N87-16020

PACOSS PROGRAM OVERVIEW AND STATUS

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NASA/DOD CSI TECHNOLOGY CONFERENCE
NOVEMBER 18-21, 1986

INTRODUCTION

Many future civilian and military large space structures (LSS) will have as performance objectives stringent pointing accuracies, short settling times, relatively fast response requirements, or combinations thereof. Many of these structures will be large, light weight, and will exhibit high structural modal density at low frequency and within the control bandwidth. The attainment of the performance objectives will be a challenge to the controls engineer.

Although it is possible in principle to achieve structural vibration control through purely active means, experience with complex structures has shown that the realities of plant model inaccuracies and sensor/actuator dynamics frequently combine to produce substandard performance.

A more desirable approach is to apply passive damping technology to reduce the active control burden. Development of the technology to apply this strategy is the objective of the PACOSS (Passive and Active Control Of Space Structures) program (Figure 1).

O FUTURE LARGE SPACE STRUCTURES (LSS) WILL REQUIRE STRUCTURAL VIBRATION CONTROL TO ACHIEVE PERFORMANCE GOALS

O VIBRATION DAMPING MAY BE ACHIEVED BY PASSIVE OR ACTIVE MEANS, OR BOTH

O MAJOR GOALS OF PACOSS PROGRAM

- DEMONSTRATE ROLES OF PASSIVE AND ACTIVE CONTROL FOR FUTURE LSS
- DEVELOP MEANS OF PASSIVE VIBRATION CONTROL
- EXPERIMENTALLY VERIFY DAMPING PREDICTIONS AND CONTROL ALGORITHMS

Figure 1

OUTLINE

A key element in the PACOSS program is the Representative System Article (RSA). The RSA is a generic "paper" system that serves as a testbed for damping and controls studies. It also serves as a basis for design of the smaller Dynamic Test Article (DTA), a hardware testbed for the laboratory validation of analysis and design practices developed under PACOSS. These topics will be discussed in greater detail (Figure 2).

- O PACOSS REPRESENTATIVE SYSTEM ARTICLE (RSA)
- O PASSIVE/ACTIVE CONTROL STUDY
- O PACOSS DYNAMIC TEST ARTICLE (DTA)
- O DTA TEST PLAN AND STATUS
- O CONCLUSIONS

Figure 2

RSA PURPOSE

The PACOSS program is generic in nature in that the damping technology being developed applies to a broad spectrum of future LSS. The RSA must contain the dynamic characteristics to be found in future systems, including the dense modal spectrum. It should contain substructures found in future LSS concepts to permit direct application of damping treatments and devices developed under PACOSS to real future systems.

The RSA also serves as the link between the DTA, the hardware test article, and future LSS. Concepts tested and validated under 1-g conditions can be evaluated under on-orbit conditions by applying proven designs to the analytic RSA model (Figure 3).

- O TRACEABLE TO REAL FUTURE SPACE SYSTEMS
  - CONTAINS SUBSTRUCTURES FOUND ON FUTURE LSS
  - CONTAINS HIGH MODAL DENSITY AND ASSOCIATED DYNAMICS PROBLEMS OF FUTURE LSS
- O GENERIC STRUCTURE FOR ANALYTIC DEMONSTRATION OF VARIOUS CONTROL APPROACHES
- O SERVES AS A LINK BETWEEN FUTURE LSS AND THE DYNAMIC TEST ARTICLE (HARDWARE)

Figure 3

REPRESENTATIVE SYSTEM ARTICLE (RSA) DESIGN

The utility of the RSA concept is in direct proportion to the number of future systems that it represents. To obtain a broad base for the design, the Military Space Systems Technology Model (MSSTM) and the corresponding NASA document (NSSTM) were examined to determine which structures and missions would benefit from passive damping technology (Reference 1). In all, it was determined that 13 future military systems and six future NASA systems would benefit from the application of passive augmentation (Figure 4).

- O REVIEW MILITARY SPACE SYSTEMS TECHNOLOGY MODEL AND NASA EQUIVALENT
- O DETERMINE SYSTEMS REQUIRING STRUCTURAL VIBRATION CONTROL
- O DETERMINE WHICH DYNAMIC CHARACTERISTICS ARE GENERIC AND IMPORTANT FOR FUTURE LSS
- O INCORPORATE THESE DYNAMIC CHARACTERISTICS INTO RSA DESIGN

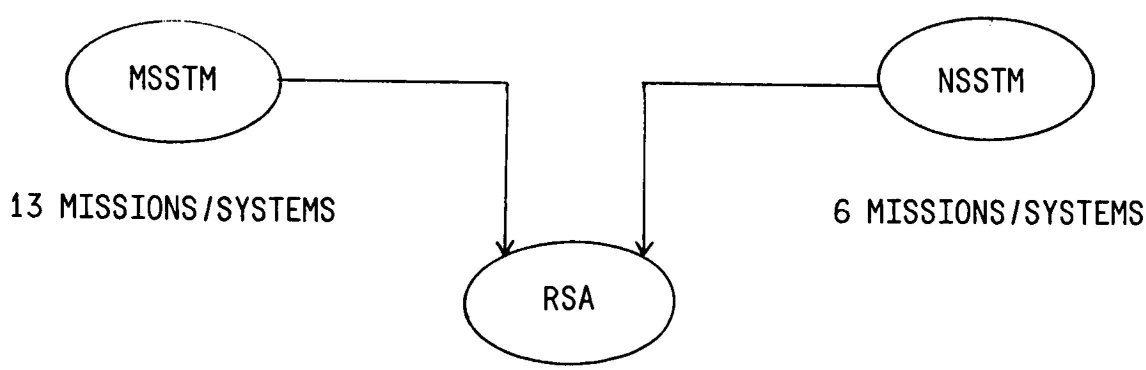

Figure 4

RSA COMPONENT SUMMARY

The substructures comprising the RSA and their respective sizes are shown in Figure 5, as well as some of the applicable future systems. It should be noted that the ring truss is the "hardback" for the system, and as such was not considered for passive damping treatments.

Each of the other components is a candidate for passive damping treatments and devices. The damping design is done at the component or substructure level, and thus is applicable to the parent real system. Naturally, the damping achieved in the overall system modes is determined by the damping in the substructure modes as well as the degree of participation of the substructure modes in the system modes.

