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Paul E. McGowan, Harold E. Edighoffer, John W. Wallace

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Langley Research Center
Hampton, Virginia 23665-5225

DEVELOPMENT OF AN EXPERIMENTAL SPACE STATION MODEL FOR STRUCTURAL DYNAMICS RESEARCH

Paul E. McGowan, Harold H. Edighoffer, and John W. Wallace
NASA Langley Research Center
Hampton, VA 23665

Design, analysis and testing of an experimental space station scale model is presented. The model contains hardware components with dynamic characteristics similar to those expected for other large space structures. Validation of analysis models is achieved through correlation with dynamic tests of hardware components and representative assembly configurations. A component mode synthesis analysis method is examined through comparisons with results from fully mated system models. Selection of input requirements for accurate component synthesis analysis predictions are assessed.

INTRODUCTION

Verifying analysis models for predicting the dynamic response of large, flexible space structures is an important research challenge. These analysis models are necessary for establishing accurate load estimates during design and operational constraints on-orbit. Historically, ground vibration tests of representative flight hardware have served an important role in analysis verification. Due to the large size of current space structure designs, ground tests may be limited to components and subassemblies. Analysis models can then be combined to produce an analysis prediction of the full spacecraft system using component mode synthesis approaches. The accuracy of these predictions depends on validation of component analysis models and modeling of component interface boundaries. Ground tests of dynamically scaled models can validate component synthesis approaches by providing verified analysis models of the components as well as verified analysis models of the fully mated system.

Ground vibration tests of flexible space structures typically involve suspension of the test article. These tests allow the simulation of expected on-orbit boundary conditions and provide a means to validate analysis predictions. With the class of structure examined recently, the development of advanced suspension methods for lightly loaded and very low frequency test articles may be required [1,2]. Prediction of the test article vibration characteristics prior to performing the suspension tests is necessary for suspension design and for pre-test analyses.

A structural model representing a generic space station design was previously studied to validate analysis models [3]. This design consisted of a cylindrical habitation module with two flexible solar panels and a radiator panel joined by a stiff connecting cube. More recently, designs proposed for space station have tended toward truss-type structures connecting a variety of flexible components (see Figure 1). Current analysis predictions indicate that flexible components will likely have natural frequencies in the same range as those dominated by global truss motions. This challenges the state-of-the-art capability for conducting accurate ground tests as well as performing on-orbit system identification.

An ongoing research program at the Langley Research Center is aimed at developing scale model technology for analysis and ground test methods for large flexible structures [1,4-6]. This program, entitled Dynamic Scale Model Technology (DSMT), is developing a hybrid-scale structural dynamic model of the space station including all of its flexible components, payloads and modules. The space station structure provides an excellent focus since it will be the first opportunity to obtain on-orbit flight data to correlate with analysis predictions for this class of structure. Prior to suspending and testing the detailed and somewhat fragile DSMT hybrid-scale model, a less refined experimental scale model has been assembled to assess test and analysis methods, and identify key technical issues and possible solutions.

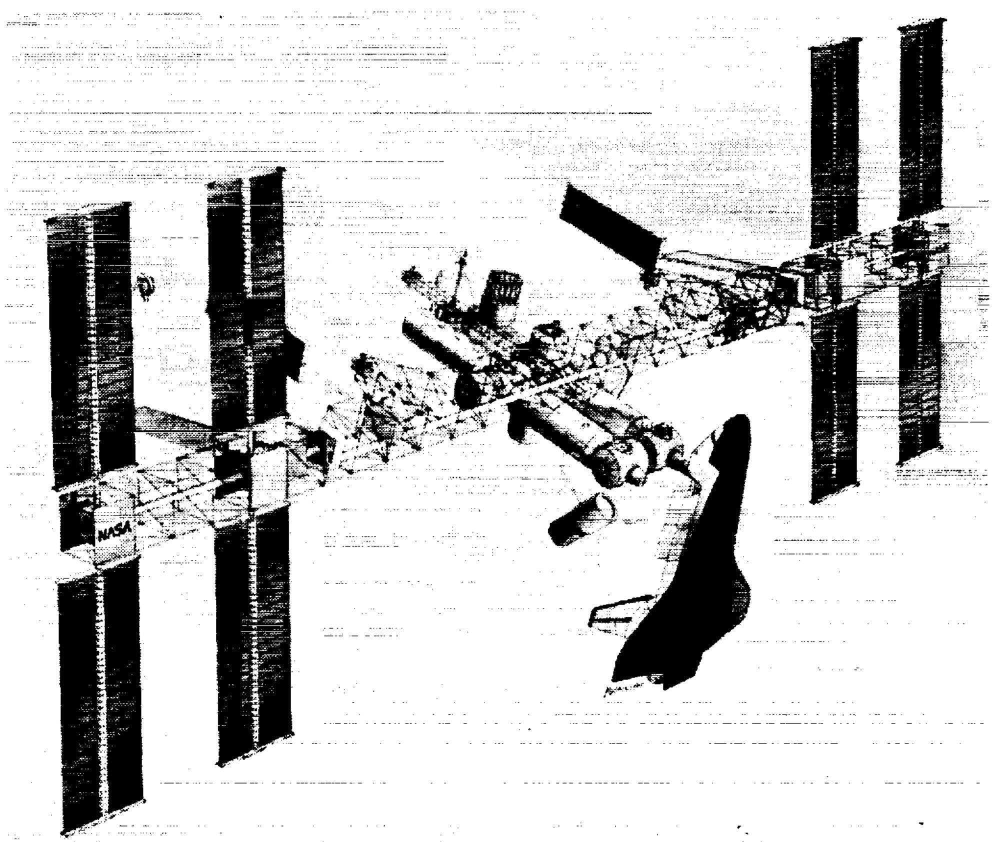

Figure 1. Design concept for space station

The purpose of this paper is to describe the development of the experimental model, including design, analysis and testing of its various components. Specifically, correlation of analysis predictions with static and dynamic tests are presented to validate the component analysis models. Also, test and analysis results for representative space station assembly configurations are presented. In addition, component mode synthesis analysis predictions are compared with analysis predictions from full system models. Selection of input modes for obtaining accurate component synthesis results is addressed. While suspended tests are not included in this work, the resulting accurate analysis models of the experimental truss hardware will be used for the suspended tests which follow.

