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Integrated Modeling and Analysis of a Space-Truss Article

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Abstract

MSC/NASTRAN is being used in the Controls-Structures Interaction (CSI) program at NASA Langley Research Center as a key analytical tool for structural analysis as well as the basis for control law development, closed-loop performance evaluation, and system safety checks. The objective of CSI research is to develop and validate the technology needed to design, verify and operate large space structures in which the structure and the control interact beneficially to meet the requirements of the 21st-century NASA and DoD missions. Guest investigators from academia and industry are performing dynamics and control experiments on a flight-like deployable space truss called Mini-Mast to determine the effectiveness of various active-vibration control laws. MSC/NASTRAN was used to calculate natural frequencies and mode shapes to describe the dynamics of the 20-meter-long lightweight Mini-Mast structure, predicting 153 modes below 100 Hz. Gravitational effects contribute significantly to structural stiffness and are accounted for through a two-phase solution in which the differential stiffness matrix is calculated and then used in the eigensolution. The frequencies of the first five modes calculated by MSC/NASTRAN are within five percent of the experimentally derived frequencies and analytical frequency response functions show good agreement with experimental FRFs. Reduced modal models are extracted for control-law design and evaluation of closed-loop system performance. Predicted actuator forces from controls simulations are then applied to the 153-mode model to predict member loads and stresses. These pre-test analyses reduce risks associated with the structural integrity of the test article, which is a major concern in closed-loop control experiments due to potential instabilities.

Introduction

NASA missions for the next century will probably employ large, lightweight spacecraft with precision performance requirements. Structural dynamicists and control engineers are developing controls-structures techniques to design and analyze spacecraft which will meet these requirements. The development and validation of that technology is the principal goal of the Controls-Structures Interaction (CSI) program at the NASA Langley Research Center. Analysis and ground testing play key roles in accomplishing the program objectives of significantly reducing system response amplitudes and accurately predicting on-orbit dynamics. Through the CSI Guest Investigator program, university and industry experts are working together with NASA researchers in solving this interdisciplinary problem by conducting comprehensive active-vibration-control experiments on a realistic large space structure. The purpose of this paper is to describe the use of MSC/NASTRAN to predict structural dynamic characteristics of the initial CSI test article, to provide a basis for design of active-vibration control laws and to verify structural integrity under test conditions.

The initial phase of the CSI Guest Investigator program uses a flight-quality 20-meter-long deployable space-truss called Mini-Mast, cantilevered vertically from the floor of NASA Langley Research Center's Structural Dynamics Research Laboratory (see figure 1). The truss, which has graphite-epoxy tubes and titanium fittings, is representative of future low-frequency space structures. The MSC/NASTRAN model was used to predict the modes and frequencies below 100 Hz, and a reduced 20-mode model was extracted for controls simulation to verify control laws developed by guest investigators. Actuator forces predicted by the simulations were then applied to the finite-element model using all the modes below 100 Hz to predict member loads and stresses. The predicted member loads were inspected to assure that the test article would not be damaged during the actual experiment.

Experimental Objectives

The technology of active vibration control for large space structures has been studied theoretically for many years (Reference 1). To date, however, there has been only limited experimental validation of these theories. Although many "proof-of-concept" experiments have been performed using small, single-component test articles such as simple beams and plates (e.g., Reference 2), these results are not fully applicable to realistic large space structures. Control techniques which work well with simple structures do not necessarily perform well with practical engineering structures due to nonlinearities, uncertain damping, measurement noise, suspension effects and other dynamic complexities. Thus there is considerable uncertainty concerning the accuracy and completeness of mathematical models used for the control designs.

The Mini-Mast facility was developed to help bridge this technology gap between simple and complex structures. Constructed using flight-like materials and workmanship, the Mini-Mast truss is representative of future large deployable trusses to be used in space. It provides a realistic benchmark for achieving the overall objective of the project, which is to obtain improved understanding of the practical performance characteristics of promising active-vibration-control techniques applied to large space structures.

Description of Mini-Mast

The Mini-Mast structure consists of a lightweight truss and two equipment platforms which provide mounting locations for actuators and sensors. The truss is composed of 18 bays, each 1.12 meters long, and its cross section is an equilateral triangle with vertices inscribed by a 1.4-meter diameter circle. Figure 2 shows the deployment of the first two-bay section of the test hardware. The maximum transverse force that can be applied at the tip of the Mini-Mast without buckling any of the structural members corresponds to a static tip deflection of approximately 5 centimeters, and the maximum torque that can be applied at the tip corresponds to a static tip twist of 0.9 degrees. The structure has been statically tested to 1.5 centimeters transverse tip deflection and 0.35 degrees tip twist. Dynamic safety limits are set to 0.762 centimeters tip deflection and 0.15 degrees tip twist.

The truss members are hollow graphite-epoxy tubes having a Young's modulus of 1.24×10^{11} N/M^2. The tubes are bonded to solid titanium fittings which connect the truss members to geometrically complex titanium structures called corner-body joints, shown in figure 3. The transverse truss members (battens), which form the triangular cross-section of the truss, have a rigid connection to the corner-body joint, ensuring cross-sectional stability during deployment. Diagonal members (diagonals) and longitudinal members (longerons) have a pinned connection to the corner-body joints, allowing the members to rotate during deployment. In addition, mid-diagonal hinges allow diagonals, folded in half during storage, to unfold during deployment and snap into a locked position. Figure 4 shows a mid-diagonal hinge in both the locked position and an intermediate position. For the ground-test program, the mid-diagonal hinges have been modified to ensure they remain locked.

The 1.45-meter square equipment platform located at the top of the mast (tip plate) is a 3.2-centimeter thick sandwich plate consisting of a hexagonal aluminum honeycomb core between two sheets of 0.24-centimeter-thick aluminum. Three torque-wheel actuators mounted on the plate, as shown in figure 5, can apply disturbance or control reaction torques about each of three orthogonal axes. Figure 6 shows the locations of the torque wheels as well as other actuators and sensors. The torque-wheel actuators, which use 0.61-meter-diameter annular inertia wheels and have a rated peak output of 67.8 newton-meters, are the only control devices available to researchers. Four linear accelerometers and a three-axis rate gyro are also mounted on the tip plate to sense tip motion. Another equipment platform (mid plate), made of the same honeycomb construction as the tip plate, is triangular shaped and attached with offset brackets within the cross-section at the top of bay 10. It is used as a mounting surface for two single-axis rate gyros and two linear accelerometers. A dummy mass of approximately 36 kg is attached to the mid-plate to simulate linear proof-mass actuators to be added in later experiments.

Other non-structural masses include circular disks attached to each corner-body joint on the structure, which serve as targets for non-contacting displacement sensors. Three independently supported shakers provide disturbance forces of up to 222 newtons at the corner-body joints at the top of bay 9. A tensioned vertical cable attached to the tip plate off-loads approximately the weight of the fully-equipped tip plate (1557 newtons) in order to prolong the fatigue life of the structure and reduce gravity-induced loads.

Description of Finite Element Model

The Mini-Mast finite element model, shown in figure 7, uses CBAR elements to represent the truss members and platform attachment brackets. An early model used rigid elements to represent the equipment platforms, but comparison of experimental and analytical results indicated that this simplifying assumption was inadequate. For the revised model, CQUAD4 and CTRIA3 plate elements are used to represent the equipment platforms. Isotropic material and physical properties for the plate elements were selected to closely approximate the out-of-plane bending stiffness and overall weight of the sandwich plate construction.

The geometrically complex corner-body joints were greatly simplified in the finite element model. Pinned connections were not modeled but were considered in establishing critical buckling loads for truss members. Modeling the pinned connections would have required a highly detailed corner-body joint model with local coordinates to identify the off-axis orientation of the pins (see picture of corner-body joint in figure 3). The idealized representation consists of short CBAR elements originating from a single point and connected by all six degrees of freedom to the adjoining truss members. This simplified model of the corner-body joints provided sufficiently accurate global response.

The MSC/NASTRAN coupled mass option was used to model the distributed mass of the truss bars and plates, and CONM2 elements were used to model the mass of the corner-body joints, corner targets, mid-diagonal hinges, dummy mass, sensors and actuators. The cable off-loading the tip mass was modeled with CROD elements. Table 1 shows some modelling details, including physical and material properties, and a mass summary.

Calculation of Normal Modes and Model Verification

The combined effects of gravity and cable tension were accounted for in the eigensolution using the two-step analysis procedure described in Reference 3. First, Solution 64 (geometric nonlinear) was executed with a combined gravity loading and temperature loading to calculate the differential stiffness terms to be added to the linear static stiffness matrix. The temperature loading was applied to the cable to simulate the effect of the force applied to the cable in the laboratory to off-load 1557 newtons of weight at the tip plate. Singularities caused by transverse deflection degrees of freedom of the cable nodes were removed using single point constraints.

The mass and stiffness matrices generated in Solution 64 were saved in a database, and a restart was performed with Solution 63 (superelement normal modes) to calculate the undamped natural frequencies and mode shapes using the Lanczos method. Transverse vibration modes of the cable were included in the solution by removing the constraints imposed during the Solution 64 differential stiffness calculations.