COMPONENT	DIMENSIONS, M	APPLICABLE SYSTEMS
BOX TRUSS	20 x 20 x 2.5	SPACE-BASED RADAR LARGE EARTH OBSERVING SYSTEM SPACE STATION
RING TRUSS	DIA: 22.4	GENERIC TRUSS STRUCTURE
TRIPOD	DIA AT BASE: 20 HEIGHT: 20	SPACE-BASED LASER LARGE DEPLOYABLE DEFLECTOR
EQUIPMENT PLATFORM	LENGTH: 10	SPACE STATION
ANTENNA	DIA: 5	SPACE-BASED RADAR SPACE STATION
SOLAR ARRAYS	LENGTH: 20	SPACE-BASED RADAR SPACE STATION

Figure 5

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REPRESENTATIVE SYSTEM ARTICLE (RSA)

Figure 6 is an artist's concept of the on-orbit RSA. All seven substructures are shown. The overall largest dimension is approximately 60 meters.

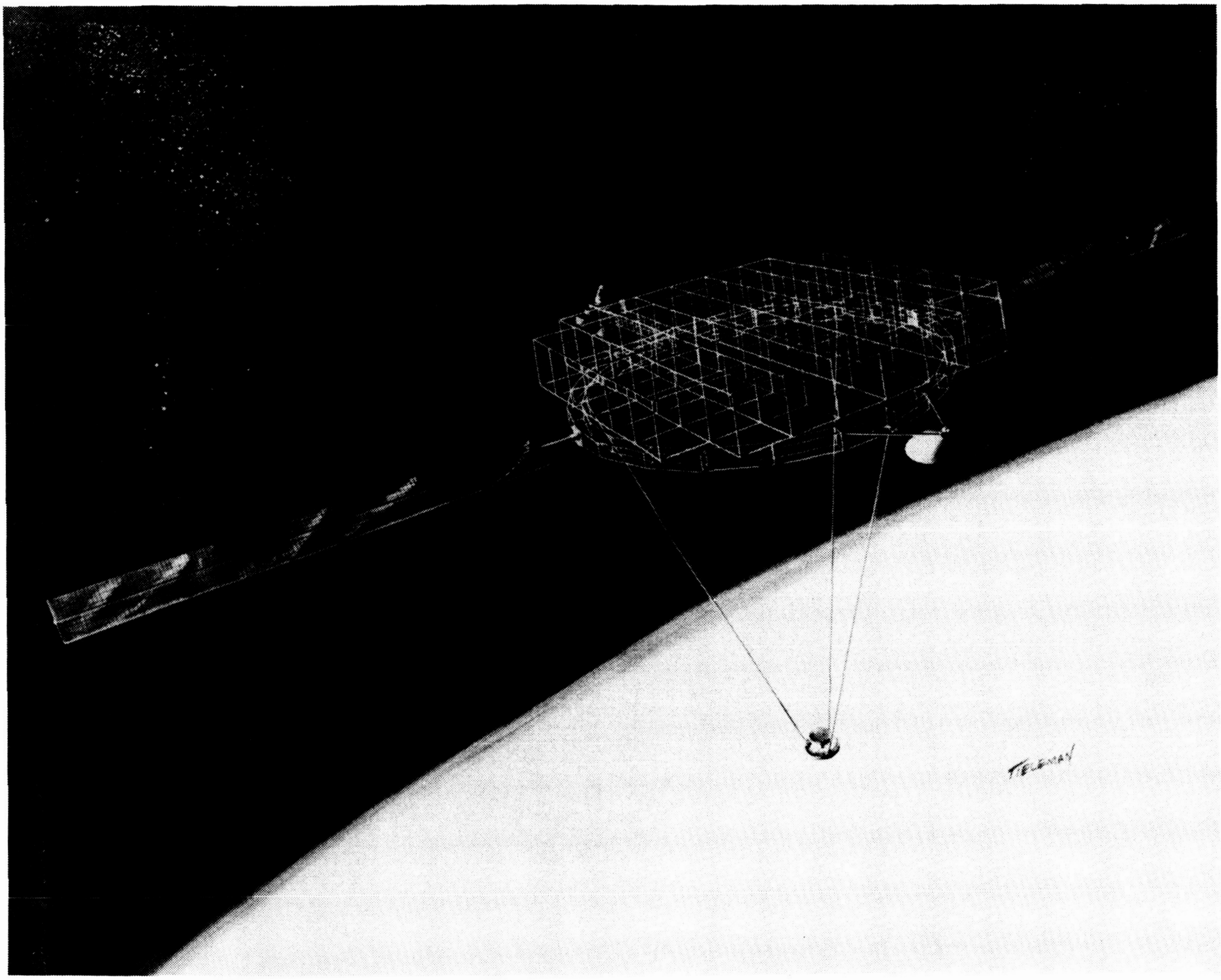


Figure 6

COUPLED SYSTEM ANALYSIS

The system normal modes were calculated with the Craig-Bampton technique. Substructure normal modes below 15 Hz and all constraint modes were used in the synthesis.

The results of the coupling produced in excess of 200 modes below 10 Hz. Many of the modes are local in nature, but there are still 34 global modes with significant modal strain energy contained in two or more substructures (Figure 7).

