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.

This series includes seven parts, each of which was released when completed:

1. Introduction to Vibration Testing (initial release April 11, 2014; Rev. B July 19, 2017)
2. Test Configuration, Fixtures, and Instrumentation (initial release April 11, 2014; Rev. A July 19, 2017)
3. Low-level Sine-Sweep Testing (initial release May 13, 2015; Rev. A July 19, 2017)
4. Sine-Burst Testing (initial release April 28, 2017; Rev. B January 19, 2018)
5. Random Vibration Testing (initial release April 7, 2016; Rev. B July 24, 2017)
6. Notching and Force Limiting (initial release May 13, 2015; Rev. A July 19, 2017)
7. Designing a Small Satellite to Pass the Vibration Test (initial January 24, 2020; Rev. B June 24, 2020)

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

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

Instar Engineering and Consulting, Inc., provides consulting services in mechanical systems engineering, concept development, requirements development, verification planning, structural design and analysis, dynamics, fatigue and fracture control, thermal analysis, and test engineering.

Part 7: Designing a Small Satellite to Pass the Vibration Test

Tom Sarafin, Poti Doukas, Mike Browning, and Lenny Demchak
Instar Engineering and Consulting, Inc.
Contact: tom.sarafin@instarengineering.com

Rev. B, June 24, 2020

With an understanding of vibration testing from Parts 1 – 6 of this paper series—especially Part 6 on notching and force limiting—we are partially armed to design a small satellite to withstand its vibration tests. For the best chances of a successful test, however, we need to expand that knowledge beyond what these papers have thus far addressed. This paper explores numerous considerations and strategies for increasing the chances of a successful test.

Table of Contents

Change Log

Revision	Date	Changes
A	3/24/2020	Corrected typo in Fig. 7-11
B	6/24/2020	Corrected several errors in section references Added this change log

7.1 Introduction

In a basic sense, a SmallSat consists of structure and components, with the latter being the items the structure supports. Components, a.k.a. units, consist of payloads, such as instruments and experiments, and equipment, such as avionics, tanks, batteries, and solar cells. Most components, particularly those not requiring a field of view to function, are enclosed inside a ***body structure*** (a.k.a. ***bus structure***), most commonly a box-like structure comprised of base plate (which is at or near the interface with the spacecraft separation mechanism), top panel, and side panels. The body structure is considered ***primary structure***, which is the structure forming the main load paths to the separation mechanism (SM), analogous to a tree trunk carrying loads to the roots and ground. Any adapter that may be placed between the SmallSat's body structure and the SM also is considered primary structure.

The cross section of the body structure, when looking normal to the interface with the SM, can be square, rectangular, hexagonal, or octagonal. Other cross sections are certainly possible as well but are not as common. Figure 7-1 shows a typical body structure for a SmallSat. The cross section referred to above is in the X-Y plane in this figure, i.e., the shape you see when looking up the Z axis.

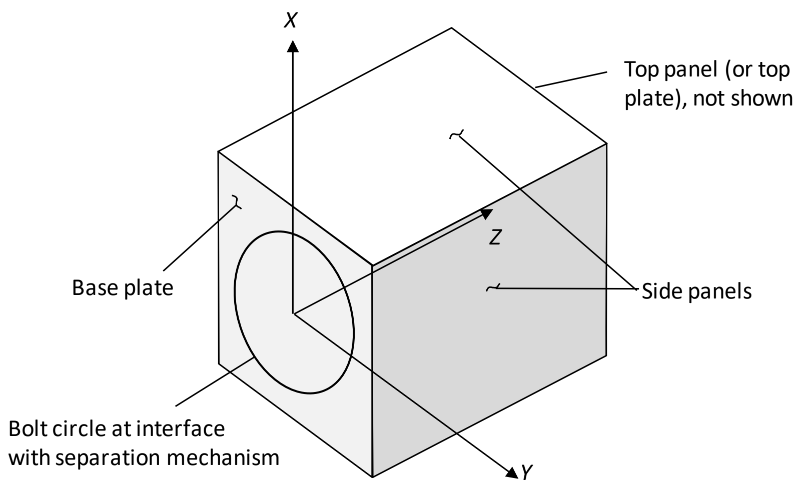

Fig. 7-1. Body Structure with a Square Cross Section. Hypothetical coordinate system shown for reference.

Components packaged inside the primary structure can mount to the side and top panels, the base plate, or internal decks. Such decks are normally in the X-Y plane of the coordinate system shown in Fig. 7-1. Equipment is sometimes stacked, with some components supporting others; the stack is most commonly supported by the base plate. As an example, Fig. 7-2 shows the configuration of the FalconSat-5 satellite, which was ESPA[1] class, just under 180 kg. The FalconSat series is developed at the United States Air Force Academy (USAFA), funded by the Air Force Research Lab (AFRL).

For a SmallSat that will be tested with sine bursts and random vibration, the primary structure will be stressed more by sine-burst testing, whereas everything else will be stressed more by random vibration testing. At least, this is the case when designing an effective sine burst test (Part 4 of this paper series) and appropriately notching or force limiting the random vibration test (Part 6).

[1] EELV Secondary Payload Adapter, where EELV is Evolved Expendable Launch Vehicle.

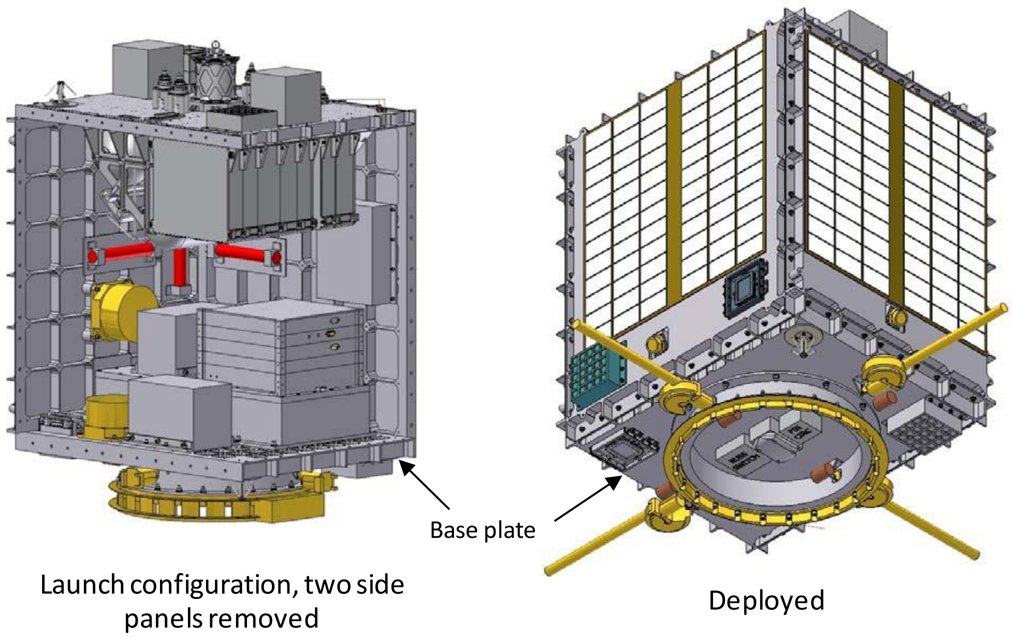

Fig. 7-2. FalconSat-5 Configuration. (From Ref. 1)

Following are some common failures encountered during vibration testing:

- Material rupture in structural parts, such as component housings and the SV primary structure
- Material rupture in small items such as electronics and solder joints
- Backing out of threaded fasteners
- Detrimental permanent deformation (e.g., misalignment that impacts performance) from materials yielding or joints slipping
- Unseating of electrical connectors

If a brittle material ruptures in a random vibration test, it is most likely a strength failure, i.e., the peak stress exceeded the material's strength. However, most parts in an integrated satellite are made of ductile materials, even when the primary structure is made of composites. By far, most ruptures of ductile materials occurring during random vibration testing are fatigue failures. In the space industry, the military standard levels and duration for qualification testing (See Table 5-1 in Part 5 of this paper series) were derived to achieve four times the maximum expected fatigue damage for flight hardware, where the factor of four is to account for variation in fatigue life from one build to the next.

7.2 Programmatic Considerations

This section explores strategies for improving the likelihood of passing the vibration test that are usually outside the control of the design team. Program management must decide whether these strategies make sense for the program and then implement them if they do. These strategies can lead not only to reduced risk but to reduced cost as well.

7.2.1 Avoiding Weight Criticality

Most SmallSat programs are constrained by budget more than by anything else (although schedule is often a driver as well). Too often we are tempted to pack too much mission or payload—instruments, antennas, experiments—into the SmallSat for the available funds. On the other hand, in some case more funding becomes available if you sign up to flying more payload. But, in such a case, there may be hidden costs associated with that decision. More payload can make the SmallSat weight critical, which means it's challenging to keep weight from exceeding the allowable weight. The result is higher cost of structural engineering and higher risk associated with a marginal structure and marginal components. When weight is not critical, we can make structural designs robust and select more robust components as well.

It's a myth that all flight hardware is weight critical. Sure, if you're designing a launch vehicle (LV), reducing weight brings in more revenue because it allows you to fly more LV payloads or more massive payloads. But, if you're designing a SmallSat, once you've selected a size and a weight class (e.g., ESPA class—up to 400 lb or 180 Kg)—you have to stay below the upper limit, but there's usually no cost savings associated with coming in below that. If you aim initially for, say, 350 lb, including allowance for typical growth, when allocated 400 lb, you can use the extra 50 lb margin to reduce both the development cost and the risk of test failure.

It's important to understand that random vibration is not well predicted, and neither are its effects on hardware. In a random vibration test, at a given frequency, NASA-STD-7001B (Ref. 2) requires the input acceleration PSD to be controlled only to within +/-3 dB of the specified level at any given frequency. Controlling more accurately than that will introduce additional fatigue damage because it takes time for the closed-loop control system to make that happen. Response depends heavily on the test article's modes of vibration—natural frequencies, mode shapes, and damping—which typically are highly uncertain, despite use of finite element models. Even if the input environment is perfectly controlled and the modes of vibration are accurately predicted, the peak response to random vibration is itself random.

Given all the uncertainty, structural designs and selected components should not be marginal for the random vibration test environment. They should be robust.