EXPERIMENTAL SCALE MODEL DESIGN AND ANALYSIS

The experimental scale model hardware consists of a generic truss structure with flexible components designed for cost-effective fabrication. Dynamic characteristics for this model resemble those expected for the DSMT hybrid-scale model [1]. These characteristics include the presence of closely spaced and relatively low frequency vibration modes. No attempt was made to reproduce exactly the expected vibration frequencies or mass properties for the DSMT hybrid-scale model. Instead, the experimental model was designed with the goal of maintaining the proper ratios of certain quantities for the flexible components relative to the truss structure. The following depicts some of these quantities, based on analysis:

Component to truss fundamental frequency ratio	0.65
Flexible component mass	25%
Rigid mass	60%
Distributed mass	15%

The generic truss is an erectable structure using commercially available hardware (trade name MEROFORM). Figure 2 shows a 23-bay truss assembly, with details of the truss nodes and struts shown in Figure 3. The nodal joint design allows the truss to be assembled into numerous configurations in any of three orthogonal directions. Each assembled truss bay is a cube with 19.7 in (0.5 m) sides which weighs approximately 7 lbs. This produces a model with planform dimensions the same as the DSMT hybrid-scale model, and is 1/10-size of a full-scale space station truss bay. All other key dimensions and properties of this generic truss are presented in Figure 4.

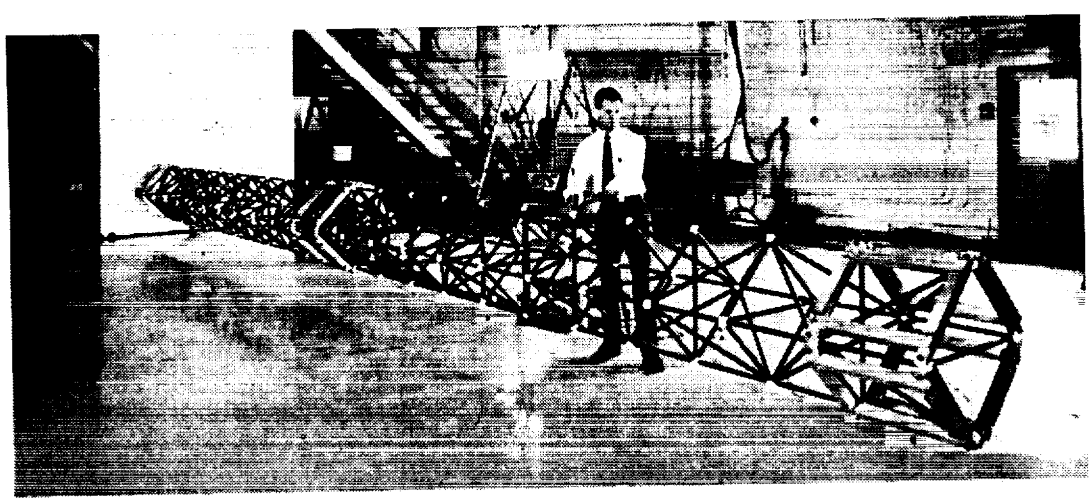

Figure 2. 23-bay generic truss assembly

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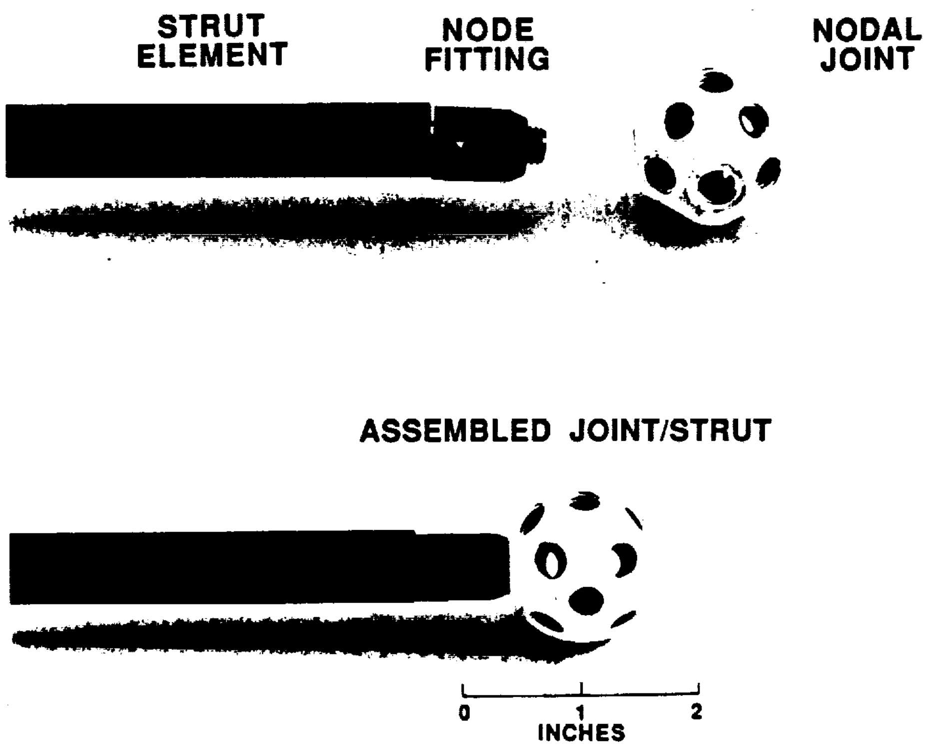
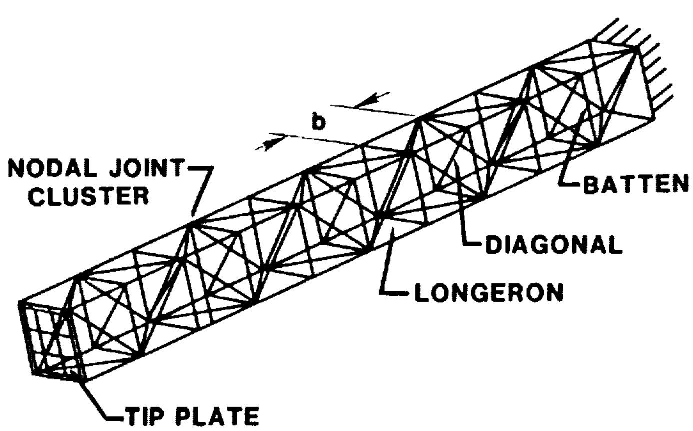


Figure 3. Generic truss structure nodal joint and strut



Truss Properties	
total weight	147.40 lb
tip weight	86.25 lb
nodal joint cluster weight	1.097 lb
strut EA	1.139×10^6 lb
b	19.7 in
total length	197.0 in
ρ diag	0.0731 lb/in^3
ρ long	0.0785 lb/in^3

Figure 4. Ten-bay generic truss structure properties

The solar array component (Figure 5) is a frame constructed of aluminum cylindrical rods and which attaches to the truss via a square, flat aluminum plate. The frame is assembled with cube-shaped attachment joints which allow any of the frame sections to be replaced without disassembly of the entire unit. Properties and dimensions of this component were selected to provide a structure with both bending and torsional modes in the frequency range of interest. Frequency separation and coupling between these modes can be controlled by adding masses at various cube joints. In addition, orientation of the solar array with respect to an axis normal to the interface plate can be varied, simulating the beta-joint articulation capability present on the space station design.