The eigensolution calculated 153 modes below 100 Hz, as shown in figure 8. The two first-bending modes, the first torsion mode, and the two second-bending modes are all below 10 Hz. A cluster of 108 "diagonal modes," with frequencies between 14 Hz and 20 Hz, is caused by various combinations of the localized first-bending modes of the 54 diagonals. The first global axial mode, the first bending mode of the tip-plate and the two first-bending cable modes also fall within the 14-20 Hz range.

Analytical mode shapes for the first and second bending modes, the first torsional mode and a typical diagonal mode are plotted in figure 9, together with a comparison of the analytical and experimentally determined natural frequencies (see Appendix). The first five global frequencies differ from the experimental values by less than 5 percent.

A set of analytical and experimental frequency response functions (FRFs) was generated to provide additional validation of the mathematical model. Experimentally-determined modal damping ratios were used in the analytical model for the first five modes, and 0.5 percent damping was used as a conservative estimate for all higher modes. Each analytical FRF was then compared to the corresponding experimental FRF (see Appendix). A representative plot shown in figure 10 compares an FRF for a displacement at bay 6 due to a force applied at bay 9. The comparison shows good agreement between the analytical and experimental FRFs below 10 Hz, which is the target range for the control experiments.

The analytical/experimental comparisons indicate that the mathematical model, with its simplifying assumptions, adequately represents the global dynamics of Mini-Mast. Not only does the finite element model accurately predict static response and natural frequencies, but the more complicated FRF comparisons provide detailed verification of input-output responses of the structure.

Control Law Development

Research is being performed to examine the practical implementation of control laws that use response measurements from sensors on lightweight, flexible space structures to generate actuator forces for vibration suppression. Individual guest investigators select a subset of sensors to use as feedback signals for their control laws designed to add artificial damping and suppress motion in the lower-frequency modes. The dynamics of the Mini-Mast structure are incorporated into the control system by reformulating the physical equations of motion, which were solved to obtain the frequencies and mode shapes, using the normal mode amplitudes as generalized coordinates (Reference 4). This process uncouples the equations of motion for the structure, and allows the physical response to be expressed as a linear combination of independent modal amplitudes, η. The modal equations of motion, scaled to unity modal mass, are written in the form

$$\{\ddot{\eta}\} + \begin{bmatrix} \diagdown & & \\ & 2\xi\omega & \\ & & \diagdown \end{bmatrix}\{\dot{\eta}\} + \begin{bmatrix} \diagdown & & \\ & \omega^2 & \\ & & \diagdown \end{bmatrix}\{\eta\} = \{p\} \qquad (1)$$

where $[\omega^2]$ is a diagonal matrix of undamped natural frequencies of the structure, ξ are the modal damping ratios, and $\{p\}$ are the modal forces. Modes are eliminated which do not contribute significantly to the global response of the structure due to a given input from the limited actuator and disturbance sources available on the Mini-Mast. Thus, the number of generalized coordinates needed to describe the structural dynamics is greatly reduced. The transformation from physical to modal coordinates also provides a convenient format for incorporating the structural dynamics into the complete control system description. The governing equations are reduced to the following 2m first order equations, where m is the number of structural modes retained in the formulation:

$$\{\dot{x}\} = [A]\{x\} + [B]\{u\} \qquad (2)$$

where the state vector, {x}, is defined as

$$\{x\} = \begin{Bmatrix} \eta \\ \dot{\eta} \end{Bmatrix} \tag{3}$$

The system matrix, [A], contains mass, stiffness and damping information; and the control and disturbance influence matrix, [B], relates the physical control and disturbance forces, {u}, to the state variables, {x}. The physical output of the structure, {y}, is expressed as

$$\{y\} = [C]\{x\} + [D]\{u\} \tag{4}$$

where the observation matrix, [C], relates the physical output to the state variables, and the matrix [D] relates the control and disturbance vector to the physical output. This state-variable description of the structural dynamics of Mini-Mast can be incorporated into a control system description, which includes details such as feedback loops and dynamic equations describing the behavior of components such as actuators, sensors and filters.

Guest investigators are provided with the full eigenvectors for the 153-mode analytical model, as well as dynamic models for actuators, sensors and filters required in the development of their control laws. They then select modes to characterize the dynamics of the system for their control law design, sometimes using only the first five modes. Digital control laws developed by guest investigators are provided in matrix format for incorporation into pre-test simulations.

Simulations and Safety Checks

Analytical safety checks consist of closed-loop simulation and member loads analysis. For the closed-loop simulation, 20 modes were selected from the 153-mode model to characterize the Mini-Mast global response to input forces. This reduced model is incorporated into a closed-loop numerical simulation that contains additional dynamic models of the torque wheels and rate gyro sensors. The dynamics of the accelerometers, shakers and noncontacting displacement sensors do not substantially affect the measurements within the experimental frequency range and are not included in the closed-loop analysis.

Simulations are performed using NASA's Integrated Multidisciplinary Analysis Tool (IMAT), which links commercial controls and finite element codes through a common engineering database (Reference 5). Through the IMAT process, MATRIXx/SYSTEM BUILD, developed by Integrated Systems Inc. (Reference 6), is used to check control law stability, compute motion at critical locations, and compute control-actuator force-time histories. Disturbance and actuator forces from this closed-loop simulation are then applied as open-loop disturbances to the 153-mode model, as indicated in Figure 11. To accomplish this, time histories of actuator and disturbance forces are saved in files which are converted to MSC/NASTRAN bulk data deck cards by MTSIM, a program developed by Structural Dynamics Research Corporation (Reference 7). Specifically, the TLOAD, DLOAD, TSTEP and LSEQ cards are created, as well as the TABLED1 tables. TABDMP1 cards for damping are created separately, together with all case control cards.

Transient analysis is performed using the 153-mode model from a restart via Solution 72 (modal transient analysis). The extended 153-mode model is used for displacement and stress recovery to avoid modal truncation errors which might occur with a model containing fewer modes. Experimental damping values used in the closed-loop simulation for the first five modes are reduced for the transient analysis to 0.5 percent in order to calculate the most conservative (highest) member loads. Deflections and rotations along the mast are calculated and checked against critical values. Stresses, moments, and axial loads are derived for longerons, battens, and diagonals at three truss locations: Bay 18 (nearest truss structure to torque inputs), Bays 9 and 10 (nearest truss structure to shaker excitation), and Bay 1 (at the cantilever root). The combined bending moment and axial loads are plotted, as shown in figure 12. Each data point plotted in figure 13 represents the worst-case combination of bending moment and axial load in any truss member for a single load case. Thus, data for many load cases are shown on the plot. The gray region represents a "soft limit" where concern for member fatigue could result in denying permission to apply a control law to the test article. The data point plotted in this region in the figure represents such a case. Both hardware and software shutdown switches are used in addition to the analytical safety checks described above.

Concluding Remarks

MSC/NASTRAN is being effectively used with other commercial codes to reduce the risk of structural damage to a flight-like testbed facility being used by researchers from government, industry, and academia. A two-step solution sequence, which included differential stiffness, provided an accurate eigensolution that is used for control law development. Time histories of disturbance and actuator forces predicted by closed-loop control simulations using a reduced modal model are applied as transient loads to the full modal model to predict truss member loads and stresses. Multiple safety checks provide assurance that the testbed facility will not be damaged due to unstable control laws, excessively high loads, or long-term fatigue.

References

1. Meirovitch, L., editor, Proceedings of the 7th VPI&SU/AIAA Symposium on Dynamics and Control of Large Structures, Blacksburg, VA, May 1989.

2. Aubrun, J.N., "Structural Control For a Circular Plate," AIAA Journal of Guidance, Control, and Dynamics, Sept.-Oct. 1984.

3. "Dynamic Analysis of Structures With Large Static Preloads," Application Note, MSC/NASTRAN Application Manual, The MacNeal-Schwendler Corporation, May 1981.

4. Cooper, P.A.; Stockwell, A.E.; and Kim, Z.C., "Interdisciplinary Analysis Procedures in the Modeling and Control of Large Space-Based Structures," Proceedings of the ASME Symposium on Engineering Data Management, Computers in Engineering Conference, New York, NY, August 1987.

5. Meissner, F.T., Ed., "Integrated Multidisciplinary Analysis Tool, IMAT, User's Guide for the VAX/VMS Computer," NASA TM 101500, December 1988.

6. MATRIXx User's Guide, V7.0, Integrated Systems, Inc., October 1988.

7. Blelloch, P., Preliminary MTSIM User's Manual, Version 1.0, Structural Dynamics Research Corporation, Engineering Services Division, San Diego, California, August 1989.

8. Hunt, D. L. and Peterson, E. L., "Multishaker Broadband Excitation for Experimental Modal Analysis," SAE Paper 831435, October 1983.

9. SDRC, I-DEAS Tdas, Test Data Analysis, User's Guide, I-DEAS Level 4, 1988.

10. Juang, J.-N. and Pappa, R. S., "An Eigensystem Realization Algorithm for Modal

Parameter Identification and Model Reduction," Journal of Guidance, Control and Dynamics, Vol. 8, No. 5, September-October 1985, pp. 620-627.

11. Pappa, R. S., "Identification Challenges for Large Space Structures," Presented at the 8th International Modal Analysis Conference, Orlando, FL, January 29-February 1, 1990.

