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4 Stiffening of the ACES Deployable Space Boom

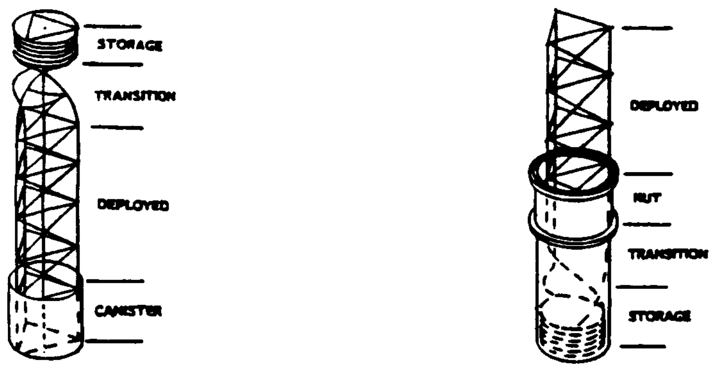
4.1 Summary

4.1.1 General Background Information

Space experiments require extremely stable and remote platforms to ensure that the experiment will operate properly. Deployable booms are used in many of these experimental platforms because of their light weight and compact size. The boom's light weight and compact size are required because of the high cost involved in transporting materials to space. Large, light structures are usually very flimsy, especially long booms. A combination of the boom's low fundamental frequencies, vibrations from the experiment itself, and vibrations at the boom's base can create conditions unacceptable for proper performance of many experiments. To make deployable space booms an effective experimental platform, the overall vibrations in the system must be minimized. One method to reduce these vibrations is by stiffening the deployable boom.

The beneficial effect of stiffening is twofold. Firstly, when the vibrational frequencies are higher, there is less energy in the system. Secondly, higher frequency vibrations are simpler to control. High performance controllers operate at high bandwidths. If the vibrational frequencies are low, it is difficult to match the controller. Therefore, if the frequencies can be increased by stiffening, the high frequency controller manipulates the system more efficiently.

The idea of using tensioned cables and spreaders, in a manner similar to a sailboat mast, allows the designers to minimize increases in both size and weight while significantly decreasing vibrations in the boom. The role of this design group is to use the concept of tensioned cables and spreaders to create an active tensioning system that will allow them to assess the effectiveness of this method to stiffen a deployable space boom.



Lanyard Deployed Boom   Nut Deployed Boom

Figure 4.1. Two Types of Deployable Booms

4.1.2 Design Objective

To design an active planar stiffening device for the existing ACES structure.

4.1.3 Abstract

The purpose of this design project was to design an active planar stiffening device for the existing ACES structure. The ACES structure was modeled using simple beam theory. Various concepts were generated about how the stiffening device should be configured in order to perform at an optimum level. The optimum configuration was selected to be a single set of spreaders located approximately 63% of the distance down the beam. Actuation was to be provided by a DC electric motor. From the test results, the design group was able to draw conclusions and make recommendations about the utility of further research into this area.

4.2 Glossary

ACES	=Active Control Evaluation for Structures
b	=length from the top to the point of maximum deflection
E	=Young's Modulus
I	=mass moment of inertia
l	=length of the boom
LMED	=Linear Momentum Exchange Device
x/l	=dimensionless length ratio
β	=weighted frequency of mode
σ	=constant

4.3 Background Information

4.3.1 Customer Requirements

The customer for this design project was defined to be the engineers at Marshall Space Flight Center in Huntsville, Alabama. Their intent was to have the group design an active tensioning system on which they could then perform control experiments. The primary contact for the definition of the customer requirements was Dr. Henry Waites. Also helping with the customer requirements were Dr. Ephrahim Garcia and Mark Whorton. The final list of customer requirements is as follows:

(1) Only the first mode of vibration must be damped for a clamped, pinned beam. It follows that if that mode can be excited by the system, the system can also damp that mode by generating it 180 degrees out of phase.

(2) The amount of deflection adequate to damp the first mode of vibration is equivalent to an approximate deflection of 15 cm at the point of maximum deflection.

(3) The weight of the control device should not exceed the weight of the existing control system. The existing control system weighs eight times as much as the ACES structure or about 40 lb. The weight of the control system needs to be roughly half of this amount, about 20 lb.

(4) The cost of the system must be reasonable enough to be constructed with available funds. This limit is considerably less than the cost of the existing system.

(5) The system should be constructed so that it can be attached to the existing structure without interfering with the existing control system or the constraints defined by the building. The limiting building constraint is a hole cut into a steel floor through which the boom passes. The

rectangular opening is 26 inches by 37 inches. This opening is located 26 bays or about 12 feet from the top of the boom. The geometry of this configuration limited the length of the spreaders to just under 3.5 feet or about 1 meter.

(7) The system should not damage or harm the ACES structure during the attachment to the structure or during the testing of the control system on the structure.

(8) It is not necessary for the system to be space qualified. It will only be used as a prototype on the ACES structure.

(9) The power supply to the control system does not need to match the power supply offered by the space station. The standard spacecraft power is 27 VDC, but the system can utilize any power source available.