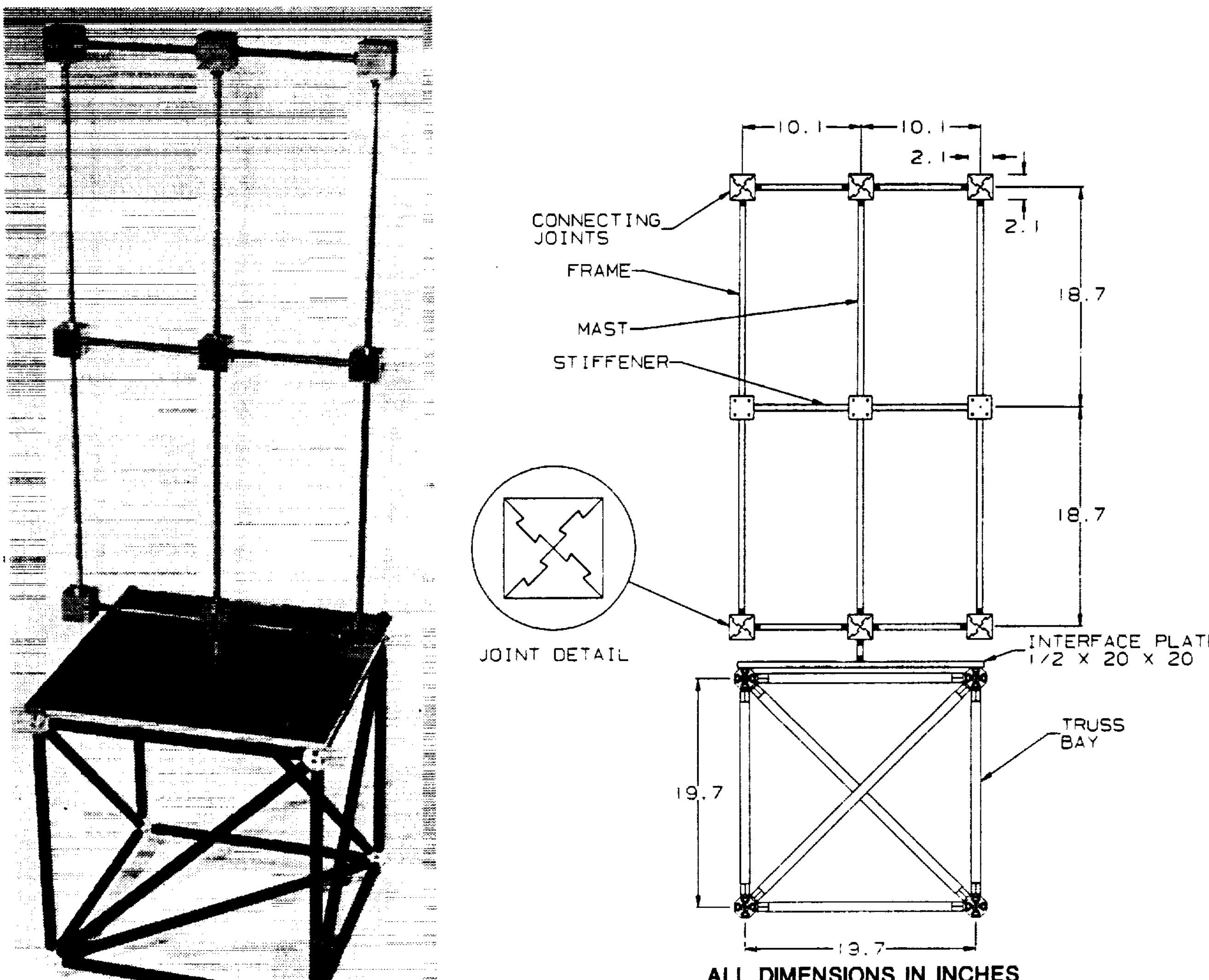


Figure 5. Details of solar array component

The radiator component (Figure 6) consists of an aluminum plate mast secured to an interface support plate attached to the face of a truss bay. The mast provides distinctly different in- and out-of-plane bending stiffnesses and vibration modes as does the full-scale radiator design. Mass-loading using four cylindrical tip masses provides lower natural frequencies.

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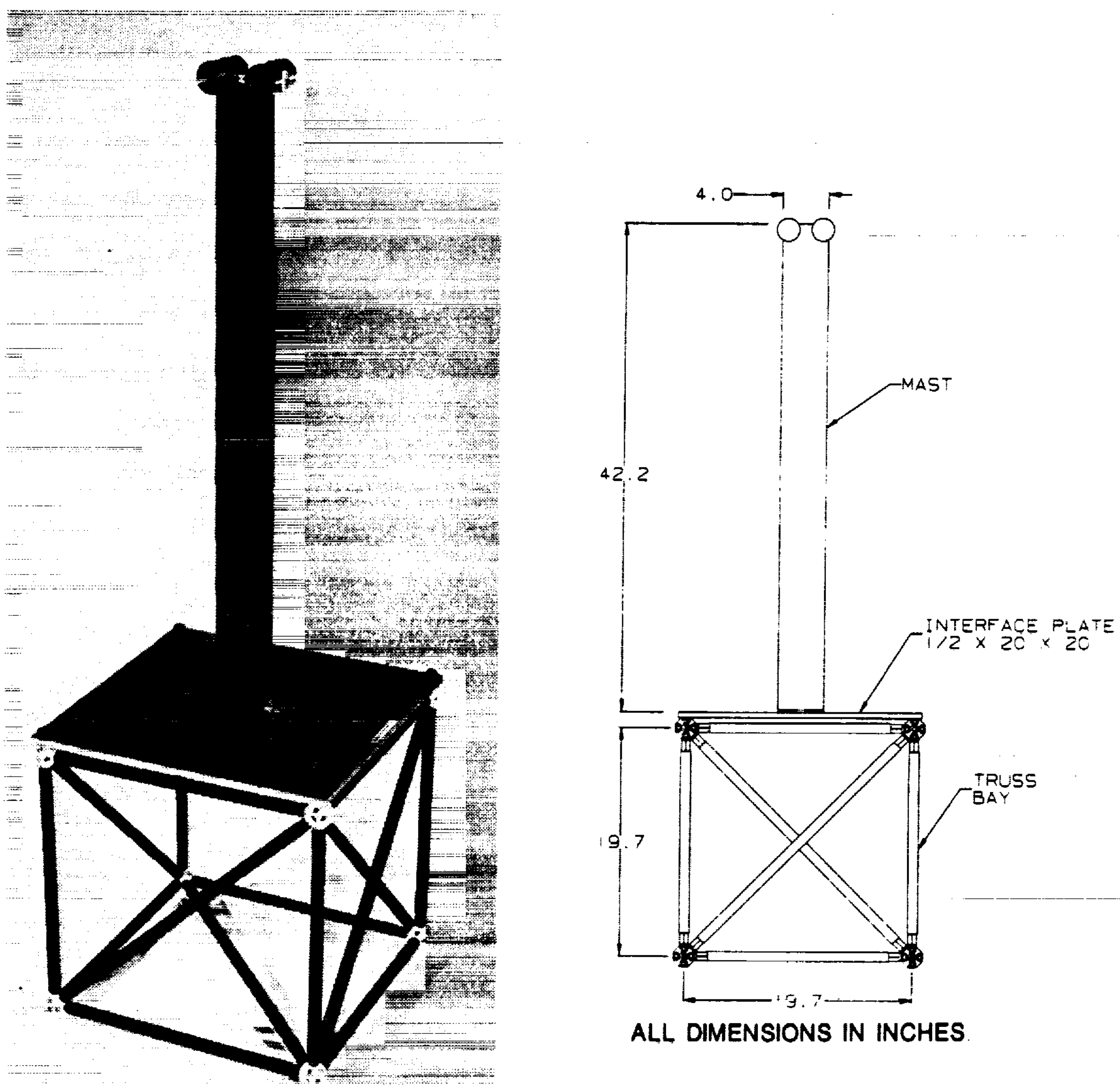

