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DESIGN AND ANALYSIS OF A KEEL LATCH FOR USE ON THE HUBBLE SPACE TELESCOPE

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This paper will be divided into two parts. The first part will deal with the mechanical design of the keel latch. The second part will be the stress analysis of the keel latch.

The first part will present (1) background information; (2) mechanical design requirements; (3) some of the initial design considerations; (4) the design considerations that led to the selection of the final design; (5) the mechanics of the final design; (6) testing that has been and will be accomplished to verify that design requirements have been met; and (7) future tests.

BACKGROUND INFORMATION

One of the initial requirements of the Hubble Space Telescope (HST), now scheduled for launch in the summer of 1986, was that it must have the capability of being maintained or refurbished on orbit. To meet this requirement, the Marshall Space Flight Center (MSFC) was directed to design, develop, fabricate, and test the necessary space support equipment (SSE) that will allow this requirement to be met. This HST SSE will be flown on a dedicated on-orbit maintenance mission.

The current planning for the HST maintenance mission is to have the orbiter approach the HST in order that the HST can be captured by the remote manipulator system (RMS). The RMS will then place the HST into a flight support system maintenance platform (FSSMP) with the HST longitudinal axis in a vertical position (see Figure 1). The FSSMP is the same hardware as used during the solar maximum repair mission. The FSSMP has the capability to rotate the HST about its longitudinal axis or to tilt it forward into a stowed position (Figure 2). All maintenance will be performed on the HST while it is in the vertical position (Figure 1). The HST will be tilted forward 57.5 degrees to its stowed position for the orbiter primary reaction control system firings or astronaut sleep cycles. When the HST is in the stowed position, it will be restrained by having its keel fitting secured by a structural keel latch.

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MECHANICAL DESIGN REQUIREMENTS

The following are the design requirements for the structural keel latch as defined in the HST On Orbit Maintenance Mission Space Support Equipment Design and Performance Requirements document.

KEEL LATCH DEVICE

The ORU carrier shall include a structural keel latch for engagement to the HST STA 240 keel trunnion, when the HST is tilted to the stowed position by the MP. Design of the keel latch shall take into account the orbiter primary reaction control system (PRCS) thruster loads when the HST is latched to the carrier structure. The keel latch shall have a capture range to accommodate an HST keel misalignment of ±2.54 cm (±1 in.) in the Y transverse direction and ±3.81 cm (±1.5 in.) in the X (longitudinal) direction. MP positioning shall correct for Z (vertical) direction alignment requirements (see Figure 1 for axis orientation). Engagement of the keel latch to the HST keel trunnion shall be provided for by a remote operation. The design of the keel latch for disconnect from the HST shall provide for a redundant remote control operation from the orbiter crew compartment. Backup disengagements shall be by means of a manual operation. The keel latch shall be designed to dampen the impact load of the HST while being tilted to the stowed position at the maximum rate without damage to the latch device or the HST.

The impact load would occur if there were a malfunction of the flight support system maintenance platform during the tilting of the HST and the HST keel fitting were allowed to impact the keel latch at maximum tilting velocity.

The keel latch also had to meet the following requirements:

Resist the following loads:

Y ± 623 kg (±1,375 lb)

Z + 812.8 kg, - 735.3 kg (+1,792 - 1.621 lb)

X no load (the HST attachment shall have freedom to move to the X direction to accommodate HST thermal growth).

The keel latch shall not apply more than 38.3 kg (85 lb) to the HST keel fitting during latching operations.

The keel latch must be two-failure tolerant for all crew safety operations. Astronaut EVA may be required after second failure.

INITIAL CONSIDERATIONS AND CONCEPTS

Figure 3 shows one of the initial concepts that was considered. It had two jaws that were driven by two individual motor drive units (MDU). As shown in Figure 3, the initial concept did not meet all design requirements.

This concept would capture the HST keel fitting, but did not have the capability to locate and capture the HST keel fitting in the required capture range. The selected design was an evolution of this concept and is shown in Figure 4. Briefly, it operates in the following manner: The HST keel fitting, with attached spool, is lowered into the capture envelope of the keel latch and, by depressing the t-bar, gives a ready-to-latch indication to the orbiter aft flight deck. The MDU for jaw 1 is turned on and jaw 1 moves toward the locked position. When its deployable bumper makes contact with the HST keel spool, the MDU for jaw 1 cuts off. The MDU for jaw 2 will then be turned on and moves toward the closed position. Jaw 2 continues to close until its stops make contact with the stops on jaw 1 (Figure 5). The spool is then secured by the jaws, but the stops maintain a clearance between the jaws and the spool. This will allow the spool to move in the X direction. To unlatch the HST keel fitting, the MDU for jaw 2 and then jaw 1 will be reversed.

DESIGN CONSIDERATIONS

The HST keel fitting is designed to provide a low friction mating surface for the orbiter keel latch used during the HST deployment mission. This made it necessary to add an appendage to the HST keel fitting to provide a positive means of resisting the required vertical loads. The shape selected for this appendage was a spool (Figure 6). This spool shape will provide a positive means of resisting the vertical loads imposed on the keel fitting. The internal sloped ends of the spool, in conjunction with the mating slope of the jaws, provide a capture envelope in the vertical direction.

The keel spool is mounted to the HST keel fitting by a spring loaded plunger (Figure 4). A mono-ball is incorporated in the spool mounting to allow for any axis misalignment between the HST and the keel latch. The Belleville spring washers will dampen any shock load as the keel fitting is placed in the keel latch.

Another design requirement of the spool was that it be visible from the aft flight deck by the astronauts, either by direct sight or by use of the orbiter closed circuit television system (CCTV). To make the spool more visible, the spool itself was gold plated.

The HST keel fitting/keel spool is placed in the keel latch capture envelope by an astronaut operating the FSSMP and observing the location of the keel fitting spool. The spool makes initial contact with the T-bar and depresses it approximately 2.54 mm (0.10 in.) (Figure 4). When the T-bar is depressed, redundant microswitches are actuated and will give a ready-to-latch indication on the aft flight deck. The switches are adjusted so that any time the ready-to-latch indication is on, the spool is in the required capture range.

During normal operating conditions, the spool impacts the T-bar with 4.5-22.5 kg (10-50 lb) force. However, a malfunction in the FSSMP could tilt the HST at full rate of 3.2 cm/sec (1.26 in./sec) that causes an impact load of 363 kg (800 lb).

When a ready-to-latch indication is obtained, the jaw 1 MDU is turned on. The MDU bevel drive gear drives a bevel gear at a 2:1 gear ratio. This driven gear has internal splines for mating with the drive screw. The two bevel gears are held in mesh by an EVA nut (Figure 4 and Figure 8). By removing the EVA nut from the driven bevel gear, the gear can be disengaged from the MDU bevel gear; and the drive screw can be rotated without having to backdrive the MDU. This is a safety feature that will allow the drive screw to be rotated in case the MDU malfunctions. The MDU has an internal brake that locks the output shaft if power is removed from the motor.

With the MDU operating, the drive screw will rotate and move jaw 1 away from the stow position. As jaw 1 begins to move, a lever system (Figure 4), will allow a bumper to be extended from the front of jaw 1 by spring force. The entire plate and bumper are moved forward (see Figure 4). Jaw 1 will continue to move until the protruding bumper makes contact with the spool. The bumper will be forced back, exerting only 3.63-4.5 kg (8-10 lb) on the spool. As the bumper is forced back, it will actuate microswitches that will give an indication jaw 1 is in the locked position on the aft flight deck. This will allow jaw 1 to locate the spool and stop whenever the spool is placed in the capture envelope.

