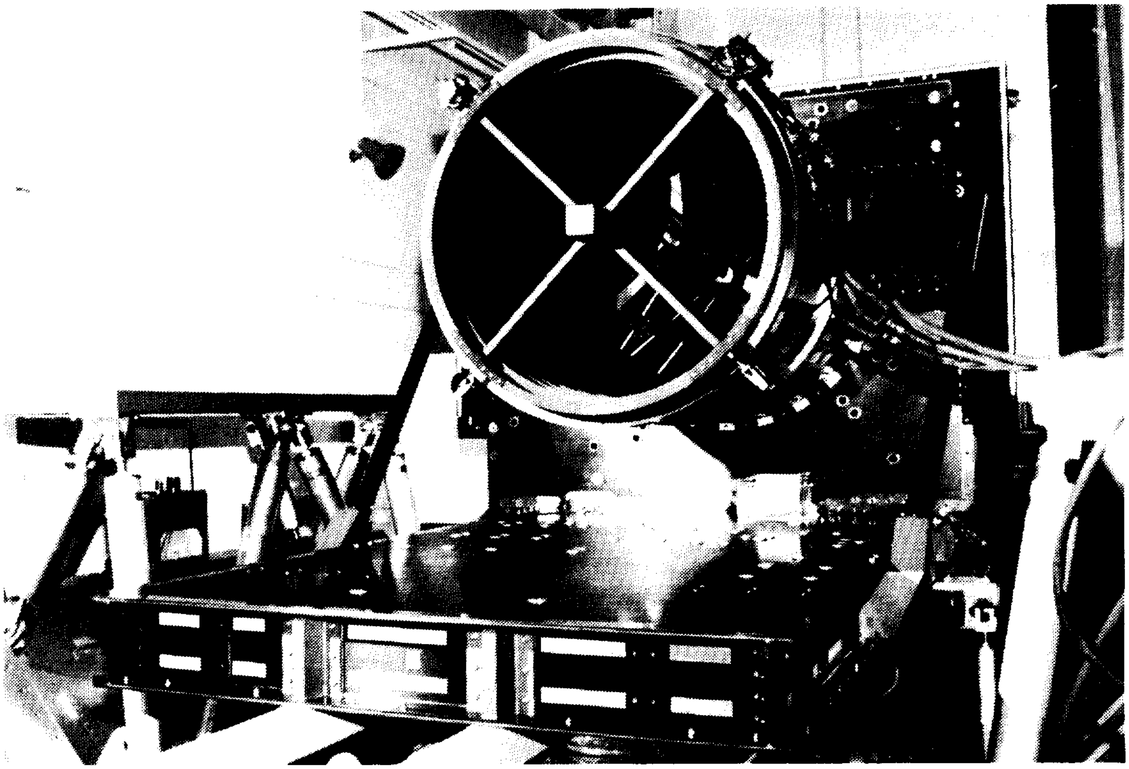

Figure 2. Aft Pallet Optical Benches and 80 cm Telescope

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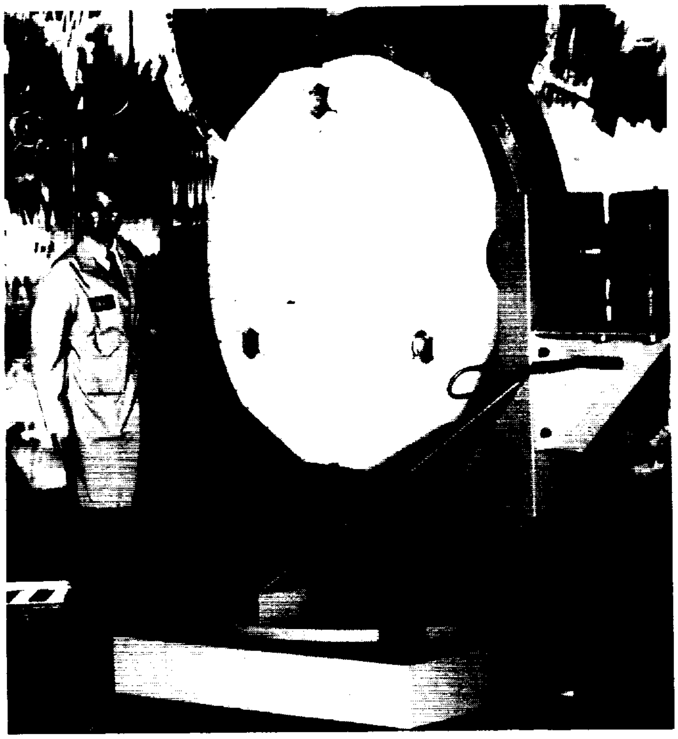

