Vibration Testing of Small Satellites

This series of papers provides a tutorial along with guidelines and recommendations for vibration testing of small satellites. Our aim with these papers is to help you (a) ensure the test meets its objectives in demonstrating flight worthiness and (b) avoid test failures, whether associated with a design deficiency or with excessive loading during test. Addressed are sine-burst testing, random vibration testing, and low-level diagnostic sine sweeps. Although much of the guidance provided in this series applies to CubeSats, the series is primarily aimed at satellites in the 50 – 500 lb (23 – 230 kg) range. Most of the guidance applies to larger satellites as well if they will be tested on a shaker.

The plan is for this series to include seven parts, each of which will be released when completed:

1. Introduction to Vibration Testing (released April 11, 2014; last revised July 19, 2017)
2. Test Configuration, Fixtures, and Instrumentation (released April 11, 2014; last revised July 19, 2017)
3. Low-level Sine-Sweep Testing (released May 13, 2015; last revised July 19, 2017)
4. Sine-Burst Testing (released April 28, 2017; last revised July 19, 2017)
5. Random Vibration Testing (released April 7, 2016; last revised July 19, 2017)
6. Notching and Force Limiting (released May 13, 2015; last revised July 19, 2017)
7. Designing a Small Satellite to Pass the Vibration Test (yet to be released)

The most recent versions of these papers are available for free download at

http://instarengineering.com/vibration_testing_of_small_satellites.html.

Part 2: Test Configuration, Fixtures, and Instrumentation

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A vibration test on a shaker requires a test fixture, which adapts the test article's mounting interface to the grid of threaded inserts in the shaker's slip table and head expander. The test may also require mass simulators to represent any significant missing items from the test-article configuration.

Test Fixture Design

If your small satellite has a standard mounting interface, such as the ESPA-class 15"-diameter bolt circle, the test lab may have a test fixture you can use. If not, you'll be responsible for design and manufacture of the test fixture. Be sure to coordinate early with the test lab to ensure your fixture is compatible with the shaker, given the considerations addressed below.

In most cases, we want the test fixture to be very stiff yet no heavier than it needs to be, and we want a flight-like interface. These characteristics may appear to be contradictory. Let's explore them:

High stiffness. You may have read or heard that the ideal fixture used for random vibration testing up to 2000 Hz has no modes of vibration below 2000 Hz. Although the intent of this adage is correct, the phrasing is not accurate. The ideal fixture is so stiff that, if you were to attach a rigid mass (and moment-of-inertia) simulator of the test article and then ground the base of the fixture at the shaker interface, the first mode of this combined structural assembly would be above 2000 Hz. This ideal ensures the test article's modes of vibration up to 2000 Hz are not influenced by the fixture. However, this ideal is often not practical, so a reasonable compromise is to ensure the fixture is stiff enough not to influence the first natural frequency (the ***fundamental frequency***) of the test article in each axis by more than one or two percent. You should analytically confirm that this is the case and also that any other key modes of vibration are not significantly affected.

Lightweight. Each shaker has a force rating, which is the maximum force it can exert on the moving mass. Force equals mass times acceleration, so the force rating translates into a peak acceleration the shaker can achieve. The moving mass includes the armature, the slip table or head expander, the test article, and the fixture. Depending on the target test acceleration, fixture mass may or may not be an issue. Check with the test lab. Regardless, a lighter fixture is easier to handle during transportation and installation, so try not to make it too heavy. Still, high stiffness is typically of higher priority than low mass. Use an aluminum alloy such as 6061-T651 for the fixture rather than steel because, for the same weight, an aluminum fixture will be thicker and thus stiffer in bending. Magnesium is used for some vibration-test fixtures, as its density is even less than that of aluminum, but we recommend aluminum when possible because it costs less and is typically more readily available.

Flight-like interface. Achieving a flight-like interface to the extent practical may be the consideration in fixture design that is most easily overlooked and most difficult to achieve. A rigid fixture cannot be truly flight like; its rigidity will constrain the test article's interface such that a failure that may occur during the mission, with a flexible mounting structure, would not occur during the test. (In such a case, structural analysis of the interface is an important supplement to testing for structural verification.) But there are things we can do in fixture design to ensure certain important aspects of the interface are flight like.

Consider Fig. 2-1, which compares flight and test configurations, with an example of over-constraint from the test fixture.

Figure 2-2 shows a suggested configuration for a test fixture for an ESPA-class spacecraft with a 24-bolt, 15-inch mounting interface.

Another way to provide a flight-like interface, of course, is to test the spacecraft on top of its separation mechanism. Doing so provides significant advantages, as noted below. A separation-mechanism simulator may be used instead of the actual mechanism if the simulator adequately represents geometry and stiffness.

