AN EVALUATION OF USING COMPONENT MODE SYNTHESIS FOR STRUCTURAL MODELS

by

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ABSTRACT

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Structural analysts have used component mode synthesis (CMS), also known as substructuring, for decades to divide large structural models into smaller, more manageable models, and to reduce the size of the associated mathematical problem. Applying CMS can also reduce the front-end effort to build models, even models that are smaller than those for which CMS has typically been applied. This paper presents an evaluation of the accuracy of two smaller substructured models so that analysts might feel confident in applying CMS to gain the benefits.

The evaluation starts with an introduction to CMS. Then, CMS is applied to a small, simply-supported beam model and a medium-sized, electronics enclosure model. Accuracy is evaluated by comparing the frequencies and mode shapes of substructured models to the full models. The electronics enclosure substructured model is evaluated further by comparing a displacement response and the solve time to that of the full model.

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CHAPTER 1

INTRODUCTION

1.1 Background and Advantages of CMS Use

Component mode synthesis (CMS) is a process that involves separating a structure into substructures, or components, calculating component mode shapes which describe the displacement of points within the substructures, and assembling, or synthesizing, this information into a reduced-order model of the full structure. This process is depicted in Figure 1.1. Structural analysts have used CMS, also commonly referred to as substructuring, since the 1960s to solve very large structural dynamics models, such as those of aircraft. Craig has discussed the history of CMS and provided an overview of the terminology and methods employed since the earliest developments [1,2].

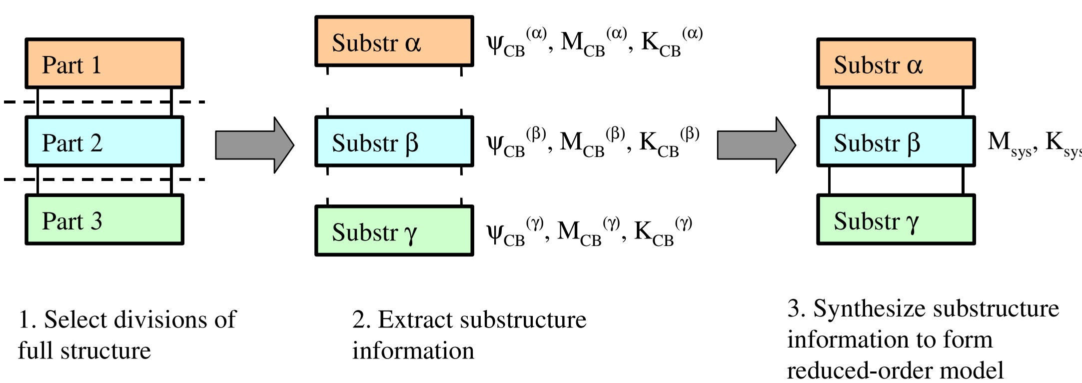

Figure 1.1 The CMS process

CMS has provided analysts with many advantages. It provided a method of reducing the size of the mathematical problem, so that analysts could obtain answers in less time or with limited computer resources. The division of the structure into substructures facilitated concurrent work by different analysts on different parts of the structure. The analysts of subassemblies could feed reduced-order models of these subassemblies to the analysts responsible for the top-level structure. This was done by choosing to retain only those component modes that had significant influence on the response of the top-level structure. The analysts of the top-level structure would assemble these reduced-order substructure models into a reduced-order top-level model. They would realize even greater efficiency if any of these substructures were used multiple times.

1.2 Questions Regarding CMS Use

Given the value that CMS has historically provided in solving very large models, and considering the capabilities of the latest finite element software with present-day computer processing speeds, might CMS prove valuable for, not only very large models, but smaller models as well? This high-level question was broken down into more detailed questions for investigation. (1) How does the accuracy of a substructured model compare to the accuracy of the full model? (2) Does increasing the number of substructures in a model reduce accuracy? (3) Does the reduced-order model result in an unacceptable amount of error for applications where a high level of accuracy is required? (4) Does substructuring reduce solver time when compared to the solve time of the full model?

1.3 Scope of this Work

The purpose of this work is to investigate questions (1) through (4) listed above by building substructured and full versions of a small model of approximately 100 DOF and a medium-sized model of approximately 1.5 million DOF. Accuracy was evaluated by comparing natural frequencies and mode shapes of substructured models to those of the full models. To further evaluate accuracy of the medium-sized substructured model, a displacement response measure was defined, calculated, and compared to that of the full model. Finally, the solve time of the medium-sized, substructured model was compared to the solve time of the full model.

1.4 Software Used in this Work

The software used to create finite element models, write analysis input files, and perform some postprocessing of results for this work was ABAQUS/CAE®. Eigenvalues, eigenvectors, and generalized masses were calculated using ABAQUS/Standard®. Both of these products, hereafter referred to as ABAQUS, are developed, produced, and distributed by ABAQUS, Inc. Additional information for these and other products can be found at http://www.abaqus.com.

Additional postprocessing of results for the electronics enclosure problem was performed using finite element postprocessing and test data processing software developed and used internally by Raytheon.

CHAPTER 2

UNDERSTANDING AND IMPLEMENTING CMS

2.1 Introduction to the Craig-Bampton Method

A number of authors have presented CMS methods over the years with the major differences pertaining mostly to the type of component modes that are used for the substructures [1,2]. Component modes can be normal modes, or eigenvectors, with interface degrees-of-freedom (DOF) that are fixed, free, or loaded. Component modes can also be a constraint mode, defined as the static deformation shape that is produced by applying a unit displacement at one DOF of a specified constrained set of DOF and the remaining DOF of this set being restrained with a displacement of zero, or an attachment mode, defined as the static deformation shape that is produced by applying a unit force to one DOF of an attachment set of DOF and the remaining DOF of this set being force-free.

One of the most straightforward and widely-implemented CMS methods was presented in 1968 by Craig and Bampton [3]. This method defines a component transformation matrix ψ_{CB} that consists of a selected set of fixed-interface normal modes and interface constraint modes. Reduced component mass and stiffness matrices are defined as

$$M_{CB}{}^{(s)} = \psi_{CB}{}^{(s)^T} m^{(s)} \psi_{CB}{}^{(s)} \quad \text{and} \quad K_{CB}{}^{(s)} = \psi_{CB}{}^{(s)^T} k^{(s)} \psi_{CB}{}^{(s)}$$

where m and k are the component mass matrix and component stiffness matrix, respectively, and the superscript s refers to a component label, such as component α or component β. The reduced component matrices are then synthesized into system mass and stiffness matrices using

$$M_{system} \equiv S^T \begin{bmatrix} M_{CB}{}^{(\alpha)} & 0 \\ 0 & M_{CB}{}^{(\beta)} \end{bmatrix} S \quad \text{and} \quad K_{system} \equiv S^T \begin{bmatrix} K_{CB}{}^{(\alpha)} & 0 \\ 0 & K_{CB}{}^{(\beta)} \end{bmatrix} S$$

where S is the substructure coupling matrix. Methods for calculating ψ_{CB} and S have been presented by Craig [2,4].

2.2 Implementation of CMS in ABAQUS

ABAQUS uses a particular case of the Craig-Bampton CMS method to solve models with substructures. An analyst can specify several levels of substructures with the top level substructures feeding the top-level model. Only one level of substructures was used in the work for this paper.

As specified by the Craig-Bampton method, ABAQUS uses constraint modes and fixed-interface normal modes to define the component transformation matrix. The ABAQUS user specifies the constraint modes in the transformation matrix by specifying the substructure's retained DOF, that is, the substructure's boundary DOF that are included in the top-level model. The DOF that are not retained are referred to as eliminated DOF. The user specifies the fixed-interface normal modes by performing a frequency extraction and specifying which of the eigenmodes should be included in the top-level model. The implementation of CMS in ABAQUS is depicted in

Figure 2.1. Details of generating substructures and using substructures in models can be found in the ABAQUS Analysis User's Manual [5].

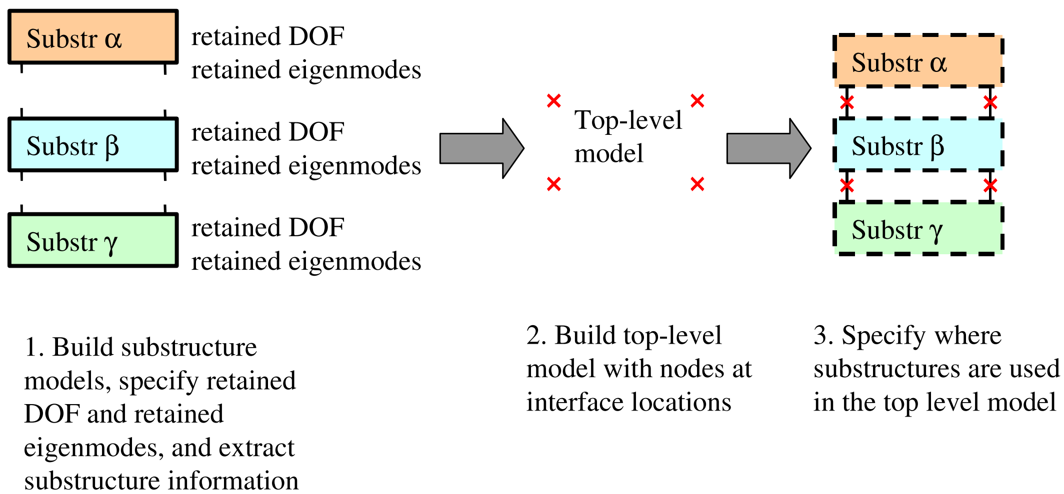

Figure 2.1 Implementation of CMS in ABAQUS

CHAPTER 3

CMS APPLIED TO A SMALL MODEL OF A SIMPLY-SUPPORTED BEAM

3.1 Beam Model Setup

The classical, simply-supported beam structure of Figure 3.1 was chosen to begin comparing the accuracy of a substructured model to the accuracy of a full model with an equivalent mesh. The beam is 100 inches long, has a circular cross-section with an area of one square inch, and is made from a generic steel material that has the following properties: density = 0.3 lbm/in^3, modulus of elasticity = 30 Mpsi, and Poisson's ratio = 0.3. The finite element models were created in two-dimensional space with three active DOF.

