MICRODYNAMIC ANALYSIS FOR ESTABLISHING NANOMETRIC STABILITY REQUIREMENTS OF JOINTED PRECISION SPACE STRUCTURES

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ABSTRACT

This paper presents a framework for analyzing a structure for nonlinear dynamic behavior in the nanometric regime, or microdynamics, and illustrates how these analyses might be used in structural design and system verification. A categorization of the different modes of microdynamic behavior as a function of frequency response and complexity of structural analysis is established. Analysis methods appropriate for some of these modes are outlined.

1. INTRODUCTION

Current concepts for deployable space optical systems require nanometric position stability of optical components. One area of great concern is the mechanical stability of the structural interfaces (e.g., joints, hinges, and latches) during flight mission operations under thermal loads, spacecraft slews and other on-board jitter sources. At such low levels of response, a potentially strong source of nonlinearity exists due to frictional microslip at the interface contact area. Assessing the structural behavior of interface components, particularly when they exhibit strong nonlinearities, becomes an important factor for accurate optical performance prediction of the instrument.

Nonlinear behavior relevant to microslip and contact mechanic behavior in the small has been generally referred to Microdynamics. Tribological models of contact mechanics and microslip using Hertzian and Mindlin theories have been developed and proposed in the past for component level analyses and test verification [1, 2, 3]. These analytical methods are derived for specific types of interfaces (e.g., ball bearing or bolted) and are based on detailed physical parameters such as contact stiffness, asperity distribution, and Coulombic yield levels. Unfortunately, such detailed information is not readily available in the early phases of the design development.

The current intent is to break down microdynamic nonlinear behaviors into elements that are meaningful when flowing down system level error budget allocations using only preliminary design information. Early in the mission project cycle, microdynamic analyses will be used to flowdown optical performance objectives to mechanical stability requirements of components. In turn the bounding stability requirements of the components will be used to guide the component design and verification tests. As such, the proposed system analysis approach should provide bounding solutions for the detailed component analysis that will be performed once the designs, the overall structural configurations and load paths are known.

This approach to microdynamic modeling is currently being implemented to define the error budget allocation for two NASA ORIGINS space missions: SIM (Space Interferometry Mission) and NGST (Next Generation Space Telescope). Several example microdynamic analyses are presented in the paper. One mode of microdynamics is the impulsive snap, a sudden energy release at an interface. Snap models have been verified using IPEX flight data, and are implemented on SIM for dynamic disturbance assessment. Another mode of microdynamics is the nonlinear response to steady-state excitations due to change of interface stiffness during microslip. Nonlinear stiffness models have been validated using the CASSINI interplanetary spacecraft modal test data, and are implemented on SIM to estimate the extent of the response distortion induced by the nonlinearity.

2. HYSTERESIS

Work performed in the area of nanometric stability over the last five years has established that microdynamics are a manifestation of nonlinear contact dynamics in the small [1, 2]. Deployment and latching mechanisms are a major contributor to microdynamic instability because they typically couple massive components and usually store the largest strain energies in the system. However, all mechanical interfaces, such as optics mounts, cables, material matrix and fiber, are capable of microdynamic nonlinearities.

Nonlinear microdynamic behavior is linked to mechanical hysteresis in the interface. As such, design mitigation strategies are aimed at reducing risk by minimizing hysteresis in the system without sacrificing stiffness and damping. These recommendations have been published in the recently released microdynamics design guidelines [1, 2, 3].

All structures are expected to exhibit microdynamic behavior under small loads, but the behavior can be made stable and bounded to within nanometers. Nonetheless microdynamic instabilities are real, as has been demonstrated in the IPEX flight experiment [4]. But what is not known yet are all the causes of these microdynamic instabilities and all the ways local microdynamics affect system response through propagation. It is also unclear how the behavior is modified in a 0-g field and what are the influences of gravity. Presumably the preload applied by the gravitational field alters the location of the stress-strain response along the hysteresis curve, but no experiment has yet validated this theory. And finally, if the bound on the microdynamic response falls beyond the requirement tolerances, then new ways will have to be devised to control microdynamics.

