Mid-Frequency Band Dynamics of Large Space Structures

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ABSTRACT

High and low intensity dynamic environments experienced by a spacecraft during launch and on-orbit operations, respectively, induce structural loads and motions, which are difficult to reliably predict. Structural dynamics in low- and mid-frequency bands are sensitive to component interface uncertainty and non-linearity as evidenced in laboratory testing and flight operations. Analytical tools for prediction of linear system response are not necessarily adequate for reliable prediction of mid-frequency band dynamics and analysis of measured laboratory and flight data. A new MATLAB toolbox, designed to address the key challenges of mid-frequency band dynamics, is introduced in this paper. Finite-element models of major subassemblies are defined following rational frequency-wavelength guidelines. For computational efficiency, these subassemblies are described as linear, component mode models. The complete structural system model is composed of component mode subassemblies and linear or non-linear joint descriptions. Computation and display of structural dynamic responses are accomplished employing well-established, stable numerical methods, modern signal processing procedures and descriptive graphical tools. Parametric sensitivity and Monte-Carlo based system identification tools are used to reconcile models with experimental data and investigate the effects of uncertainties. Models and dynamic responses are exported for employment in applications, such as detailed structural integrity and mechanical-optical-control performance analyses.

Keywords: Mid-frequency, dynamics, uncertainty, non-linearity, component mode subassemblies, reconcile, experimental data

1. INTRODUCTION

Analytical prediction of spacecraft primary structure dynamic response to launch and on-orbit environments has become a well-established engineering discipline. The (base) frequency band associated with primary structural dynamic response is generally below 100 Hz. The analytical tools used for dynamic response prediction include (1) the finite element method (FEM)[1], (2) component mode synthesis (CMS)[2], and (3) explicit and implicit numerical integration[3]. Analysis of measured data, test-analysis correlation and model reconciliation (for the purposes of dynamic model verification and validation) are based on (4) correlation and spectral analysis[4], (5) experimental modal analysis[5], and (6) systematic mode orthogonality and cross-orthogonality relationships[6]. All of the above noted tools and procedures rely heavily on assumed linear elastic structural mechanics and real, normal modes of vibration. A wealth of laboratory and flight experience points to the fact that linear structural dynamic tools do not reliably and accurately describe reality. Dynamic responses of complex spacecraft structures, which include a variety of joint connections, are both uncertain and non-linear as evidenced by differing apparent modal characteristics associated with (a) low versus high intensity dynamic environments and (b) initial conditions resulting from prior (non-damaging) loading events. Modal testing of the International Space Station P5 Truss[7] provides a clear illustration of uncertain and nonlinear behavior attributed to joint connections.

The challenges associated with spacecraft dynamic response prediction become more pronounced in the mid-frequency band (typically 100-500 Hz). Modal density in the mid-frequency band is typically below 10 modes per 1/3 octave band. However, for truss, thin plate, and shell assemblies, local modal densities may be significantly higher. As a result of close modal spacing in this frequency band, sensitivity of dynamic response to parametric uncertainty and non-linearity (especially at joints) is generally more pronounced than in the base frequency band. When modal density is above 10 modes per 1/3 octave band (the high-

frequency band), it is generally accepted that dynamic response characteristics may be appropriately described by statistical energy analysis (SEA) techniques[8]. While SEA takes advantage of asymptotic simplifications in the high-frequency band, it does not provide rigorous means for evaluation of non-linear effects.

In the low-frequency band, excitation and instrumentation arrays for measurement and detailed mapping of normal modes (for model verification and validation) may be rigorously defined within practical limits (i.e., location and number of sensors)[9]. As modal density increases in the mid- and high frequency bands, detailed mapping of normal modes becomes impractical due to excessive demands on instrumentation. Alternative measurement, correlation and model reconciliation strategies are therefore needed to address the task of model verification and validation. A reasonable alternative to detailed normal mode mapping is frequency response functions for linear structural dynamic subassemblies. In addition, measured load and displacement time histories may provide required information for verification and validation of local joint non-linear models.

Recent work, sponsored by the U.S. Army (TACOM)[10] and NASA's Jet Propulsion Laboratory (JPL) has led to development of a MATLAB based toolbox, designed to address requirements for mid-frequency range dynamic analysis. The toolbox, called ISYSD (Integrated System Dynamics), (a) imports Craig-Bampton components from NASTRAN or IMOS (a JPL finite element code), (b) assembles complete system dynamic models via component mode synthesis (CMS), (c) defines linear and non-linear joint models, (d) computes linear and non-linear structural dynamic response histories, (e) provides tools for parametric sensitivity studies and test-analysis reconciliation, and (f) exports models and results for other applications (e.g., fatigue and fracture analysis and optical-mechanical analysis). Moreover, ISYSD includes a series of graphical and tabular display features designed for practical review of analysis results. ISYSD addresses issues of importance to advance military ground vehicles, spacecraft systems, and automobiles. A preliminary concept of a Structurally Connected Interferometer (SCI) for JPL's Terrestrial Planet Finder (TPF) mission is used as an illustrative example in this paper.

