N76-19184

12. A PRECISION SIX-METER DEPLOYABLE BOOM FOR THE MARINER-VENUS-MERCURY '73 MAGNETOMETER EXPERIMENT

By Harry F. Burdick

Goddard Space Flight Center

SUMMARY

A unique deployable boom has been developed for accurately positioning magnetometers 6 meters (19.7 feet) from a spacecraft. Position accuracy within ± ½° can be maintained. Weight, mounting system, magnetic cleanliness, thermal dimensional stability, and natural frequency were critical constraints that were met. The boom was flown on Mariner 10 and deployed flawlessly. The design, development, and testing of the boom and optical alignment of the sensors are described. Design trades and problem solutions are discussed.

INTRODUCTION

A Goddard Space Flight Center Magnetic Fields Experiment was one of the science instruments on the Mariner 10 (Mariner-Venus-Mercury '73) mission launched November 3, 1973. This experiment was designed to measure the magnetic fields in interplanetary space and during Venus and Mercury encounters. Since the Mariner type spacecraft was not designed to be magnetically clean, the effect of the spacecraft field had to be determined. This was accomplished by providing a 6 meter (19.7 foot) long deployable boom that would place three orthogonal fluxgate sensors seven meters from the spacecraft Z axis and a second set of sensors 4.5 meters (14.7 feet) from this axis.

Unique design restraints were imposed by spacecraft and experiment considerations. The length of the boom when folded about its two hinged joints for launch was 3 meters (9.8 feet). This length precluded end restraint using the spacecraft structure. End tie-down was obtained by latching to the solar array tip tie via a swivel fitting and tie rod. The lack of rigid restraint of this end and the large excursions of the solar panel during ascent imposed high dynamic loads on the boom and sensors. In addition, possible dynamic interactions of a long boom with the Mariner three-axis stabilization system required a minimum natural frequency of 0.57 Hz.

REQUIREMENTS

Some significant design requirements of the boom are listed below.

1. Deploy an outboard magnetometer package to a maximum practical distance ($\geq$ 6m) (19.7 ft) from the Z axis.

2. The outboard magnetometer axes should be known throughout the mission within $\pm\frac{1}{2}^{o}$ with respect to the spacecraft axes.

3. Deploy a second magnetometer package inboard of the first by approximately 2m (16.6 ft). The axes should be within $\pm\frac{1}{4}^{\circ}$ of the first package.

4. The natural frequency of the supporting boom should be 0.57 Hz minimum.

5. The temperature range of the sensors must be controlled.

6. The supporting boom must be compatible with the spacecraft and launch vehicle constraints.

7. Nonmagnetic materials shall be used wherever possible.

DESIGN FEATURES

A. Deployment Sequence

The boom in the prelaunch configuration is folded about its two hinged joints. One hinge (inboard hinge) is attached to an outrigger on the spacecraft body. The other hinge (outboard hinge) joins the two tubular boom sections. Figure 1 illustrates the deployment sequence.

Simultaneously with release of the two spacecraft solar panels, the tie rod on the outboard hinge is released. The first stage of deployment takes place with the two boom elements latched together rotating through an angle of 106°. At completion of that phase, lockup of the inboard hinge takes place and the outboard element is released by a sequencing mechanism which is incorporated in the inboard hinge. The outboard element then swings through an angle of 180° and locks in place to complete the deployment. All mechanical functions were monitored by switches.

B. Deployment Mechanisms

Energy for boom deployment was provided by individual constant torque spring motors at each hinge. Figure 2 illustrates the torque spring configuration on the inboard hinge. A spool which is free to rotate on its axis is mounted to the nonrotating hinge half. A spring that is prestressed in the heat treatment process, such that it desires to coil up on this drum, applies torque to a spool fixed to the boom element. This spool is centered over the hinge pin and mounted to the rotating hinge half, thus the torque is applied to deploying the boom.