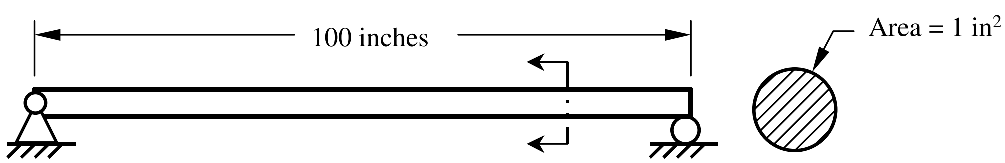

Figure 3.1 Simply-supported beam structure

3.2 Varying the Number of Substructures in the Beam Model

First, a full model of the structure was created to establish the reference for natural frequency and mode shape accuracy. The beam was meshed with twenty-one nodes and twenty, two-noded cubic beam elements. Next, three substructured models were created by dividing the model into two, four, and five substructures. These models are shown in Figure 3.2. The divisions were selected to preserve the mesh density. Substructures were generated, retaining two fixed-interface normal modes and retaining all constraint modes.

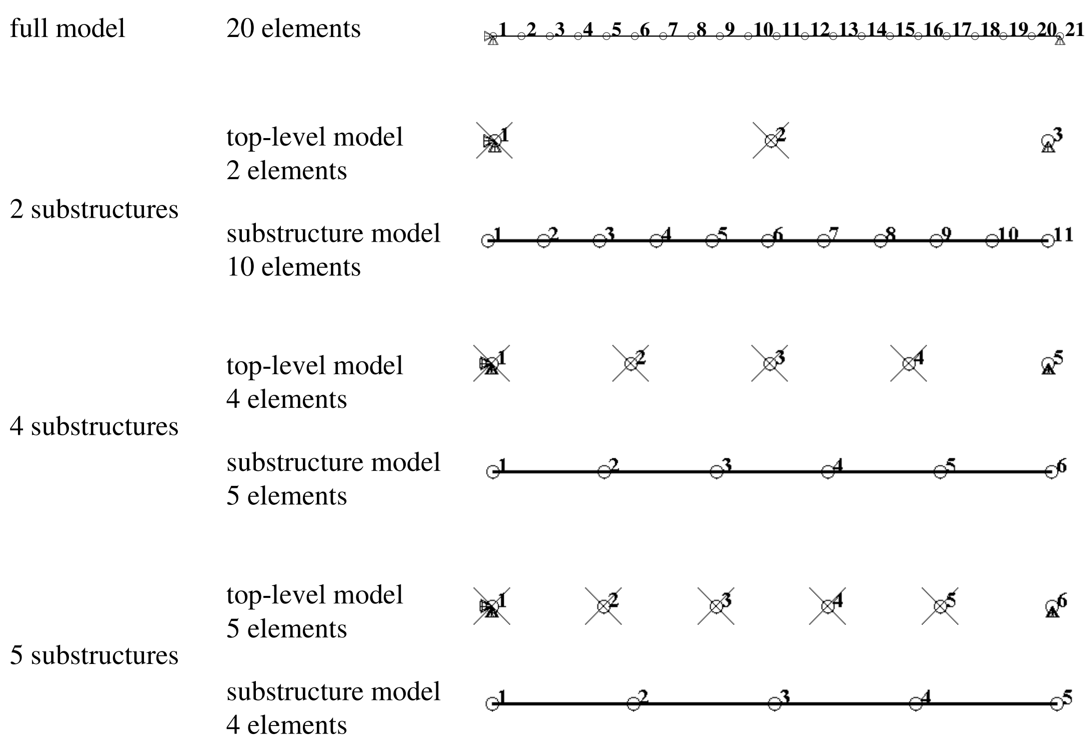

Figure 3.2 Full and substructured models of the simply-supported beam

The full and substructured models were solved and the resulting frequencies were compared as shown in Table 3.1. All errors in frequency were below one percent, except for the frequency of mode five for the two-substructure model. The vertical-direction generalized displacements were used to create plots for a comparison of the mode shapes. These are shown in Figures 3.3 through 3.7. In general, the differences in mode shapes appeared indistinguishable, except for that of mode five where the shape of the two-substructure model appeared to differ greatly.

Table 3.1 Natural frequencies for a varying number of substructures of the beam model

Mode	Frequency of Full Model (Hz)	Frequency of Substructured Models (2 Eigenmodes Retained)					
		2 Substructures Used		4 Substructures Used		5 Substructures Used	
		Frequency (Hz)	Error (%)	Frequency (Hz)	Error (%)	Frequency (Hz)	Error (%)
1	8.7058	8.7060	0.002	8.7058	0.000	8.7058	0.000
2	34.824	34.826	0.006	34.824	0.000	34.824	0.000
3	78.355	78.427	0.092	78.360	0.006	78.357	0.003
4	139.31	139.40	0.065	139.32	0.007	139.32	0.007
5	217.70	227.31	4.414	217.75	0.023	217.72	0.009

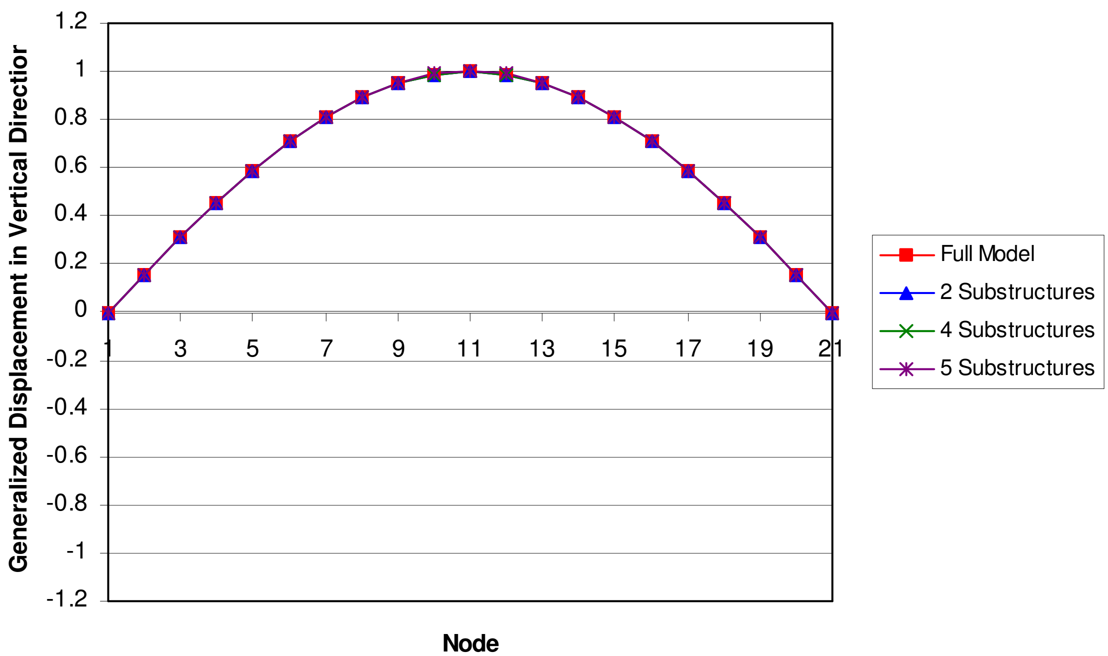

Figure 3.3 Mode shape 1 of the beam models for varying numbers of substructures using two retained eigenmodes