2. SIGNIFICANCE OF JOINT UNCERTAINTY AND NONLINEARITY

Verification and validation of structural dynamic models are often misjudged on the basis of limited modal testing, especially when structural joints are pre-conditioned in the laboratory (e.g., shimmed) to eliminate play and non-linearity and when measured data is "linearized" to conform to linear modeling assumptions. During International Space Station P5 Truss modal testing at NASA/MSFC, estimates of normal modes changed after successive modal measurement sessions. Careful investigative testing revealed that ~3 mil gaps in several spherical bearings (and friction) caused sufficient local stiffness sensitivity to alter natural frequencies and mode shapes of even the lowest frequency modes. Moreover, substantial local non-linearity was noted in the behavior of the sensitive spherical bearings, as indicated in Figure 1.

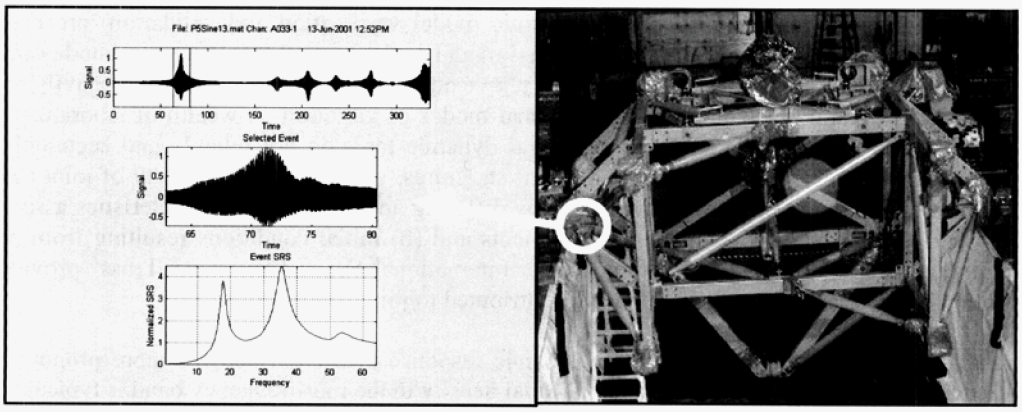

Figure 1. International Space Station P5 Truss Investigative Analysis of Spherical Bearing Behavior

The upper two data plots in the left-hand portion of Figure 1 detail the local acceleration time history during a swept sine test, the lower plot in the left-hand portion of Figure 1 represents a shock response spectrum indicating significant second harmonic distortion in the time history. Not shown are plots of acceleration data at locations away from non-linear joints, which indicate negligible or minor harmonic distortion (possibly indicating the validity of a "linearized" structural dynamic model).

Erroneous estimation of dynamic response due to "linearization" is illustrated in simulations of cargo response to ground vehicle random vibration (using the MIL-STD-810E common carrier transportation vertical environment). Response associated with a heavy cargo model, supported by wire-rope isolators (characterized by stick-slip friction) and having a lightweight, sprung mass (representative of an electronic component) is illustrated in Figure 2.

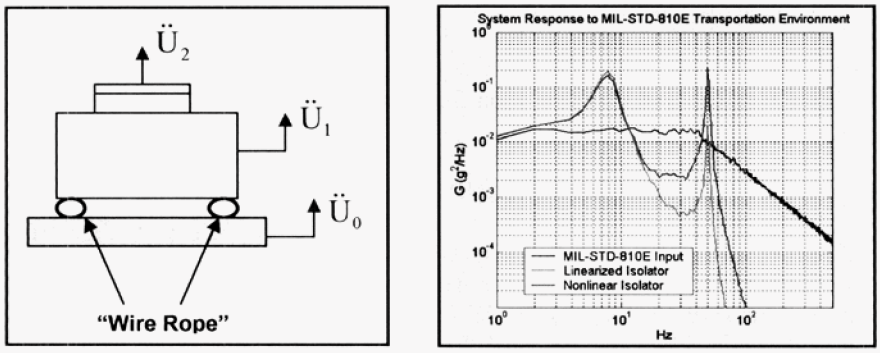

Figure 2. Wire-rope isolated heavy cargo simulated response to random vibration

Following wire-rope vendor recommendations, a linear spring and viscous damper approximation is used to describe isolation characteristics of the wire-rope hardware. The peak response power spectral density estimate for the lightweight subassembly (at its resonance), using the linear approximation, is about .02 g^2/Hz. In contrast, when the more appropriate non-linear model is used to represent the wire-rope isolator, the peak power spectral density estimate (at its resonance) is about one order of magnitude greater that the linearized system estimate. This result clearly illustrates potential inadequacy of linearized dynamic models and the need for practical non-linear structural dynamics tools.

3. OVERVIEW OF THE ISYSD TOOLBOX

The ISYSD toolbox was initially developed under U.S. Army (TACOM) sponsorship and is currently being refined and upgraded under NASA/JPL sponsorship in support of TPF research. Routines developed for a previous MATLAB toolbox, namely ITAP (Integrated Test-Analysis Processor)[11] have been modified and adapted for use in ISYSD. The ISYSD Process Controller GUI front panel, illustrated in Figure 3, summarizes general features of the toolbox.

