



VTSS—Vibration Testing of Small Satellites

Course Book

Last revision: April 2026

Developed and taught by

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Disclaimer

Our intent is to provide dependable information and recommendations regarding vibration testing, but we cannot guarantee their accuracy.

Vibration Testing of Small Satellites

- Objectives: Improve your understanding of how to ...
- establish an effective vibration test program
 - identify and clearly state test objectives
 - design (or recognize) a test that satisfies the objectives while minimizing risk of an over test
 - design and technically justify a force limited, response limited, or manually notched vibration test
 - establish pass/fail criteria and interpret test data
 - design a SmallSat that will withstand its vibration test

Audience: All engineers involved in ensuring small spacecraft can withstand mission environments

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My Bio—Tom Sarafin

Engineering consultant and teacher in the space industry, in the fields of structures and mechanical systems

- BS in Civil Engineering (1979), with an emphasis on structures, from the Ohio State University
- Started my career in 1979 as a stress analyst at Martin Marietta, Denver, gaining valuable experience in structural analysis, preliminary structural design, structural test engineering, verification planning, technical writing, making presentations, and managing teams and projects
- Editor and principal author of *Spacecraft Structures and Mechanisms, from Concept to Launch* [1995] (nearly 4 years, full time; left Martin Marietta in 1993 to finish this book)
- Currently president, chief engineer, and sole owner of Instar (founded Instar in 1993)
- Have consulted for NASA, DARPA, DOD Space Test Program, Lockheed Martin, DigitalGlobe (now Vantor), U.S. Air Force Academy, Sierra Nevada Corp (Sierra Space), Planetary Systems Corp (part of Rocket Lab USA), and others
- Key member of the team that developed NASA-STD-5020 for design, analysis, quality assurance, and verification of threaded fastening systems (bolted joints)

continued

My Bio, continued

- Other publications:
 - Contributing author to *Space Mission Analysis and Design* (1st, 2nd, and 3rd eds) [1991, 1992, & 1999] and *Human Spaceflight: Mission Analysis and Design* [1999]
 - Multiple technical papers, including a series on vibration testing of small satellites
- Have taught over 300 2-, 3-, and 4-day courses since 1995 to more than 6000 engineers and managers:
 - SCS—Spacecraft Structures, from Concept to Launch
 - SDA—Structural Design and Analysis for Aerospace Engineers
 - DABJ—Design and Analysis of Bolted Joints for Aerospace Engineers
 - STDI—Structural Test Design and Interpretation and VTSS—Vibration Testing of Small Satellites
 - TenP—Ten Principles for Successful Space Programs

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} Not
covered
in class

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Appendix: Test Documentation and Reviews

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Summary

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Conversion of Units

- One inch (in or ") = 2.54 centimeters (cm) = 0.0254 meters (m)
- One mil = 0.001 in
- One micron (μm) = 10^{-6} m = 3.937E-5 in
- One foot (ft) = 12 in
- One nautical mile (nmi) = 1.151 mile (mi) = 1.852 kilometers (km)
- One pound (lb) = 4.4482 Newton (N)
- One pound of weight corresponds to 0.45359 kilogram (kg) of mass
 - Something having a mass of one kg will weigh about 2.2 lb
- One lb-s²/in of mass = 12 slug (sg) $\cong$ 175 kg
- Mass moment of inertia: one lb-s²-in = 0.11298 kg-m²
- One kip = 1000 lb
- One pound per square inch (psi) = 6894.8 Pa = 6894.8 N/m²
- One kip per square inch (ksi) = 6.8948 MPa
- g = gravitational acceleration $\cong$ 386.1 in/s² $\cong$ 9.81 m/s²
- To convert between Fahrenheit ($^{\circ}\text{F}$), Celsius ($^{\circ}\text{C}$), and Kelvin (K):
$$^{\circ}\text{C} = (5/9)(^{\circ}\text{F} - 32)$$
$$\text{K} = ^{\circ}\text{C} + 273.15$$

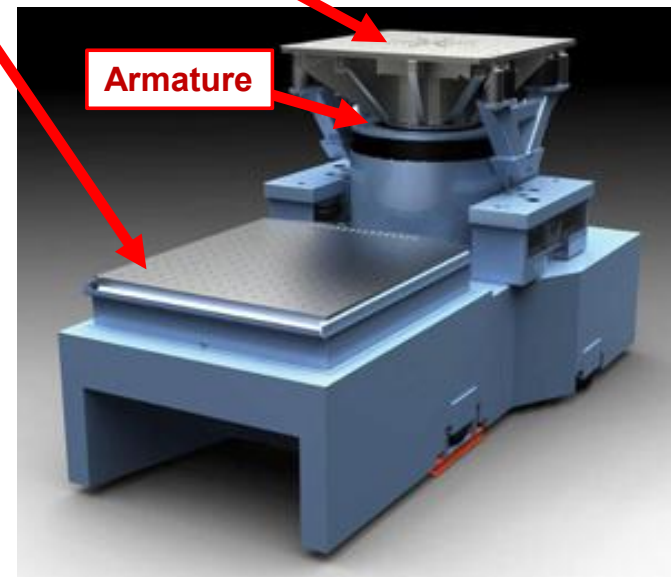
1. Overview of Vibration Testing

- Electrodynamic Shakers and Test Configuration
- Vibration Test Objectives
- Review of Mechanics
- Transmissibility
- Common Types of Vibration Tests
- Qualification, Acceptance, and Protoflight Testing
- Limitations of Testing on a Shaker
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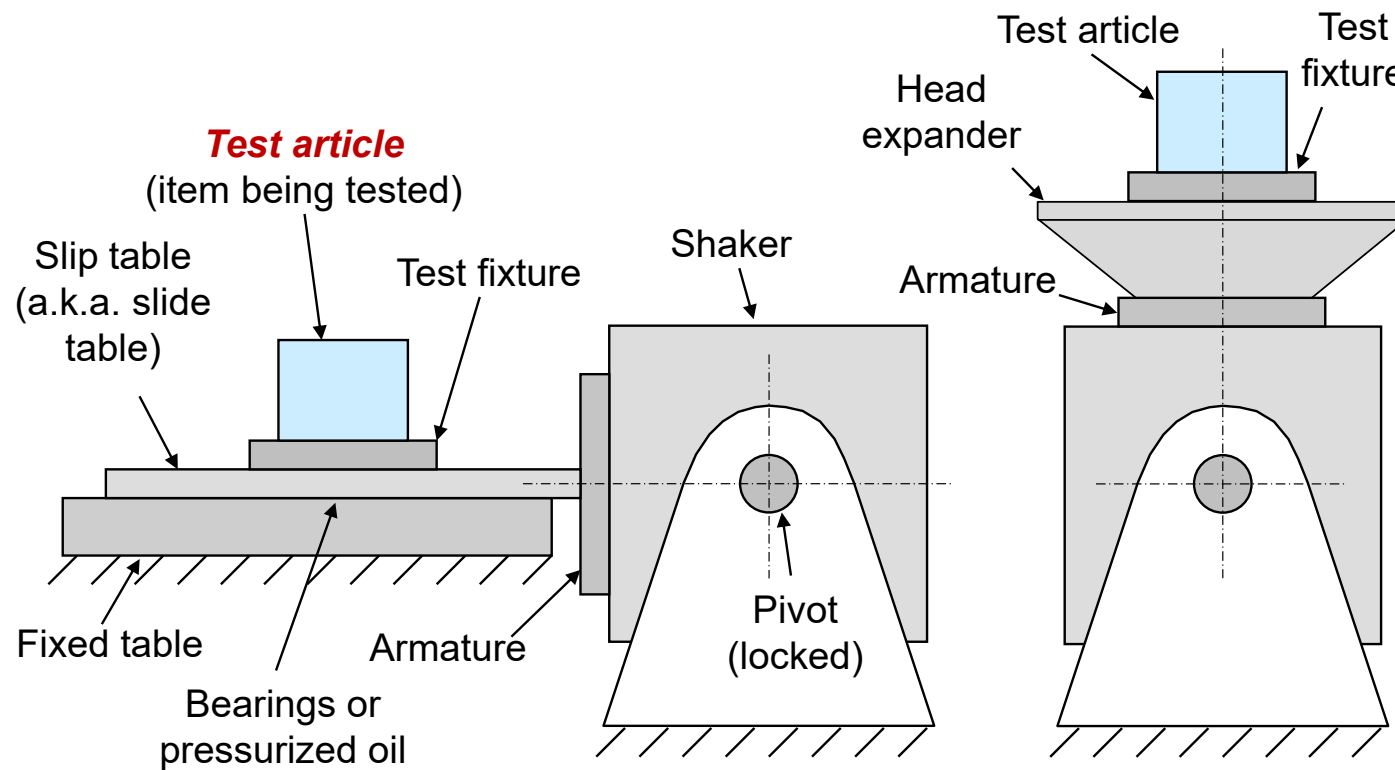
Electrodynamic Shakers

- An **electrodynamic shaker** is a device that imparts force that is proportional to electrical current passing through a coil of wire suspended in a magnetic field.
- Force causes acceleration that depends on the amount of mass moving.
 - Test article plus armature and **slip table** or **head expander** (a.k.a. slide table and expander head)
- Acceleration is controlled by use of a control channel.
 - Typically the average of two or more accelerometers placed at the base of the test article.
 - Closed-loop control system: Current is automatically adjusted until the control channel measures the desired acceleration.



Dayton T. Brown (DTB) T5500-3 shaker

Typical Configurations for Vibration Tests



What Are Some Likely Reasons for Doing a Vibration Test?

~~It's a requirement imposed by the launch service provider (LSP) or your customer.~~

- Contractually this might be the case, but a test is not a true requirement. It's a method of verification.
- You, the engineer responsible for the test, must understand and embrace the actual reason (why the LSP or your customer is requiring the test).

The main reason for vibration testing is to establish stakeholder confidence (which is what “verification” is all about) that the SmallSat can withstand launch environments and still function as required—the true requirement.

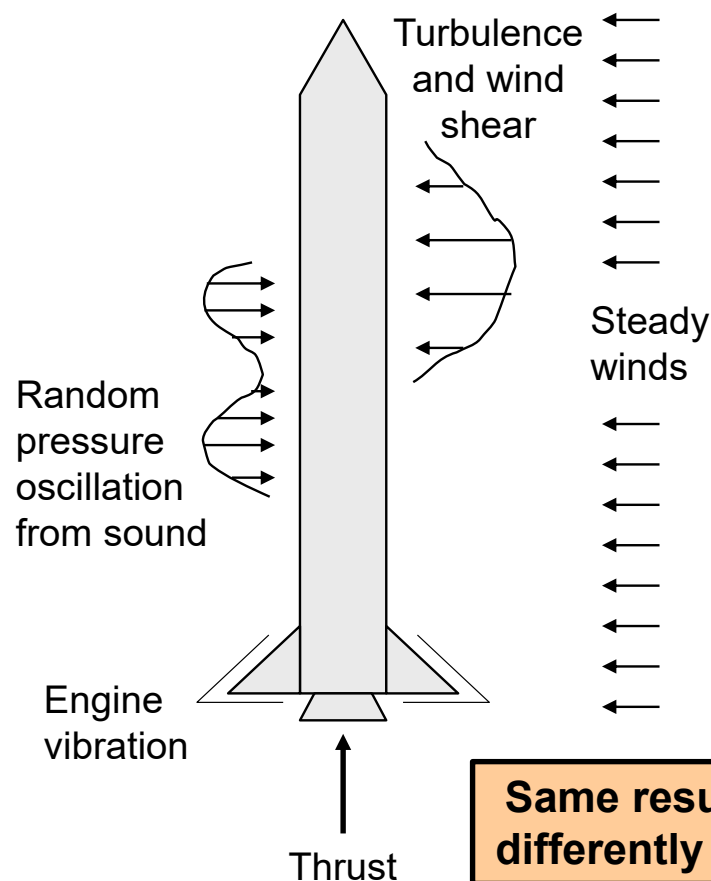
Other reasons can be to ...

- build confidence in a design before committing to it (development testing).
- determine natural frequencies.
- obtain information needed to correlate a finite element model in order to improve accuracy of predicted responses to mission environments.

Two main reasons—or objectives—for doing a test:

- **Verify compliance with a true requirement**
- **Obtain information**

Effects of Flight Loading Environments



Two types of loads act on a flight vehicle: steady-state (constant) and time-varying.

- Steady-state loads cause uniform acceleration along with resisting inertia loads.
 - Inertia loads cause stress in the materials.
- Time-varying loads cause structures to vibrate.
 - Acceleration and corresponding inertia loads vary according to the shapes of the excited modes of vibration.
 - Inertia loads cause stress in the materials.

Same result at the material level—but materials can fail differently when stress is cyclic, such as from vibration.

Effects of Stress on Materials

- A material can take only so much stress before failure occurs.
 - Under one-time application of stress, failure can be ...
 - Rupture
 - Collapse (a structural failure rather than a material failure)
 - Yielding that results in permanent deformation that is detrimental to form, fit, or function
- referred to as **ultimate failure**
- yield failure**

Yielding is considered failure only if it is detrimental.

- Under cyclic stress, ...
 - a crack may form in a metal and then grow to full rupture (**fatigue**)
 - materials at contacting surfaces can suffer detrimental wear

Two structural characteristics required of all hardware:

Strength: the highest load a structure can withstand (or highest stress a material can withstand) without failure

Life: the number of cycles (or duration) of load or stress before failure

Review

Before we look at different types of tests on a shaker, let's review some basic physics and the mechanics of vibration.

We'll start with terminology ...

Quasi-Static Loads and Load Factors

- A **static load** is a steady-state external load (force or moment) acting on an object and is a term typically associated with static equilibrium, with the object having no acceleration.
- An **inertia load** is the internal load that resists acceleration.
- A **quasi-static load** is a term used for (a) uniform steady-state acceleration and (b) the inertia load associated with uniform acceleration.
- A **dynamic load** varies over time and either causes or is associated with vibration.
- A **load factor** is a multiple of g (Earth's gravitational acceleration), representing a quasi-static inertia load; thus, it's also a multiple of weight on Earth.
- Quasi-static loads are often defined in g units; however, an inertia load acts in a direction opposite that of the acceleration and thus is of opposite sign from the acceleration:

$$a = -ng \quad (\text{Eq. 1-1})$$

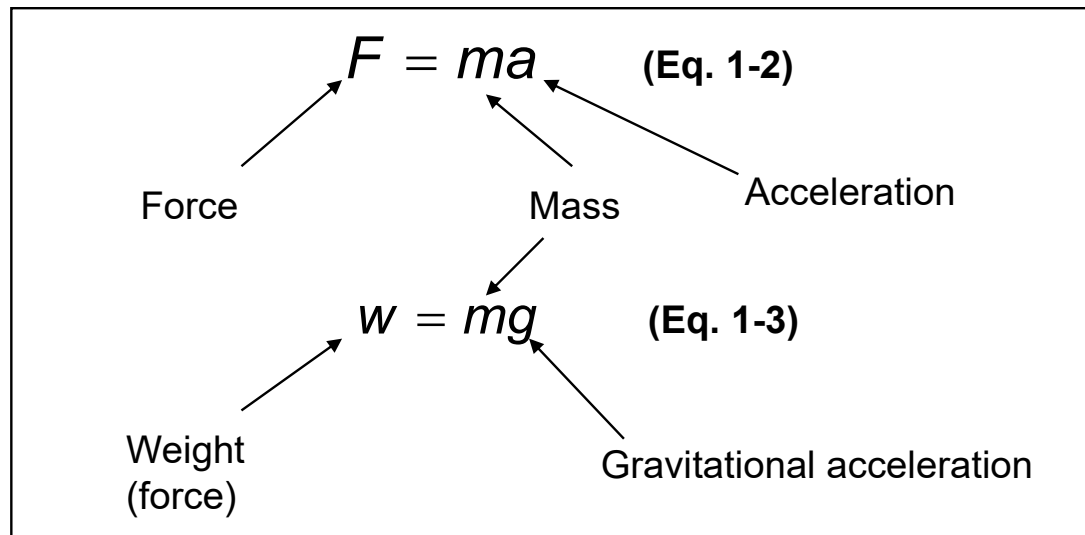
where a = acceleration (in/s² or m/s²)
 n = load factor
 g = gravitational acceleration (in/s² or m/s²)

Limit load is the maximum expected load during launch or some other event.

In a vibration test, acceleration is reversible, so the sign difference between acceleration and load factor is typically not important.

Before we review the mechanics of vibration,
Remember: Weight Is Not the Same As Mass!

The weight of an object is less on the moon than on Earth, but the mass is the same.



Weight is in units of force.

Kilogram is a unit of mass, not force.

Pound is a unit of force, not mass.

In this course, I do not recognize the “pound mass” unit. It’s unnecessary, and I believe it’s caused a great deal of confusion.

	US customary units	SI units
Weight	lb	N
Acceleration	in/s ²	meter/s ²
Mass	lb-s ² /in (=“slinch”)	N-s ² /meter (= kg)

The Convenience of Using Acceleration in “g” Units

When using g as the unit for acceleration, we use weight (lb or N units) in place of mass and load factor (or number of g 's) in place of acceleration.

$$F = ma \quad (\text{Eq. 1-2})$$

$$F = \left(\frac{w}{g} \right) (-ng)$$

Recall: $m = w/g$ from Eq. 1-3,
and $a = -ng$ from Eq. 1-1

$$\text{External force} \Rightarrow F = -nw \quad (\text{Eq. 1-4a})$$

$$\text{Internal force} \Rightarrow F_i = nw \quad (\text{Eq. 1-4b})$$

where

m = mass

w = weight

a = acceleration

g = gravitational acceleration

n = load factor

Example: An object weighs 100 lb (445 N).