Structures and components can't be robust if weight is critical—at least not when staying within the constraints that are typical of SmallSat programs. When weight is critical …

- more-detailed analysis—and more-frequent testing to substantiate the analysis—are needed to ensure robustness, and you probably won't have the budget or schedule for these activities.
- you may not be able design the structure to accommodate a total mass of the SmallSat that's greater than predicted in order to protect against the risk of mass growth—the common situation of a spacecraft's mass becoming greater than anticipated because of unanticipated contributors.
- you may have to select components that weigh less and are less robust or more costly than other available options.
- for components that historically are prone to failure in space missions, including spares for redundancy won' be feasible.

Many of the strategies explored in Sec. 7.3 are feasible only when weight is not critical.

7.2.2 Building and Testing an Engineering Model of the Spacecraft Structure

With the advancement of computer technology, the trend in the space industry has been to reduce the extent of structural and environmental testing, relying on analysis instead to provide the desired confidence. This management philosophy is based on the perception that testing is expensive, and that building hardware dedicated for testing is even more so.

Our experience says otherwise, at least in many cases: The extra structural and mechanical analyses deemed necessary by the design team when there is no test substantiation prior to system (SmallSat)-level testing can be more expensive and time consuming than building and testing dedicated hardware. When an engineer knows the first test will be of flight hardware, he or she naturally spends more time trying to ensure the hardware will pass the test: checking models and analyses, pouring over details, and assessing sensitivity to uncertainties.

If your SmallSat's configuration and structural design are new—or at least new to your design team—strongly consider providing up-front budget and schedule for building and testing an engineering model before committing to the final flight design. Make it known to your team and any stakeholders that test failure prior to design release is okay; the objectives are to build understanding, learn how to improve the design, anchor finite element models, and get early indication of component environments. If failure were to occur, however, you probably would want to retest an engineering model of improved design.

This approach was used in the early years of the FalconSat program at the U.S. Air Force Academy—and, per our understanding, is still used on that program when schedule permits. The cadets build an engineering model of the primary structure, fabricate and install mass simulators representing SmallSat components, and then do a vibration test on a shaker to qualification levels and durations. In exchange, structural analysis is minimal, as testing of both the engineering model and the eventual flight vehicle becomes the main method of structural verification.

When this approach was adopted for FalconSat-2 in 2001 and 2002, the engineering model was intentionally designed to be robust, with little chance of test failure; after the test, the flight design was made more efficient (but still robust) for weight savings and other reasons. FalconSat-3 was larger but used the same structural concept. The structural design evolved with each subsequent FalconSat, leveraging test-qualified aspects of the design in order to reduce cost and risk. The net result of this building-blocks process is that the FalconSat program seldom if ever has structural failures during vibration testing, and structural analysis remains minimal.

Use of engineering models can be expanded to include flight-like representations of SmallSat components. The higher the fidelity of the engineering model, the more useful the test data will be.

On the other end of the spectrum regarding fidelity, before building a structural engineering model, identify any materials or joints in the design for which you don't have dependable allowable stresses or methods of analysis. Either replace those materials or joints with ones for which such data and methods are available, or plan to spend budget and time testing specimens in the lab before progressing too far with the design.

7.2.3 Qualification and Acceptance Testing of Components Prior to Integration

A basic principle of risk reduction is the use of a "building blocks" approach to verification: Test at the lowest level of assembly possible rather than deferring the risk of deficient hardware to a higher level of assembly. This principle certainly applies to spacecraft components. Selected components should be of designs that have been qualified by test to environments that are suitably higher than acceptance test levels. And each flight component should be tested prior to spacecraft integration, to either acceptance levels or protoflight levels, as discussed below. See Part 5 of this paper series for a discussion of government standards associated with levels and durations for qualification, protoflight, and acceptance tests.

Vibration testing of each component prior to spacecraft integration makes the risk of failure during spacecraft-level vibration testing negligible—but only if the component test levels are at least as high as the levels the component will see during the SmallSat test. If the spacecraft will be tested to protoflight levels, 3 dB above acceptance levels, the components should be tested to protoflight levels separately beforehand. A common mistake in the SmallSat industry is to use the GEVS (Ref. 3) 14.1 g_{rms} environment for qualification random vibration testing. These levels are too low for SmallSat components; see discussion in Part 5 of this paper series.

7.3 Configuration Development and Structural Design

The time to begin thinking about the launch environment and the test program is the time we begin to develop the spacecraft configuration. A good configuration, including efficient structural architecture, makes it much easier for the SmallSat to withstand the vibration test.

Following are some basic strategies related to launch loading and vibration testing for configuration development and structural design:

- Make the structural design robust. Minimizing structural margins of safety may be a sound strategy for designing weight-critical flight structures, but most SmallSats are not weight critical, and the effects of random vibration environments are not well predicted. (See Sec. 7.2.1.)
- In each axis, with the SmallSat constrained at the separation interface, aim for a ***fundamental frequency*** (lowest natural frequency involving much of the mass) of at least 50 Hz, and preferably higher to account for uncertainty in the finite element model. Doing so will reduce the chances that the SmallSat's modes of vibration couple with those of the launch vehicle (LV).
- Put the heaviest items aft, near the separation interface. (See Sec. 7.3.4.)
- Make ***load paths*** (paths through which load travels within structures) direct. (See Sec. 7.3.1.)
- Keep natural frequencies of other modes of vibration well separated from the SmallSat's fundamental frequency for each test axis. Doing so will avoid modal coupling and high loads during test. (See Secs. 7.3.2 through 7.3.4.)
- Design good joints. Most failures occur at joints. (See Sec. 7.3.6.)
- Design the structure for appropriate loads. (See Sec. 7.3.7.)

7.3.1 Making Load Paths Direct

A good load path is a direct load path; the best is in a straight line. Indirect load paths result in higher stresses, lower stiffness, and lower natural frequencies. Figure 7-3 shows the concept of direct and indirect load paths.

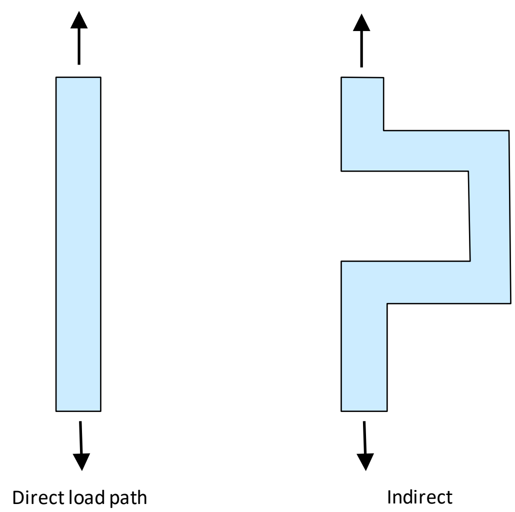

Fig. 7-3. Examples of Direct and Indirect Load Paths.

As a practical example, consider the interface between the SmallSat and its separation mechanism (SM). For the most direct load paths, we want to match the spacecraft body dimensions to those of the SM. The spacecraft body should overhang the SM only a small amount, as shown in Fig. 7-4. When the SmallSat body structure greatly overhangs the separation system, there is more bending deformation and stress in the base plate, and it becomes impossible to keep the fundamental frequency of the cantilevered SmallSat as high as desired. (See Sec. 7.3.5 for a discussion of how the primary structure carries loads.) Likewise, an under-size SM introduces a soft spring in the system that reduces the fundamental frequencies of the cantilevered SmallSat even further.

If the needed dimensions cause the SmallSat to greatly overhang the SM, as shown at left in Fig. 7-4, select a larger-diameter SM; or, if the SM diameter is specified, reduce the lateral dimensions of the SmallSat structure.

Of course, the size of a SmallSat body is normally dictated by the number and sizes of the SmallSat's components. Packaging density for items housed within the body of a satellite typically ranges from 15 to 25 lb/ft^3 (240 to 400 kg/m^3) (Ref. 4, Sec. 14.4). See Sec. 7.3.4, below, for considerations in selecting component locations.

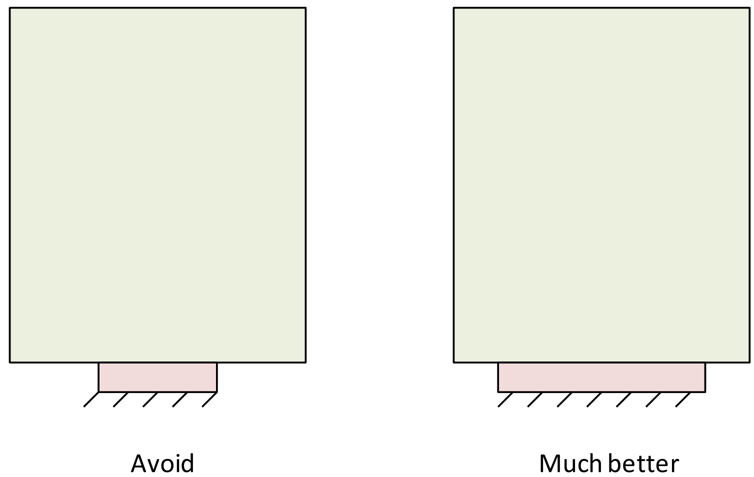

Fig. 7-4. Matching the Dimensions of the Separation Mechanism to the Dimensions of the Body Structure.

7.3.2 Separating Natural Frequencies

Try to keep the frequencies of local modes of vibration, such as for components and panels, at least 50% higher in frequency than the fundamental frequency in each axis. Because you won't be able to predict these natural frequencies exactly, we suggest aiming for a separation of two to one, i.e., predicted local frequencies at least twice as high as the predicted fundamental frequency.

Separating modal frequencies ensures the modes won't couple, which can lead to high loads, especially during random vibration testing. The more separation from the fundamental frequency, the more the local modes will be isolated and the lower the response will be during a base-driven vibration test.

Following is a case history of test failure caused by a component's fundamental frequency coinciding with the SmallSat's fundamental frequency.

Case History: FalconSat-2 Antenna

FalconSat-2 (FS2) was a SmallSat developed at the US Air Force Academy in the early 2000s. The FS2 body structure was cubic, approximately 12" on a side, and the integrated satellite weighed about 43 lb (20 kg in mass). In the acceptance random vibration test, lateral axis, the S-band antenna suffered a fatigue crack. Figure 7-5 shows FS2 and the antenna.