As stated above, hysteresis incorporates the many features that form the basis for the proposed modes of microdynamic behavior. Hysteresis describes the response of nonlinear static components. It depends on the loads applied at the interfaces. Geometrically, the width of the hysteresis curve describes position uncertainty and creep. The area inside the curve represents the amount of energy stored and possibly dissipated. The backbone of the hysteresis curve represents stiffness variations as a function of loading amplitude. Hence hysteresis is also recommended as a measure of dynamic nonlinearity and stability to be used in system models in lieu of detailed physical models of joints and other mechanical interfaces. Of course this implies that the interface geometry and the load transfer across the interface become critical parameters for these models. Furthermore, models of static hysteresis will have to be enhanced with parameters such as loading velocity and energy release velocity to incorporate dynamic effects into the model.

3. MODES OF MICRODYNAMIC BEHAVIOR

Six modes of microdynamic behavior are proposed. They are ranked in terms of response frequency and analysis complexity and cost. These modes of microdynamics also reflect parameters that then become requirement quantities within error budgets, and can then be used as drivers for component designs.

The first mode of microdynamic stability is the D.C. offset generated by a slip at an interface. The analysis of the static slip offset entails applying a geometric offset in the finite element model (FEM) and assessing the effect on the alignment of the rest of the precision structure. This type of analysis is particularly meaningful to missions such as Next Generation Space Telescope (NGST) [6] where the 8-meter wide segmented telescope will have to maintain nanometer stability of the primary mirrors for hours without the use of control mechanisms.

The second mode of microdynamics is the quasi-static effects of creep and time-dependent strain relaxation. Creep can be modeled as piece-wise static offset error, using the method described above.

The third mode of microdynamic behavior is the dynamic instability or "snap crackle and pop" induced by a sudden slip across joints. These are typically exhibited as impulsive disturbances in the response measurements. Approaches for modeling these snaps will be discussed below.

The fourth mode of microdynamic behavior is nonlinear stiffness. The inherent stiffness of the joint will change as the load cycles back and forth, and the response follows the behavior dictated by the hysteresis curve. The system is said to be softening if the stiffness decreases with amplitude, or the system is hardening if the stiffness increases with amplitude. The nonlinear stiffness is depicted in the hysteresis curve as the backbone of the curve. In the frequency domain, a nonlinear stiffness is mapped into a distorted transfer function, as shown in figure 1 for a softening system. The immediate implication is that there is a zone of instability within which the response at a given frequency can have more than one amplitude, as illustrated in Figure 1. Another implication of nonlinear stiffness is harmonic distortion of the response, in which for a single frequency input load, the response has multiple frequency components other than the frequency of the input (a classic definition of nonlinearity). Harmonic distortion has been observed in the IPEX flight hardware both on the ground and on -orbit (Figure 2). Hence, nonlinear stiffness degradation effects would be

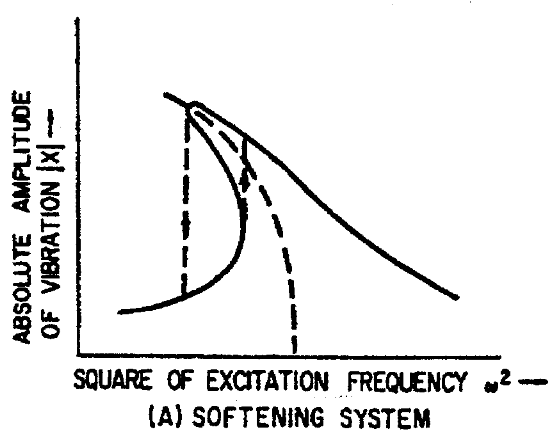

Figure 1. Transfer function of a softening system.