12. Pappa, R. S., Schenk, A., and Noll, C., "ERA Modal Identification Experiences with Mini-Mast," Proceedings of 2nd USAF/NASA Workshop on System Identification and Health Monitoring of Precision Space Structures, Pasadena, CA, March 27-29, 1990.

APPENDIX: Experimental Procedures

Calculation of Frequency Response Functions

Experimental frequency response functions (FRFs), such as shown in figure 10, were generated using modern, multiple-input random excitation techniques (Reference 8). Uncorrelated random signals were applied simultaneously to each of the three shakers located at Bay 9 of Mini-Mast. Displacement, velocity, and acceleration responses, together with the three excitation forces, were recorded. These time histories were processed into FRFs using the SDRC I-DEAS Tdas software (Reference 9). This procedure consists of four steps: overlap processing of time history segments using 50- percent overlap, application of Hanning windows to minimize leakage, fast Fourier transformation to obtain frequency spectra, and accumulation of both auto- and cross-spectral matrices. Finally, these auto- and cross-spectra are used to calculate frequency response functions between all response measurements and each of the three excitation forces acting independently. Fifty ensemble averages were made to minimize sensor and background noise and to randomize the effects of moderate structural nonlinearities.

Modal Identification

The primary method used for experimental modal identification is the Eigensystem Realization Algorithm (Reference 10). ERA is a time-domain technique which decomposes free-response data into constituent modal components. Data for ERA consists of directly measured free-response time histories, impulse response functions obtained by inverse Fourier transformation of FRFs, or correlation functions. For Mini-Mast, results were derived using both free-response time histories and impulse response functions obtained from FRFs.

Experimental modal identification proved to be difficult for Mini-Mast due to the high modal density and nonlinearities arising from numerous joints. Results for the first five modes (two 1st bending modes, 1st torsion, and two 2nd bending modes), shown in figure 9, were relatively straightforward to obtain. For higher frequency modes, however, considerable identification scatter and uncertainty was encountered. A brief discussion of these difficulties is available in Reference 11. Further results, including a summary of overall findings, is contained in Reference 12.

Vibration Control Experiments

Considerable freedom is given to researchers using the Mini-Mast facility concerning experimental procedures. However, all hardware components have been selected and installed and are not readily changed. A simplified block diagram of the major facility components is shown in figure 13. Of the 97 analog signals available from the test article, only 35 are available to the guest investigators as feedback signals linked via fiber optic cables to a real-time digital controls computer, a CDC Cyber 175. The signals available for feedback are the output from 12 displacement sensors, 5 rate gyro sensors, 6 accelerometers, and both input and output signals from the 6 actuators. Redundant hardware and software safety switches stop the experiment if any preset critical limits are exceeded.

Individual GIs can perform experiments in many different ways. For example, either the shakers attached to Bay 9 or the torque-wheel actuators at Bay 18 can be used for disturbance generation. The type of disturbance signal, e.g., sinusoidal, random, or impulsive, can also be selected by the individual researcher. These disturbances, derived from either analog or digital sources, can be combined with the control commands sent to the torque-wheel actuators or they can be separately applied at different times than the control commands. Selection of disturbance characteristics is made based on the particular research objectives of the investigator. The specific objectives of the work are also selected by the individual researcher. In most cases, the objective is to perform a thorough validation of the effectiveness of a particular theory that has been developed by the researcher.

GRID POINTS/ELEMENTS

618	GRID POINTS
10	CROD Elements
548	CBAR Elements
160	CQUAD4 Plate Elements
6	CTRIA3 Plate Elements

MASS SUMMARY

Truss members (total)	20.90 Kg
Joints and hinges (total)	83.38 Kg
Fully-equipped tip plate	151.69 Kg
Fully-equipped mid plate	49.64 Kg
Total Mass	305.59 Kg

MATERIAL PROPERTIES

Tubes (Graphite-Epoxy)	E = 1.24E11 N/M**2
Joints (Titanium)	E = 1.13E11 N/M**2
Plates (Aluminum)	E = 6.90E10 N/M**2

PHYSICAL PROPERTIES

	LENGTH(M)	OD(mm)	ID(mm)
Longerons	1.092	20.2	14.9
Battens	1.212	15.1	11.9
Diagonals	1.623	15.1	11.2

DYNAMIC DOF

3481 Dynamic Degrees of Freedom

Table 1 Finite element model summary

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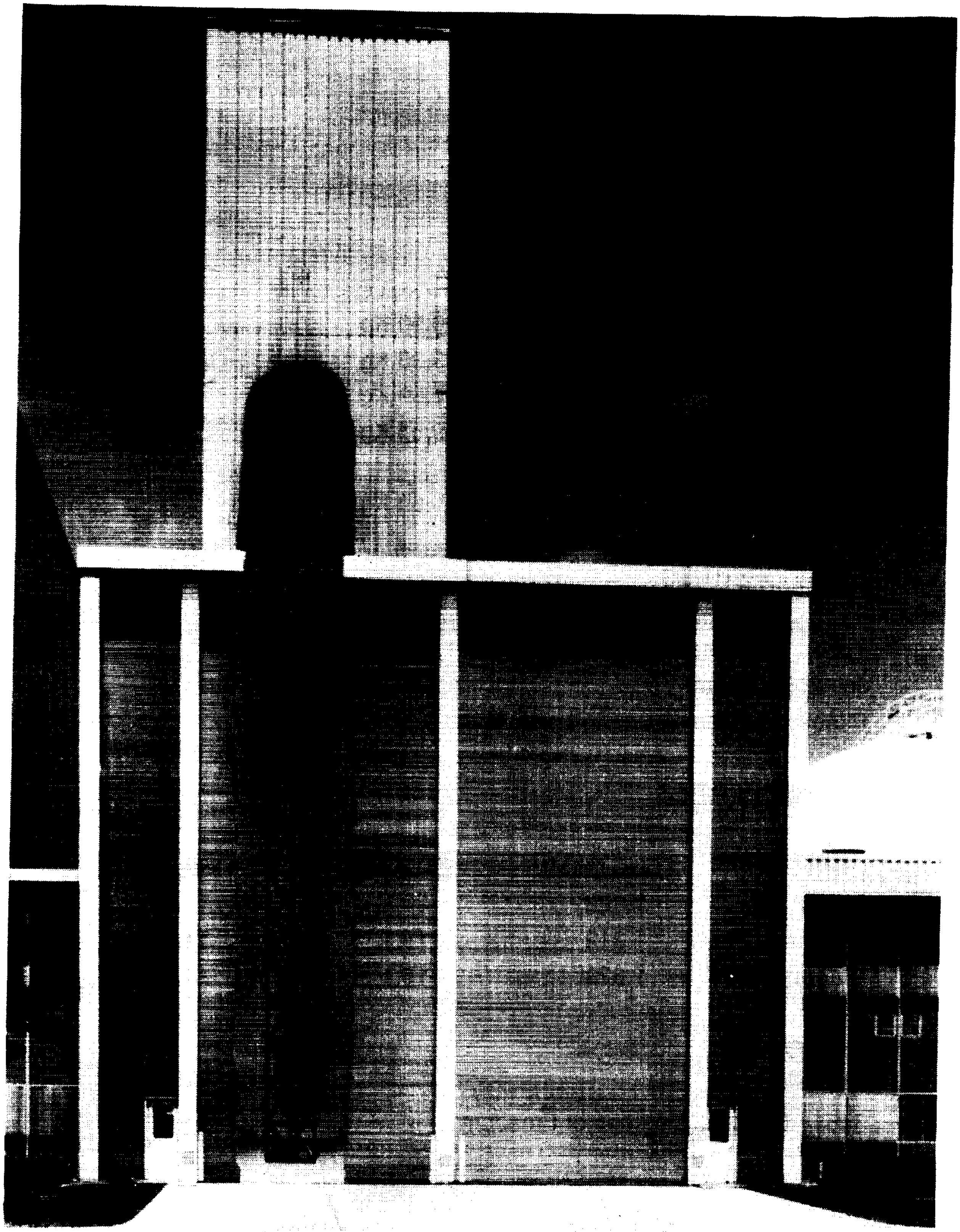

Fig. 1 Sketch of Mini-Mast inside Structural Dynamics Research Laboratory at NASA Langley Research Center