Figure 6. Details of radiator component

ASSEMBLY CONFIGURATIONS

The experimental model hardware can be assembled into different configurations thereby providing structures with varying degrees of complexity. At present three focus configurations are being investigated using a ten-bay truss section, two solar arrays and a radiator component. These three configurations represent varying levels of assembly during the build-up of an early space station flight configuration. Test and analysis results are presented in this paper for two configurations. The first configuration (Figure 7a) consists of a ten bay truss structure with one flexible solar array component located at the truss tip bay. A second configuration is obtained by adding a radiator component to the first configuration at the fourth bay from the truss tip bay. Finally, a third configuration (Figure 7b) consists of the truss with two solar arrays and a radiator. Ultimately, the experimental hardware will be configured into a structure resembling that in Figure 8 which in addition to the solar arrays and radiators, consists of several pallets and two simulated alpha-joints.

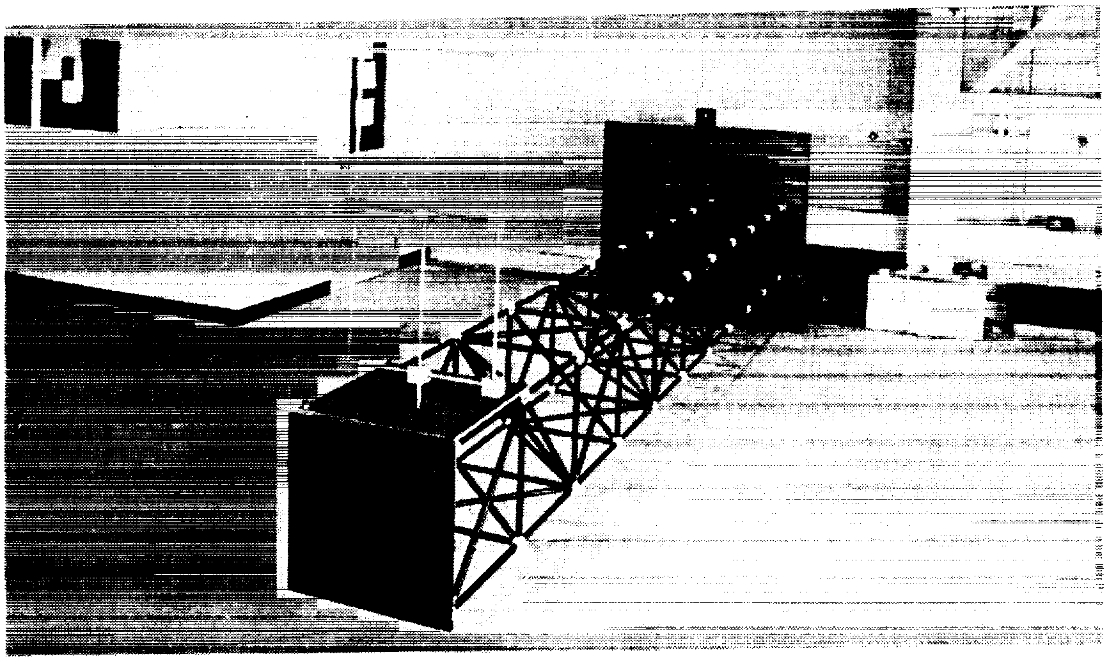

Figure 7a. First assembly configuration

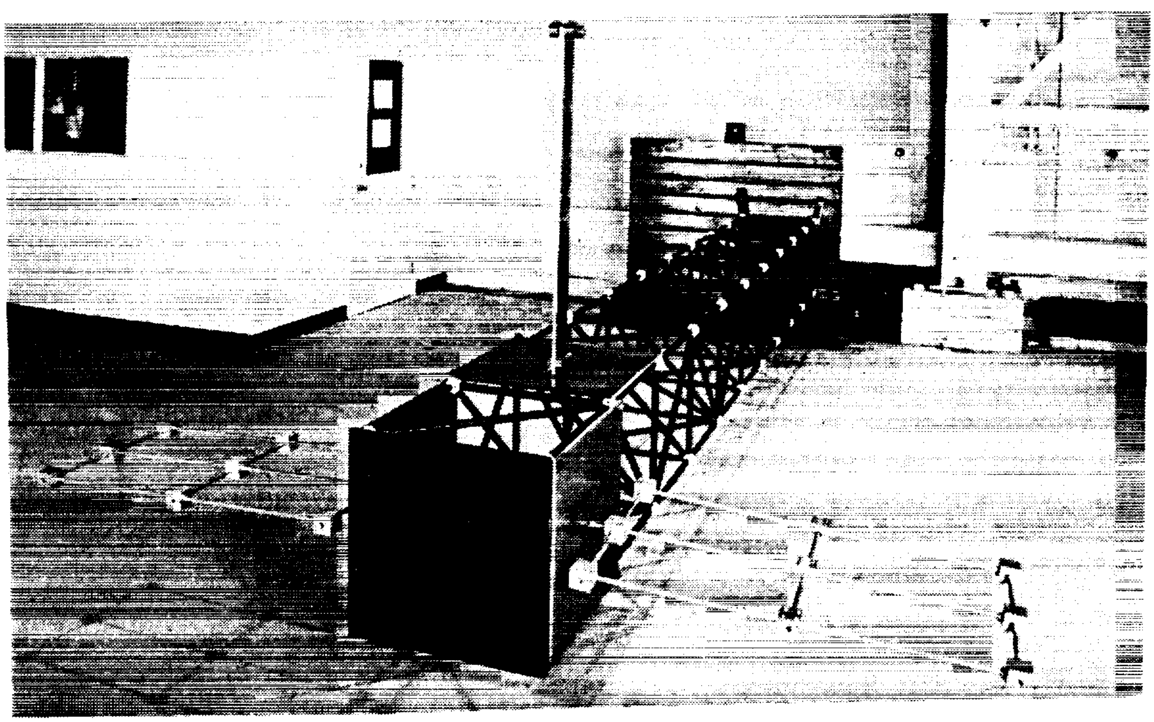

Figure 7b. Third assembly configuration

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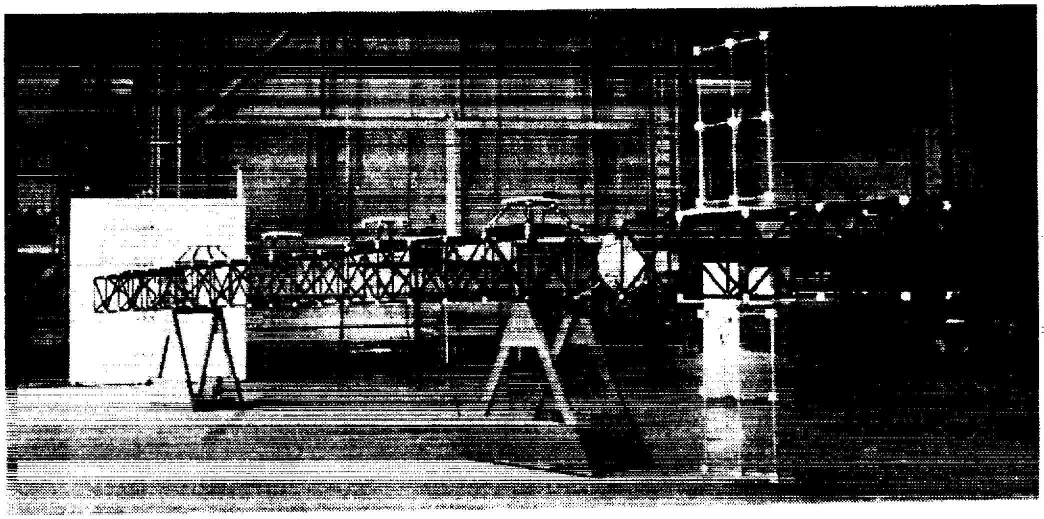


Figure 8. Fully assembled experimental model configuration

FINITE ELEMENT ANALYSIS METHOD

