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Nanometer-Scale Spontaneous Vibrations in a Deployable Truss Under Mechanical Loading

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Evidence is reported of spontaneous vibrations of a deployable spacecraft structure at optical levels of motion under mechanical loads. In these experiments a truss was mechanically loaded while in a thermally and vibrationally stabilized test environment. The applied loads were smaller, by a factor of 20, than the load that would have caused gross slippage in the frictional interfaces of the structure. Even so, results indicate that infrequent, spontaneous vibrations occurred. Most of these vibrations occurred after the application and removal of a mechanical load. The response spectra of the vibrations occurred in narrow bandwidths around the dominant modal frequencies of the structure. The vibrations ranged in amplitude from 4 to 20 nm and in velocity from 2 to 8 μ/s. Potential error sources are eliminated, including unmeasured environmental excitations and sensor errors. These vibrations are presumed to arise from the sudden release of strain energy stored in the hysteretic mechanisms and materials of the structure. An analysis based on this hypothesis shows that the amplitude of the observed vibrations is quantitatively consistent with this theory.

Introduction

THE ability of a structure to remain dimensionally stable under applied loads is a fundamental requirement for any precision structure. How a design achieves a desired level of structural stability depends in part on the scale of deformation considered. For large deployed optical telescopes in space, this stability must be maintained to within a fraction of a wavelength of light, perhaps as little as a few nanometers, and over a bandwidth of perhaps 1000 Hz.[1,2] Unfortunately, this range of structural deformation is so small that little information is available in the literature about structural mechanics and stability at this scale.

Although there are accounts of structural and material instability in micron levels going back 100 years,[3] it was only in recent years that instruments were developed to the required nanometer and microgravity levels of resolution needed to systematically study this issue. Also recently, there have been theoretical developments that indicate friction and anelasticity, long recognized as large-motion nonlinearities, are important nonlinearities at small levels of motion as well. Although our understanding of these issues is incomplete, there is reason to expect that structures might exhibit mechanics under very small deformations which are different from what would be expected from large-deformation models.

For example, Bullock and Peterson showed that hysteresis and stiffness nonlinearity were present in a deployed structural joint even under nanometer deformations.[4,5] These responses are caused by the uneven distribution of stresses present within any metallic interface. Hinkle and Peterson[6] demonstrated further that these intrinsic properties of rough, frictional contacts sometimes lead to instabilities under shear load. Not only can frictional interfaces exhibit microslip,[6,7] but they can also exhibit rate dependence, which sometimes is manifested as a time-dependent stiffness degradation and viscous-like mechanics in the frictional interface.

A major concern about deployed optical structures follows from these observations. At large-scale motion (i.e., on the order of many tens of microns or larger) it is known that the stability of a frictional interface under shear load can become unstable, resulting in a sudden vibration of the structure.[8–10] Sometimes, these vibrations are referred to as stick-slip instabilities. They are behind the mechanics of what causes a violin string to vibrate,[11] what causes a tennis shoe to emit sounds when dragged across a floor,[12] and what causes the sound of fingernails dragged across a blackboard. These phenomena are familiar to the dynamics literature.

Perhaps what may be little appreciated is that these frictional instabilities are triggered at a particular magnitude of interface shear displacements and speeds.[10] This critical shear displacement is on the order of the mean size of the contact asperities in the interface, which is typically on the order of 1–10 μ. The critical shear speed is a threshold beyond which the friction force becomes unstable, as a result of rate-dependent effects. Unless the load and load rate causes the frictional interface to approach both the shear displacement and speed threshold, the common empirical models of friction predict the interface should be stable. Under these conditions a jointed structure should retain its stability, undergoing at most progressive microslip shearing in the friction interfaces. Said another way, if gross slip is not induced in the friction interfaces in such a way to create an unstable stiffness there should be no sudden release of strain energy.

If this were indeed the case, precision structures could simply be designed to avoid this condition, remaining in the so-called microslip loading regime. Most space structure and mechanism designs probably already do this. By operating below the friction load capacity or by increasing it through increased assembly preload, the expectation is that structural instabilities will not happen. However, this expectation is based on models derived entirely from empirical evidence gathered at the macroscopic and microscopic deformation scale. Space telescopes need to be stable at the nanometer scale (and perhaps below). With no validated theory to guide expectations, then the first steps must necessarily be experimental.

Because of the potential significance to large space telescopes, a few recent experimental studies have attempted to identify nanometer-scale instability phenomena. The first evidence of such spontaneous vibrations were found on a shuttle-based flight experiment,[13–16] the Interferometry Program Flight Experiment II (IPEX II). IPEX II was flown on an ASTRO-SPAS on STS-85 in August 1997 and measured the response of a deployed mast to thermal and mechanical disturbances. Perhaps the most important

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observation occurred during one particular transition from nighttime to daylight, when the ASTRO-SPAS systems were completely quiet. As reported in Ref. 14, the heating of the truss over several minutes resulted in the generation of spontaneous, unexpected vibrations in the truss. An analysis of the thermal deformations confirmed that the induced loads should have never exceeded the Coulombic friction limit within the truss joints, although it was determined that Coulombic friction limits might have been exceeded in the mounting hardware at the base of the truss.[16]

Ingham repeated similar thermal cycling experiments in 1 g on a different structure, the Middeck 0-Gravity Dynamics Experiment (MODE) truss.[17] Like the IPEX-II results, these data also indicate spontaneous vibrations under thermal loads. Transient displacements induced by these spontaneous vibrations were estimated to be on the order of nanometers concentrated at 9, 14.5, and 18 kHz. It was not noted whether the induced loads were within the microslip regime; however, we presume that was the case, based on the scale of the observed vibrations.