- O CRAIG-BAMPTON COUPLING (FREQUENCY CUTOFF 15 Hz)
- O 210 SYSTEM MODES BELOW 10 Hz
- O 34 "GLOBAL" SYSTEM MODES

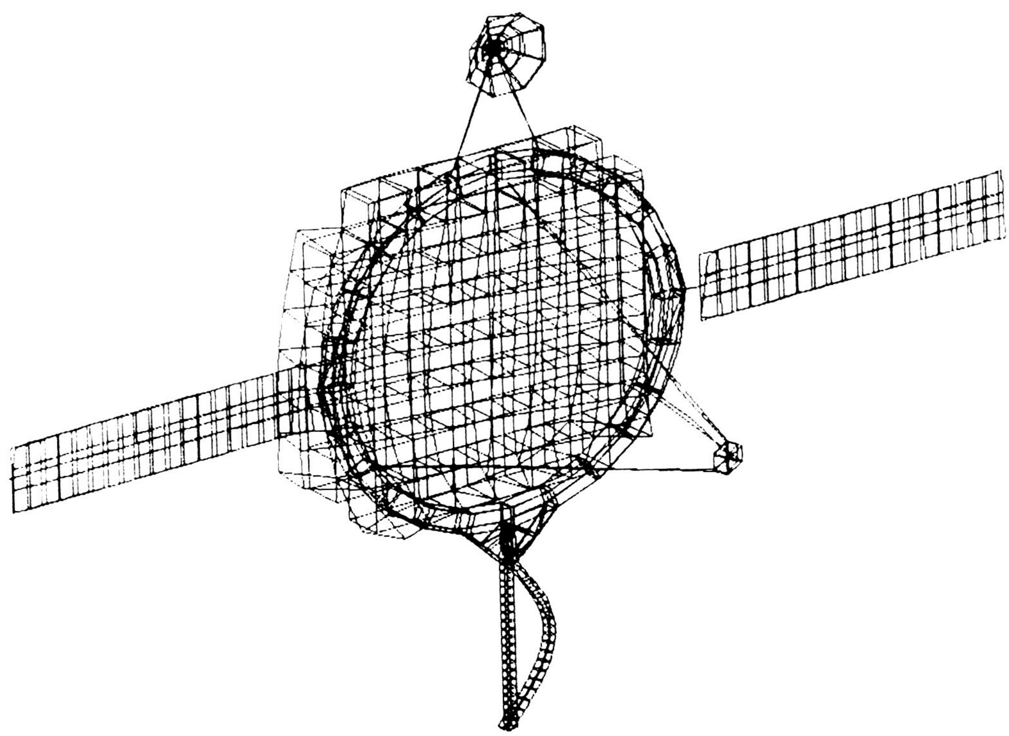


Figure 7

CONTROL SYSTEM/PASSIVE DAMPING STUDY

To demonstrate the benefits of passive damping on RSA performance and to select target levels for passive damping design, a simple control study was undertaken (Figure 8 and Reference 2).

The controlled variables were system line of sight (LOS), which will be defined on a later chart, and attitude. A 0.01 radian slew maneuver was commanded, and the system was considered to have settled when LOS excursions remained in a 100-microradian (zero-peak) band.

The RSA is not a real system, and selection of the desired settling time is somewhat arbitrary. It was decided to select rigid-body control torque levels producing reasonable settling times of 3.25 seconds for the pitch (about the solar arrays) axis and 5.0 seconds about the yaw (perpendicular to pitch and LOS) axis for the rigid RSA. These times then became performance requirements for the flexible system.

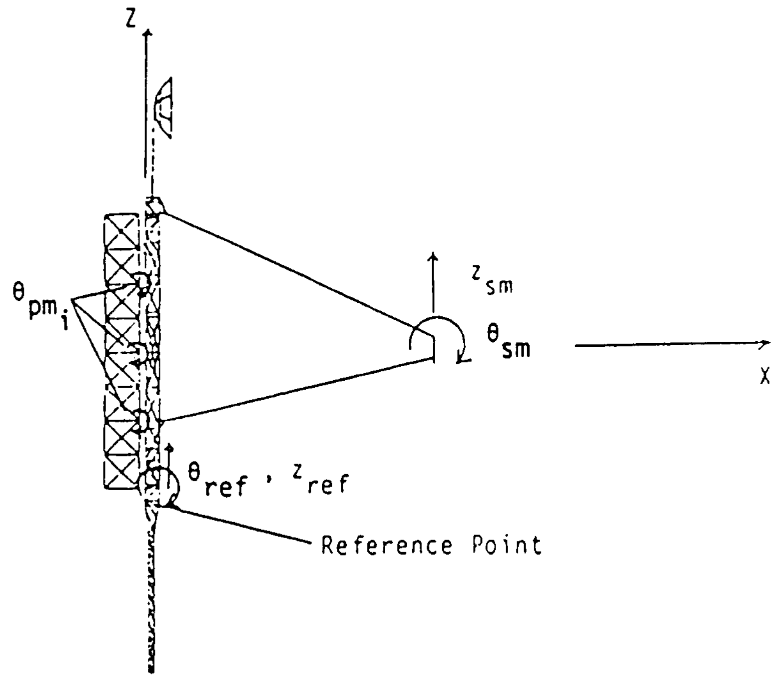
Various realizable levels of passive damping were designed into the system, and active modal control was utilized as required to achieve the performance goals. Total control energy for the maneuver was calculated for each passive/active combination.

- O CONTROL LINE OF SIGHT DURING SLEW
- O SELECT DESIRED PERFORMANCE FROM RIGID-BODY RESPONSE
- O FOR VARIOUS LEVELS OF REALIZABLE PASSIVE DAMPING, DETERMINE CONTROL ENERGY REQUIRED TO ACHIEVE DESIRED PERFORMANCE
- O USE ACTIVE MODAL CONTROL ONLY AS REQUIRED

Figure 8

LINE-OF-SIGHT DEFINITION

Figure 9 defines the LOS as used in this study. Note that the LOS response due to primary reflecting surface deformation is approximated by an average of selected rotations on the box truss.



$$LOS_y = \left(2\frac{f_s}{f_p} - 1\right)\theta_{ref} + \frac{2}{13}\sum_{i=1}^{13}\theta_{pm_i} - 2\frac{f_s}{f_p}\theta_{sm} - \frac{1}{f_p}(z_{ref} - z_{sm})_{flex}$$

Where

f_s = Focal length of secondary mirror

f_p = Focal length of primary mirror

$(z_{ref} - z_{sm})_{flex}$ = Relative z deflection of secondary mirror to reference point due to structural deformation

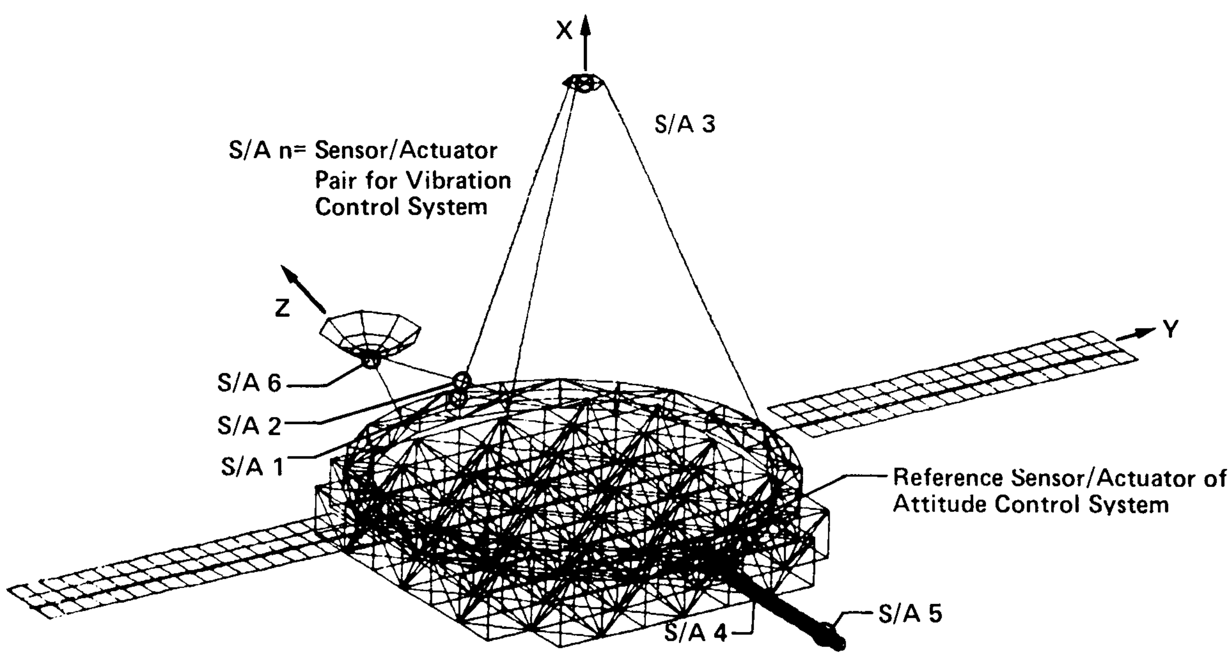
Figure 9

SENSOR/ACTUATOR LOCATIONS (PITCH DYNAMICS MODEL)