4.3.2 ACES Structure Background

The existing ACES structure as shown in Figure 4.2 is a 45 foot astromast boom with a 10 foot antenna attached at its bottom. The ACES structure is excited by a two axis active traverse that can simulate the firing of a positioning thruster. The current vibration suppression system consists of two sets of two-axis linear momentum exchange devices (LMEDs). An LMED is a solenoid that accelerates an attached reaction mass. The vibration in the astromast is evaluated by a laser that is bounced off of a mirror on the antenna at the end of the mast to an independent, powered gimbal mirror, back to an optical detector pad on the antenna.

TEST ARTICLE: 45 FOOT ASTROMAST BOOM
WITH TEN FOOT ANTENNA
REAL-TIME COMPUTER SYSTEM PROCESSES
37 SENSORS AND 11 ACTUATORS

1. BASE EXCITATION TABLE
2. 3-AXIS BASE ACCELEROMETERS
3. 3-AXIS GIMBAL SYSTEM
4. 3-AXIS BASE RATE GYROS
5. 3-AXIS TIP ACCELEROMETERS
6. 3-AXIS RATE GYROS
7. OPTICAL DETECTOR
8. MIRRORS
9. LASER
10. 2-AXES POINTING GIMBALS
11. LMED SYSTEM

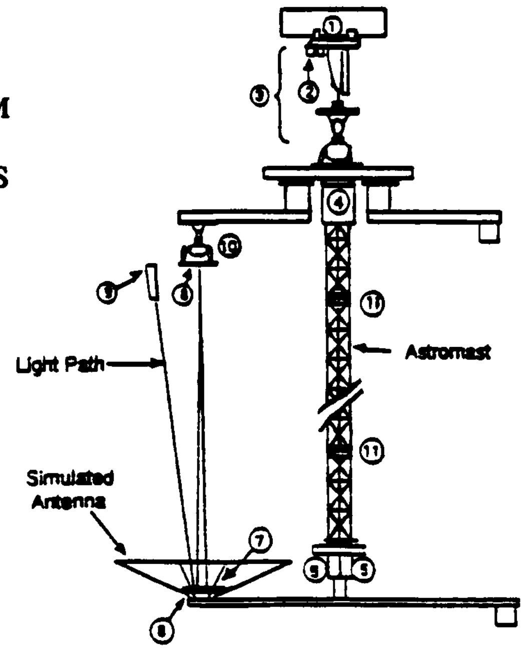


Figure 4.2. ACES structure.

The astromast boom is made up of 96 bays (Figure 4.3) that are triangular prisms 5.625 inches in height and 7.875 inches in length on each side. The overall astromast boom is a linear triple helix with a 260 degree twist from top to bottom. The fact that the astromast boom consists of over 1100 separate members made a Finite Element Analysis model an inconceivable goal for a one semester project.

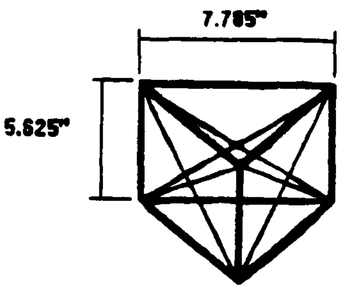


Figure 4.3. Single ACES Bay

4.4 Analysis

Due to time and logistical constraints, only the motion in one plane will be considered. Controlling the motions in one plane will sufficiently prove the viability of the tendon-spreader configuration as an active vibration suppression system.

Certain assumptions must be made in order to model the boom. The first assumption is that the boom can be modeled as an elastic, slender beam in transverse vibration. This assumption allows the use of simple beam vibration equations to determine the point of maximum deflection. The next step is to determine the appropriate mode to be damped and define the corresponding boundary conditions. The first two modes are the most critical in that they contain the majority of the vibrational energy. The first vibrational mode is the first mode of vibration of a clamped-free beam, while the second mode is the first vibrational mode of a clamped-pinned beam.

The clamped-pinned arrangement was chosen because the mass at the top of the structure is significantly large when compared to the mass of the beam. The end mass is also significantly large, however some rotational movement can be observed. These facts lead to the clamped-pinned model.

Ideally, the first mode for the clamped-free case would be damped because it contains more energy. The configuration necessary to damp this mode would have to be able to generate enough force to overcome the motion of the large end mass. The combination of weight limitations and spatial constraints prevents this from being an option.

In order to damp the first mode for the clamped-pinned case, a choice must be made between dealing with the maximum deflection or the maximum slope of that mode. While the maximum slope will occur at the bottom of the boom, the maximum deflection will occur at some point near the middle of the boom. In order to deal with the maximum slope of the mode, the spreaders must be located at the bottom and force a rotation of the entire end mass. This would require a large motor which would be capable of supplying a necessary force to overcome the rotational inertia of the bottom mass. A more realistic goal is to increase the natural frequency of the boom itself. This goal can be obtained by supplying a force normal to the boom at the point of maximum deflection.