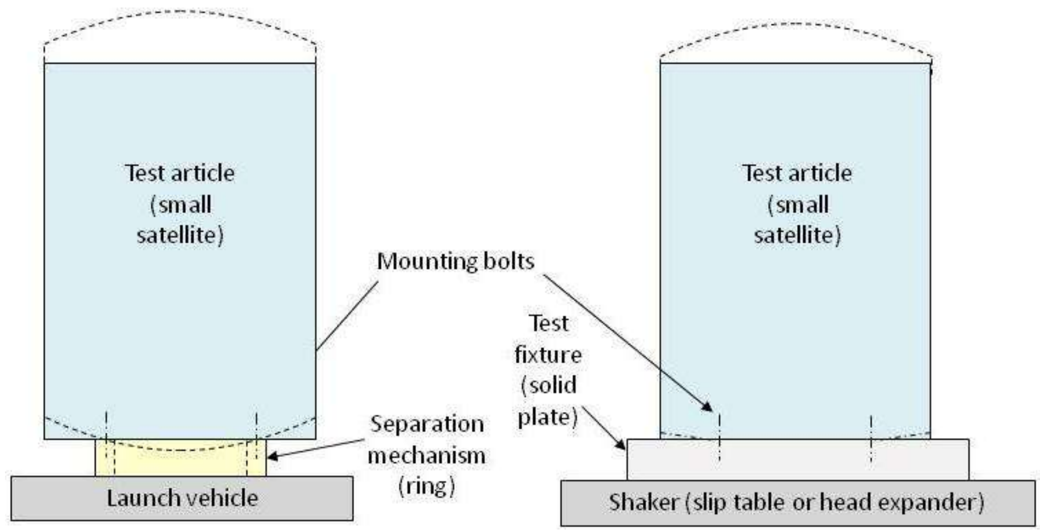


Fig. 2-1. Example of Over-constraint from a Test Fixture. The drawing at left shows the typical first axial mode of vibration for a small satellite during launch, with the base plate at the bottom of the satellite bending back and forth about the bolts that attach to the separation mechanism. With a solid fixture in place of the separation mechanism, as shown at right, the base plate can't flex below the surface of the fixture, as it can within the open cavity of the separation mechanism. The corners can move up, as shown by the lower dashed lines at right, but they can't deform below the fixture surface. As a result, the structure behaves nonlinearly, the first mode is at higher frequency than when in flight configuration, the interface bolt-load distribution is not flight-like, and the satellite is not properly tested.