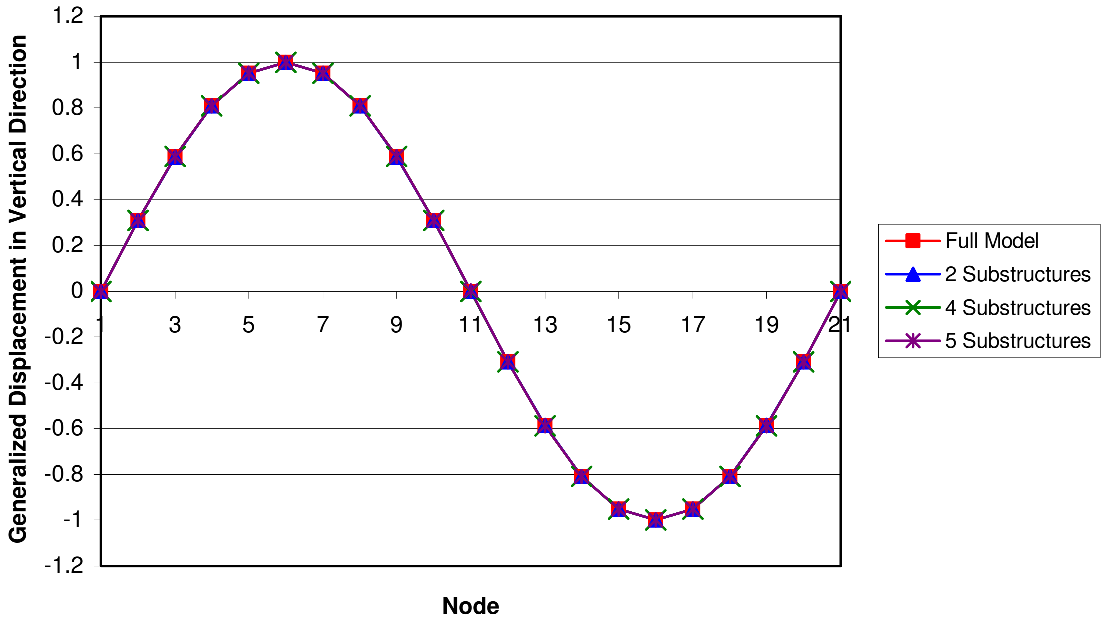

Figure 3.4 Mode shape 2 of the beam models for varying numbers of substructures using two retained eigenmodes

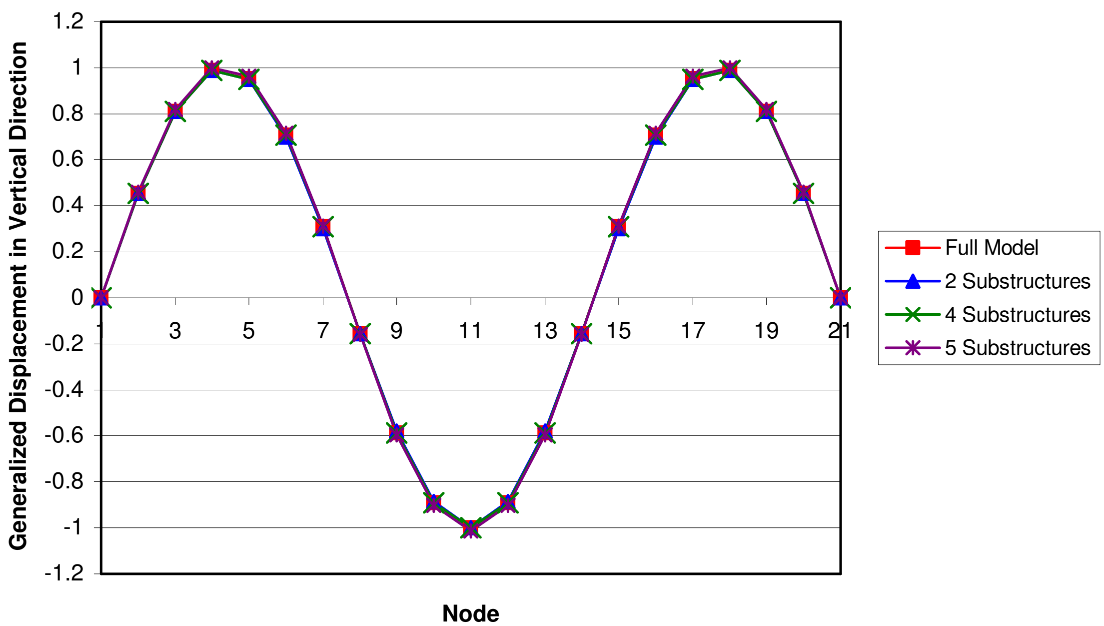

Figure 3.5 Mode shape 3 of the beam models for varying numbers of substructures using two retained eigenmodes

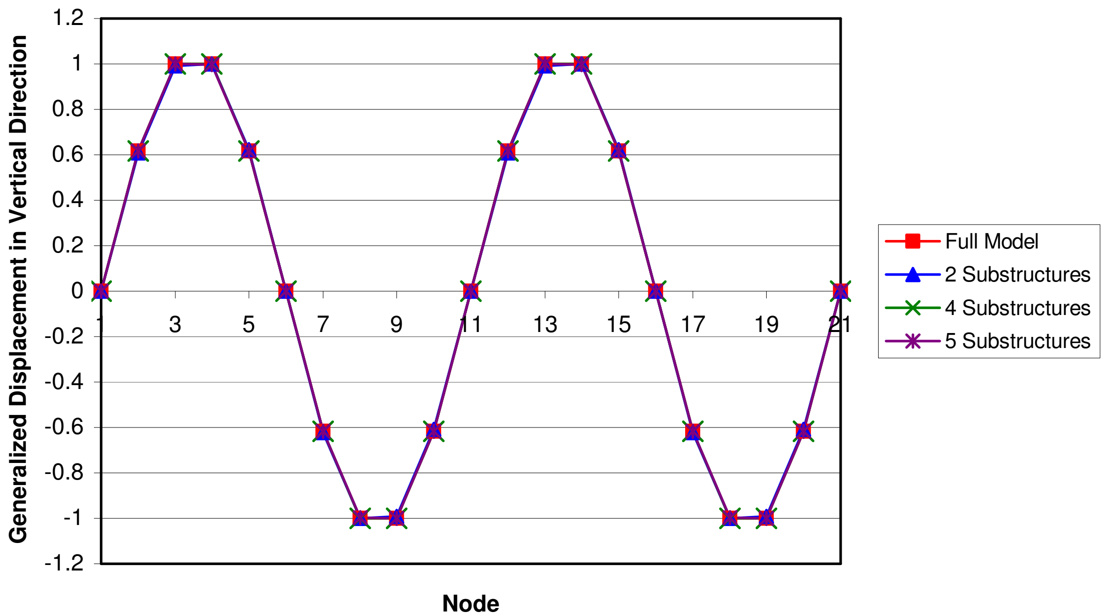

Figure 3.6 Mode shape 4 of the beam models for varying numbers of substructures using two retained eigenmodes

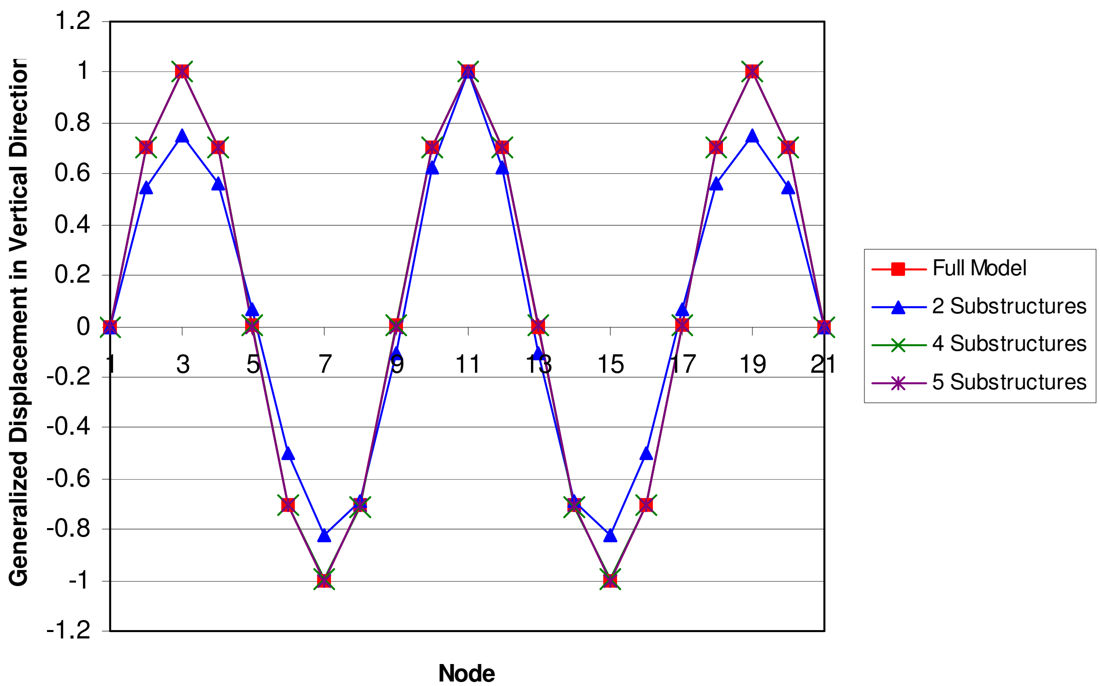

Figure 3.7 Mode shape 5 of the beam models for varying numbers of substructures using two retained eigenmodes

3.3 Varying the Number of Retained Eigenmodes for the Substructures

Considering the errors observed in the two-substructure model, additional solutions were obtained for this model with a varying number of retained fixed-interface normal modes. These results are given in Table 3.2 and show that the accuracy improves significantly as more eigenmodes are retained. Retaining three eigenmodes resulted in a frequency error of less than one percent. Figures 3.8 through 3.12 show the mode shapes for the two-substructure model using the vertical-direction generalized displacements. These plots indicate that only one eigenmode is needed to reasonably describe modes one and two, two eigenmodes are needed to describe modes three and four, and three eigenmodes are needed to describe mode five.