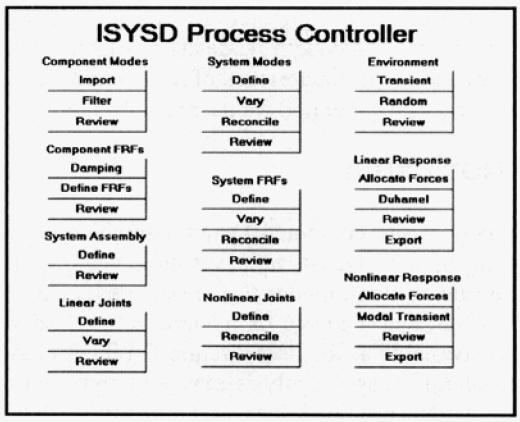

Figure 3. ISYSD Process Controller GUI Front Panel

3.1 COMPONENT MODES FUNCTION GROUP

A Craig-Bampton modal component, generated in NASTRAN or IMOS is imported and converted into a MATLAB (.mat) data file. The ISYSD Craig-Bampton data file includes component mass and stiffness matrices, physical geometry and coordinate system information. A modal filter may be used to eliminate insignificant modes from the component model. Insignificant modes are designated as those modes with modal effective mass below a user-specified threshold. A thorough graphical review of an ISYSD Craig-Bampton model may be conducted at the user's discretion.

3.2 COMPONENT FRFs FUNCTION GROUP

An ISYSD Craig-Bampton model is augmented with viscous damping parameters. Boundary motion excited frequency response of the component is computed over a specified frequency band. Graphical displays of boundary reaction loads due to unit boundary accelerations are displayed. In addition, a complete set of component physical acceleration frequency response functions is computed and stored for subsequent detailed graphical review of results.

3.3 SYSTEM ASSEMBLY FUNCTION GROUP

A collection of ISYSD Craig-Bampton components is assembled to form the basis of a structural system model. Coincident boundary grid points are automatically joined if they are co-located (within a specified position tolerance). The assembled system model consists of generalized component mode displacements and merged plus other non-coincident, physical degrees of freedom. The explicit system boundary points are used in subsequent ISYSD operations as (a) joint connection points and (b) external loads application points. Physical and coordinate system information is included in the ISYSD system model data file. Graphical displays illustrating generalized mass and stiffness topologies are generated along with a bar graph summary of component mode frequencies.

3.4 LINEAR JOINTS FUNCTION GROUP

In order to complete definition of a linear structural dynamic system, the non-coincident component boundary points must be connected by "joint" elements as required. A joint element assembly for a linear model consists of beams connecting non-concurrent boundary node pairs. In addition to a joint element data file, additional matrix and tabular data, required for modal sensitivity and FRF (test-analysis) reconciliation analyses are generated. Included in the generated data are residual modes, appended to the system model file, which are required for effective modal sensitivity analysis. A graphical and tabular review of a linear joint file may be conducted at the user's discretion.

3.5 SYSTEM MODES FUNCTION GROUP

Linear system modes and frequencies are computed for a Craig-Bampton component and linear joint assembly. A thorough graphical review of system modes is available to the user. Displays are designed to enable the user to clearly understand the characteristics of modes associated with large, complicated system models. Modal sensitivity maps may be generated for user specified ranges of joint stiffness parameters.

3.6 SYSTEM FRFS FUNCTION GROUP

Linear frequency response functions are computed for an assembled structural dynamic system described by a truncated set of system modes. Quasi-static residual vectors, associated with applied loading distributions, augment the truncated system modes to minimize the effects of modal truncation. Frequency responses are computed for all physical degrees of freedom associated with the applied forces. Detailed graphical reviews of FRFs are available at the user's request. When measured test FRFS are available for an assembly, a Monte-Carlo random search analysis may be conducted to optimally reconcile measured data and the assembled system model (provided linear joint sensitivity data has been previously defined).

3.7 NONLINEAR JOINTS FUNCTION GROUP

A linear joint data file is augmented with nonlinear joint parameters for one of four model types, namely (1) cubic stiffness, (2) gap (or dead-band), (3) stick-slip friction, (4) graduated stick-slip. In the near future, "hooks" to include user-defined nonlinear joint models will be introduced. If a nonlinear joint test data file is available, the nonlinear joints may be reconciled with measurements using a time domain Monte-Carlo search strategy.

3.8 ENVIRONMENT FUNCTION GROUP

In preparation for specification of applied external force time histories, physical distributions of applied loads are specified. As part of this process, residual modes, which account for static response of modes not included in the truncated mode set, are computed and appended to the system mode set. Time history files associated with user-defined applied force tables are generated. If the applied forces are specified as time histories, a transient applied force data file is defined by linear or spline interpolation for a uniform sampling rate consistent with the system model frequency band. Alternatively, if the applied forces are specified as uncorrelated frequency-force power spectral density pairs, applied force time histories are generated using a specialized digital filtering process. A thorough graphical review of generated force history data, employing well-known data analysis tools, may be invoked at the user's discretion.