Figure 3. Pointing Mirror Mounted on Test Stand

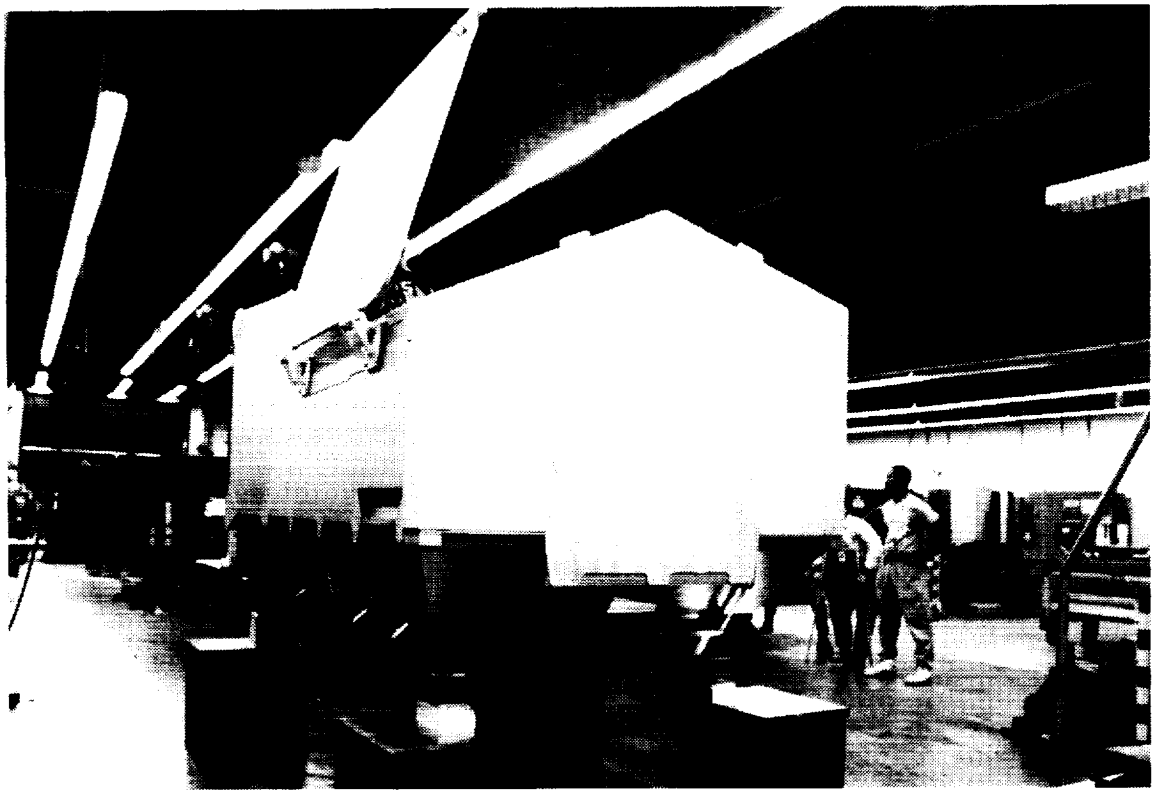

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Figure 4. Aft Protective Enclosure Assembly, Port Door Partially Open