All finite element analysis results presented herein were obtained using the Engineering Analysis Language (EAL) computer program [7]. Schematics of the analytical models used for components of the experimental model are shown in Figure 9. Table 1 contains information regarding the level of detail used in each analysis model. Subsequent sections of the paper present the results for each component and the assembly configurations.

Table 1. Description of component analysis models

Component	Degrees-of-Freedom	No. Bar Elements	No. Plate Elements
Truss	360	143	18
Solar array	240	54	0
Radiator	306	9	20

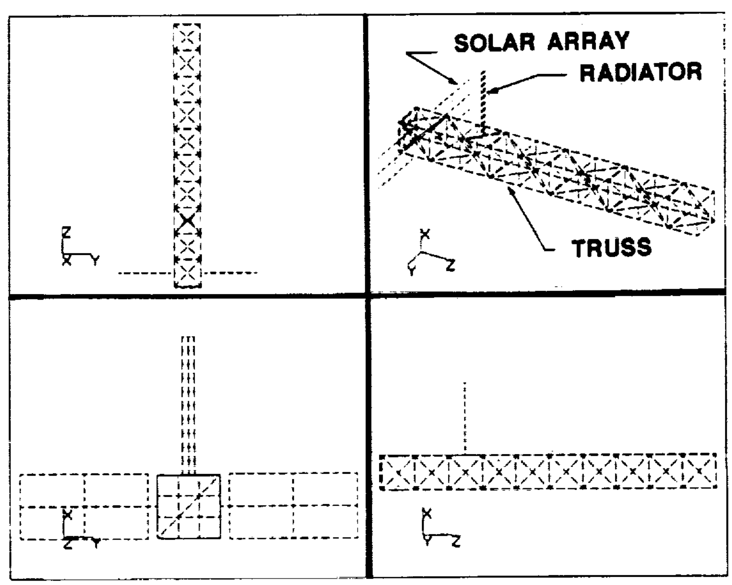


Figure 9. Schematic of finite element analysis models

TEST METHOD

Modal tests were conducted using impact testing techniques. An instrumented force hammer was used as an excitation source. There were 32 acceleration measurements made from 16 pairs of accelerometers located throughout the structure. Each accelerometer and the force hammer were calibrated to achieve accurate mode shape estimates. A GenRad 2515 dynamic test system was used to acquire and reduce all of the test data. Acceleration and force measurements were processed with the SDRC Modal-Plus software [8] and used to estimate modal parameters. Ensemble averaging was used to minimize any noise present. Frequencies and mode shapes were extracted from frequency response functions calculated in the modal analysis. Mode shapes were animated to facilitate identification and correlation with analysis modes.

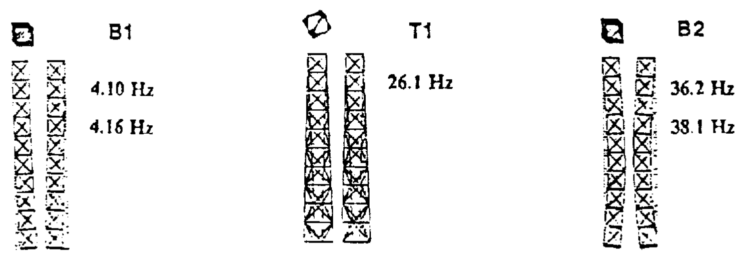
COMPONENT TEST/ANALYSIS CORRELATION

The following sections present the test/analysis correlation achieved for the model components. Nominally the frequency range of interest for this study was from 0 to 40 Hz. This frequency range is consistent with that anticipated for testing of the DSMT hybrid-scale model [1,2].