The object is under 5g acceleration.

The load factor for that object is -5.

The inertia force is -500 lb (-2224 N).

The external force that causes 5g acceleration is 500 lb.

$$w = 100 \text{ lb}$$

$$a = 5g$$

$$n = -5$$

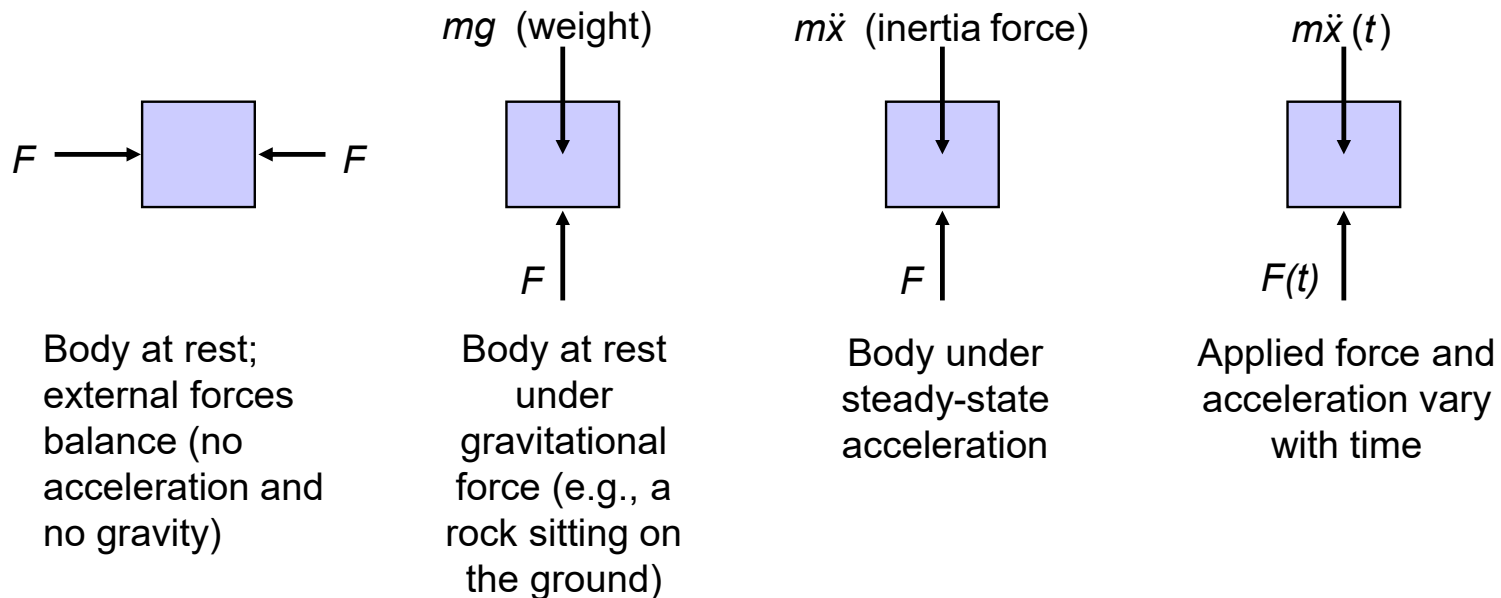
$$F_i = nw = -500 \text{ lb}$$

$$F = -nw = 500 \text{ lb}$$

Force Equilibrium for Accelerating Objects

Static equilibrium applies when there is no acceleration: $\Sigma F = 0$, where F is external force.

For an accelerating object, external force is balanced by an equal and opposite internal force :



F = force

m = mass

g = gravitational acceleration

$\ddot{x}$ = acceleration

t = time

**Everything is in force equilibrium—always.
External forces, gravitational forces, and
inertia forces always sum to zero.**

From my course "Spacecraft Structures, From Concept to Launch" (SCS)

Modes of Vibration

Every structure has a near-infinite number of DOFs (degrees of freedom), with a **mode of vibration** for each DOF. Each mode is characterized by ...

- **Natural frequency**: frequency at which the structure vibrates once excited and after the excitation is gone
- **Mode shape**: deformed shape of the vibrating structure
- **Damping**: loss of energy in a vibrating structure

The **fundamental frequency** is the structure's lowest natural frequency, corresponding to the fundamental (first) mode of vibration, which usually ...

- has the most mass participation
- and is the easiest mode to excite

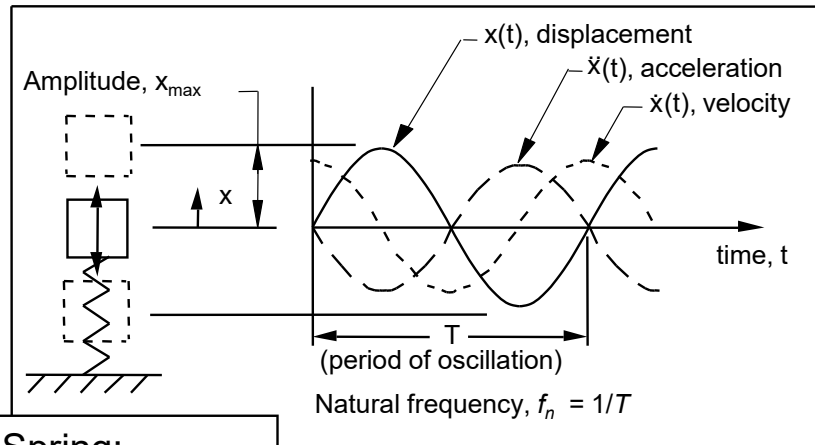
Note: If the first mode for a spacecraft is bending of a small antenna, that mode is considered the fundamental mode of the antenna but not of the spacecraft.

To simplify structural analysis, we try to idealize or model a structure with as few DOFs as needed for the desired accuracy.

Often we can use just one with what we call a **single-degree-of-freedom system (SDFS) (mass on a spring).**

Harmonic Motion of an SDFS

- Once disturbed by a time-varying force, a structure will vibrate.
- **Free vibration** is vibration that continues after external forces are removed.
- In absence of damping, each point in a freely vibrating, linear structural system experiences **harmonic motion**, in which displacement is a sinusoidal function of time.
- Velocity and acceleration are the first and second derivatives of displacement with respect to time, and are thus also sinusoidal functions of time.



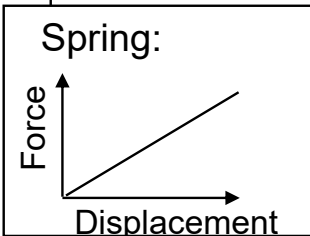
$$\text{Displacement, } x(t) = x_{\max} \sin(\omega_n t) \quad (\text{Eq. 1-5})$$

$$\text{Velocity, } \dot{x}(t) = \omega_n x_{\max} \cos(\omega_n t) \quad (\text{Eq. 1-6})$$

$$\text{Acceleration, } \ddot{x}(t) = -\omega_n^2 x_{\max} \sin(\omega_n t) \quad (\text{Eq. 1-7})$$

Note: Acceleration is proportional to displacement.

$$\begin{aligned} \omega_n &= \text{natural frequency in rad/s} \\ &= \sqrt{k/m} \quad \begin{array}{l} k = \text{stiffness} \\ m = \text{mass} \end{array} \end{aligned}$$



Natural frequency in Hz:

$$f_n = \frac{1}{T} = \frac{1}{2\pi} \sqrt{k/m} = \frac{\omega_n}{2\pi} \quad (\text{Hz}) \quad (\text{Eq. 1-8})$$

Damping

- If there were no energy-dissipating forces, a structure, once disturbed, would vibrate forever.
- **Damping**, which is the loss of energy in a vibrating structure (e.g., from friction), allows vibrations to die out by applying forces that resist motion.
- **Viscous damping**, the most commonly assumed damping model, is proportional to velocity:

$$\text{Damping force} = -c \dot{x}(t) \quad (\text{Eq. 1-9})$$

c = damping factor
 $\dot{x}(t)$ = velocity

- The **critical damping factor** is the value that would allow the mass, once disturbed, to return to its initial position without vibration:

$$\text{Critical damping factor, } c_c = 2m\omega_n = 2\sqrt{km} \quad (\text{Eq. 1-10})$$

m = mass
 ω_n = natural frequency, rad/s

- The **damping ratio**, ζ , is a measure of a structure's damping relative to the critical damping:

$$\zeta = \frac{c}{c_c} \quad (\text{Eq. 1-11})$$

The damping ratio is difficult to predict for most structures. Until we test the structure, we must rely on data from tests of similar structures.

From my course "Spacecraft Structures, From Concept to Launch" (SCS)

The Equation of Motion for a Linear SDFS

All forces acting on an object (mass) must always sum to zero:

Applied force + inertia force + damping force + spring force = 0

$$F(t) + [-m\ddot{x}(t)] + [-c\dot{x}(t)] + [-kx(t)] = 0 \quad (\text{Eq.1-12})$$

$\ddot{x}$ = acceleration (d^2x / dt^2) $\dot{x}$ = velocity (dx / dt) x = displacement

where

m = mass

c = damping factor

k = stiffness

t = time

Rearranging terms results in the **equation of motion** (an equation of equilibrium) for a linear SDFS:

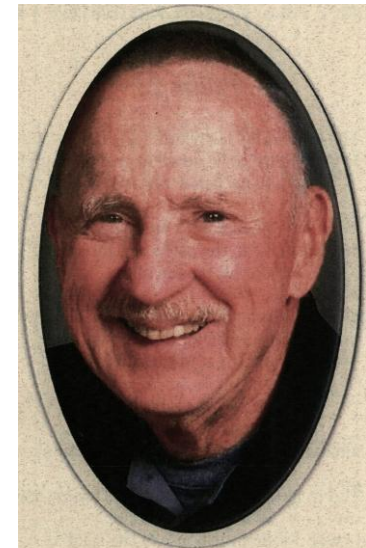
$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t) \quad (\text{Eq. 1-13})$$

or
$$m\ddot{x}(t) + 2\zeta\omega_n m\dot{x}(t) + \omega_n^2 mx(t) = F(t)$$

ω_n = natural frequency (rad/s)

ζ = damping ratio

The equation of motion is the basis for dynamic analysis and can be easily modified to apply to any mode of vibration when damping is relatively low. (See SSAM (Ref. 9) Sec. 7.2.)



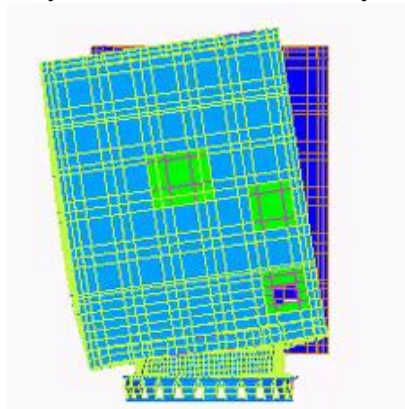
Peter Abbott
January 14, 1939 – May 16, 2025
(dynamics mentor)

Mode Shapes

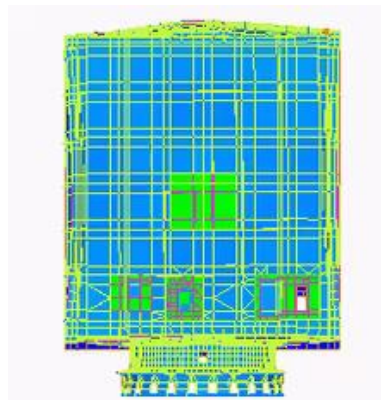
- Each mode of vibration has a different shape
- Some modes are analogous to that of a mass on a spring
 - Concentrated mass, concentrated spring
- Others are more like the bending modes of a beam
 - Distributed mass, distributed spring

FalconSAT-3 engineering model:

Courtesy U.S. Air Force Academy

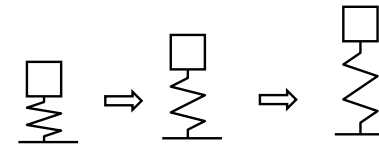


Fundamental lateral (rocking) mode

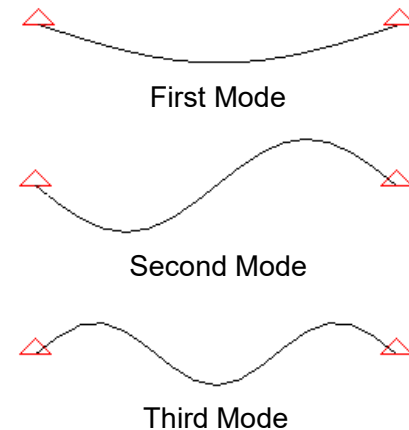


Fundamental axial mode

Mass on a spring:



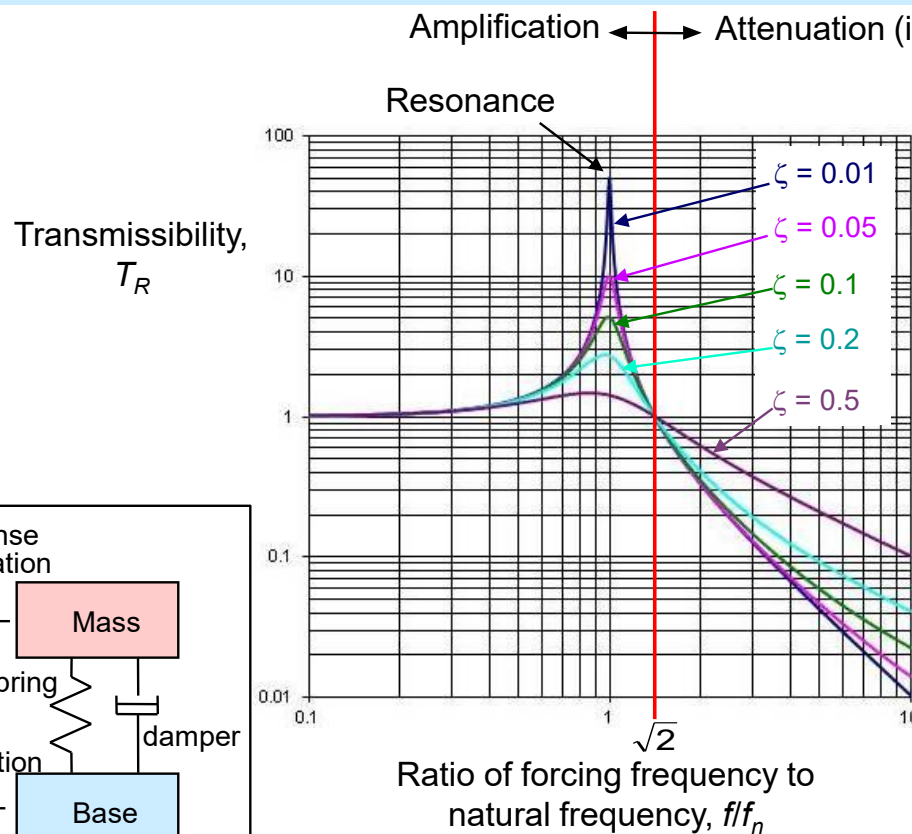
Beam:



For a structure vibrating in a single mode, when ignoring damping, each point moves as a sinusoidal function of time.

Response to Base-driven Sinusoidal Vibration: Transmissibility

Transmissibility is the gain in a sinusoidally base-driven linear SDFS: peak response acceleration divided by peak base (input) acceleration as a function of frequency.



$$T_R = \sqrt{\frac{1 + \left(\frac{2\zeta f}{f_n}\right)^2}{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \left(\frac{2\zeta f}{f_n}\right)^2}} \quad (\text{Eq. 1-14})$$

where

f = Forcing frequency in Hz

f_n = System's natural frequency in Hz

ζ = Damping ratio

Quality factor, $Q \cong \frac{1}{2\zeta}$

= transmissibility at resonance for a mass on a spring

(Eq. 1-15)

Common Types of Vibration Tests on Shakers

- **Sine burst test** (This is NOT a static loads test; it's dynamic.)
 - A strength test of the main (primary) structure.
 - Shaker imparts sinusoidal acceleration at a frequency that is low relative to the test-article's fundamental frequency so that there will be minimal dynamic amplification.
 - Result: near-uniform acceleration.
- **Sinusoidal vibration (sine vibe) test**
 - A test of general mechanical integrity, usually for integrated spacecraft.
 - Shaker imparts sinusoidal acceleration at gradually increasing or decreasing frequency to exercise the assembly in a manner representative of flight loading.
- **Random vibration test**
 - A test of general structural integrity, including resistance to cyclic loading and strength of relatively small structures. Qualification testing is intended to verify fatigue life.
 - Shaker imparts acceleration at multiple frequencies simultaneously across a given range (20 – 2000 Hz in the space industry), at random amplitude.
- **Low-level sine sweep**
 - Like a sine vibe test, but done at low levels to determine natural frequencies and damping, and to assess structural integrity after high-level testing on a shaker.

(Shock testing also can be done on a shaker, but typically only at low levels.)