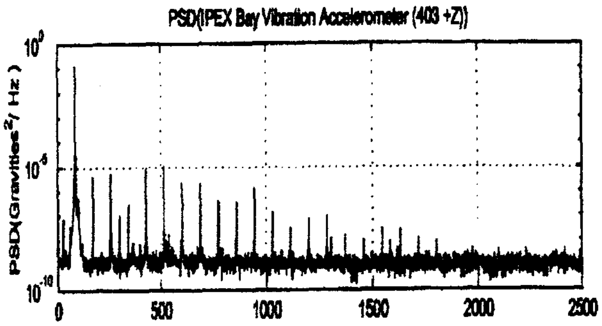

Figure 2. Harmonic distortions measured in the IPEX bay during sine dwell tests [7].

most significant when assessing the system response to steady-state disturbances, such as reaction wheels.

Means of modeling nonlinear stiffness degradations have been implemented on the data obtained during the CASSINI spacecraft modal tests, and will be discussed in a later section.

The fifth mode of microdynamic behavior is nonlinear damping or velocity dependent effects. This issue refers to approximations of frictional damping as modal damping in standard structural models, and how amplitude and frequency of vibrations affect these approximations. Damping can also be viewed as the energy dissipated per cycle of the hysteresis loop. Damping is a function of frequency in that it relates to the velocity at which the friction is exerted at the interfaces, and hence it implies a relationship between static hysteresis and dynamic hysteresis.

It is not clear at this point how to incorporate analysis tools that would provide damping bounds for the error budget allocation process. The typical way of dealing with damping uncertainties in models is to assign modal damping values that correspond to the lowest value that has been historically measured on comparable structures. This same modal value is applied equally across the whole frequency range of interest. This approach is certainly sufficient as long as the selected damping value has a very low probability of ever being exceeded during spacecraft operations. However, there is little flight data available to base current damping values for precision space instruments such as SIM and NGST. Both instruments will operate at very low background jitter levels. Hence it is suspected that frictional damping will be small, and that material damping will predominate. This becomes even more of an issue for NGST, since at cryogenic temperatures of 50°K material damping is extremely low. Parasitic damping sources such as friction induced from cable harnesses is extremely difficult to predict and these effects are currently ignored in model predictions. There is even less information on the expected damping trends between the lower frequency modes and the higher frequency modes, although test data suggests that there are wide variations. And then there still are large uncertainties as to the damping trends between 1-g and 0-g response, as has been illustrated on IPEX [4].

The overriding problem is that damping values cannot be made artificially low, as it then becomes design drivers for other subsystems within the error budgets. One course of action would be to insure that specific damping in achieved in the system through either passive or active means. However, damping strategies typically target a narrow frequency band. Hence, damping of the uncontrolled modes still remains a requirement factor.

The sixth and final mode of microdynamic behavior is wave propagation effects. This mode refers to the response uncertainty due to the propagation of transient waves through a jointed structure. The analysis proposed above for microdynamic snaps uses standard linear models, and no allowance is made for localized energy dissipation at joints, or for wave confinement within a component of the structure. This mode is documented here for completeness of the microdynamics phenomena, and implementation models have yet to be proposed. However, analysis techniques similar to the ones adopted by the statistical energy analysis (SEA) community should be investigated.

4. MODELS OF JOINT SLIPS

The susceptibility for a joint to slip is a function of the specific joint design, the mechanical load currently being transferred by the joint, and possibly the loading history. Transfer of forces across the interface by tangential friction, a design detail discouraged for precision structures, results in strain energy being stored in the joint, which in turn could potentially be released into the structure. If the strain energy is released slowly enough, there will be no structural dynamics excited, although some static offset and misalignment will be present after the slip. Such a gradual release of energy might be associated with frictional microslip across the surface. A more conservative (and interesting) assumption, suitable for preliminary design and requirements validation, would be that the release of energy is caused by the sudden slip of the interface, a sudden breaking of frictional lock, resulting in dynamic excitation of the entire structure.

The precise mechanics that transpire during a joint slip are complex and not generally amenable to precise modeling. Fortunately, though, if the purpose of the analysis is to bound the dynamic response, the exact kinematics the joint slip need not be known, as long as it can be shown that the slip used for the analysis leads to a reasonable bound on the structural response. Using as an analog a simple SDOF model subjected to support motion, it can be demonstrated that the bound for structural response is caused by the most rapid change in boundary position; ie, a step function. Due to the inertia of the moving elements, a ramp function of very short rise time is more appropriate than a step function, but as argued below, this is a fine distinction.