The point of maximum deflection was determined by locating the point of zero slope. The equation for the mode shape of a clamped-pinned beam is:

$$0=\cosh(\beta_r x)-\cos(\beta_r x)-\sigma_r\left(\sinh(\beta_r x)-\sin(\beta_r x)\right)$$

$$\text{where } \sigma_r=\frac{\cosh(\beta_r l)-\cos(\beta_r l)}{\sinh(\beta_r l)-\sin(\beta_r l)}$$

$$\text{and } \beta_r l=7.06858275.$$

Solving the first equation for x/l, the point of maximum deflection was found to be approximately 63 percent of the length from the top of the boom. Since the boom is 13 meters long, the point of maximum deflection is 8.2 meters from the top.[1]

In order to determine a preliminary size for the motor, the relationship between tension and beam displacement was found. The configuration in Figure 4.4 was used to determine this relationship.

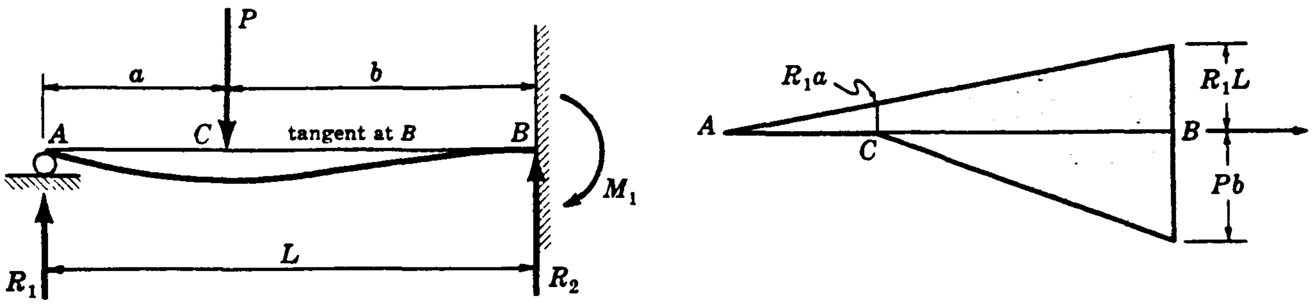


Figure 4.4. Clamped-pinned beam model and moment area diagram

Equations for the reactions at points A and B, as well as the moment at B, were obtained from the following equations.

$$R_1=\frac{3Pb^2}{2L^3}\left(a+\frac{2}{3}b\right)=\frac{Pb^2}{2L^3}(2L+a)$$

$$R_2=\frac{Pa}{2L^3}(3L^2-a^2)$$

$$M_1=\frac{Pa}{2L^2}(L^2-a^2)$$

$$EI\Delta c=abR_1+\tfrac{1}{3}b^3R_1+\tfrac{1}{3}b^3P$$

The following values were used for the variables in the equations:

$L = 13$ m $\quad$ $I = 3.35 \times 10^{-7}$ m^4 $\quad$ $b = 8.19$ m $\quad$ $E = 2.86 \times 10^{10}$ N/m^2 $\quad$ $a = 4.81$ m

Using the reactions found above and the moment area method, the final equation relating force and displacement was determined.

4.7 $\qquad$ $\Delta c = .003\ [\text{m/N}] \times P$

From the geometry of the system, the tension in the tendons can be directly related to the displacement at the spreader. This equation is given below.

4.8 $\qquad$ $T = .0935[\text{N/m}]\times\Delta c$

4.5 Design Concept

The design concept generation phase involves both brainstorming and consideration of previous designs. The purpose of the design is functionality and reliability, so simplicity of design is a major consideration. Therefore, the customer requirements are kept in mind at all times with complex configurations eliminated in lieu of simpler, yet innovative systems.

The first step in concept generation involves examining prior designs. The present configuration of the ACES stiffening device includes linear momentum exchange devices. While this concept of damping the structure's vibration is effective, the weight of the linear momentum exchange devices comprises a large portion of the entire structure's mass. The design team is thus faced with finding an alternate design that is just as effective yet which is of lower mass. Group brainstorming resulted in a concept that seemingly fits the low-mass requirement: a spreader/tendon design. Tendons placed internal to the structure are considered, but dismissed due to the amount of motor torque necessary to produce the desired vibration damping. An external spreader design is chosen so as to reduce the power requirements of the motor and reduce the possibility of the buckling. This design requires fewer electrical considerations (the motor is the only source of energy consumption), is a fraction of the prior design's mass, and is of much lower cost.

Now that the basic design concept is selected, a means by which to tension the tendons is considered. Again, a previous design is examined. A similar design project by Barry Dunn, a graduate student at Vanderbilt University, employs linear motors in order to excite vibrational modes in a structure similar to the ACES, but on a smaller scale. Linear motors are investigated as a means of quick tendon retraction. Although these motors are ideal for the structure studied by Dunn, the very limited motor travel is not suitable for the length of tendon retraction required for the substantially longer ACES structure. In addition, the linear momentum of the motor mass introduces additional vibrational modes that need further damping. The design group is thus faced with finding a method that is capable of retracting several inches of tendon in a short time. A high torque, high speed DC motor attached to a spool is chosen. It must be capable of high angular accelerations. The cost of such a motor is less than that of a linear motor and does not contribute to vibrations in the plane of study.