3.9 LINEAR RESPONSE FUNCTION GROUP

Modal transient responses associated with applied transient excitations are efficiently calculated using a Duhamel integral computation algorithm. Modal acceleration, velocity and displacement histories are stored in the system dynamic model data file for further review and analysis. Physical responses are computed and made available for detailed graphical reviews using well-known data analysis tools. Dynamic model matrices and response time histories may be optionally exported for other applications.

3.10 NONLINEAR RESPONSE FUNCTION GROUP

Modal transient responses associated with applied transient excitations are efficiently calculated using an Adams-Bashforth computation algorithm. Modal acceleration, velocity and displacement histories are stored in the system dynamic model data file for further review and analysis. Physical responses are computed and made available for detailed graphical reviews using well-known data analysis tools. Dynamic model matrices and response time histories may be optionally exported for other applications.

3.11 SUMMARY OF THE ISYSD RESPONSE SIMULATION PROCESS

A general overview of the ISYSD response simulation process is summarized in Figure 4. Note that the only step in the nonlinear response process, departing from standard linear response computation, is the "nonlinear joint equations" operation.

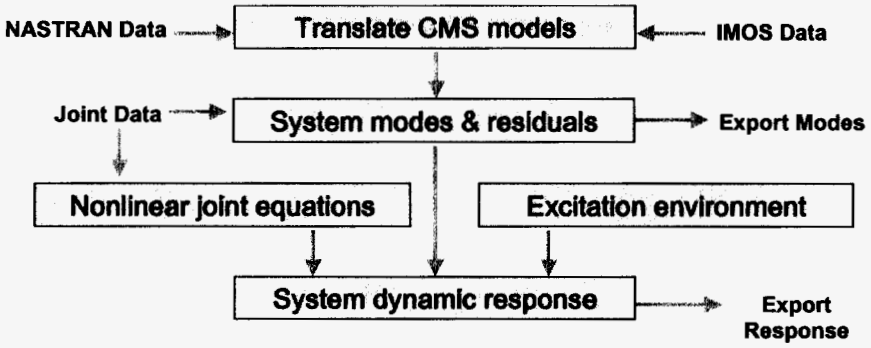

Figure 4. ISYSD Response Simulation Process

4. DEFINITION OF RELEVANT MID-FREQUENCY DYNAMIC MODELS

Modern preprocessors permit automated definition of structural finite element models directly from engineering drawings. As a result, models are often characterized by many more degrees of freedom than necessary for components defined from detailed engineering drawings. In contrast, components not based on detailed drawings (i.e., in a preliminary stage of design) may not be described with enough degrees of freedom or, in some cases, be defined as rigid bodies. This type of inconsistency leads to erroneous system models, especially in the mid-frequency band.

4.1 DYNAMIC FREQUENCY BAND BASED ON EXCITATION ENVIRONMENTS

Dynamic environments are generally (a) harmonic, (b) transient, (c) impulsive or (d) random. For all categories, the dynamic bandwidth of the environment or cut-off frequency (f^*) is reliably determined by shock response spectrum analysis[12] as illustrated in Figure 5.

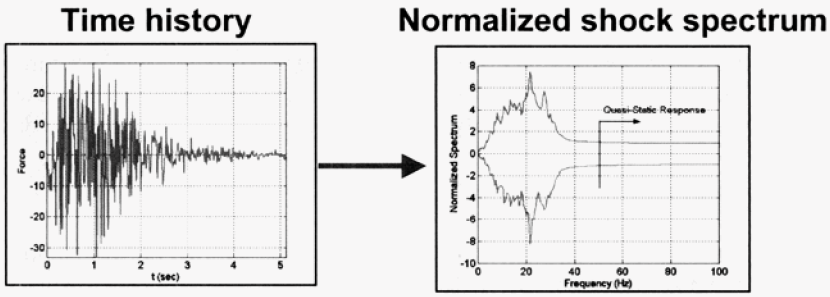

Figure 5. Example Excitation Time History and Normalized Shock Spectrum

A characteristic of the normalized shock response spectrum for any time history is that its value will settle to unity above the cut-off frequency (f*) indicating that the response of a single degree-of-freedom oscillator having its natural frequency above f* responds quasi-statically to the subject environment. For linear systems f* clearly defines the frequency requirement for relevant model fidelity.

4.2 FINITE ELEMENT MODEL GRID SPACING REQUIREMENT

With the cut-off frequency (f^*) established, the shortest relevant wavelength of a forced vibration for components in a structural assembly may be calculated. For finite element modeling, the quarter wavelength (L/4) is of particular interest, since it defines the grid spacing requirement needed to accurately model the dynamics. The guidelines for typical structural components are summarized in Table 1. In addition to the above grid spacing guidelines, the engineer must also consider the limitations associated with beam and plate theories. In particular, if the wavelength to thickness ratio (L/h) is less than about 10, a higher order theory (pure flexure) or 3-D elasticity modeling should be considered. Moreover, modeling requirements for the capture of stress concentration details may call for a finer grid meshing than suggested by the cut-off frequency. Finally, if the dynamic environment is sufficiently high in amplitude, nonlinear modeling may be required, e.g., if plate deflections are greater than the thickness, h.