The MDU for jaw 2 will then be turned on. Jaw 2 will travel toward the closed position until stops on jaw 2 make contact with stops on jaw 1 (Figure 5). The closing force for jaw 2 is reacted by the jaw 1 stops; no force is exerted on the spool. The MDU for jaw 2 will continue to drive until a 1,134 kg (2,500 lb) preload is applied between the mating stops. When the predetermined preload is reached, jaw 2 MDU is cut off by the power nut being moved in relation to the rest of jaw 2 (Figure 7). The power nut is held in position by four Belleville spring stacks. These spring stacks allow the movement between the power nut and jaw 2.

This preload between the jaws will capture the spool between jaw 1 and jaw 2. The stop lengths are determined at assembly to provide for 0.127 cm (0.005 in.) clearance between the jaws and the keel fitting spool. This allows the jaws to resist keel fitting spool loads in the ±Z and ±Y direction, but will allow the HST keel fitting to move in the X direction to meet the requirement of allowing for thermal expansion and contraction of the HST structure.

The spool is released from the latch by actuating the jaws in the reverse order from which they are latched. Jaw 2 is released first; and when it is no longer in contact with the spool, jaw 1 MDU can be reversed. This order of releasing the spool is necessary because of the spring load that would be applied to the spool by jaw 2 if jaw 1 were released first. When the jaws are 0.95 cm (0.375 in.) from their stowed position, they come in contact with individual preload switches (Figures 4 and 8). These switches serve two functions: (1) They will cut off the operation of the MDU's when they are operated in the reverse direction and the jaws reach their stowed position. (2) The preload switches are also spring loaded so that, when compressed by the jaws while in the stowed position, they will provide stability for the jaws during launch and landing vibration.

MANUFACTURING PROBLEMS

One of the first problems discovered during assembly operations was that the flange on the drive screw for jaw 2 was galling on the side under pressure when jaw 2 was being closed. This was attributed to the flange flexing during application of the 1,134 kg (2,500 lb) preload. The design solution was to apply only a 227 kg (500 lb) preload; measure the gap between the power nut and jaw 2; machine a spacer 0.51 mm (0.020 in.) thinner than the measured gap (Figure 7) and install it between the power nut and jaw 2. The lesser preload was enough to initially secure the spool and the installed spacer would not allow jaw 2 to move back enough to releasee the spool. Only 227 kg (500 lb) was applied to the flange while it was rotating. No galling has been detected after the spacer was added to the engineering unit.

Another problem discovered during initial assembly was the stops with ramped ends did not provide the necessary dimensional repeatability accuracy for the required gap between the jaws when the jaws were closed. The ramps were initially provided to produce a wedging action between the stops and the tracks they slide in during the latching action. After initial assembly, it was decided to use flat surfaces with a slot on one stop and a mating groove on the opposite stop. This prevents the jaws from racking during locking operations and also provides a positive indexing of the jaws in relation to the spool.

LUBRICATION

Dry lubricant was used throughout the keel latch except in the MDU and the two ball bearings supporting the MDU bevel gear. These two applications have Brayco 601 grease. External heaters have been provided to prevent freezing of the grease at low temperatures.

NPI 425 dry lubricant was used in all applications where the cure temperature of 302°C (575°F) would not damage the parent metal. NPI 14 was applied to all other sliding surfaces.

The drive screw end bushings are made from Nitronic 60 and the drive screw is made from A-286 stainless steel with NPI 425 applied to all mating surfaces. The keel latch engineering unit has been subjected to approximately 80 cycles and there have been no problems with lubricant break down.

TESTING PROGRAM

The first test the keel latch engineering unit was subjected to was a proof of concept test conducted in a 6 degrees-of-freedom (DOF) test facility. The test facility was computer controlled and powered by hydraulics. It has the capability to produce motions and forces that simulate the HST being tilted forward into the keel latch capture envelope. It also has the capability to measure forces that are induced into the HST keel fitting by the keel latch during latching operations. Figure 11 shows the keel latch installed in this facility. Equations of motion were developed that considered the mass of the HST, the flexibility of the FSSMP, and the elasticity of the keel latch support structure. These equations were input to the computer and tests were accomplished which prove, first, that an astronaut could place the keel fitting spool into the keel latch capture envelope; second, the keel latch has the required capture envelope; third, that it would not exert more than the specified loads on the HST keel fitting during a latching operation; and fourth, the keel latch could survive and operate after the specified impact load.

The ability of an astronaut to place the spool in the keel latch capture envelope using the FSSMP manual controls was verified during this series of tests. A direct visual line of sight was available for the initial testing, but later the berthing operation was accomplished by using television cameras only. It was very apparent during this testing that proper camera placement was critical for a satisfactory berthing. It was verified that the FSSMP could position the spool in the required capture envelope and the keel latch could successfully capture and retain the spool over the entire specified capture range. During this series of tests, it was proven that the keel latch could retain the spool when it was subjected to the maximum specified loads in both Y and Z axis.

An impact loads test was also accomplished in the 6 DOF test facility. As stated earlier, for normal operation the force that the spool impacts the keel latch is 4.5-22.5 kg (10-50 lb); but for a malfunctioning FSSMP, the force could be as much as 363 kg (800 lb) for an initial impact and build up to 1,225 kg (2,700 lb) if the power is not cut to the FSSMP. The keel latch sustained no damage during this test.

By observing the motion of the spool in relation to the keel latch and operating the tilt control of the FSSMP in the proper rhythm, the spool can be brought into the keel latch with very little oscillation and very little impact force. It did require some learning on the part of the operator but was easy to accomplish once learned.

During berthing tests it was discovered that the clearance between the spool and the forward part of the keel latch was inadequate. When the spool was being placed in the ready-to-latch position and positioned in the center of the keel latch capture range, the spool clearance at the forward edge (X direction) of the keel latch was 2.29 mm (0.09 in.). This was due to the geometry of the FSSMP and the HST. The spool entered the keel latch on an arc instead of a straight line. A redesign of the keel latch was required to increase this clearance to 3.18 cm (1.25 in.).

The keel latch was then subjected to a vibration test in all three axes. The unit was vibrated in both horizontal axes and operated satisfactorily during post-vibration testing. The unit was then transferred to the vertical test facility and vibrated. Jaw 1 operated satisfactorily, but the MDU for jaw 2 would not cut off.

The engineering unit was disassembled and inspected. It was found that the clearance that provides for microswitch over-travel was less than dimensionally specified. This clearance is required to insure repeatable microswitch operation. Interference between the power nut and jaw 2 prevented proper mating of the power nut with jaw 2. This condition resulted in the actual over-travel available being less than indicated. The interference problem was corrected by removal of material.

The engineering unit will next be subjected to thermal vacuum and life cycle testing which will complete the required test program.

STRESS ANALYSIS

The stress analysis of the keel latch was originally begun assuming that the structure was determinate. However, once all the load paths and reaction points were determined, it became obvious that the keel latch was an indeterminate structure.

To solve the system of loads and reactions, a finite element computer model of the latch was built (Figures 13 and 14). A detailed model which would provide stresses was decided against since the keel latch is a fairly complicated mechanism, and there were time constraints. Instead, a more simplified model which would provide reactions was used. Once the reactions were found, stresses for different parts were calcualted by tracing the load paths and using the appropriate reactions for a part and the part geometry.