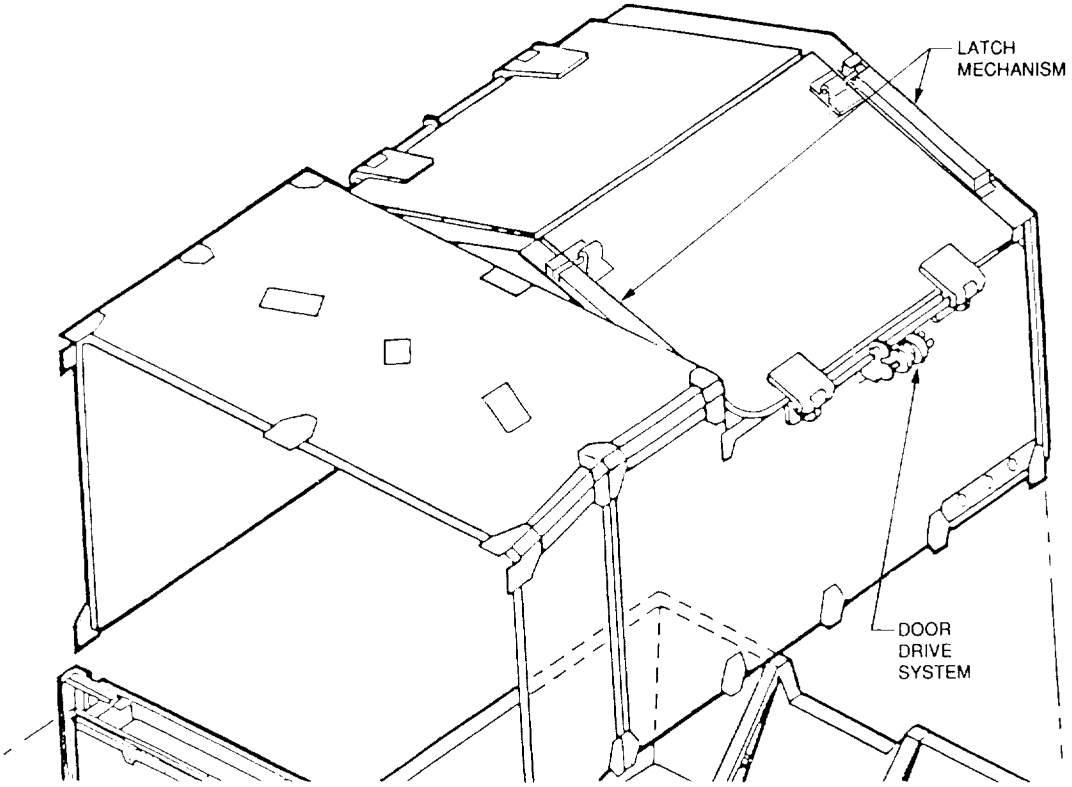

Figure 5. Aft protective Enclosure Assembly, Doors Closed

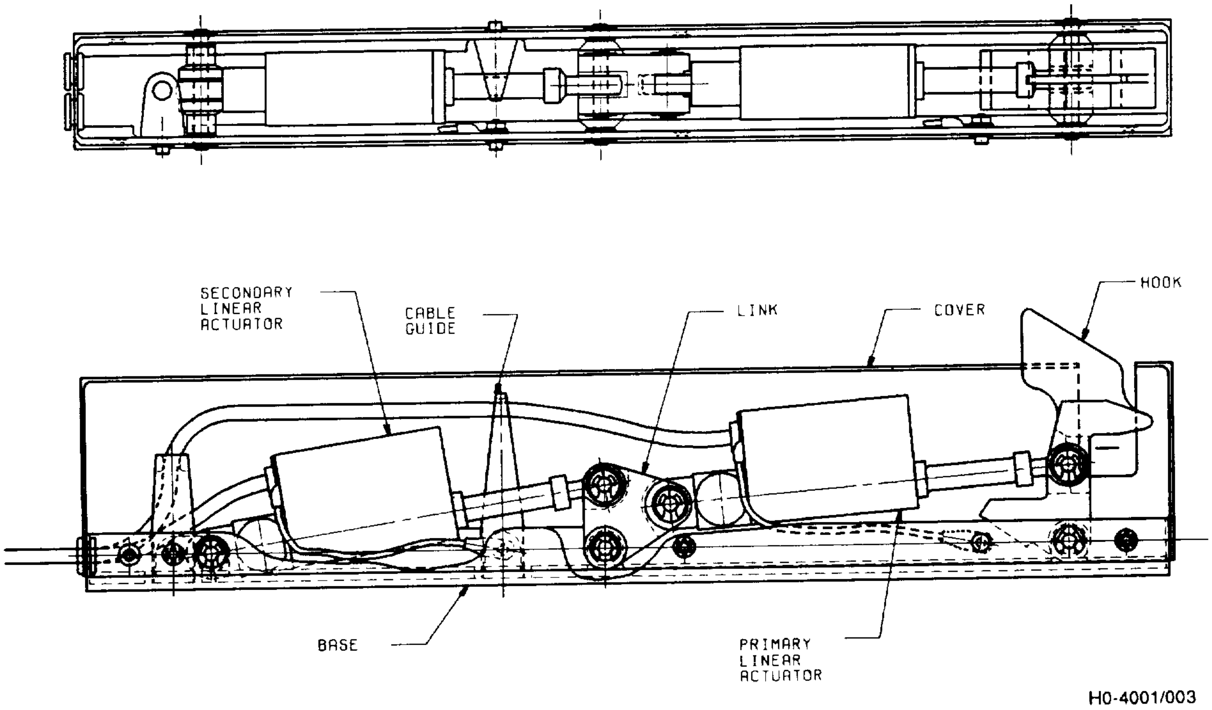