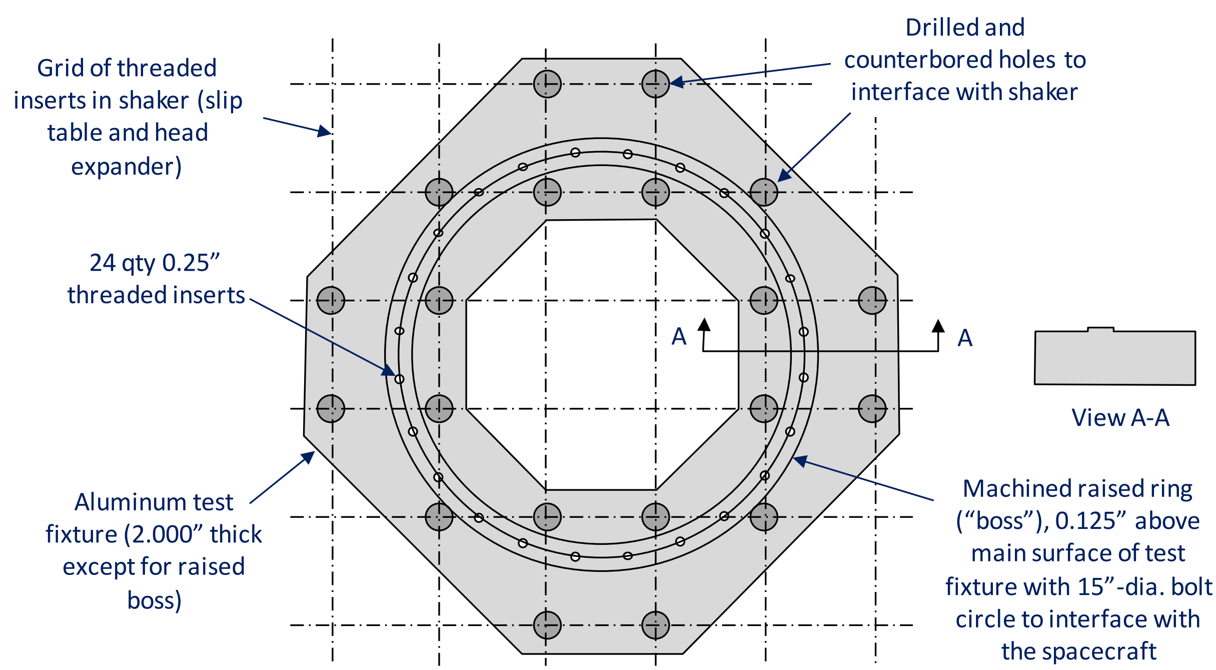


Fig. 2-2. Suggested Design for a Test Fixture. This design is compatible with a 4-inch shaker-insert grid and a 15-inch bolt circle for mounting the spacecraft; modify as needed to accommodate actual fastener locations. The counterbored holes keep the bolt heads from interfering with the spacecraft structure. The boss raises the spacecraft bottom surface from the fixture so that the structure can flex without making

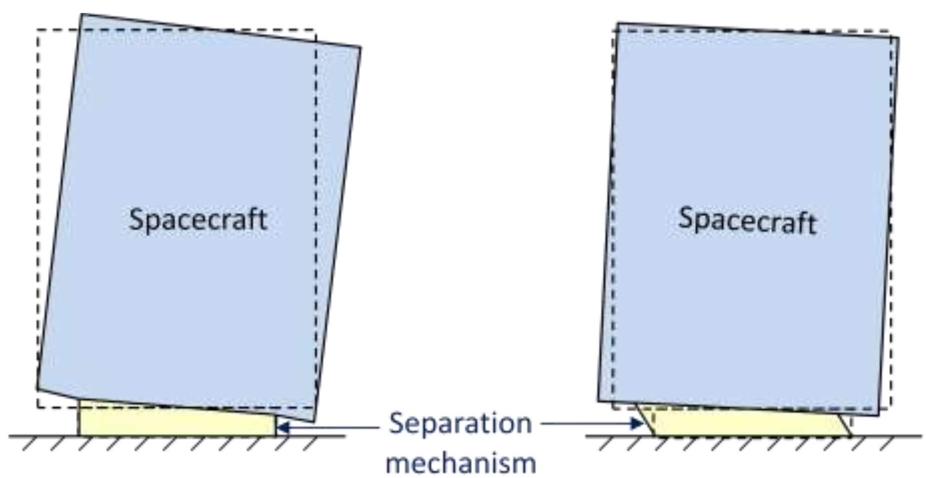
contact elsewhere on the fixture. The shape of the fixture, with center octagonal cut-out, minimizes weight while still providing near-uniform stiffness under the boss, given the bolts attaching to the shaker.

Advantages of Testing on the Separation Mechanism

When practical, the separation mechanism should be included in the random vibration test because it better represents the flight boundary conditions to ensure a more flight-like test. Inclusion of the separation mechanism allows the raised boss shown in Fig. 2-2 to be omitted from the test fixture.

During launch, a separation mechanism acts like a spring between the spacecraft and the launch vehicle. The first lateral mode is typically a rocking mode (Fig. 2-3a), with much of the strain energy in the separation mechanism. If the spacecraft's ***base plate*** (bottom plate that attaches to the separation mechanism) overhangs the bolt circle for the separation mechanism, additional strain energy for the rocking mode is in bending of the base plate. When testing without a separation mechanism, such a spacecraft still will exhibit a combined rocking-bending mode, as long as the fixture provides proper boundary conditions, as noted above, but the mode will be at higher frequency than when the separation mechanism is present.

Lateral shear flexibility of the separation mechanism introduces another mode of vibration (Fig. 2-3b) in which the bottom of the spacecraft moves laterally and the top moves very little. This mechanism-shear mode is typically well higher in frequency than the rocking frequency. When testing without the separation mechanism, this second mode is typically not present unless the spacecraft structure includes an adapter or spacer ring sandwiched between the base plate and the separation mechanism in flight configuration.



a. Rocking mode. Most of the strain energy is in the separation mechanism and in bending of the spacecraft's base plate.

b. Shear mode. Most of the strain energy is in shearing of the separation mechanism.

Fig. 2-3. Rocking and Shear Modes with Separation Mechanism.

A key advantage of including the separation mechanism in a random vibration test of a small satellite is that the shear mode provides isolation such that the spacecraft payloads and equipment see less high-frequency response to lateral excitation. Such isolation is analogous to that of a base-driven SDOF system, as shown in the transmissibility plot (Fig. 1-2). If you predict unacceptable high-frequency acceleration of spacecraft components as a result of the specified random vibration environment, investigate the effects of including the separation mechanism in the test configuration. In addition, include any other structures, such as adapter rings or spacer rings, that, in flight configuration, will be between the spacecraft and the separation mechanism or between the separation mechanism and the LV structure. Their presence during test will reduce the frequency of the shear mode and thus enhance isolation at higher frequencies.