Detent mechanisms latched the hinges in the deployed position to provide the required boom alignment and stiffness. The outboard hinge detent mechanism is shown in Figure 3 in the stowed position. This hinge deploys through 180°. Stops machined into each hinge half determine the deployment angle and absorb the impact loads. A spring-driven tapered detent pin drops into a tapered hole to solidly latch the hinge. The hole and pin are angularly displaced ¼° at lockup so that the tapered pin rests on one side of the tapered hole and acts as a wedge driving the hinge stops together so that there is no play.

The sequencing mechanism that phased the deployment so that the two hinges opened sequentially is shown in the prelaunch position in Figure 4. A saddle located at the tip of the outboard boom section is locked to the inboard hinge by two sequencing pins. Completion of inboard hinge deployment allows the detent pin to drop into the hole, simultaneously withdrawing the trapezoidal-shaped wedge and releasing the spring-loaded sequencing pins so that they are withdrawn from the holes in the saddle. The saddle and outboard boom sections are then free to move away under the torque generated at the outboard hinge.

STRUCTURE

The structural design of the boom tubes was complicated by a combination of design requirements--strength (imposed by the launch environment), stiffness and weight (imposed by the spacecraft stabilization system), nonmagnetism (imposed by experiment), and straightness under solar heating (imposed by experiment axis orientation requirements and spacecraft trajectory). A graphite/epoxy composite material was selected as the best candidate material to meet these requirements. It exceeded the strength and stiffness of aluminum with approximately one-half the weight, and had a near zero coefficient of expansion. This latter property eliminates the problem of thermal bending from being a serious concern. After completion of design trades, the design that evolved was a 38-mm (1.49-in) diameter tube with a 1.8-mm (0.07-in) wall having six longitudinal plies of high strength/low modulus graphite/epoxy and six radial plies of a synthetic fiber/epoxy.

Titanium fittings were bonded to the boom tubes for attachment of the magnetometers and hinges. Titanium was selected over aluminum and magnesium primarily because its lower coefficient of expansion creates less stress on the bond. The hinge structural parts were machined from solid blocks of aluminum to shapes resembling I beams with cross web stiffness.

In the prelaunch configuration, the boom tubes are preloaded by spacers located approximately 1 m (3.28 ft) from each end. This load is applied when the saddle (Figure 4) is latched by the sequencing mechanism. The saddle, in addition to applying the preload, also acts as a torsion restraint on the outboard boom section. The preload prevented relative motion between the tubes during vibration and increased stiffness of the boom assembly.

DYNAMICS

A. Deployment

Deployment parameters as calculated during the design phase are shown in Figure 5. For a given mass distribution and deployment angle the only independent variable is the deployment spring motor torque. The table (Figure 5) shows some of the parameters generated for a family of spring net torque values. The upper half of the table represents inboard hinge deployment and the lower half is the outboard hinge deployment. A net torque of 2.26 N·m (20 in-lb) was the upper limit because the resulting bending moment generated at the hinge was close to the design limit load. Any net positive torque would be sufficient to assure deployment. Component tests indicated the resistive torques, due to friction in the hinges and flexing of the electric cable service loop at the hinges, could vary from about 0.34 N·m (3 in-lb) to 0.79 N·m (7 in-lb) depending on deployment angle and temperature. The springs selected for flight were 1.69 N·m (15 in-lb) for the inboard hinge and 1.13 N·m (10 in-lb) for the outboard hinge.

Fifty percent higher torque springs were used for an overtest of the prototype. The maximum estimated net torque for this test was less than 2.26 N•m (20 in-lb).

B. Natural Frequency

The requirement that the deployed boom have a minimum natural frequency of 0.57 Hz proved to be the most troublesome of the design requirements. Design trades were made to maximize strength at a sacrifice in the margin of safety of predicted stiffness.

During the development, some unanticipated structural weight was added to the magnetometer packages and to the thermal blankets. This resulted in an adverse reduction of the natural frequency to 0.50 Hz.

A systematic weight reduction effort was initiated in order to raise the natural frequency. A NASTRAN model of the boom was utilized to determine the effect of the various options and combinations of options on the boom natural frequency.