Figure 6. Latch Mechanism Assembly

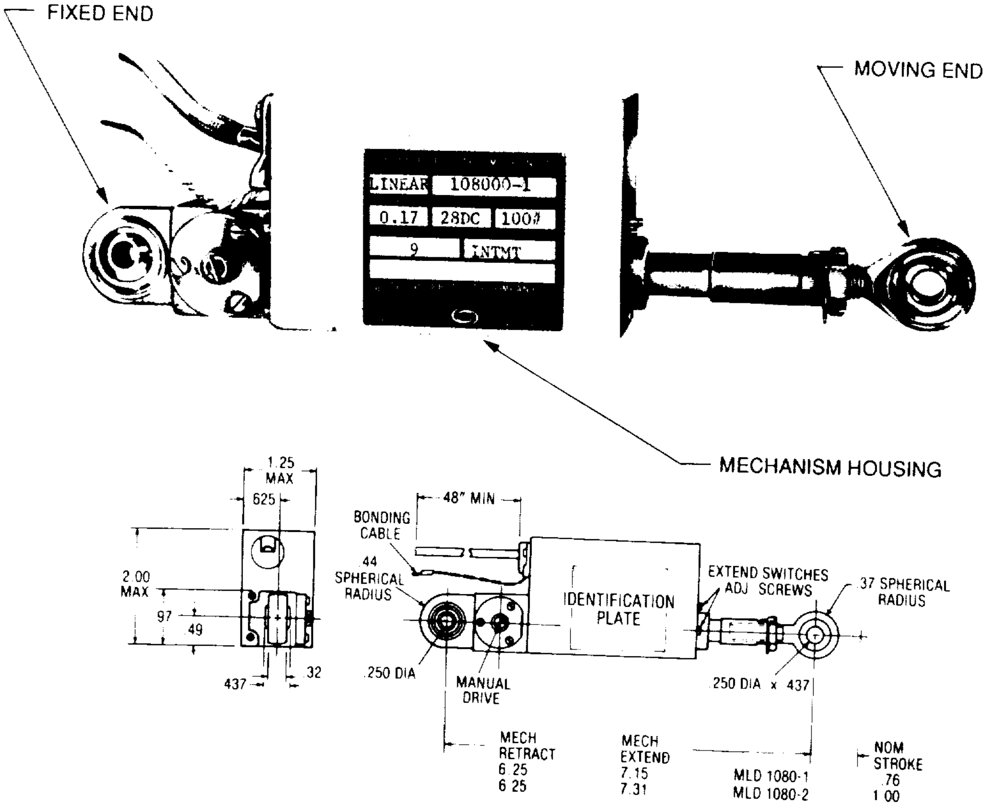

Figure 7. Linear Actuator