The separation mechanism also may add beneficial damping to the test, thus reducing response of the fundamental modes of vibration.

The disadvantages of testing on the separation mechanism are that, unless a dedicated mechanism is available, it adds risk of handling damage and introduces fatigue damage to the flight mechanism. Using a stiffness simulator in place of the flight mechanism avoids these issues if such a simulator is available or can be developed. A simulator may not add beneficial damping, but it provides a more flight-like interface and can introduce flight-like isolation.

Before deciding to include the separation mechanism or a mechanism simulator, or spacer rings, however, you must ensure that the specified random vibration environment applies at the base of the separation mechanism or spacer ring as well as at the base of the satellite. During launch, the environment will differ between these locations. However, sometimes there is not enough available flight data or fidelity of predictions to distinguish between these two locations. Let the launch-vehicle provider know that you would like to include the separation mechanism in the vibration test, and ask if any modifications of the specified random vibration environment are needed.

For the sine-burst test, it often is advantageous to test the satellite without the separation mechanism even though the flight boundary conditions aren't simulated as well. This is because the fundamental frequency in any axis is higher without the separation mechanism, so there will be less dynamic gain for a given frequency of shaker input. The shaker stroke limits how low the input frequency can be. (See Part 4 for discussion.) The transmissibility function can be used to estimate dynamic gain. If you plan to do the sine-burst test without the separation mechanism, the test fixture should have the raised boss shown in Fig. 2-2 (or a separate adapter ring) to provide a flight-like interface.

Mass Simulator Design

Testing a spacecraft structure on a shaker before mounting the flight payloads, equipment, and electrical harness requires mass simulators to represent the missing items. Each major item—anything having mass greater than perhaps 1% of the total predicted mass of the integrated satellite—should be represented with a mass simulator having the same mounting interface as the missing item. Items with distributed mass, such as harness, usually are not discretely simulated; instead, their mass is distributed to other mass simulators. For example, if the predicted weight of the battery is 15 lb, we may design the battery mass simulator to weigh, say, 18 lb to represent some of the harness mass.

As noted in Part 1 of this series, we want the total mass of the test article, when including mass simulators, to be at least equal to the vehicle's maximum design mass, which includes an appropriate weight-growth allowance. We also want the test article's center of gravity (CG) and mass moments of inertia to be about the same as we calculate for the integrated vehicle when set at maximum design mass. To ensure mounting structures will be adequately tested, each discrete mass simulator also should have representative CG and MOI.

Mass simulators should be carefully designed to have the intended mass properties without introducing modes of vibration in the frequency range that is important to the test (unless the stiffness of the represented component also is simulated, as discussed below). Any such modes will combine with those of the test article to make the system modes different than those for the integrated flight vehicle. The same can be said if rigid mass simulators are used in place of actual flight components, of course, but rigid mass simulators present several advantages:

- Modal strain energy is in the structure being tested rather than in the mass simulators, thus better ensuring the structure is adequately stressed.
- Measured response acceleration of a rigid mass simulator during a random vibration test is a good indication of the environment the flight component will see during the random vibration test of the integrated satellite. The response spectrum can be used to derive appropriate environments for component-level testing.
- They allow for a simpler finite element model of the test configuration, with point masses and moments of inertia for the simulators, and make it easier to correlate the model with test results by eliminating the need to correlate the models of the mass simulators.

Figure 2-4 shows examples of designs of mass simulators to avoid.