Test Classification: Qualification and Acceptance Testing

Qualification (“qual”) **test** (a.k.a. **prototype test**)

Objective: Demonstrate the adequacy of a design by testing a dedicated non-flight article—built just like a flight article—to conditions that are more severe than expected for the mission. The combination of test level and duration in excess of acceptance testing is considered the **qualification margin**, which is intended to account for build-to-build variation.

Applications: When we are not confident in our ability to predict a design’s key characteristics or response or resistance to an environment.

Limitations: Effectiveness depends on how well manufacturing processes are controlled (i.e., whether the qualification margin bounds the build-to-build variability).

Acceptance test

Objective: Demonstrate the adequacy of each product by testing it to the maximum predicted environment for the mission.

Applications: When we are not confident that our manufacturing processes will limit product variability within the range covered by the qualification margin.

Limitations: Not effective in demonstrating life (no way of knowing how much life remains in the flight article).

A qualification random vibration test provides confidence that the flight article will have sufficient fatigue life after acceptance testing.

Test Classification: Protoflight and Protoqualification Testing

Protoflight (NASA) and **Flightproof** (military):

- No dedicated qualification hardware.
- Each flight article is tested to environments that are higher than acceptance levels but equal to (NASA) or lower than (military) qualification levels.

Protoqualification (military):

- No dedicated qualification hardware.
- The first-built flight article is tested to environments that are higher than acceptance levels but lower than qualification levels.
- Subsequent builds are then acceptance tested.

Remember: These approaches do not demonstrate or verify fatigue life, as does traditional qualification testing.

- The flight article must have enough fatigue life after the test to make it through launch.

Risk is higher with these approaches than with qualification and acceptance testing.

To reduce the risk, put additional emphasis on fatigue analysis and life testing of development articles.

Shaker Limitations

To understand whether a particular shaker has the capability of performing a given test, you must understand its force rating and its stroke*.

* A shaker's velocity limit and maximum CG offset for test article can also be of consideration.

Force rating—the maximum force the shaker can generate (depends on amplifier power)

- Determines the amount of acceleration that can be achieved for a given mass ($F = ma$)
- The mass to be considered is not only that of the test article but also of the shaker's armature, the slip table or head expander, and the test fixture used to mount the test article.

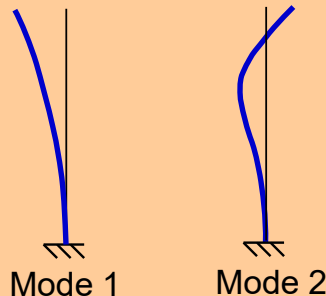
Stroke—the maximum range of displacement for the armature

- Limits the acceleration that can be achieved at a given frequency
- Section 4 addresses this limitation for sine burst testing.

Limitations of Traditional Testing on a Shaker Related to Test Effectiveness

- Introduces acceleration uniformly through the mounting interface, and energy is limited only by the shaker's force rating
 - Tends to over-stress the hardware (discussed in Sec. 6)
- One axis at a time
- Translational input only; no angular (rotational) acceleration
- Can't excite all modes of vibration from base translational input
 - Depends on the **modal effective mass** as a ratio of total mass, which tells us ...
 - how much mass relative to total mass is moving in phase in a given direction,
 - and whether (or how effectively) a mode can be driven by cyclic base motion in a given axis.
 - The mathematics are beyond the scope of this course, but modal effective mass can be computed with FEA

Which of these two modes can be more easily excited with lateral motion at the base?



Answer: Mode 1.
For Mode 2, mass is moving in opposite directions, so the net mass motion is low.

If one of the test objectives requires exercising a mode that can't be excited with a shaker, you need to find a different test!

Pass/Fail and Success Criteria

In addition to clearly defined objectives, all tests need pass/fail or success criteria.

- How will we know whether the test achieved its objectives (**success criteria**) or whether the test article passed the test (**pass/fail criteria**)?
- Criteria must be clearly defined—in advance of the test.
 - Otherwise, how will you know what to look for? How will you know you have the right instrumentation?
- However, failure to meet certain criteria does not necessarily mean test failure.
 - Pass the criteria: no-brainer; move on.
 - Fail to meet criteria: investigation and disposition required.
 - Make sure this philosophy is clear before starting the test.

For a test to be meaningful, valuable, and cost effective, it must be carefully designed to meet the objectives and enable assessment of the pass/fail or success criteria.

You must assess the criteria before tearing down the test configuration so you'll be able to do the proper investigation when needed.

Dry Running the Test

Before subjecting expensive flight hardware to vibration testing on a shaker, it may be a good idea to run the shaker through the entire test.

- Especially if the test includes anything new or challenging or if the shaker hasn't been used in a while
- Preferably with a mass simulator in place of the test article; if a mass simulator is not available, any approximate mass will do
- Even a dry run with test fixture only can be worthwhile

Why?

- Ensures the shaker is mechanically working properly
- Ensures the test environment can be properly controlled

Let's look at a case history that highlights the importance of a dry run ...

HESSI Test Mishap (Refs. 1 – 3)

High Energy Solar Spectroscopic Imager (HESSI) suffered over \$1M damage in a test mishap.

- Sine burst test, intended to input 1.88 *g* (-12 dB from full 7.5-*g* level)
- Actual input acceleration hit 21 *g*, damaging the spacecraft

What went wrong:

- The shaker was old, the slip table and supporting granite block were misaligned, and a bearing had failed.
- Input force was increased to overcome “stiction” (greater than normal static friction).
- Evidence of the problem from previous tests went unnoticed.
- No validation test (dry run) was performed beforehand.



HESSI after repair. The test in March 2000 that caused the failure was lateral, on a slip table.

To protect expensive test articles, maintain and check test equipment and ...

- **dry run the test with a mass simulator if the shaker hasn't been used recently for a similar test.**
- **pay attention to test data and noises, and pursue anomalies.**
- **do the sine burst test before vehicle integration, or consider static loads testing instead.**

Taking Responsibility for Verification

When it comes down to it, **verification** in the space industry means making yourself and other stakeholders confident that your SmallSat or other product will meet its requirements.

Vibration testing is for verification of compliance with the requirement to withstand launch environments ...

- and still function as needed—the interest of your customer.
- without failing in a way that could jeopardize the launch vehicle (LV) or its other payloads or that prevents successful separation from the LV—the interest of the launch services provider. **The LSP has no reason to care whether the SmallSat functions properly in orbit!**

Verification is the responsibility of the product developer, not the customer or the LSP. Again, the reason you're doing a vibration test should not be that your customer or the LSP requires it!

Design your test program accordingly.

**Write a test plan that makes you confident, then present it to other stakeholders along with justification, and request concurrence ...
rather than simply asking them what they want you to do.**

The appendix, at the end of this course book, provides thoughts on test plans and other forms of test documents and controls.

Key Points of This Section

- Identify and carefully define the test objectives. We do tests not because they are requirements, but to ...
 - verify compliance with true requirements
 - or obtain information.
- Take time to understand how tests relate to the mechanics of vibration.
- Recognize the limitations of testing on a shaker when designing a test program.
- Clearly define the pass/fail or success criteria before the test, and be ready to investigate violations of the criteria quickly during the test.
- Dry run the test, preferably with a mass simulator for the test article, if the test requires something new or something the shaker hasn't done recently.
- Take responsibility for verification; the LSP's motivation is different than yours or that of your customer, so don't blindly test like the LSP requires.
- But be sure to get buy-in from all stakeholders.

2. Configuration, Fixtures, and Instrumentation

- Test Fixture Design
- Test Article Configuration
- Mass Simulator Design
- Test Configuration
- Providing a Flight-like Interface
- Instrumentation and use of Accelerometers
- Strategies for Determining Base Force and Moment

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Test Fixture Design

The **test fixture** is a simple structure that adapts the test article's mounting interface to the hole pattern in the shaker's slip table and head expander.

In most cases, we want the test fixture ...

1. to be very stiff (rigid)
 - To keep from influencing the test environment and test-article response
 - Ideally, when loaded with a rigid mass equivalent to that of the test article, fundamental frequency > 2000 Hz for a random vibration test to 2000 Hz
2. yet no heavier than it needs to be
 - The shaker can exert force only up to its force rating, and some of that force accelerates the fixture; so mass of the test fixture limits the peak achievable acceleration for the test article.
3. and to provide a flight-like interface
 - to ensure proper modal response and interface load distribution

These objectives often conflict!

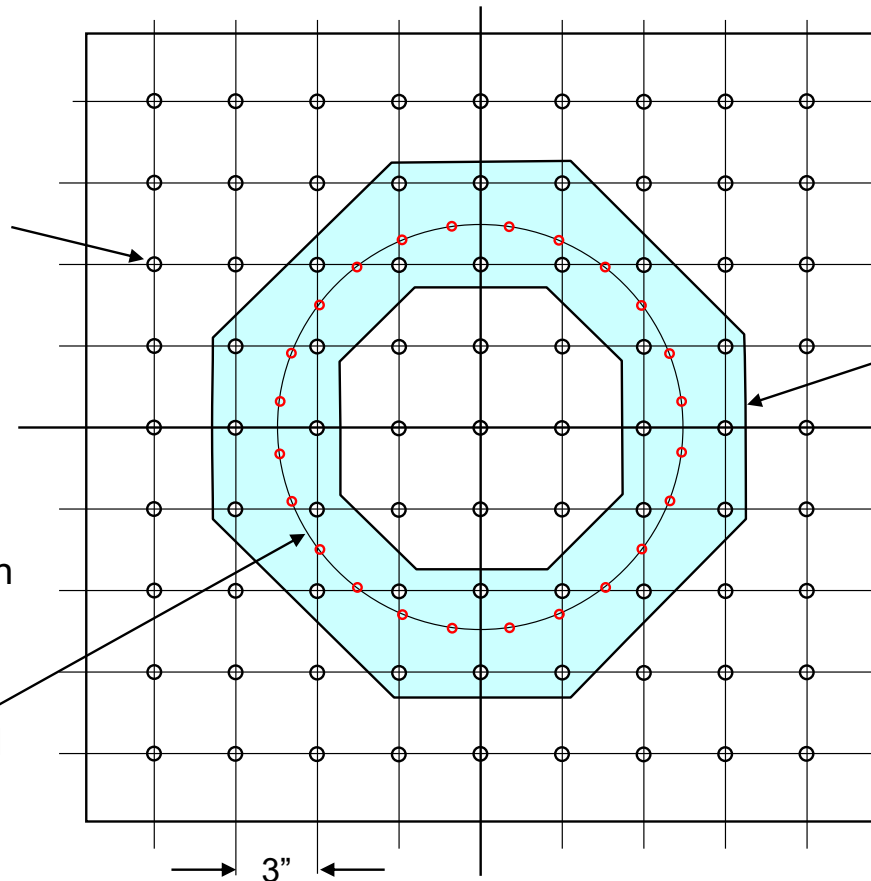
For a SmallSat, we typically go with items 1 and 2, above, and then provide a flight-like interface by testing the satellite on its separation mechanism (discussion upcoming).

Example Design of a Test Fixture for an ESPA*-Class Spacecraft

For this example:
3" grid for threaded
inserts in the slip
table and the head
expander

Note: The actual
bolt pattern
depends on the
shaker. Check with
your test facility.

15" diameter circle
of 24 evenly spaced
 $\frac{1}{4}$ " bolts for the
separation
mechanism



This fixture
footprint is efficient
in that it provides
consistently direct
load paths between
bolt patterns
without being
heavier than it
needs to be.

Make the fixture
thick so that it will
be stiff, perhaps
1.85" or 2.35",
either of which
allows a standard
stock thickness to
be planed off for
flatness.

*ESPA is EELV Secondary Payload Adapter, where EELV is Evolved Expendable Launch Vehicle.
An ESPA-class spacecraft typically weighs between 150 and 400 lb.

Test Fixture Analysis

Assuming the goal is a rigid fixture ...

- If unsure whether the test fixture will affect test results—i.e., is it stiff enough?—make a finite element model of it.
- With a FEM of the test article (SmallSat), calculate modes of vibration with and without inclusion of the fixture FEM.
- The fixture is stiff enough if it reduces the calculated fundamental frequency in each axis by no more than a couple percent.

Note that the shaker itself is not infinitely stiff and will affect the test article's modes of vibration.

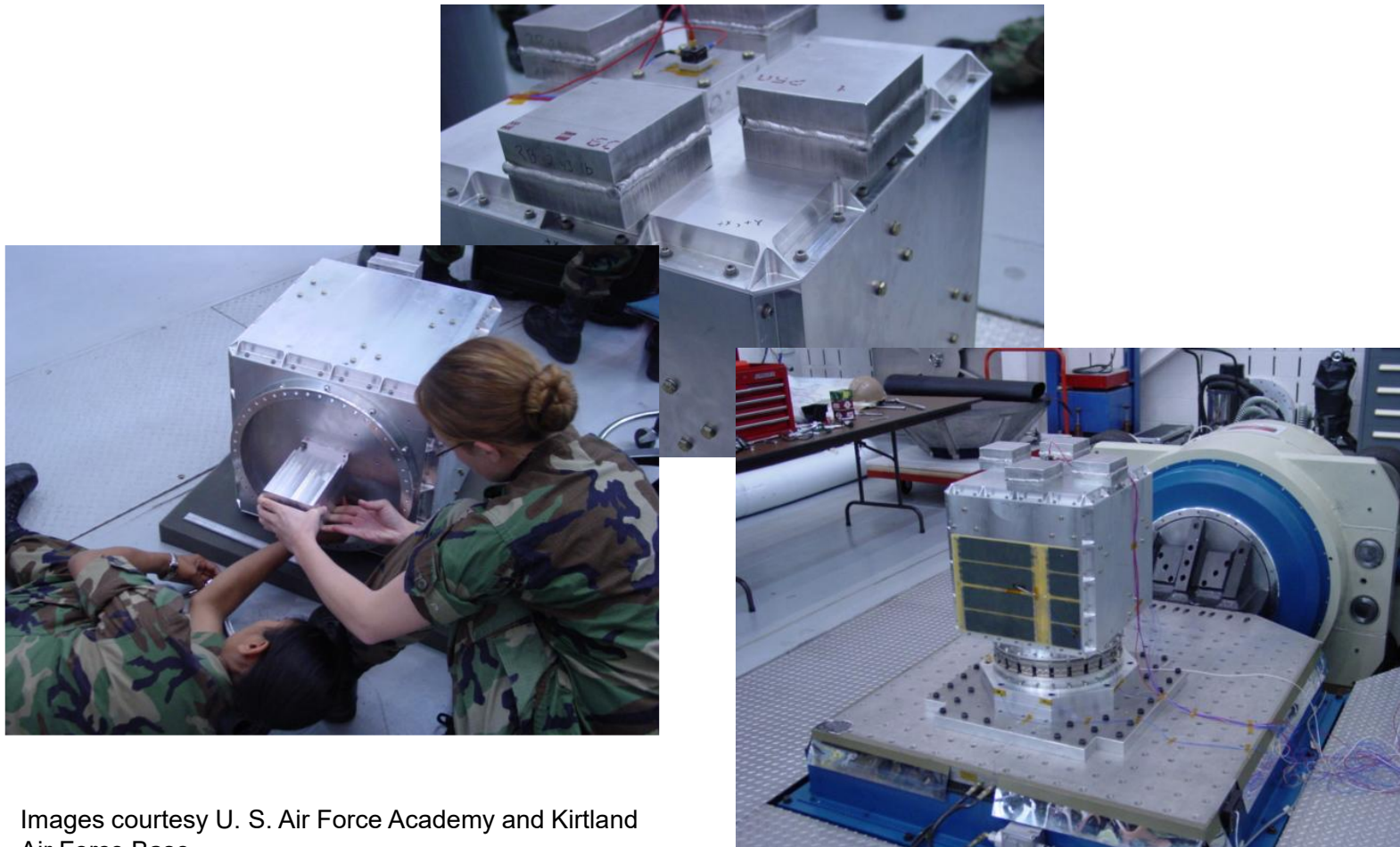
For example, on one shaker, in a 0.1 g sine sweep, a test article was found to have a fundamental lateral (rocking) frequency of 59 Hz.

When we moved the test article to a larger shaker, with a thicker slip table, again in a 0.1 g sine sweep, the rocking frequency increased to 64 Hz—an increase of 8%!

Test Article Configuration

- For the first vibration test of a spacecraft, the configuration of the test article may not be the same as the flight configuration.
 - A strategy for reducing risk is to test the structure (sine burst (or static loads) and random) before integrating the spacecraft.
 - The integrated SmallSat is tested later for random vibration only.
- To ensure a good test on a shaker, ...
 - load paths and stiffness must accurately reflect the flight configuration.
 - mass properties must be representative, with accuracy of representation depending on test objectives.
 - if the modes of vibration for a component are important to the test, that item must be present or represented with a mass and stiffness simulator.
 - Otherwise, use rigid mass simulators for missing flight items.
 - sources of damping, such as electrical cables, may be important to represent.
- The integrated spacecraft should be tested in full flight configuration.
 - And nothing should be disassembled after test unless necessary (e.g., to replace a defective component). Ensuring properly mated interfaces is a key objective of spacecraft-level testing.