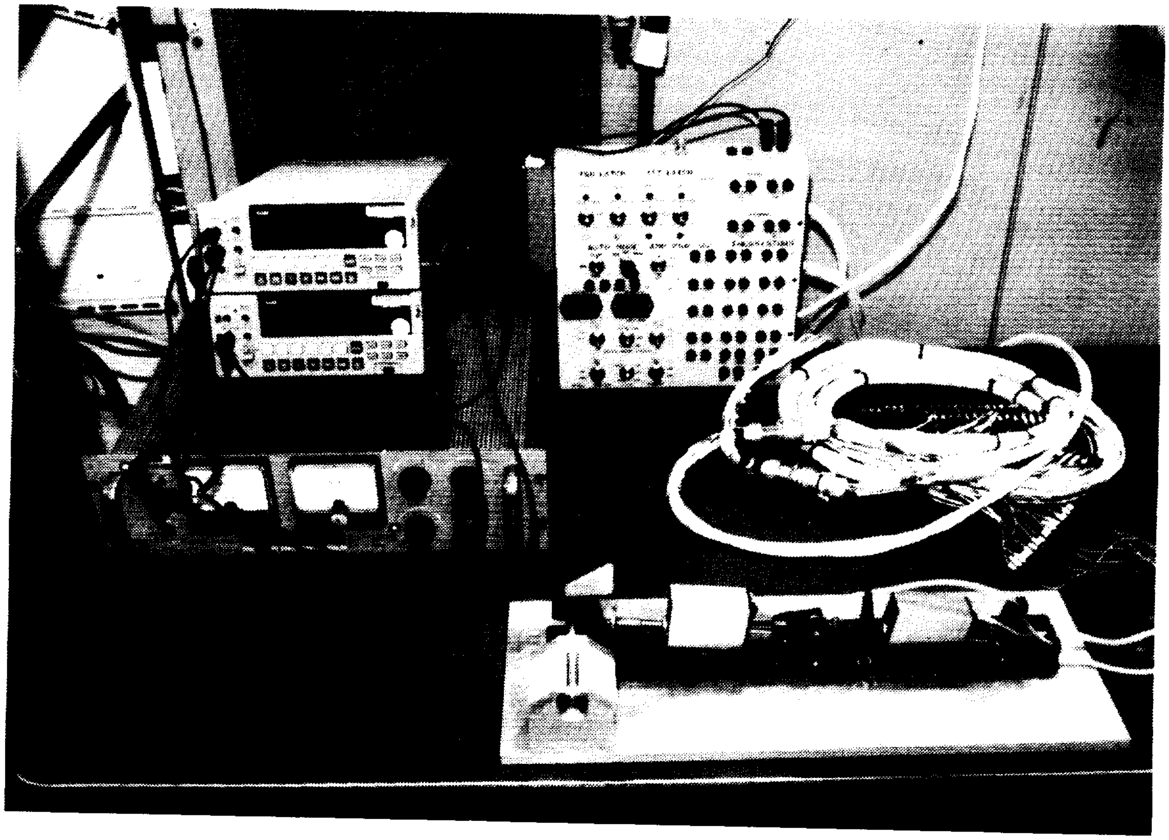

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Figure 8. Latch Mechanism Assembly Under Test