Generic Truss Structure

Results from static characterization tests of the truss elements and a ten-bay truss section were reported previously [9,6]. These tests provided validation of stiffness properties used to construct finite element models of the truss. Excellent correlation with dynamic test results was obtained using these finite element models [10]. For the dynamic tests, the truss was cantilevered and loaded with a heavy tip mass. Figure 10 contains selected excerpts from [10] indicating the excellent test/analysis correlation achieved.

Mode Shapes



Test/Analysis Correlation

mode no.	description	frequency (Hz)		percent difference	MAC
		analysis	test		
1	B1	4.10	3.94	3.9	0.998
2	B1	4.16	4.00	3.8	0.999
3	T1	26.1	26.1	0.0	0.999
4	B2	36.2	36.0	0.55	0.999
5	B2	38.1	37.8	0.78	0.997

* did not attempt to measure

MAC=Modal Assurance Criterion

Figure 10. Mode shapes and test/analysis correlation for ten-bay generic truss

Solar Array

Dynamic tests were performed on the solar array component mounted on a truss bay. Thus, the interface support plate was loaded as in the fully assembled model configuration. As shown in Table 2, 10 modes were experimentally identified. Table 2 also shows comparisons of finite element results for the solar array model. In the finite element analysis model each of the frame members was modeled using beam finite elements. In addition, the interface support plate was represented in the finite element model by four beam elements. These four beam elements attach the base of the solar array to the four nodes in the truss bay face. Figure 11 depicts representative analytical mode shapes from the finite element model. In this figure, each of the four quadrants contains orthographic and isometric views of the given mode shape. The mode shape label indicated in the upper right portion of each quadrant refers to the mode shape designation from Table 2. Using the initial analysis model relatively poor correlation was achieved due to a coarse finite element discretization at the attachment joints and uncertainties in modeling the stiffness at the interface support plate. A detailed revision of the mass and stiffness properties of the solar array frame was made. In addition, the interface support plate was statically tested to determine its stiffness properties when attached to the face of a truss bay. With these revisions the excellent agreement shown in Table 2 was achieved.

Radiator

Dynamic tests were also performed on the radiator component, with similar boundary conditions to the solar array. Four test modes were identified and are reported in Table 3 along with finite element analysis predictions. Corresponding analysis mode shapes are shown in Figure 12. The interface support plate was also represented by four beams using information from static tests to adjust their stiffness properties. Because of the attachment geometry of the radiator, a fifth and sixth beam element was added to attach the radiator base to the four nodes of the truss bay face. Good test/analysis agreement was achieved for three of the first four vibration modes. The fourth mode is not as accurately represented, but since it is outside the frequency range of interest, no further refinement of the model was deemed necessary.

Table 2. Test/analysis correlation for solar array component

Mode Number	Mode Shape	Frequency (Hz) Test	Initial Analysis	% Diff.	Frequency (Hz) Revised Analysis	% Diff.
1	B1-o	2.36	2.18	7.6	2.43	3.0
2	B1-i	3.13	2.80	10.5	3.31	5.8
3	T1	7.27	6.36	12.5	7.29	2.8
4	B2-o	17.30	15.49	10.4	17.04	1.5
5	T2	21.11	18.46	12.1	21.43	1
6	B2-i	30.59	26.24	14.2	31.91	4.3
7	T3	31.64	31.03	2.0	35.24	11.4
8	B3-o	36.22	32.46	10.4	38.86	7.3
9	B3-i	41.16	33.37	18.9	41.57	1.0
10	A1	43.50	38.91	10.6	44.87	3.1

B=Bending mode T=Torsion mode A=Axial mode
o=out-of-plane i=in-plane

Table 3. Test/analysis correlation for radiator component

Mode Number	Mode Shape	Frequency (Hz) Test	Analysis	% Diff.
1	B1-o	5.04	4.92	2.3
2	B1-i	20.40	20.93	2.6
3	T1	42.46	41.41	2.5
4	B2-o	81.99	73.22	10.7

B=Bending mode T=Torsion mode A=Axial mode
o=out-of-plane i=in-plane

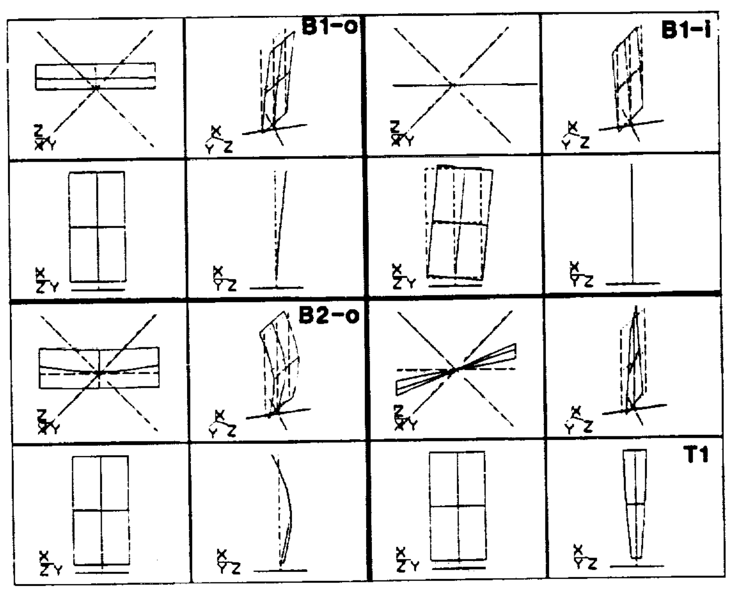


Figure 11. Typical analysis mode shapes for solar array component

COMPONENT MODAL SYNTHESIS ANALYSIS PREDICTIONS

An evaluation of a component mode synthesis approach for predicting the full system dynamics was made using the method of Craig-Bampton [11]. This method is a solution option within the EAL computer program [7]. In this method the substructure generalized coordinates consist of a set of interface coordinates and a set of normal mode coordinates. These normal modes are obtained with all interface coordinates fully constrained. For the present analysis each of the truss, solar array and radiator components was treated as a separate substructure with interface coordinates at all attachment points between substructures and at the truss cantilevered end