The most likely explanation of these observed vibrations is that a thermal load induces mechanical loads within the joints that exceed some unknown stability condition of the frictional interfaces. In the research reported in this paper, the goal is to test this hypothesis by applying a mechanical load, not a thermal load, to a structure, and recording the response, even during periods of no loading. Unlike experiments in which the loading is thermoelastic, mechanically induced loads within the structure can be accurately predicted and controlled, making it possible to assess quantitatively different hypothetical explanations for the ensuing vibrations.

In the experiments reported in this paper, a heavily instrumented single bay of the IPEX-II structure was mechanically loaded in a stabilized test environment. The applied loads were smaller, by a factor of 20, than the load that would have caused gross slippage in the frictional interfaces of the structure. Even so, results indicate that spontaneous, though infrequent, spontaneous vibrations occurred. The vibrations occurred in narrow bandwidths around the dominant modal frequencies of the structure. They ranged in amplitude from 4 to 18 nm, and they ranged in velocity from 2 to 8 μ/s. Potential error sources are eliminated, including unmeasured environmental excitations and sensor errors. This paper therefore presents the first evidence in the literature of a mechanical quasi-static load inducing vibrations in a structure under microslip conditions.

In addition to these observations, the paper poses a theoretical explanation of the phenomena. An hypothesis is presented in which the vibrations are presumed to arise from the sudden release of strain energy stored in the hysteretic mechanisms and materials of the structure. An analysis based on this shows that the amplitude of the observed vibrations are quantitatively consistent with this theory.

The paper is organized as follows. The next section presents the experimental configuration, apparatus, and protocols. Then, the discovery of the observed phenomena is explained, and potential error sources are systematically eliminated. This is followed by a presentation of the theoretical explanation, in which an energy balance is shown to bound the magnitude of the vibrations. Such a bound may pose a design methodology that compensates for these spontaneous vibrations.

Experimental Configuration and Protocols

Test Apparatus

The test article studied in these experiments is a single-bay truss built from components identical to those of the IPEX boom. The single bay, like the boom, has both a high structural efficiency (stiffness to mass ratio) and high packaging efficiency (deployed to packaged volume ratio, approximately 17:1). The battens and longerons of the truss are made of high-modulus graphite/epoxy composite. The corner fittings, including the fitting housing and the ball joints, are stainless steel. The diagonals are made of braided steel cable and are pretensioned by entrapment within a steel "detent" at the completion of deployment. This design ensures the joints are preloaded under a range of temperatures and mechanical loads.

For these experiments the bay was constrained between two aluminum plates and then bolted to an isolated optics table. Figure 1 shows a rendering of the experimental configuration, and Fig. 2

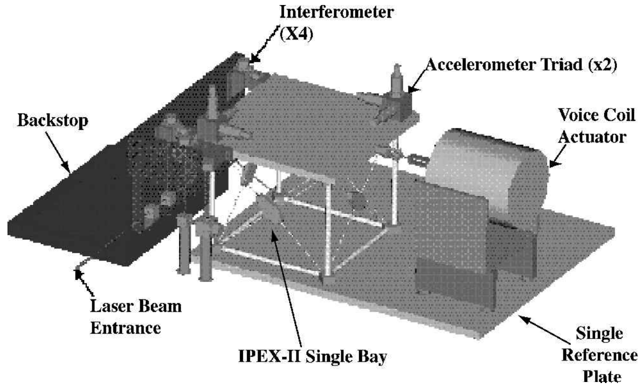

Fig. 1 Rendering of the experimental configuration shows the major components.

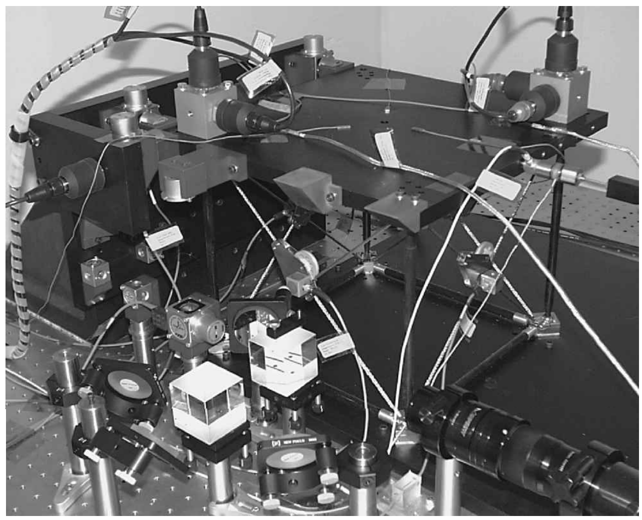

Fig. 2 Experiment included many sensors as well as mechanical actuation.

