A FINITE ELEMENT METHODOLOGY FOR PREDICTING RELATIVE MOTION OF AVIONICS MODULE CONNECTOR CONTACTS

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ABSTRACT

This paper presents a methodology for modeling and predicting electronic module connector displacements in an advanced avionics system. The system is modelled using finite element theory and the equations of motion solved using MSC/NASTRAN. Emphasis in this paper is placed on the finite element modeling (FEM) approach and reduction scheme. In order to achieve the desired accuracy in analysis, the initial FEM contained over 20,000 degrees of freedom. A FEM of this size is typically cost prohibitive to use and sensitive to numerical instabilities, particularly when the frequency range of interest may be as low as 10 Hertz and exceed 1000 Hertz. Several model reduction schemes and various superelement approaches are examined in an effort to reduce model size and improve numerical stability. Results include guidelines for model reduction of similar type structures as well as selection of the appropriate eigenvalue solver and associated parameters when using MSC/NASTRAN. Of particular interest to the authors was the random response of the connector. To this end, modal analysis is employed to identify natural modes of vibration. Forced frequency response analysis using the large mass method provides transfer functions between the source excitation and the response at the connector. The transfer functions along with a user specified input power spectral density function are used as input to a random analysis. Results include rms displacements and frequencies of the connectors.

INTRODUCTION

Recently, connector reliability has become of increasing concern in the avionics community. The connector has become a significant component in determining system cost, performance, and reliability. Unfortunately, the connector is considered the weak point in many avionics systems.

In the past, design efforts have focused on such characteristics as current capacity, dielectric strength, contact density, shielding, and cost, Ref 1-5. Accumulative vibrations and shock wreaked havoc with many avionics systems and over time caused fatigue failure. System design engineers did not sufficiently address the connector's dynamic environment, and design objectives did not include minimizing the relative motion between connector contacts. According to David Dylis, who manages the Air Force's Field Failure Return Program at Rome Laboratory, vibration, shock, and electrical overload account for up to half of all avionics failures, Ref 6.

In today's advanced digital avionics systems much attention is focused on predicting and minimizing relative motion between connector contacts. Typically such motion is due to high vibration environments or transportation shock spectra and leads to fretting corrosion of the contact base metals. As a result of frictional wear, relative motion between connector contacts removes the protective coatings, typically made of gold or nickel. Subsequent motion between the base metals, along with high contact pressures, lack of lubricants, and presence of oxygen leads to fretting corrosion. The result of fretting is an increase in resistance and eventually intermittent contacts and signal transmission, constituting connector failure.

Failures due to intermittences have proven to be very hard to identify. Often the failure is noted in the operational environment of the aircraft, Ref 4,6. However, when the system is removed and bench tested, the failure is no longer present. According to Dylis, 35% of the failed components show no problem when retested, Ref 4. Buf Slay, manager of quality and reliability for military semiconductors at Texas Instruments, estimates 50-70% of the parts they receive test okay, Ref 4. Often the system must be tested on a ''shake and bake'' device that simulates flight vibration and shock, requiring expensive and specialized facilities.

This paper presents an analytical methodology for modeling and predicting the relative motion of avionics module connector contacts. The avionics system considered consist of 3 bays containing nearly 100 line removable units (LRMs) or modules. The finite element modeling approach and reduction scheme is discussed in detail. Guidelines for FEM reduction and modeling are presented. Observations as related to predicting connector contact motion using MSC/NASTRAN are also discussed.

SYSTEM DESCRIPTION

The avionics system considered in this study was quite large, measuring 21.5x9x21.5 inches and weighing approximately 200 pounds. The system was composed of 3 bays, each containing as many as 33 modules or LRMs. An avionics system, with no LRMs, similar to the one in this study is seen in Figure 1. A module consists of integrated circuits mounted on printed circuit boards which are laminated to an aluminum heat sink, Figure 2. Modules are connected to a backplane by ultra high density connectors, each having over 370 pins or contacts. Half of the connector is fastened to the module heat sink, the other half is fixed to the backplane. Modules are held in place using camming type devices which clamp the modules at the heat sink/coldwall interface.

The backplane is a multilayer composite mixture with integrated circuitry made of copper. The avionics system considered in this study consists of 4 backplanes. The upper bay consists of 2 backplanes separated at the center of the bay. The mid and lower bays each have one backplane spanning their respective bay periphery. Fasteners are used to attach the backplanes to the rack structure.