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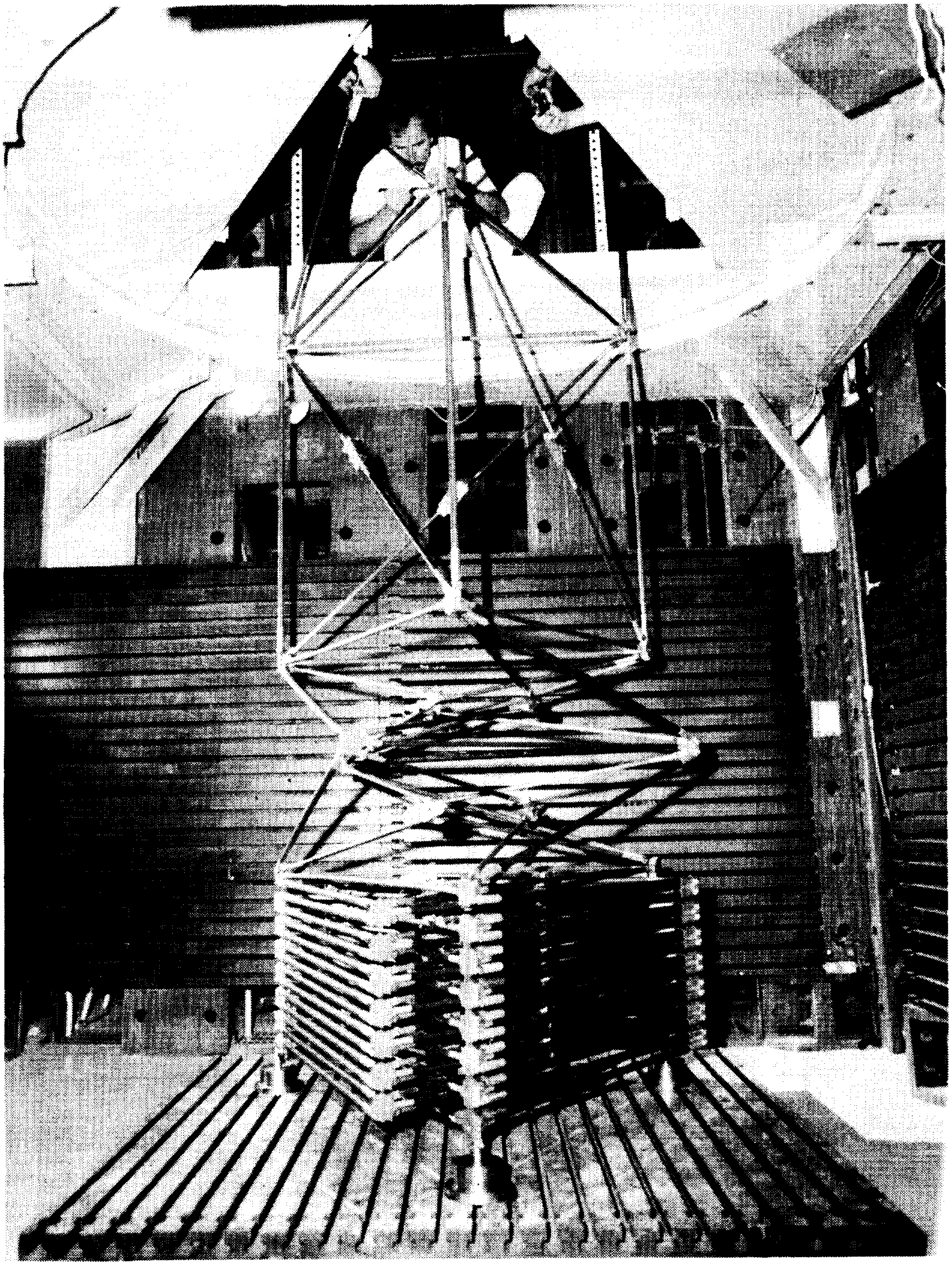

Fig. 2 Mini-Mast with top two bays deployed

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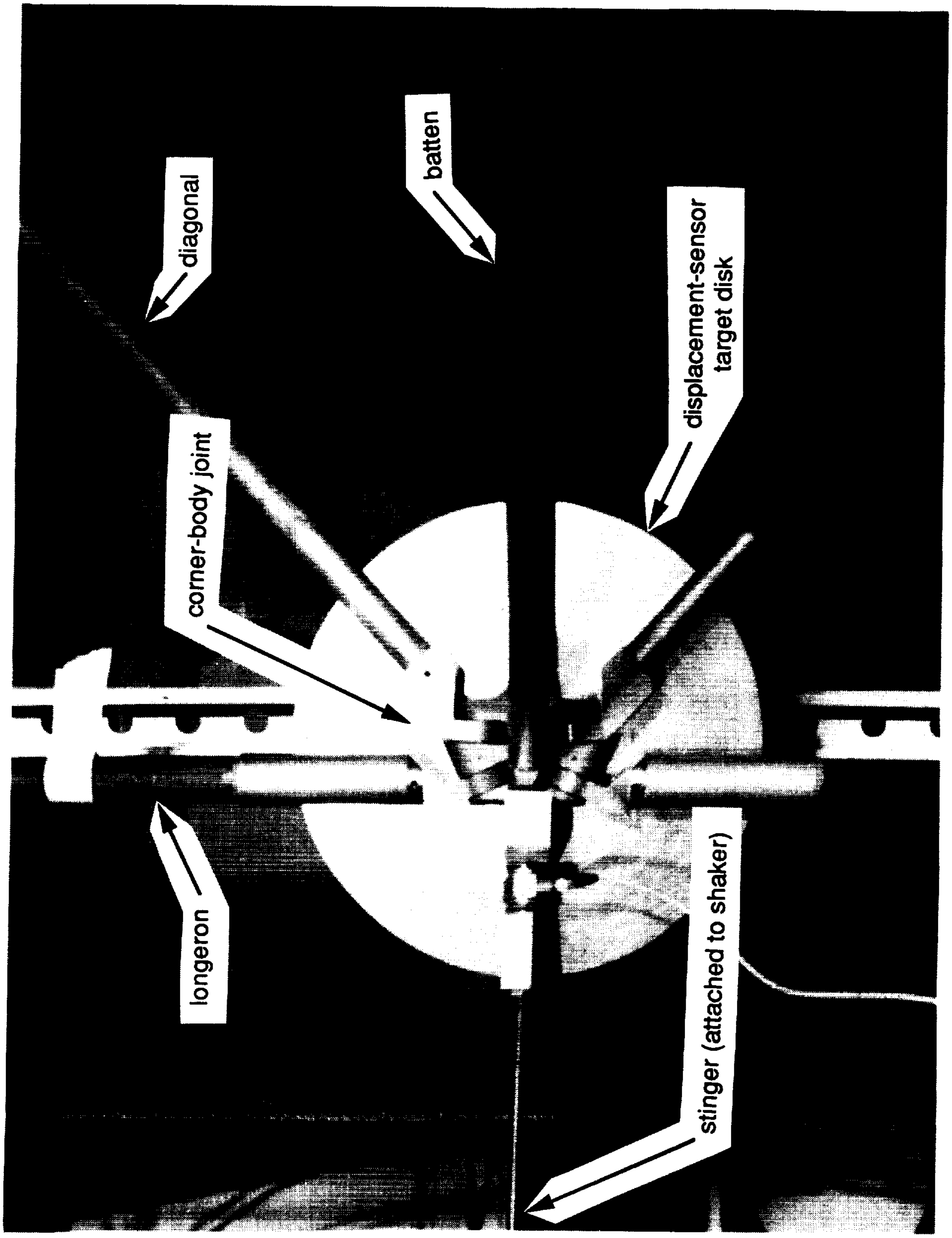

Fig. 3 Mini-Mast corner-body joint

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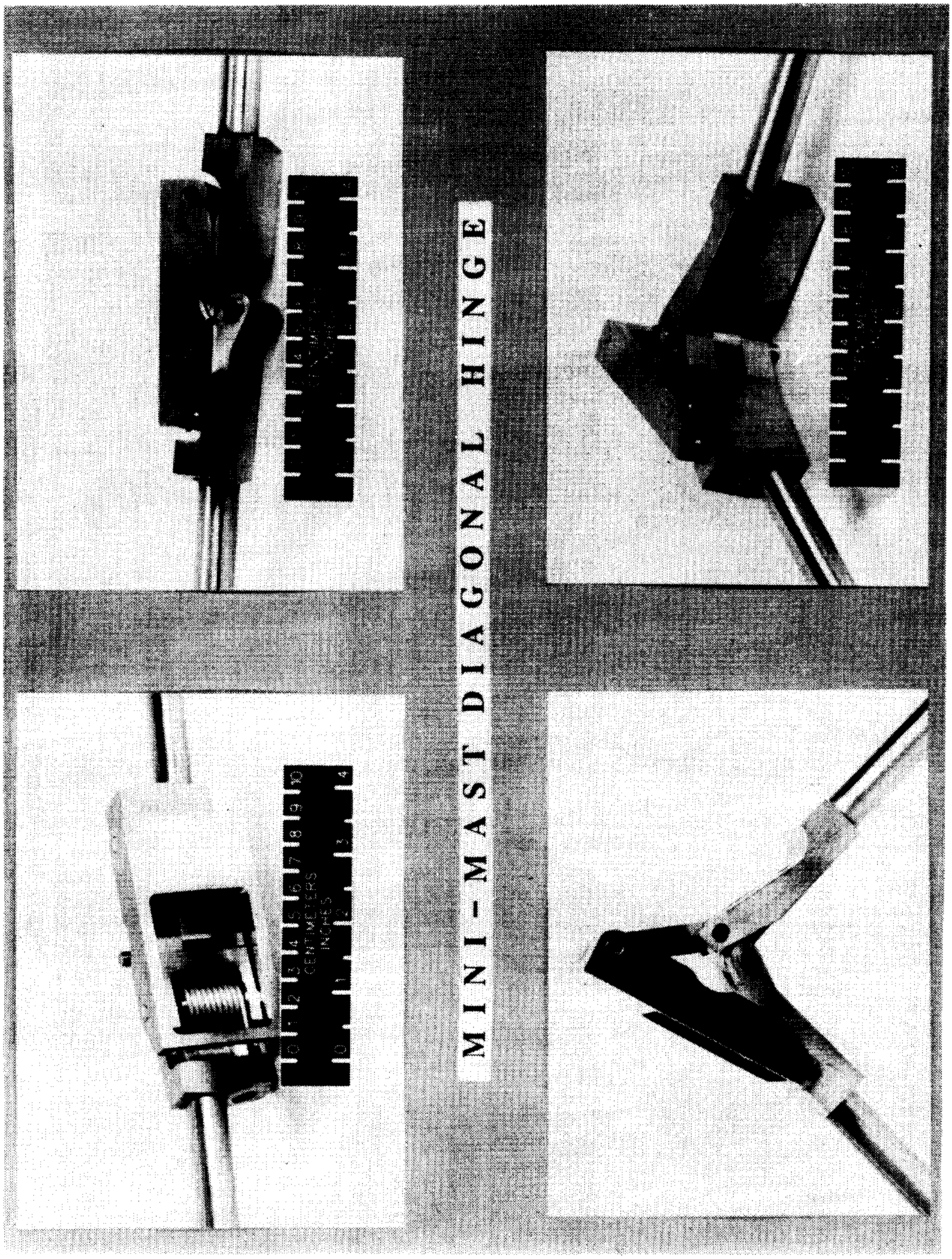