Table 3.2 Natural frequencies of the two-substructure beam model for a varying number of retained eigenmodes

Mode	Frequency of Full Model (Hz)	Frequency of Substructured Models (2 Substructures Used)									
		1 Eigenmode Retained		2 Eigenmodes Retained		3 Eigenmodes Retained		4 Eigenmodes Retained		5 Eigenmodes Retained	
		Frequency (Hz)	Error (%)	Frequency (Hz)	Error (%)	Frequency (Hz)	Error (%)	Frequency (Hz)	Error (%)	Frequency (Hz)	Error (%)
1	8.7058	8.7066	0.009	8.7060	0.002	8.7059	0.001	8.7058	0.000	8.7058	0.000
2	34.824	34.826	0.006	34.826	0.006	34.824	0.000	34.824	0.000	34.824	0.000
3	78.355	80.610	2.878	78.427	0.092	78.397	0.054	78.363	0.010	78.360	0.006
4	139.31	177.12	27.141	139.40	0.065	139.40	0.065	139.31	0.000	139.31	0.000
5	217.70	297.11	36.477	227.31	4.414	218.13	0.198	218.04	0.156	217.77	0.032

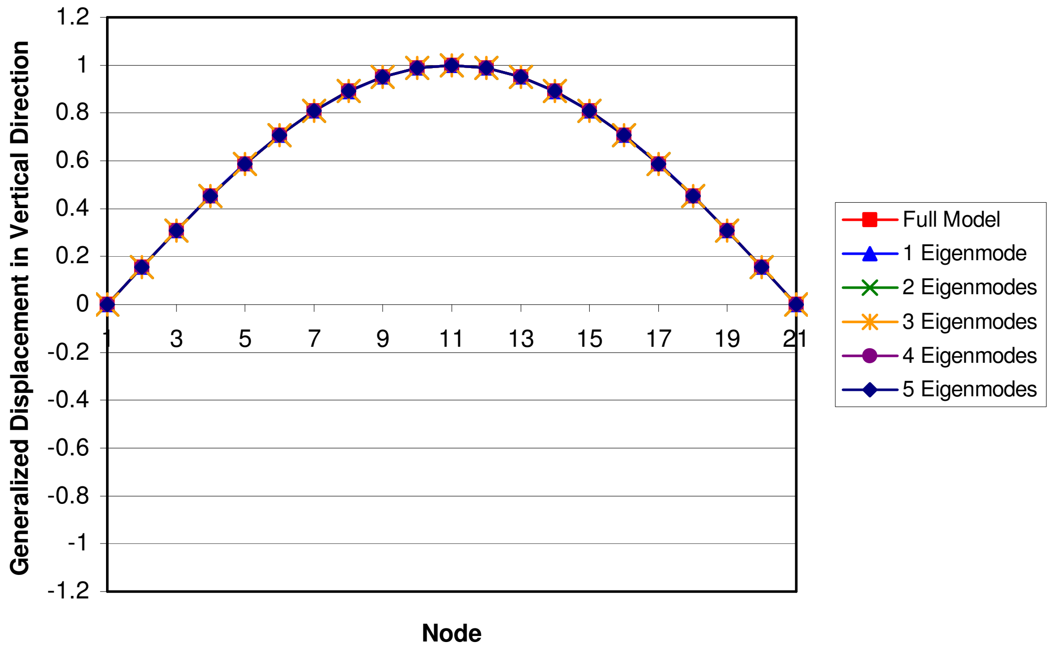

Figure 3.8 Mode shape 1 of the full beam model and two-substructure beam model using varying numbers of retained eigenmodes

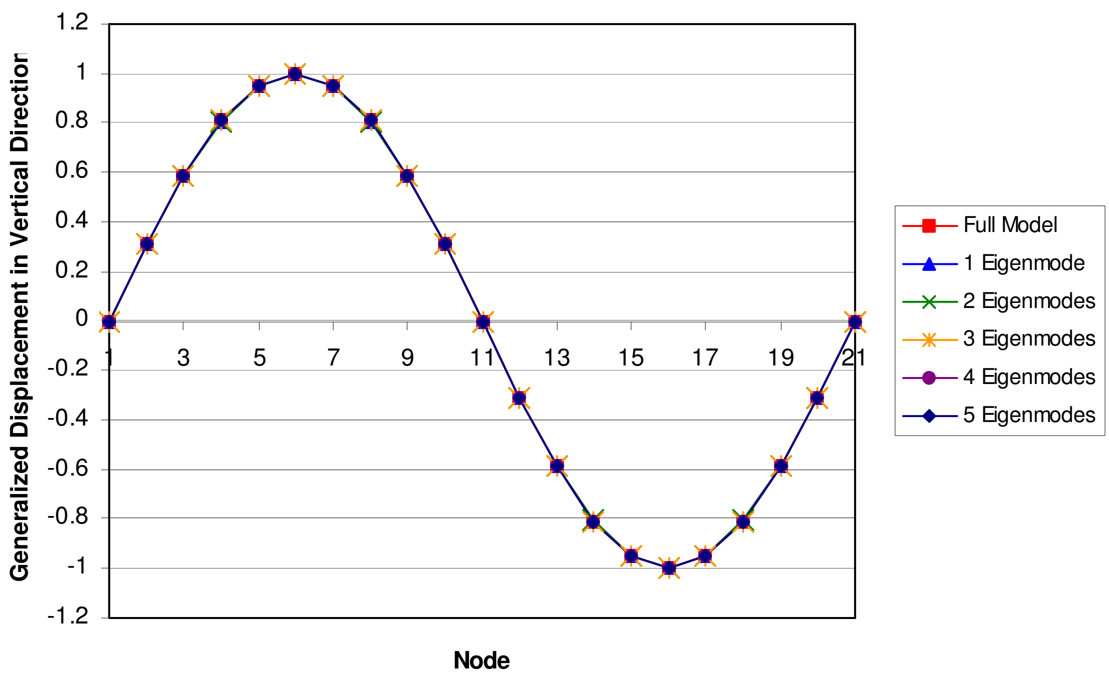

Figure 3.9 Mode shape 2 of the full beam model and two-substructure beam model using varying numbers of retained eigenmodes

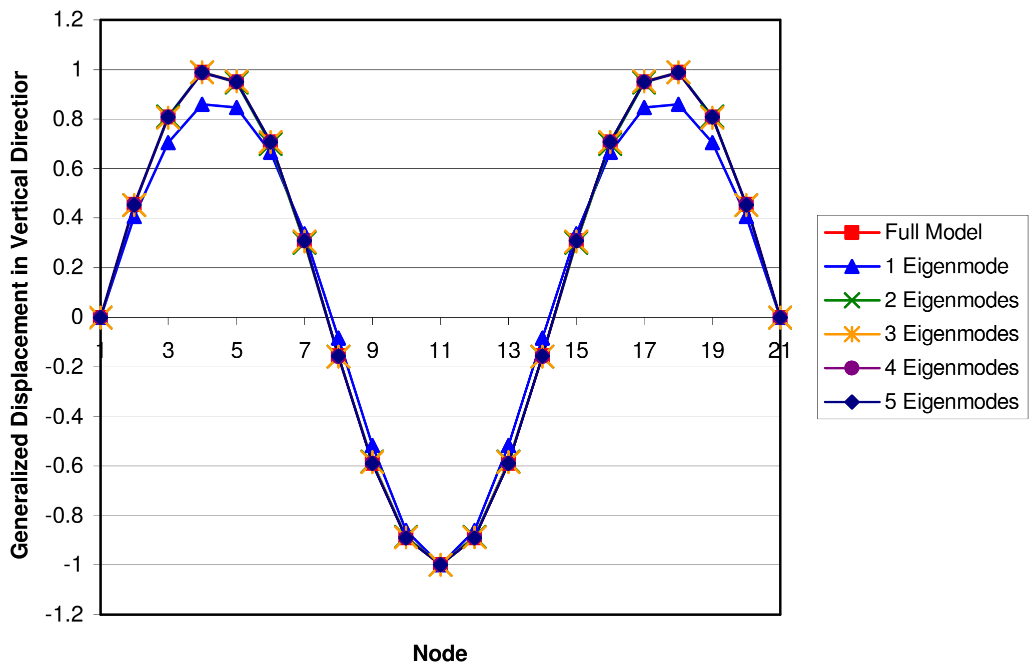

Figure 3.10 Mode shape 3 of the full beam model and two-substructure beam model using varying numbers of retained eigenmodes