The shelves or coldwalls have very complex cross sections which utilized both liquid flow through and air flow through cooling, Figure 3. The system is hermetically sealed using front and rear covers. The avionics system also contains nonstructural mass items such as a power conditioner, I/O devices, wiring, splitters, and filters.

FINITE ELEMENT MODEL

The FEM was constructed for the purpose of predicting module/rack interface transmissibilities, backplane motion, and connector contact motion associated with a random environment. The frequency range of interest was 10-1000 Hertz. Objectives also included studying the effects of partially populated racks, as well as a parametric study to examine the effects of varying damping and stiffness of both the backplanes and the aircraft/rack interface structure.

The initial FEM of the avionics rack consisted of over 20,000 degrees of freedom. The FEM included a finite element representation of each module which were modeled using 2D plate elements, CQUAD4s. Physical properties associated with the modules were included using the PCOMP property card. This allowed easy representation of the circuit board and heat sink. Test data indicated the fundamental mode of vibration, corresponding to a 1.5 lb. module clamped at the coldwall interface, to be 576 Hz. with 2% structural damping. Material properties of the module were calculated to match this test data assuming clamped boundary conditions at the coldwall/module interface. The module physical properties were further adjusted to account for the difference in size when comparing a physical module to one in the FEM. The size difference resulted from the coldwalls also being modeled using 2D plate elements. Nodes on the module periphery are located on the coldwall midplanes. The FEM modules are thereby larger than the physical modules by half the thickness of the coldwall midplanes and thus would be too flexible with identical material and physical properties.

Since few modules in the avionics system are identical, but, most are similar, exact correlation with the test data was not considered essential. For simplicity, each module in the FEM was considered identical.

The backplanes are multilayer circuit boards where copper is used for circuit routing. Material properties were supplied by the manufactures. Large mass items such as couplers, splitters, filter boxes, and I/O connectors were lumped by hand. Wiring was assumed uniformly distributed with 1/2 of the wiring mass distributed to the backplanes and the remaining mass distributed to the I/O connector on the backcover. The backcover is an environmental seal made of aluminum, and is attached to the rack and backplane using multipoint constraint (mpc) relationships. Additional mpcs were used to connect the backplane to the rack. Using mpcs relationships allowed the backcover and backplane to be attached to one another and to the rack at intermittent locations. Also, the mpcs more accurately represented the interface connection by not transferring bending moments between the rack, backplane, and backcover. A frontcover was attached in a similar fashion.

Connectors were modeled using 2D plate and beam elements. The physical connector is composed of a .02 inch aluminum housing surrounding an interstructure containing high density contacts. The load path is such that only bending and transverse shear loads are transferred between connector halves. The aluminum housing was represented by beam elements. Beam elements span one side of each module and the adjacent backplane. Test data indicted that the connector undergoes a 0.02 inch out of plane displacement under a 60 lb side force. The material properties of the interstructure were determined from this information. The interstucture was assumed to only carry bending and transverse shear.

The sliding friction force due to the connector contact was represented by structural damping in the backplanes. The primary influence of damping is to limit the amplitude of the response at resonance. For an oscillatory system, damping has little influence on the response in frequency regions away from resonance. A simplistic approach to represent the Coulomb damping force due to the sliding pin motion of the connector contacts is to determine an equivalent viscous damping force . In the case of viscous damping, the amplitude of resonance is proportional to the excitation force and inversely proportional to the damping coefficient and resonant frequency. For Coulomb damping, an equivalent damping is determined by equating the energy dissipated by viscous damping to that of the nonviscous damping force with an assumed harmonic motion. The energy lost per cycle is proportional to the work done by the system. The work done per cycle for a Coulombed damped system is $W_d=4F_dX$, where F_d is the frictional force and X is the amplitude at resonance. The equivalent viscous damping for a Coulomb damped system is then $\pi c_{eq}\omega X^2=4F_dX$.

For most structural metals, the energy dissipated per cycle is independent of frequency over a wide frequency range, and proportional to the square of the amplitude of vibration. Damping fitting this classification is structural damping. The energy dissipated per cycle is proportional to the square of the vibration amplitude and is equivalent to $W_d=\alpha X^2$, where α is a constant with units of force/displacement. The equivalent viscous damping is expressed by $\pi c_{eq}\omega X^2= \alpha X^2$ or $c_{eq} =\alpha/\pi\omega$. Using the concept of complex stiffness, the structural damping factor is expressed by $G=\alpha/\pi K$ which reduces to $G=4F_d/(\pi KX)$.