The computer model proved to be responsive to load changes similar to the actual system and was very useful in selecting the weight reduction options to be implemented. Figure 6 shows the results of several runs. For successive runs, the weight reduction was cumulative. Run Numbers 013 and 014 represent reduced thermal material weights. Run Numbers 015 and 016 represent proposed redesign of the experiment packages to reduce their size and weight. Run Number 017 is a proposed structural modification to the boom that was not implemented because the weight reduction was adequate to meet the requirement. The final flight unit had a frequency of 0.61 Hz and a weight of 8.1 kg (17.8 lbs). It should be noted here that the frequency variations we were looking for were in the second decimal place of frequency. The inability to model the true system precisely enough to get this second decimal place of frequency to coincide with the absolute value obtained empirically did not seriously degrade the usefulness of the model in selecting the modifications to be implemented.

TESTING

A. Static

Static load tests were conducted on structural components such as the hinges and boom tubes to verify that they met design load requirements. Each tube was fabricated with extra length to be cut off for test specimens. The specimens were tested to failure. These tests verified uniformity in the graphite/epoxy composite tube fabrication process.

B. Vibration

All of the vibration testing of the boom was done on the spacecraft. The complexities of the system, with mounting interfaces at each end of the folded boom having different input loads and frequencies simultaneously, made realistic testing off the spacecraft virtually impossible. The first test was conducted on an instrumented boom with mass models simulating the magnetometer packages and thermal control system. Component test specifications were generated from this test for the magnetometer packages and sun shades so that they could be tested prior to being installed on the boom. Two complete boom assemblies (the prototype and flight unit) with magnetometers were also tested on the spacecraft.

C. Deployment

Boom deployment tests were conducted under simulated "zero-G" conditions. The boom was supported in a horizontal position by overhead cables attached to the center of gravity of each boom segment. The cables were runover pulleys and counterweighted. Two setups were required for each full deployment, one to test the primary hinge deployment and the other to test the secondary, so that the overhead support point was directly above the deploying hinge. The second setup permitted an overlap of the two tests in that dynamic loads incurred during lock-in of the primary hinge were repeated. The location of the support cables over the outboard hinge created a lateral force component on the boom during the overlap portion of the test. The lock-in velocity was duplicated by releasing the boom from a position between the stowed position and inboard hinge lock-up position that gave the same boom velocity just prior to inboard hinge lock-up as that obtained with the first setup.

A functional check of the hinge mechanisms was performed following each boom vibration test by conducting a restrained "walk-out" deployment. During the first test, the sequencing pins failed to withdraw from the holes in the saddle (Figure 4) preventing release of the outboard boom section. Upon inspection, galling was noticed at the tip of the sequencing pins. In order to correct the problem, the tip of the pins was changed from a truncated conical shape to a spherical shape. This modification prevented reoccurrence of the galling and there were no malfunctions during subsequent tests.

Low temperature deployment tests were conducted with the hinges and flexing cable service loops cooled by supplying liquid nitrogen to a thermal encasement that was removed when the desired temperature was obtained and deployment then commenced.

Strain gauges were installed on the hinges at the same locations that were monitored in the static load tests for the first series of deployment tests. These tests verified that the deployment loads did not exceed the calculated loads (Figure 5).

D. Natural Frequency

Measurement of the boom's natural frequency was performed in a test configuration similar to that used for deployment. The overhead support points were directly above the center of gravity of each boom segment. The inboard hinge was attached to a very heavy, rigid fixture. An optical tracking system monitored the tip of the boom. A voltage proportional to displacement was recorded on an oscillograph along with a precise 1 Hz reference signal. The boom was displaced and tip motion was recorded as it decayed from about 3 cm zero-to-peak (1.17 in) to less than 0.2 cm (0.79 in). The pendulum effect of the 23-m (75.5-ft) overhead support lines would tend to increase the frequency by about 0.01 Hz; this was subtracted to obtain the true zero-G natural frequency. The test was performed in two orthogonal planes - one in the plane of deployment.

The first boom assembly was also tested by a second method. The natural frequency was computed as the root mean square of two measurements taken with the boom mounted vertically - one with the tip up, and the other, tip down. It can be shown mathematically that this method's results is the zero-G frequency. It was found in this test that there were two uncoupled directions of oscillation. These were the two planes in which the horizontal tests were run. The test results for each plane were within 0.005 Hz for the horizontal and vertical test method.