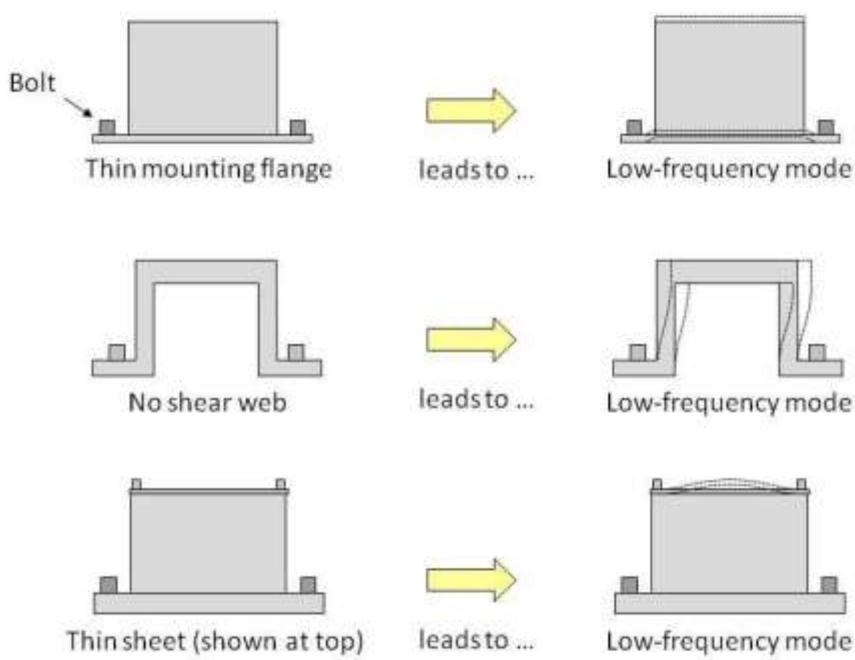


Fig. 2-4. Mass-simulator Designs to Avoid.

Use of rigid mass simulators, as advised above, makes it likely that the mounting bolts at simulator interfaces will not see flight-like load distribution and the mounting panels subsequently will not be adequately stressed. This is because the simulators are stiffer than the simulated flight equipment, and they act as stiffer structural doublers to mounting panels. Unfortunately, ensuring proper stiffness and load distribution entails the design and build of stiffness simulators, which can be labor intensive. Compromising with simulators that only approximate the proper stiffness without supporting engineering usually introduces other test errors associated with simulator-specific modes of vibration, as discussed above. Thus, improper load distribution at component interfaces as a result of over-stiff mass simulators is normally an accepted test deficiency, given that the fully integrated flight vehicle will be tested for random vibration, which should confirm structural integrity for component interfaces.

An exception to the above guidance is a simulator representing a relatively large item whose modes of vibration contribute significantly to the satellite's overall modes of vibration at frequencies that are important for meeting the test objectives. For example, consider a component or structural assembly that will not be present during the satellite's structural qualification test. The component hypothetically has a mode of vibration at 150 Hz. If the mounting structure (say, a structural panel) being qualified in the test is stressed most in flight configuration by modal response of the mounted component, using a rigid mass simulator for that component probably will not adequately stress the mounting structure during test. In this case, the component's mass simulator should be designed in a way that also simulates the component's stiffness and modes of vibration.

The decision on whether mass simulators should also simulate stiffness should be based on analysis, in most cases using a finite element model. Configure the satellite model two ways: (a) flight configuration, with mass and stiffness representations of the larger items, and (b) test configuration, with items represented with concentrated masses and moments of inertia attached to the structure with rigid elements (RBE2 elements, if using Nastran). Then calculate and compare modes of vibration, responses to the specified test environments, and load distribution or stresses within the structures being qualified. Any significant and important differences indicate that better stiffness representation is warranted.

Instrumentation and Pretest Analysis

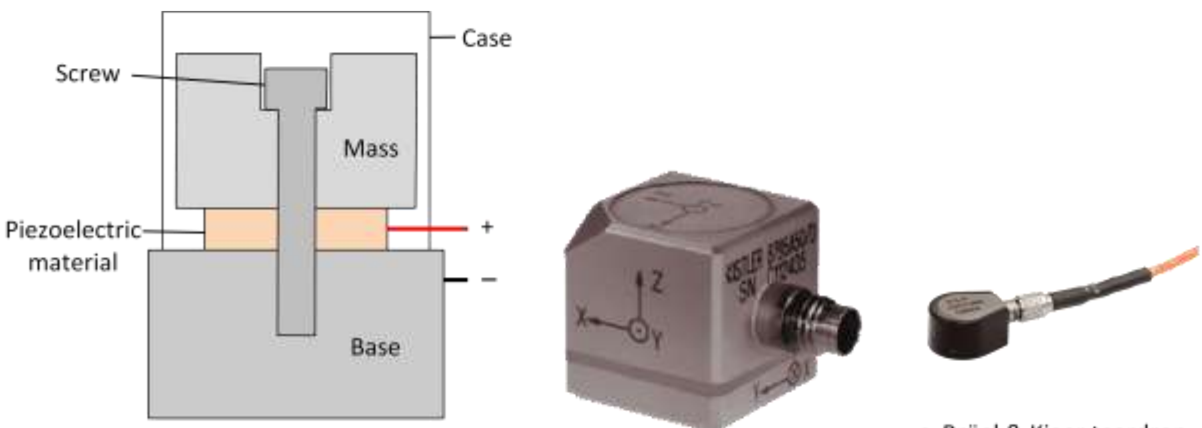
An ***accelerometer*** is an instrument that measures acceleration indirectly from deformation, which in turn relates to inertia force that resists acceleration. Piezoelectric and piezoresistive accelerometers are the most commonly used types.