Spreader material must also be considered. Brainstorming the problem yielded several possibilities. A thin wooden dowel, metal rod, plastic rod, and graphite rod were considered. Wood dowels are strong enough in tension and compression, but any bending might result in fracture. Metal rods are adequate in this area yet are high in mass. Plastic rods are difficult to break, yet bend far too easily and would lead to inaccurate force transmissions. Therefore, graphite rods were chosen for their extremely high yielding point, stiffness, and low mass. In addition, Dunn's project used the same graphite rods (manufactured by a kite company) with excellent results.

Also adopted from Dunn's project is the tendon material. DuPont Spectra kite string is used with a test strength of 500 lb., more than adequate for the forces required.

Installing tendons on the ACES structure introduces the problem of exerting forces via the spreaders on the structure, presenting the possibility of local buckling. The use of a mounting plate alleviates this problem. The mounting plates for the LMED's were used as a pattern for a spreader mounting plate. A machine drawing for the mounting plate can be found in Figure 4.10 in the appendix.

4.6 Active Tensioning Mechanism

The active tensioning system of the Aces structure consists of a motor that alternately tensions opposing sides of a continuous tendon running down opposing sides of the structure. The force from the tendons is translated to the structure by a set of spreaders located along the beam. The system can be divided into five main parts as illustrated in Figure 4.5.

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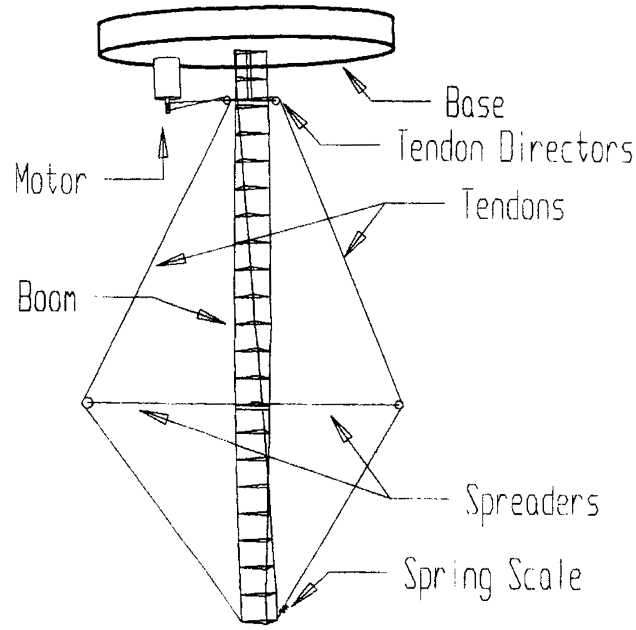

Figure 4.5. Active tensioning mechanism

The system is driven by a DC electric motor. The motor is mounted to a counterweight that is connected to the baseplate. A spool is attached to the shaft of the motor and the tendon wraps around the spool and exits at both the top and bottom of the spool.

The tendon that is used in the tensioning system was chosen for its light weight and low elasticity. A logical choice for the tendon in this case was high performance kite string. The tendon selected is braided DuPont Spectra with a breaking strength of 500 lb. It has approximately three percent stretch before breaking and is very lightweight. We should never encounter any tensions exceeding ten percent of the yield strength. Therefore it is reasonable to assume that there is negligible stretch in the tendon, ensuring consistent and instantaneous response of the system to an active control device.

The dimensions of the spreaders were defined by the spatial constraints of the building surrounding the ACES structure as well as material considerations. At approximately 11 feet down from the base of the structure, the boom passes through a steel floor.

The hole in the floor is 26 inches by 37 inches. This geometry defined the length of the spreaders. This length was 96 centimeters from the centerline of the boom. Therefore, the total length of the spreader assembly would have to be 192 cm.

The other constraint on the length of the spreaders was available material. The material used for the spreader arms is a blend of fiberglass and graphite. The longest available length was 32.5 inches (82.55 cm). This meant that to assemble a 192 cm rod, two full sections and a short section would have to be connected. This extra section would weaken the system considerably so it was decided to use a spreader length of 82.5 cm from the centerline of the structure.

Low friction pulleys in which the tendons would ride were mounted on the ends of the spreaders. It was decided to use only a normal force to damp vibration, so pulleys had to be used to prevent a moment from being applied to the structure.