ALIGNMENT

In order to meet the magnetometer axis alignment requirements, it was necessary to measure and/or simulate the boom curvature (deviations from its theoretical axis) that would occur in a zero-G field. Boom deviations in a horizontal plane were measured while the assembly was supported by floats in pans of water so that the boom was level and free to move horizontally. Measurements of curvature were made utilizing a theodolite that was collimated to a mirror which was coplanar to the boom-to-spacecraft mounting interface and perpendicular to the boom axis. The boom was then rotated 90 degrees and leveled. Straightness was measured again. The data obtained was a useful approximation of the zero-G curvature.

The magnetometer packages were then installed for alignment as shown in Figure 7. The supporting floats were adjusted in height so that the boom was forced into a curvature in a vertical plane which duplicated that shape previously measured. The boom was again free to move horizontally. A three dimensional curvature approximating that which would occur in a zero-G field was obtained. An optical alignment fixture was then installed on the magnetometer sensor packages. Alignment adjustments were made and checked using the theodolite.

CONCLUDING REMARKS

A deployable boom has been developed for positioning magnetometer sensors at a distance of 6 m (19.7 ft) from a spacecraft. Use of nonmagnetic constant torque spring motors at each of the two hinges produces controlled and reliable deployment forces. A sequencing mechanism provides an orderly unfolding motion. Spring driven pins are used to keep the hinges tightly locked after deployment. Each critical mechanical function is monitored by a switch so that its status can be determined through telemetry. The boom was flown with complete success on the Mariner 10 mission.

Questions and Answers Pertaining to

Mr. Burdick's Paper (#12) Given at

9th Aerospace Mechanisms Symposium

QUESTION - Member of the audience from the Boeing Company

What sort of force margins did you attempt to maintain; that is, how much did the force available for the deployment exceed the spring torque?

ANSWER

The resistance torques as measured at the component level with a hinge and cable in a cold box varied between 3 and 7 in./lb. depending upon temperature. The lower the temperature, the more the torque, that was for the inboard hinge which had the 15 in./lb. springs. There was a smaller cable with less resistance at the outboard hinge.

QUESTION - Member of the audience

How were the bearings lubricated?

ANSWER

There were no ball bearings, just hinge pins in close tolerance holes and sliding contact lubricated with molybdenum disulfide.

QUESTION - George Sandor, RPI

Which modification of the sequencing pins helped - the stronger springs or the rounded point? Were they tested separately?

ANSWER

No, the rounded point, I am quite sure, is the one that eliminated the galling and that in itself may have done the job, but increasing the spring force gave an extra margin of safety, particularly if there were side loads on it. I didn't mention, but also, these sequencing pins and that saddle had to absorb the torsional vibration loads of the boom so it also served that function as a structural member as well as a sequencing system.

QUESTION - Gilbert

How were the hinged joints protected from the sun - a bellows arrangement?

ANSWER

The inboard hinge was protected with coatings, you can look at that on the table in the rear. The spacecraft did provide some protection for the inboard hinge. The outboard hinge was completely in full time sunlight, that had a flexible sunshade which included the same materials we used in our thermal blanket's outer layer (silver teflon coated outer layer), and it was foldable.

QUESTION - Gilbert

Was it wrapped around the hinge?

ANSWER

No, it was a flat piece held by bracketry and a frame at each end, but no framework longitudinally; it was just folded up like a sheet of paper.