The sliding friction force of an individual contact was known from test data supplied by the connector manufacturer. In the FEM, five nodes span a single connector. One fifth of the connector's contacts were considered lumped at each node. The stiffness value, K, was determined by placing a unit load at the location where the stiffness was to be determined on the FEM. Examination of the frequency response function of the connector associated with a user specified input power spectral density function revealed the system behaved much like a single degree of freedom resonator, Figure 4. The response was characterized by a single spike near 90 Hertz. A ''nominal'' structural damping factor of 0.3 was determined for the upper bay and a factor of 0.5 for the lower bays. The damping at a particular location is inversely proportional to displacement. The values used for ''nominal'' structural damping correspond to locations of maximum deflection and therefore minimum damping. Analysis considered high and low values for structural damping to represent upper and lower bounds. The upper bound was associated with a critically damped backplane and the lower bound was associated with no structural damping in the backplanes.

Contact relative motion was defined as the relative motion between opposite nodes on the backplane and module in the direction normal to the backplane. Rotations of the backplane and connector were determined to be a second order effect and were not included in analysis. Connector relative motion was determined in the FEM using mpc relationships.

Boundary conditions of the FEM represent the avionics system to aircraft interface. Aircraft flexibility is incorporated into the FEM using 1D scalar spring elements at the rack/aircraft interface boundaries. Where structural damping was to be included, uniaxial rod elements replaced the spring elements.

ANALYSIS - INITIAL FEM

Of particular interest to the authors was the response of the connector due to a random excitation. Prior to predicting rack/module interface transmissibilities and connector displacements, the system natural modes of vibration were determined. Initial modal analysis was conducted using generalized dynamic reduction (GDR) and the modified Givens method for eigenvalue extraction. The frequency range of interest was 10 - 1500 Hertz.

Initial runs on an IBM 3090 mainframe fataled due to insufficient memory and spill problems. The Sturm sequence indicated 417 modes below 1500 Hertz, requiring over 600 generalized degrees of freedom. The frequency range was reduced to 10 - 1000 Hertz and spill problems were still encountered. Obviously, an unacceptable amount of cpu and I/O time was associated with the spill problems. Memory was expanded to the maximum allowable, but spill problems remained. The frequency range was further reduced to a maximum of 600 Hertz. This ran successfully, however, all modes were not recovered. Results indicated a small group of low frequency modes characterized by the rack moving as a rigid body with respect to the aircraft, at slightly higher frequencies a few rack flexible body modes, and the remaining modes were associated with packets of modules undergoing first bending, Figure 5. Missing modes were expected to be in the regions of high modal densities associated with the module resonances. This assumption was verified using the Lanczos eigenvalue solver which recovered all modes in the range of interest.

Approximately 45 minutes of cpu was required to obtain modes through 600 Hertz. This was not considered acceptable. A parametric study was to be conducted requiring many runs and results through 1000 Hertz. This study was to examine the sensitivity of aircraft/rack interface stiffness, backplane stiffness, and module growth or reduction (a partially empty or full rack). Since the basic configuration was a randomly populated rack, there was no symmetry to take advantage of to reduce model size. Several superelement approaches were examined in hopes of reducing cpu, expanding the frequency range, and improving numerical stability.

SUPERELEMENT APPROACH

The initial superelement approach included modeling a single module as a primary superelement and all others as image superelements. The remaining rack structure and aircraft interface constituted the residual structure. This superelement configuration failed due to an excessive bandwidth of the residual structure. The large bandwidth was contributed to no resequencing of the residual structure. Currently MSC/NASTRAN provides "full" or optimal resequencing for tip superelements. However, when a downstream superelement is partitioned, resequencing data associated with interior points that are exterior points of upstream superelements are not used. A multi-level tree superelement approach was attempted. This configuration consisted of primary and image superelements representing the modules, one down stream superelement representing the rack structure, and the residual structure representing the aircraft/rack interface structure. As expected, resequencing of the rack superelement was not sufficient and resulted in the same spill problems as previously encountered.

Following the failed attempts to minimize bandwidth, an entirely new approach was taken. Close examination of the natural modes of vibration indicated that the modules move in packets of at least three throughout the frequency range of interest. A new FEM was generated where each module represent three in the physical rack. Module physical parameters were adjusted so to maintain the first 3 fundamental modes and frequencies of vibration of the module. Obviously a single module in the new FEM had three times the mass of a single physical module. When comparing natural frequencies and modes of vibration with the original FEM, frequencies varied by less than 5% and the order of all natural modes were preserved, Figure 6.