The Z loads were reacted at the eight rollers that ran in the side plate slots (Figure 4). Each jaw had four rollers, two on the +X side and two on the -X side. The Y loads were reacted at the two drive screw flanges (Figure 4), one on the +Y end and one on the -Y end. Although there were no applied X loads, moments induced into the system by the Y and Z applied loads created X loading. The X loads were reacted by four recirculating rollers (Figure 4). Each jaw had two recirculating rollers, one on the +X side and one on the -X side. Figure 15 shows the reaction points on the computer model.

It can be seen by looking at Figure 15 that the jaw stops were modeled as several skewed bars instead of one straight bar. The reason there are several bars instead of one is that there has to be a grid point at each reaction point, and the bars have to begin and end at a grid point. The reason the bars are skewed instead of straight is that the reaction points don't all fall in line in the XY plane. To avoid getting inaccurate reactions due to the deflection of the skewed bars under load, the bars were made very stiff so as to act like one straight bar.

Several assumptions about the behavior of the keel latch under load had to be made in order to define the proper constraints in the model. The first assumption was that all eight rollers could react at the same time. The second assumption was that both drive screw flanges could react at the same time. The third assumption involves the recirculating rollers, and it had to be approached from a different aspect than the other two assumptions. As stated earlier, the only X loading comes from moments induced by the applied Y and Z loads. After studying the actual keel latch engineering unit, it was decided that when the Y and Z loads were applied, the jaws would form a couple and try to rotate about the Z axis. The reaction to the couple would come from one recirculating roller on one jaw and one recirculating roller on the opposite side of the other jaw. Therefore, the third assumption was that only two recirculating rollers, one on each jaw, could react at the same time and that they would always be on opposite sides of the jaws from each other. Which two would react was a function of where the external loads were applied. The fourth assumption was that the drive screws would take no bending, therefore, the shear and moment reactions at the drive screw flanges were released.

The next assumption that had to be made in the model was how to incorporate the 227 kg (500 lb) preload between the jaw stops. It was decided that in order to initiate the preload, a forced displacement of one of the drive screw flange ends would be used. By displacing one of the Y-force reaction points in the Y direction, a compression force in the Y direction between the jaw 1 stops and jaw 2 stops was achieved. A dimensionless spring element in the Y direction was placed between the end nodes of either pair of stops in order to detect the amount of compression present for a given forced displacement. The spring elements were dimensionless because the end nodes of either pair of stops were located in the same geometric location in order to represent the hard contact between the stops when the jaws are closed together (Figure 15).

The initial displacement used was a calculated guess based on the spring rate of the jaw stops and the known compression force that was needed. It took only two iterations of holding the spring rate constant, varying the forced displacement, and determining the compressive force in the spring before a 227 kg (500 lb) preload was achieved. Determining the preload case for the model was a very good check for the model because it showed that it was acting symmetrically under the preload, and the actual keel latch acted symmetrically under the preload when it was tested.

Once a method for incorporating a preload had been determined, the worst load case for each reaction point had to be determined. The preload case was considered the first load case, and the other eight load cases contained the preload and the external load. The external loads were applied in four different locations on the model (not simultaneously) (Figure 15) with a +Z and -Z load case at each location for a total of eight cases. The two points identified on each jaw represent the end points of the envelope where the spool could be located when the external forces are applied, and there is an envelope on each jaw because the Y load is fully reversible. It was found that when the external load was applied in all eight cases, the preload was completely relieved, and the worst reactions were a function of only the external load. The load case which was the worst for each reaction point varied.

Two minor design changes were made due to findings from the model and stress analysis. First, the analysis showed that the worst case roller load would fail the side plate slot upper flange. To correct this, the flange was increased to an adequate thickness. Second, the analysis showed that the bending forces on the jaw stops due to the tongue and groove mating at the ends would fail the stops at a section where the rollers went through the stops. To correct this, the jaw stop material was changed from aluminum, which has a low bending strength, to Inconel, which has a high bending strength.

All the load cases had to be run again when the keel latch was redesigned to solve the clearance problem between the spool and the forward part of the keel latch. The jaw stops were redesigned such that the envelope where the spool could be located when the external forces are applied was increased. When the new loads were determined, the appropriate changes were made in the stress analysis.

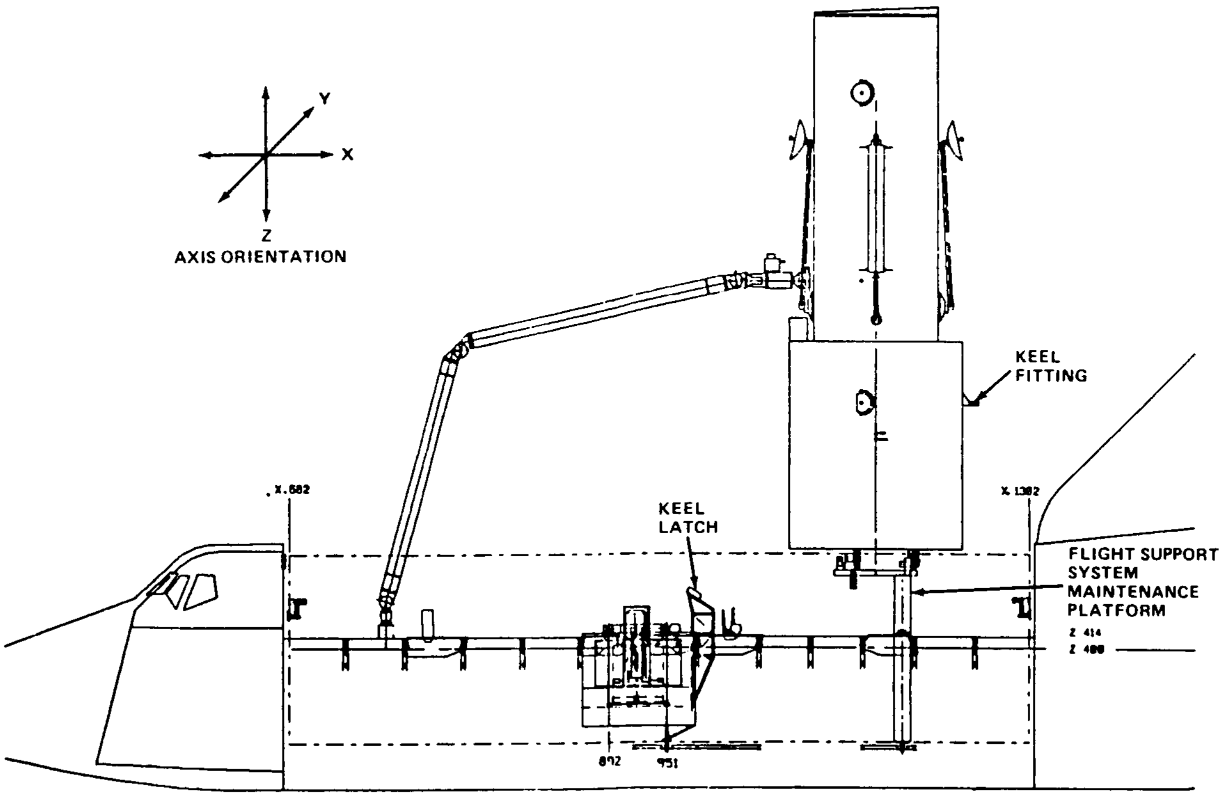

Figure 1. - Berthing configuration.