Figure 10 shows the sensor/actuator locations and the selection criterion for their location. Symmetry of the structure results in pitch and yaw dynamics being uncoupled. The ten symmetric modes with the highest gain into LOS were retained for the pitch study, and the nine antisymmetric modes with highest gain into LOS were retained for the yaw-axis simulation.

A nominal modal damping level of 0.2% viscous was assumed to represent the untreated structure. It should be noted that, without additional active augmentation, this system exhibited a mild instability due to the presence of a low pass filter in the attitude control system, which had a 1-Hz cutoff frequency.

In the pitch case, it was determined that for low levels of passive damping, modal control was required for six modes. As the amount of passive augmentation was increased, the number of modes requiring active augmentation was reduced to two.



- 0 IDEAL ANGULAR RATE SENSORS AND TORQUE WHEELS
- 0 LOCATIONS SELECTED TO GIVE $(\phi_{REL})_C$ THE LARGEST DETERMINANT
- 0 6 S/A PAIRS REQUIRED FOR ACTIVE DAMPING ALONE
- 0 2 S/A PAIRS USED FOR PASSIVE + ACTIVE DAMPING ABOVE CERTAIN LEVELS

Figure 10

LOS RESPONSE TO 0.01 rad SLEW

When designing damping treatments, the analyst utilizes the modal strain energy (MSE) method to design a damping treatment to produce the desired amount of damping in a target mode. There is "damping spillover" into other modes, wherein non-target modes also receive some damping. Thus, each mode will receive in general a different level of damping.

Figure 11 shows the LOS response for different levels of average modal damping for the modes included in the simulation. Note that, for low levels of average passive damping, six-mode active control was used. Higher levels required active control of only two modes. It is obvious that the settling times for all systems are virtually identical.

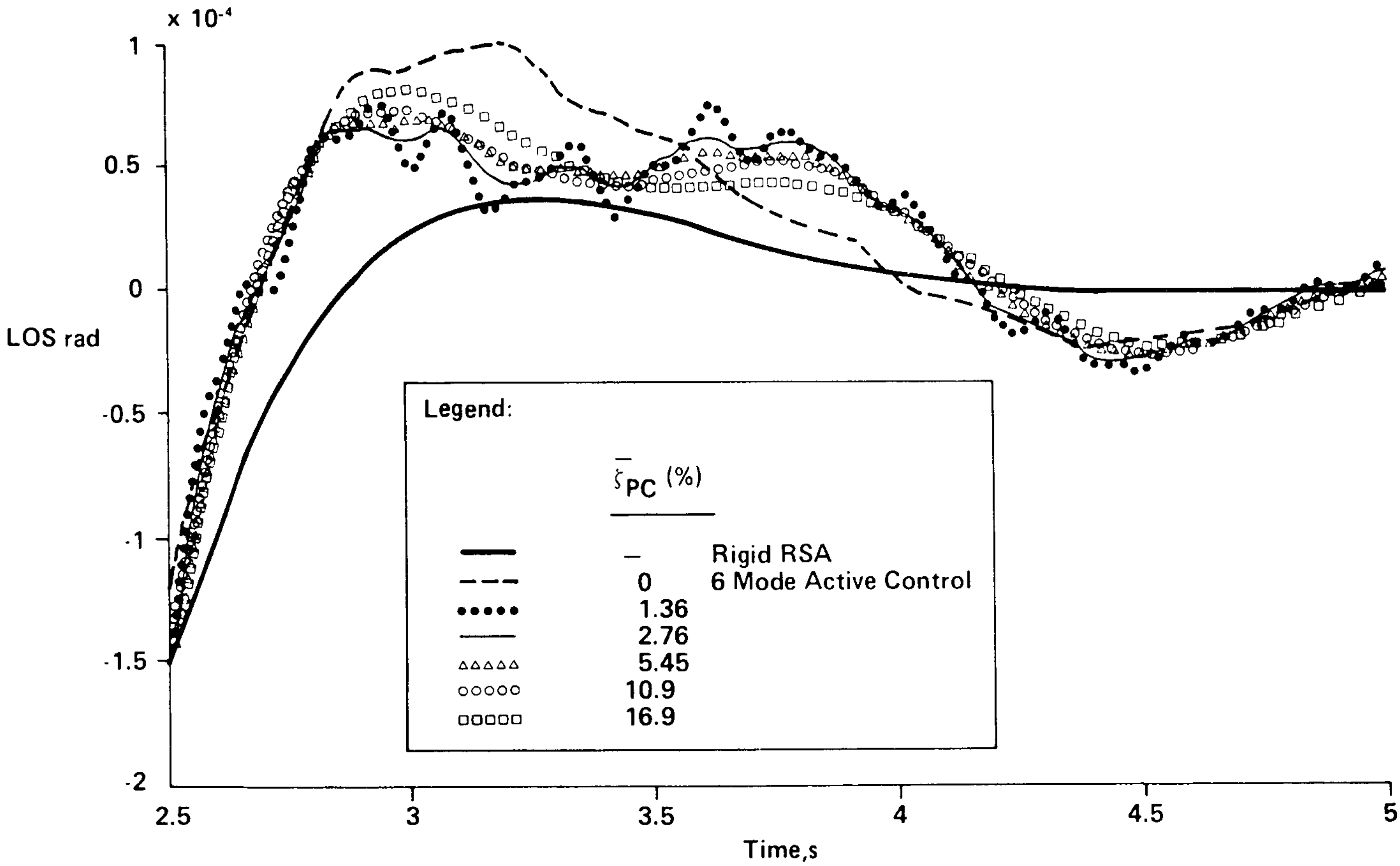


Figure 11

The control energy requirements for the pitch-axis simulation are shown in Figure 12, where the energy has been normalized to the value required for the 0.2% nominally damped (untreated) system.