Random vibration testing in the space industry traditionally goes up to 2000 Hz. For such tests, ***piezoelectric accelerometers***, which incorporate materials that produce a voltage when they deform (***piezoelectric materials***), are preferred over other types because they are more accurate for high-frequency vibration. Figure 2-5a shows a common configuration in which a small screw preloads a mass against a piezoelectric material. As the mass accelerates, the piezoelectric material deforms. Calibration of this instrument leads to accurate measurement of acceleration. A piezoelectric accelerometer can measure only the alternating acceleration, not steady-state acceleration such that associated with gravity (one-g loading) or centrifuges, and it is also not a good choice for low-frequency (less than 2 Hz) vibration.

A ***piezoresistive accelerometer*** incorporates ***piezoresistive strain gages***, which change electrical resistance in proportion to material strain. Such accelerometers can measure sustained acceleration or acceleration at extremely low frequencies, but they're not as accurate as piezoelectric accelerometers for the typical range of frequencies involved in satellite vibration testing.

An accelerometer is designed to measure acceleration in a single axis, with cross-axis acceleration having negligible effect on the measurement. A ***triaxial accelerometer*** (Fig. 2-5b) is simply three accelerometers packaged orthogonally in a compact unit. A ***teardrop accelerometer*** (Fig. 2-5c) is a single-axis accelerometer with extremely low mass.

We use accelerometers to control the test and to measure test-article response. For most tests on an electrodynamic shaker, the control system is closed loop. The shaker produces a force proportional to electrical current, and the force accelerates the mass. One or more control accelerometers mounted on the test fixture, close to the test article, measure acceleration. The measured acceleration is fed back to the control system, which then changes the current until the acceleration hits the specified level, plus or minus a specified tolerance. The average of two or more fixture-mounted accelerometers is used as the control acceleration when the test article interacts with the fixture and the shaker to produce a mode in the test frequency range. A significant difference between measurements at these accelerometer locations tells us we should use the average for controlling the test environment.



a. Common configuration of a piezoelectric accelerometer

b. Kistler triaxial accelerometer 32 grams (1.1 ounce)

c. Brüel & Kjaer teardrop accelerometer, 1 gram (0.035 ounce)

Fig. 2-5. Piezoelectric Accelerometers.

An accelerometer can be attached by a screw or a stud, but more commonly they're bonded in place, as doing so has less impact on the test article.

Where we put accelerometers on the test article depends on our objectives. In most tests, we want to understand responses that most affect the spacecraft's primary structure along with acceleration of any relatively high-mass items, such instruments, batteries, and electronics boxes. Table 2-1 provides guidance. For convenience, orient accelerometer axes to match the coordinate system used in your finite element model. With care, we should be able to align accelerometers by eye within +/-5˚, which has insignificant effect on measurement accuracy.

Put accelerometers on corners, edges, and other stiff locations that won't be affected much by local modes of vibration, such as panel bending, unless those modes are of interest. Recognize that the mass of the accelerometer may significantly affect certain modes. When concerned that this may be the case, use a low-mass teardrop accelerometer. Figure 2-6 shows an example of how local shell modes can influence response data.

Table 2-1. Guidance for Accelerometer Locations on the Test Article. Other objectives may apply as well, depending on the type of test. Start with objectives before defining instrumentation. (See Part 3 of this series for how to derive damping from sine-sweep data.)

Objective	Where to put accelerometers	Comments
Determine fundamental frequency	Any location predicted to have relatively high motion for the first mode, typically in axis with the excitation	Do pretest analysis to predict modes of vibration for the test configuration.
Determine mode shape and damping	Multiple locations, as needed to adequately define the shape	For all but the simplest modes (e.g., rocking), determining mode shape requires phasing—identification of which accelerations are positive in sign and which are negative at any point in time. Phasing requires either acceleration time histories or frequency response functions (FRFs) that include magnitude and phase. FRFs are helpful in determining damping for a given mode, as discussed in Part 3 of this series.
Identify impending structural failure	Locations and directions that have a lot of motion for the key modes of vibration, e.g., top of spacecraft and locations with high mass	There's no sure way of doing this, but, as test levels increase, you can look for abrupt changes or nonlinearities in response data, which may indicate yielding or fatigue cracking.
Identify structural failure after it has occurred (health check)	Same as above	Look for natural frequencies that have dropped. A natural frequency that continues to drop (along with response amplitude) with test duration or with cycles most often indicates either a growing fatigue crack or fasteners losing preload.
Define random vibration environments for component-level testing	If a component is represented in the spacecraft-structure test as a rigid mass simulator, put the accelerometer on the simulator, in line with the CG. If the actual component is present, put an accelerometer at the mounting base and another on the component in line with the CG.	In the spacecraft-level test (and during launch), the component will see angular acceleration as well as translational acceleration, whereas the component-level test will see translation only. Acceleration at the CG of a rigid mass simulator is best for defining the component test environment. With the actual component present, two accelerometers should be used so that response associated with component modes can be removed.