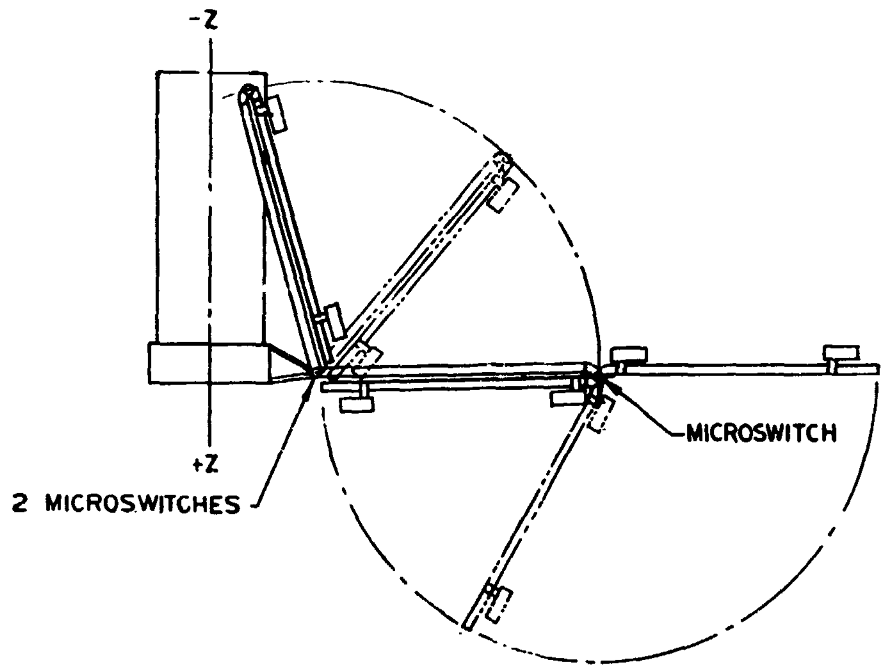

Figure 1. Deployment Sequence

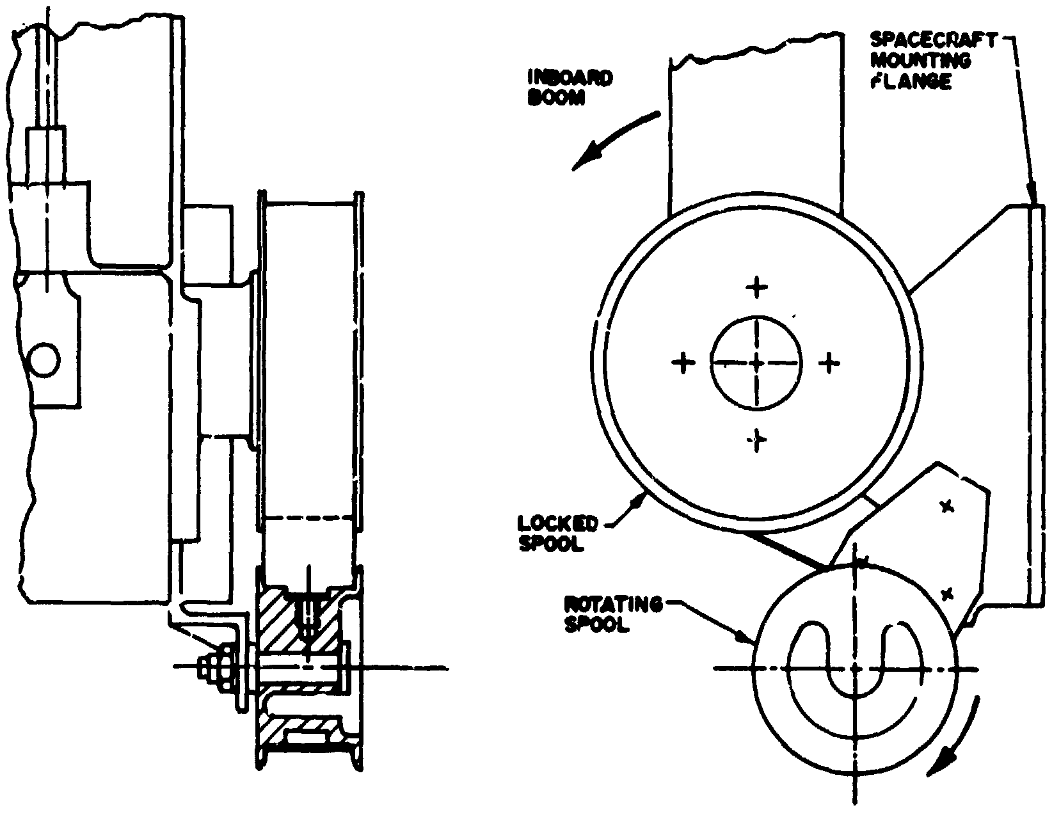

Figure 2. Torque Spring

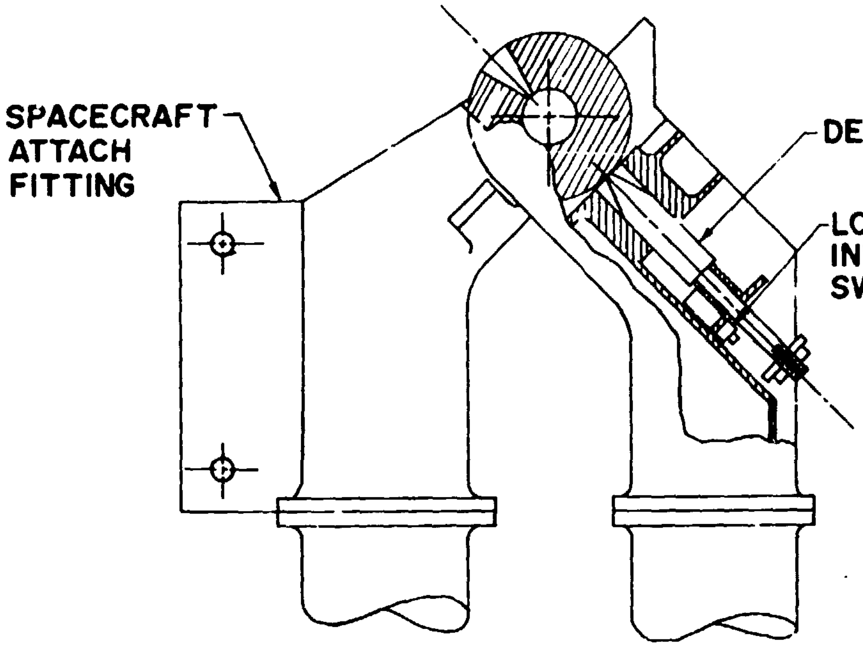

Figure 3. Detent Mechanism Outboard Hinge

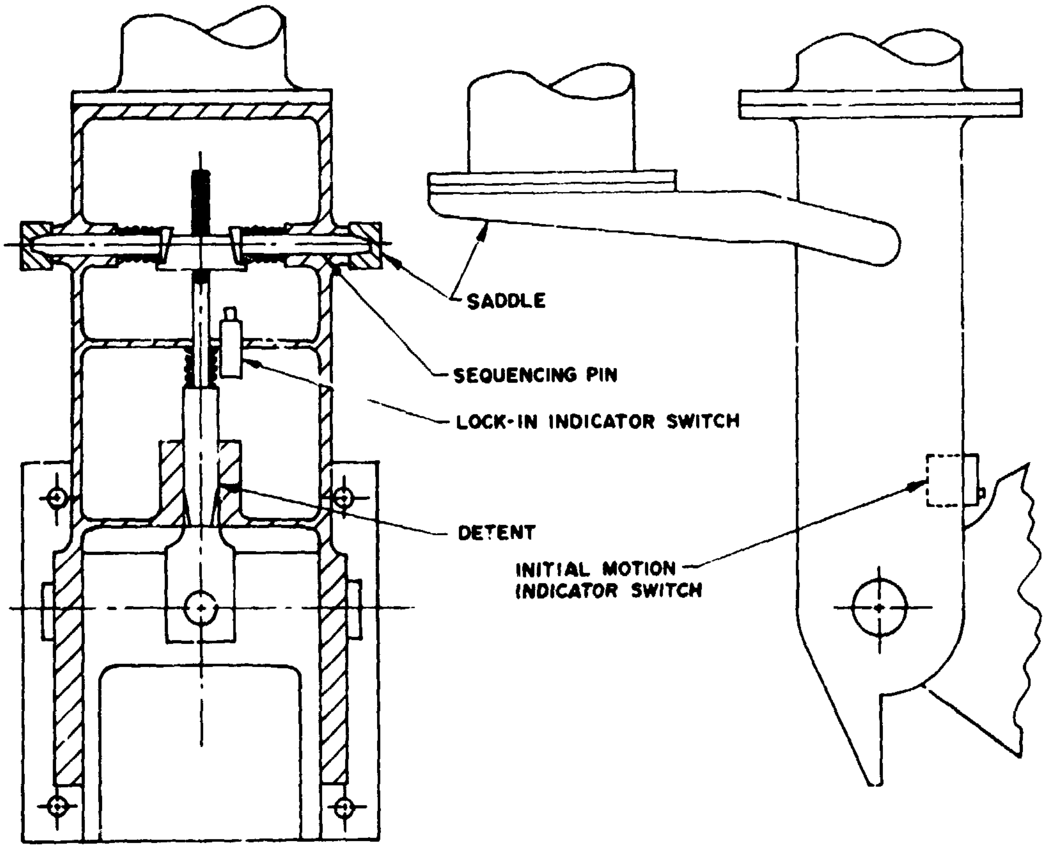

Figure 4. Sequencing Mechanism Inboard Hinge

	CASE NO	NET TORQUE N.m (IN-LB)	DEPLOYMENT ANGLE DEG.	DEPLOYMENT TIME SEC.	VELOCITY AT LOCKUP RAD/SEC	MAXIMUM BENDING MOMENT N.m (IN-LB)	TIP DEFLECTION Cm.
INBOARD	1a	0.56 (5.0)	106.5	12.22	0.304	114 (1006)	7.2
	1b	1.13 (10.0)	106.5	8.64	0.430	161 (1422)	10.2
	1c	1.41 (12.5)	106.5	7.73	0.481	180 (1590)	11.4
	1d	1.69 (15.0)	106.5	7.06	0.527	197 (1743)	12.4