Two horizontal scales have been furnished. The lower scale is the average level of passive damping for all modes in the system. This scale is of interest for low levels of passive damping in which active augmentation is required for six modes.

The upper scale is the average level of passive damping for the first two modes. Above the level of passive damping for which two-mode active control is sufficient, the response is dominated by these modes, and increasing the damping level on the passively controlled modes produces little effect on system performance.

The PACOSS DTA design goal was selected to be an average damping value of 5%. At this level, the RSA requires only one third of the control energy of the untreated system for pitch-axis control. A similar study for the yaw axis resulted in an even greater reduction.

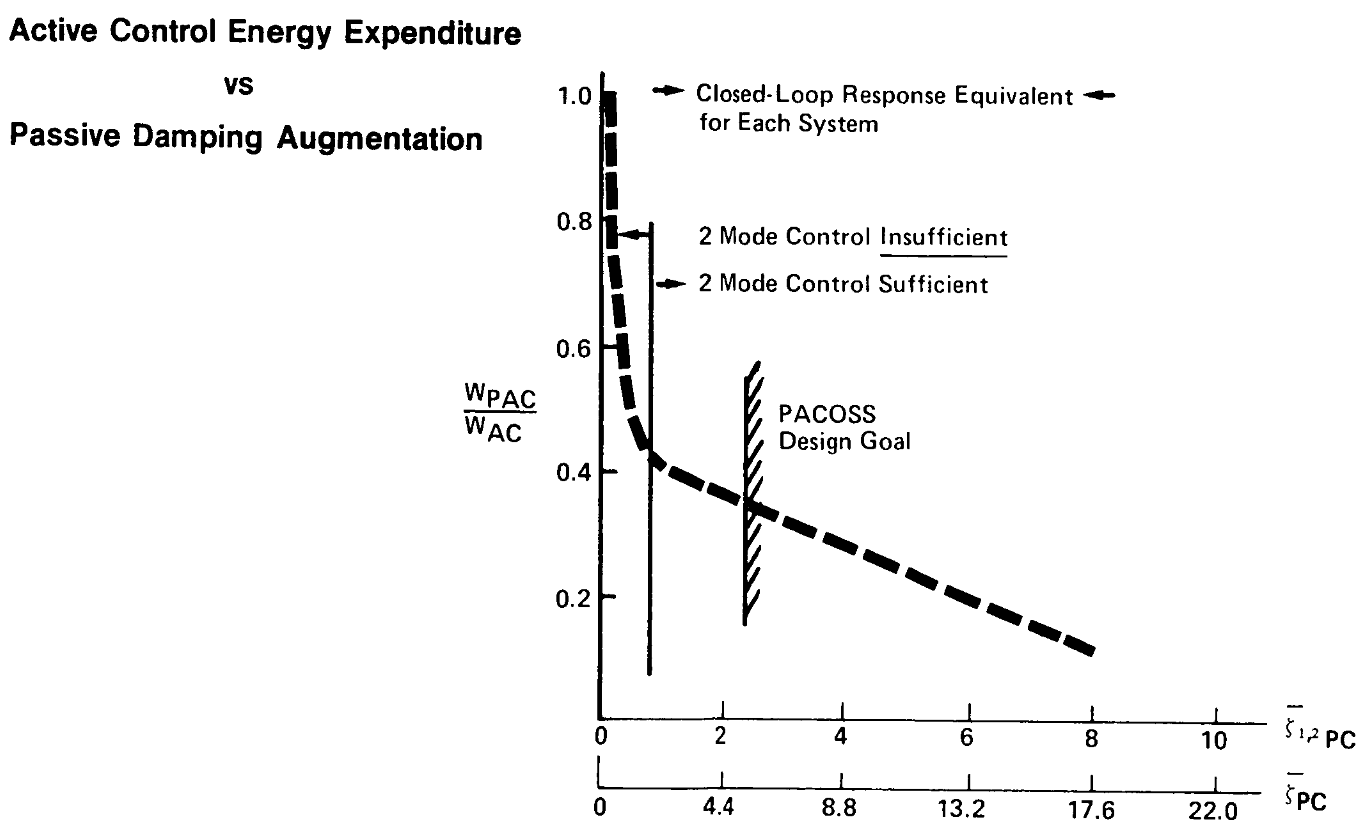


Figure 12

BENEFITS

The benefits of the passive/active control strategy on the system level are obvious (Figure 13). With reduced sensor/actuator requirements, the system becomes simpler, less expensive, and more reliable. Reduction in control system complexity results in weight savings, both for the control components as well as reduced structural strength because of the lower control torque levels. These savings, together with increased robustness and reliability will compensate for, if not totally offset, the weight of the viscoelastic damping treatments.

It is essential, however, that the structure be designed for passive damping augmentation. It is inefficient to post-treat an existing structure.

O PASSIVE + ACTIVE DAMPING GIVES DESIRED PERFORMANCE WHILE

- REDUCING NUMBER OF ACTIVE CONTROL COMPONENTS
- REDUCING ENERGY AND POWER REQUIREMENTS

O THIS CAN LEAD TO

- MORE ROBUST AND RELIABLE SYSTEMS
- LESS EXPENSIVE SYSTEMS

O FUTURE LSS SHOULD BE DESIGNED TO FACILITATE PASSIVE DAMPING TECHNOLOGY

Figure 13

DYNAMIC TEST ARTICLE (DTA) REQUIREMENTS

The DTA design was chosen to be dynamically similar to the RSA and will serve as a hardware validation of design and analysis techniques for passive damping treatments. The requirements shown in Figure 14 have been selected to assure traceability to the RSA within the realities of the test environment and budgetary constraints. In addition, a program requirement is that the DTA be space-qualifiable. Among other implications, delivery in a single Shuttle flight was assumed.

- VALIDATION OF DAMPING TREATMENT DESIGN AND ANALYSIS TECHNIQUES
- DYNAMICALLY SIMILAR TO RSA
- DELIVERABLE TO ORBIT AS SINGLE SHUTTLE PAYLOAD
- NEGLIGIBLE UNPREDICTED DAMPING
- SUITABLE FOR 1-G TEST
- EASILY AND INEXPENSIVELY FABRICATED

Figure 14

DTA DESIGN APPROACH

The DTA was designed to be essentially a scaled-down version of the RSA. Because of the 1-g test environment, it was necessary to shift the fundamental frequency upward by approximately 2 Hz. This will provide for a structure that can withstand the 1-g test environment, but will not compromise the control interaction aspects of the dynamics because the control bandwidth will be shifted accordingly.

Low-cost materials are being used to the maximum extent possible, but careful fabrication techniques are being utilized to assure that the structure is well known. In addition, the design is such that inadvertent sources of damping are avoided (Figure 15).