The fundamental lateral mode for FS2 was a rocking mode, with a frequency of 148 Hz. Unfortunately, the cantilevered fundamental bending frequency of the S-band antenna was nearly the same—a fact not recognized prior to the test. The test environment was notched at 148 Hz to ensure the base force and moment did not exceed the maximum predicted launch levels. During the test, the acceleration of the top panel was about 20 g_{rms} (Fig. 7-6), and the accelerometer on the shaft of the S-band antenna (shown in Fig. 7-5) measured 83.1 g_{rms} (Fig. 7-7). The tip of the antenna no doubt saw much greater acceleration than that, probably well higher than 200 g_{rms}. (We couldn't put an accelerometer there without its mass significantly affecting the antenna's bending frequency.) The solder joint between nested tubes, just above the accelerometer on the shaft, failed in fatigue.

After this test, the S-band antenna was redesigned to have a fundamental bending frequency of about 290 Hz. With nearly a two-to-one separation from the spacecraft's fundamental frequency, the redesigned antenna passed its vibration test with ease, with much lower response acceleration and lower stress.

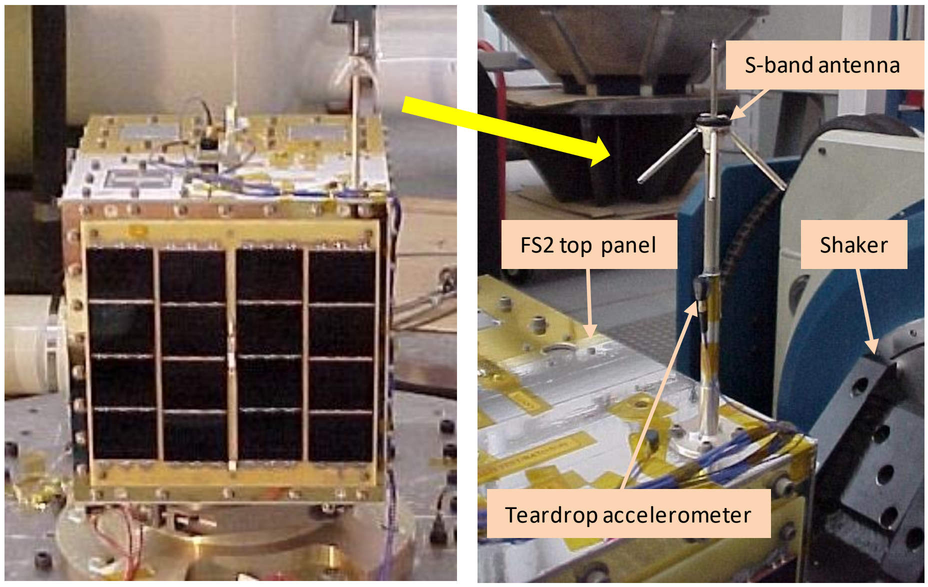

Fig. 7-5. FalconSat-2 and Its S-Band Antenna.

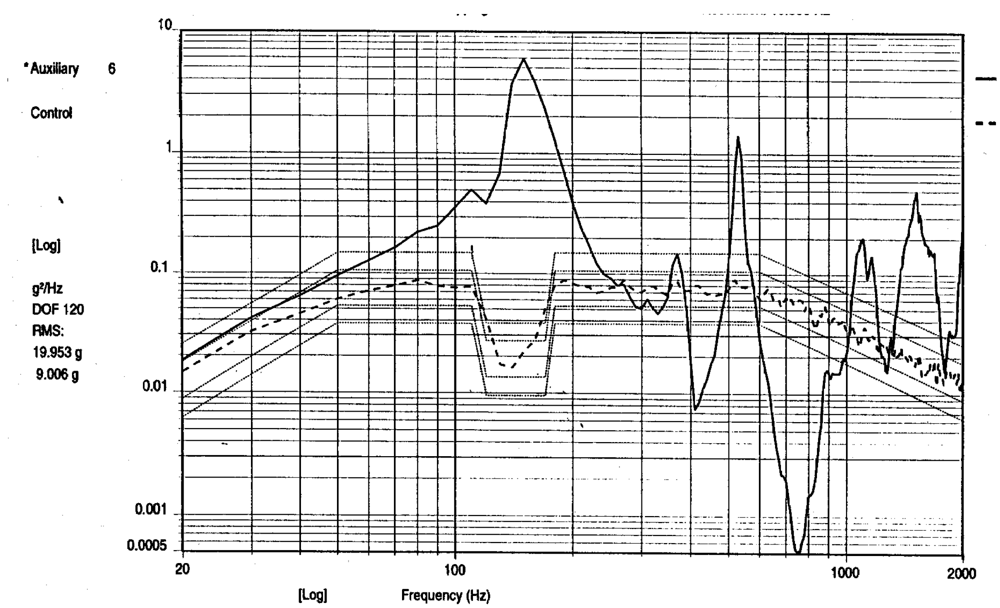

Fig. 7-6. Measured Acceleration at the Top of FalconSat-2. Even with the notch shown in the input environment, response of FS2's rocking mode led to 20 g_{rms} at the top panel.

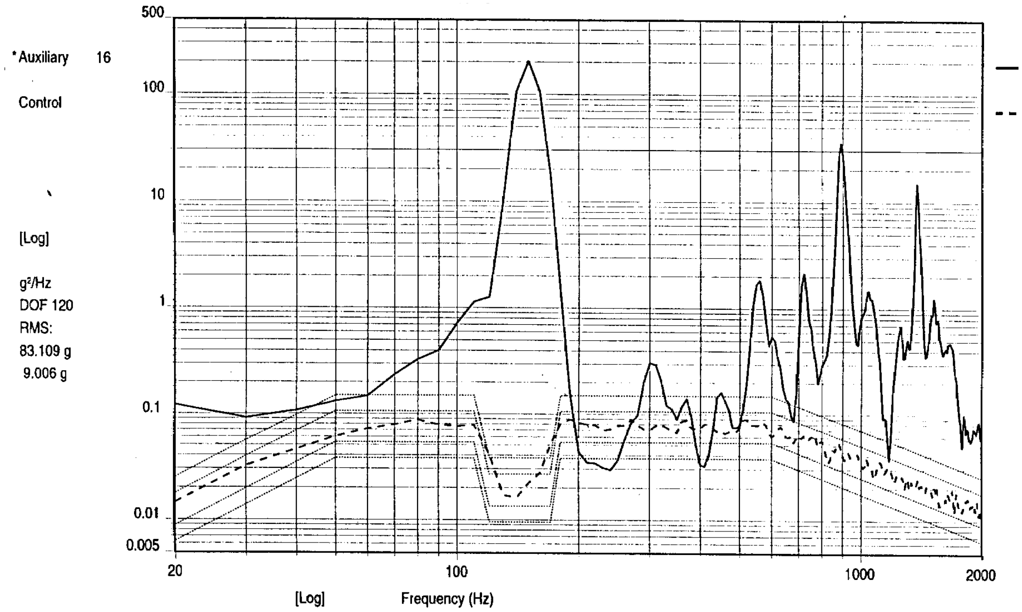

Fig. 7-7. Measured Acceleration on the Antenna Shaft. The 20 g_{rms} response at the top of FS2 was the input acceleration for the antenna, tuned to the antenna's fundamental (bending) frequency, resulting in 83 g_{rms} response measured part-way up the antenna's shaft. Acceleration at the tip of the antenna was estimated to be over 200 g_{rms}, or 600 g at 3 sigma.

Figure 7-8 shows a common situation: The SmallSat's fundamental axial mode involves a lot of in-phase bending of the top panel. By "in phase", we mean that the top panel is bending upwards while the rest of the SmallSat mass is also moving upwards. This is often the case when a lot of mass is mounted on the top panel. The same can occur when a lot of mass is mounted on an internal deck that is parallel to the plane of the SmallSat's mounting (and separation) interface (X-Y plane, using the coordinate system shown in Fig. 7-8).[2]

For a mode such as the one shown in Fig. 7-8, we'll most likely need to notch or force limit the random vibration test (and the sine vibe test, when planned) in order to avoid excessive, non-flightlike loading. As discussed in Part 6 of this paper series, such notching should be based on force-limiting principles, regardless of whether the test is manually notched, response limited, or actually force limited. Because force limiting relates directly to limiting the base force (force at the separation interface) to a technically justified level, we still can expect whatever is mounted on the center of the top panel to see high loads.

We can avoid this situation by making the fundamental frequency of the mass-loaded top panel well higher than the SmallSat's fundamental axial frequency. If we can do this, the top panel will act nearly as a rigid body with the rest of the SmallSat for the SmallSat's first mode, and, for the mode in which the top panel or deck bends, most of the rest of the SmallSat mass will move in the opposite direction. In other

[2] It's more likely that modes of vibration for panels and decks in the X-Y plane, using the coordinate system shown in Fig. 7-8, will couple with the SmallSat's Z-axis fundamental mode than it is for modes of panels in other planes to couple with the SmallSat's fundamental lateral modes. This is because the fundamental lateral frequencies are normally much lower than the fundamental axial (Z-axis) frequency.

words, the top panel will have vibration isolation. The result is much lower loads for the components mounted to the top panel.

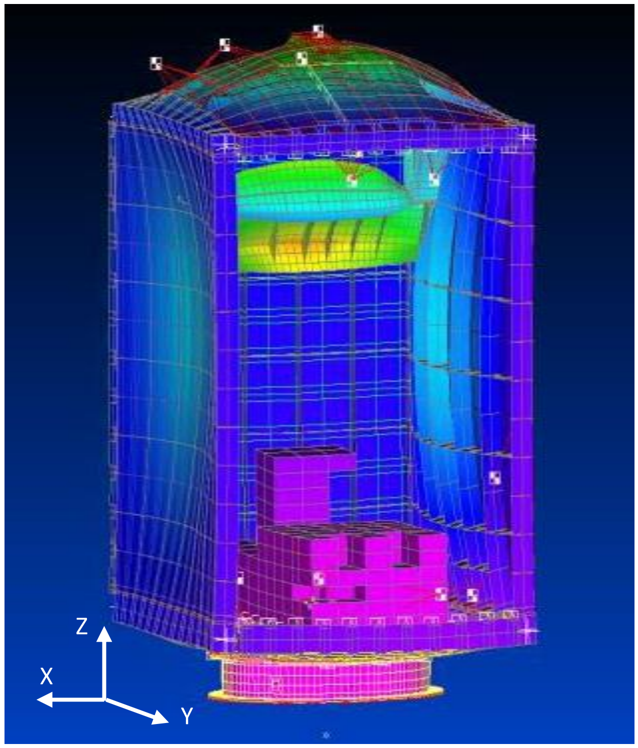

Fig. 7-8. Example of an Axial (Normal to Separation Interface) Mode Shape.