shows a photograph. The table was located inside a thermal and acoustic stabilization chamber (TASC). The TASC was designed to provide a stable mechanical and thermal environment sufficient for studying nanometer-scale vibrations and mechanics of components and small deployable structures. The chamber walls provide both an acoustic shield and thermal insulator. Because of this, the isolated optics table within the chamber has a mean ambient vibration level of 6–10 μg over a 1-kHz band. Also, the temperature of the table and test articles mounted to it have a long-term stability of approximately $\pm 0.1^\circ$C and 1% relative humidity over periods of 5–10 days. The thermal stability over a few hours is approximately 0.01°C. More details of the experiment, the design and evaluation of the test environment, and other test results from the same test article can be found in Ref. 18.

A voice coil actuator applied the mechanical loads to the top plate of the bay. Two different locations were used, one in the center and one at a corner. In both cases the load was a shear load in the plane of the top plate. Both ac and dc load cells, with resolutions of 0.002 and 0.01 lb, respectively, were placed in the path between the actuator and the structure to measure the actual load being applied. The current and voltage going into the voice coil actuator were also monitored.

The bay was instrumented with multiple accelerometers. Two triads of three accelerometers, with resolutions of 8 μg, were located on the top plate. Triaxial accelerometers, with a resolution of 40 μg, were placed on three of the detent housings at the intersection of the cables on the faces of the bay. Temperature sensors with a resolution of 0.01 K were co-located with the accelerometer triads. The strain gauges, with a resolution of 50 picostrain, were located on the top plate at 0, 120, and 240 deg (where 0 deg is along the X axis). The acceleration and temperature of the TASC optics table were also monitored.

In addition to the sensors just described, four interferometers were used to measure the rigid-body motion of the top plate. Two measurements were in the X direction (the direction of actuation) and one each in the Y and Z directions. (Z is vertical from the table.)

All cables were taped down with nonconductive, insulating Kapton® tape. Cables that were particularly susceptible to external vibration, such as the triaxial accelerometer cables, were encased in an RTV compound. This served to control the configuration of these small load paths, and to control loads that might have been passed through the instrument cables.

Pretest Analysis

The principal independent experimental variable in these experiments was the applied load. The intent was to study loads that should be much smaller than the frictional load capacity of the structure. Thus an important experimental design analysis was completed to determine the structural load for which gross slip would be expected in the mechanisms.

A linear finite element model of the test article was constructed for this purpose. The configuration of the joints on the IPEX bay is such that a full microslip analysis including friction would challenge the state of the art in finite element modeling. In this case not only are the interface patterns caused by machining tolerances of the ball within the socket unknown, there is additional load on the joint from the bolts holding the assembly together and also from the diagonals. However, a simplified linear slip analysis was conducted in MSC/NASTRAN in order to determine where the slip and stick regions of the interface might be located. This analysis assumes full stick throughout the region and uses linear constraints. Although such an analysis cannot predict the details of the progression of microslip, it is useful for predicting the load corresponding to complete slippage in the interface.

The model included details of each of the mechanisms and modeled each of the joints with solid elements. Rigid gap elements were used to recover the internal loads induced normal and tangentially within the structural joints. As detailed in Ref. 18, this model was able to estimate the load for which the induced shear across the weakest joint exceeds the frictional load capacity of that joint. Although such a linear model cannot predict the deformation under microslip, it should predict the load distribution at the onset of gross slip. Any correction in this assumption should scale with the redundancy of the truss members in bending and therefore should be small. Consequently, the experiments ensured microslip conditions by keeping the applied forces to less than 5% of the load capacity computed with this model.

Testing Protocols

To begin the test, the laser interferometers were aligned. The chamber doors were sealed closed, and the environment was allowed to stabilize for 12–24 h.

Data were acquired at a 5-kHz sampling rate with a window collection length of 13.1 s. It was desired to have the sampling frequency between 10 and 20 kHz. However, hardware constraints allowed only for a maximum of 5 kHz when sampling all sensors. (In contrast, the spontaneous vibration response seen in the IPEX-II Boom on orbit was sampled at 1 kHz.)

A suite of test protocols was applied to characterize nonlinear mechanics in a standard format for comparison with analytical models. These protocols were extensions of those commonly used for material testing under nonlinear yield conditions. They were initially explored for nonlinear mechanism frictional yield in previous research.[5,6,19,20] Each test type and the information gained are presented in more detail in Ref. 18. The results of these tests on the IPEX Bay test article are the subject of another paper.

Testing was completed between 2300 hrs and 0600 hrs, when interruptions caused by ambient vibrations (such as traffic, building systems, etc.) could be avoided for long periods of time. Seventy gigabytes of data were collected over the course of four months. These data were scanned for possible spontaneous vibrations, as described in the following section.