- ACHIEVE SIMILAR DYNAMIC CHARACTERISTICS ON COMPONENT LEVEL IN CONSTRAINED MODES
- GEOMETRIC SCALE FACTOR OF 1:7.72
- FREQUENCY SHIFT OF 2 Hz FOR DENSE BANDWIDTH
- USE SCALED-DOWN MEMBER CROSS SECTION
- ELIMINATE PIN JOINTS
- MINIMIZE AIR RESISTANCE
- ACCURATELY MODEL SUSPENSION MECHANISM AND INCLUDE EFFECTS OF GRAVITY ON COUPLED SYSTEM
- LOW-COST MATERIALS IN STANDARD SIZES
- STANDARD FABRICATION PROCESSES

Figure 15

SUMMARY OF FINAL DTA DIMENSIONS

The status of the DTA design at the time of this writing is shown in Figure 16, together with the component sizes and weights.

The ring truss has been fabricated and tested, and results are presented herein. This structure, as mentioned previously, was designed to have very low damping. Bonded joints were used extensively to simulate more expensive, precision hardware.

The tripod was designed to be heavily damped, with both constrained-layer damping treatments on the legs and novel viscoelastic rotational dampers for the secondary mirror. At the time of this writing, testing has not been completed.

COMPONENT	DIMENSION (M)	MASS (KG)
1) BOX TRUSS	2.59 X 2.59 X 0.324	180.5
*2) RING TRUSS	DIAMETER: 2.9	59.7
**3) TRIPOD	DIAMETER AT BASE: 2.59 HEIGHT: 2.59	29.9
4) EQUIPMENT PLATFORM	LENGTH: 1.295	7.04
5) ANTENNA	DIAMETER: 0.648	4.52
6,7) SOLAR ARRAYS	LENGTH: 2.59	12.0

* FABRICATED AND TESTED

** FABRICATED

Figure 16

DTA FINITE ELEMENT MODEL

The DTA finite element model is shown in Figure 17. Note that the extremely soft solar arrays have been rotated to lie in the plane of the gravity vector in the test configuration. This compromise was necessary to avoid buckling during test.

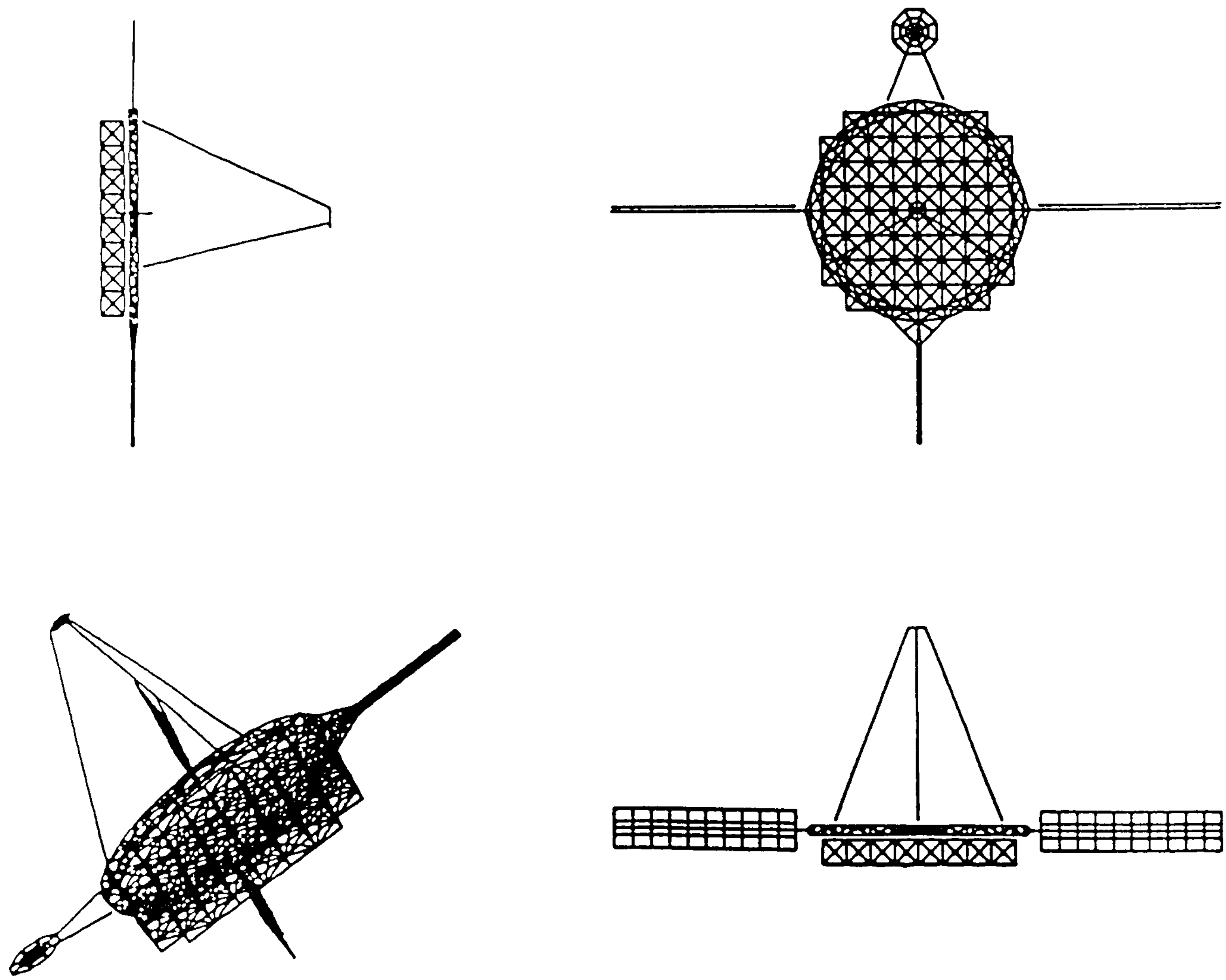


Figure 17

DTA TEST PROGRAM

An extensive test program is planned for the DTA, beginning at the component level (Figure 18). Each component will undergo a modal survey. As mentioned previously, the Craig-Bampton technique is used for system synthesis. For this reason, fixed-interface boundary conditions will be used for the appendage modal surveys.

The ring truss modal survey has been completed, and was done with the ring truss interfaces loaded with mass simulators. The ring truss response is critical to the overall system, and this approach was deemed most suitable to check the critical interface modeling.

In addition, the ring truss survey served to verify that inadvertent damping can be avoided with careful fabrication techniques. This test also served as a check on the suspension system, which will be described later.