The concept of vibration isolation is shown for a base-driven mass on a spring in the transmissibility plot (Fig. 1-2 in Part 1 of this paper series). The situation here—isolating a panel's first bending mode from the SmallSat's fundamental mode—is somewhat more complex, as there are multiple modes involved. Think of the effective "spring" associated with the fundamental mode acting as the isolator for a panel's first bending mode that is at considerably higher frequency. Because of the isolation, the panel's first bending mode will not respond as much to base acceleration during launch or during the SmallSat's vibration test.

The following strategies may make it practical to isolate the first bending mode of a mass-loaded panel or deck from the SmallSat's fundamental axial mode:

- Mounting less mass on the top panel or deck; putting more components on the base plate or spreading components more evenly between available mounting panels.
- Stiffening the top panel or deck. (See Sec. 7.3.3.)
- Mounting components closer to the edges of the top panel or deck rather than in the center, and making the edges act more "fixed" (constrained against rotation) than "pinned" (free to rotate). (See Sec. 7.3.4.)

We can stiffen mounting panels most efficiently with the forms of construction discussed in the following subsection.

7.3.3 Panels Used To Mount Components

Spacecraft panels used to mount components should be made out of either sandwich construction or a machined plate with ribs in a pattern such as isogrid (equilateral triangles) or orthogrid (rectangles) (Fig. 7-9). This includes the base plate, the side panels, the top plate, and any internal component-mounting decks. Don't mount components on unstiffened (nonreinforced) sheet metal. Doing so will cause multiple low-frequency modes and potential problems, including fatigue failure of the sheet metal from bending stress.

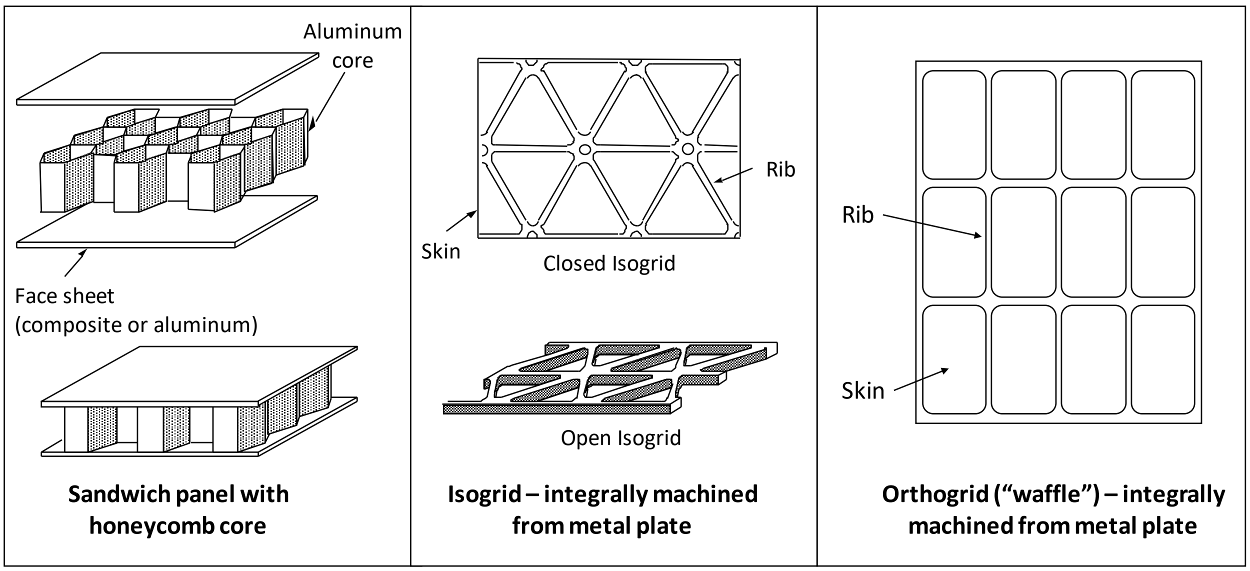

Fig. 7-9. Common Forms of Construction for Structural Panels. (From Instar's course "Space Mission Structures, from Concept to Launch" (SMS))

As discussed in section 7.3.2, we want to separate modal frequencies, i.e., keep component and panel frequencies well above the fundamental frequency of the spacecraft. Using stiff but lightweight panels such as sandwich construction or integrally machined isogrid allows us to do that.

When weight is not critical, machined panels are preferable over sandwich panels for two main reasons:

1. The total cost associated with sandwich panels is usually higher. To attach to sandwich panels, we need to install metal inserts that are held in place with a potting material. To ensure dependable pull-out strength for these inserts, we'll need a development program that includes pull testing of specimens, and we'll also want to proof test each flight insert to protect against potential low strength arising from processing variables and poor workmanship.

2. Sandwich panels have more potential failure modes than panels machined out of aluminum alloy. More things can go wrong. Drop a wrench on a machined panel and you'll normally dent it, but the panel probably will still do its job. Drop a wrench on a sandwich panel and the damage is

more severe, including potential debonding—and potential delamination when using composite face sheets.

Still, a sandwich panel offers an advantage over machined panels: We can install an insert at nearly any location at the last minute to accommodate a needed interface. A machined panel, with fixed locations for inserts, requires use of a structural adapter (extra machined part) to accommodate a late change in interface bolt pattern.

7.3.4 Selecting Component Locations and Mounting Schemes

The decision on where to put components on or within a SmallSat is made early in the design process. We can avoid problems during SmallSat vibration testing and during launch if we select locations wisely.

A simple rule of thumb is to put heavy items aft, on or near the base plate and the interface with the separation mechanism. A liquid-propellant tank is a good example. Moving heavy components aft lowers the SmallSat's CG, increases the fundamental lateral frequency, and reduces the base moment under lateral loading.

Of course, other considerations may have higher priority than the desire to keep heavy items aft. For example, depending on the mission, the SmallSat's payload (typically heavy) may need to be on the top panel, farthest from the base plate, in order to keep it away from the source of separation shock or to provide the needed field of view.

Another good practice relates to the goal of separating frequencies, which is discussed above in Sec. 7.3.2: For components that will be mounted on panels, avoid putting them in the middle of the panel, farthest from the panel's supported edges. Putting components closer to edges and corners makes it easier to keep the loaded panel's bending frequencies well higher than the SmallSat's fundamental frequency. However, for this to be the case, the panel edges must be supported with other panels that are themselves stiff in bending, with the joints being able to transfer moments. If the mounting panel has pinned edge fixity, mounting components near the edges may lead to lower natural frequencies and higher loads than mounting components in the center of the panel.

As an example of how component mounting schemes can cause problems in vibration testing, consider the configuration shown in Fig. 7-10. The reaction wheel is mounted off a cantilevered bracket, which is mounted to a honeycomb sandwich deck. The deck would have been quite stiff in bending if it didn't have the large circular hole in the center, for the purpose of allowing another portion of the spacecraft to pass through.

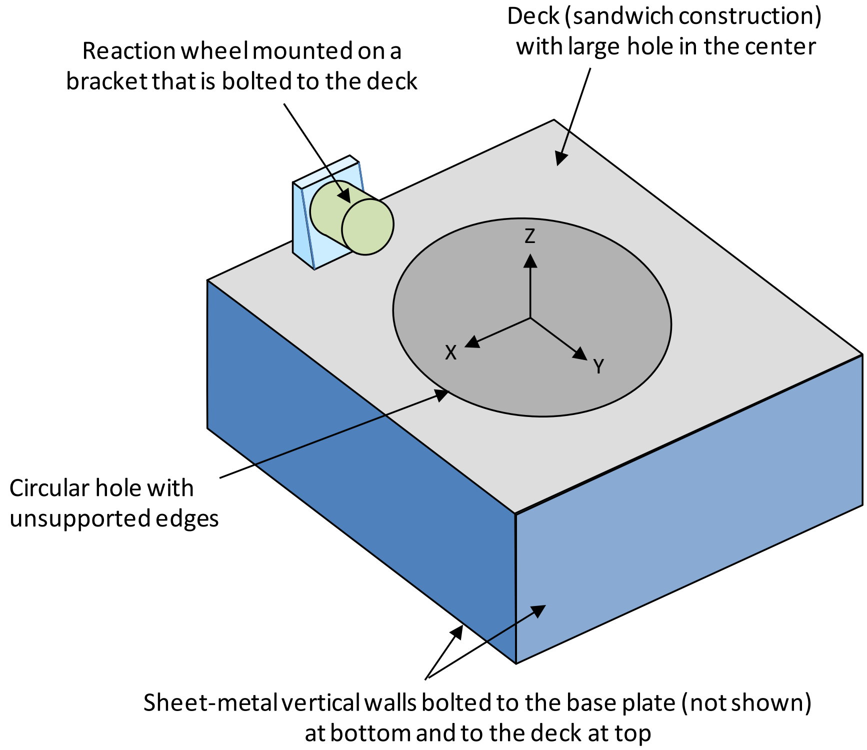

Fig. 7-10. Approach to Mounting a Reaction Wheel that Led to High Responses during the Random Vibration Test.

Figure 7-11 shows a side view of the reaction wheel on the deck, along with the approximate shape of the problem mode: rocking of the wheel-bracket assembly on a rotating deck. The outer edges of the deck were essentially simply supported, with virtually no moment fixity provided by the thin sheet-metal wall. This mode was below 200 Hz, with very low damping, and led to high response acceleration in the random vibration tests in the axial (Z) and lateral (Y) axes. To compound the problem, the reaction wheel had been designed and qualification tested to the GEVS (Ref. 3) 14.1 g_{rms} environment for components weighing less than 50 lb. (This reaction wheel weighed about 2 lb.) The response levels during the SmallSat test were well higher than the GEVS environment at the frequency of the wheel assembly's rocking mode. The reaction wheel could not sustain such loading and still meet requirements for smooth operation.[3] Resolving this problem severely impacted cost and schedule.

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[3] ***Brinelling*** (local yielding or denting) of bearing races is a common problem with reaction wheels. Operation of reaction wheels with brinelled bearings causes on-orbit vibration (***jitter***), which can adversely affect mission performance.