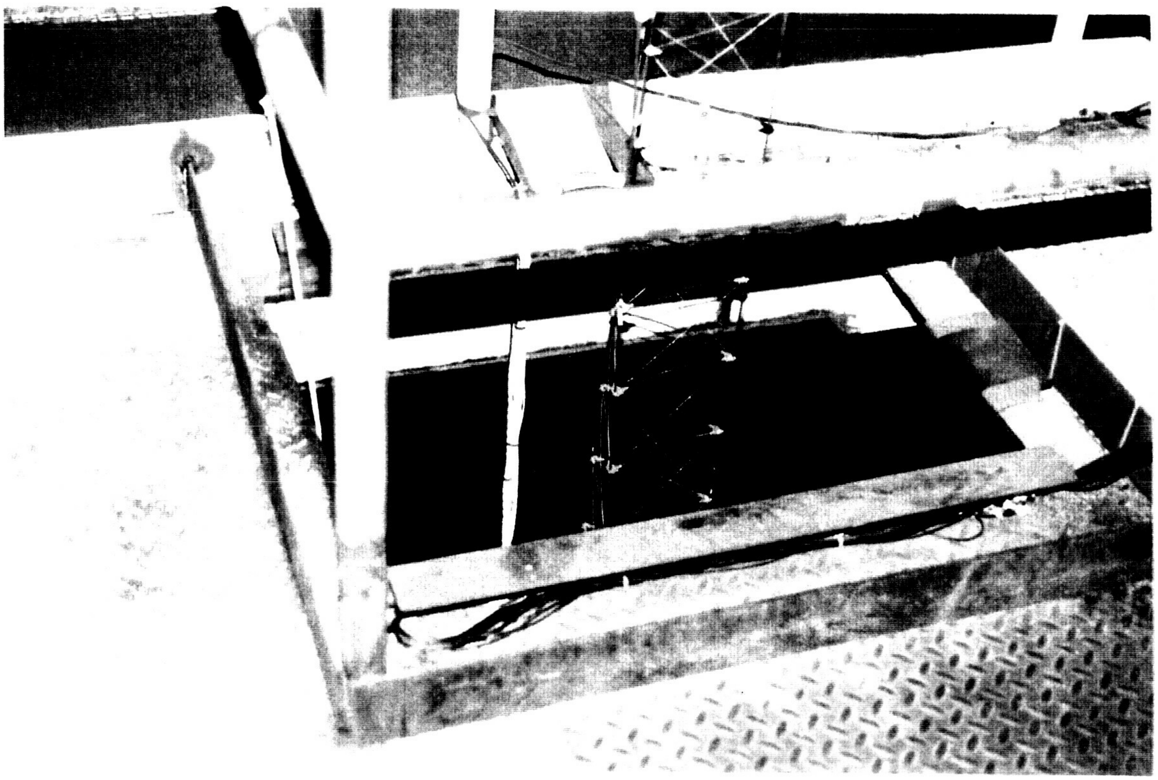

Figure 4.6. Main structural constraint.

The spreaders were mounted to the structure by the use of a mounting plate. The spreader rods connected through a specially designed clamp that attached to the plate with one bolt. The use of only one bolt allows the spreader to be correctly directed with respect to the structure when mounted. Both the plate and the clamp are shown in Figure 4.8.

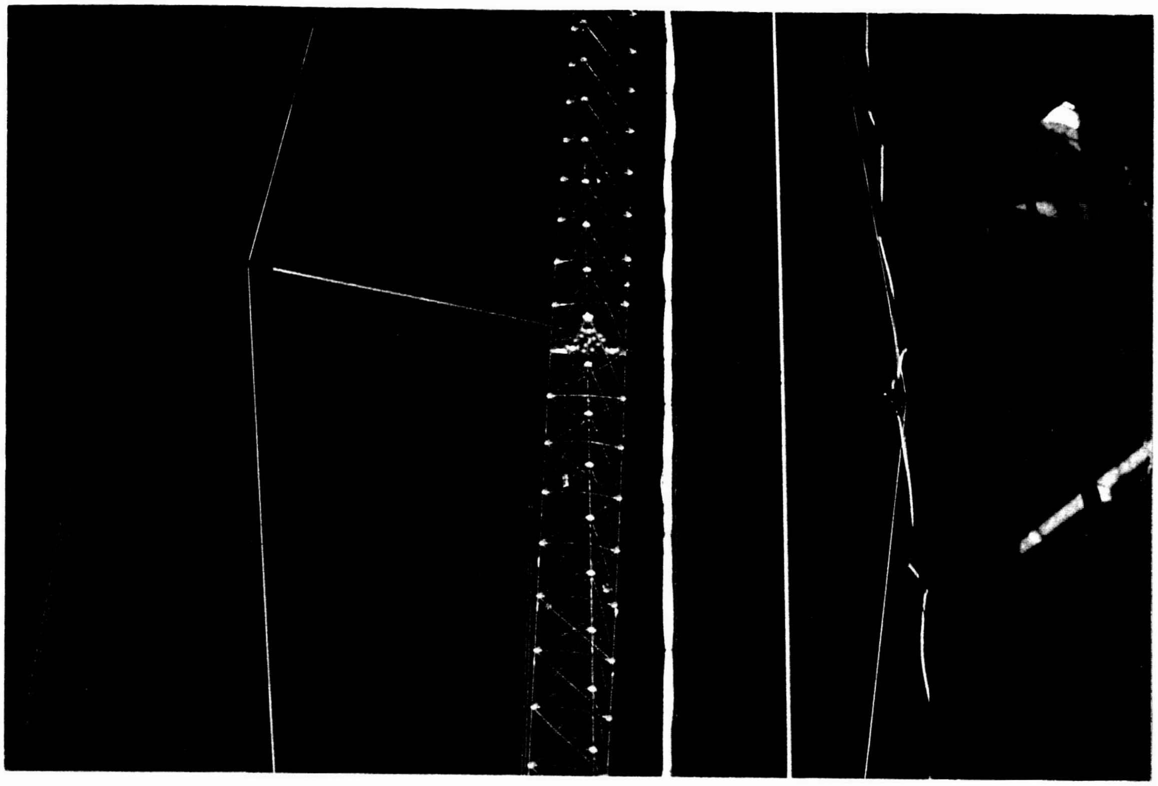

Figure 4. 7. Spreader assembly.