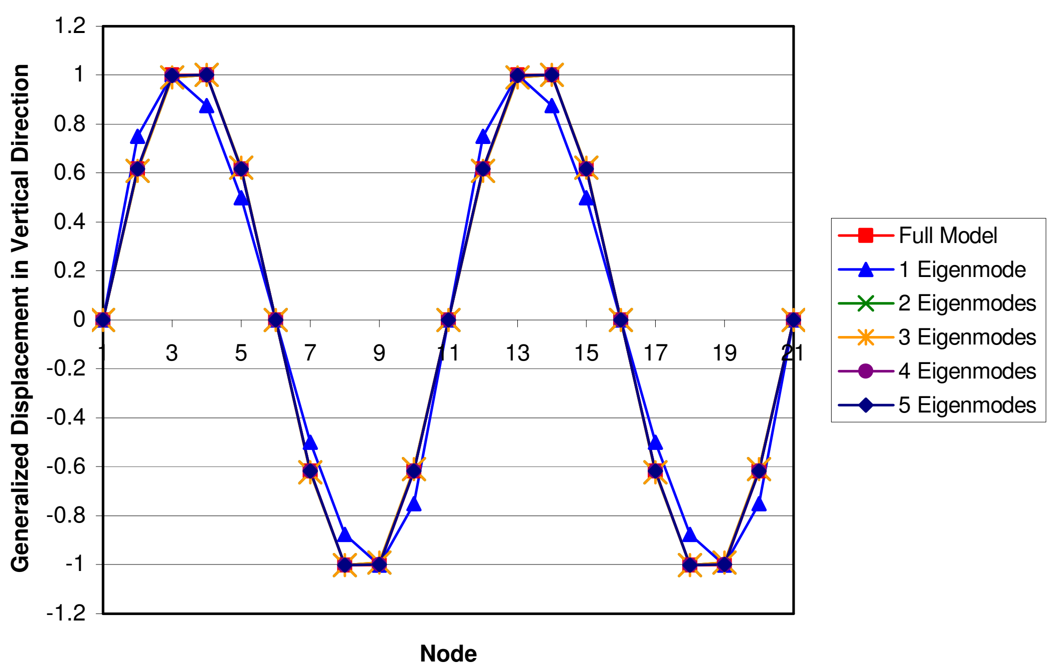

Figure 3.11 Mode shape 4 of the full beam model and two-substructure beam model using varying numbers of retained eigenmodes

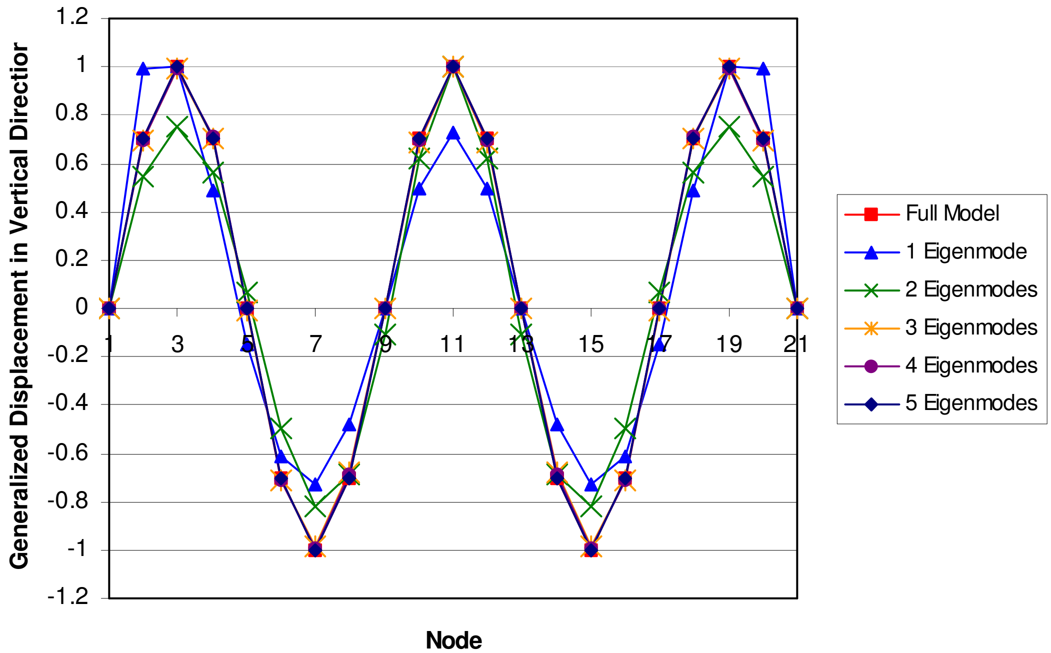

Figure 3.12 Mode shape 5 of the full beam model and two-substructure beam model using varying numbers of retained eigenmodes

Since differences in mode shape are difficult to quantify with these plots, it was determined that the generalized masses for each mode should be compared. These values provide a single, quantifiable measure of the mode shape. The generalized mass values and errors are listed in Tables 3.3 and 3.4.

Table 3.3 Generalized mass values for a varying number of substructures of the beam model

Mode	Generalized Mass of Full Model (lbf-sec^2 / in)	Generalized Mass of Substructured Models (2 Eigenmodes Retained)					
		2 Substructures Used		4 Substructures Used		5 Substructures Used	
		Generalized Mass (lbf-sec^2/in)	Error (%)	Generalized Mass (lbf-sec^2/in)	Error (%)	Generalized Mass (lbf-sec^2/in)	Error (%)
1	0.038860	0.038857	-0.009	0.038860	-0.001	0.038894	0.088
2	0.038859	0.038780	0.039	0.038856	-0.008	0.038892	0.085
3	0.038855	0.038570	-0.733	0.038845	-0.025	0.038810	-0.115
4	0.042944	0.042579	0.043	2.459200	0.043	0.042848	-0.224
5	0.038820	0.027047	-30.327	0.038745	-0.193	0.038742	-0.201

Table 3.4 Generalized mass values of the two-substructure beam model for a varying number of retained eigenmodes

Mode	Generalized Mass of Full Model (lbf-sec^2/in)	Generalized Mass of Substructured Models (2 Substructures Used)									
		1 Eigenmode Retained		2 Eigenmodes Retained		3 Eigenmodes Retained		4 Eigenmodes Retained		5 Eigenmodes Retained	
		General-ized Mass (lbf-sec^2/in)	Error (%)	General-ized Mass (lbf-sec^2/in)	Error (%)	General-ized Mass (lbf-sec^2/in)	Error (%)	General-ized Mass (lbf-sec^2/in)	Error (%)	General-ized Mass (lbf-sec^2/in)	Error (%)
1	0.038860	0.038847	-0.035	0.038857	-0.009	0.038859	-0.002	0.038860	-0.001	0.038860	0.000
2	0.038859	0.038780	-0.204	0.038780	-0.204	0.038885	0.068	0.038885	0.068	0.038837	-0.057
3	0.038855	0.032555	-16.21	0.038570	-0.733	0.038718	-0.351	0.038826	-0.073	0.038840	-0.039
4	0.042944	0.040158	-6.488	0.042579	-0.851	0.042579	-0.851	0.042889	-0.128	0.042889	-0.128
5	0.038820	0.030348	-21.82	0.027047	-30.33	0.037987	-2.145	0.038296	-1.349	0.038621	-0.512

CHAPTER 4

CMS APPLIED TO A MEDIUM-SIZED MODEL OF AN ELECTRONICS ENCLOSURE

4.1 Introduction to the Electronics Enclosure Assembly

The electronics enclosure assembly shown in Figure 4.1 was selected to evaluate the use of CMS because it is typical of a structure that would require a medium-sized finite element model. Also, it can easily be divided into substructures that would represent subassemblies that are designed by separate engineering organizations. The outer structure is an enclosure chassis with a cover. Modules B1 and B2 are fastened to the bottom of the mounting plate. This mounting plate is fastened to bosses on the inside of the chassis and is used to support all of the internal modules. Modules A1 and A2 are fastened to the top of the mounting plate, and the module of greatest interest, the electronics unit (EU), is fastened to the top of modules A1 and A2. The EU consists of a housing and cover with three printed circuit boards (PCBs): the main board and two mezzanine boards. The EU assembly is shown in Figure 4.2. The mezzanine boards are fastened to the main board using steel standoffs.