Fig. 4 Mini-Mast mid-diagonal hinge showing both locked and intermediate positions

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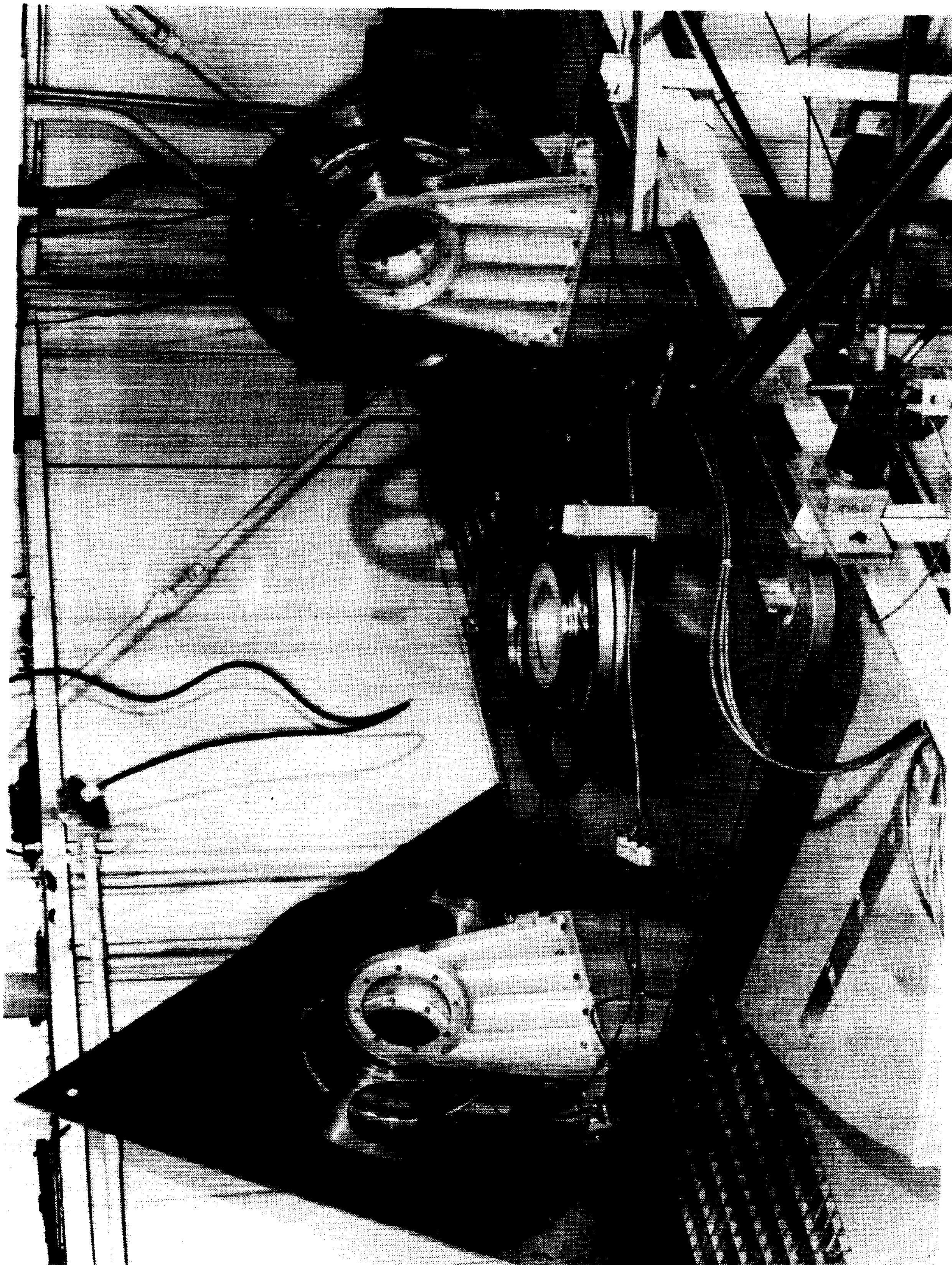


Fig. 5 Mini-Mast tip plate with torque-wheel actuators, shown prior to being mounted to truss structure