	1e	1.98 (17.5)	106.5	6.53	0.569	213 (1881)	13.4
	1f	2.26 (20.0)	106.5	6.11	0.608	227 (2010)	14.4
OUTBOARD	2a	0.56 (5.0)	180.0	7.06	0.598	102 (906)	30.1
	2b	0.85 (7.5)	180.0	6.34	0.701	120 (1062)	36.0
	2c	1.13 (10.0	180.0	5.82	0.790	135 (1196)	40.6
	2d	1.41 (12.5)	180.0	5.43	0.871	149 (1320)	44.7
	2e	1.69 (15.0)	180.0	5.10	0.945	162 (1432)	48.6
	2f	2.26 (20.0)	180.0	4.62	1.077	184 (1632)	55.3

Figure 5. MVM '73 Magnetometer Boom Calculated Deployment Data

RUN NO.	N(Hz)	W kg (LBS)	C.G. m (IN)	I_{TOTAL}* Kg.m^2 (SLUG-FT^2)	CUMULATIVE WEIGHT REDUCTION
009	.52	9.06 (19.98)	3.55 (139.6)	141.24 (104.17)	--
013	.526	8.82 (19.45)	3.55 (139.8)	137.74 (101.59)	Removed .18 kg (.4 lb) as NSM[1] up to O.B. Mag[2], reduced O.B. Mag. by .05 kg (.1 lb) and I.B. Mag[3] by .02 kg (.05 lb)
014	.539	8.40 (18.53)	3.53 (139.85)	131.33 (96.86)	Removed .30 kg (.66 lb) as NSM up to end, reduced O.B. Mag. by .07 kg (.15 lb) and I.B. Mag. by .05 kg (.10 lb)
015	.607	7.65 (16.87)	3.37 (132.68)	108.60 (80.10)	Reduced O.B. Mag. by .61 kg (1.35 lb), I.B. Snubbers reduced .02 kg (.05 lb ea), O.B. Snubbers reduced .05 kg (.1 lb ea)
016	.618	7.17 (15.80)	3.37 (132.53)	102.98 (75.95)	Reduced I.B. Mag by .48 kg (1.063 lb)
017	.864	8.14 (17.95)	3.19 (125.67)	108.95 (80.36)	Same as 016 except with stiffeners

*Taken about inboard hinge pivot.