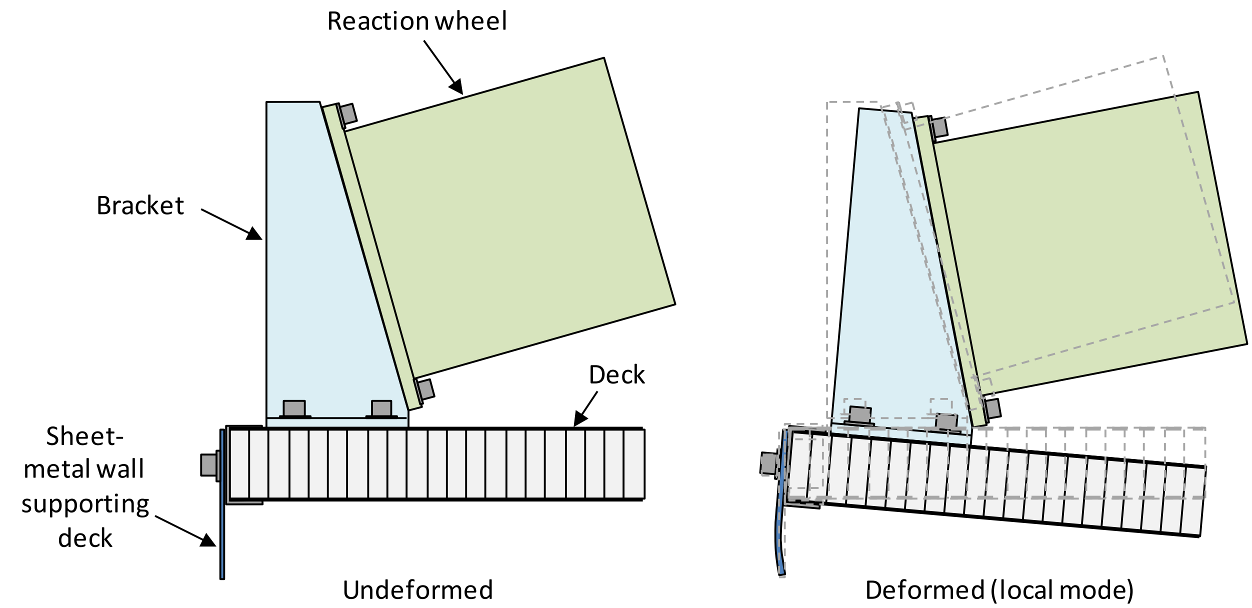
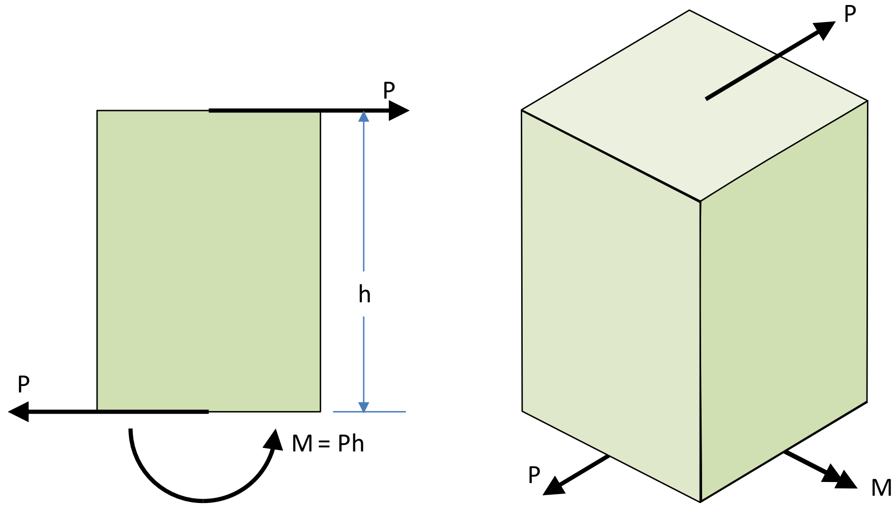


Fig. 7-11. Problem Mode for the Reaction Wheel Mounting Approach Shown in Fig. 7-10. The large cutout in the deck combined with sheet-metal walls supporting the deck edges led to the relatively low-frequency (< 200 Hz) rocking mode shown and high response acceleration during the SmallSat random vibration test.

7.3.5 Understanding How the Primary Structure Carries Loads

Most SmallSat primary structures are stressed more by lateral (X and Y in Fig. 7-1) loading than by axial (Z) loading. When designing such a structure, it's important to understand how it carries lateral loads. Inertia loading during launch is a distributed load, but it's easier to understand the load paths if we start with a concentrated lateral load, *P*, applied to the top panel, reacted at the interface between the base plate and the separation mechanism with load *P* and moment *M*. (*P* and *M* are shown as concentrated loads for simplicity.) Figure 7-12 shows simple free-body diagrams for this situation.

Figure 7-13 shows an exploded view for lateral loading, with the primary role of each panel illustrated by ***running loads*** (force per unit length) normal to the panel edge, ρ, and along the edge in shear, ν. Looking only at the top panel, the applied load *P* is reacted by equal shear running loads ν. Side panels A and B carry this lateral load down to the base plate; if we make the simplifying assumption that these panels are in pure shear, the running shear load reacted by panels C and D is equal to ν. The running shear loads introduced into panels C and D from panels A and B cause an increasing tensile load (panel C) and an increasing compressive load (D), which, with separation distance *b*, react the linearly increasing moment on the assembled structure caused by applied force *P* on the top panel. In actuality, side panels A and B will carry some of the moment as well, but their main roles, as shown in the figure, are to carry the lateral load and to shear axial loads into panels C and D so that C and D can react the moment.



a. 2-dimensional view b. 3-dimensional view

Fig. 7-12. Free-Body Diagrams for a Cantilevered Box Structure under Idealized Lateral Loading.

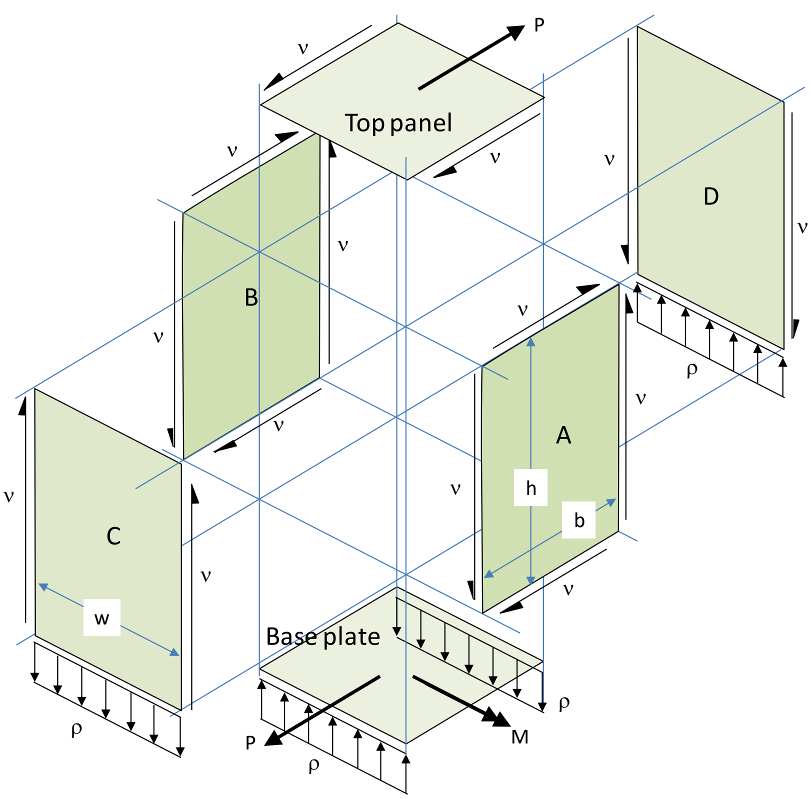


Fig. 7-13. Idealized Loads Carried by Each Panel for a Box Structure under Lateral Loading. The reactions, P and M, are the total force and moment at the base plate.

Based on the assumptions that the lateral load is applied solely at the top panel, as shown in Figs. 7-12 and 7-13, and that panels A and B are in pure shear, as described above, we can solve for the running loads ν and ρ in units of force per unit length, e.g., lb/in:

$$\nu = \frac{P}{2b} \tag{7.1}$$

$$\rho = \frac{2\nu h}{w} = \frac{Ph}{bw} \tag{7.2}$$

Section 7.3.6.1 explains how to use an understanding of how the primary structure carries loads to ensure the panel-to-panel attachments are structurally sound.

Example 7-1. Estimating Running Loads on Panel Edges

Consider the 120 kg spacecraft shown in Fig. 7-14, under a design load of 15 g uniform (quasi-static) lateral acceleration. The spacecraft has a square cross section, with construction of the type shown in Fig. 7-13 ($b = w = 40$ cm). Let's say we want to estimate the running loads ν and ρ, as defined in Fig. 7-13.

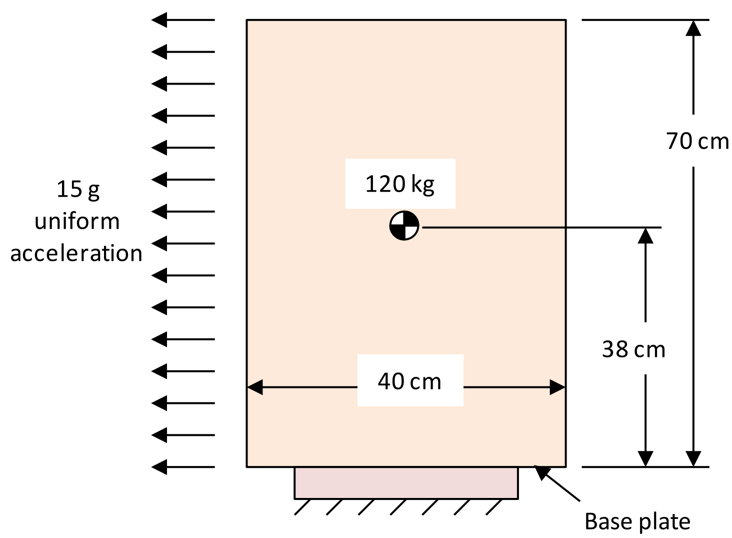


Fig. 7-14. Dimensions and Acceleration for Example 7-1.