Ramp functions are completely described by the rise time, or slip duration, and the amplitude, or slip distance. The amplitude must be dependent on joint design and possibly loading history, and might be determined from joint geometry or perhaps from quasi-static testing of the actual hardware. For rise time, an engineering estimate is that joint slip velocity lies in the range 1 to 1000 micron/sec. However, knowledge of the specific rise time (or slip duration) may not be necessary either, since if the slip is less than 1/10th the period of the fastest mode of interest, the resulting structural displacement is largely insensitive to rise time and deviates less than 2% from the maximum displacement due to a step function; i.e., fast ramps are indistinguishable from step functions. Velocity of structural response is more sensitive to rise time, but like displacement, velocity becomes less sensitive as slip duration decreases.

These observations and the assumptions that they lead to form the basis of a rational method to bound the effects of joint slips, before the structure is built, before the joints are designed, and before the loads acting on the joints are even known. Using these bounding models for joint slips, requirements can be established and validated for the interface components.

A Case Study: The SIM Model

The Space Interferometer Mission (SIM) is a space-based 10-m baseline interferometer under development at Jet Propulsion Laboratory, and targeted for launch in 2006 [8]. As illustrated in Fig 3, the spacecraft consists of four primary systems: Precision Support Structure (PSS or optical bench), external metrology boom, the spacecraft bus, and solar array. At this point in the project evolutionary cycle, the emphasis is on mission requirements definition and preliminary design phases. Using a reference system design, a multidisciplinary (structural, thermal, control, optical) finite element model has been constructed using the Matlab-based program IMOS [9]. This integrated model is capable of predicting structural responses to mechanical and thermal loads, as well as predicting critical optical performance metrics such as interferometer fringe positions and wavefront tilt angles.

To facilitate the rapidly changing design of the various elements of the spacecraft, and to keep the system-level model to a reasonable size, the Craig-Bampton component-mode-synthesis substructuring method was used [10]. In the Craig-Bampton CMS technique, the dynamic response of each component of the system is represented as a set of constraint displacement patterns and a small number of normal modes and generalized coordinates. The components are joined together by kinematic compatibility equations written for the substructure boundary degrees of freedom. The SIM model has sixteen unique components, and in general each component is modeled with just enough of the normal modes to represent the critical response. Fully assembled, the system model has about 1275 normal modes. Historically, the SIM integrated model has been used to

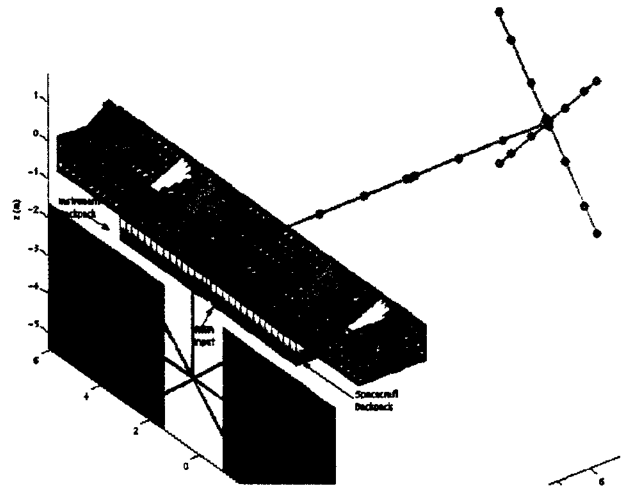

Figure 3. Finite element model of SIM preliminary design. Note the four pairs of telescopes on top of the PSS.

generate transfer functions for analysis of reaction wheel disturbances; the use of the integrated model to analyze the system for microdynamic joint slips represents a new area of investigation.