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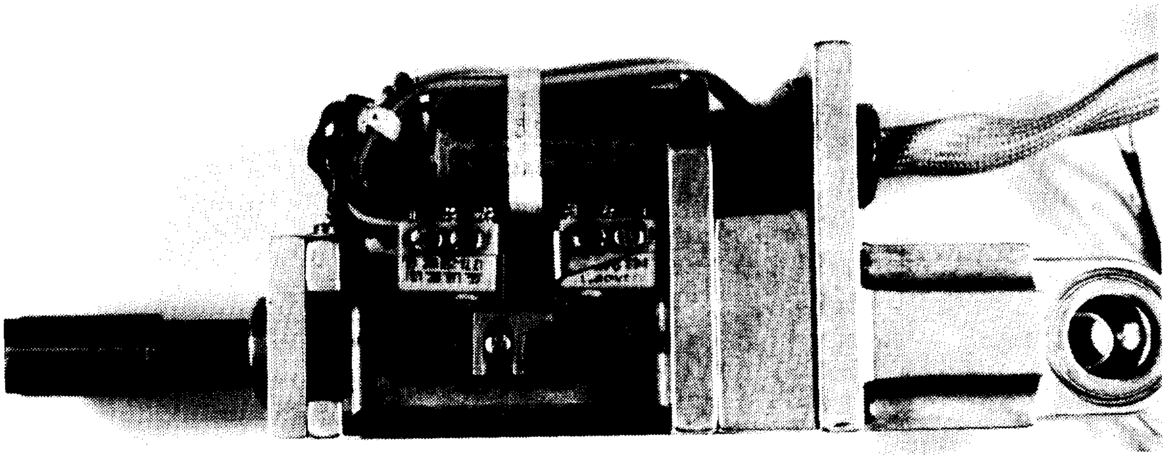