After the component modal surveys are completed and analytic models refined as necessary, the DTA will be assembled and a system-level modal survey will be accomplished. Finally, a control system will be installed and closed-loop dynamic response tests will be performed.

O MODAL SURVEY OF EACH INDIVIDUAL COMPONENT

- RING TRUSS "FREE-FREE" WITH MASS LOADING AT INTERFACES
- APPENDAGES WITH FIXED INTERFACES

O MODAL SURVEY OF ASSEMBLED DTA

O CLOSED-LOOP TRANSIENT RESPONSE TESTS

Figure 18

DTA RING TRUSS MODAL SURVEY

The ring truss modal survey was completed in August. For the test, the truss was mass loaded and suspended by a zero spring rate mechanism (ZSRM) from long cables. This system resulted in three virtually zero frequency rigid-body modes and three approximately 0.20-Hz pendulum modes. The ZSRM suspension produced very little restraint in the vertical direction, effectively simulating a free condition for vertical motion.

Three separate excitation points were chosen to assure that good data were obtained for all modes of interest. In addition, tuned decay tests were undertaken to achieve accurate damping data.

O SUSPENDED FROM THREE ZERO SPRING RATE DEVICES

- VIRTUALLY NO RESTRAINT IN VERTICAL DIRECTION
- THREE PENDULUM MODES OF APPROXIMATELY 0.25 Hz

O APPENDAGES REPLACED WITH MASS SIMULATORS TO LOAD INTERFACES

O SINGLE-POINT RANDOM EXCITATION AT THREE DIFFERENT DRIVE POINTS

O TUNED DECAY TESTS TO BETTER ESTIMATE DAMPING

Figure 19

SETUP

ORIGINAL PAGE IS OF POOR QUALITY

The ring truss modal survey was performed in the Martin Marietta Reverberant Acoustic Lab (RAL). This chamber (Figure 20) provides a quiet temperature-controlled environment. Temperature control is essential for future tests where viscoelastic damping treatments are to be used.

Note the suspension system with the ZSRMs on the overhead beams.

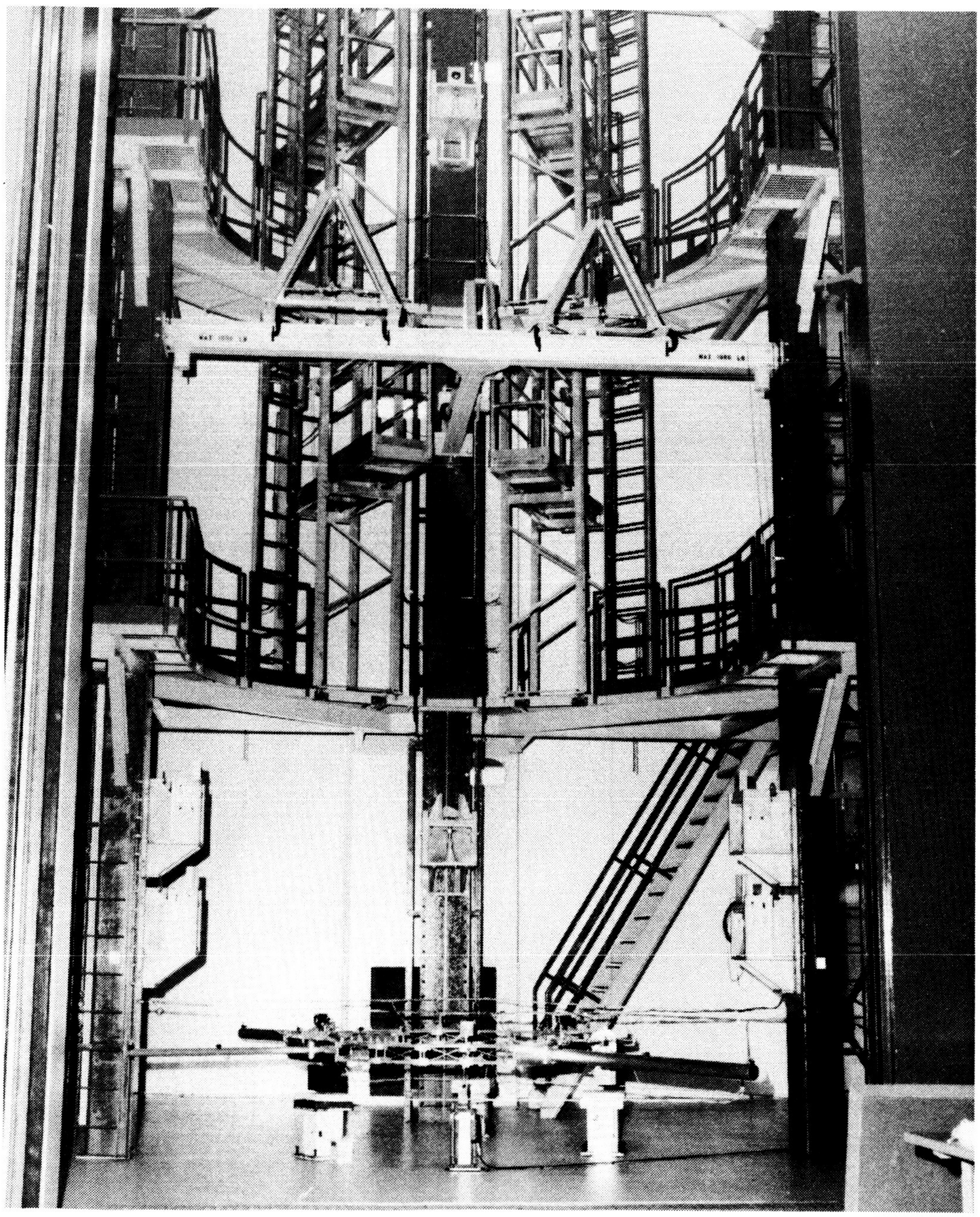


Figure 20

Figure 21 is a closeup of one ZSRM. The spring constants for all three springs are chosen based on the weight to be supported. In the equilibrium position, the mechanism is adjusted so that the side spring rods are in the horizontal position. The side springs are preloaded in compression in this position.

With proper spring sizing and adjustment, the additional force in the vertical spring due to a downward motion of the cable is offset by the component of the compressive loads in the initially horizontal springs in the vertical direction due to a downward rotation of the horizontal springs.

For this particular test, each ZSRM supported about 200 lb. The effective spring rate of the mechanism was approximately 0.4 lb/in. at a 1/2-inch deflection in the vertical direction.