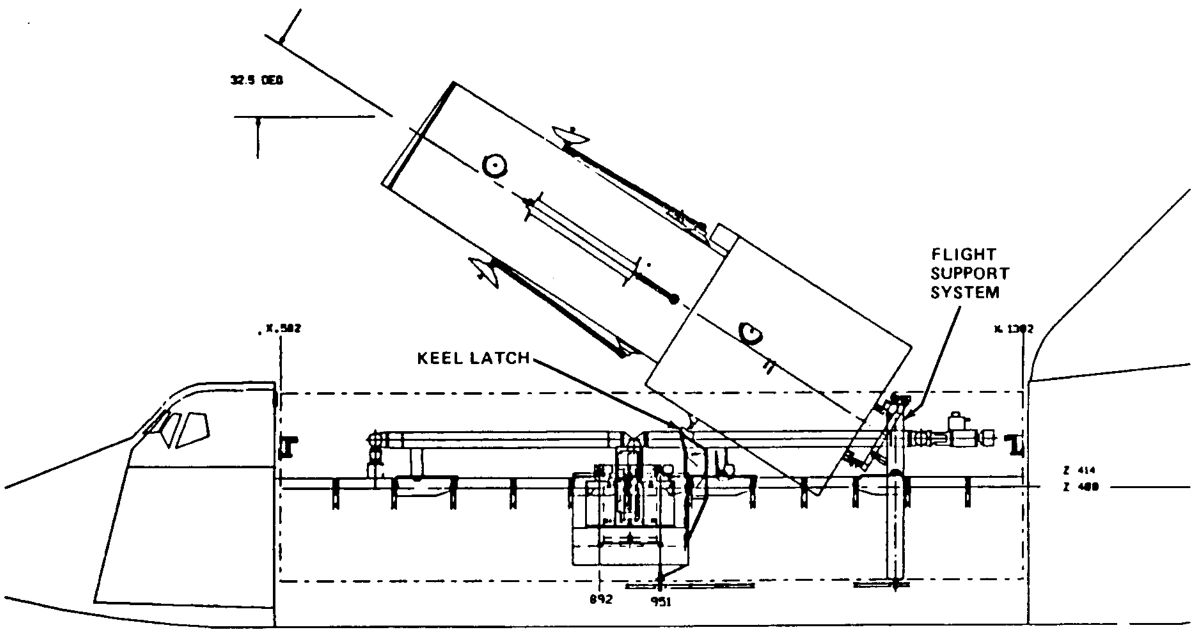

Figure 2. - Stowed/reboost configuration.

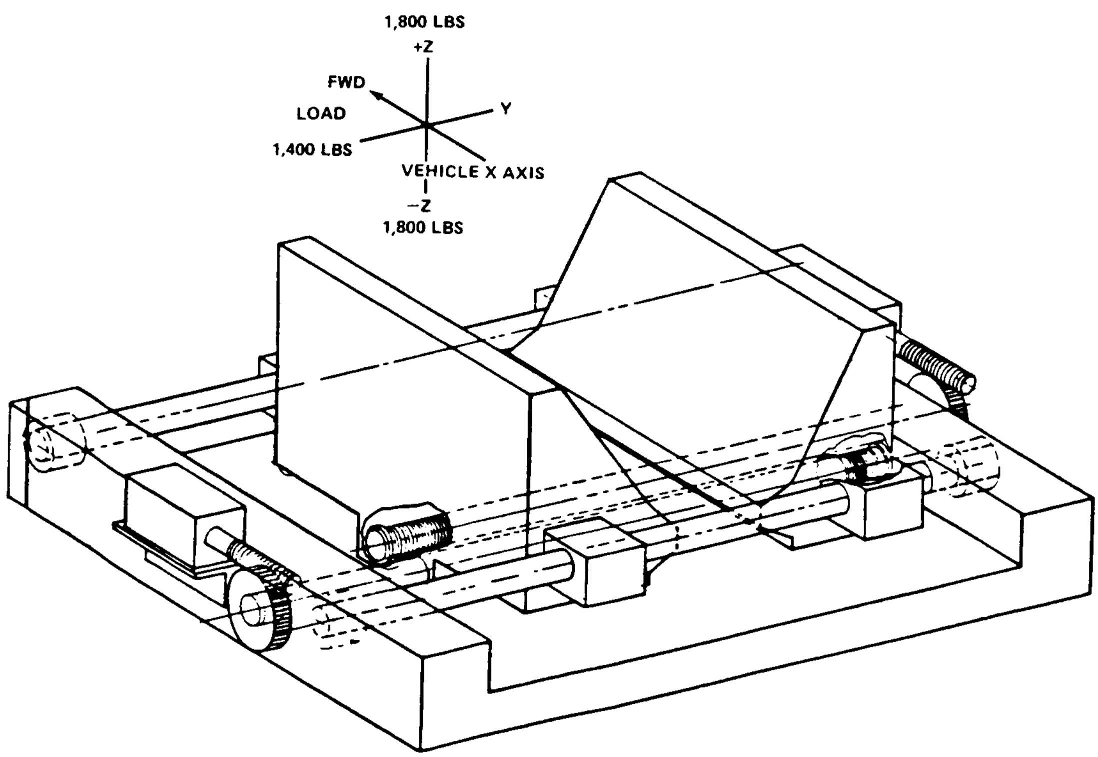

Figure 3. - S. T. keel latch concept.

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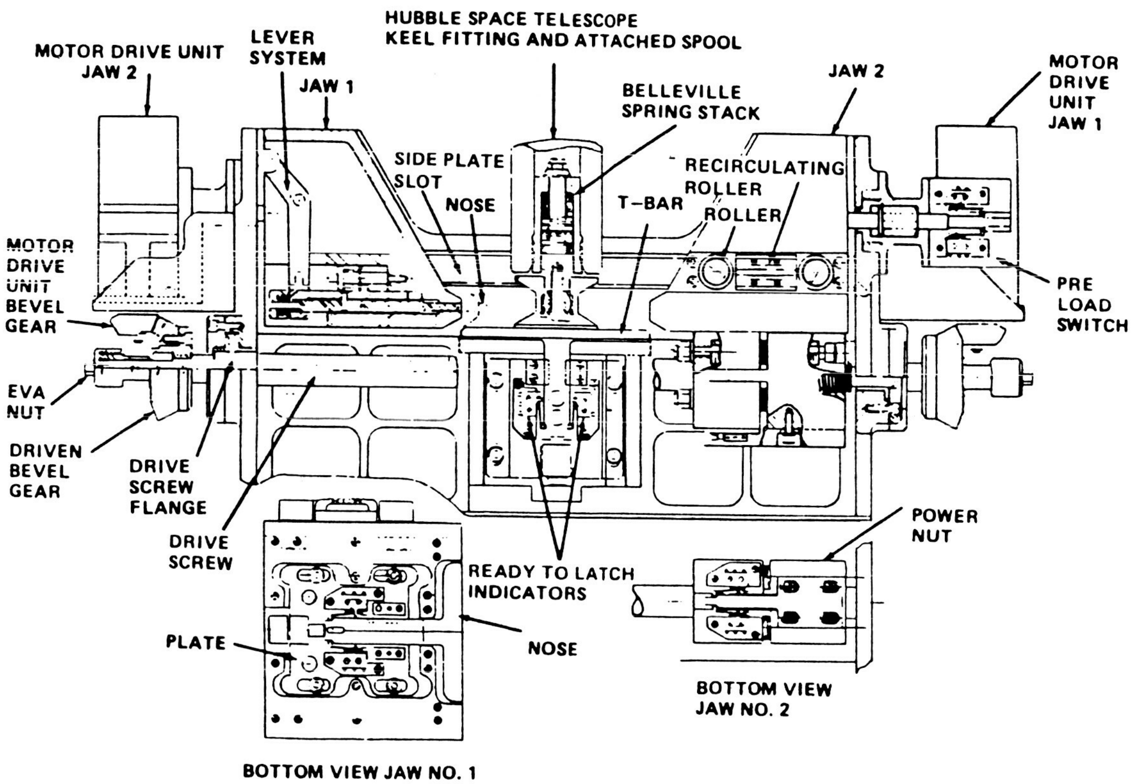

Figure 4. - Keel latch sectional views.