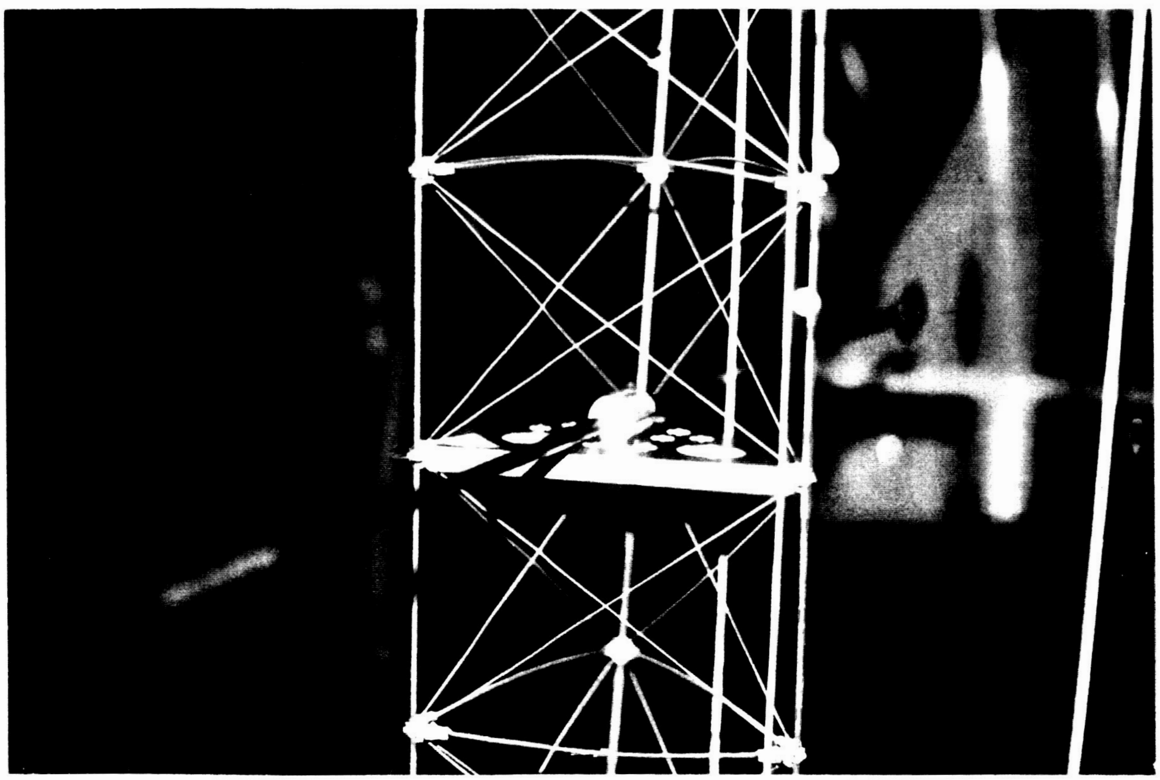

Figure 4.8a. Spreader mounting plate assembly

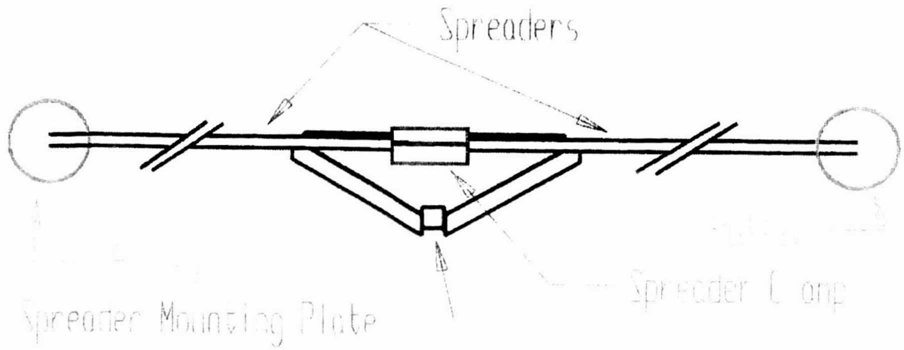

Figure 4.8b. Spreader mounting plate assembly drawing

Two different mounting mechanisms were used to attach the tendon to the bottom of the structure. On one side, a strap of webbing was used. This was ideal because it allowed for nondiscreet adjustments in tendon length, enabling the static tension in the tendons to be adjusted. On the other side, a spring scale was mounted between the structure and the tendon, showing the amount of static tension in the system. The bottom mounting system is shown in Figure 4.9.