Figure 9. Linear Actuator Showing Cam and Microswitches

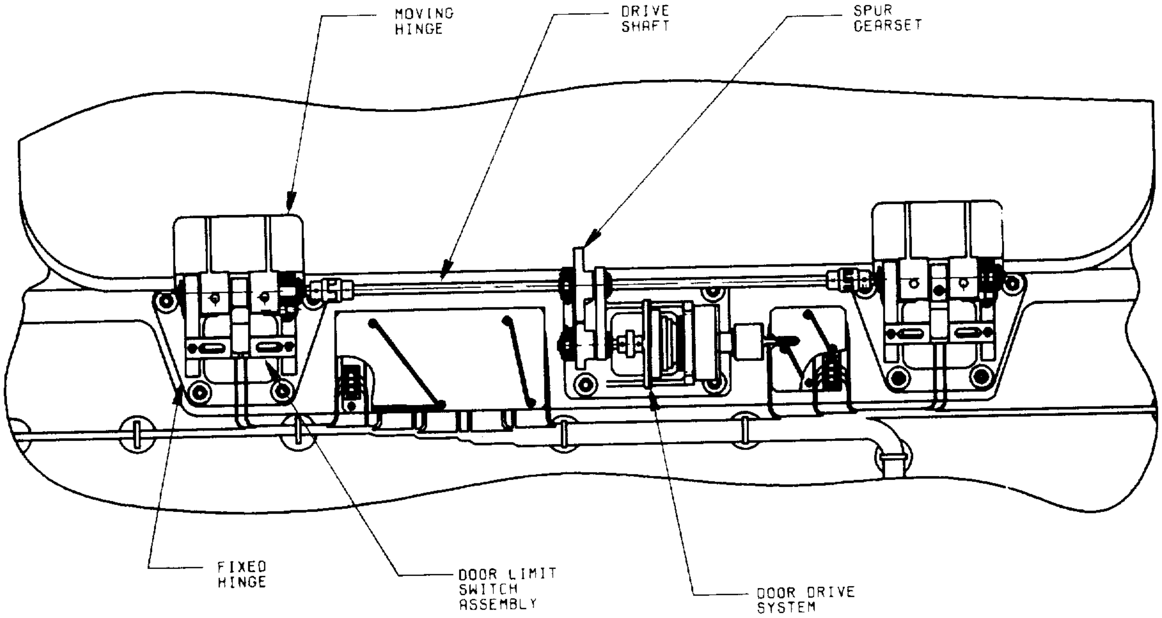

Figure 10. Door Drive System. Portside Shown

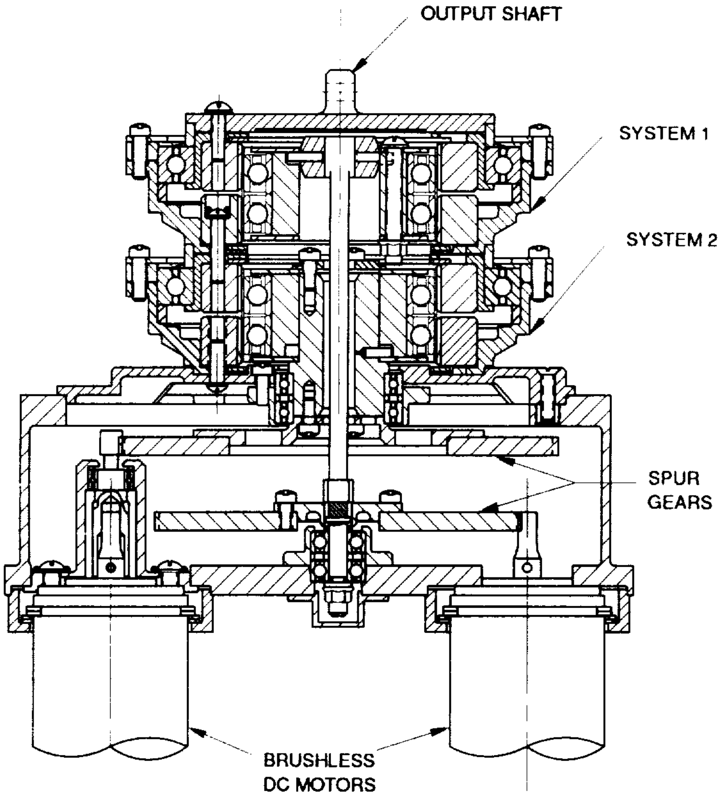

Figure 11. Dual Drive Assembly Cross-Section