ANALYSIS - REDUCED MODEL

The reduced finite element model of the avionics system now consisted of 1365 nodes and 1846 elements. Specifically, the FEM was composed of 1408 2D plate elements, 428 2D beam elements, 98 multipoint constraint relationships, 10 1D scalar springs and 19 rigid masses. This avionics system configuration possesses ''module growth capability'', i.e. vacant module slots for I/O growth were included in the FEM, Figure 7.

Using the reduced FEM, approximately 24 cpu minutes were required to solve for the natural modes of vibration through 750 Hertz. A value of 5% modal viscous damping (G) was assumed for the entire structure. This damping value was derived from the 2% damping value measured in test which considers an isolated module and damping due to other structure such as wiring etc. In addition to modal damping, the backplanes included structural damping due to the sliding frictional forces of the connector pins.

The natural modes and frequencies of vibration were used as inputs to a forced frequency response analysis, Figure 8. Forced response results were obtained using the ''large mass'' method. In the ''large mass'' method, a mass several orders of magnitude larger than the total mass and inertia of the structure is placed at the degree of freedom where motion is to be enforced. The magnitude of this mass is very important in obtaining accurate results. The mass must be large enough, typically five to six orders of magnitude larger than the total mass or inertia of the structure, to ensure adequate decoupling of the rigid-body mass motion from the flexible body motion of the structure.

Care must be taken to not make the mass too large when using the modal frequency response or the orthogonality of the natural modes may not be preserved. Section 3.5.4 of Reference 7 contains a detailed description for applying enforced motion. Since a combination of structural and modal damping was used in the FEM, the damping matrix was no longer proportional. A coupled algorithm was used to solve for forced responses and obtain the desired transfer functions. The forced responses are solved in the modal domain to minimize cpu. Connector responses were determined from 0-1000 Hertz at 1 Hertz intervals and included all resonance frequencies.

The forced frequency response analysis provided transfer functions due to unit accelerations applied at the aircraft/rack interface locations. The inputs excited motion normal to the backplane and included roll, yaw, and lateral (normal to the backplane) excitations. These results were used as input to a random analysis. Connector response associated with a user defined power spectral density input was determined. Connector responses were characterized as rms displacements and frequencies.

MODELING GUIDELINES & ANALYSIS OBSERVATIONS

The following guidelines are suggested for modeling structures similar to the avionics system in this study.

Mesh density of the modules should be sufficient to allow accurate representation of the module dynamic behavior in the frequency range of interest. Implementation will often result in a large number of degrees of freedom when including a FEM representation of each module. If the number of degrees of freedom are too large for the available resources, module ratioing is suggested. A 2 to 1 or 3 to 1 ratioing between physical and finite element modules can reduce an excessively large FEM to a manageable size while maintaining a high level of accuracy for determining connector motion.

For structures similar to the avionics rack in this study, exact correlation of the FEM module with modal test data is not essential. Typically, each module will be slightly different and lack of test data or time often does not allow a unique FEM representation of each module. If modules are similar, it is suggested each module be considered identical. Obviously if modules are substantially different, then a FEM representation of the associated modules will be required.

Boundary conditions of the backplanes at the rack interface should be considered carefully. A pinned or clamped boundary will substantially change backplane resonant frequencies.

The impact of modeling Coulomb damping as equivalent structural damping for predicting connector motion must be carefully assessed. This assumption is only true at a given frequency. The damping is nonlinear and inversely proportional to displacement. The problem should be bounded with high and low values of structural damping in the backplanes to assess connector motion sensitivity to damping.

The transfer function of the connector, as a function of frequency, should be examined to assess the validity of modeling coulomb damping as equivalent structural damping. This assumption is a reasonable approximation if the connector is primarily responding at a single frequency.

Rotations of the modules and backplane were second order effects in determining connector motion for this structure. Analysis should verify these motions are small before these terms are neglected.

When conducting a modal frequency response analysis using the large mass method, care must be taken when selecting the mass and/or inertia magnitudes. Too large a value results in a set or nonorthogonal modal vectors while too small a value results in modal coupling between flexible body motion of the structure and the rigid body motion of the large mass.