Gravitational acceleration on Earth, g, is approximately 9.81 m/s^2. Thus, one kilogram of mass weighs about 9.81 newtons (N) on Earth, so the weight of the spacecraft in this example is 120 x 9.81 = 1,180 N. Figure 7-15 shows the lateral load of 15 x 1180 = 17,660 N at the CG of the spacecraft—recall that inertia load is opposite in sign from the acceleration—along with the equivalent lateral loads acting on the top panel and the base plate, based on the assumption that the dimensions in Fig. 7-15 are dimensions between panel centerlines.

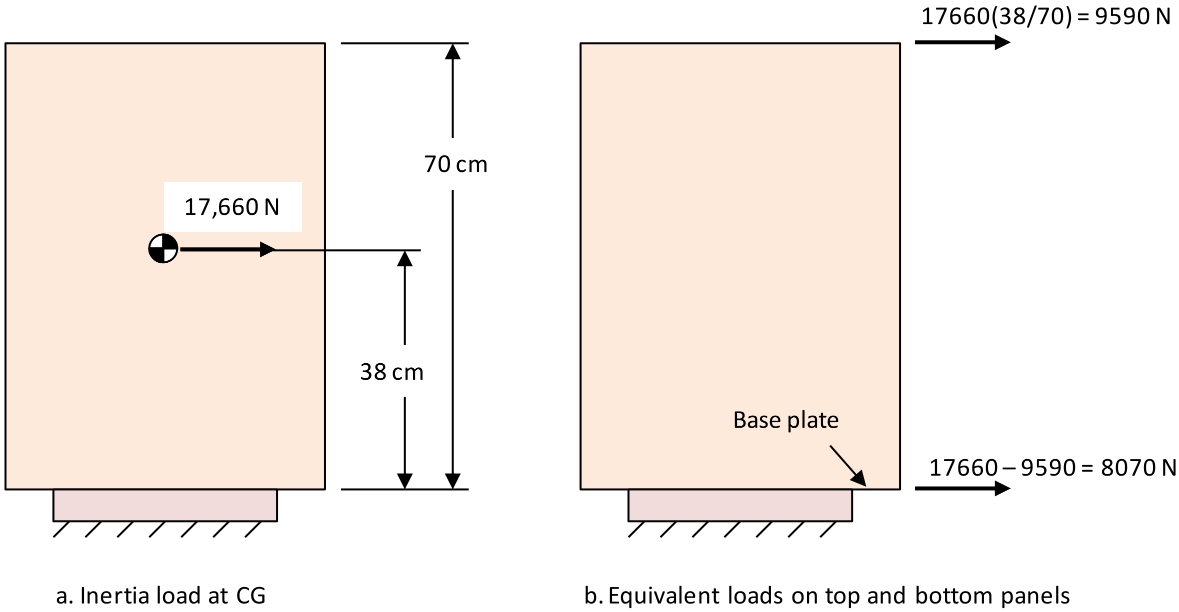


Fig. 7-15. Inertia Loads and Equivalent Loads at Top and Bottom for Example 7-1. These equivalent lateral loads, 9590 N and 8070 N, cause the same shear and moment on the base plate as 17,660 N acting at the CG.

Referring to Fig. 7-13, with h = 70 cm, b = w = 40 cm, and P = 9,590 N per Fig. 7-15b, we can use Eqs. (7.1) and (7.2) to estimate the running loads on the panel edges:

$$v = \frac{9590}{2(40)} = 120 \text{ N/cm } = 68.4 \text{ lb/in}$$

$$\rho = \frac{2(120)(70)}{40} = 419 \text{ N/cm } = 239 \text{ lb/in}$$

Section 7.3.6.3 revisits this example by converting the above loads into bolt loads at panel-to-panel joints.

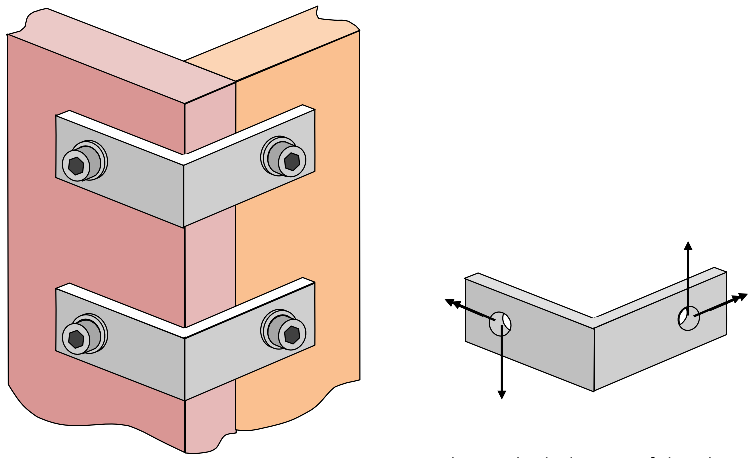
7.3.6 Bolted Joints

Most structural failures occur in joints. Bolted joints are the most commonly used methods of attachment for spacecraft. Structural integrity of bonded and welded joints is much more sensitive to process variables, and most SmallSat programs with low production volume (e.g., one of a kind) can't afford to undergo extensive process development supported by coupon strength testing in order to ensure adequate strength and fatigue life. (On the other hand, such development programs can be very cost effective for programs planning large constellations of SmallSats.)

For information and guidance on designing joints with threaded fasteners, beyond what is provided below, we recommend NASA-STD-5020A (Ref. 5) and (biased as we are!) Instar's three-day course "Design and Analysis of Bolted Joints" (DABJ), developed and taught by Tom Sarafin, and the course book (handbook) by that title (Ref. 6), which is used in the course.

7.3.6.1 Attaching Panels in the Primary Structure

When designing attachments for structural panels, start by reviewing how the structure carries loads. The attachments between panels should be designed to carry loads such as those shown in Fig. 7-13 efficiently. Figure 7-16 shows a poor method of attaching panels—which we've seen for a particular SmallSat, some years ago.



a. A poor way to attach panels

b. Free-body diagram of clip when transferring shear between panels

Fig. 7-16. Poor Way to Attach Panels. Each leg of the clip will be in torsion and will pry on the bolt head, resulting in low strength and low stiffness.

Figure 7-17 shows a better way to attach structural panels. Load paths are direct and stiff.

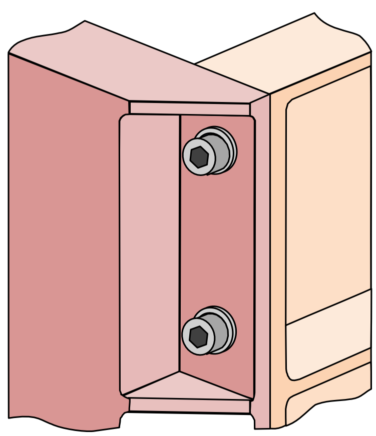


Fig. 7-17. A Better Way to Attach Panels.

When attaching sandwich panels, it's best to use metal parts that are bonded or bolted to both face sheets of each panel. Figure 7-18 shows an example. Engaging a single face sheet on a panel at attachments between panels is inefficient and can lead to unforeseen failure.

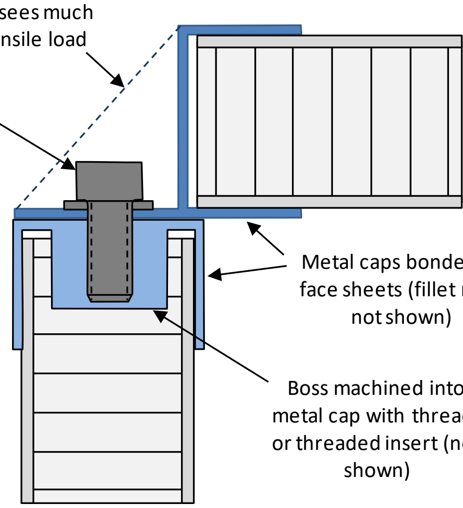


Fig. 7-18. One Way to Attach Sandwich Panels.

7.3.6.2 The Importance of Preload

The most important characteristic of a bolted joint for ensuring integrity is ***preload***: the tensile load in the bolt that is developed during installation, balanced by the equal compressive ("clamp") load between the mating parts (Fig. 7-19). It's the clamp load that's important. Without it, using threaded fasteners in tension joints or in shear joints with clearance holes is a bad idea!

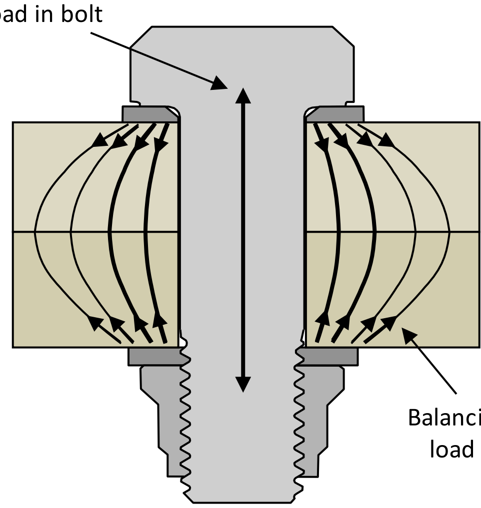


Fig. 7-19. Preloaded Joint. (From Instar's course "Design and Analysis of Bolted Joints" (DABJ))

When a bolted joint without preload is subjected to cyclic tensile loading, the bolt's tensile load varies one for one with the applied load. Fatigue failure is likely in regions where stress concentrates, at the threads and under the bolt head. In addition, such a joint is normally much less stiff when under tensile loads than when under compressive loads, making structural behavior not predictable with linear-elastic FEA when loads can reverse, such as during launch or in a vibration test.

When a bolted joint without preload is subjected to shear loading, one part will slip relative to the other until the bolt is loaded in shear. We certainly can't allow that to happen if the structure must maintain alignment between component interfaces. Even if alignment is not of concern, cyclic slipping of joints under dynamic loads, such as those during launch or during a vibration test, can cause a number of problems:

- The structure won't be predictable with linear-elastic finite element models or analyses.
- While the hardware is in Earth's atmosphere, cyclic slip can lead to fretting corrosion.
- The bolt may fail in fatigue, especially if the threads are in the shear plane, as they are when using threaded inserts such as Heli-Coils®. There's always some bending moment in the bolt associated with shear loading, and cyclic bending stress at the threads invites fatigue failure because of the stress concentration there.