Table 1: Guidelines for Dynamic Finite Element Model Meshing

Component	Mode Type	L/4	Additional Data
String	Lateral	$(\sqrt{T/\rho A})/(4f^*)$	T=tension, A=area ρ = mass density
Rod	Axial	$(\sqrt{E/\rho})/(4f^*)$	E = elastic modulus
Rod	Torsion	$(\sqrt{G/\rho})/(4f^*)$	G = shear modulus
Beam	Bending	$(\pi/2)(EI/\rho A)^{1/4}/\sqrt{2\pi f^*}$	EI = flexural stiffness
Membrane	Lateral	$(\sqrt{N/\rho h})/(4f^*)$	N = stress resultant
Plate	Bending	$(\pi/2)(D/\rho h)^{1/4}/\sqrt{2\pi f^*}$	D=plate flexural stiffness h=plate thickness
3-D Elastic	Dilatational	$(\sqrt{E/\rho})/(4f^*)$	
3-D Elastic	Shear	$(\sqrt{G/\rho})/(4f^*)$	
Acoustic	Dilatational	$(\sqrt{B/\rho})/(4f^*)$	B=Bulk Modulus

4.3 STRUCTURALLY CONNECTED INTERFEROMETER (SCI) DYNAMIC MODEL

The structurally connected interferometer concept for the Terrestrial Planet Finder (TPF) program, shown in Figure 6, serves as an illustrative example dynamic system with a 300 Hz bandwidth requirement.

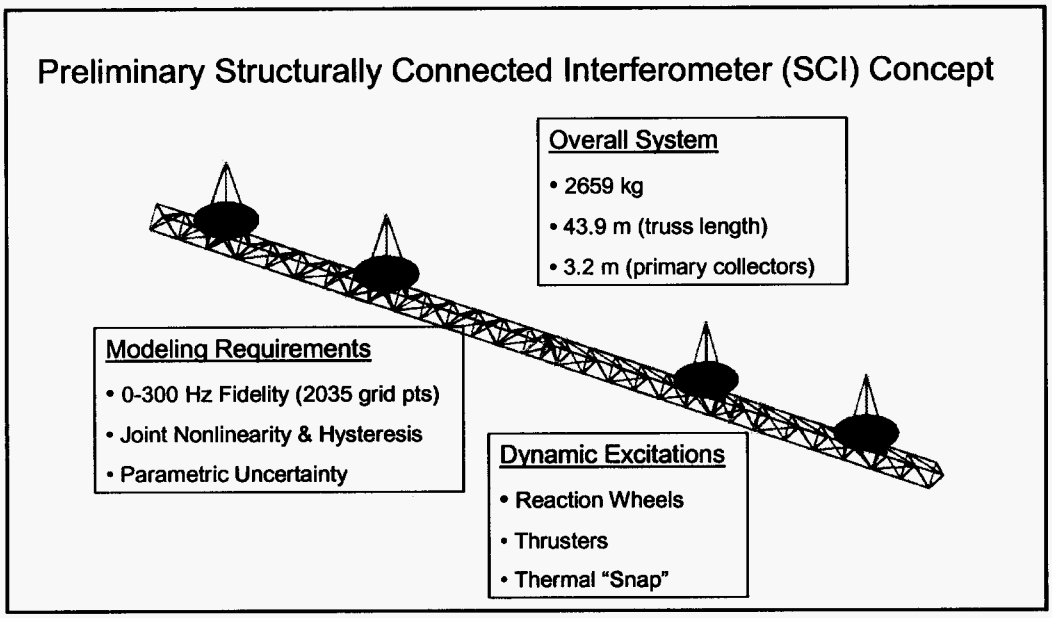

Figure 6. Preliminary Structurally Connected Interferometer (SCI) Concept

The concept system is composed of tubular composite beam members for the truss and four (3.2 meter diameter) primary collectors that are defined as shell structures. In addition, there are six hinge locations in the truss with a total of 18 mechanical joints that permit the truss to be folded into its launch configuration. Following the modeling guidelines outlined in Table 1, the 300 Hz mid-frequency finite element model was defined as illustrated in Figure 7. Note that this structure has been segmented into seven subassemblies (subassembly 1 is highlighted in the lower right corner caption in the figure) with the separations defined by the six hinge locations. The grid layout for substructure 1 is detailed in the figure.