When using Version 66 or 67 of MSC/NASTRAN in conjunction with enforced motion of structures similar to the one in this study, the Lanczos eigenvalue solver is recommended. The Lanczos solver is recommended due to its robustness and ability to find all frequencies in areas of high modal densities. This is particularly true for structures where the first flexible body mode is above 30 Hertz and modal densities are high. In Versions 66 and higher, field 8 of the EIGRL card provides for an estimate of the first flexible body mode. If left blank, the program estimates a value for this field. The default, however, may not be sufficient for the algorithm to find the first flexible body root. Version 65 does not contain the option for estimating the first flexible body frequency. When using Version 65, GDR in conjunction with modified Givens or other appropriate solver is suggested when using the large mass method for enforced motion. The Lanczos solver most likely will not converge. Care should be taken to examine the Sturm sequence to determine if all modes were found and evaluate the significance of any modes that may be missing.

CONCLUSIONS

The finite element analysis procedures associated with predicting the relative motion of avionics connector contacts has been discussed. Several modeling reduction schemes and superelement approaches were presented where efforts were directed towards reducing model size and cpu, and improving numerical stability. Module ratioing was found to be the most effective means of reducing the number of degrees of freedom and improving numerical stability in the finite element model. Damping remains to be hard to define, however, an adequate representation of the Coulomb damping due to the sliding friction force of the connector contacts as equivalent structural damping was developed. Guidelines for conducting dynamic analysis are also presented. Implementation of these guidelines was demonstrated in a random analysis of a complex avionics system.

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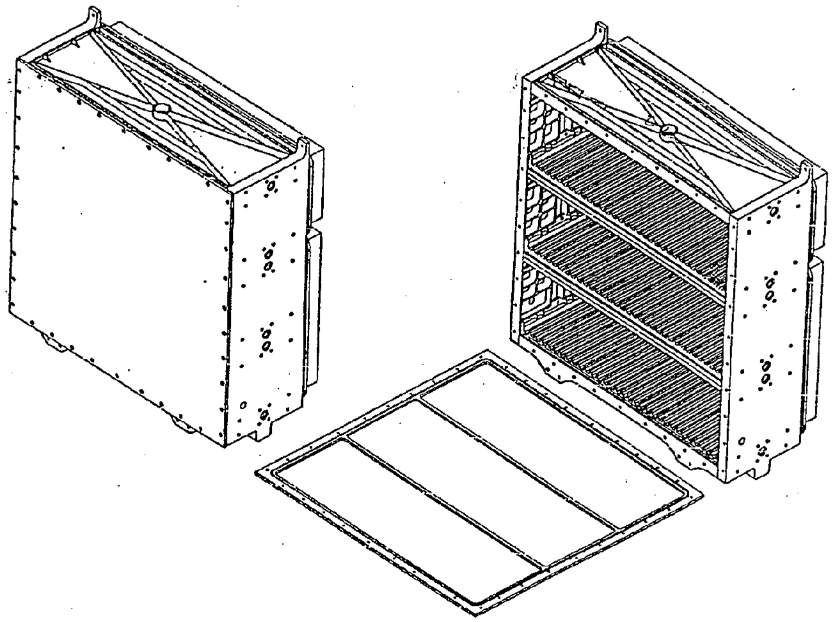