Observed Spontaneous Vibrations

Initial Detection

Detection of spontaneous vibrations proved to be difficult because of the volume and bandwidth of the data that were collected. For a long period of time, it appeared that there were not any spontaneous vibrations in the data at all. In fact, the apparatus was disassembled and returned to the Jet Propulsion Laboratory, Pasadena, California, while data analysis was being completed. In hindsight, plots of the full-bandwidth data were masking the vibrations, which were actually concentrated in narrow frequency bands near the lowest shear mode of the test article. It was not that the spontaneous vibrations were below the resolution but that pixelation in the full bandwidth hid the vibrations from the human eye.

The first discovered spontaneous vibrations were found almost by accident when a narrowband filter was applied to a data set following a quasi-static mechanical load. Figures 3 and 4 show this event. This particular vibration occurred after the load had been removed from the system. As can be seen in Fig. 3, the vibration was 200 μg in the X direction, 60 μg in the Y direction, and 6 μg in the Z direction. As seen in Fig 4, the maximum displacement in the interferometer is 6–8 nm in the X direction and 2 nm in the Y direction. If there is any response in the Z direction, it is less than 0.5 nm. The acceleration magnitude and the interferometric displacement magnitudes are consistent.

This spontaneous vibration is identified as event 1 in Table 1. This particular event was greater than five times the standard deviation of the background persistent response, and so it was not a random excitation. The event was also narrowband, occurring near the first shear and torsion modes. A spectrogram indicating this is shown in Fig. 5. These modes, with a natural frequency of near 86 Hz, were also the modes with the greatest projection on the applied loads. The response was noticeably symmetric about the peak, with an increase and decrease from maximum amplitude at similar rates. Because the measured vibration damping of the first several modes was approximately 0.5% (Ref. 18), the decay rate is far too quick to be an ordinary linear transient response.

Summary of Other Detected Spontaneous Vibrations

Many such events were discovered using an automated threshold detection algorithm. The number identified depends on the threshold chosen for the detection. In this paper only spontaneous vibrations above five times the standard deviation of the persistent vibration are presented, as these have less than a one in a million chance of being random responses to the persistent background. These events are identified in Table 1. Event 3 had very similar characteristics to event 1. Events 2 and 4, however, have a few differences. Event number 2 is largest in the Y direction, as opposed to the X direction of the other three. The response is plotted in Fig. 6, and the load is in Fig. 7.

Table 1 Dynamic instabilities detected in the IPEX-II bay

Response number	File name (date recorded)	Record number	Acceleration amplitude $401+X$, μg	Velocity amplitude $401+X$, nm/s	Displacement amplitude $401+X$, nm	Time of response, s	Applied load
1	19990605_A	1	200	4	8	3	0
2	19990605_B	6	100[a]	2[a]	4[a]	4	0
3	19990612_D	36	200	4	8	1.5	0
4	19990613_E	6	400	8	13	1.5	1

[a] In $401+Y$ direction.

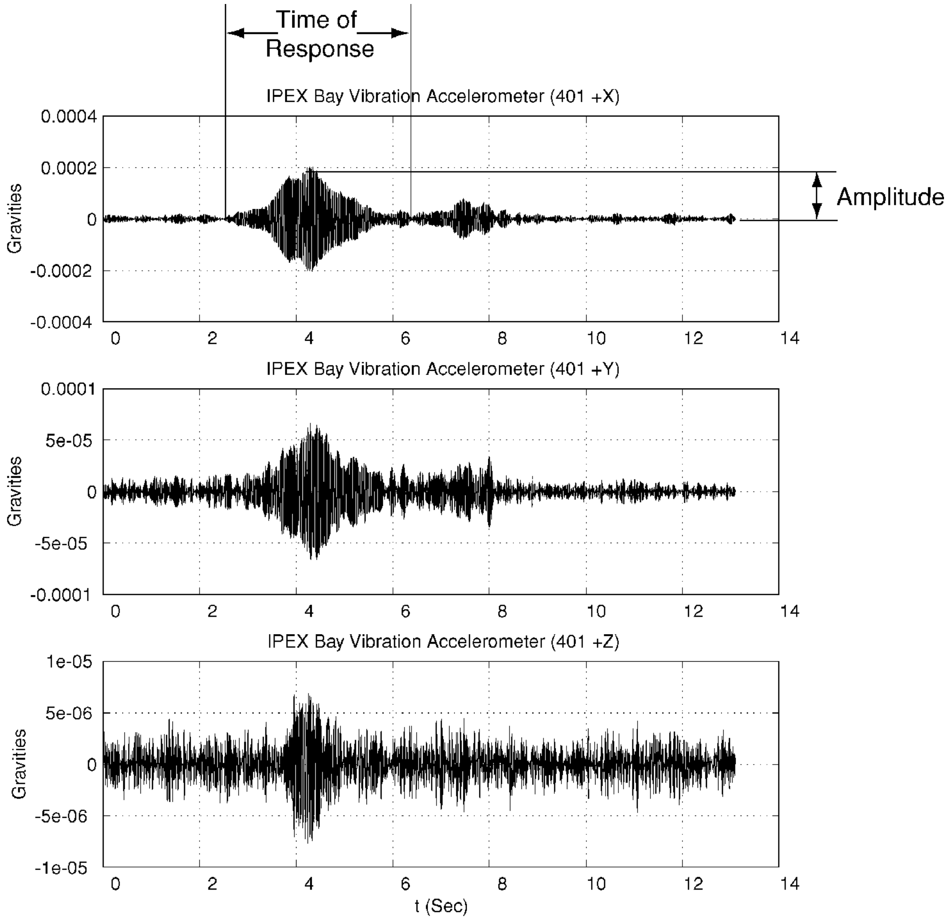

Fig. 3 Anomalous transient vibration response in structure accelerometers: event 1.