Figure 4.9. Bottom mount with strap and spring scale

4.7 Testing

The original goal was to be able to excite the first mode of the clamped-pinned case. However, due to equipment considerations discovered after mounting, it was decided that this was not feasible. Consequently, the system was evaluated under static tension. The natural frequency within the plane of the tendon system was noticeably higher.

4.8 Conclusions

The frequency of the structure in the plane considered was markedly higher than the frequency in the orthogonal plane. This was due to the static tension applied by the system. Therefore, the conclusion can be drawn that the tendon system has stiffened the structure. No conclusions can be drawn as to the effect of the active tensioning system since the active system was never operational.

4.9 Recommendations

There are many problems associated with the current tensioning system design. Many of these are relatively simple hardware considerations. Others involve major theoretical overhauls and significant design changes. One of the main problems with the design of this system is the difficulty of trying to design a system to fit a structure that is not readily accessible. This led to some fitting problems with the hardware.

It is imperative that all of the pulleys are mounted in the same plane. If the pulleys do not line up correctly, the tendon will slip out of the track. A way to alleviate this problem would be to design an adjustable system which could be properly aligned after the system is mounted.

Another improvement to the system would be to utilize stronger pulleys, possibly with deeper tracks. The existing pulleys are rated at 4.4 kg, hardly enough to withstand active tensioning. Deeper tracks or a tendon guide would help to keep the tendon from jumping out of the track.

When considering the overall purpose of the system, it would stand to reason that to remove energy from a system, it should not be necessary to input energy into that system. This is especially important in space where energy must be conserved as much as possible. An electronically-controlled variable-damping clutch could be a viable alternative.

The application of the system is crucial to the design considerations. It is possible that the reduction of end mass motion is critical to some applications. A possible alternate system could be designed that utilizes moments. These moments would be produced by fixing the tendons on the ends of the spreaders and crossing over the boom. This configuration would control the motion of the end mass.

4. 10 References

(1) Inman, Daniel J., "Engineering Vibrations" p. 335 Table 6.4.