Figure 1 Three Bay Avionics System

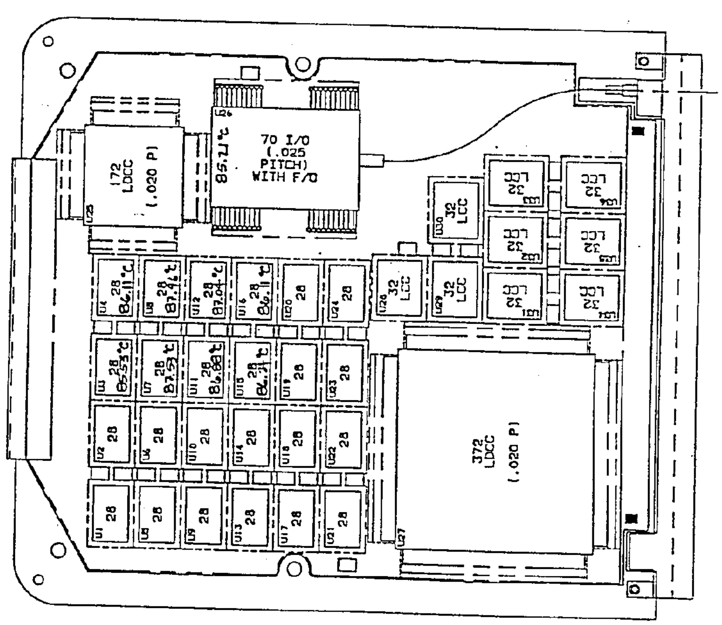


Figure 2 Line Replacable Module

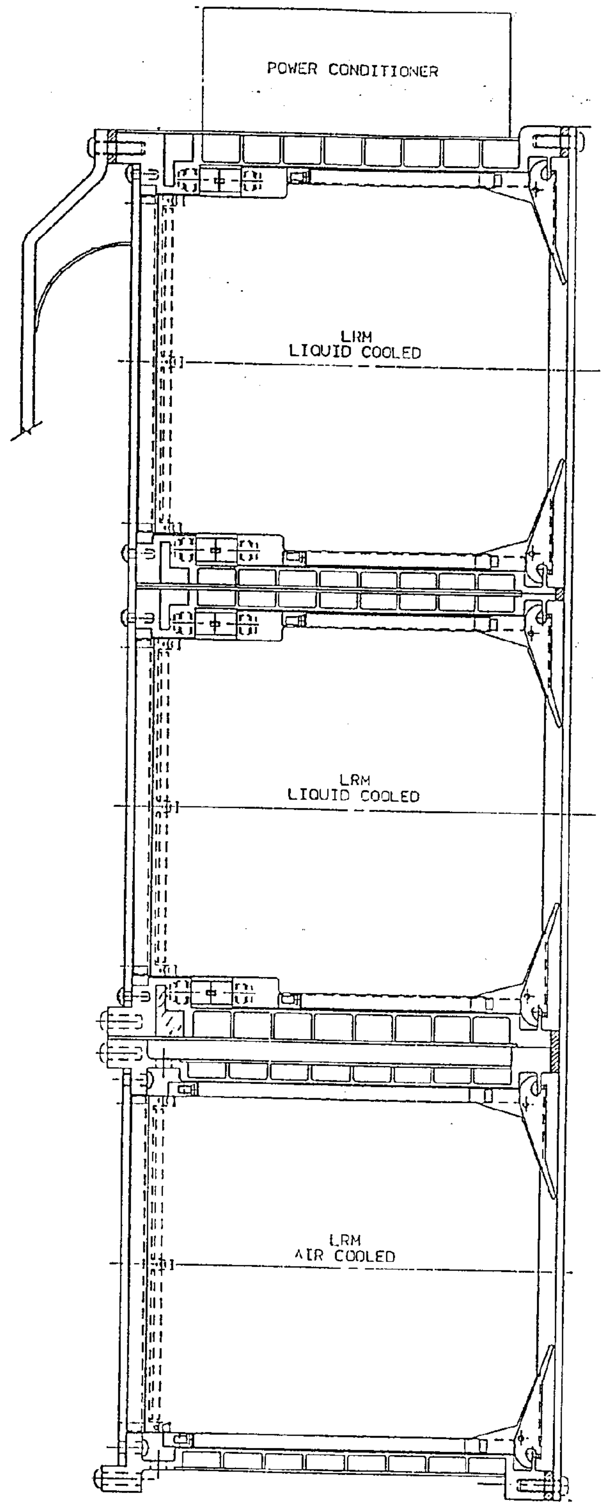