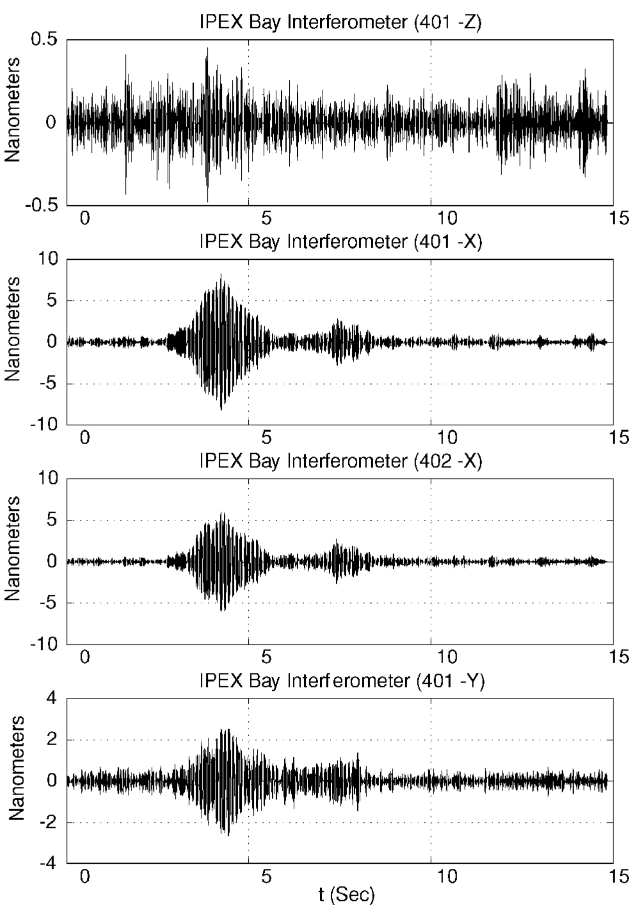

Fig. 4 Transient vibration response in the interferometer: event 1.

Event 4 is shown in Fig. 8. The applied load during this event is in Fig. 9. As can be seen, the transient occurred in the X direction, and it occurs shortly after a ramp-hold load was applied and before the load was reduced, but long after the load transient ended.

Estimation of Experimental Error

The source of the spontaneous vibration responses described was not immediately apparent. Because the theoretical expectation was that it should not have occurred, the possible experimental error sources needed to be eliminated. After considering many such error sources, including operator error, two particular experimental error sources required further scrutiny. These are 1) inadvertent actuation in the voice coil and 2) table vibration. The results of this analysis are summarized in the following paragraphs.

Inadvertent Actuation in the Voice Coil

The applied load was always measured, with not one but three signals: a dc coupled load cell (100-Hz bandwidth), an ac coupled load cell, and a voltage proportional to the applied current in the voice coil. The two load cells, which were located between the applied load and the structure, did not register a signal component from the voice coil that correlated with the observed response. However, it could have been caused by a force below the resolution of the load control system and measurement system (as will always be the case). This resolution was 2×10^{-5} lb at the frequency of the vibrations over the bandwidth of the window. Suppose that such a load signal error of this magnitude was created in the system, and all of its spectral energy were concentrated in the first mode of the structure (by random chance), and suppose that this caused a resonance of the first mode. The transient growth of such a resonance over a few cycles, followed by a decay in the same number of cycles is inconsistent with the observed light damping (found experimentally) in the structure. More important, no events were detected in the quiescent data sets, in which the voice coil actuator was powered, but no signal was sent to the command a load. The events were detected

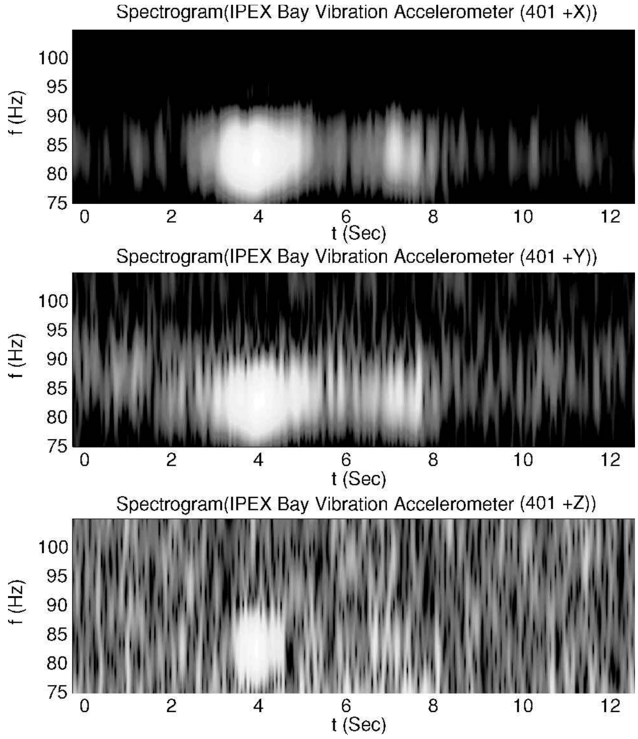

Fig. 5 Spectrogram of event 1.