A number of alternatives were considered to implement the joint slip in the finite element method. The selected method must be computational efficient, and of course be consistent with the analysis software and analysis methods available. In itself this can present a significant challenge if working with a general-purpose finite element program. After weighing the alternatives, it was decided to use a double-node at the slip location, and to apply equal-but-opposite external forces at the two distinct nodes, in the direction of the desired joint slip. The forces have a ramp time variation that forces the two nodes apart according to the desired rise time; the magnitude of the forces is determined by the desired slip distance divided by stiffness of the artificial spring spanning the joint in the slip direction. Stiff springs also connect the other five DOFS at the slip joint. Selection of spring stiffnesses can be trick because the added flexibility should not interfere with the structural dynamics, but should not be too stiff that computational costs get excessive. It is absolutely essential to include a deformation pattern that describes the joint slip deformation in the component representation. A normal mode or a Ritz vector work equally well for this. This generalized coordinate must be retained throughout the system analysis to obtain accurate results.

After the system model is synthesized and its normal modes have been determined, the system normal modes are numerically integrated to obtain the transient response due to the joint slip. Economy is realized at this stage by first performing a response spectrum analysis and ranking each mode according to the contribution it makes to each response quantity of interest. Only the modes with significant contribution need be retained in the solution. Depending on the desired output, the pre-analysis can eliminate between 60% and 90% of the computational expense of the numerical integration compared to integrating all 1275 system modes. In fact, the solution can be terminated entirely after the response spectrum analysis if the predicted responses are far below the allowable levels. Efficient numerical integration of the modal equations is accomplished with an unconditionally stable Newmark method [11].

To illustrate the analysis method, a joint slip has been applied to the base of the metrology boom. This location was chosen because it was thought to be the worst slip location on the spacecraft (although that has not been verified yet) and because in some ways was the easiest location to work with. A rotational slip adjacent to the attachment to the PSS and with a vector direction transverse to the boom longitudinal axis was simulated. The magnitude of the slip was arbitrarily chosen to be 1 micron of displacement with a 0.3m lever arm, or about 3.3 microradians of rotation. A slip duration of 0.01sec was assumed, and 30 seconds of transient response were computed. Shown in Figure 4 is the first five seconds of computed structural motion of an optical reflector (corner cube) located at a vertex of the metrology boom-and-kite structure. As briefly described below, a total of 16 structural and optical performance metrics are of interest to the SIM mission requirements.

For the error budget flowdown process, the joint slip analyses described above are being used in the following manner. SIM science objectives were flowed down to optical quality requirements, which in turn were flowed down to structural motion requirements at key points, particularly at support points for the optics. These

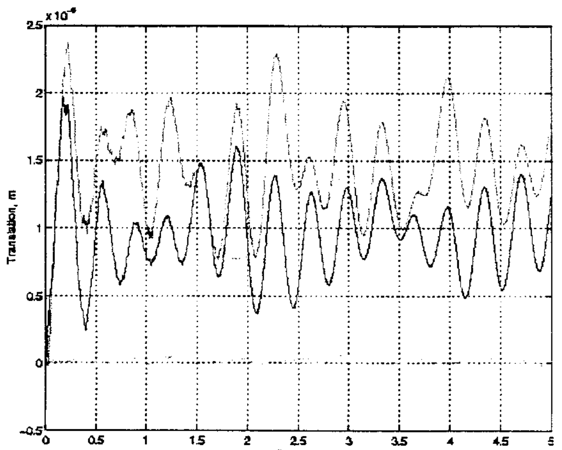

Figure 4 Translational motions in three dimensions of metrology kite vertex.

requirements include: 1) structural motions of the kite vertices relative to optical elements located on the precision support structure, 2) rate of path length difference for the external metrology laser beams, 3) optical path difference for incoming starlight between interferometer pairs, and 4) wavefront tilt. Some of these requirements relate to maximum instantaneous quantities, while others relate to a root-mean-square value for 30 seconds. The simulated output is then compared to the allowable requirement, and the ratio of allowable to simulated can be used to scale up or down the assumed joint slip magnitude, thus determining the allowable joint slip magnitude. Therefore, there is a distinct and manageable flowdown from mission science requirements down to allowable joint slip magnitudes.