Figure 3 Avionics Rack Cross Section

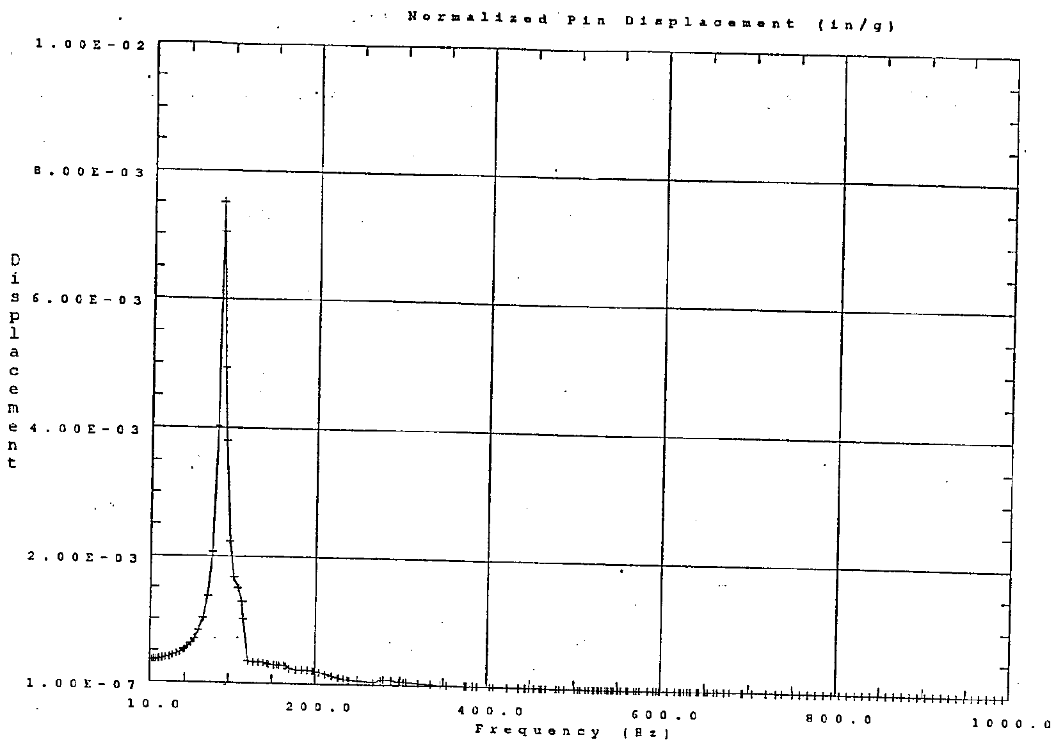
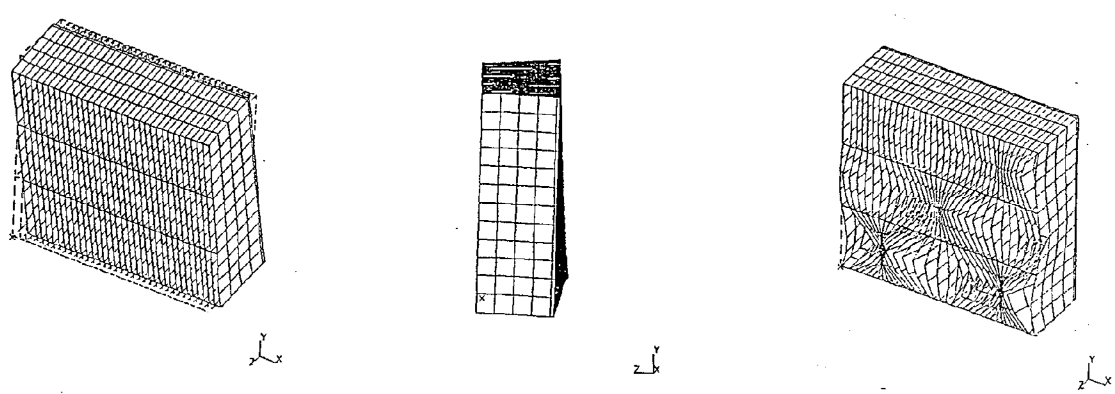