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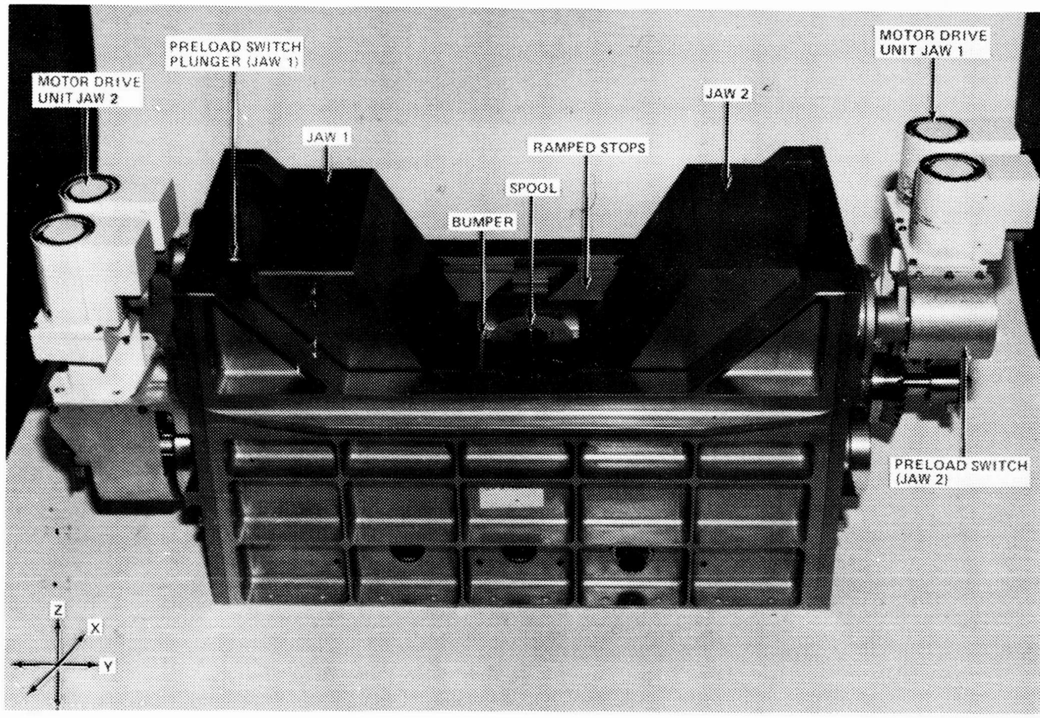

Figure 5. - Side view of keel latch.

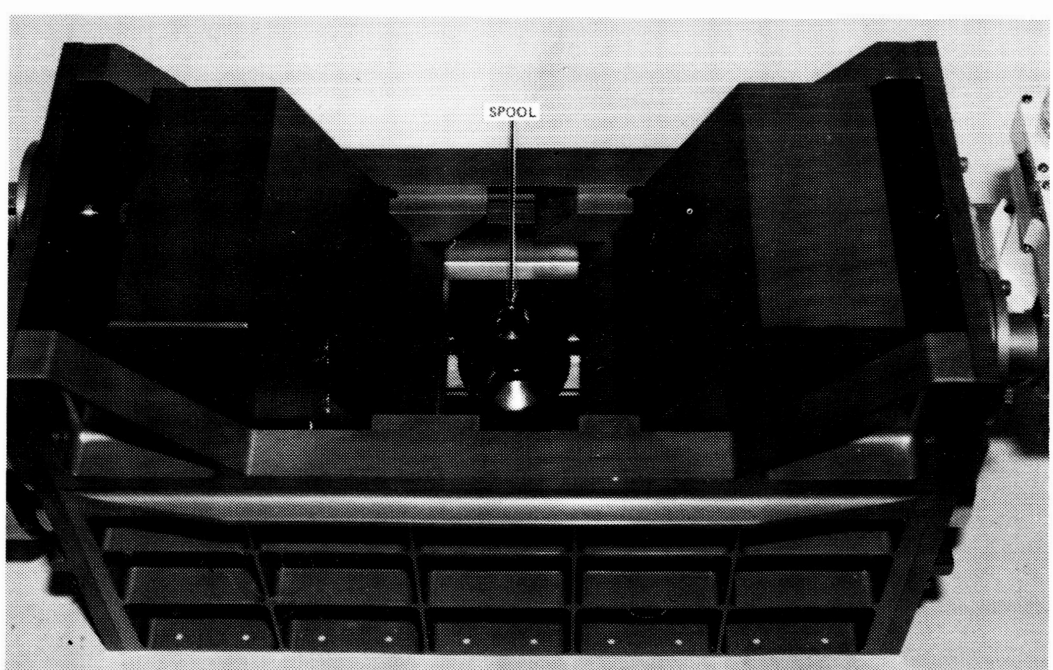

Figure 6. - Keel latch top view.

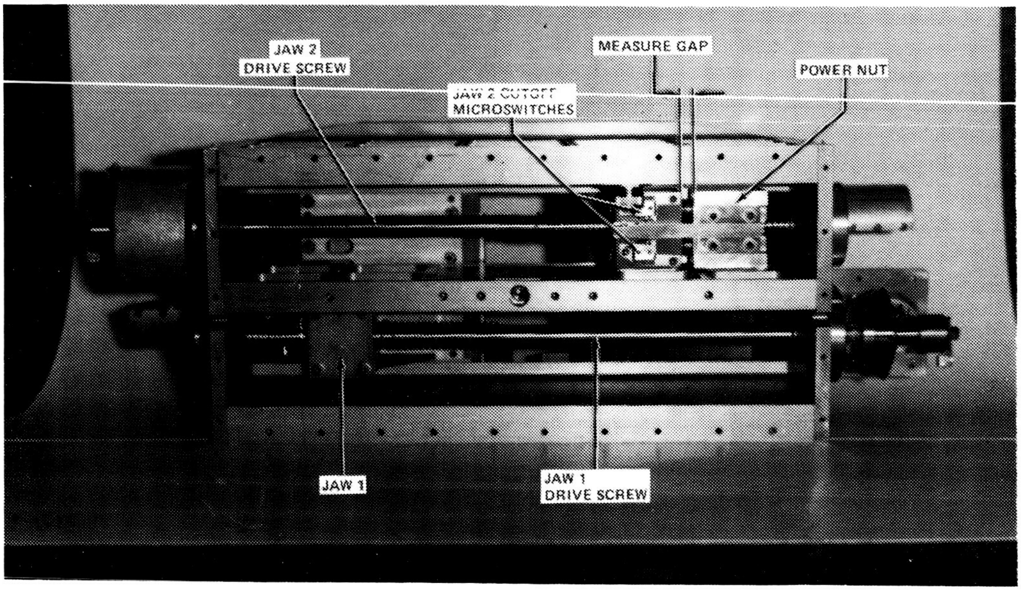

Figure 7. - Bottom view of keel latch.