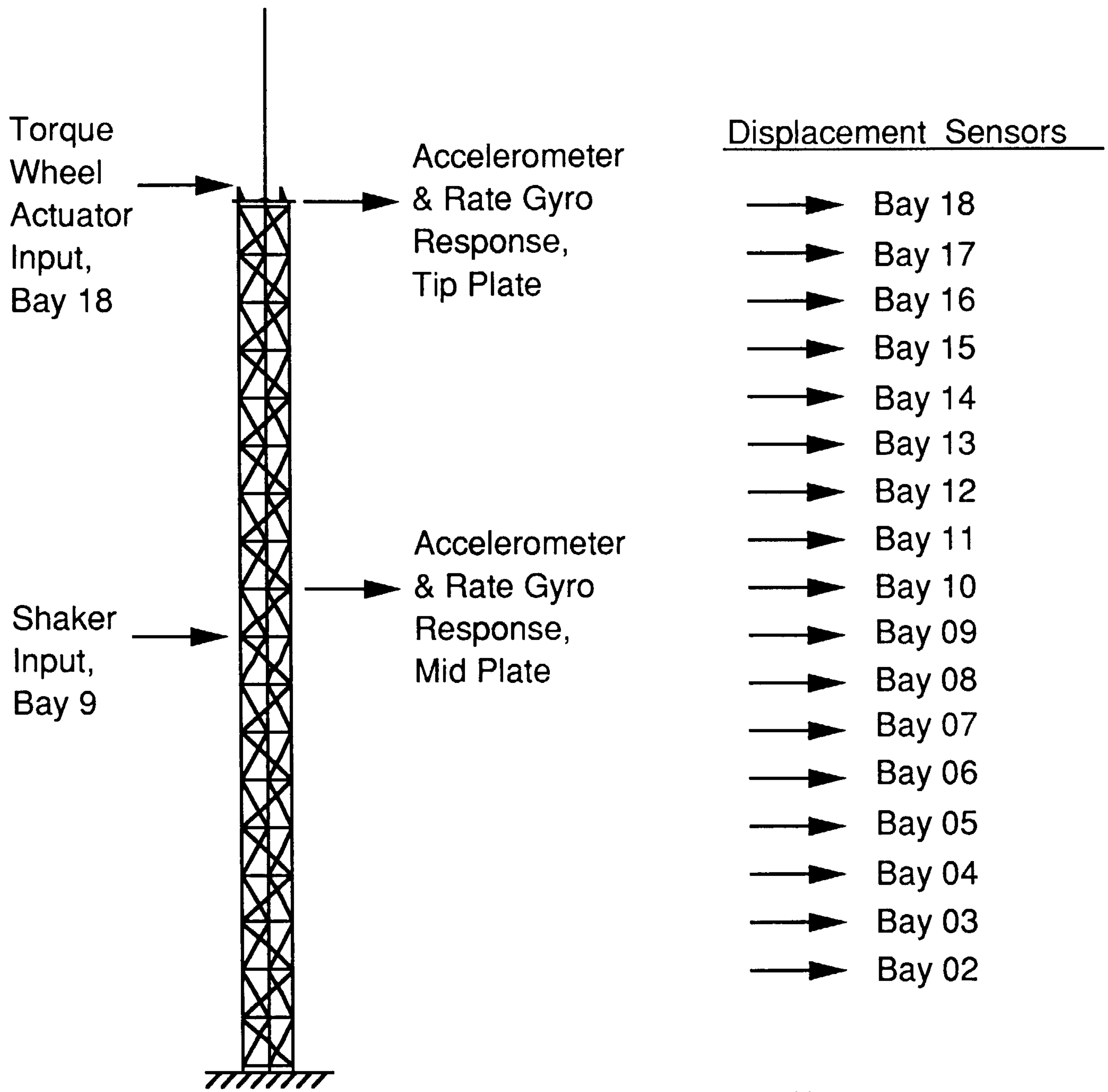


Fig. 6 Sensor and actuator positions

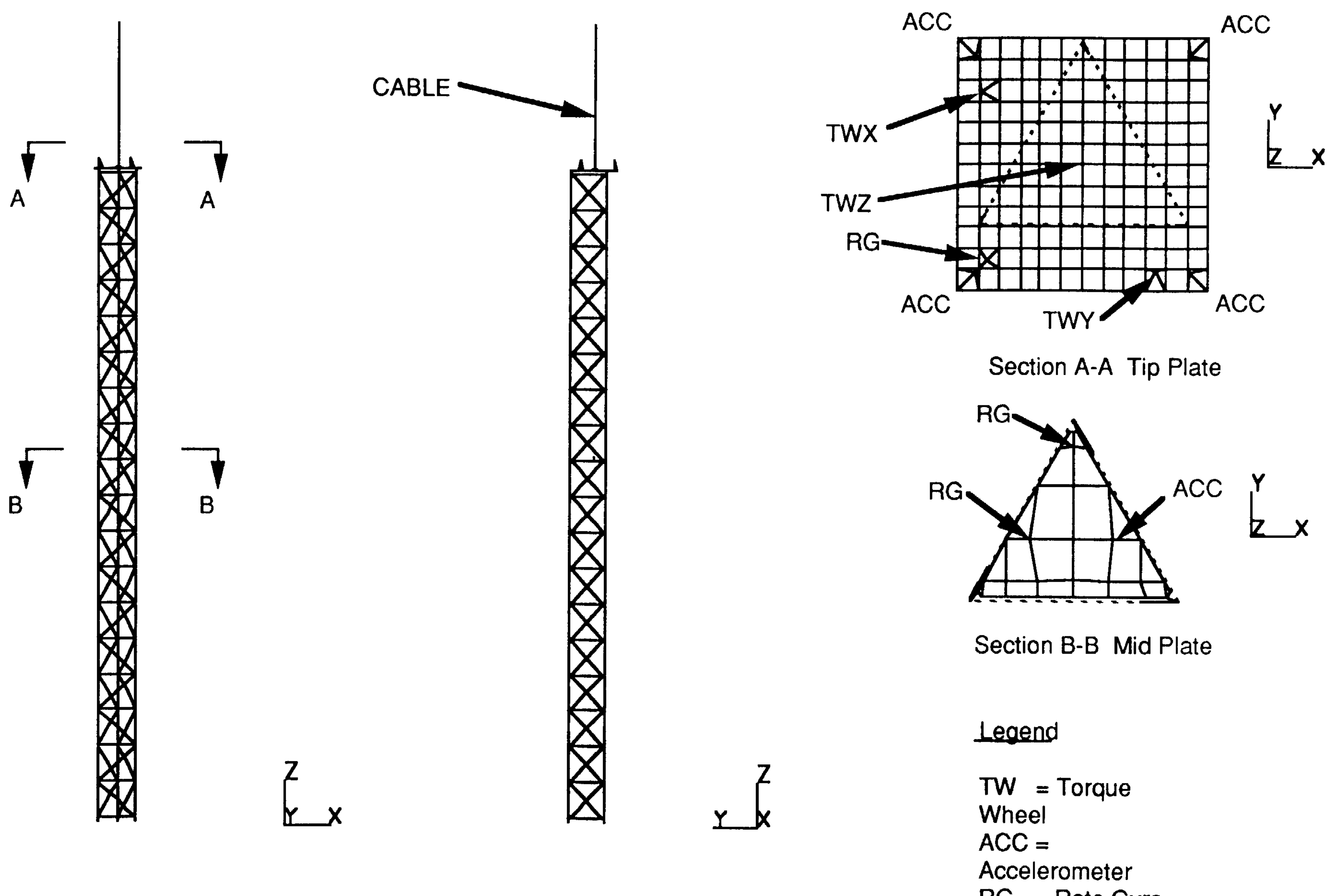


Fig. 7 Finite element model of Mini-Mast

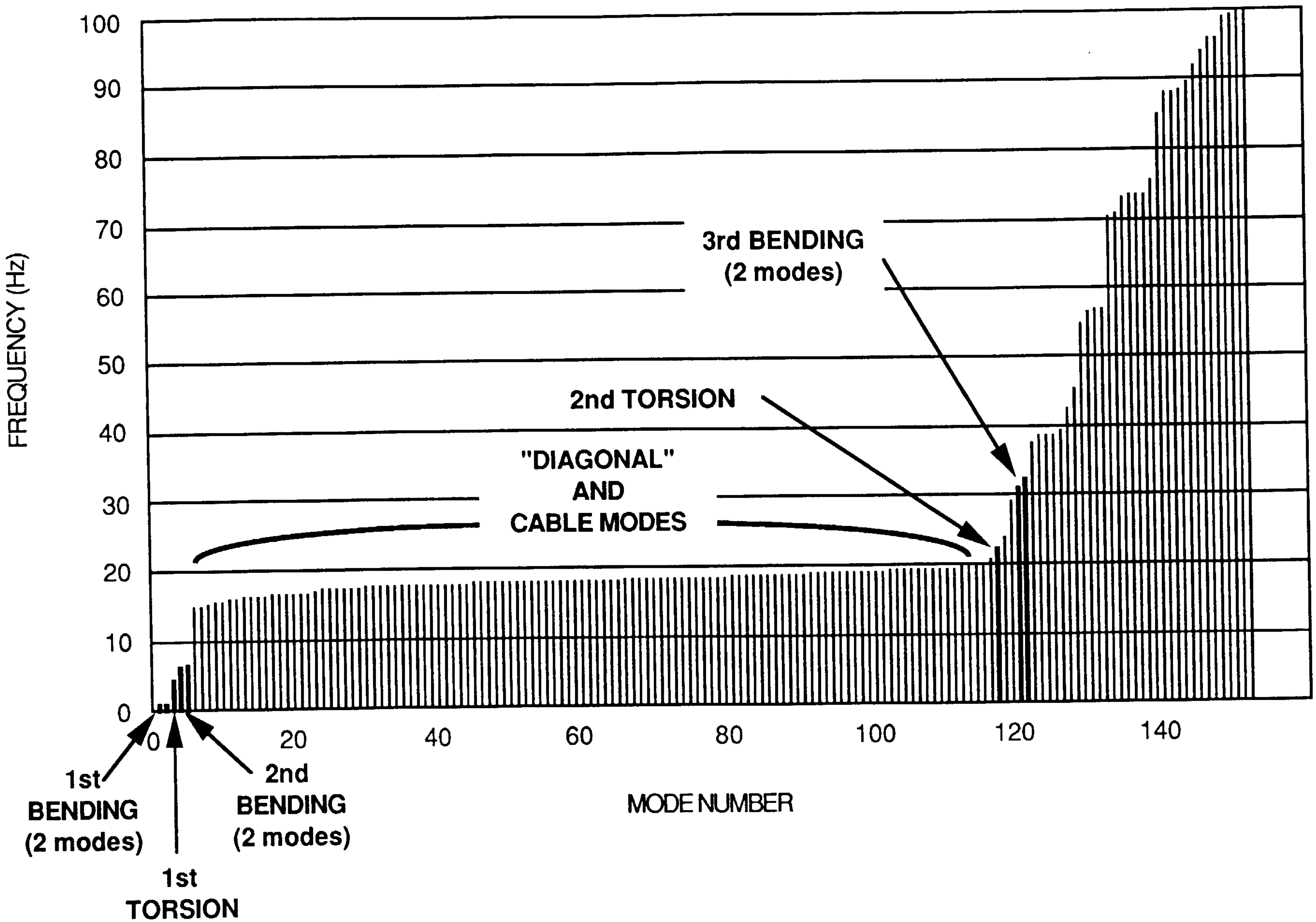
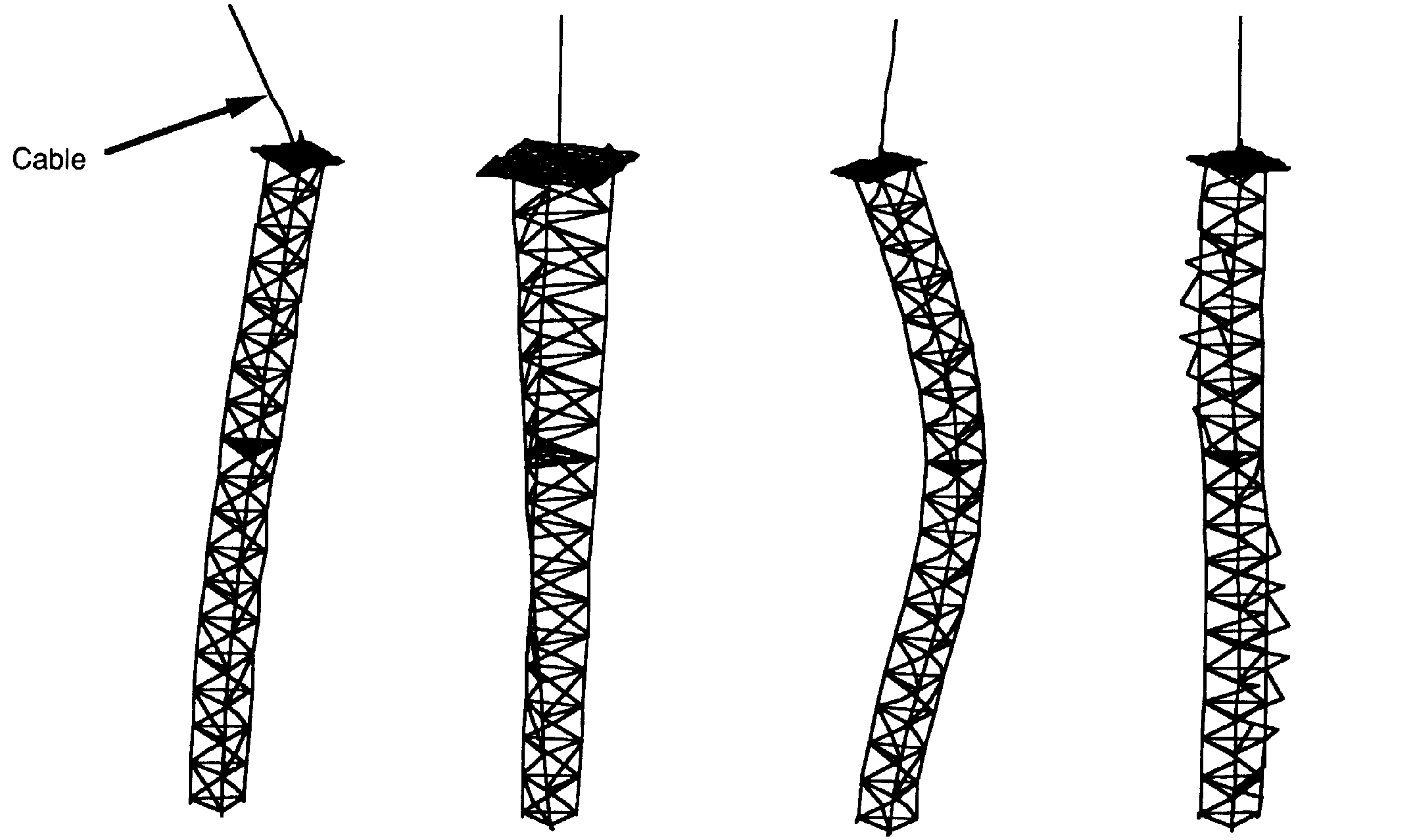