FalconSAT-3 Engineering Model with Mass Simulators



Images courtesy U. S. Air Force Academy and Kirtland Air Force Base

Designing Mass Simulators

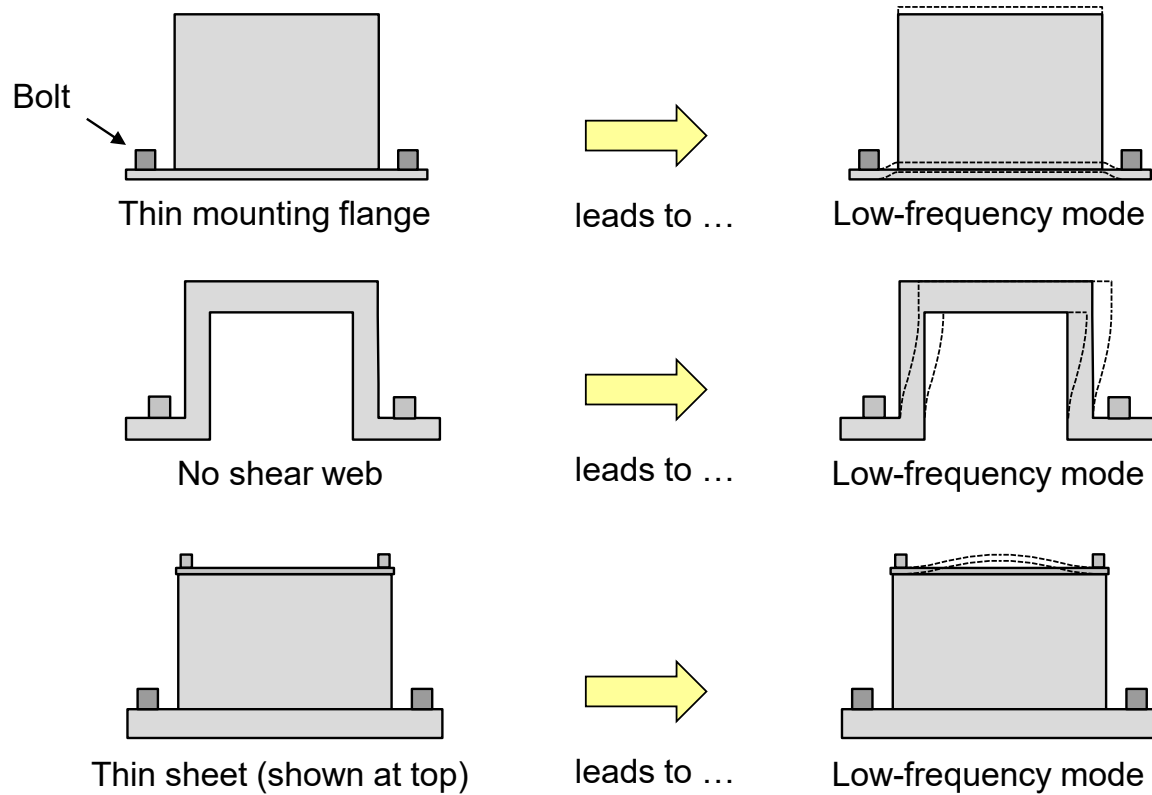
In most cases, we want mass simulators to add mass without otherwise affecting the test article's key modes of vibration.

- Which means we want them to be stiff, with no modes of vibration within the range of test frequencies.
- Which in turn means they need to have efficient load paths.

In addition to mass, it's usually important to match the flight hardware's CG and, for larger items, its mass moments of inertia (MOIs).

- Component CG and MOIs affect local (e.g., panel) modes of vibration and overall mass properties of the test article.

Example Mass-simulator Designs to Avoid



Example of a Mass Simulator that Caused Problems in Test

This mass simulator was meant to provide representative mass and mass moments of inertia, simulating a spacecraft (primary payload) on top of a rideshare structure.

Steel plates,
 $\frac{1}{4}$ " thick,
spot-welded
together as
shown



Rideshare structure

- The thin plates introduced relatively low-frequency shell modes and an uneven bolt-load distribution into the RSA.
- Result: A great deal of extra time during test and afterwards, reducing test data to meaningful information and ensuring a valid test.
 - Additional labor cost that far exceeded the savings in going with steel rather than aluminum.
- For the same weight, aluminum plates would have been nearly $\frac{3}{4}$ " thick. The shell modes would have been of much higher frequency, with no impact on the test.

Potential Exception for Mass Simulators

In some cases, a rigid mass simulator will not meet test objectives.

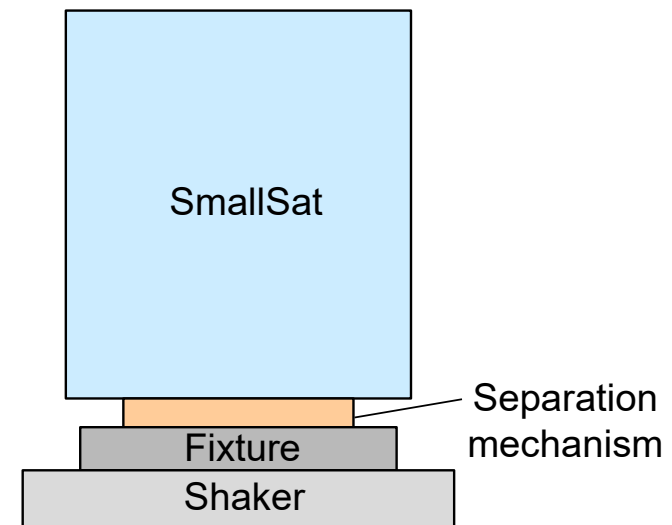
Example: Part of the structure is loaded most by modal response of a missing component.

- To adequately stress the structure, the mass simulator must also be a stiffness simulator.
- Even damping may be important to simulate.

Simulating mass, stiffness, and damping can be expensive, so try to use the actual flight component or an engineering model in such cases.

Test Configuration

- In most cases, we want to do the random vibration test with the SmallSat on its separation mechanism.
 - Provides flight-like boundary conditions
 - And often provides beneficial damping and isolation at high frequencies
- If the separation mechanism is not available, a geometry and stiffness simulator for the mechanism is the next best option.
- However, it may be better to do the sine burst test without the separation mechanism.
 - This is the case when the SmallSat's fundamental frequency in this configuration is low and the shaker doesn't have enough stroke to do a good sine-burst test (discussed in Sec. 4).



Motorized Lightband Simulator (MLBS)

- Simulates the stiffness of the Motorized Lightband (MLB) separation system for use in vibration testing of small satellites
 - Intended for structural qualification testing
- All one piece, machined out of aluminum alloy
- Basic design by Planetary Systems Corp. (now part of Rocket Lab USA), refined and tested by Instar
- Has been qualification tested (strength and fatigue life), and has test-verified stiffness
 - Somewhat stiffer than the actual MLB in order to provide longer fatigue life



15"-diameter version

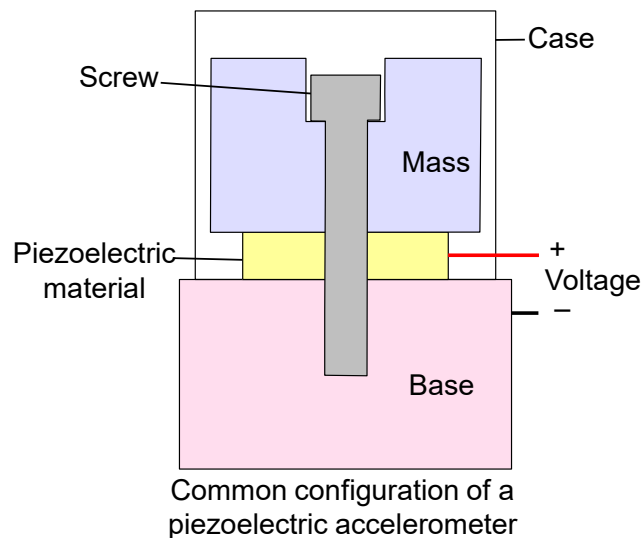
Instrumentation

- By far, the most commonly used instrument in vibration testing is an **accelerometer**:
 - An electromechanical device that measures acceleration indirectly from deformation, which in turn relates to inertia force that resists acceleration
 - Mounted to the test article and on the test fixture, close to the test article, for use in controlling the test environment
- Less commonly used is a **laser Doppler vibrometer**:
 - Uses a laser to measure acceleration of a surface without physically contacting the surface
 - Thus, the mass of the instrument doesn't affect the vibration
- Often not considered is use of **strain gages**:
 - Small devices with electrical resistance that varies linearly with length change
 - Particularly useful when strain is calibrated with force (example upcoming)

Accelerometers

Piezoelectric accelerometers are more accurate for vibration > 500 Hz than other types and are thus most commonly used in aerospace.

- They incorporate **piezoelectric materials**, which produce a voltage when dynamically deformed.
- Good for vibration but not for steady-state acceleration.



Kistler triaxial accelerometer
32 grams (1.1 ounce)



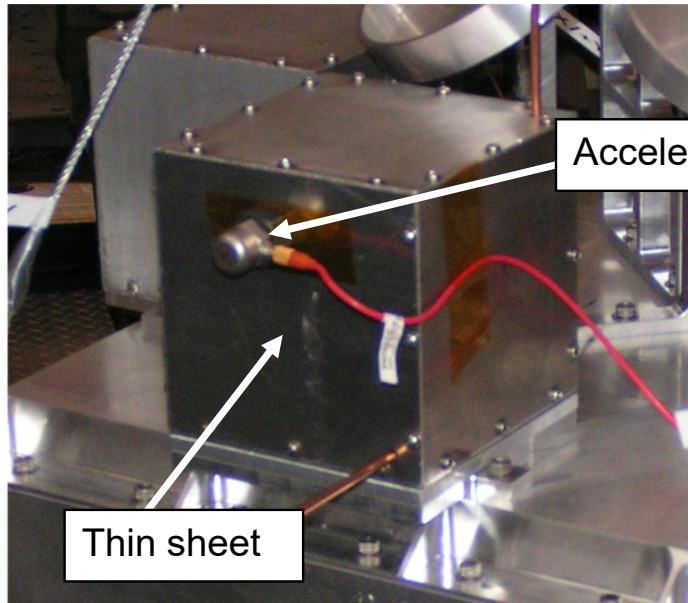
Brüel & Kjaer teardrop
(single axis)
accelerometer, 1 gram
(0.035 ounce)

Piezoresistive accelerometers, which use strain gages, can be better for shock and impact testing.

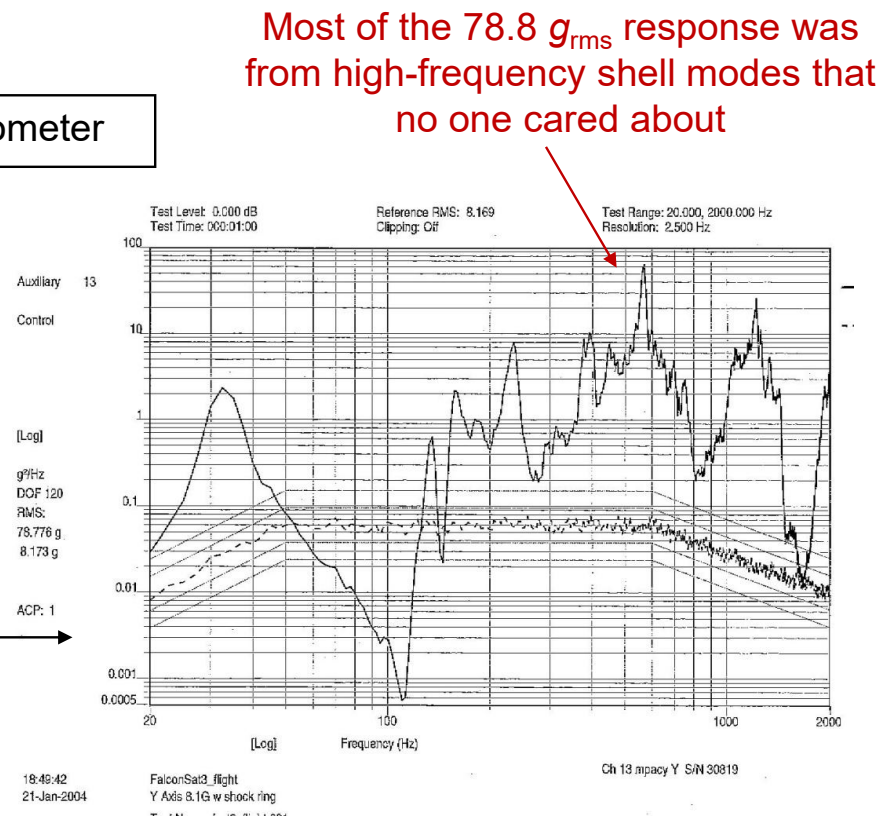
Guidelines for Where To Put Accelerometers

- Use a finite element model to guide you to locations that have the most motion for the vibration modes of interest.
 - Put them also at mounting interfaces for key components.
- For convenience, align the accelerometer's axes with the loading directions, which are usually in the axes of the test article's coordinate system.
- In locations where the mass of the accelerometer may significantly affect a vibration mode, use small teardrop accelerometers instead of tri-axial ones.
- Find locations that won't be influenced by unimportant local modes.
 - Avoid thin shells (e.g., sheet metal)
- If you want to mount the accelerometer on a thin shell in order to characterize a shell mode, you'll have two challenges:
 - If you mount the accelerometer directly to the shell, bending strain can affect the instrument's voltage and thus the measurement.
 - If you mount the accelerometer on a thick metal block bonded to the shell, the mass of the block may influence the shell mode.

Example of How Local Shell Modes Contaminate Data



Response data from random vibration test



Don't put accelerometers on thin sheets!

More Guidelines for Accelerometer Use

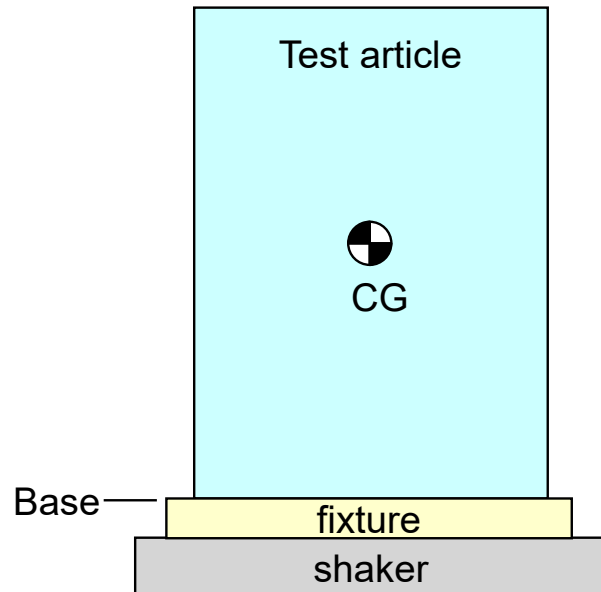
Objective	Where to put accelerometers	Comments
Determine fundamental frequency and damping	Any location predicted to have relatively high motion for the first mode, typically in axis with the excitation	Do pretest analysis to predict modes of vibration for the test configuration. See Sec. 3 for how to derive damping.
Determine mode shape	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.
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.

Even More Guidelines for Accelerometer Use

Objective	Where to put accelerometers	Comments
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.

Example Problem 2-a:
Determining Base Force and Moment during Test

Hypothetical situation:



Your small satellite will undergo vibration testing (sine sweep, sine burst, and random) on a shaker in each axis.

During test, you would like to know the overall forces and moments at the base.

Let's look at optional strategies ...

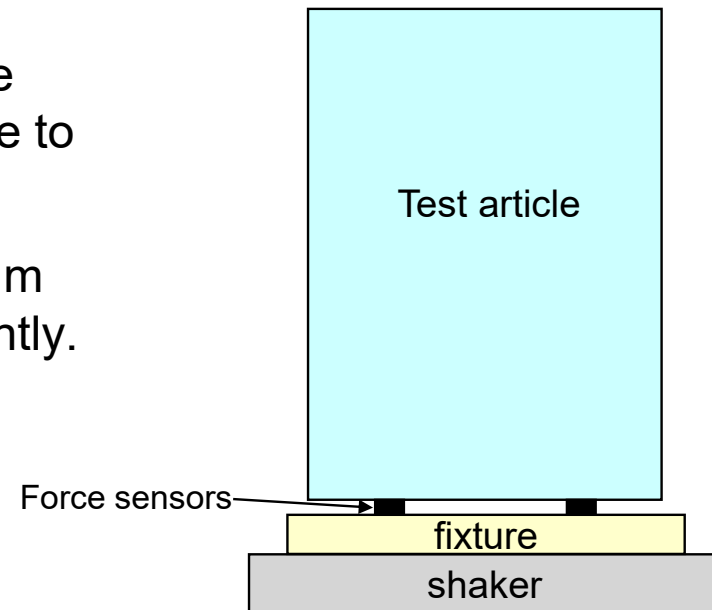
continued

Example Problem 2-a, continued:

Option 1: Use Piezoelectric Force Sensors

Use 3-axis force sensors (aka force gages) at the base of the test article to calculate forces during test.

Use a custom electronics box to sum forces and moments time-consistently.



This approach leads to accurate knowledge of loads at the mounting base.