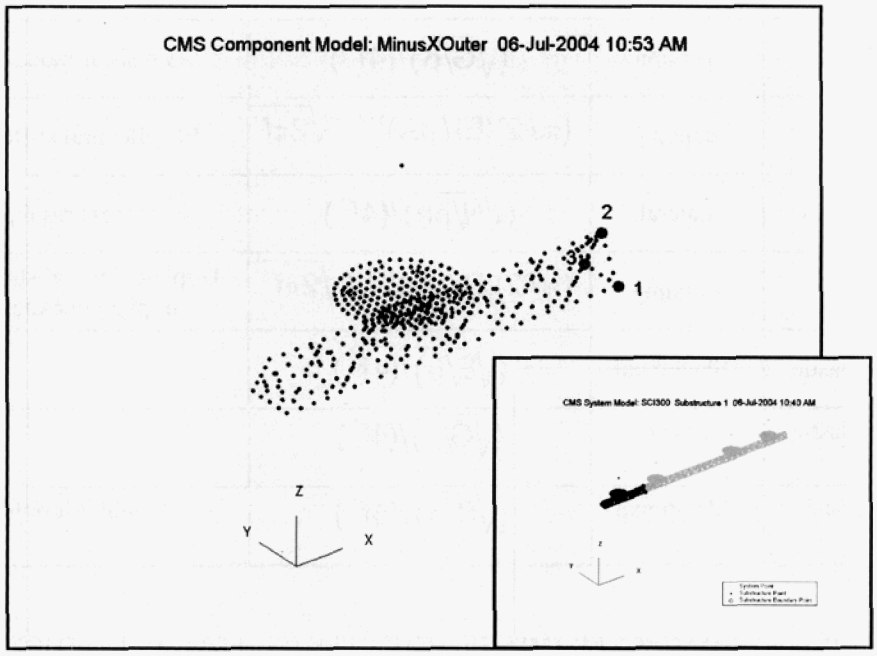

Figure 7. Mid-Frequency Band SCI Model Grid Layout with Details of one Particular Subassembly

5. MID-FREQUENCY BAND VIBRATION MODES OF THE SCI MODEL

A linear dynamic model of the SCI configuration was assembled from the seven individual subassemblies. The eighteen joints, which connected the seven subassemblies, were modeled as short, stiff beams. A summary of calculated system mode frequencies (320 modes up to 200 Hz) is provided in Figure 8.

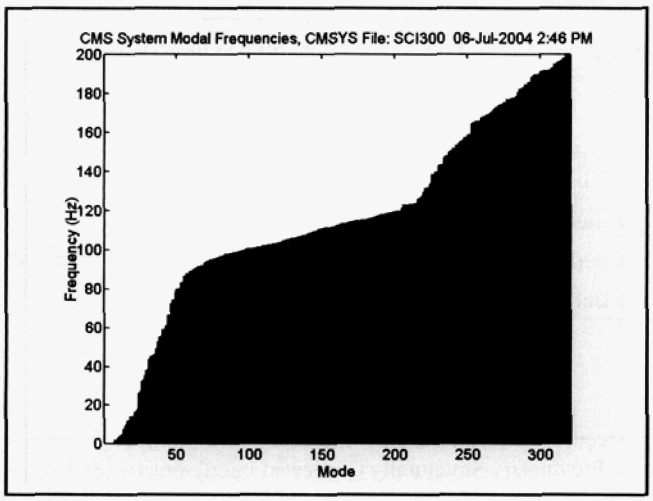

Figure 8. SCI System Mode Frequencies

The lowest frequency flexible body modes of the SCI are characterized by overall body motion. The first flexible body mode, lateral "Y" bending at 1.76 Hz, is illustrated in Figure 9.

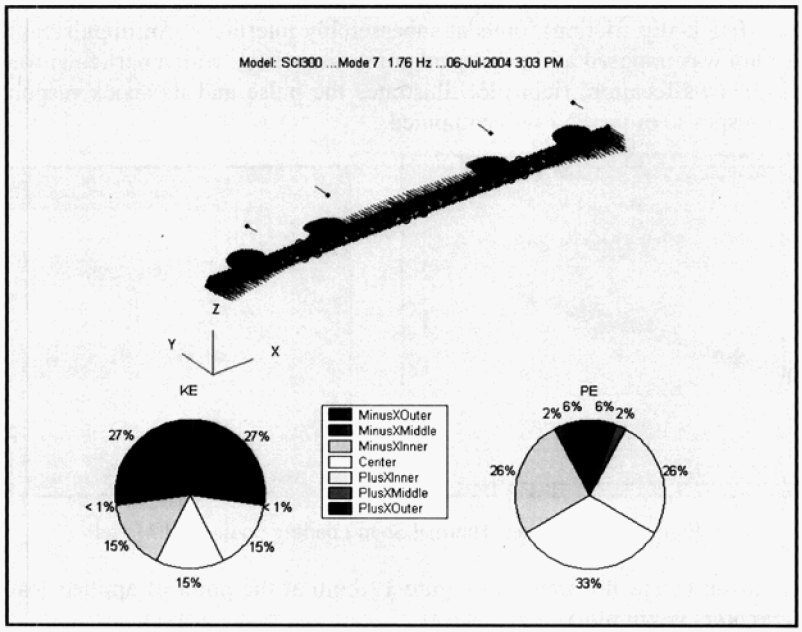

Figure 9. Lowest Frequency Flexible Body Mode at 1.76 Hz

Note that the character of the mode is indicated by distributions in kinetic energy and potential (or strain) energy among the subassemblies, as well as by geometric deformation. This type of display is extremely useful for describing the character of higher frequency, local modes.

The distribution of modal density for this system, summarized in Figure 10, indicates that modal density above 80 Hz is on the order of 20+ modes per 1/3 octave band; this level of modal density, strictly speaking, is representative of the high-frequency band normally treated by the SEA method. The high modal density is due to the fact that local truss member bending modes occur above 80 Hz.