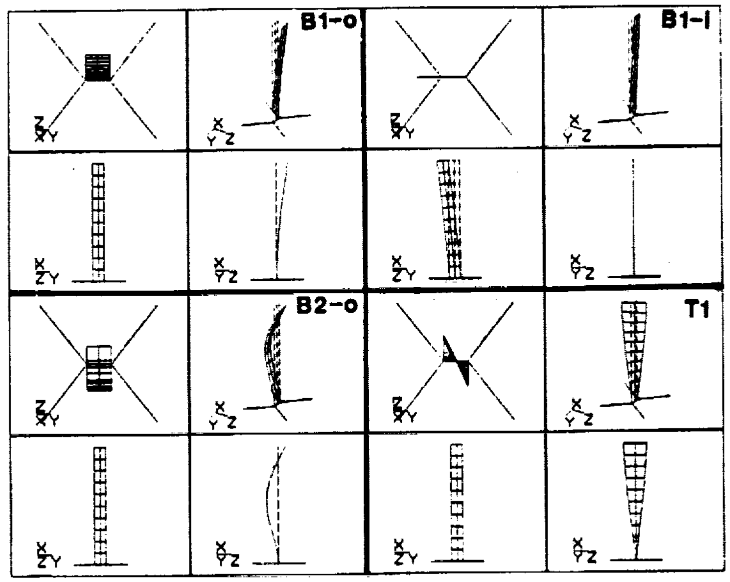
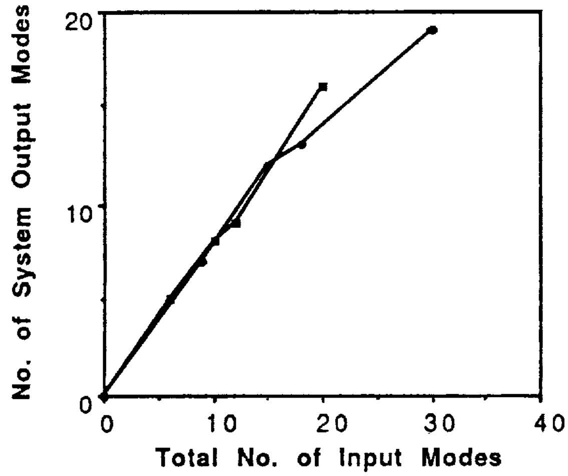


Figure 12. Typical analysis mode shapes for
radiator component

Input requirements for the component mode synthesis were assessed by examining the number of and frequency content of the substructure input modes. The number of consecutive reliable system output modes was evaluated versus the total number of input modes. In addition, the highest reliable system output frequency was evaluated versus the highest input frequency of the most flexible substructure. For the configurations studied herein the solar array component was the most flexible substructure. A reliable system mode was defined as a consecutive mode with a frequency deviation from the full model frequency of less than one percent. Results from the full system model analysis are contained in a subsequent section of the paper.

Shown in Figure 13 are the effects of varying the total number of input modes on the number of system output modes obtained. For this analysis an equal number of modes from each substructure were used. Four evaluations were made with 3, 5, 6, and 10 lowest modes for each substructure used as inputs. Two curves are presented in Figure 13 representing analyses using two and three substructures (the first and second assembly configurations, respectively). For the two substructure analysis the number of reliable system output modes is 80 percent of the total number of substructure input modes. In addition, the three substructure analysis indicates that 63 percent of the total input modes are obtained as system modes. Future studies will consider including a variable number of input modes for each substructure.

The effects of substructure mode input frequency variation on the highest reliable system output frequency are shown in Figure 14. This represents another method for evaluating input requirements for the component mode synthesis analysis. These results indicate that the highest reliable system frequency is greater than or equal to the highest input frequency for the most flexible substructure. Thus, for the two configurations studied in this paper the system frequency output would be accurate to at least the highest frequency of the solar array component. Further studies with additional and more complex model configurations are required in order to establish input requirement criteria for more general use.



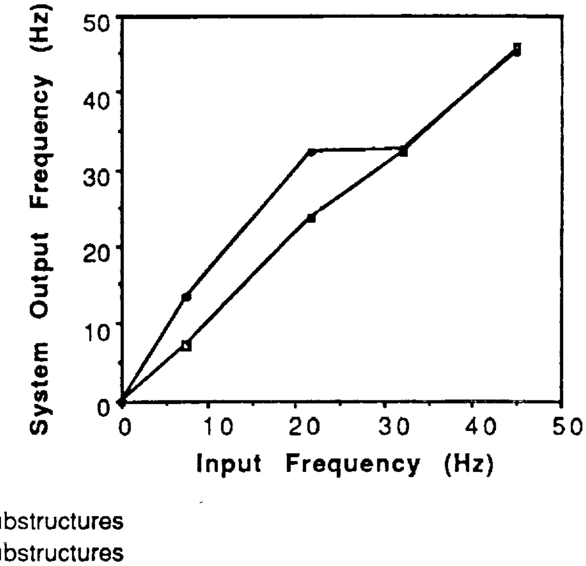


Figure 13. Effects of input mode selection on component mode synthesis results

Figure 14. Effects of input mode frequency on component mode synthesis results

ASSEMBLED MODEL VALIDATION

First Assembly Configuration

An analysis model of the first assembly from Figure 7a was obtained by connecting test verified models of the truss section and solar array at appropriate interfaces. Typical analysis mode shapes are depicted in Figure 15. Dynamic tests were performed on this configuration using a single input excitation in one direction at a given truss nodal joint. Accelerations were measured (in two directions) at each of four truss joints at three bays along the truss length and at four locations on the solar array. The identified test and analysis frequencies and corresponding mode shapes are reported in Table 4. Analytical frequencies in Table 4 represent the results from a complete finite element model of this configuration. These analysis results were identical to those achieved when using 20 input modes for each substructure in the component mode synthesis analysis. There are 13 test modes identifed in the 40 Hz frequency range. Specifically, 5 truss modes and 8 solar array modes. Each mode involves dominant motion of one component which corresponds to a comparable mode shape identified in the component tests but with a slightly lower frequency. Although there is considerable motion of the flexible solar array evident in modes dominated by global truss motion, the results indicate minimal interaction between modes for this configuration. This is most likely due to the local stiffness provided by the interface support plate at the solar array base.

Second Assembly Configuration

The second assembly configuration was dynamically tested in a manner similar to that described above. Additional sensors were added in two directions on two points at the radiator tip. Test results and identified mode shapes are listed in Table 5. The test verified radiator analysis model was joined with the first assembly model to provide the analysis predictions. Representative analysis mode shapes are shown in Figure 16 with the analysis predictions also listed in the Table. Analytical frequencies listed in Table 5 represent results from a complete finite element model of this configuration. The modal synthesis analysis using 20 input modes for each substructure again resulted in the same frequencies as the complete model results. In this case there are 15 test modes identified. These consist of the original 13 modes from the first assembly and two additional modes involving the radiator component. All modes from the second assembly retain the same character as previously displayed in the first assembly; however, mode identification was complicated by the presence of the additional flexible component. Mode shapes involving predominantly the solar arrays remained essentially unchanged indicating the presence of the radiator has little effect on them. An interaction between modes 7 and 11 was noted. These modes both involve torsion of the truss structure coupled with in-plane bending of the radiator component. In addition, the second truss bending mode pair was affected by the mass of the radiator, which was placed at the point of maximum displacement for these modes. As a final validation of the analysis models developed in this work, dynamic tests of the third configuration shown in Figure 7b are required. Testing of this structure was underway at the time of this writing.