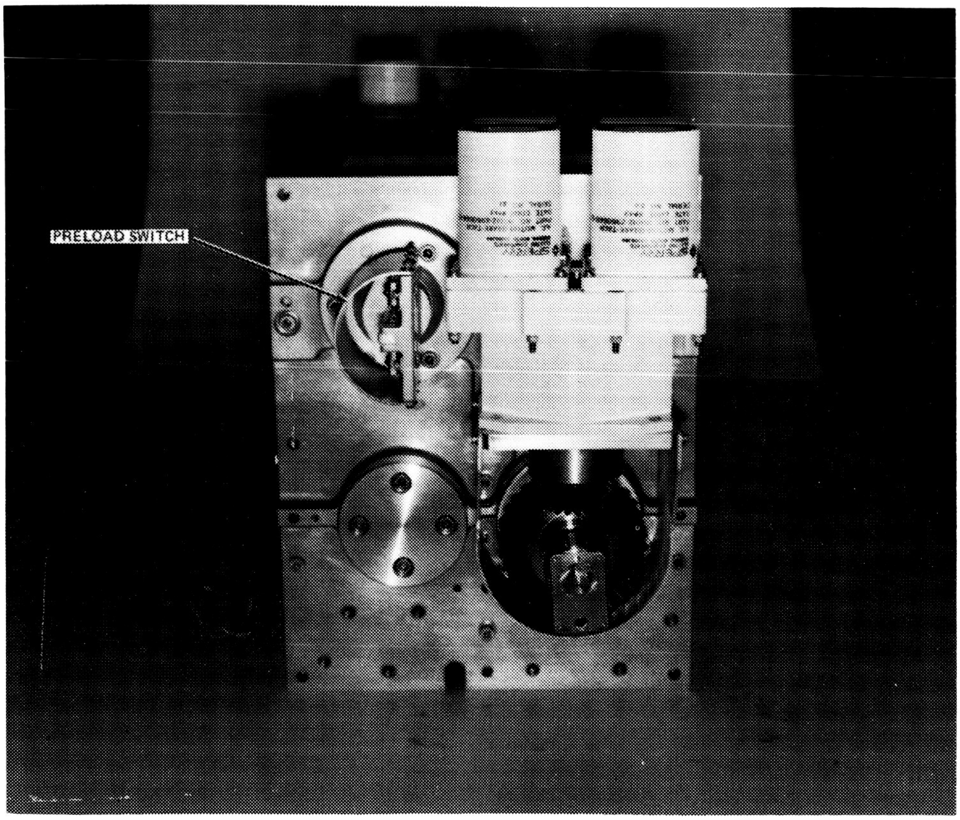

Figure 8. - Keel latch end view.

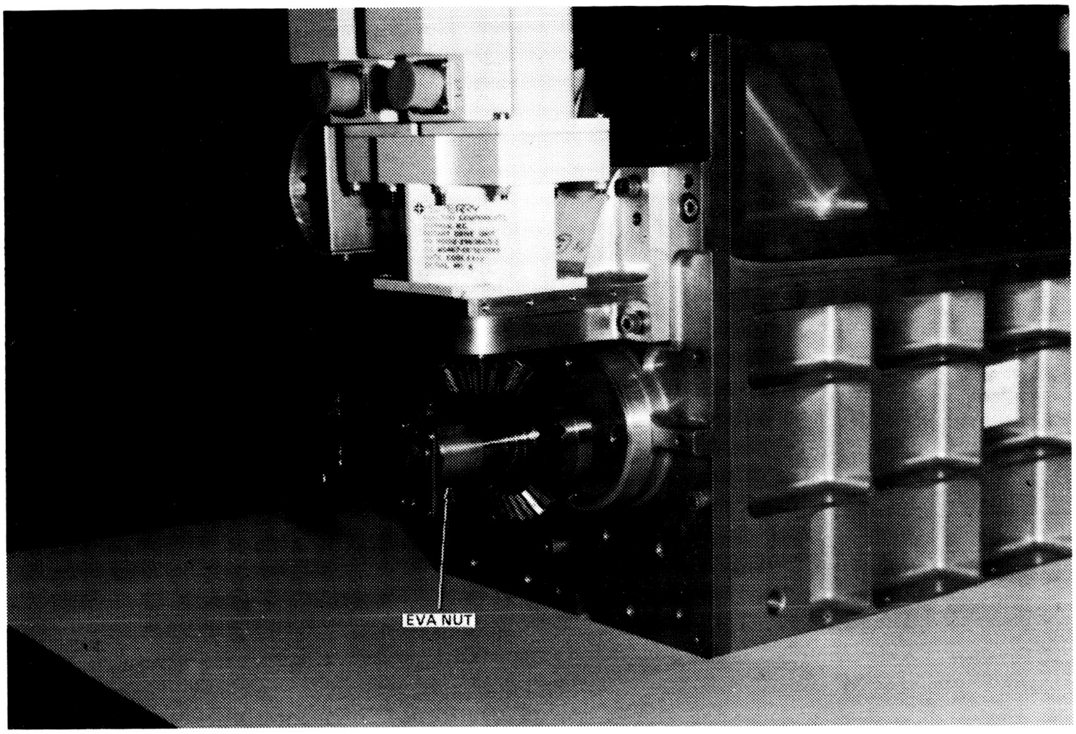

Figure 9. - Keel latch EVA nut detail.