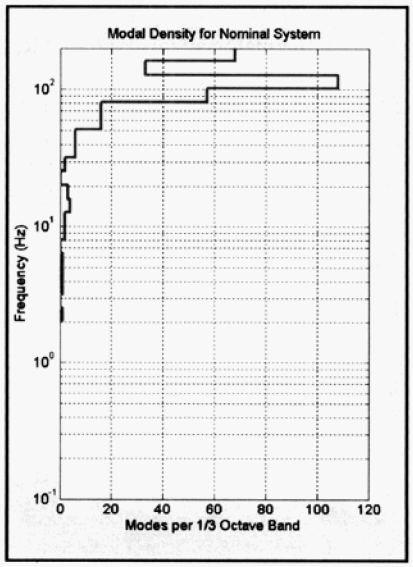

Figure 10. Distribution of SCI Modal Density

6. RESPONSE OF THE SCI TO SIMULATED THERMAL SNAP LOADING

The linear SCI system model, with modal damping specified at 0.1% of critical damping, was modified by including non-linear (stick-slip friction) joints at subassembly interfaces. An impulsive (half-sine) thermal snap loading condition was imposed at the two grid points associated with a particular joint as illustrated in Figure 11 (left plot shows location, right plot illustrates the pulse and its shock response spectrum) and non-linear transient response of the SCI was computed.

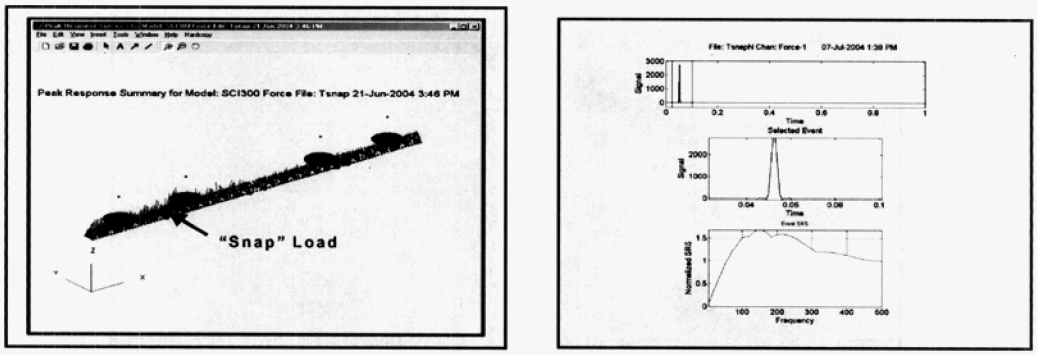

Figure 11. Simulated Thermal Snap Loading on the SCI Model

Transient response histories are illustrated in Figure 12 both at the point of applied load application (left plot) and at a remote point (right plot).

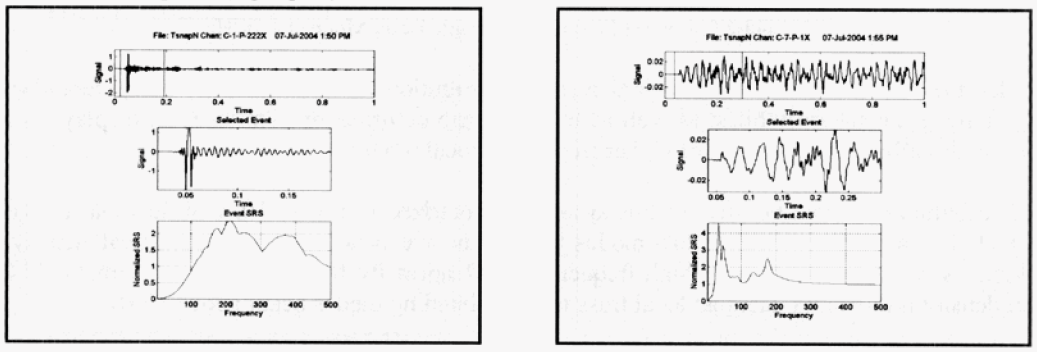

Figure 12. Typical SCI Thermal Snap Load Induced Response Histories

The physical distribution of peak acceleration response, shown in Figure 13, indicates that peak response amplitude generally decreases exponentially with distance from the force application point.

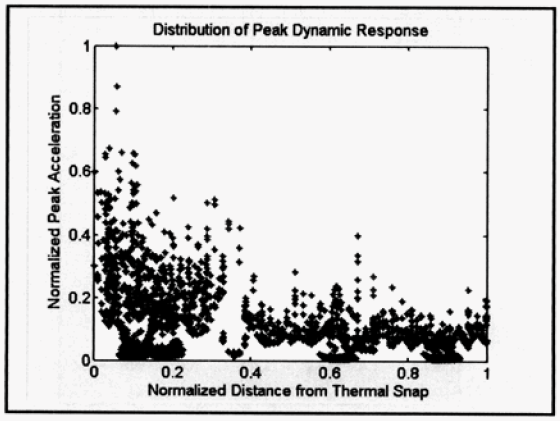

Figure 13. Physical Distribution of Peak Response

7. RESPONSE OF THE SCI TO APPLIED SINUSOIDAL LOADING

The SCI model with non-linear joints, described in the previous section is subjected to a centrally located sinusoidal loading as illustrated below in Figure 14 (left plot shows location, right plot illustrates the force history and its shock response spectrum) and non-linear transient response of the SCI was computed.