**Downside: complicates test setup and adds cost.
The force sensors are expensive!**

We'll discuss use of force sensors more in Sec. 6.

continued

Example Problem 2-a, continued:
Option 2: Put an Accelerometer at the CG

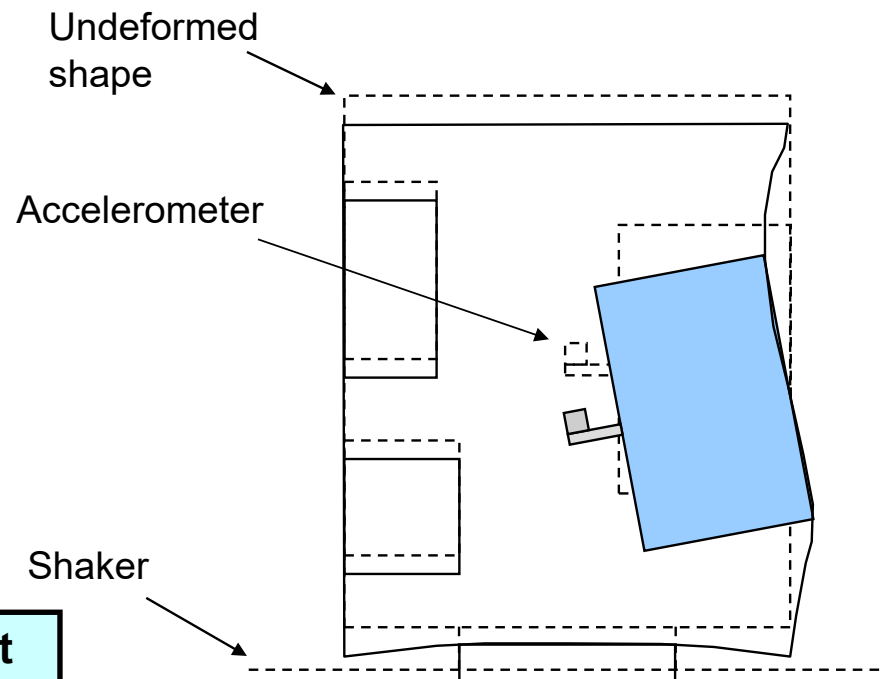
We place an accelerometer at the test article's CG by cantilevering a mounting platform off a nearby component that mounts to a side wall.

The shaker introduces base acceleration in the vertical axis.

Not a good plan.

Putting an accelerometer at the CG does NOT mean the measured acceleration will be the average or net acceleration.

Be careful how you interpret accelerometer data!

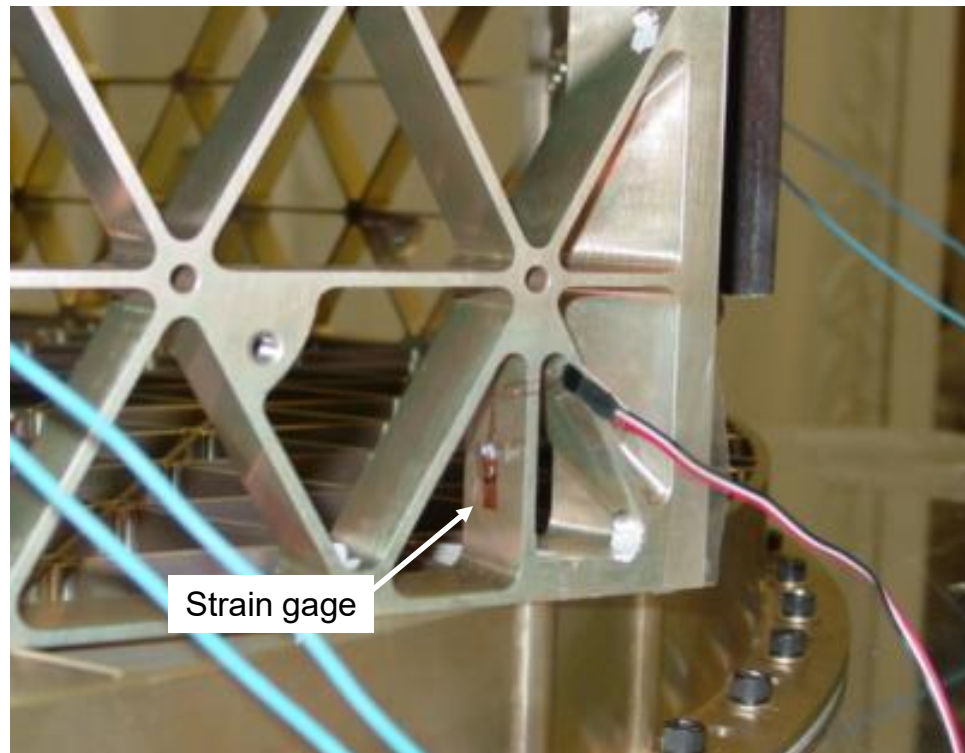


continued

Example Problem 2-a, continued:
Option 3: We May Be Able to Use Strain Gages to Measure Base Force

This strain gage was calibrated with force by applying low-frequency sinusoidal vibration with a shaker, thus achieving near-uniform acceleration throughout the test article.

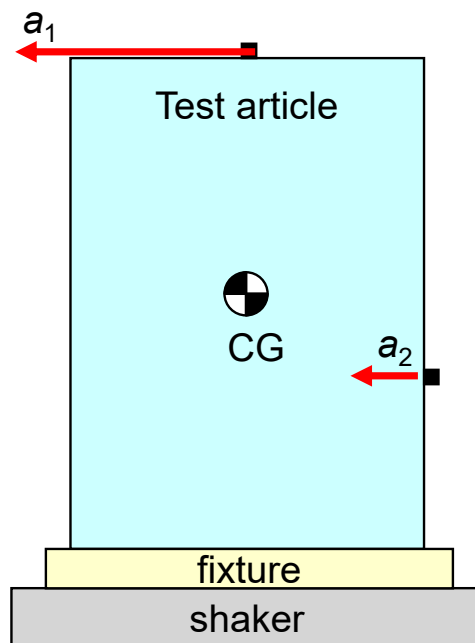
The calibration applied only for low-frequency measurements.



Strain gage

continued

Example Problem 2-a continued:
Option 4: Use FEM-derived Transfer Functions

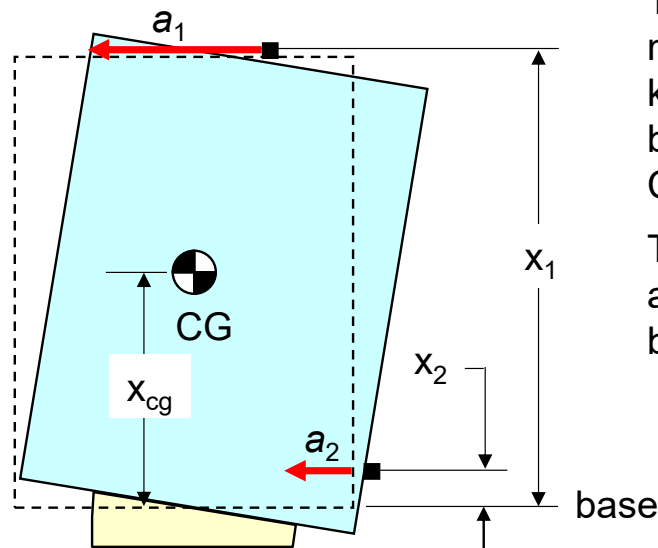


With the FEM, calculate the frequency-dependent transfer functions relating base loads to measured acceleration at one or more locations.

- This approach can give a good estimate of base force and moment.
- Depends on FEM accuracy.
- Typically useful only for the contributions of the first mode or two.

continued

Example Problem 2-a, continued: Option 4 Without Use of FEM Transfer Functions: Rocking Mode



Rocking mode: The test article rotates as a rigid body on an angular spring at the base

To convert measured accelerations a_1 and a_2 to moment at the base of the test article, M , we need to know the amount of mass, m , that is moving as a rigid body and the moment of inertia of that mass, J , at the CG.

The lateral acceleration at the CG, a_{cg} , and the angular acceleration, R , of the inertia both contribute to the base moment, M .

$$a_{cg} = a_2 + (a_1 - a_2) \frac{x_{cg} - x_2}{x_1 - x_2} \quad (\text{Eq. 2-1})$$

With a_1 and a_2 in g units,

$$R = \frac{(a_1 - a_2)g}{x_1 - x_2} \quad (\text{Eq. 2-2})$$

(units: rad/s²)

$g = 386 \text{ in/s}^2$ when x_1 and x_2 are in inches.

continued

Example Problem 2-a, conclusion:
Option 4 Without Use of FEM Transfer Functions: Rocking Mode, continued

$$\text{Base moment, } M \approx ma_{cg}x_{cg} + RJ \quad (\text{Eq. 2-3})$$

where, in U.S. customary units,

M = moment (in-lb)

m = mass (lb)

a_{cg} = acceleration at CG (as a load factor, without units)

R = angular acceleration (rad/s²)

J = MOI at CG (in-lb-s²)

Recall from Sec. 1:
When acceleration is in g units, we use weight as mass and drop the g .

This approach is intended for any case in which the first lateral mode can be approximated as rocking of a rigid body on a base spring.

Key Points of This Section

- If schedule permits, test the spacecraft structure prior to spacecraft integration, with mass simulators in place of flight components.
 - Reduces risk and allows the design team to acquire useful knowledge earlier in the program.
- Design mass simulators that don't have unwanted modes of vibration.
- Design a test fixture that ...
 - is stiff enough to not significantly affect test-article modes of vibration.
 - provides boundary conditions that are as flight-like as is practical.
 - is no heavier than it needs to be.
- Select accelerometer locations and axes based on test objectives and criteria.
- Find an effective way to measure or estimate base force and moment.

3. Low-Level Sine Sweep Testing

- Objectives and Test Parameters
- Examples of Response Data
- Deriving Damping from Test Data
- Criteria for Pre/Post Comparisons
- Common Reasons for Differences between Pre- and Post-test Data
- Limitations of Detecting Failure from Sine Sweep Tests

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Typical Objectives of Low-level Sine Sweeps

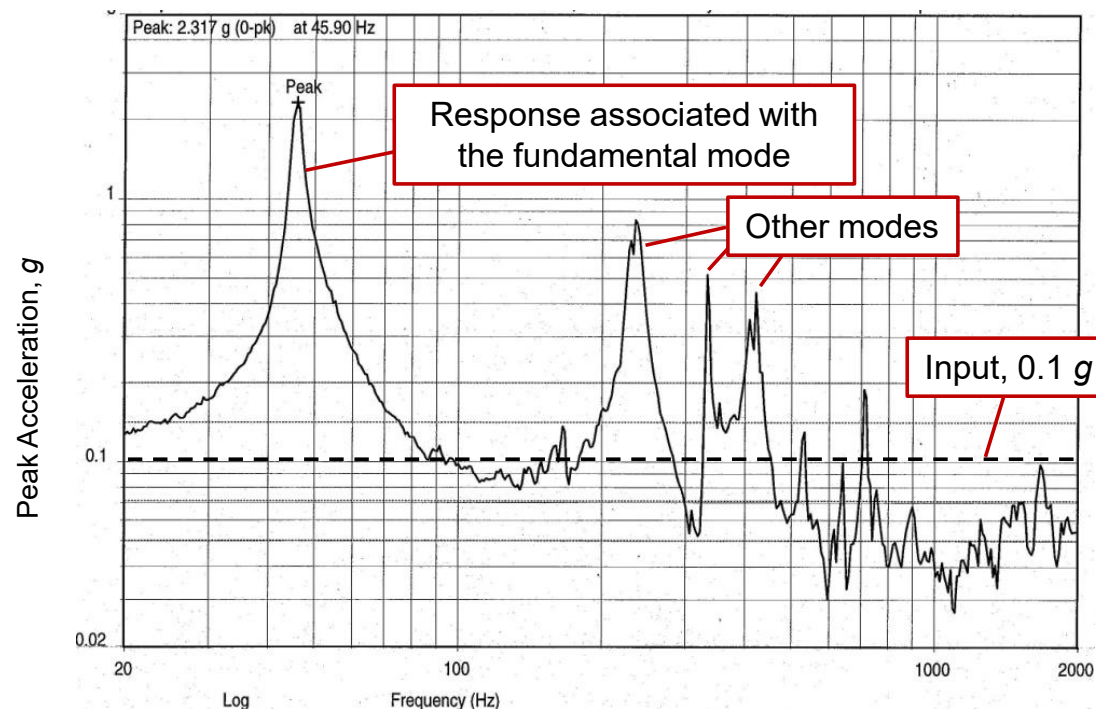
- Determine natural frequencies.
- Acquire data to help you understand mode shapes.
 - Compare relative response accelerations at various locations.
- Determine modal damping.
- Allow an assessment of structural integrity as part of the pass/fail criteria for high-level tests (e.g., random vibration and sine burst):
 - Establish baseline response characteristics before the high-level tests.
 - Repeat the sine sweep after each high-level test, and compare the signature traces with the baseline ones. There should be no significant differences.
 - Further discussed later in this section.

Parameters for Low-level Sine Sweeps

- Frequency range: xx Hz to yy Hz
 - xx no higher than 20 Hz, but should be 50% or less of the predicted fundamental frequency to account for error in the prediction.
 - yy typically 2000 Hz (same as for random vibration).
- Level (peak input acceleration)
 - Upcoming discussion.
- Sweep rate: frequency typically increases logarithmically
 - Not too fast or the test article won't hit true resonance and the peaks may not be as repeatable
 - High-Q (low damping) structures require a slower sweep rate to get good data.
 - Not too slow because you don't want to waste time
 - General recommendations:
 - 2 octaves/minute when $Q > 25$
 - 3 octaves/minute otherwise

An **octave** is a doubling of frequency.

Example of Sine Sweep Response Data

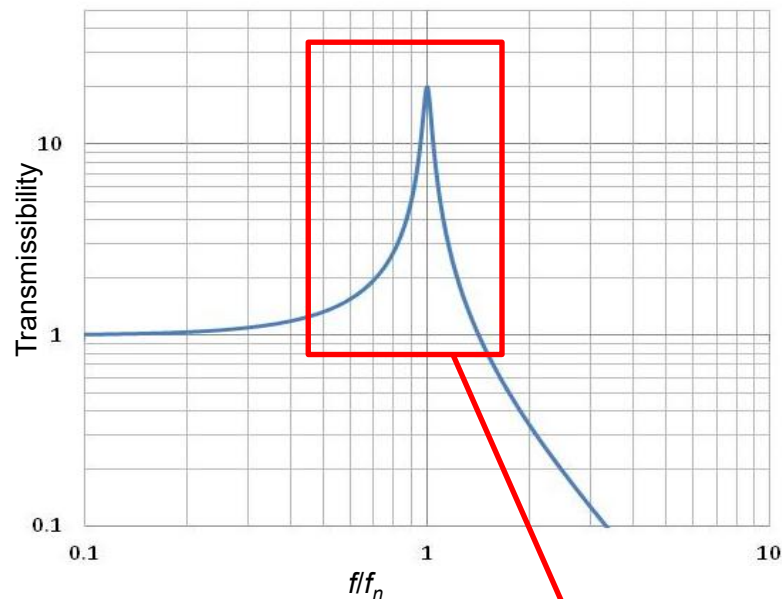


This response plot is for ***in-axis response***: response acceleration that is in the same axis as the input acceleration (e.g., Y-axis response during a Y-axis test).

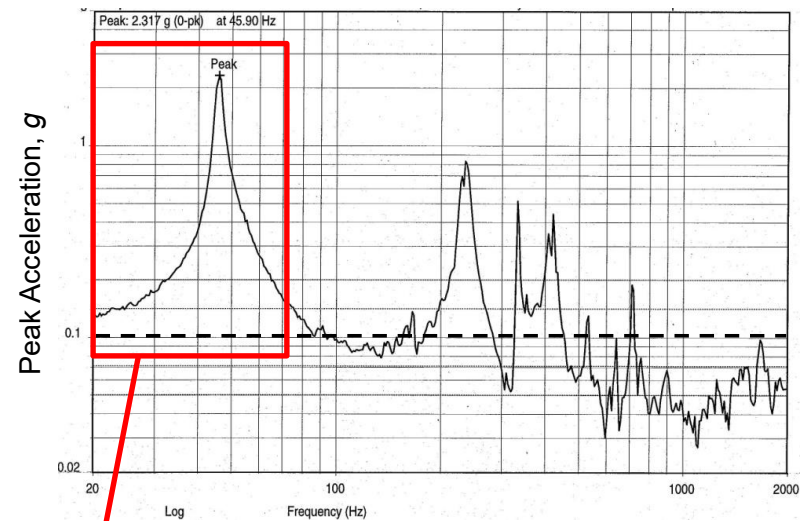
What clue tells us this?