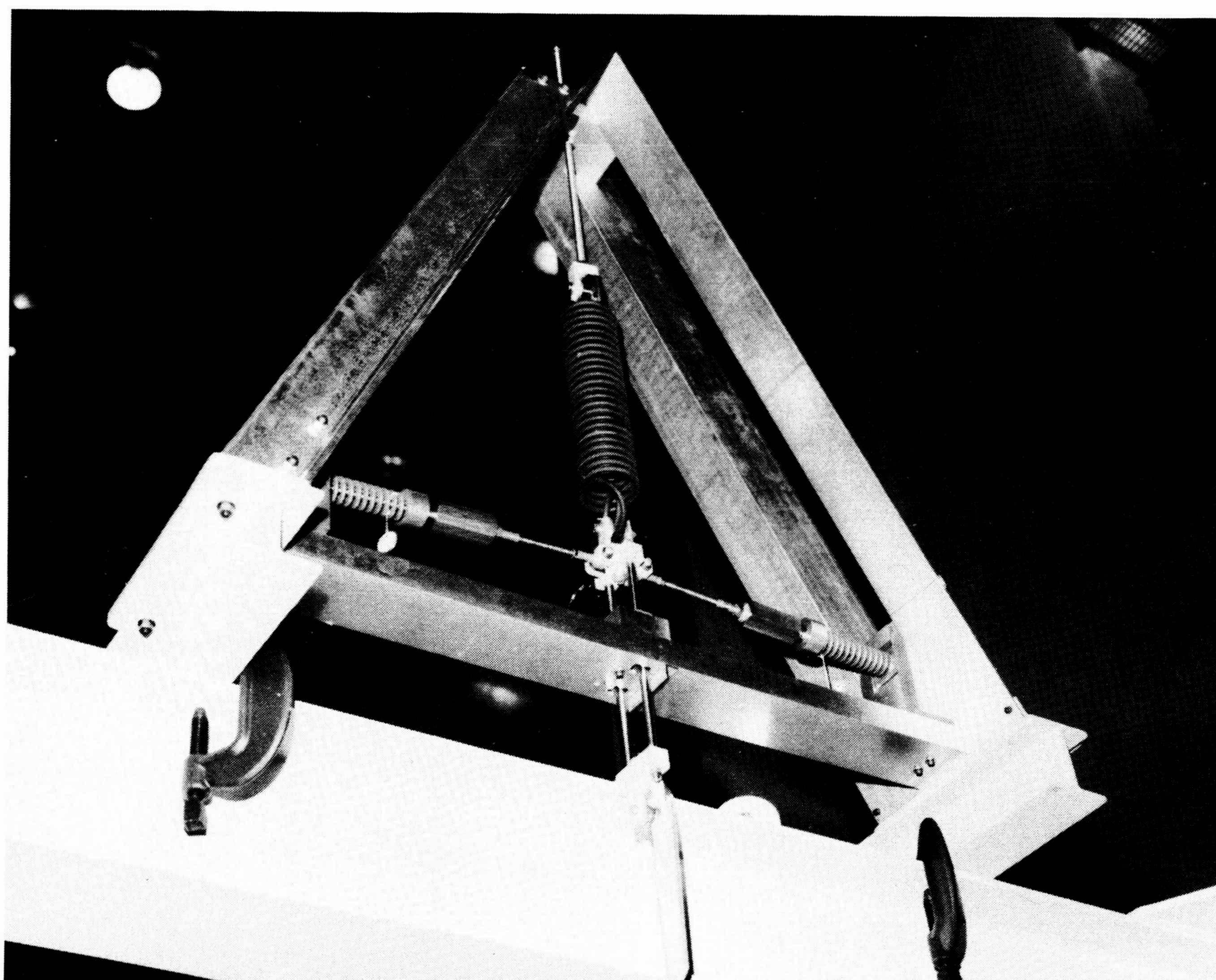


Figure 21

DTA RING TRUSS MODAL SURVEY RESULTS

The results of the ring truss modal survey are shown in Figure 22. The first two modal frequencies agreed very closely with predicted values. These modes are characterized by large motions at suspension points, and the agreement indicates that the ZSRMs were contributing negligible stiffness. The first mode, a symmetric mode, had large motion of the centerline suspension point and very little at the other two supports. The second mode, an antisymmetric mode, had virtually no motion on the centerline suspension point and large out-of-phase motions at the remaining two suspension points.

The higher damping ratios of these modes relative to modes involving small suspension point motion is indicative of slight friction in the ZSRMs.

Notice the very low damping in the remaining modes. For those modes that could be tuned separately, the decay traces provide more accurate damping values than the curve fits from the single-point random tests. For those modes where a range of values or no value for the tuned decay tests is given, satisfactory tuning was not achieved.

The uniformly low damping in the higher modes deserves emphasis. It is often assumed that higher modes have more inherent damping. This is not always the case in precision structures.

FREQUENCY (Hz)	MODAL (VISCOUS) DAMPING (PERCENT)			
	TUNED DECAY	TEST 1	TEST 2	TEST 3
3.24	0.30	0.80	0.42	
6.36	0.70	1.06	0.98	
8.78	0.14	0.21		
9.28	0.17	0.20	0.20	
12.47	0.2-0.3	0.33		
12.68	0.2-0.3	0.22	0.27	
13.05	0.16			0.25
13.22		0.18		
15.10	0.15	0.22	0.14	
16.2	0.17	0.20		

Figure 22

CONCLUSIONS

Although much of the research in the PACOSS program lies ahead, several conclusions can be drawn at this point in time (Figure 23). First, the best control strategy will result from the combined efforts of structural designers, control engineers, and damping designers working together early in the design phase, when sufficient latitude in the design exists to permit trades among the disciplines. Second, the common assumption that damping values increase with frequency does not always hold true with precision structures, and the implications on system performance, should such assumptions prove invalid, must be determined.

- BEST CONTROL STRATEGY FOR FUTURE LSS IS A COMBINATION OF PASSIVE DAMPING AND ACTIVE CONTROLS
- HIGHER MODES OF PRECISE STRUCTURES DO NOT NECESSARILY HAVE SIGNIFICANT INHERENT DAMPING
- OPTIMUM SYSTEMS WILL RESULT FROM INTERACTION BETWEEN STRUCTURAL DESIGNERS, CONTROL ENGINEERS, AND DAMPING DESIGNERS EARLY IN THE DESIGN PHASE

Figure 23

REFERENCES

1. Morgenthaler, Daniel R.; and Gehling, Russell N.: Design and Analysis of the PACOSS Representative System. Damping 1986 Proceedings, May 1986 (AFWAL TR 86-3059).

2. Gehling, Russell N.: Active Augmentation of a Passively Damped Representative Large Space System. Damping 1986 Proceedings, May 1986 (AFWAL TR 86-3059).