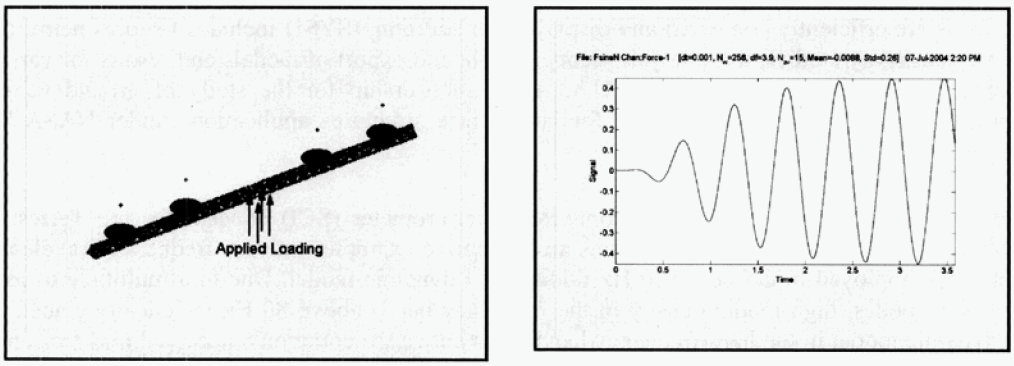

Figure 14. Applied Lateral Sinusoidal Loading

The overall system transient response is sinusoidal, however, the local responses at the nonlinear joints indicate clear hysteresis as illustrated in Figure 15 (which compares joint axial strain and stick-slip force).

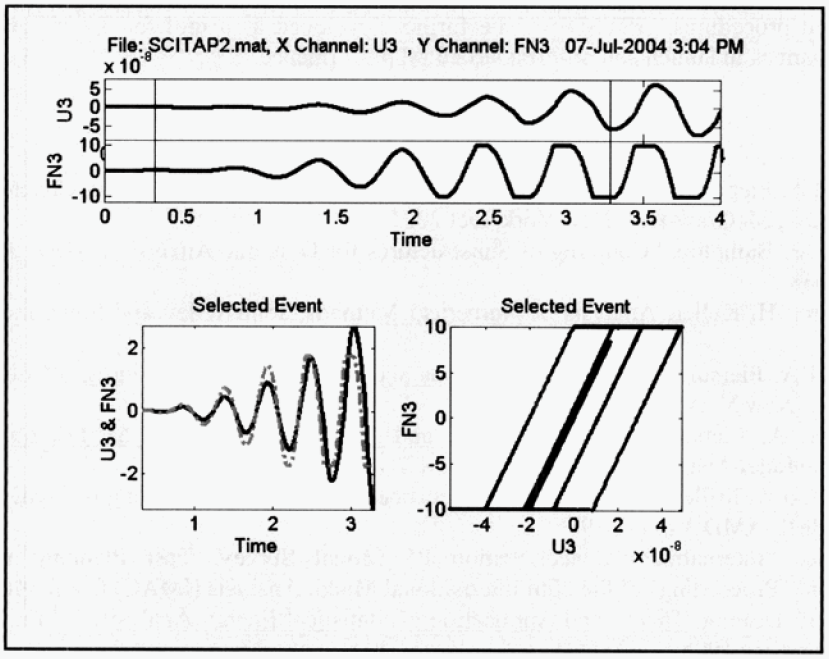

Figure 15. Joint Axial Strain and Stick-Slip Force Histories

8. CONCLUDING REMARKS

ISYSD has been developed as a new MATLAB toolbox addressing mid-frequency band locally non-linear dynamics of structural systems. The toolbox assembles structural subassemblies, described as Craig-Bampton modal components, computes, and displays assembled system modes. Component interface joints may be modeled as having linear or nonlinear behavior and transient responses to applied loads are efficiently computed and displayed. In addition, ISYSD includes features permitting parametric sensitivity studies, test-analysis reconciliation, and export of models and results for various applications. Originally developed under U.S. Army sponsorship for the study of ground vehicle dynamics, this toolbox is being enhanced for large space structures applications under NASA/JPL sponsorship.

Modeling and analysis of a structurally connected interferometer (SCI) concept for the Terrestrial Planet Finder (TPF) mission was presented as an illustrative example study. A frequency-wavelength guideline was employed to define a 300 Hz fidelity SCI dynamic model. Due to a multitude of local truss member modes, high modal density in the frequency bands above 80 Hz technically placed the system dynamic model in the high-frequency band (well above the commonly accepted mid-frequency band). Response to a simulated thermal snap pulse was investigated indicating decreasing response transmission with increasing distance from the point of excitation. Response due to a low frequency sinusoidal loading was calculated to demonstrate hysteretic behavior of stick-slip friction joints.

Studies are ongoing with laboratory test activities expected to begin in the near future. The primary goal of this continuing work is to develop realistic mid-frequency band modeling and analysis guidelines and procedures. ISYSD will be further developed as a tool for determining the role of nonlinear dynamics in launch and on-orbit structural performance.

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