Fig. 8 Undamped natural frequencies of Mini-Mast finite element model



	1st Bending	1st Torsion	2nd Bending	Typical "Diagonal" Mode
Analytical	0.80 Hz; 0.81 Hz	4.37 Hz	6.11 Hz; 6.16 Hz	14.6 - 19.7 Hz
Experimental	0.84 Hz	4.18 Hz	6.08 Hz	-----
% Difference	-4.76	4.55	0.49; 1.32	-----

Fig. 9 Analytical mode shapes and frequencies

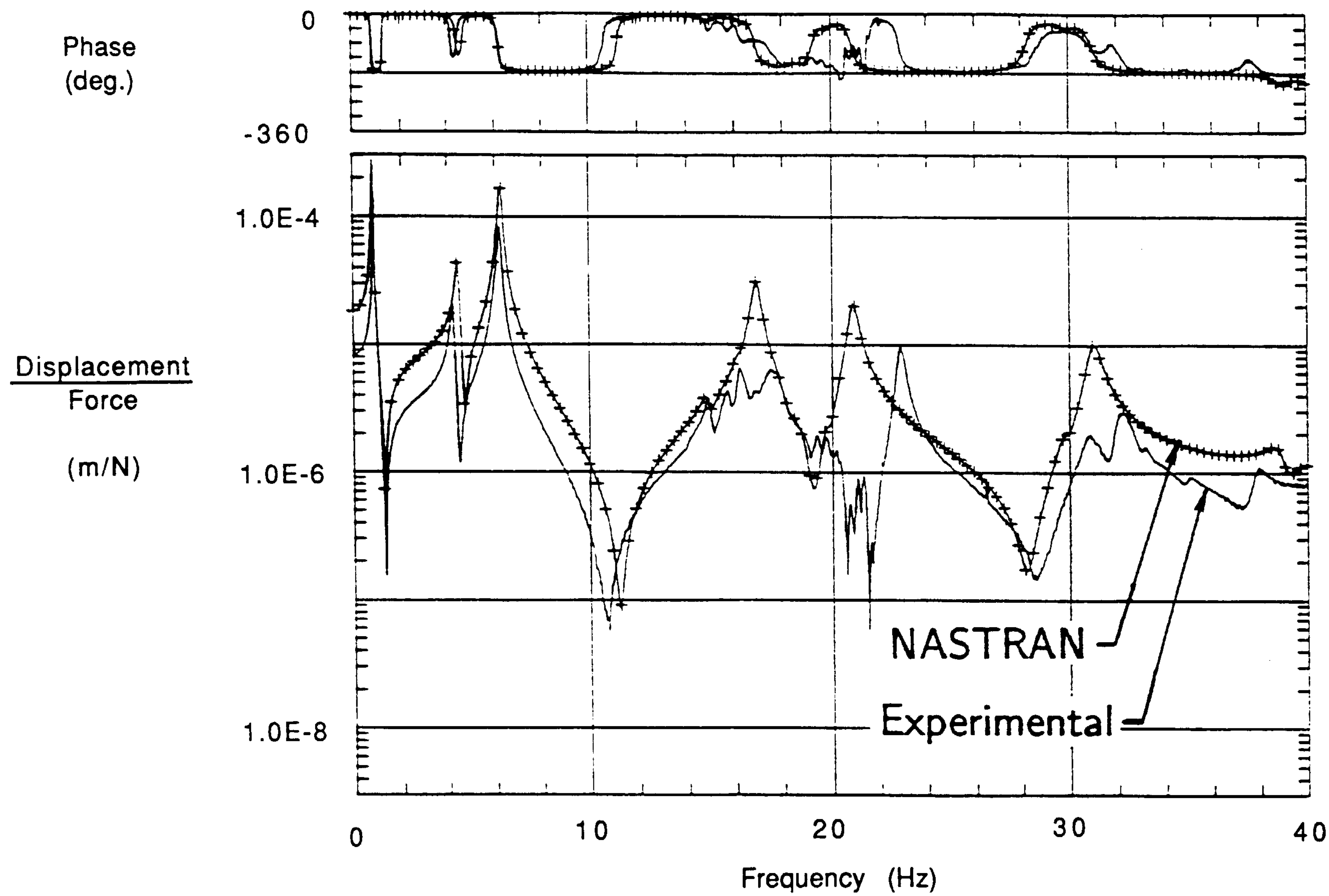


Fig. 10 Frequency response function of Mini-Mast from input at Bay 9 to output at Bay 6