Comparing Sine Sweep Data with the Transmissibility Plot

Transmissibility for $Q = 20$

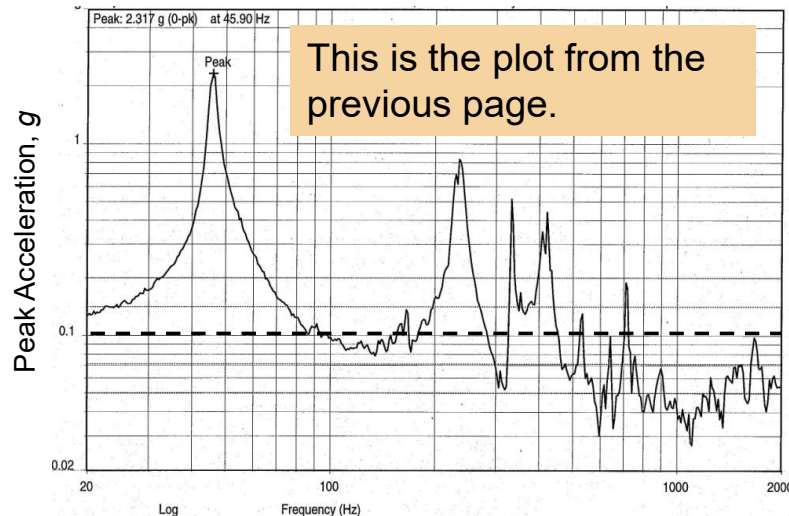


In-axis response data



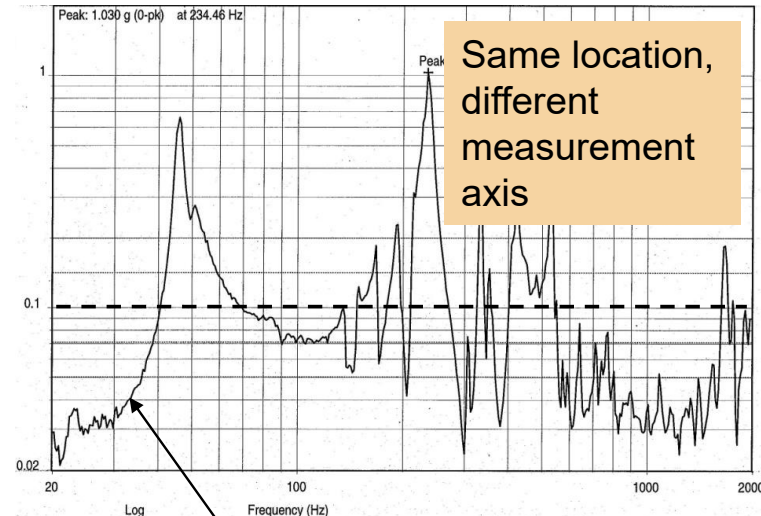
Note the similarity between the response and transmissibility plots in the red-boxed regions.

Two Response Plots from the Same Test



The above plot is for **in-axis response** (response that is in the test axis).

- At frequencies below the first response frequency, in-axis responses are at least as high as the input level.
- See transmissibility plot.

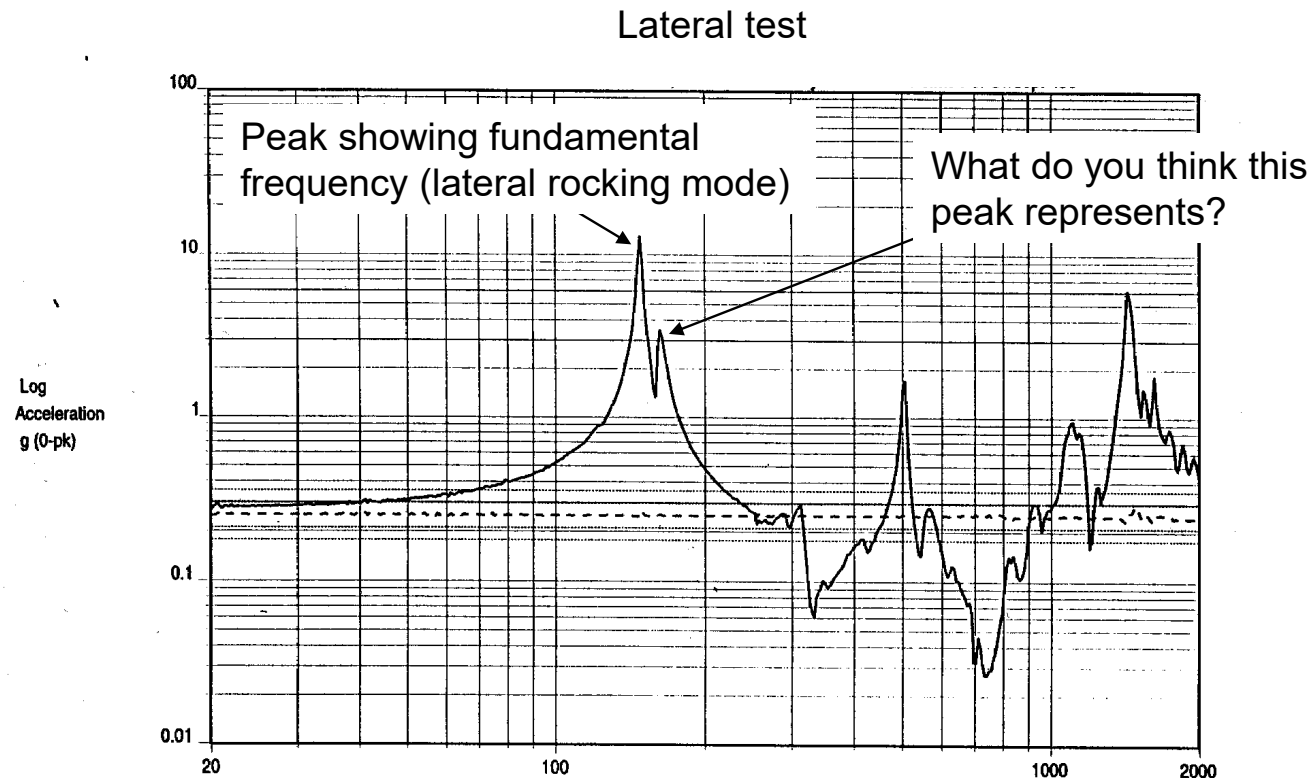


The above plot must be for **off-axis response** (response that is in a different axis than the test axis) because the response is less than the input at low frequency.

Use this principle to interpret test data and recognize when the response acceleration is mislabeled.

Note: In-axis response can be lower than input at 20 Hz only if the test article has a mode of vibration < 20 Hz.

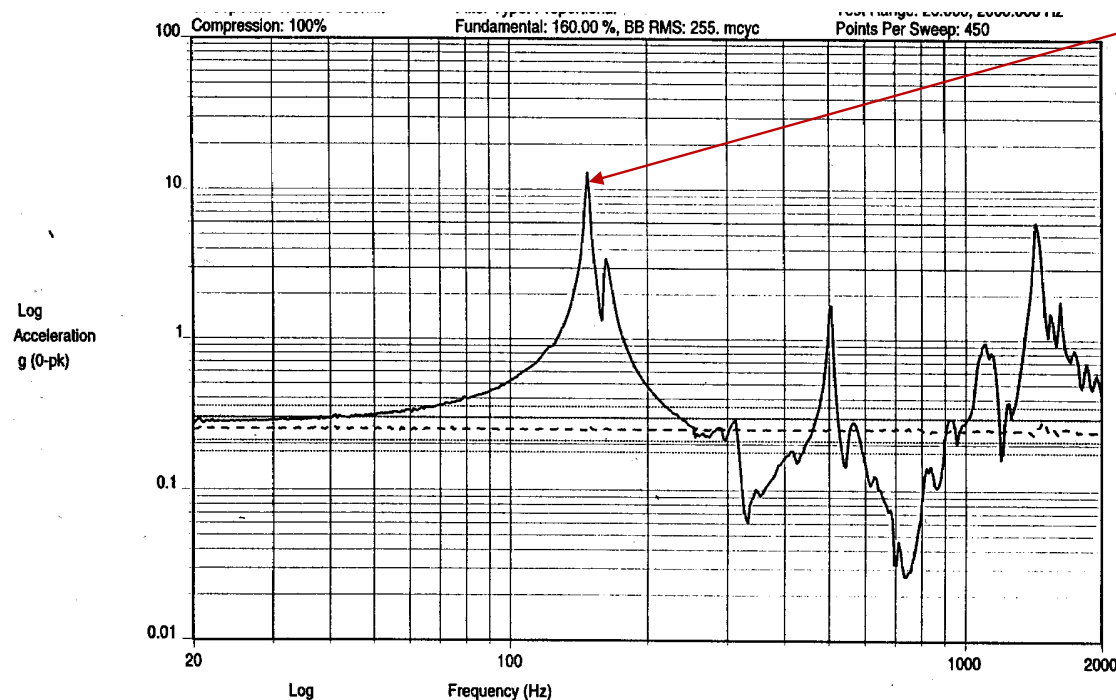
Another Example of Sine Sweep Data



Answer: It's for the second rocking mode. The test article is rocking about axes that are skewed from the test axis, so both modes are excited.

What's the Appropriate Level for a Diagnostic Sine Sweep?

This test was run at an input of 0.25 g peak acceleration (dashed line).



Notice the response of the fundamental mode: about 13 g. That's pretty high! This was a lightly damped structure.

Keep the input acceleration low (say, 0.1 g) to avoid unnecessary fatigue damage.

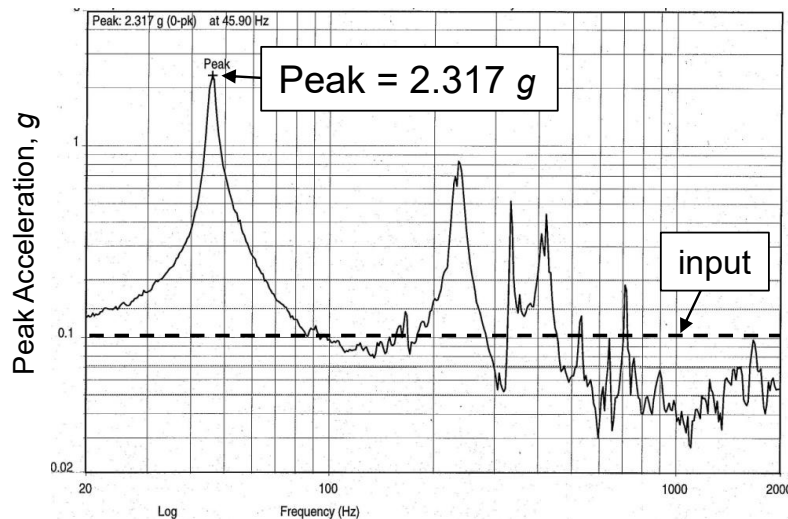
Note: Some test labs prefer to do random vibration tests at low, constant levels (white noise) instead of low-level sine sweeps. Transfer functions (response divided by input) provide the same information as a sine sweep. (Discussed in Sec. 5.)

Deriving Damping from Sine Sweep Data

Returning to a data plot shown previously in this section, let's say we want to derive damping for the fundamental mode.

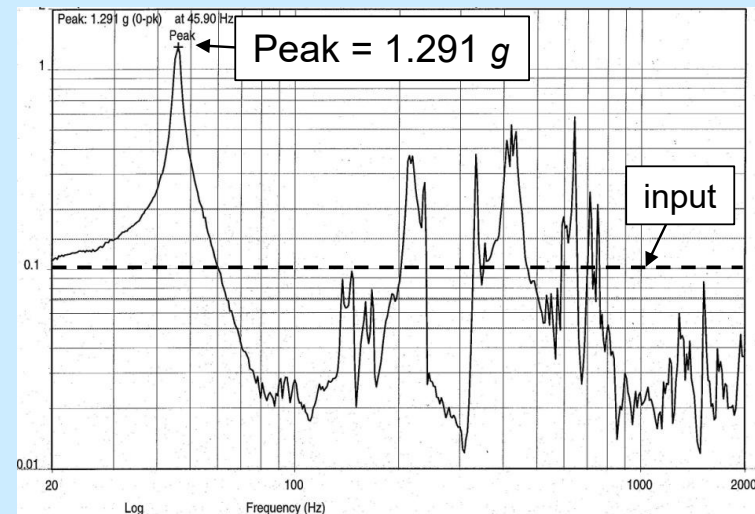
We might think that $Q = 2.317/0.1 \approx 23$

Which makes the damping ratio $\zeta \approx 1/(2Q) \approx 2.2\%$ of critical



However, the plot shown below is from the same test; in-axis response at a different location.

Now it appears $Q = 1.291/0.1 \approx 13$, and $\zeta \approx 3.9\%$ of critical



The peaks are different because of the mode shape. We can't derive damping directly from measured acceleration magnitude. (See next page.)

Example Problem 3-a: Half-Power Method of Deriving Damping from Sine Sweep Data

Peak acceleration, $a_{\max} = 2.317 \text{ g}$

Full power, $P = (2.317 \text{ g})^2$

Half power, $P/2 = (1/2)(2.317 \text{ g})^2$

Half-power acceleration,

$$a_{hp} = \frac{\sqrt{2}}{2} (a_{\max})$$

$$= 0.707(2.317) = 1.638 \text{ g}$$

$$Q \approx \frac{f_n}{\Delta f_{hp}} = \frac{45.9}{2.5} = 18 \quad (\text{Eq. 3-1})$$

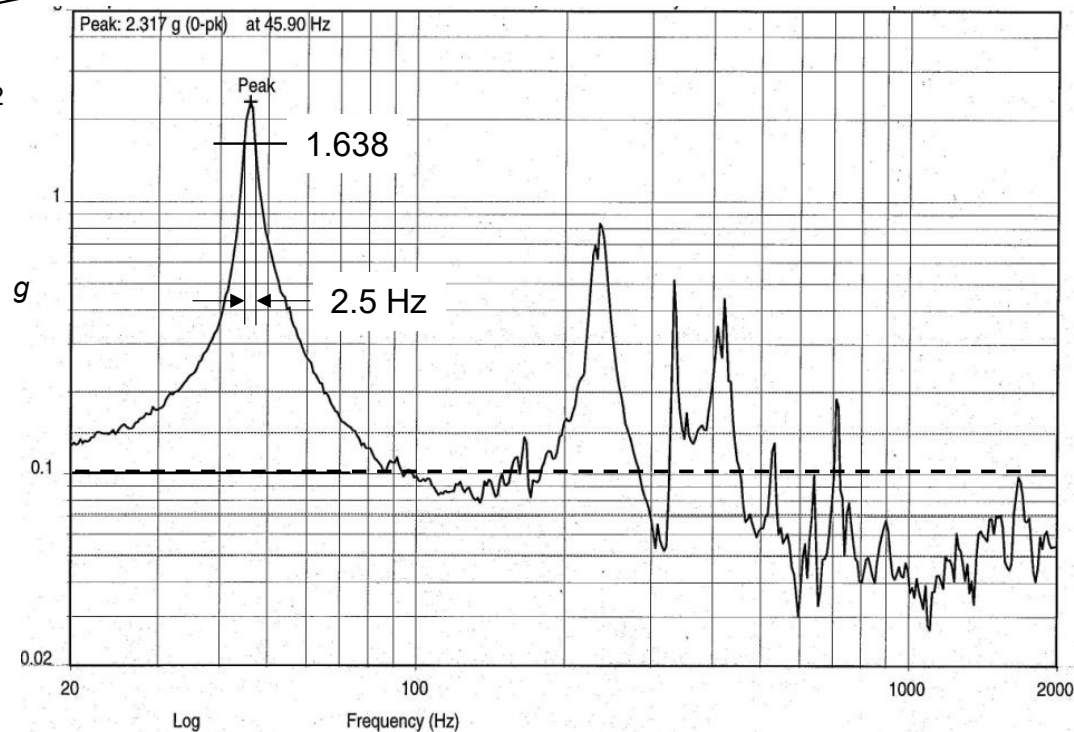
where

f_n = natural frequency

Δf_{hp} = width of plot at the half-power acceleration

$$\zeta \approx \frac{1}{2Q} = 2.7\%$$

Power in this sense is acceleration squared.



for the fundamental mode.

continued

Example Problem 3-a, continued: Half-Power Method of Deriving Damping

Now check to see if we get the same damping from the other response plot:

Maximum acceleration,

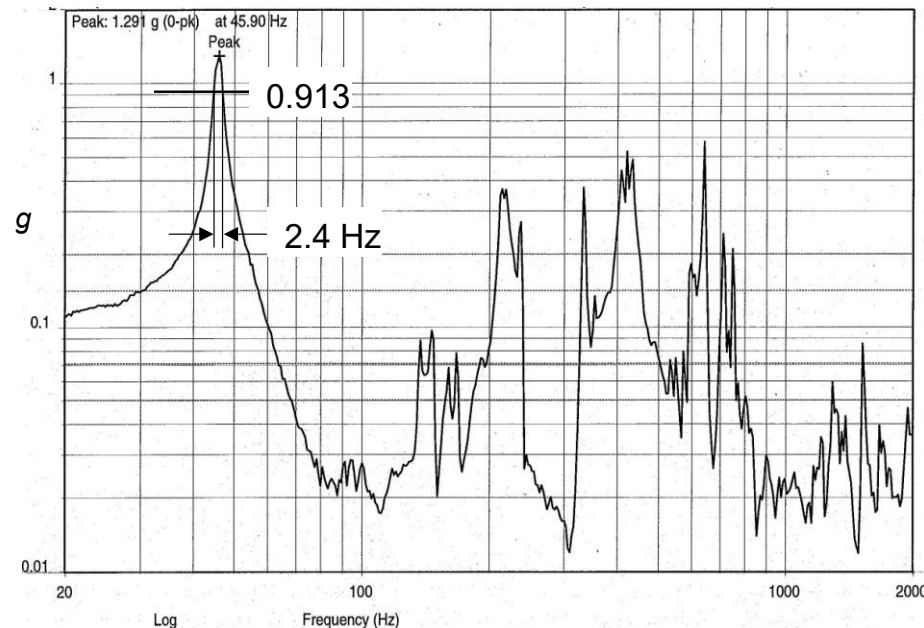
$$a_{\max} = 1.291 \text{ g}$$

Half-power acceleration,

$$a_{hp} = 0.707(1.291) = 0.913 \text{ g}$$

$$Q \approx \frac{f_n}{\Delta f_{hp}} = \frac{45.9}{2.4} = 19$$

$$\zeta \approx \frac{1}{2Q} = 2.6\%$$



← Very close to the 2.7% we derived from the previous plot.