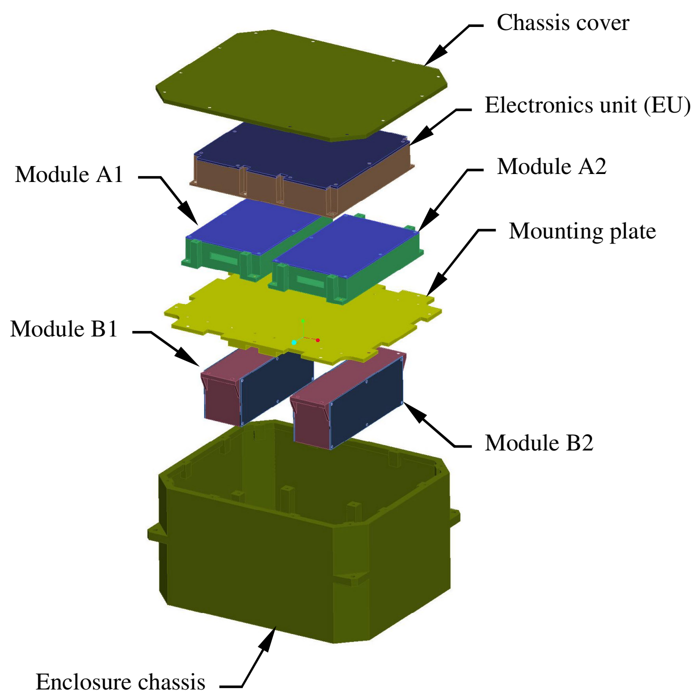

Figure 4.1 Exploded view of the electronics enclosure assembly

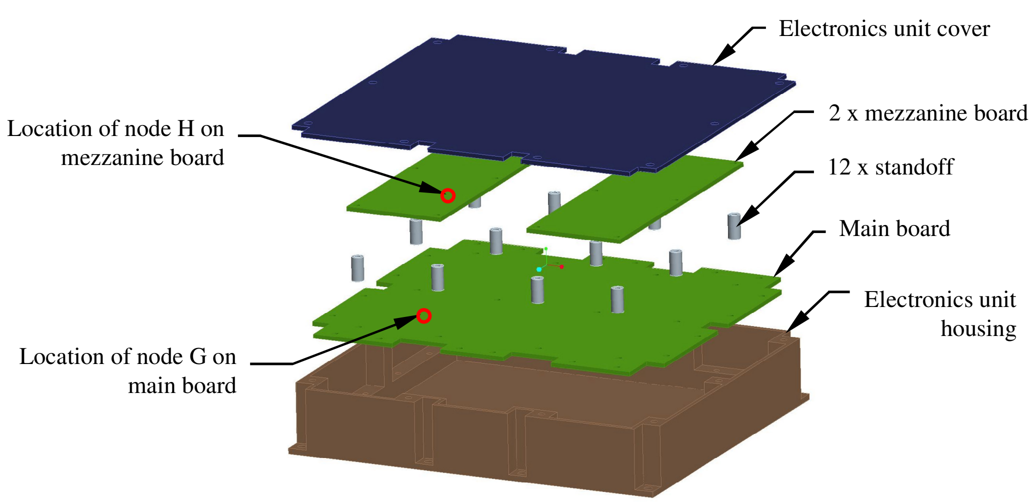

Figure 4.2 Exploded view of the electronics unit

The enclosure chassis is fixed to ground at four holes, each hole being on a lug at a corner of the enclosure. The chassis cover and each of the internal parts are held in place by fasteners that go through clearance holes and into tapped holes. For this model, these fasteners were considered to be completely rigid from the tapped hole to the fastener head, but allowed to freely rotate between the fastener head and the clearance hole.

The materials and material properties for all of the parts are listed in Table 4.1. In general these materials are commonly used in electronics assemblies. The material properties for PCBs can vary greatly, so the values listed in the table are typical homogeneous material values that might be assumed to represent the entire PCB assembly. The internal electronics in modules A1, A2, B1, and B2 are not represented in the models, so the density values of the housings for these modules were increased to account for the weight of these electronics.

Table 4.1 Material properties used in the electronics enclosure models

Parts	Material	Density used in model (lbm/in^3)	Modulus of elasticity (Mpsi)	Poisson's ratio
Enclosure chassis	Aluminum Alloy A356	0.097	10.4	0.33
Chassis cover	Aluminum Alloy 6061	0.098	9.9	0.33
Mounting Plate	Aluminum Alloy 6061	0.098	9.9	0.33
Module A1 and A2 housings	Aluminum Alloy A356	0.250	10.4	0.33
Module A1 and A2 covers	Aluminum Alloy 6061	0.098	9.9	0.33
Module B1 and B2 housings	Aluminum Alloy A356	0.250	10.4	0.33
Module B1 and B2 covers	Aluminum Alloy 6061	0.098	9.9	0.33
Electronics unit housing	Aluminum Alloy A356	0.097	10.4	0.33
Electronics unit cover	Aluminum Alloy 6061	0.098	9.9	0.33
Electronics unit PCBs	Generic PCB	0.150	2.5	0.12
Standoffs	Generic Steel	0.290	29.0	0.30

4.2 Criteria for Evaluation

Given the size and complexity of this model, only one substructured version was built. In order to evaluate the eigenmode accuracy of this substructured model, the natural frequencies and generalized masses were compared to that of the full model for 100 modes.

In addition to evaluating eigenmode accuracy, a displacement measure was defined and used to compare the response of the models to a base motion input. The displacement measure selected was the relative displacement between the z-axis DOF of node G on the EU main board and the z-axis DOF of node H on one of the EU mezzanine boards. The locations of these nodes are shown in Figure 4.2. These two DOF were selected because they contributed only slightly to the strain energy of the parts and had closely spaced peaks in the frequency response plots, making them sensitive to errors in the modes shapes. The response of this displacement measure was calculated using mode-superposition for a base motion, random vibration input of 0.001 G^2/Hz from five to 2000 Hz.

4.3 Setup of the Electronics Enclosure Models

First, a full model of the enclosure was created to establish a baseline for the evaluation criteria. The number and type of elements used for each part are listed in Table 4.2.

Table 4.2 Mesh details of the electronics enclosure finite element models

Parts	Name of ABAQUS® element used	Description of element	Number of elements in each part
Enclosure chassis	C3D10M	10-node modified quadratic tetrahedron	34580
Chassis cover	S4R	4-node doubly-curved thick or thin shell, reduced integration	2933
Mounting Plate	C3D10M	10-node modified quadratic tetrahedron	45551
Module A1 and A2 housings	C3D10M	10-node modified quadratic tetrahedron	9776
Module A1 and A2 covers	S4R	4-node doubly-curved thick or thin shell, reduced integration	909
Module B1 and B2 housings	C3D10M	10-node modified quadratic tetrahedron	13089
Module B1 and B2 covers	S4R	4-node doubly-curved thick or thin shell, reduced integration	551
Electronics unit housing	C3D10M	10-node modified quadratic tetrahedron	29554
Electronics unit cover	S4R	4-node doubly-curved thick or thin shell, reduced integration	2360
Electronics unit main board	S4R	4-node doubly-curved thick or thin shell, reduced integration	4609
Electronics unit mezzanine board	S4R	4-node doubly-curved thick or thin shell, reduced integration	459
Standoffs	B33	2-node cubic beam in 3-D space	5
Fasteners	CONN3D2	2-node connector in 3-D space	1

Next, the full model was divided into top-level and substructure models as shown in Figure 4.3. The substructures for module A and module B were each used twice. In an effort to eliminate concerns about error due to mesh density, the mesh for all of the parts was kept identical to that in the full model.

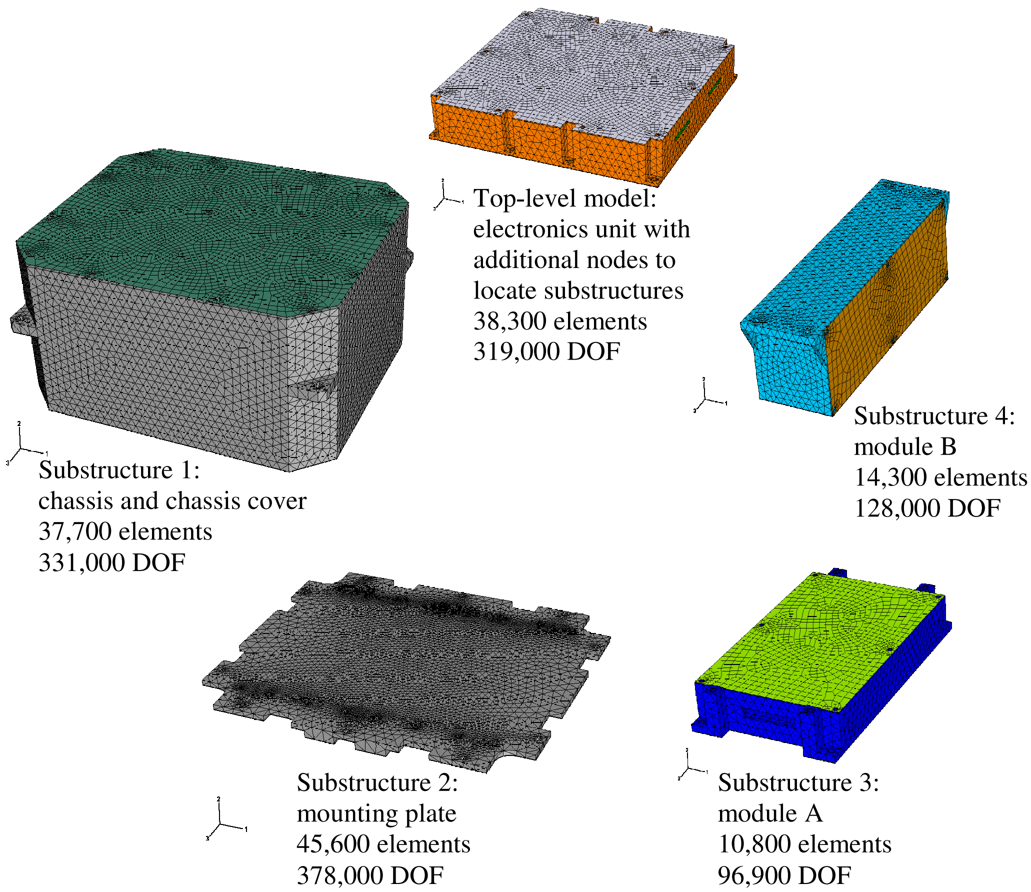

Figure 4.3 Divisions used in the electronics enclosure substructured model

A frequency extraction analysis was performed for each substructure to identify all of the fixed-interface normal modes within the input spectrum of five to 2000 Hertz. Substructure generation procedures were run where all of the fixed-interface normal modes and all of the interface constraint modes were retained for each.