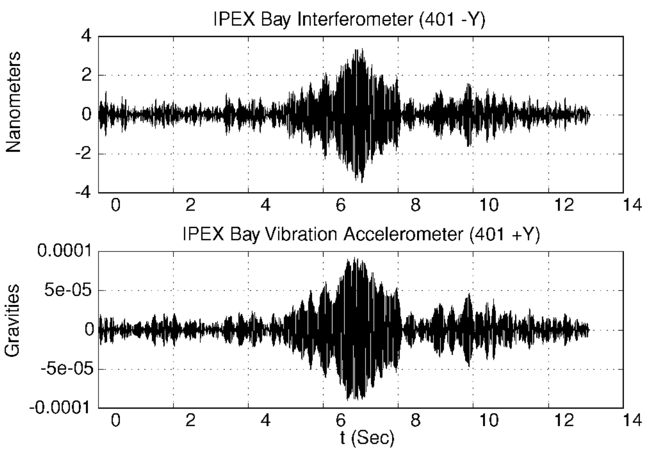

Fig. 6 Transient vibration event 2.

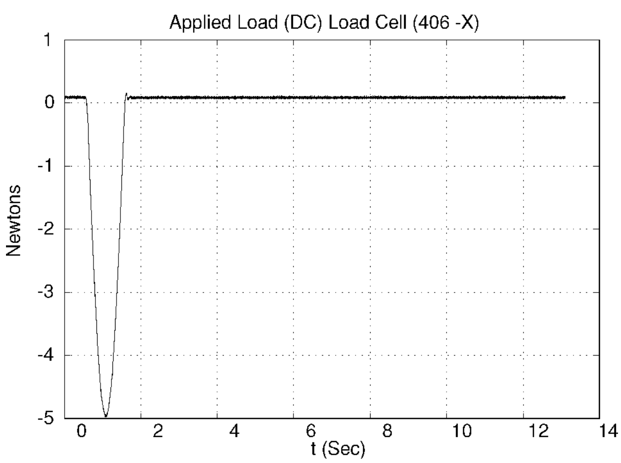

Fig. 7 Applied load for event 2.

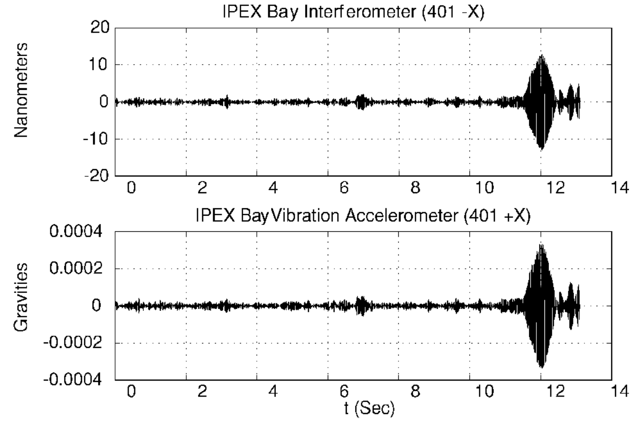

Fig. 8 Transient vibration event 4.

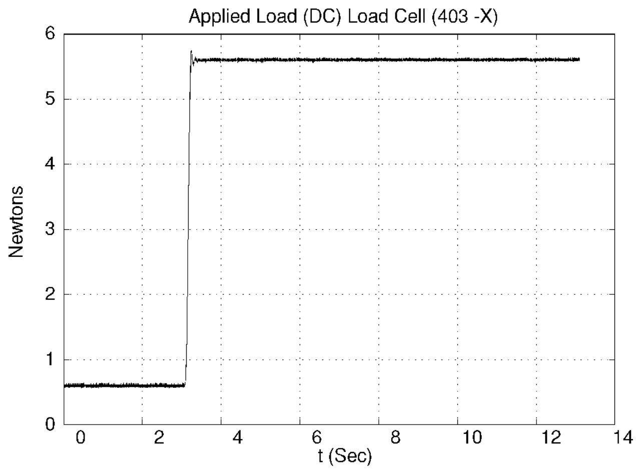

Fig. 9 Applied load for response number 4.

only in data sets immediately following an applied load and were not found in the long period quiescent data.

Table Vibration

At first we applied a discriminator to each event that asked, "did it also appear in the table vibration?" Upon further analysis, however, it was noted that this is not a satisfactory discriminator. If the table were reacting to a vibration originating within the structure, it would also vibrate a resolvable amount. The table mass was large but finite (2000 lb), and the structures moving top plate mass was small but finite (22 lb).