5. MODELS OF NONLINEAR STIFFNESS EFFECTS

As was argued previously, nonlinear stiffness errors principally affect response predictions of steady state sinusoidal motions because of instabilities in the regions of the modal frequency [Fig. 1], and because of harmonic distortions (Fig. 2). This is particularly relevant to space precision platforms, since reaction wheels are the predominant source on on-orbit disturbance. Furthermore these types of harmonic distortions have been observed both in microslip conditions, as observed in the IPEX data (Fig. 2) as well as in gross slip situations as observed during testing of the CASSINI spacecraft (Fig. 6) [13]. Hence it is believed that model forms representing stiffness degradation at high load levels, should also be valid for low load levels.

Iwan has proposed models for generating frequency response functions of hysteretic systems subjected to steady-state excitations [12]. The model is a distributed

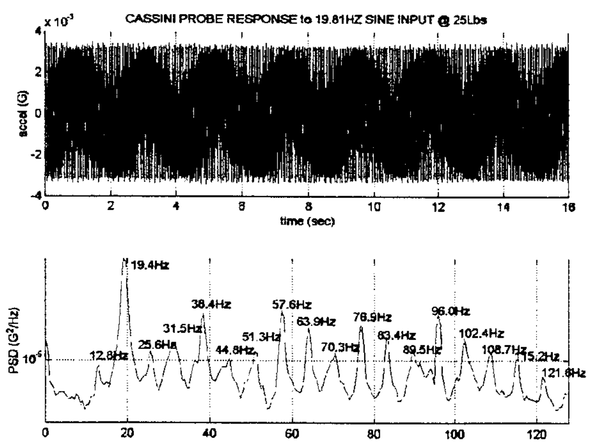

Figure 6. Probe response measured during sine dwell tests at the fundamental frequency of the bounce mode

system of discrete elasto-plastic elements that represents progressive stiffness degradation. This type of hysteresis models falls under the broad classification of elasto-plastic systems, which encompasses macroscopic behavior such as gross Coulombic slip in bolted interfaces down to microslip stiffness degradations and non-deteriorating yield of materials. This is in agreement with our experimental observations, whereby harmonic distortions have been measured in both high level and low level test data.

The test data obtained during the high level CASSINI modal tests is used to validate the Iwan modeling approach for elasto-plastic degrading systems. The localized nonlinearity observed during the CASSINI spacecraft modal test has been previously reported in [13]. In short, the support joints between the CASSINI spacecraft and the HUYGENS probe mount produced highly nonlinear response during the step-sine tests performed around the primary bounce mode of the component at 19.8Hz. A sample of the step-sine data is shown in Figure 6. The power spectral density (PSD) of the output response shows the characteristic harmonic distortions of nonlinear stiffness, as reported earlier. The transfer functions, obtained from step-sine tests performed around the bounce frequency for force amplitudes ranging from 2lbf to 60lbf, showed severe stiffness degradation as a function of forcing amplitude. (Figure 7). In particular, each data set exhibits a jump in the transfer function estimate as the sine-step frequency moves up across the zone of instability (Figure 1).

A model of the nonlinear stiffness degradation was implemented using the Matlab and Simulink software tools [14]. Using a single degree of freedom representation, the individual microslip stiffness elements were computed such that at the lowest force level the effective stiffness of the sum of the individual microslip elements approached the linear modal frequency. Each individual microslip element was associated with a yield level above which it did not contribute to the effective stiffness of the overall system. For the purpose of approach validation, only 11 microslip elements were included in the analysis. More elements are recommended to increase the fidelity of the prediction. Detailed description of the microslip models and parameter identification process will be documented in a future publication.

A simulation of the actual step-sine test process and estimation was implemented in SIMULINK, in which for each step-sine frequency and for each input sine amplitude the time history response was run until steady-state conditions were reached. Then for each forcing frequency, the amplitude of the transfer function at the frequency of excitation was identified. It is noted here that since the behavior was nonlinear, modal distortion similar to those in the actual test data induced responses at frequencies other than the input frequency [Fig. 6]. The transfer functions were then assembled incrementally for the whole bandwidth of the step-sine test, using as initial condition the last data point from the previous sine frequency response. The analysis can be run for either increasing step frequency or decreasing step frequency. Results for the increasing step-sine test simulation using the distributed elasto-plastic model is shown in Figure 8 for forcing amplitudes of 2 lbf, 25 lbf, 50 lbf. By comparison to the test data in Figure 7, the analytical simulations produced almost identical results, including the location of the jump frequencies. Similar agreements between the data and the analysis were obtained for the decreasing step-sine transfer functions.