The top-level model was constructed of the EU parts along with additional nodes at each substructure interface point. User defined elements, or super-elements, were defined at these nodes to specify which substructure was used at that location.

4.4 Frequency and Generalized Displacement Results

Frequencies and mode shapes were calculated for the first 100 modes of the full model and the top-level substructured model using the ABAQUS Lanczos eigensolver. The resulting frequency and generalized mass for each mode of both models are listed in Table 4.3. The shading in the table changes every 100 Hz to aid in identifying frequency bands. All of the errors in frequency are shown to be less than two percent, but the generalized mass values for many of the higher order modes had very large errors. However, a visual comparison of the mode shapes revealed that modal displacements often occurred at different frequencies in the substructured model than in the full model. Sometimes, this is due to symmetrical modes being in a different order. Consideration was given to reordering the modes of the substructured model so that the mode shapes would compare most closely with those of the full model. This process proved to be very subjective, so the results were left as originally given.

Table 4.3 Frequencies and generalized mass values for the enclosure models

Mode	Full Model		Substructured Model and Error Values			
	Frequency (Hz)	Generalized Mass (lbf-sec^2 / in)	Frequency (Hz)	Percent Error (%)	Generalized Mass (lbf-sec^2 / in)	Percent Error (%)
1	294.94	0.000831	295.00	0.02	0.000828	-0.44
2	304.21	0.000645	304.21	0.00	0.000645	0.00
3	313.16	0.000569	313.17	0.00	0.000571	0.26
4	329.69	0.001526	329.69	0.00	0.001526	-0.04
5	332.83	0.000408	332.85	0.01	0.000407	-0.04
6	333.81	0.000435	333.82	0.00	0.000435	-0.08
7	379.68	0.000480	379.68	0.00	0.000480	0.00
8	408.34	0.000268	408.34	0.00	0.000268	-0.03
9	409.21	0.000269	409.21	0.00	0.000269	-0.13
10	415.20	0.001997	416.05	0.20	0.001927	-3.49

Table 4.3 - *Continued*

Mode	Full Model		Substructured Model and Error Values			
	Frequency (Hz)	Generalized Mass ($lbf\text{-}sec^2$ / in)	Frequency (Hz)	Percent Error (%)	Generalized Mass ($lbf\text{-}sec^2$ / in)	Percent Error (%)
11	425.30	0.000338	425.30	0.00	0.000338	0.00
12	436.86	0.002758	438.94	0.48	0.002731	-0.98
13	474.47	0.001844	474.72	0.05	0.001854	0.53
14	485.74	0.001658	488.29	0.53	0.001739	4.90
15	503.69	0.000591	503.69	0.00	0.000591	-0.08
16	517.59	0.001097	517.85	0.05	0.001039	-5.26
17	518.70	0.000447	518.71	0.00	0.000447	-0.02
18	518.92	0.000420	518.92	0.00	0.000414	-1.46
19	519.70	0.000447	519.70	0.00	0.000446	-0.06
20	519.70	0.000435	519.70	0.00	0.000433	-0.45
21	571.48	0.007581	572.95	0.26	0.007411	-2.24
22	577.84	0.000322	577.91	0.01	0.000321	-0.11
23	614.67	0.001029	614.75	0.01	0.001030	0.03
24	622.67	0.000448	622.70	0.00	0.000449	0.17
25	630.52	0.000315	630.54	0.00	0.000316	0.05
26	643.60	0.002281	648.60	0.78	0.001979	-13.23
27	653.90	0.008824	660.13	0.95	0.007745	-12.22
28	689.79	0.000469	689.79	0.00	0.000470	0.28
29	689.79	0.000469	689.79	0.00	0.000470	0.26
30	694.12	0.000407	694.59	0.07	0.000407	0.06
31	719.92	0.000691	720.24	0.04	0.000611	-11.54
32	723.87	0.000144	723.92	0.01	0.000140	-2.33
33	730.88	0.000499	731.34	0.06	0.000496	-0.68
34	731.52	0.000239	731.62	0.01	0.000152	-36.24
35	731.59	0.000347	731.73	0.02	0.000699	101.79
36	733.73	0.000895	735.85	0.29	0.000987	10.24
37	739.45	0.000071	739.50	0.01	0.000081	13.25
38	739.51	0.000072	739.55	0.01	0.000079	9.67
39	739.69	0.000601	740.06	0.05	0.000599	-0.45
40	740.20	0.001558	740.32	0.02	0.001607	3.16
41	754.10	0.000084	754.10	0.00	0.000084	-0.08
42	754.16	0.000084	754.18	0.00	0.000084	0.05
43	770.05	0.000135	770.06	0.00	0.000134	-0.81
44	770.08	0.000136	770.08	0.00	0.000134	-0.85
45	819.20	0.002092	821.85	0.32	0.001840	-12.05
46	831.70	0.001588	832.91	0.15	0.001481	-6.73

Table 4.3 - *Continued*

Mode	Full Model		Substructured Model and Error Values			
	Frequency (Hz)	Generalized Mass (lbf-sec^2 / in)	Frequency (Hz)	Percent Error (%)	Generalized Mass (lbf-sec^2 / in)	Percent Error (%)
47	841.72	0.000568	845.63	0.46	0.000202	-64.36
48	848.84	0.000277	862.16	1.57	0.002905	948.04
49	871.04	0.000605	871.05	0.00	0.000605	0.05
50	873.57	0.001353	873.89	0.04	0.001352	-0.09
51	893.15	0.000166	893.32	0.02	0.000166	0.13
52	911.19	0.000375	911.68	0.05	0.000349	-6.86
53	914.31	0.000329	916.74	0.27	0.000349	6.03
54	927.28	0.000507	937.57	1.11	0.000280	-44.76
55	935.44	0.000570	939.00	0.38	0.000139	-75.57
56	939.20	0.000086	942.16	0.31	0.000313	266.25
57	939.31	0.000088	950.98	1.24	0.000501	471.56
58	996.79	0.000210	996.85	0.01	0.000210	0.37
59	1002.41	0.000199	1003.92	0.15	0.000186	-6.55
60	1023.22	0.001884	1028.41	0.51	0.001615	-14.28
61	1028.26	0.001518	1044.08	1.54	0.003581	135.87
62	1060.36	0.000235	1061.03	0.06	0.000223	-5.16
63	1076.74	0.000272	1078.30	0.14	0.000160	-41.07
64	1078.62	0.000158	1078.64	0.00	0.000158	-0.26
65	1080.67	0.000126	1080.68	0.00	0.000157	24.78
66	1080.68	0.000157	1080.68	0.00	0.000159	1.29
67	1080.92	0.000215	1090.23	0.86	0.002396	1012.21
68	1101.01	0.001190	1114.24	1.20	0.000863	-27.49
69	1114.04	0.000455	1119.87	0.52	0.000344	-24.54
70	1137.27	0.000249	1137.67	0.04	0.000250	0.06
71	1146.95	0.001148	1149.50	0.22	0.000226	-80.33
72	1149.39	0.000220	1171.41	1.92	0.000260	18.33
73	1172.38	0.000249	1175.95	0.30	0.000098	-60.52
74	1176.49	0.000095	1178.26	0.15	0.000092	-2.99
75	1177.64	0.000093	1187.69	0.85	0.001166	1153.85
76	1226.88	0.000371	1234.99	0.66	0.001076	190.02
77	1229.51	0.001174	1235.31	0.47	0.001164	-0.83
78	1233.52	0.001234	1235.97	0.20	0.000411	-66.70
79	1240.87	0.000100	1241.20	0.03	0.000105	5.47
80	1242.62	0.000251	1250.14	0.60	0.000202	-19.47
81	1253.32	0.001311	1264.78	0.91	0.001278	-2.50
82	1298.37	0.000365	1299.03	0.05	0.000288	-21.16