So, how does preload help?

In a tension joint, the clamping preload creates a load path that is considerably stiffer than the bolt itself: applied tensile load causes the clamp load to decrease and the bolt load to increase. After adequately preloading aluminum parts with a steel bolt, the increase in bolt load typically is only around 15% to 20% of the applied tensile load (Ref. 6). As a result, properly preloaded bolts hardly ever fail in fatigue. A high preload also can make a joint nearly as stiff in tension as it is in compression, so that linear-elastic analysis is more valid. The added stiffness increases modal frequencies as well.

In a preloaded shear joint, the clamping force allows the applied load to be carried by friction. As long as friction is not overcome, the joint has high linear stiffness and the bolt sees virtually no cyclic stress.

Outside the region affected by preload, it becomes more important to avoid sudden changes in geometry, which lead to stress concentrations and risk of fatigue failure. Use gradual transitions and generous fillet radii.

7.3.6.3 Strategies for Taking Full Advantage of Preload to Ensure Structural Integrity

- **Use high-strength bolts that are controlled by specifications so that you can preload them highly.**

  The most commonly used threaded fastener in the space industry is the socket head cap screw, controlled by specifications NAS1351 (fine thread) and NAS1352 (coarse thread). These fasteners can be procured in three materials: alloy steel, corrosion-resistant steel (300-series CRES), and heat-resistant steel (A-286). For spaceflight hardware, select A-286. This material is compatible with the space environment and has a minimum ultimate tensile strength of 160 ksi, as compared with 80 ksi for CRES. A-286 fasteners are more expensive than CRES fasteners, but the price is reasonable, and the fasteners are readily available. CRES fasteners should be used only for light-duty applications.

- **Use a bigger (larger diameter) bolt than you think you need.**

  Assuming you're using the A-286 cap screws discussed above, going from a #8 to a #10 increases the ultimate strength from 2360 lb to 3200 lb. Potential preload goes up proportionally. You may be thinking, "Wow, a #10 screw can support the weight of a small car! Why do I need multiple such screws for my 300 lb SmallSat?" You may not <u>need</u> them, but using them will make your structure more robust and predictable. And the weight impact of going from a #8 to a #10 is usually negligible.

  There's a wide range of uncertainty in the preload generated by a given torque. Many variables—particularly coefficients of friction on sliding surfaces, which are in turn affected by many other variables—are the cause of this uncertainty. Depending on the whether the hardware is lubricated at assembly (see below), you probably can ensure the preload for a #10 A-286 screw is at least 1000 lb when aiming for something well higher than that (perhaps 1600 to 1700 lb), given the variability in the torque-preload relationship. If the coefficient of friction between the mating joint members is 0.2, with a minimum preload of 1000 lb each fastener location can carry 200 lb in friction before the joint slips.[4] With enough fasteners in the joint, you can be assured all shear loads during test and during launch will be carried by friction[5]. Joints don't fail when that's the case, and the structure is much more predictable.

  In Example 7-1 we estimated maximum running loads along the panel edges of 68.4 lb/in in shear and 239 lb/in in a direction that is normal to the edges of the vertical panels. If we space the fasteners two inches apart, that makes loads of 2(68.4) = 137 lb and 2(239) = 478 lb per bolt. If we use #10 A-286 fasteners, we should be able to preload them highly enough that the joints won't slip within the clearance holes under the 137 lb of shear per bolt. The 478 lb load, however, could cause slip if that load also is a shear load on a bolt, so it's better to orient those fasteners to be in line with that load (ρ in Fig. 7-13) so that they will be loaded in tension rather than shear. With properly preloaded #10 A-286 fasteners that are so oriented, the joints will not gap under the 478 lb applied tensile load.

- **Don't be afraid to preload your A-286 fasteners highly.**

  In our opinion, there is excessive, unjustified concern about potential yielding of bolts, either while torquing them or afterwards, when load is applied. This concern causes engineers to reduce the installation torque and, as a result, under-utilize their bolts—i.e., not take full advantage of preload.

  There's an adage that "if the bolt doesn't fail during installation, it won't fail in service." (We probably should start that saying with "In a well designed joint, …", though.) This is because, during installation, the applied torque causes torsional shear stress in the fastener, which interacts with the tensile stress caused by the preload. Once the torque wrench is removed, the torsional

[4] NASA-STD-5020A (Ref. 5) advises us to count on no higher than a 0.2 coefficient of friction, but only for cleaned, bare-metal surfaces, and 0.1 for all other surfaces, including metal that has a conversion coating such as Alodine®.
[5] Note that most space programs (see Ref. 5) use a criterion that analysis for ultimate strength, using an ultimate factor of safety, must not count on friction as a beneficial load path. Calculation of ultimate margins of safety must be based on the assumption that the entire applied shear load is carried by the fasteners. That does not mean, however, that we should stop doing all we can to maximize friction in joints!

stress relaxes and thus does not contribute with tensile stress to cause noticeable yielding or rupture.

Yielding is considered failure only if it is detrimental. A small amount of yielding during installation simply strain hardens the material. Yielding under applied loads, when combined with a high preload, typically causes a loss of only 10 to 15% of the original preload (Ref. 6). Given that you won't really know the preload for a given applied torque—you'll know only a range of likely preload—yielding under applied load will occur only if the initial preload is at the high end of that range; residual preload after yielding will then be closer to the middle of that range.

To help you understand how highly you can torque your fasteners, do a simple test: With the same combination of materials, hardware, and lubrication (if applicable—see below) as for the flight design, but with dedicated hardware, torque a few fasteners to failure. With A-286 socket head cap screws, you should be able to feel the material yielding; you'll be turning the wrench and the torque is no longer increasing. Continue until rupture occurs. For the flight hardware, use an installation torque that is perhaps 75% of the torque that caused noticeable yielding during the test. Given the variability of preload for a given applied torque, this approach should ensure there will be no excessive yielding during installation under the combination of tensile and torsional stresses.

- **To maximize preload, lubricate fasteners prior to installation.**

  Lubrication reduces the coefficient of friction, turning more of the applied torque into preload and reducing torsional stress in the fastener. For best results, lubricate all sliding surfaces, including threads and under the bolt head or nut. Lubricants with acceptable outgassing properties for space use include molybdenum disulphide dry film and Braycote 601EF and 602EF greases.

- **Use nuts and threaded inserts that are controlled by specifications and that are at least as strong as the bolts.**

  A common mistake in the design of bolted joints is to not match the strength of the fastening hardware. When the specified strength of the nut or the insert is lower than that of the fastener, we must reduce the preload accordingly. First, though, we must recognize when this is the case.

  For example, consider the #10 A-286 cap screw discussed above, with a specified minimum tensile strength of 3200 lb. That screw is frequently paired with an MS21043 nut, but that nut has a specified minimum strength of only 2460 lb. The specified strength of a nut is not necessarily related to failure of the nut material. A nut expands under load, pulling its threads away from the bolt threads. When the nut and bolt materials are of comparable strength, the bolt threads strip (fail in shear) before the nut threads strip. The external thread form used for bolts is weaker by geometry (lower shear-engagement area) than the internal threads used for nuts. The internal threads would strip first only if the nut material is considerably weaker than the bolt material.

  Socket head cap screws are even more frequently used with Heli-Coils® (Fig. 7-20). The strength of a Heli-Coil is limited by the strength of the parent material (the material in which the insert is installed). Heli-Coils are sold in length increments of 0.5D, where D is the bolt diameter (0.190" for a #10). With 6061-T651 aluminum alloy as the parent material (in which the insert will be installed), the procurement specification (NASM8846) for the NASM21209 (formerly MS21209) helical coil insert says we should use a 2D insert to be sure that threads in the aluminum parent material won't strip before reaching 3200 lb for this #10 screw. We've seen many spacecraft designs that use Heli-Coils shorter than 2D installed in 6061-T651 for A-286 fasteners. This combination often works, but the reduced insert pull-out strength introduces avoidable risk and limits the preload that can be achieved.

Fig. 7-20. Heli-Coils® of Different Length.

- **Make sure the nut or threaded insert does not encroach on the runout threads.**

  The first two threads, starting at the full-diameter body or the bolt head, depending on the fastener, are not full threads. These ***runout threads*** (Fig. 7-21) transition gradually from fully formed threads to the full-diameter body. The purpose of this gradual transition is to reduce stress concentrations and improve fatigue life. If the internal threads in the nut or the insert encroach onto the runout threads, the resulting interference ensures you won't get the intended preload after applying the full torque. This detail is commonly overlooked in design. A simple check is to see if the washer spins after torquing. This check is not foolproof, though, because there may be a small amount of preload; interference with runout threads is gradual.

- **When you can do so, install threaded inserts in holes that are tapped all the way through the part rather than part way (referred to as *blind*) (Fig. 7-22).**

  Blind holes are sometimes necessary to maintain a seal or to ensure particulates aren't pushed into sensitive items such as bearings. If there's no need for a blind hole, though, drill the pilot hole (for the tap that will be used to cut threads) all the way through the part, and then run the tap all the way through as well. With a blind hole, dimensions have to be carefully designed and controlled to ensure the fastener won't bottom out in the hole—which again would mean little or no preload. We all know Murphy 's Law—"Anything that can go wrong will go wrong"—so design in a way that makes sure nothing can go wrong!