1. Non-Structural Mass
2. Outboard Magnetometer
3. Inboard Magnetometer

Figure 6. Results of NASTRAN Model Runs

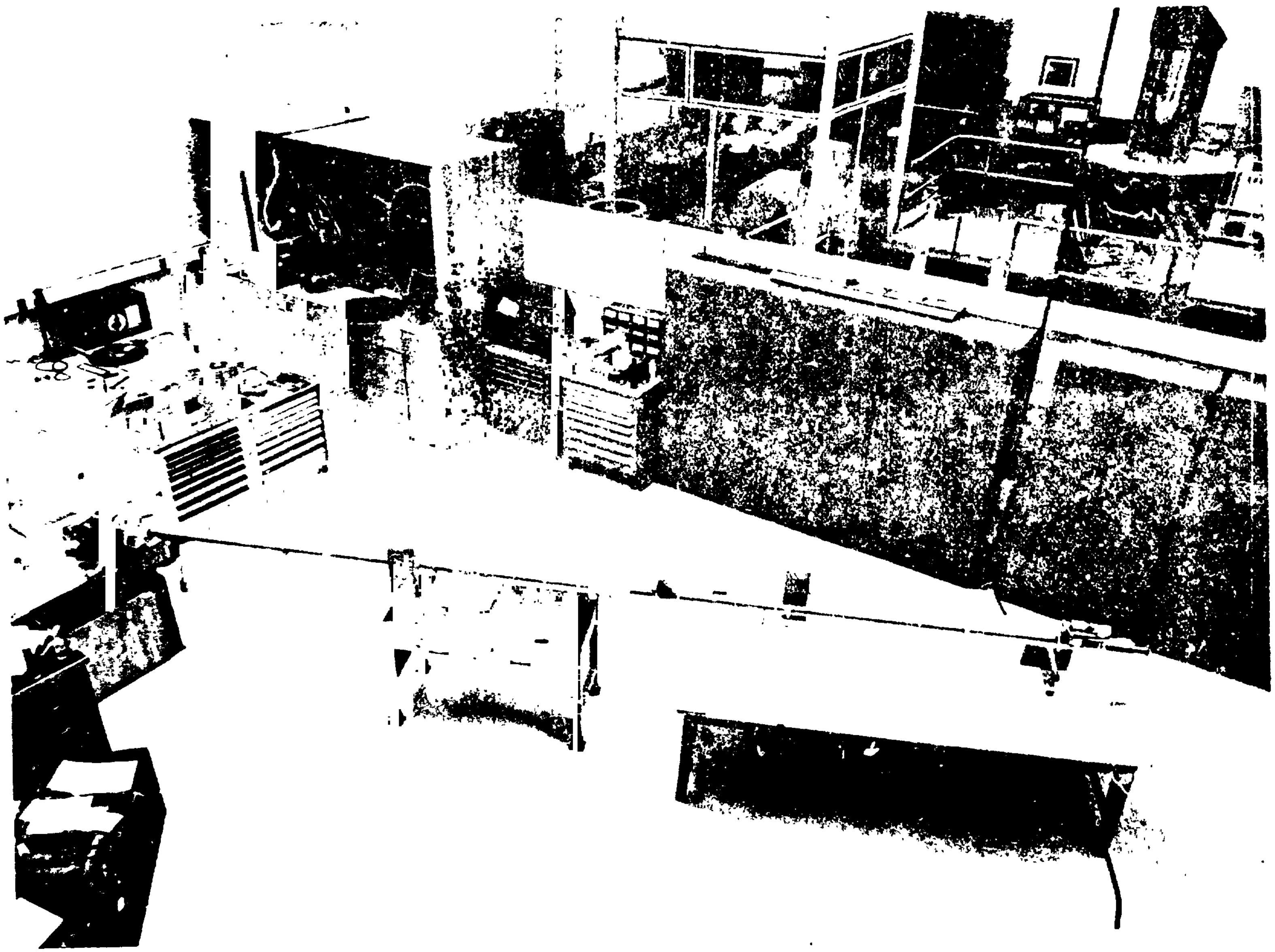

Figure 7. Magnetometer Package Alignment Installation