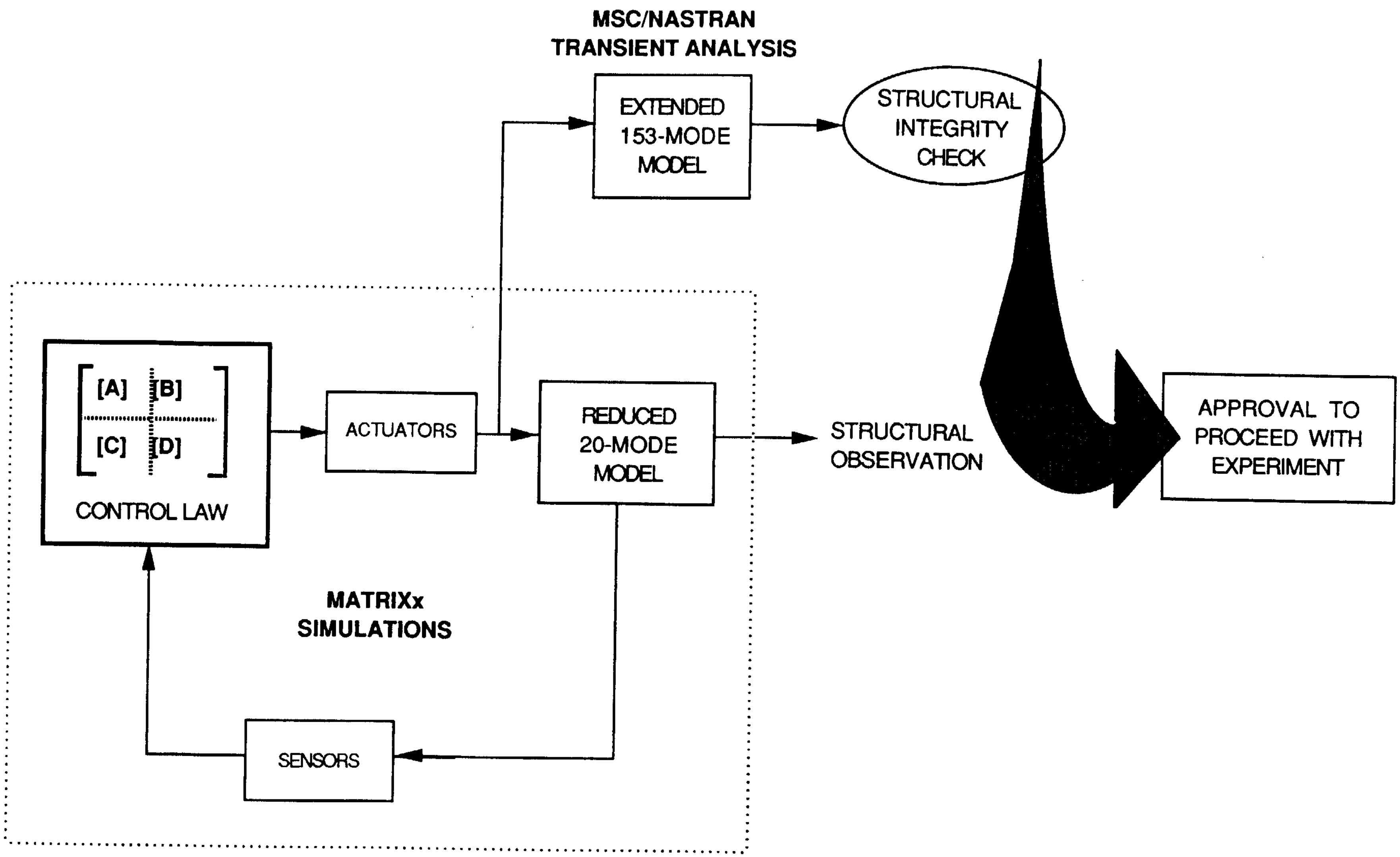


Fig. 11 Simulation and safety check process

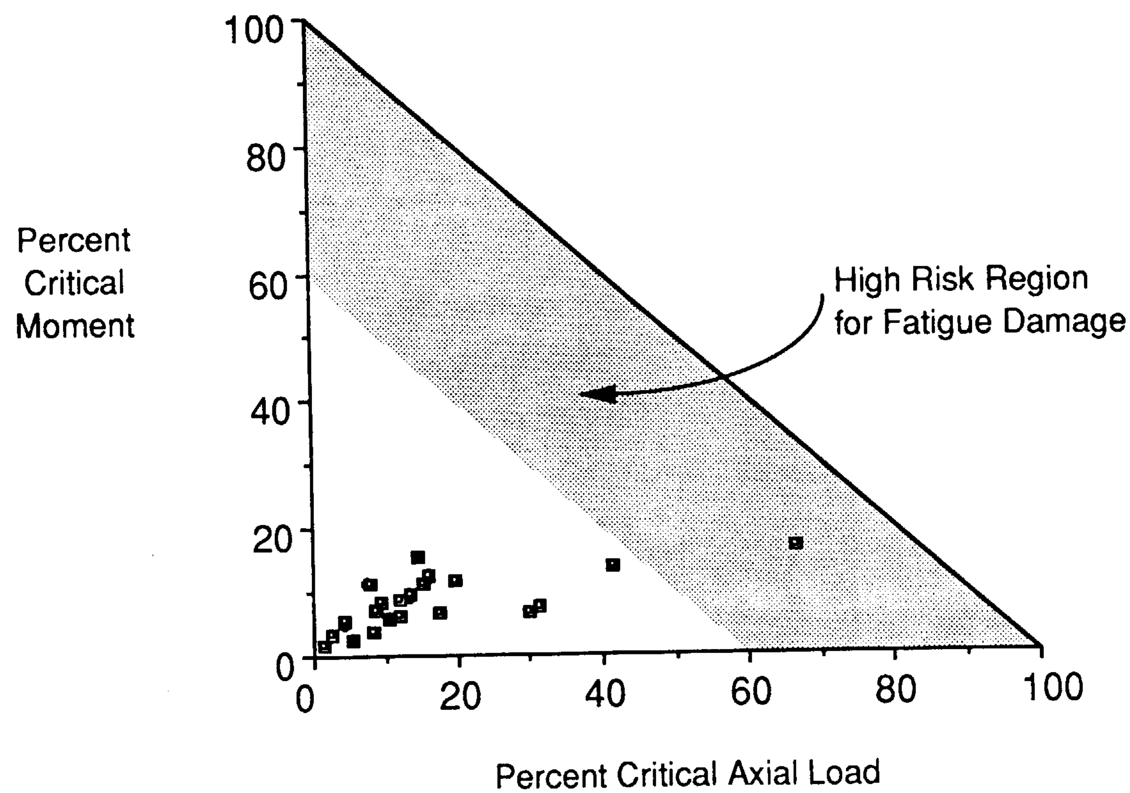


Fig. 12 Structural integrity check

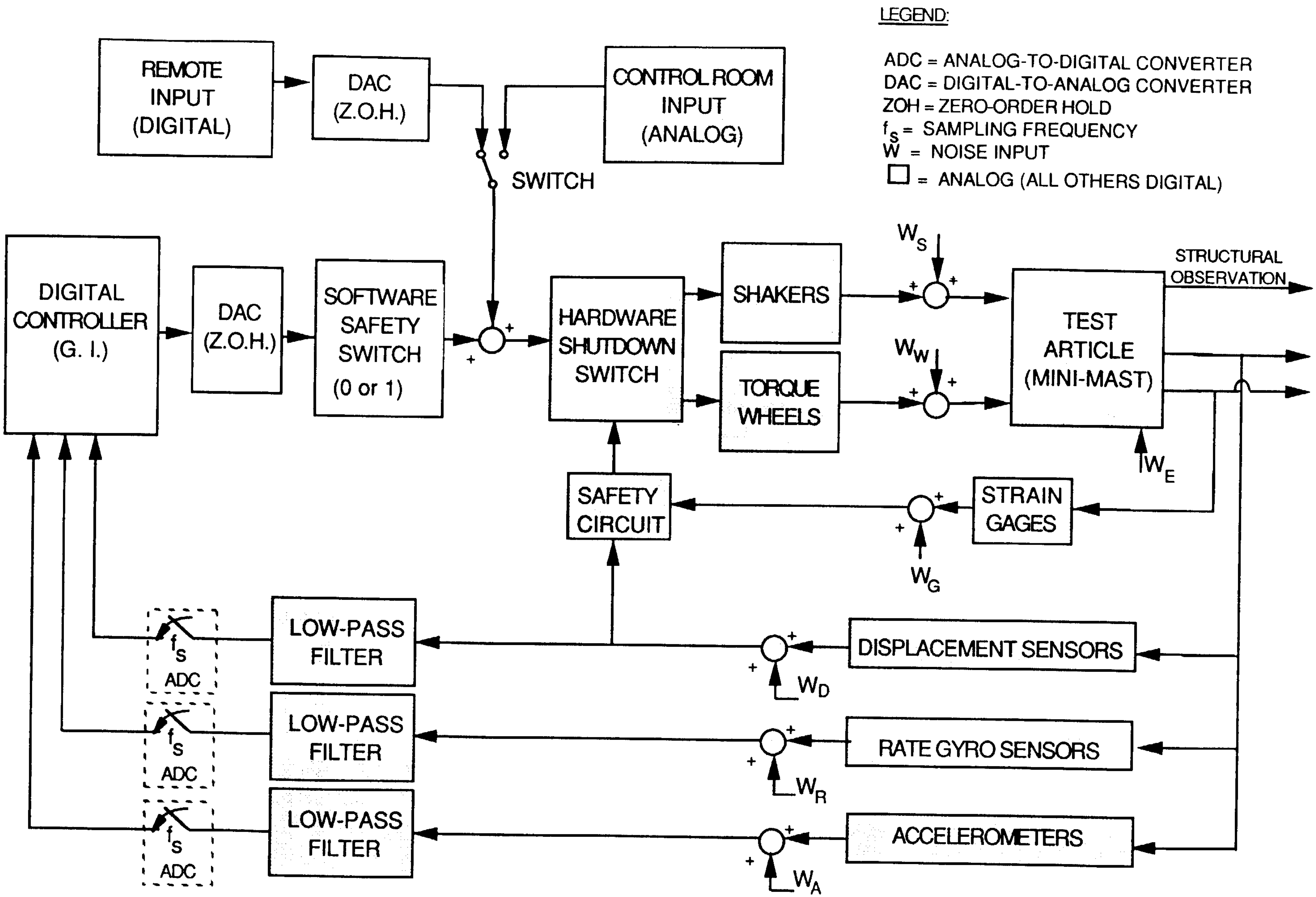


Fig. 13 Mini-Mast test configuration

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15. Supplementary Notes

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16. Abstract

MSC/NASTRAN is being used in the Controls-Structures Interaction (CSI) program at NASA Langley Research Center as a key analytical tool for structural analysis as well as the basis for control law development, closed-loop performance evaluation, and system safety checks. Guest investigators from academia and industry are performing dynamics and control experiments on a flight-like deployable space truss called Mini-Mast to determine the effectiveness of various active-vibration control laws. MSC/NASTRAN was used to calculate natural frequencies and mode shapes below 100 Hz to describe the dynamics of the 20-meter-long lightweight Mini-Mast structure. Gravitational effects contribute significantly to structural stiffness and are accounted for through a two-phase solution in which the differential stiffness matrix is calculated and then used in the eigensolution. Reduced modal models are extracted for control-law design and evaluation of closed-loop system performance. Predicted actuator forces from controls simulations are then applied to the extended model to predict member loads and stresses. These pre-test analyses reduce risks associated with the structural integrity of the test article, which is a major concern in closed-loop control experiments due to potential instabilities.

17. Key Words (Suggested by Author(s))		18. Distribution Statement
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Structures	Analysis	
Interaction	Finite Element	
Dynamic	Space	Subject Category 39
Test	Simulation	

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