Table 4. Test/analysis correlation for first assembly configuration

		Frequency (Hz)	
Mode No.	Mode Shape	Test	Analysis
1	S, B1-o	2.27	2.42
2	S, B1-i	2.88	3.07
3	TR, B1	3.43	3.62
4	TR, B1	3.73	3.96
5	S, T1	7.07	7.29
6	S, B2-o	17.12	16.99
7	S, T2	20.78	21.43
8	TR, T1	23.03	23.58
9	S, B2-i	30.60	32.22
10	S, T3	33.12	35.23
11	TR, B2	35.19	35.82
12	TR, B2	37.76	37.75
13	S, B3-0	36.31	38.92

Table 5. Test/analysis correlation for second assembly configuration

		Frequency (Hz)	
Mode No.	Mode Shape	Test	Analysis
1	S, B1-o	2.25	2.42
2	S, B1-i	2.88	3.05
3	TR, B1	3.30	3.50
4	TR, B1	3.63	3.87
5	R, B1-o	5.16	4.89
6	S, T1	7.10	7.29
7	R, B1-i	13.89	13.49
8	S, B2-o	17.13	16.99
9	S, T2	20.78	21.43
10	TR, B2	26.09	26.84
11	TR, T1	27.34	28.09
12	TR, B2	31.59	31.93
13	S, B2-i	33.40	32.55
14	S, T3	33.12	35.25
15	S, B3-0	36.49	38.23

S=Solar array mode   R=Radiator mode   TR=Truss mode
B=Bending mode   T=Torsion mode   A=Axial mode
o=out-of-plane   i=in-plane

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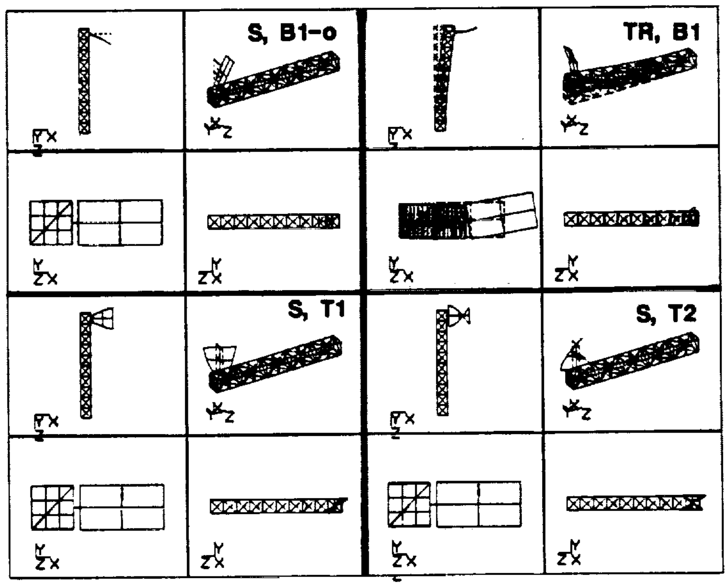

Figure 15. Typical analysis mode shapes for first assembly configuration

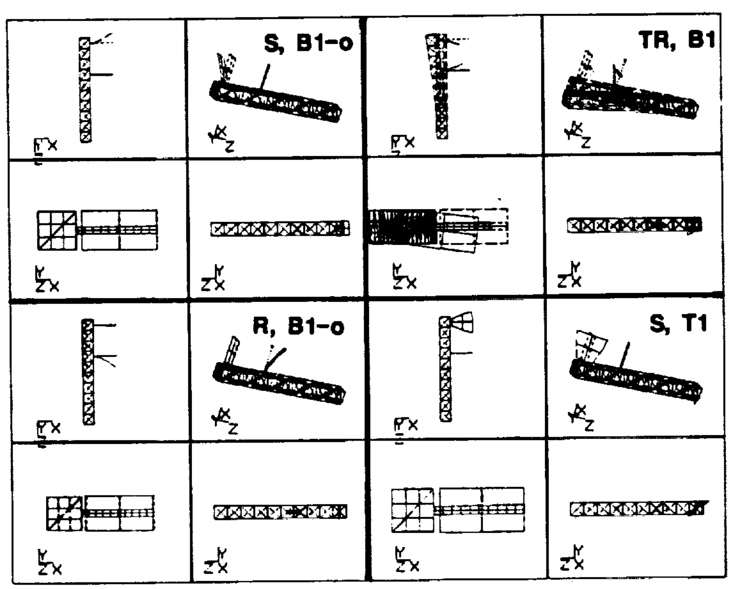

Figure 16. Typical analysis mode shapes for second assembly configuration

CONCLUDING REMARKS

An experimental scale model has been developed with the dynamic characteristics of a large, flexible space structure, such as space station. This provides a capability to assess ground test and analysis methods for this class of structure, investigate interactions between flexible components and global truss vibration modes, and to establish the modeling detail required for accurate system predictions. Finite element analysis models of a generic truss and two flexible component structures were verified by correlation with dynamic test results. These analysis models were used to develop analysis predictions for two configurations representing two different space station assemblies. Test and analysis correlations for the two focus configurations are excellent. Results from a component mode synthesis analysis provide insight into the selection of input requirements for generating accurate predictions of full system dynamics. For the two configurations examined at least 63 percent of the total number of input modes were found to be reliable system output modes when an equal number of input modes are used for each substructure. In addition, all system output frequencies up to the highest input frequency of the most flexible substructure were found to be accurate. Development of criteria for more general use requires evaluation with more complex structures involving multiple components.

ACKNOWLEDGEMENTS

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P. E. McGowan: Langley Research Center, Hampton, Virginia.
H. H. Edighoffer: AS&M, Inc., Newport News, Virginia.
J. W. Wallace: Langley Research Center, Hampton, Virginia.
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16. Abstract

Design, analysis and testing of an experimental space station scale model is presented. The model contains hardware components with dynamic characteristics similar to those expected for other large space structures. Validation of analysis models is achieved through correlation with dynamic tests of hardware components and representative assembly configurations. A component mode synthesis analysis method is examined through comparisons with results from fully mated system models. Selection of input requirements for accurate component synthesis analysis predictions are assessed.

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