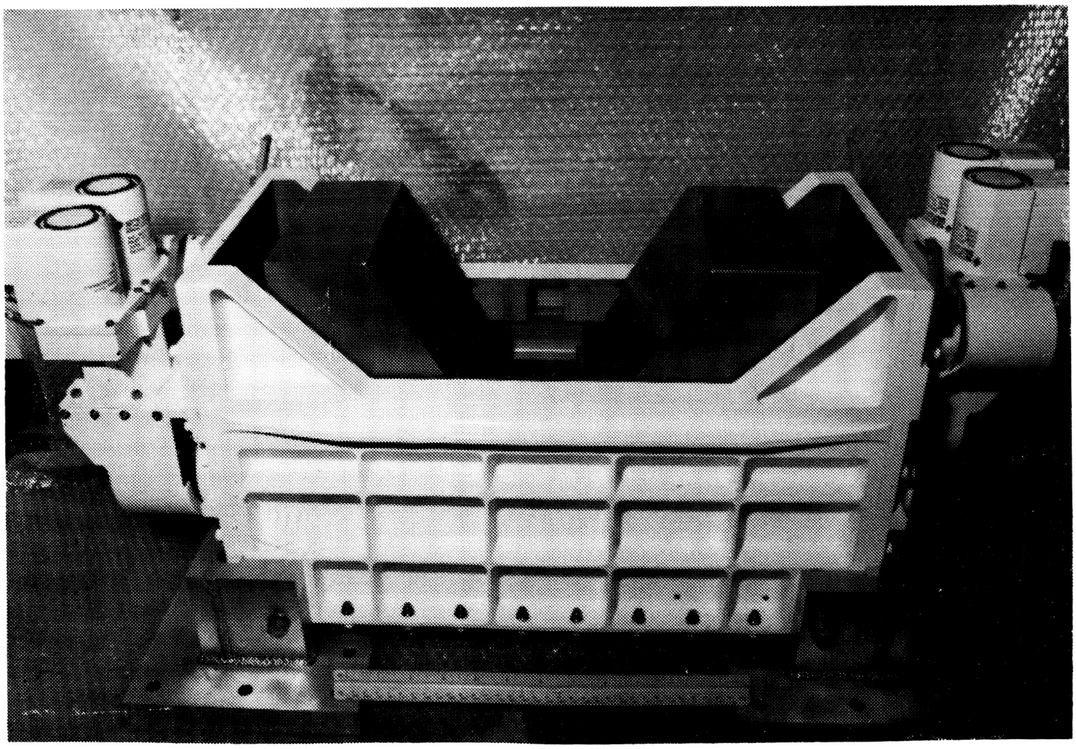

Figure 10. - Keel latch test configuration.

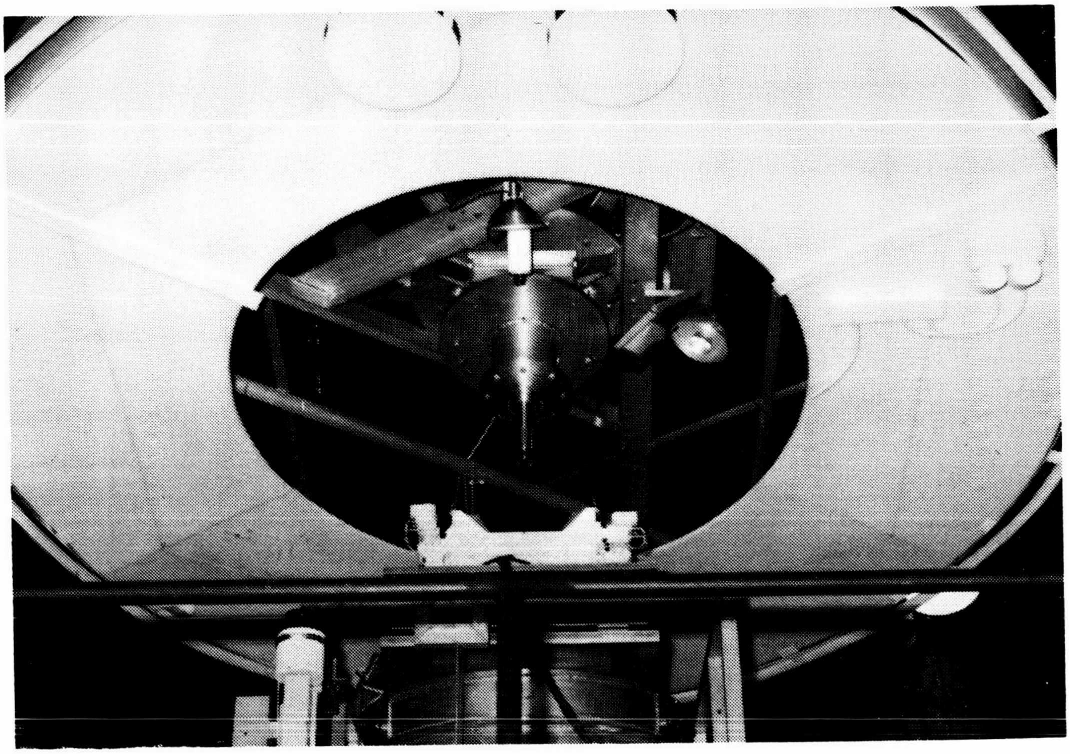

Figure 11. - Six DOF test facility.

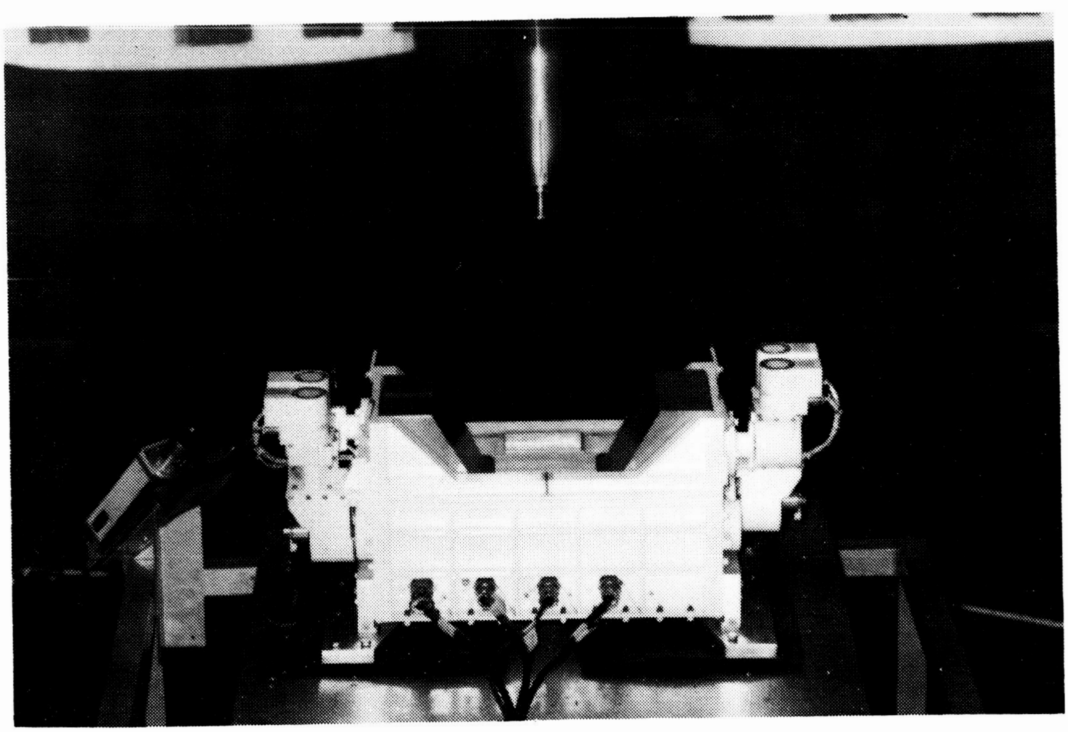

Figure 12. - Six DOF test configuration.