This method also works with random vibration data, but you would use 0.5 (rather than 0.707) times the peak PSD, which is in power units (g^2/Hz). (Discussed in Sec. 5.)

Note: The half-power method of determining damping becomes less accurate for higher damping. Reference 4, NASA/TM-2016-218227 suggests using this method only when $\zeta \leq 5\%$ of critical.

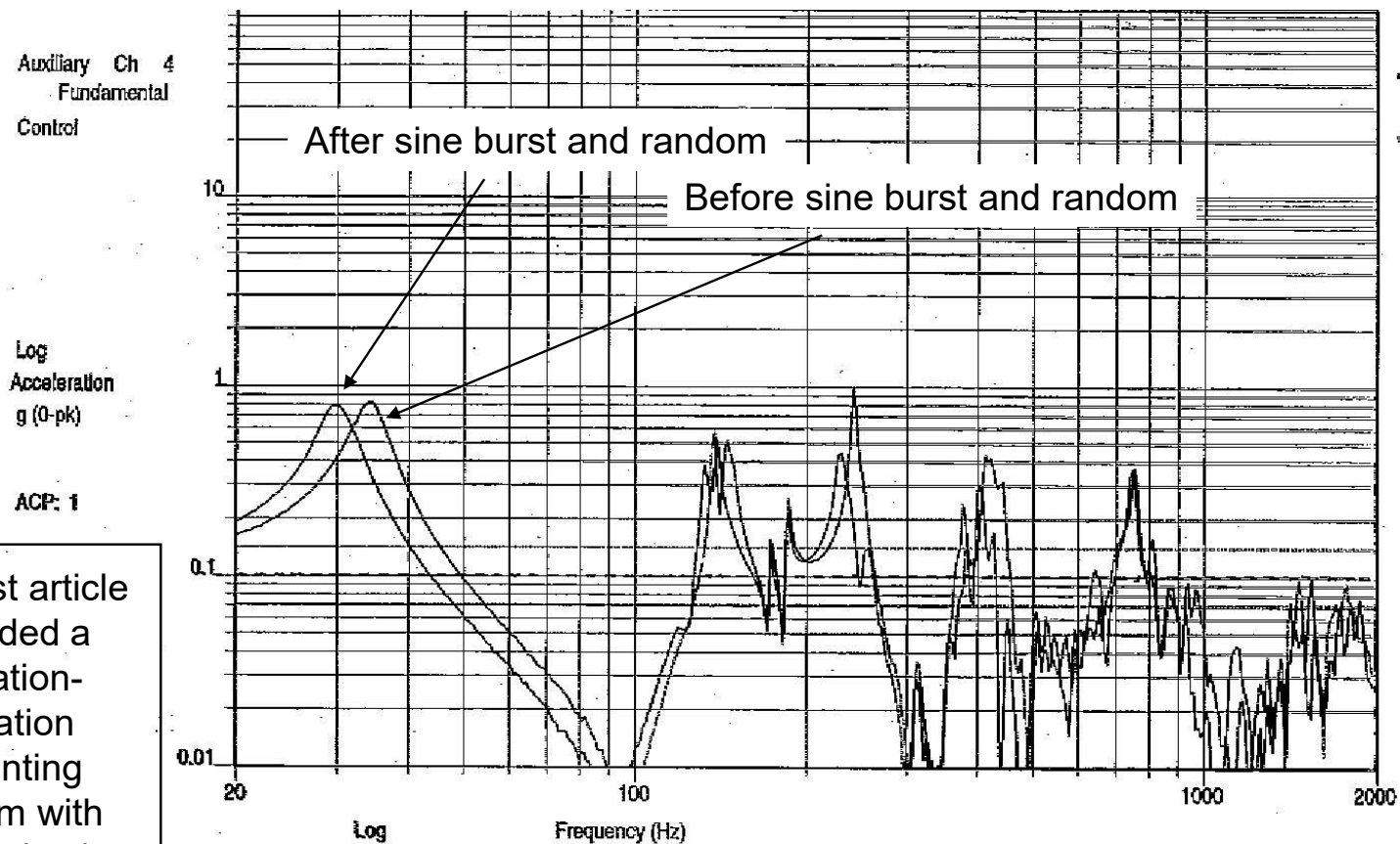
Criteria Related to Low-level Sine Sweeps

Differences in the response signature between pre- and post-test sine sweeps mean something in the structure has changed.

If the differences are significant, you must understand what has changed in order to assess the health of the hardware and whether the hardware passed the test.

- Establish criteria up front: limits that would trigger test stoppage and an investigation—perhaps 1% to 5% change in natural frequency and 25% drop in response peak, depending on the type of structure—especially for the high-mass modes of vibration.
 - Maybe 5% frequency change for structures with many bolted joints
 - As low as 1% for optical instruments
- But make sure all stakeholders agree prior to the test:
 - Failing the criteria does not mean the hardware failed the test.
 - It simply means an investigation and a disposition are required.

Example Comparison of Pre- and Post-test Sine Sweep Data



The test article included a vibration-isolation mounting system with visco-elastic materials.

What could have caused this difference?

Common Reasons for Differences between Pre- and Post-Test Data

- Loss of preload for threaded fasteners
 - Any loss of preload can cause a reduction in stiffness.
 - A small loss is to be expected and is unavoidable, despite use of most locking features, as a result of ...
 - Relative rotation of the threaded parts, caused by cyclic slip of preloaded surfaces
 - Local yielding in the threads or elsewhere
 - Differential thermal expansion under temperature change
 - When preload loss is suspected, retorque, then repeat the sine sweep.
- A fatigue crack or bond-line separation
 - If warranted, inspect for cracks with nondestructive evaluation (NDE) following the test.
- Localized yielding around fasteners that changes load paths
 - Can reduce elastic stiffness of the joint if load is not carried by friction



Torque striping to detect nut rotation in a vibration test

The right way to retorque bolts:

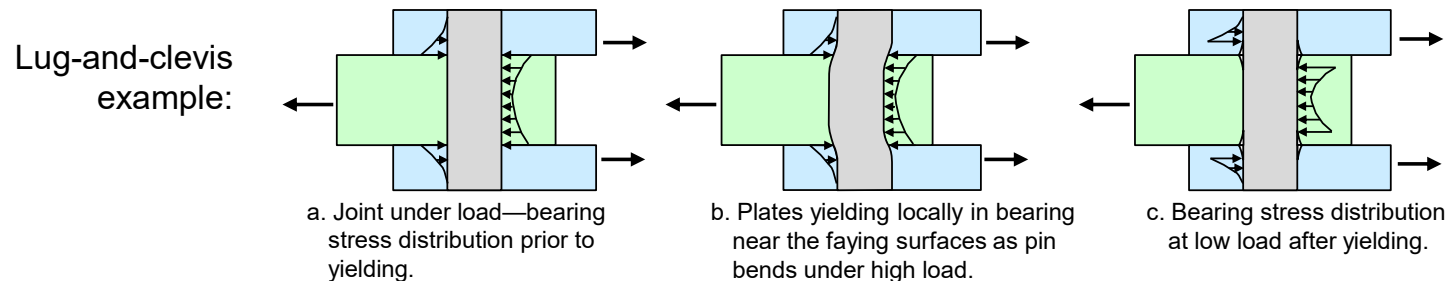
Back off with the wrench first, and then apply the full torque.

The static coefficient of friction is higher than the sliding COF.

Limitations of Detecting Yielding with Sine Sweep Tests

Structural materials can yield during high-level testing, and the structure may show no difference between pre- and post-test sine sweep data.

If yielding causes a permanent change in load paths, as does preload loss in a bolted joint, the natural frequencies at low excitation levels may change.



But if yielding causes no permanent change in load paths, natural frequencies at low excitation levels do not change.

- Yielding does not change the elastic modulus. Once the load that causes yielding is removed, the elastic modulus is the same as it originally was.
- Low-level sine sweeps typically stress materials solely within their elastic range.

If the structure must maintain alignment of component interfaces, you'll need to find some way of detecting detrimental deformation other than low-level sine sweep comparisons, such as performance testing for alignment.

Key Points of This Section

- Don't sweep at too fast a rate, or you may not get the data you need.
- Use the half-power method of deriving modal damping when the calculated damping ratio is no greater than 5% of critical.
- Make sure you understand the reason for any significant differences between pre- and post-test sine sweep results before disturbing the test configuration.
 - To flag instances that warrant investigation, establish criteria associated with comparisons of pre- and post-test data.
 - Not satisfying the criteria should trigger a test stoppage to allow investigation.
- Remember that meeting such criteria does not necessarily mean the test article didn't suffer detrimental permanent deformation.
 - Make sure you have another dependable method of ensuring there is no detrimental permanent deformation.

4. Sine Burst Testing

- Introduction
- Applicable Standards
- Do We Need a Dedicated Strength Test?
- Comparing Options for Strength Testing
- Limitations of Sine Burst Test Effectiveness
- Recommended Success Criteria
- Designing a Sine Burst Test
- Establishing the Test Axes and Deriving the Target Accelerations
- Selecting the Sine Burst Frequency
- Accounting for Dynamic Amplification
- Potential Problems and Recommendations
- Appendix: Example 4-7 (a variation of Class Problem 4-1)

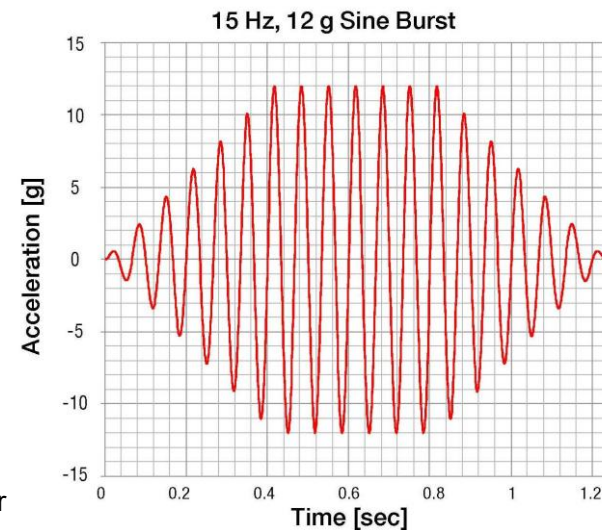
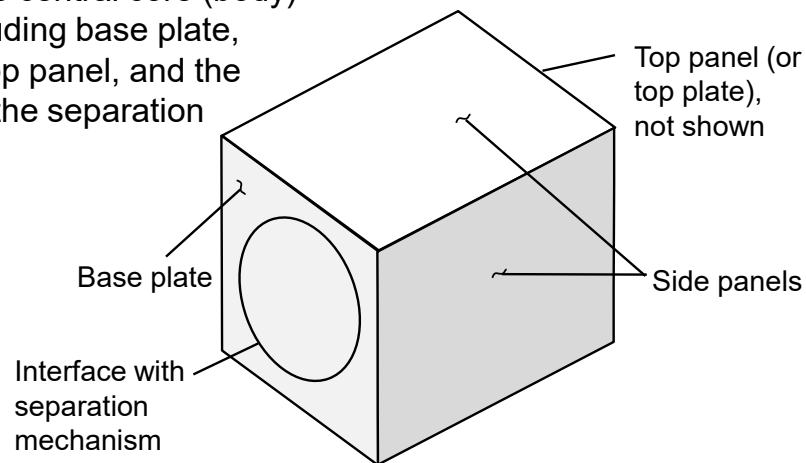
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Introduction

In a sine burst test, the shaker introduces sinusoidal acceleration at the base of the test article, with gradually increasing, then maintained, and finally decreasing amplitudes, all at constant frequency.

The objective is to verify strength of the primary structure.

The **primary structure** of a SmallSat is the central core (body) structure, including base plate, side panels, top panel, and the interface with the separation mechanism.



NASA wants to see at least 5 cycles at full load (Ref. 26). To minimize fatigue damage, don't go much beyond that.

NASA and Military Test Factors for Strength Testing (spaceflight hardware)

Test Approach	NASA	U.S. Space Force (human-rated mission)	U.S. Space Force (no humans on board)	Comments
Prototype (qualification) test	1.4	1.4	1.25	Test article structurally identical to flight article but dedicated for test
Protoflight (NASA) or Protoqualification (DOD) test	1.2	1.25	1.25	The test article is the flight structure
Proof test (combined with Prototype or Protoqual approach)	1.05	1.1	1.1	Each flight structure of certain types, materials, or joints*
Reference	NASA-STD-5001B (Ref. 5)	AIAA S-110-2005 (Ref. 6) combined with SMC-S-016 (Ref. 7), derived from the inactive DOD-HDBK-343 (Ref. 8)		

The structure should be tested to the specified or derived limit load multiplied by the applicable test factor.

*Proof testing applies to pressure vessels and structures made of materials or joints that have wide variation in strength, such as composites and adhesive bonds.

Introduction, continued

We want the frequency of the sine burst to be significantly lower than the fundamental vibration frequency of the test article in order to minimize dynamic response or amplification of the acceleration.

- In this way, we subject the test article to near-uniform acceleration.

A sine burst may be the preferred method of strength testing if ...

- the structure is going to be on a shaker for other vibration testing, and
- the structure may not be adequately stressed by the other vibration tests, and
- the potential modes of failure can be adequately screened out by cyclic loading from its base rather than by static loading (explored in coming slides).

Do We Really Need a Dedicated Strength Test?

Why can't we use the random vibration test as a strength test for our SmallSat's primary structure?

A random vibration test is just that, random. There's no way to predict or control the peak load achieved in such a test.

- Response is highly dependent on damping, which is hard to predict and nonlinear (typically more damping at higher levels of excitation)
- Unpredictable peak response load for the nominally specified test levels; will be considerably higher than 3 sigma—but may not have enough energy to cause failure that would occur in a different type of test (discussed in Sec. 5)
- Unpredictable actual test levels, given test-control tolerances

Thus, to make sure the primary structure sees the required test load, we'd have to aim for loads that are higher than required for strength verification.

- Adds risk, especially when considering the unnecessary fatigue damage introduced. Most failures caused by random vibration are fatigue failures.

Using a random vibration test as a strength test should be reserved for items that can be designed for the potential high loads and associated loading cycles, which may be the case for very small satellites.

What About Using a Sine Vibe Test as a Strength Test?

- Recall: A sine vibe test introduces sinusoidal vibration at gradually increasing or decreasing frequencies.
- Although peak response loads in a sine vibe test, when sweeping through the test article's fundamental frequency, are more predictable than for random vibe we still won't know the peak loads very well before the test.
 - Mainly because of higher damping at higher test levels, making it inaccurate to extrapolate peak loads from lower-level tests.
- So we'd again have to aim for higher loads than are required and design the primary structure for the potential peak loads, given uncertainty.

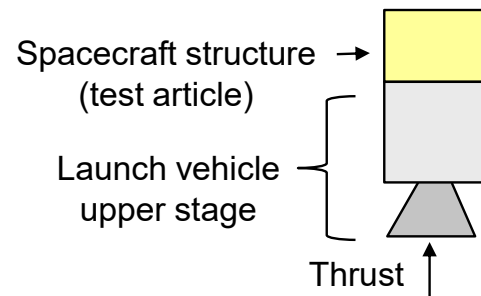
When we can't over-design the primary structure to withstand the uncertain loads associated with random vibe or sine vibe testing, we're left with the more traditional options for strength testing:

Static loads, centrifuge, or sine burst.

Let's compare them ...

At the end of this section, we'll look briefly at another option: a low-frequency sine vibe test in which the test article moves with the shaker with insignificant dynamic amplification.

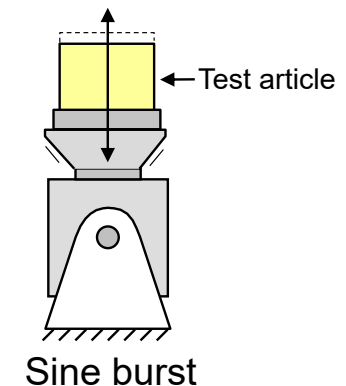
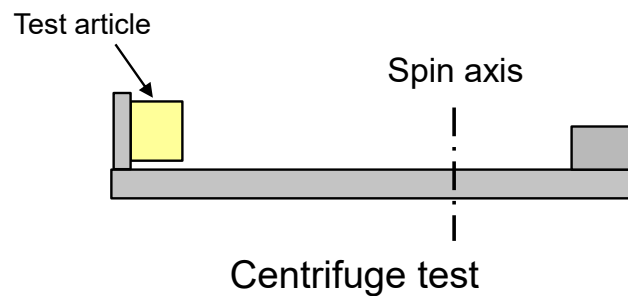
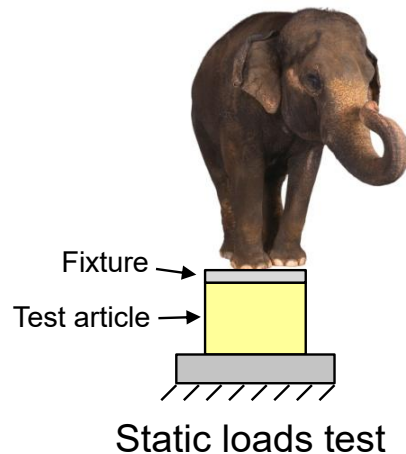
Options for Strength Testing



We want to test a spacecraft's primary structure to verify that it has enough strength to withstand the design ultimate load without rupture or collapse.