A more telling question might be that of temporal causality between the table and structural sensors. A finite element model might have provided insight into this question. However, the basic inverse modeling question here is one of deducing the location and nature of a dynamic system input, given only observations of the system response. This remains an unsolved problem, even for linear systems, much less for nonlinear systems such as the one in question here. As an alternative, signal correlation methods, similar to those used in earthquake analysis, were considered. Unfortunately, the 5-kHz sample rate proved too small to lead to conclusive results because the wave transmission time between the test article and the table vibration sensors was too small for this sample rate. Temporal causality could therefore not be determined.

Instead, consideration of the shape, magnitude, and spectral content of the table response does eliminate the table as the source. For example, the large transient seen in Fig. 3 can be seen in the optics table in Fig. 10. Because responses occur at the natural mode of the structure instead of the table, a disturbance traveling through or originating within the table would have had to be at or near the natural mode of the structure. The spectrogram in Fig. 11 shows the table vibration was narrowband in the mode of the motion. As can be seen, the frequency content in both the table and the structure

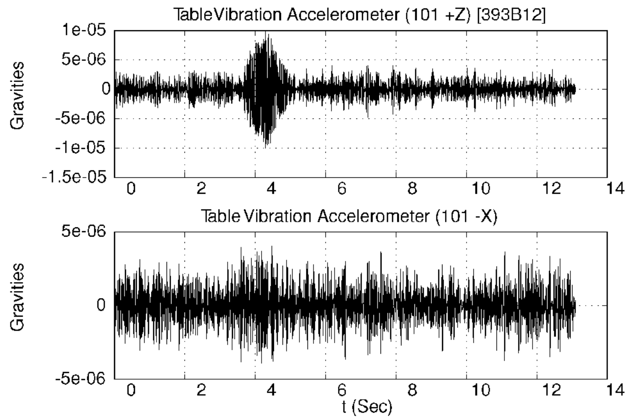

Fig. 10 Table acceleration during transient vibration event 1.

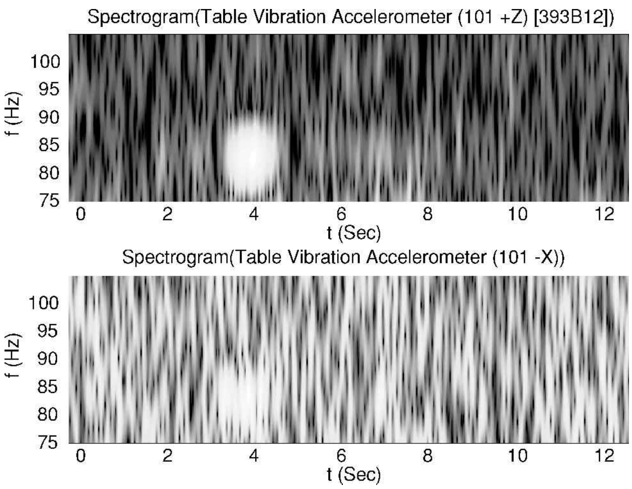

Fig. 11 Spectrogram of the table acceleration response during event 1.

is monotonal, indicating that the response is not a reaction to a broadband disturbance or a disturbance with traveling frequency that happens to cross the first mode (such as was typical with traffic noise). Moreover, the table vibration was vectorally different from the structural vibration. The greatest motion in the test article was in the horizontal plane, usually aligned 90% or better with the applied load. The greatest motion in the table was in fact vertically at a magnitude much smaller than either the horizontal or vertical motion in the structure. Thus, the table response is consistent with a reaction to a vibration originating within the structure, not from outside the environment of the test.

Analytical Explanation of the Observed Vibrations

The authors concede that doubt remains in these experimental error analyses and would argue this might always be the case with such high-resolution tests. The fact that the experiment was dismantled before the events were detected made it impossible to check the results by repetition. However, an analytical examination of a possible mechanical source within the structure provides evidence that the vibrations were instabilities originating within the test article. At least, that, the data are consistent with the following theory.

The basic assumption is that if such spontaneous vibrations, such as those observed here and elsewhere, are originating within the structure then they are the release of strain energy entrapped in the hysteretic mechanisms of the structure. (These mechanisms might include not only the joints but also the materials of the structure.)

The mechanics of hysteresis in frictional interfaces under microslip loads has both static and dynamic manifestations, including a viscous-like behavior resembling creep.[18] This viscous component in fact creates a large dynamic hysteresis, measured as the width of a load-cycle plot, even at load-cycle rates of 0.1 Hz or less, depending on the system being studied. Here, however, we limit consideration to entrapped energy held by the much smaller, truly static steady-state hysteresis, which remains after all loads have subsided. In the

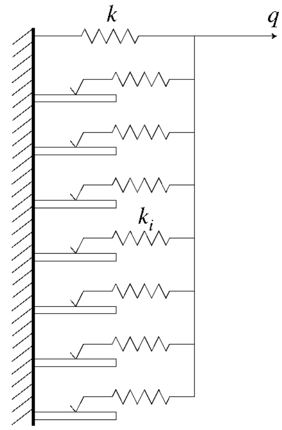

Fig. 12 Pictorial representation of the distribution of microslip elements within a structure.

case of the single-bay experiments just reported, this hysteresis was on the order of 20 nm, depending on the amplitude of the applied load. More details can be found in Ref. 18.