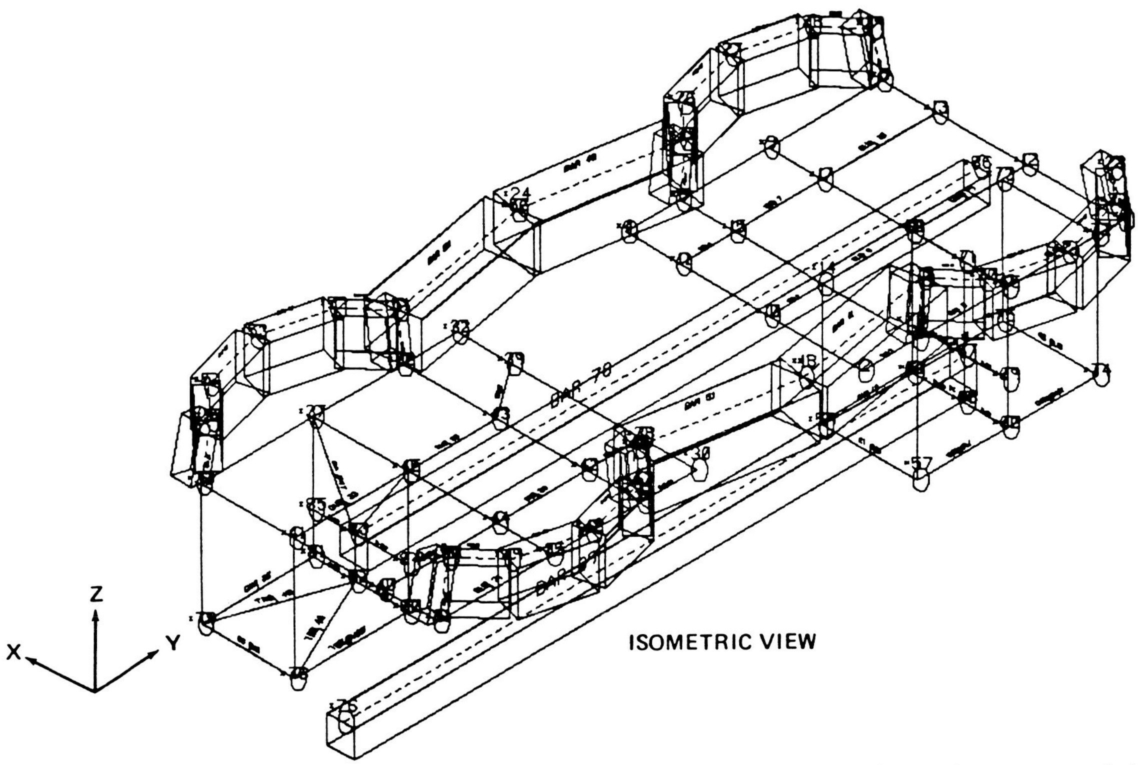

Figure 13. - HST maintenance mission keel latch finite element model.

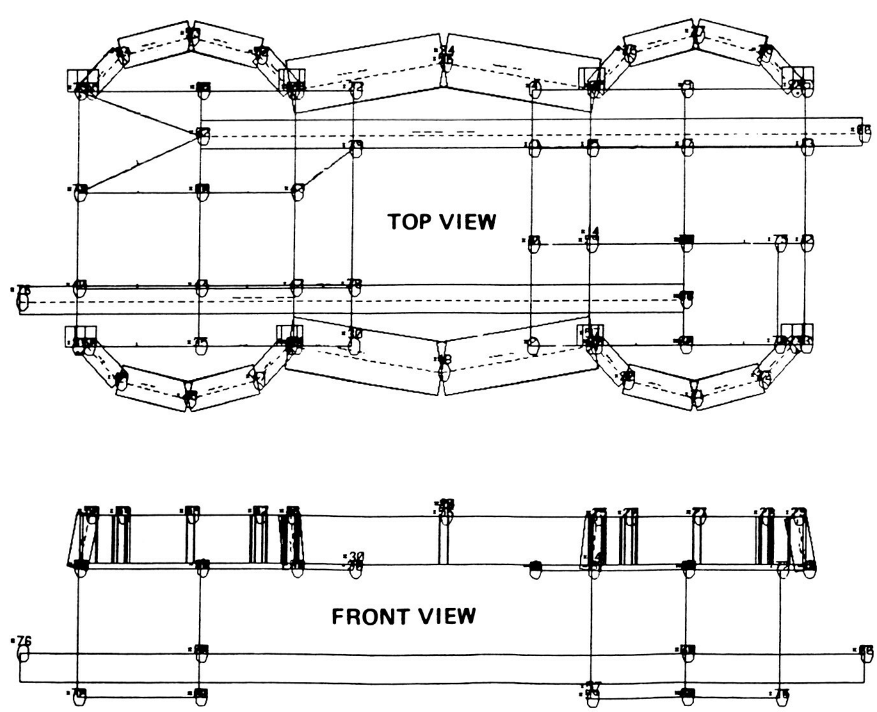

Figure 14. - HST maintenance mission keel latch finite element model.

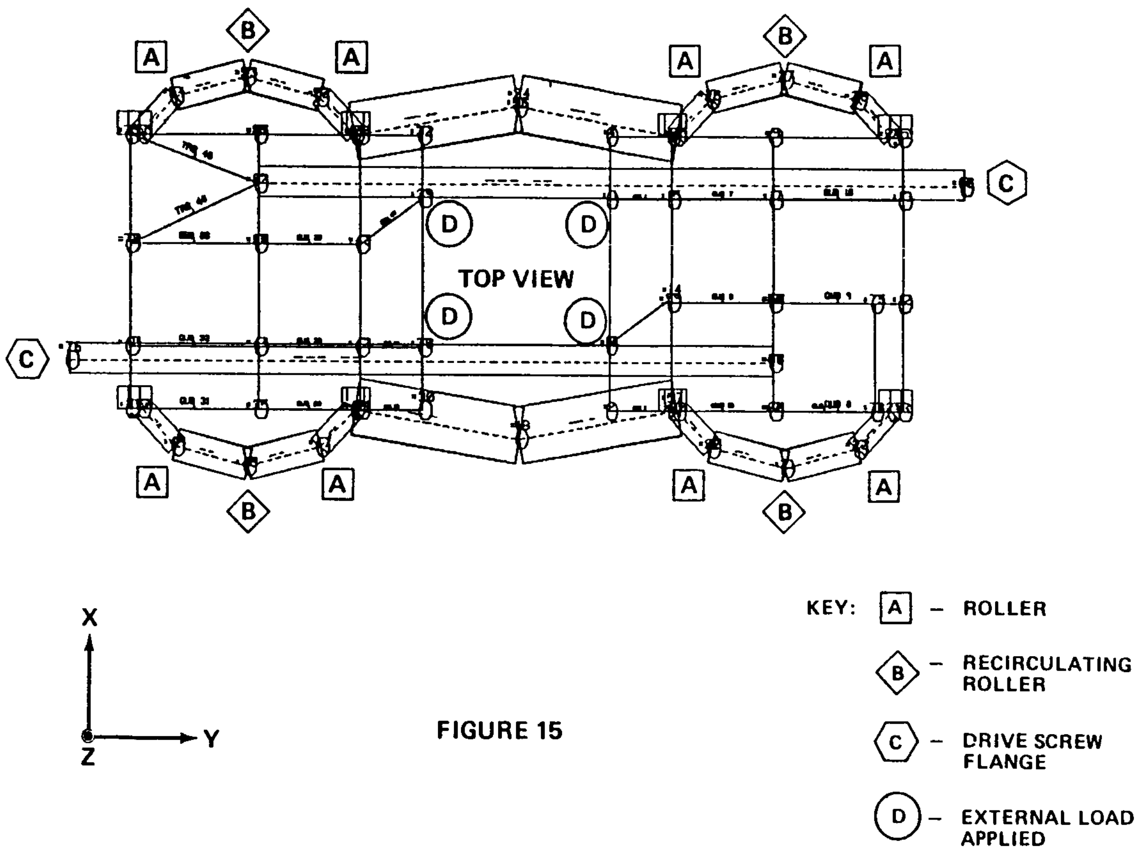

Figure 15. - Reaction and load application points.