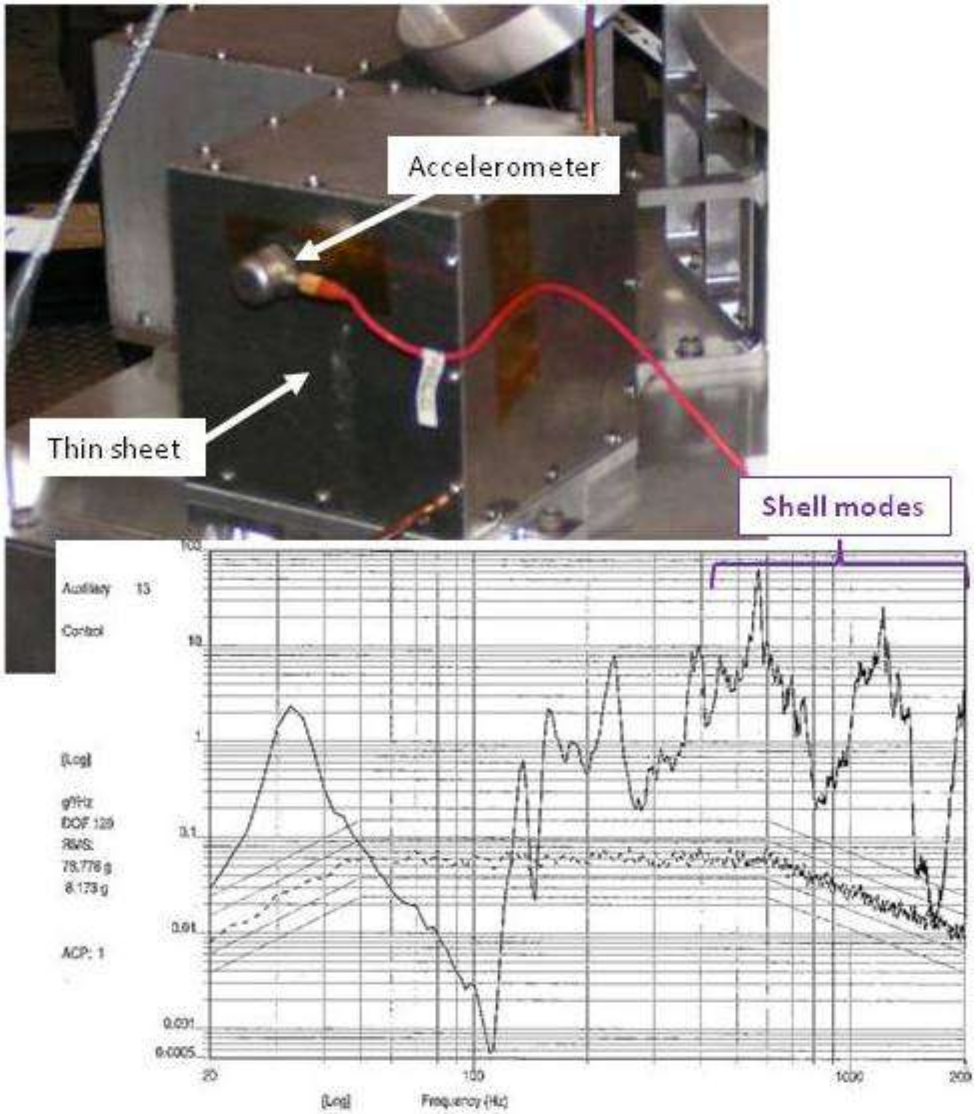


Fig. 2-6. Example of How Local Shell Modes Can Contaminate Data. During random vibration testing, the response at this accelerometer was 78.776 g-rms. Much of that response was from local shell modes that no one cared about in this test; when discounting the modes above 300 Hz, the response is only about 18 g-rms. On an item such as this, with thin sheet metal on each side, the accelerometer should be at a corner, where shell modes have less influence, or at the mounting base. If you really want to know the shell-mode response, recognize that the mass of the accelerometer may have a significant influence on the modes, as it no doubt did here. (See Part 5 of this series for discussion of random vibration testing.)