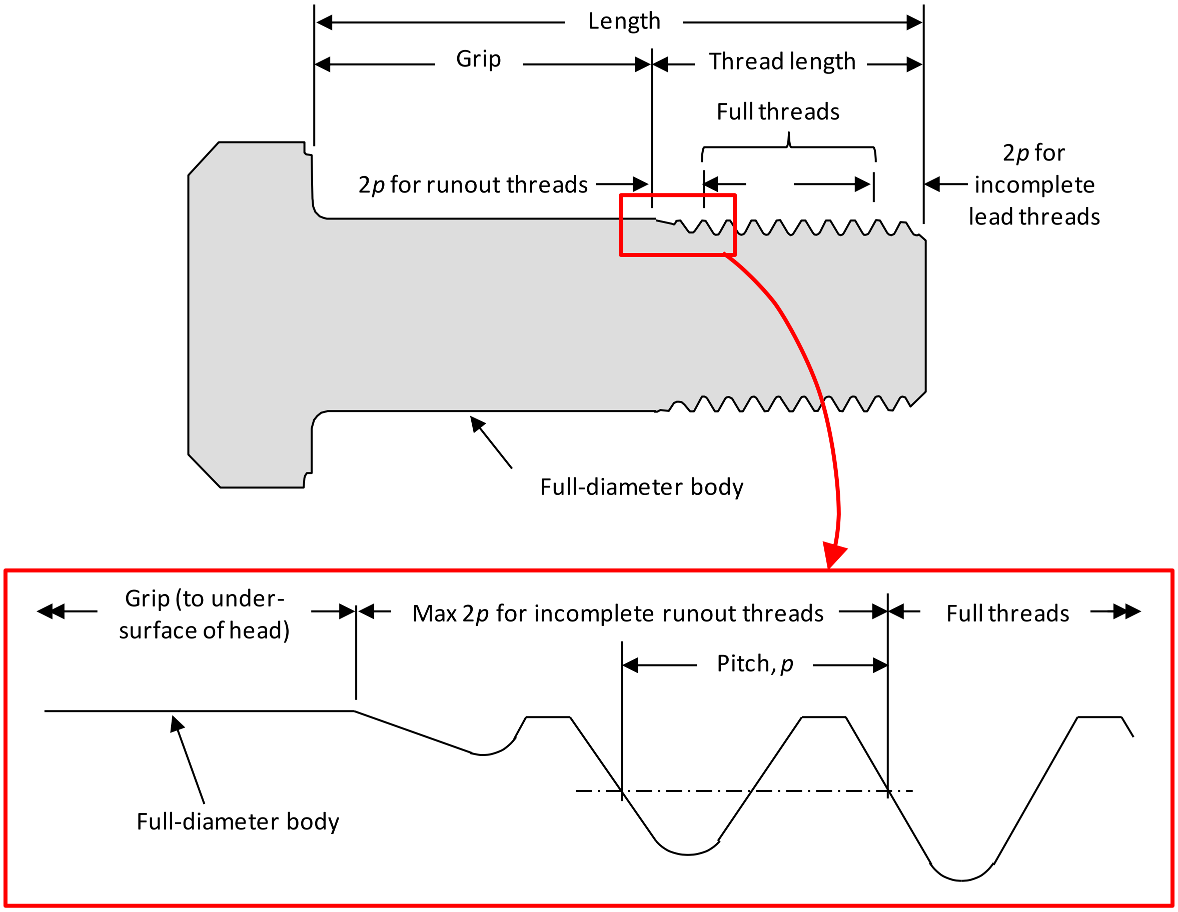


Fig. 7-21. Profile of Runout Threads. (From Instar's course "Design and Analysis of Bolted Joints" (DABJ))

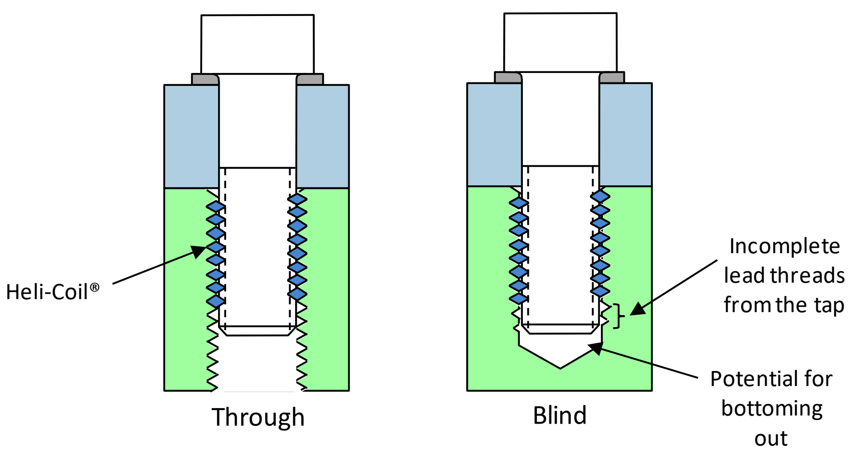


Fig. 7-22. Through Hole and Blind Hole for Threaded Inserts. A blind hole has to be deep enough to ensure (a) the insert does not encroach upon the incomplete lead threads from the tap and (b) the fastener doesn't bottom out in the hole. (From Instar's course "Design and Analysis of Bolted Joints" (DABJ))

7.3.7 Design Loads

Defining or identifying appropriate design loads at the SmallSat level and at the component level can be difficult. Clearly, we invite problems if we don't design the spacecraft or its components to loads that are at least as high as those that will be experienced in test. And, if we don't design the test to envelop the effects of launch, as discussed in Parts 4 and 6 of this paper series, we invite problems during launch.

We can't assume that the maximum loads resulting from coupled loads analysis (CLA) fully encompass the effects of launch. CLA is typically done only up to 50 or 100 Hz, depending on the launch-vehicle contractor (LVC). This upper limit on frequency is referred to as the ***truncation frequency*** because modes of vibration in the coupled system model are truncated above that frequency, meaning that they are not included in the analysis. The effects of random vibration are either included in the CLA only up to the truncation frequency or are not included at all. As the mass of an LV payload (e.g., SmallSat) or a SmallSat component decreases, the portion of total launch load caused by random vibration increases.

At the SmallSat (LV payload) level, maximum expected equivalent quasi-static loads are normally specified by the LVC either in a user guide or in an interface control document (ICD). Whether those loads fully encompass the effects of launch, including random vibration introduced at the base of the SmallSat, depends on how the LVC derived those loads. Over the last ten years or so, many new LVs have been (and are being) developed, and other established LVs are being used nowadays to fly SmallSats as part of rideshare missions. Engineers at such companies may not be fully tuned in to your considerations as a SmallSat developer, and some may not include the effects of high-frequency loads in the derived quasi-static loads.

A key point from the above discussion: It's important for the SmallSat developer to understand (a) the truncation frequency for the CLA and (b) whether the CLA results and the specified quasi-static loads include any of the effects of vibroacoustics. Be sure to ask the LVC.

Even if you find out that the effects of random vibration produced by the LV are adequately included in the loads provided by the LVC, the vibroacoustic response of your SmallSat may contribute significantly as well. This can be the case if your SmallSat has solar panels or shells that expose a lot of surface area in the launch (stowed) configuration.

When in doubt, or when the analysis required to predict loads becomes too difficult or time consuming, many programs have used a ***mass-acceleration curve*** (MAC) to define maximum expected ("limit") quasi-static loads for design and test. Such a curve relates load to mass, with lower-mass items understandably expected to see higher equivalent quasi-static load. (It takes less energy to accelerate a lower mass.) A MAC is derived based on some combination of flight data, test data, and analysis. Without more information and resources than are usually available, a SmallSat developer won't be able to generate an appropriate MAC. The LVC or the integration contractor may be able to do so, however.

Figure 7-23 shows two MACs for LVs that no longer fly; as far as we know, these are the first such MACs published. Although many other MACs have been developed through the years, most are not in the public domain, so we can't include them here, but they're not too far off from those in Fig. 7-23. Although any MAC that you might be able to find probably applies to components of relatively large spacecraft, such as the MACs in Fig. 7-23, the MAC should be reasonable for use at the SmallSat level. SmallSats mounted on a rideshare structure (sometimes referred to as a "dispenser") are physically analogous to components mounted on a large spacecraft.

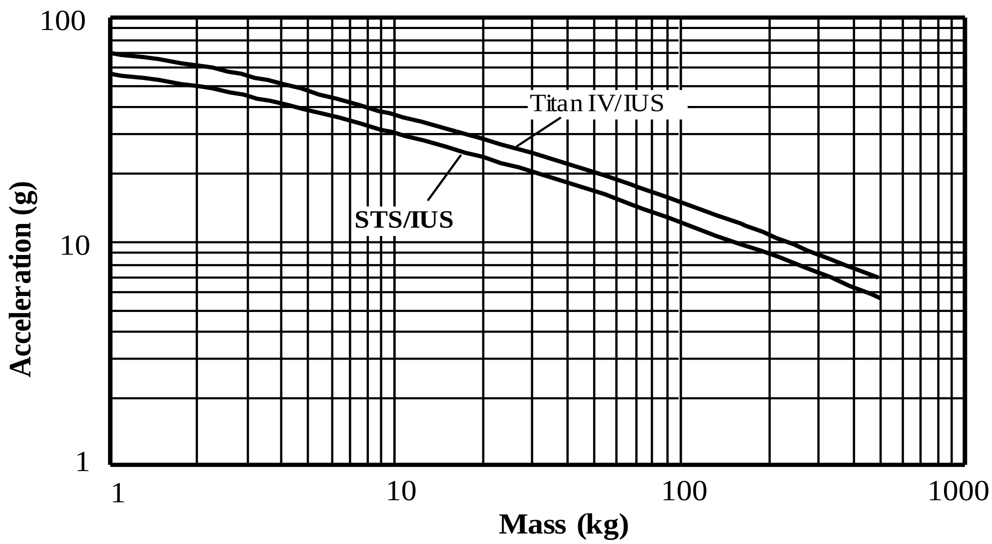


Fig. 7-23. JPL's Mass-Acceleration Curves for Space Shuttle and Titan-IV Launches (Ref. 7). These curves were intended for preliminary design of components of spacecraft significantly larger than today's SmallSats. The load applies in each axis, one axis at a time, with an additional 1.5 g for Shuttle and 3.9 g for Titan-IV in the thrust axis.

An important caution: Most MACs do not apply to SmallSat components, at least not for the levels that they may see during your SmallSat random vibration test, especially if the test is not notched or force limited. Most MACs are derived from flight data and data from acoustic testing of large spacecraft. Some SmallSat programs have used MACs to justify notching or response limiting during the random vibration test.

Even if you don't have an applicable MAC for your SmallSat or its components, you can use Fig. 7-23 as a sanity check for your design loads, sine burst test levels, and response limits intended for random vibration testing at the SmallSat level. For example, for a 100 kg SmallSat, using the higher of the two curves in Fig. 7-23 and a test factor of 1.25 for qualification, the sine burst test should be done to a level approximately equal to 1.25 x 15 = 18.75 g, one axis at a time. For a related discussion on deriving appropriate random vibration test environments for components, see Part 5 of this paper series.

Summary

We hope this series of papers helps you improve your structural designs and plan, perform, and interpret your vibration tests. Feel free to contact us with questions: tom.sarafin@instarengineering.com

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