Table 4.3 - *Continued*

Mode	Full Model		Substructured Model and Error Values			
	Frequency (Hz)	Generalized Mass (lbf-sec^2 / in)	Frequency (Hz)	Percent Error (%)	Generalized Mass (lbf-sec^2 / in)	Percent Error (%)
83	1301.02	0.000243	1301.54	0.04	0.000192	-20.68
84	1316.90	0.000134	1317.52	0.05	0.000130	-2.82
85	1318.32	0.000136	1318.32	0.00	0.000133	-2.29
86	1318.32	0.000138	1318.32	0.00	0.000133	-3.73
87	1323.85	0.000134	1325.39	0.12	0.000134	-0.12
88	1334.63	0.000056	1334.67	0.00	0.000055	-0.14
89	1358.63	0.001440	1358.65	0.00	0.001439	-0.04
90	1365.58	0.001759	1376.02	0.76	0.000560	-68.18
91	1366.00	0.000460	1380.19	1.04	0.001408	205.89
92	1391.46	0.000137	1393.91	0.18	0.000129	-5.97
93	1408.28	0.000052	1409.29	0.07	0.000056	7.34
94	1423.50	0.000082	1425.47	0.14	0.000060	-26.50
95	1428.26	0.000170	1432.14	0.27	0.000362	112.87
96	1428.32	0.000274	1432.55	0.30	0.000630	129.94
97	1429.38	0.000192	1442.10	0.89	0.000050	-73.91
98	1443.18	0.000046	1447.14	0.27	0.000093	103.06
99	1453.00	0.001089	1460.54	0.52	0.000056	-94.85
100	1458.40	0.000059	1474.36	1.09	0.000074	26.46

4.5 Displacement Response Results

The eigenvalues, eigenvectors, and generalized masses for both the full and substructured enclosure models were transferred from ABAQUS to postprocessing software for response calculations by mode superposition techniques. This transfer was accomplished using Raytheon internally-developed software. The random vibration input of 0.001 G^2/Hz was applied to both models and the frequency response of the displacement measure was calculated. These frequency response curves are shown in Figure 4.4. The calculated RMS values of the displacement measures over the entire input frequency band were 1.891 microinches for the full model and 1.808 microinches for the substructured model. This indicates a 4.4 percent error for the substructured model. To further evaluate error in the displacement measure at various frequencies, the RMS values were calculated for various frequency bands. These results are listed in Table 4.4.

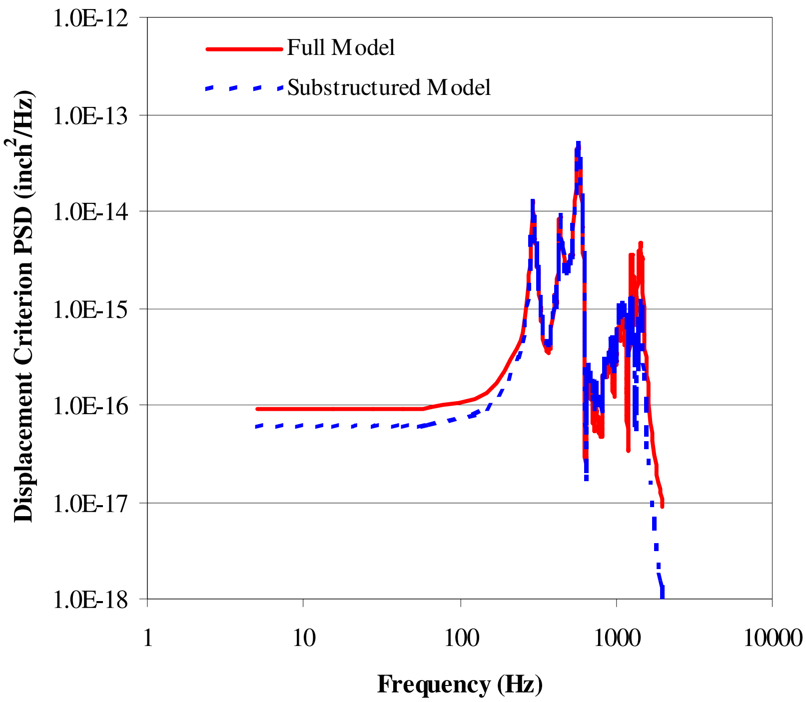

Figure 4.4 Frequency response of the displacement measure for the full and substructured enclosure models

Table 4.4 Comparison of RMS values of the displacement measure for various frequency bands

Frequency Band (Hertz)	RMS of Displacement Criterion for Full Model (micro inch)	RMS of Displacement Criterion for Substructured Model (micro inch)	Error
5-2000	1.891	1.808	-4.4%
5-1460	1.855	1.797	-3.1%
5-640	1.646	1.681	2.1%
640-1460	0.856	0.636	-26%
5-200	0.159	0.132	-17%
201-400	0.588	0.586	-0.3%
401-600	1.455	1.485	2.1%
601-800	0.471	0.514	9.2%
801-1000	0.205	0.240	17%
1001-1200	0.309	0.411	33%
1201-1400	0.579	0.310	-46%
1401-1600	0.599	0.304	-49%
1601-1800	0.119	0.049	-59%
1801-2000	0.056	0.019	-65%

4.6 Finite Element Model Solve Time

One of the questions asked regarding CMS was whether an analyst can save solve time when using substructuring. The electronics enclosure problem resulted in models of appropriate size to make a comparison of solve times. These are listed in Table 4.5 and show that the substructured model, being of reduced-order, was solved in 40 percent less time.

Table 4.5 Solve times for the electronics enclosure models

Model	Solve Time (seconds)	
	Full Model	Substructured Model
Chassis Substructure		221
Mounting Plate Substructure		346
Module A Substructure		52
Module B Substructure		71
Usage-Level with Electronics Unit		605
Total for Enclosure Model	2084	1295

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions from Work Performed

The simply-supported beam models showed that error in the natural frequency and mode shape of the substructured models increases as the order of the mode increases. Increasing or decreasing the number of substructures does not necessarily have a significant impact on accuracy, but increasing the number of eigenmodes retained for the substructures can significantly improve accuracy for higher order modes.

Although the investigation of the beam models provided some indication as to how accuracy changes when substructures are used, it did not indicate how much error might be acceptable. An allowable error of the frequencies and mode shape depends on the desired accuracy of the response when loading conditions are applied to the structure.

A comparison of the electronics enclosure models showed that the frequencies and modes shapes of the substructured model were very accurate for modes up to approximately 640 Hz. Comparing mode shapes and frequencies in the higher order modes proved to be very difficult, or impossible for some modes, because generalized displacements could not readily be matched due to the complexity of the shapes.

The displacement measure of the enclosure substructured model proved to be reasonably accurate from five to 640 Hz and highly accurate between 200 and 640 Hz. The results above 640 Hz showed significant error; however, the contribution of the higher order modes to the total RMS displacement is small.

Considering the many sources of error in finite element models, the error introduced by implementing CMS should be acceptable for many structural dynamics problems. If this amount of error is considered not acceptable, the many other error sources should be considered as well. When these error sources are compared, an analyst may desire to accept the error caused by substructuring and seek to reduce error from other sources.

Beyond the questions addressed in this work is a question regarding the time spent in creating finite element model idealizations. Idealization is a common technique used to reduce the solver time by modeling three-dimensional parts with a mesh of shell and beam elements instead of solid elements. Many parts are easily represented in this manner, such as using shell elements for sheet metal covers and using beam elements for shafts and bolts. However, many times, when the parts are more complex, such as with castings or detailed machine parts, the time and effort spent to create these idealizations can be significant. Also, simply automeshing these parts with solid elements can result in a very large number of DOF. Can the combination of CMS with automeshing reduce modeling time compared to creating idealizations?

It was not feasible to track the time spent creating the substructured model of the electronics enclosure in this present work because much time was required for concurrent investigation and model rework. Nevertheless, it was apparent that automeshing the various housings in the model was much easier than creating idealizations. Considering the significantly reduced solve time that CMS provides and the ability to reuse substructures and simplify model creation, CMS is a good alternative to creating idealizations.

5.2 Recommendations for Applying CMS and Opportunities for Further Investigation

This evaluation has led to the following recommendations for applying CMS.

1. Substructure divisions and boundary locations should be selected to minimize the number of interface points, thereby reducing the modeling effort. Fastener locations provide ideal interface points.
2. The analyst should be careful to match the order of retained DOF for the substructure to the order of nodes specified for the super-element in the top-level model. Otherwise, super-elements may be oriented incorrectly. The ABAQUS documentation discusses techniques to ensure proper ordering [5].
3. An investigation of the mode shapes of the substructures can help the analyst eliminate unnecessary modes and reduce solve time. Although all of the substructure modes were retained for this work, some of the modes were unnecessary. For example, the modes for the module covers could have been eliminated with little impact on the response calculations.

Furthermore, this evaluation has generated questions for further investigation.

1. How does multilevel substructuring differ from the single-level substructuring used in this work and how might it affect the accuracy of response calculations?
2. What methods are available to aid the analyst in selecting fixed-interface normal modes for elimination?
3. What methods are available for estimating the frequency range over which CMS will meet particular accuracy requirements?

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BIOGRAPHICAL INFORMATION

Richard Scott is a structural analyst at Raytheon Company in McKinney, Texas. His engineering interests include structural dynamics, finite element modeling, stress analysis and dynamics testing. He has worked as a mechanical engineer for defense and commercial electronics companies, performing design, thermal analysis, and structural analysis, since graduating from Oklahoma State University in 1992 with the degree of Bachelor of Science in Engineering Technology. He expects to graduate in May of 2007 with the degree of Master of Science in Mechanical Engineering with an emphasis in solid mechanics. He is a Registered Professional Engineer in the state of Texas.