Accelerometers are the most commonly used instruments in vibration testing, but they're not the only types we can use. Strain gages and force gages also have their use.

A ***strain gage*** is a small device whose electrical resistance varies linearly with length change. (There are other types of strain gages, but the electrical-resistance gage is most commonly used because of its accuracy, versatility, low cost, and dependability; hence it's the only one we'll discuss.) When adhesively bonded to a material, a strain gage translates the material's strain into an interpretable electrical signal, using a Wheatstone bridge to measure change in resistance.

Some strain gages, referred to as ***single-leg*** or ***single-axis*** gages, measure strain in a single direction. A ***rosette*** is a combination of two to four single-leg gages combined into a single unit intended to measure strain in two to four directions. To completely understand the two-dimensional state of stress (actually strain), we need to measure strain in three directions. As long as the material is stressed in the linear-elastic range, without onset of yielding, we can readily transform strain to stress based on mechanics of materials.

We can use strain gages in a vibration test, but seldom for the purpose of determining the maximum stress—for two main reasons:

- Strain typically peaks near discontinuities, such as fastener holes, where it usually is not possible to put a strain gage.
- In ductile materials, such strain concentrations cause local yielding, such that stress is not proportional to strain, don't relate directly to failure, and cause loads to redistribute.

As a result, even if we can put the strain gage in the region of peak strain, we may not know how to interpret the measured strain regarding the objective of predicting or avoiding potential failure. Of course, there are exceptions—such as when we can put a gage at the location of peak stress in a brittle material—but we seldom use strain gages this way in a vibration test. (Actually, the above logic applies for static loads testing as well.)

Instead, the most common use of strain gages in a vibration test is to understand how load distributes in a statically indeterminate structure. In such applications, we want to keep strain gages away from discontinuities and the associated strain concentrations. Figure 2-7 shows an example of how a strain gage was used to indicate force near the mounting base of a structure during vibration testing.

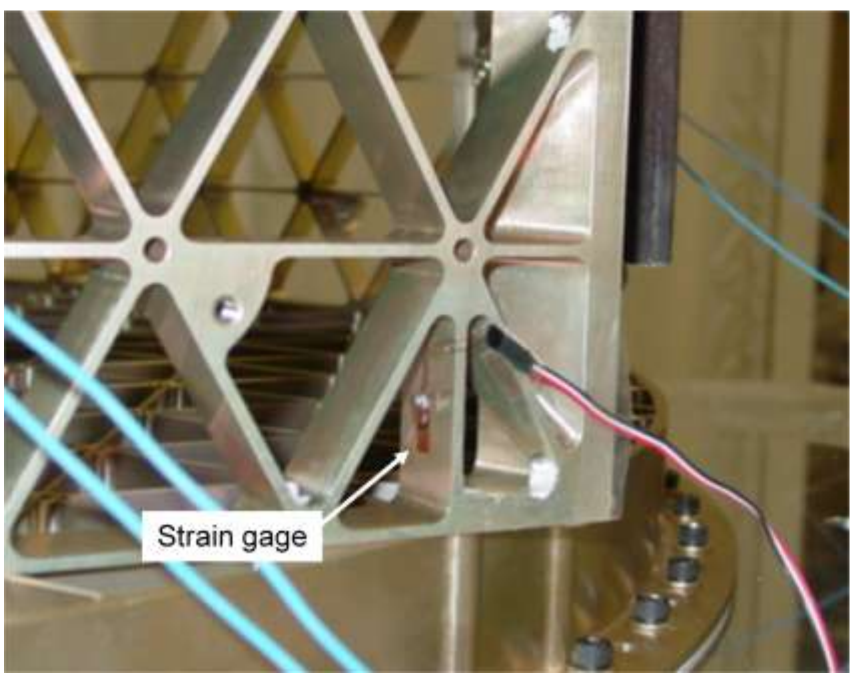


Fig. 2-7. Strain Gage Used to Indicate Load. This strain gage was calibrated with force in the axial (vertical, in the figure) direction by applying low-frequency sinusoidal vibration at known acceleration. The gage was used to indicate force at the mounting interface during random vibration testing.

A ***force gage***, also referred to as a ***load cell***, is an instrument calibrated to measure force, typically by use of one or more strain gages. In vibration testing, force gages are most commonly used to measure the total force or moment at the base of the test article. (Individual force gages can be electrically bridged to measure total force or moment.) Such a measurement is used for force limiting to avoid excessive loading, as discussed in Part 6 of this series.

Part 3 of this series addresses low-level sine-sweep testing.