Figure 4 Connector Response

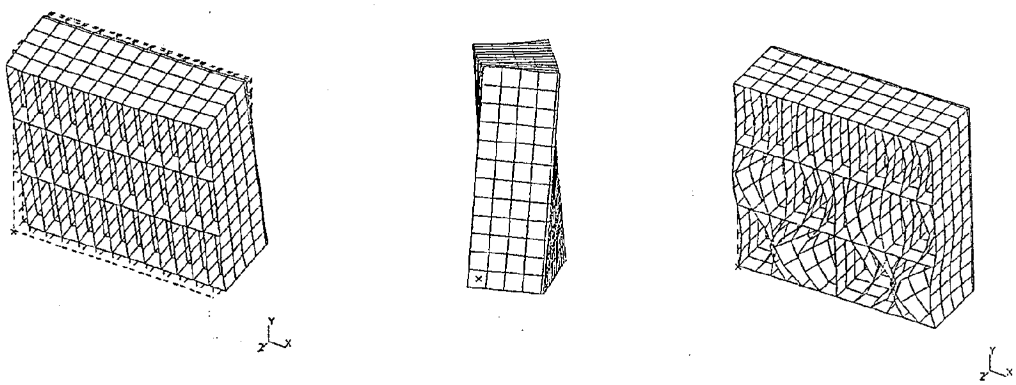


107 Hz
Rack Roll

122 Hz
Rack Torsion

497 Hz
Module Bending

Figure 5 Mode Shapes - Initial Finite Element Model



106 Hz
Rack Roll

115 Hz
Rack Torsion

477 Hz
Module Bending

Figure 6 Mode Shapes - Reduced Finite Element Model

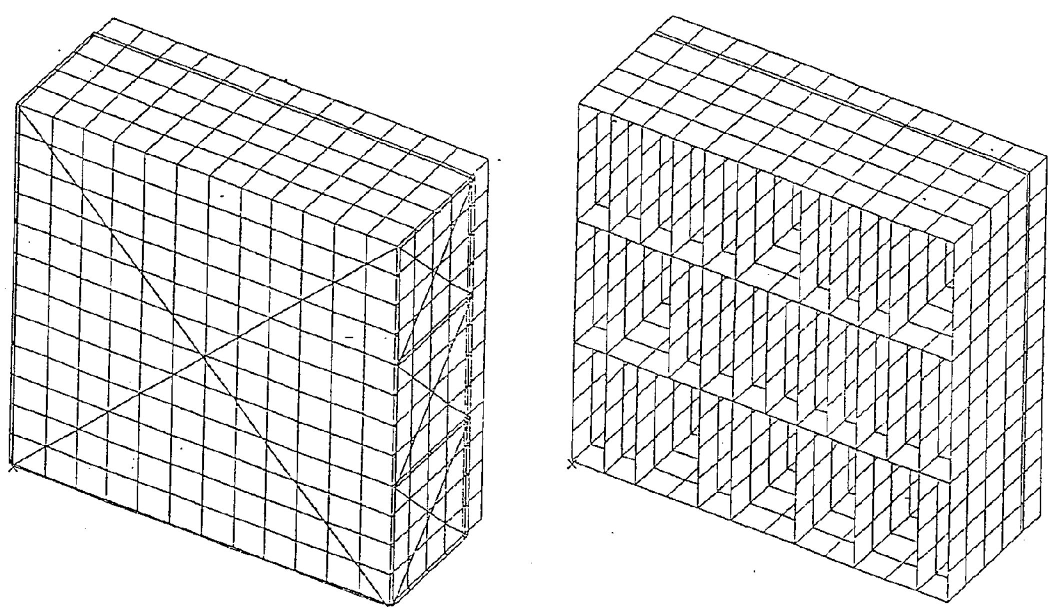


Figure 7 Reduced Finite Element Model with I/O Growth Capability

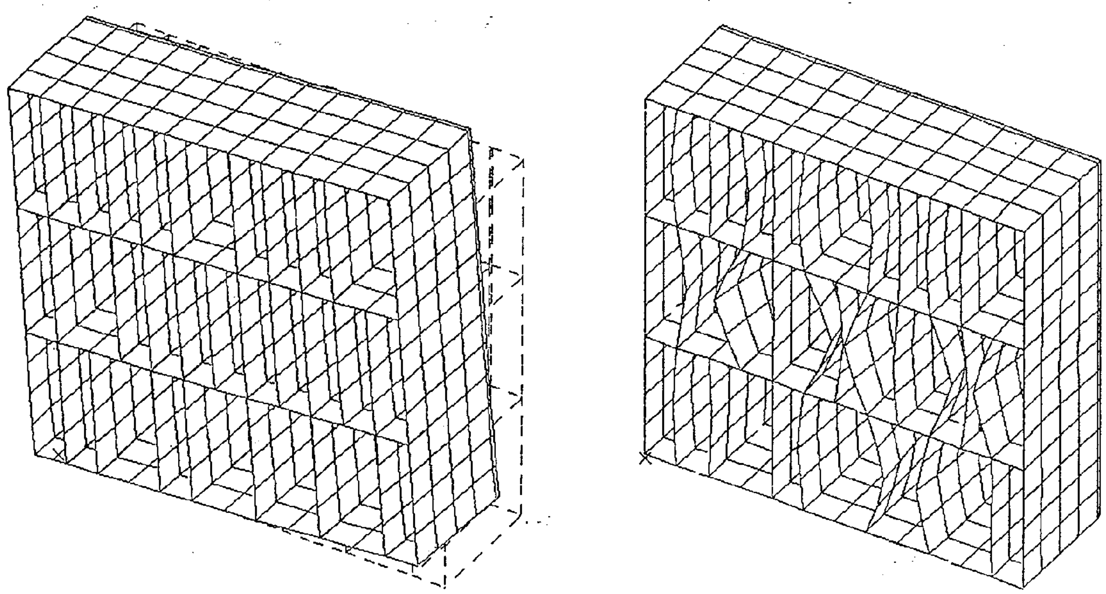


Figure 8 Mode Shapes - Reduced Finite Element Model
with I/O Growth Capability