The CASSINI data has validated a model form for degrading elasto-plastic systems. These nonlinear models can be incorporated as components into system level models, and then exercised for system level performance assessments. This implementation as yet to be executed on SIM for error budget allocation. However, it is envisioned that stiffness degradation will be parameterized in terms of deviation from linear stiffness over the expected load range.

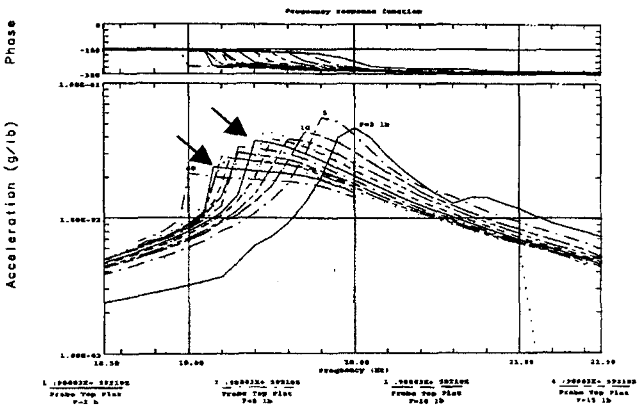

Figure 7 Transfer functions obtained through step sine testing about the bounce mode frequency of 19.8Hz, for input forcing amplitudes ranging from 2lbf to 60lbf.

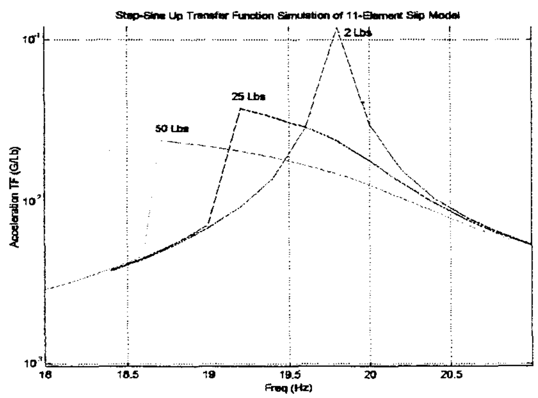

Figure 8. Analytical transfer function obtained using microslip model.

6. CONCLUSION

The microdynamic effects of nonlinearities in the small have been decomposed into six modes that can be traceable to system level behaviors relevant to future precision space structures. These modes of microdynamics are static slip, creep, impulsive snaps, nonlinear stiffness, nonlinear damping or velocity dependent effects, and propagation effects. Modeling approaches have been suggested for each of these microdynamic modes. These parametric models are to be used for the error budget allocation process in the early phases of the flight system design. Implementation of the snap disturbance modeling approach has been shown for the space instrument SIM, and validation of the microslip model used for stiffness degradation has been demonstrated on the CASSINI spacecraft test data.

It is recognized that the system level implementations of microdynamics effects are new, and are susceptible to change as our understanding of the phenomena mature. The proposed implementation is based on our best engineering judgment, and is intended to be as generic as possible and not dependent on any particular design configuration. This strategy has never been applied before, compared to the 40 or more years of the launch loads analysis and testing procedures. Furthermore, working in the small poses special challenges in itself, whereby standard linear mechanical testing and analyses techniques are no longer applicable. Hence, there is a scarcity of experimental data for validation of the process and of the models. Furthermore, there has been no "microdynamic failures" to guide us towards the elements of highest risk for the missions. For this purpose, future proposed flight experiments such as NEXUS [15] and MADE [16] will help validate the implementation processes defined herein.

7. ACKNOWLEDGEMENTS

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