In general, the load paths, distribution of entrapped energy, and the details of the vibration will be complex. However, in this particular case there is the mass of the large top plate, which effectively makes this structure a single-degree-of-freedom (DOF) oscillator to a good degree of approximation. The stiffness of this single-DOF oscillator is the net shear stiffness of the entire bay, and the accumulated static hysteresis in all of the joints is manifested as the static hysteresis in the single DOF.

Figure 12 is a representation of the distribution of microslip elements within the structure.[21] Such a diagram is often used to represent an arbitrary number of multiple parallel slipping elements distributed within frictional interfaces. Each of the slipping elements is meant to have a different friction load capacity and material stiffness. The total static hysteresis in the structure, manifested in q, is the combined hysteresis in all of these sliding elements, projected through equilibrium against the elastic stiffness k.

Suppose that the structure has undergone an arbitrary load cycle, during which some of the slipping elements have slipped, and others have retained their unslipped state; this is the common explanation of microslip. At this point the deformation remaining in q will be the static hysteresis δ, and each of the individual slipping elements will have unknown hysteresis offsets of δ_i. Then, the total strain energy entrapped within the structure is

$$U_0 = \frac{1}{2}k\delta^2 + \frac{1}{2}\sum_i k_i(\delta - \delta_i)^2 \tag{1}$$

(The typical amount of energy involved here can be pico-joule in the single bay.) In most high-precision deployable structures, including the one considered here, the elastic stiffness k dominates, and as a reasonable upper bound the second term in the preceding equation can be neglected.

Now suppose that this entrapped potential energy is suddenly released. That is, the frictional slider elements suddenly and completely loose their ability to carry load. Now suppose that all of this energy is manifested as kinetic energy in the single-DOF q. Then, energy balance gives

$$U_0 \leq \tfrac{1}{2}k\delta^2 = \tfrac{1}{2}m\dot{q}^2 \tag{2}$$

Assume also that this ensuing vibration is nearly sinusoidal (there is only one DOF), so that velocity and displacement amplitude can be related by frequency in the following way:

$$\dot{q} = \omega q_0 \tag{3}$$

and if the amplitude of the vibration is solved for

$$q_0 = \delta \tag{4}$$

Thus it is that in this special, single-DOF case that the amplitude of the motion is limited by the magnitude of the static hysteresis in the structure. The multi-DOF case produces a much more complicated response, in which the energy can be expressed into many different modes, and the amplitude is not bounded by static hysteresis at a single point on the structure. Fortunately, the single-bay experiment produced vibrations in the single mode most actuated by the applied shear load. So the preceding bound would apply in the limit that the structure was dominantly a single-DOF system.

So, does this analysis predict a useful bound on the experimental observations? In fact, the preceding observed spontaneous vibrations were all in the range of the static hysteresis measured in the load-cycle tests, some approaching 90% of this value.[18] Thus, the data are consistent with this theory, and the theoretical bound is not overly conservative. Had the vibrations been caused by some unknown, random external load, as investigated in the error analysis of the preceding section, they should not have been bounded in this manner. The fact that they were provides internal consistency to the hypothesis that the observed vibrations were indeed spontaneous, arising from within the structure, and were caused by releases of energy entrapped by the static hysteresis.

The analysis also predicts that any release would result in an observable, possibly measurable, shift in position. The suggests that there might have been some low-frequency motion, but low-frequency noise made direct correlation with the transient vibrations inconclusive. What in fact might be the case is that the release was multidirectional and could not be predicted by a single-DOF model.

Conclusions

This paper reports the first evidence of spontaneous vibrations of a deployable spacecraft structure at optical scales of motion under mechanical loads. In these experiments a single-bay of the IPEX-2 flight-test article structure was mechanically loaded in shear within a stabilized test environment. An analysis showed that the applied loads were smaller, by a factor of 20, than the frictional load capacity of the structure. Even so, results indicate that spontaneous, though infrequent, vibrations occurred, most following, not during, the application of a mechanical load. The vibrations were monotonal at the dominant shear mode of the structure. They ranged in amplitude from 4 to 20 nm and in velocity from 2 to 8 μ/s. Potential error sources were eliminated, including unmeasured environmental excitations and sensor errors. An hypothesis was presented in which these vibrations are presumed to arise from the sudden release of strain energy stored in the hysteretic mechanisms and/or the materials of the structure. An analysis based on this shows that the amplitude of the observed vibrations are quantitatively consistent with this theory.

Because, for programmatic reasons, the test apparatus was dismantled before these vibrations were detected in the data, it was not possible to repeat the experiments and further study their causes and manifestation. It would be expected, for example, that the trigger mechanism of a frictional instability, even at this nanometer scale of motion, should depend on not only load magnitude but load rate. Future studies of this kind of response should examine this dependency.

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