4.11 Appendix

4.11.1 Additional Figures

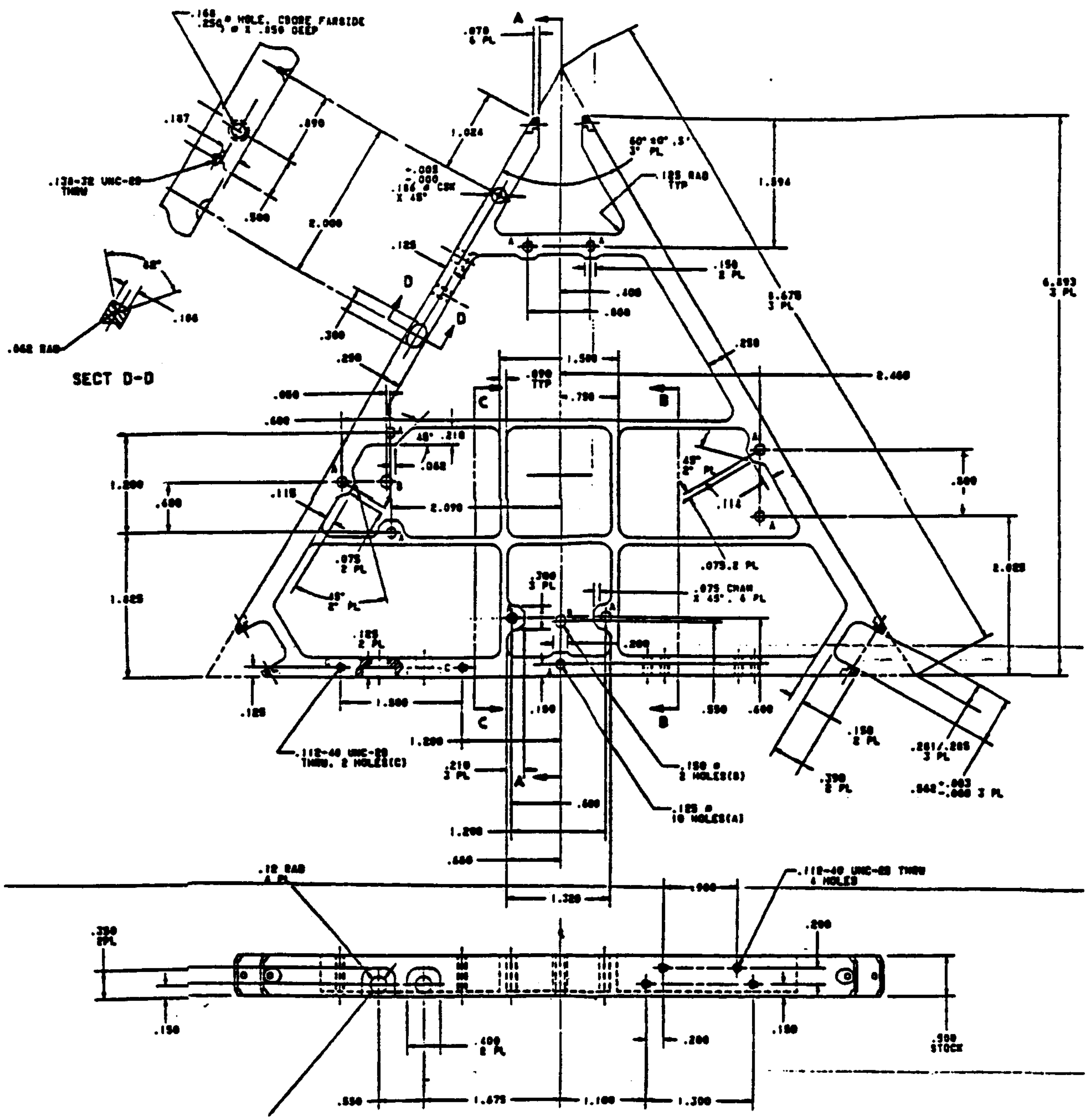

Figure 4.10. Spreader Mounting Plate

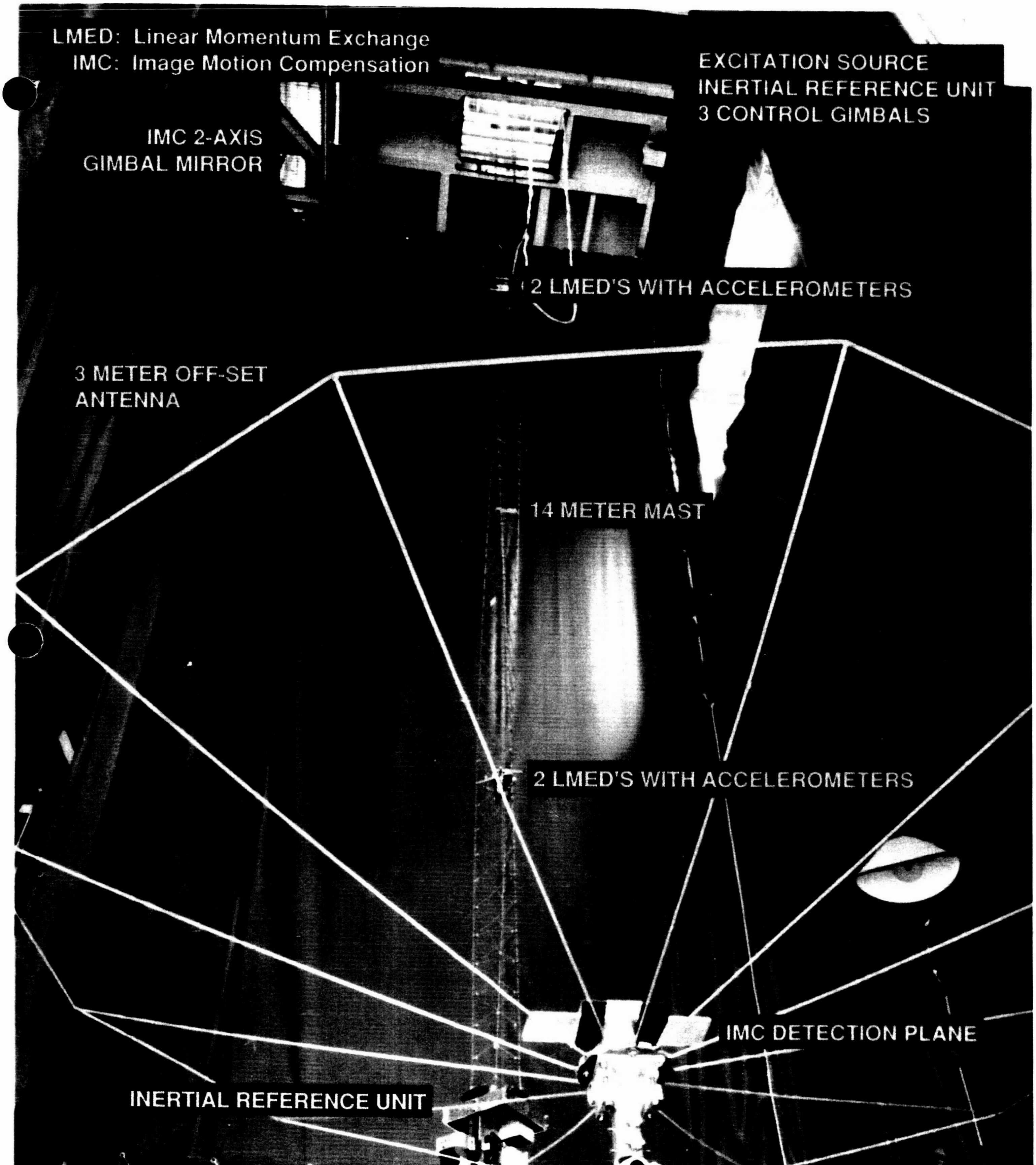

Figure 4.11. ACES structure and previous control system