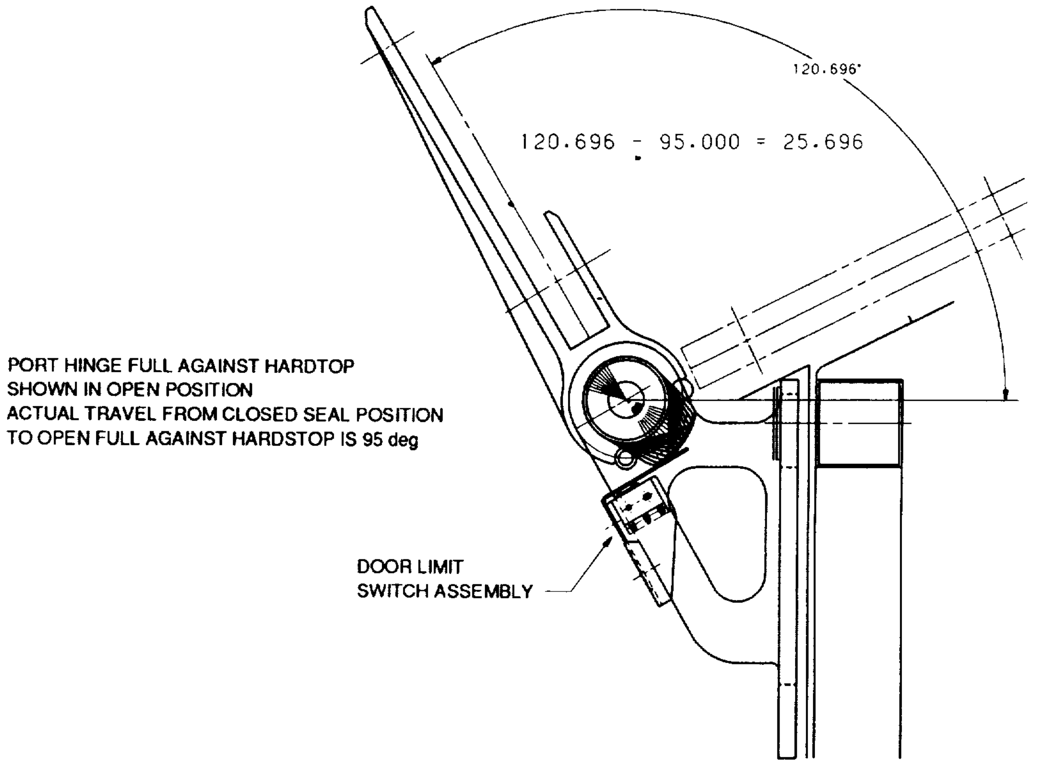

Figure 12. Door Limit Switch Arrangement

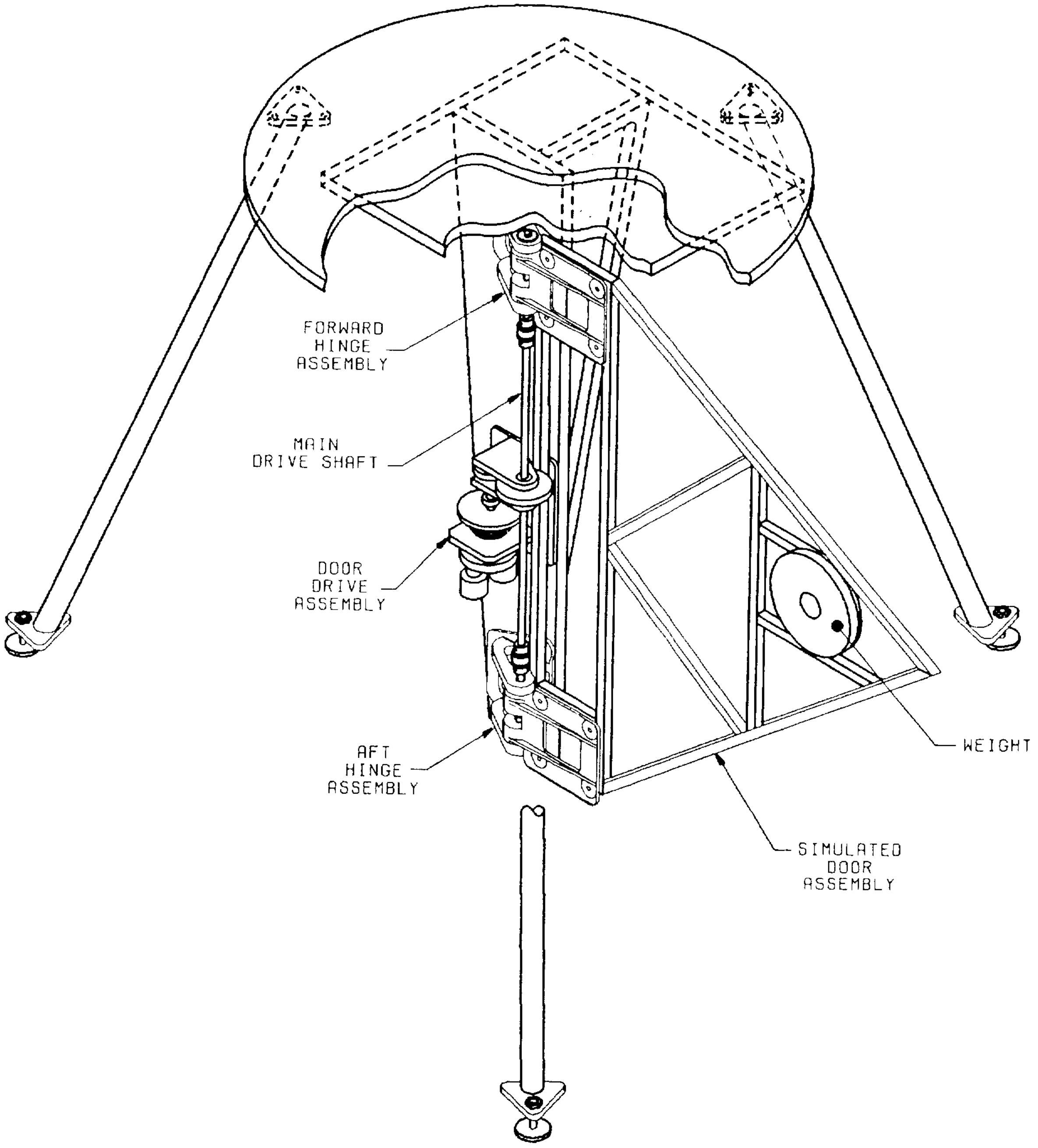

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Figure 13 . Driveline Special Test Equipment