For this simple example, the limit load is specified in terms of uniform axial acceleration caused by thrust, with no lateral loads.

Options:



What are the pros and cons of each?

Comparison of Strength Test Options

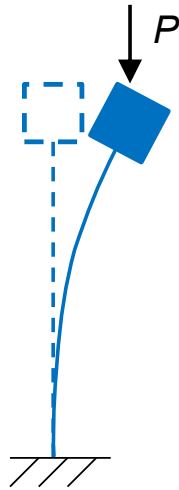
Type of test	Advantages and applications	Disadvantages and limitations
Static loads	Can simulate inertia loading from response of one or modes of vibration combined with steady-state loads. Often best for primary structures of large vehicles.	Application of concentrated loads can be more stressful than inertia loads in certain locations. Often not practical for relatively small structures, including SmallSat primary structures.
Centrifuge	The most flight-like test for structures loaded most in flight by steady-state acceleration. Good for relatively compact structures for which quasi-static loads apply.	Not the choice for large structures loaded significantly in flight by response of their modes of vibration.
Sine burst	Cost effective for items that will be on a shaker for other tests, such as random vibration. Can be quite simple (although we'd need to ensure the objectives are met, given the test is one axis at a time--discussed in this section).	Often effective for strength testing only if displacement is nearly proportional to load up to failure; not always effective when the critical mode of failure is ductile or some form of buckling. Thus, normally not a good strength test for large structures. Another downside is a sine burst is historically difficult to control (discussed later in this section).

Let's explore some of the limitations of sine burst testing ...

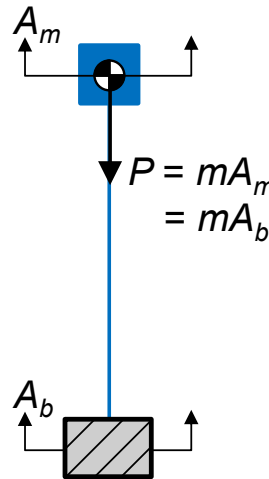
Example, Limitations of Sine Burst for Strength Verification: Column Buckling



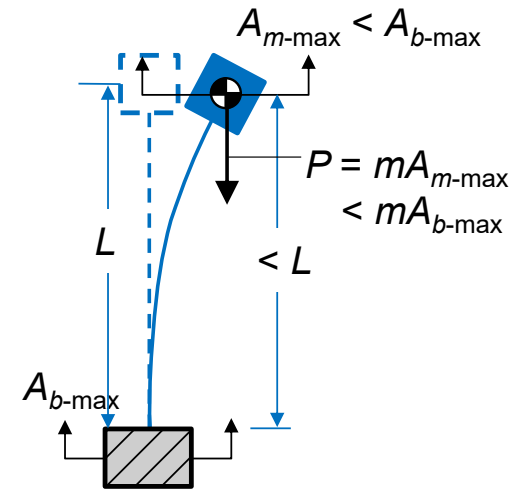
a. Static load, P , applied to mass, m .



b. Column buckles. The static load then typically causes the column to collapse.

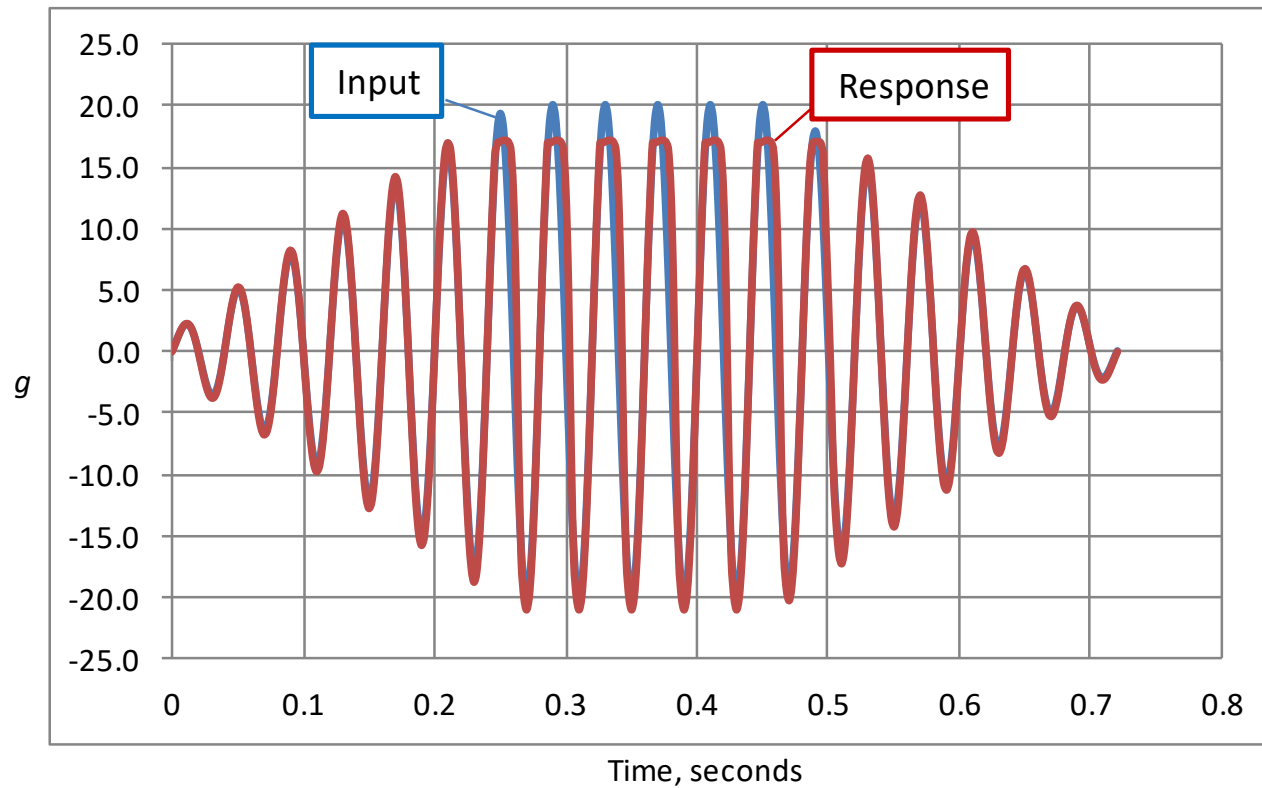


c. Column under sinusoidal base acceleration, A_b . If the column is rigid, $A_m = A_b$.



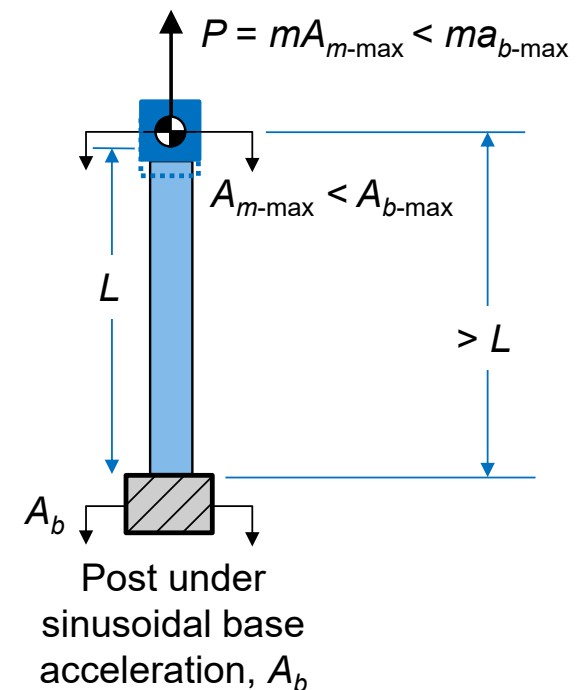
d. Column buckles. Because of relative displacement between mass and base, peak mass acceleration A_{m-max} is less than peak base acceleration A_{b-max} . Result: column may sustain the test without collapsing.

What the Response Data Might Look Like in a Sine Burst Test If the Structure Starts to Buckle



Example 4-2 , Limitations of Sine Burst for Strength Verification: Yielding

- Replace the column in the previous example with a shorter or broader “post” for which buckling is of no concern.
- The post has a constant cross section but no mass. The mass at top weighs 400 lb.
- The post is subjected to a 10 *g* sine burst test, which is predicted to cause an inertia load of $10(400) = 4000$ lb.
- The column’s ultimate tensile strength is 3800 lb.
- When the base acceleration hits 7.5 *g* and the load hits 3000 lb, the post material begins to yield in tension (for acceleration shown).
- As a result, the post stretches inelastically, and the mass doesn’t displace as much as the base and sees only 9 *g* acceleration.
- The peak inertia load is $9(400) = 3600$ lb, and the post withstands the test.



**Is the post deficient?
Is the test deficient?**

So, Is a Sine Burst Test a Valid Strength Test?

- Clearly, the structures in the two examples just given are deficient for static or quasi-static applied loads even if these structures withstand a sine burst.
- But only some of the load during a flight event is steady state, such as from constant engine thrust, and some is dynamic, such as from random vibration.
- As a result, none of these tests—static loads, sine burst, centrifuge—are truly flight-like.
- A static loads or centrifuge test tends to be more severe than launch, and a sine burst test tends to be less severe.
- As LV payloads become larger, the effects of dynamic loads on the primary structure during flight events lessen, while steady-state loads stay the same.
- For a SmallSat, a higher percentage of total flight load is dynamic.

Thus, a large spacecraft structure should be tested with static loads or centrifuge, whereas random vibration testing is more applicable for small CubeSats.

A sine burst test is somewhere in between a static loads (or centrifuge) test and a random vibration test and is most applicable for items weighing between 50 and 500 lb or so, such as many SmallSats. But, to use a sine burst as a strength test, we need some additional criteria ...

Special Considerations Related to Sine Burst Test Effectiveness

- Recapping: If the structure starts to buckle, or if materials yield, the intended load may not be achieved in a sine burst test.
- In addition, “floppy masses” (e.g., flexible cable bundles and thermal blankets) probably will not accelerate as much as the shaker does.
 - A good reason the sine burst test should be done on the structure with mass simulators rather than on an integrated flight vehicle.
- To compensate for the possibility that a sine burst test doesn’t adequately encompass the required loads, we need pass/fail criteria that are more stringent than simply ensuring the test article doesn’t break.
- For a sine burst test to do what it’s supposed to do,
 - either every part of the test article must see the intended acceleration,
 - or the target base loads (loads at the interface to the test fixture) must be achieved, as evidenced by direct measurement with force gages.
- Buckling and yielding may be evident only in the response data.

A Recommended Success Criterion

If force gages are present between test article and shaker: “Overall base loads (force and moment) shall be at least as high as the required load levels.”

If no force gages are present: “Measured acceleration at each item of mass greater than ____ shall be at least as high as the required or target levels.”

- Input and required levels should be increased somewhat to account for cables and thermal blankets, if present, which may not be moving one for one with the shaker.

If the above criterion, as applicable, isn’t met, try increasing input levels to see if the target responses are achieved.

If not, something may be yielding or buckling, and you should stop the test to investigate.

Designing a Sine Burst Test

Common problem:

- Limit quasi-static loads are typically specified as load cases, with acceleration acting in two or three axes simultaneously (and sometimes with angular acceleration as well).
- Whereas a sine burst test is performed one axis at a time.
- How can we be sure the test meets its objective of verifying strength prior to launch?

It's the responsibility of the SmallSat developer to design a sine burst test that meets the objectives.

Three parameters must be defined:

- Test axis
- Input acceleration
- Frequency of the sinusoidal input acceleration

Process:

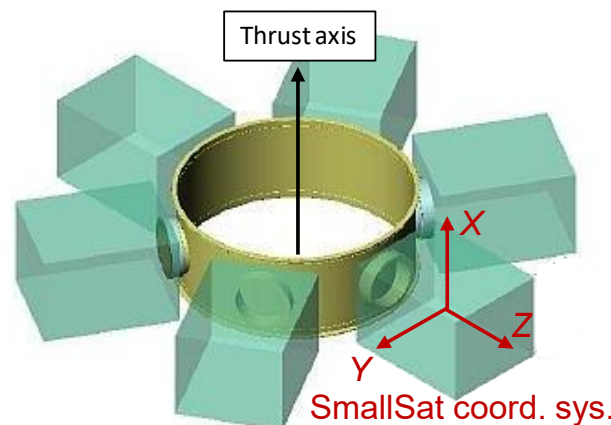
- 1. Establish the test axes relative to the test article's coordinate system.**
- 2. For each test axis, derive the target acceleration for the test article.**
- 3. Select the sine burst frequency.**
- 4. Adjust the input acceleration to account for dynamic amplification.**

Let's look at an example ...

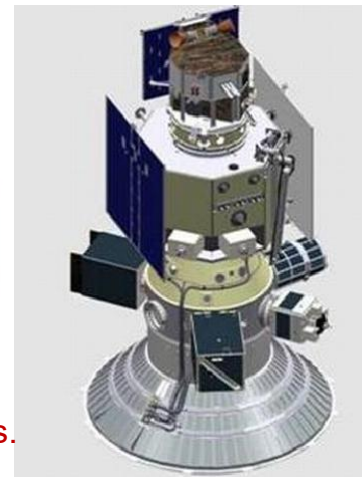
Establishing Test Axis and Target Acceleration

Example: The 2007 STP-1 rideshare mission

- Orbital Express was the primary payload.
- There were 5 auxiliary payloads (SmallSats) on the ESPA*.



A. ESPA with Six Auxiliary Payloads

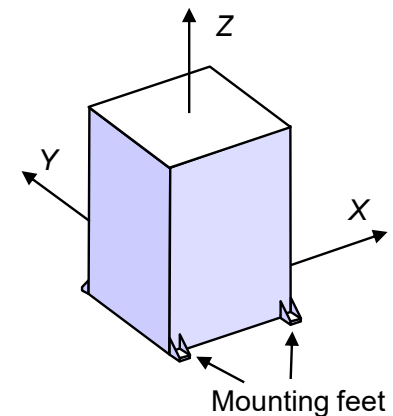


B. STP-1 Configuration

*ESPA is EELV Secondary Payload Adapter, where EELV is Evolved Expendable Launch Vehicle

Example Problem 4-a: Deriving the Target Acceleration for a Sine Burst

- You're planning a sine burst test for a SmallSat that will be one of the ESPA payloads on the STP-1 rideshare mission.
 - Let's simplify this example by assuming the SmallSat is mounted to the ESPA with four bolts in a square bolt pattern, rather than the actual 24 bolts in a circular pattern.
- The specified limit quasi-static loads are as follows:
 - $\pm 8.5\text{ g}$ in the LV thrust axis (SmallSat X axis)
 - acting simultaneously with 8.5 g in either of the SmallSat's Y or Z axis
- Must test to 1.25 times the limit loads to verify strength



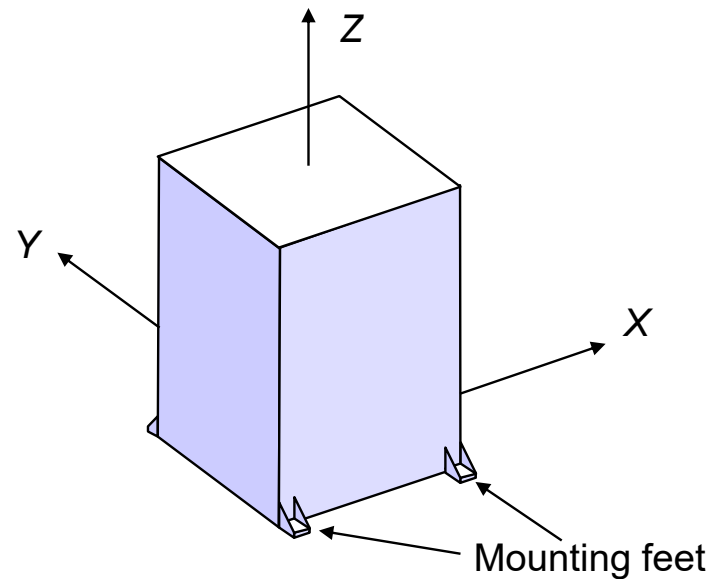
Problem statement:

Decide how to perform this test: How should the test article be oriented on the shaker, and what acceleration needs to be achieved?

continued

Example Problem 4-a, continued: Option 1

Test in each of the three orthogonal axes shown to 1.25 times $8.5\text{ g} = 10.6\text{ g}$, one axis at a time.



Option 1 clearly would not adequately load or stress the structure because the specified limit loads are in two axes simultaneously.

continued

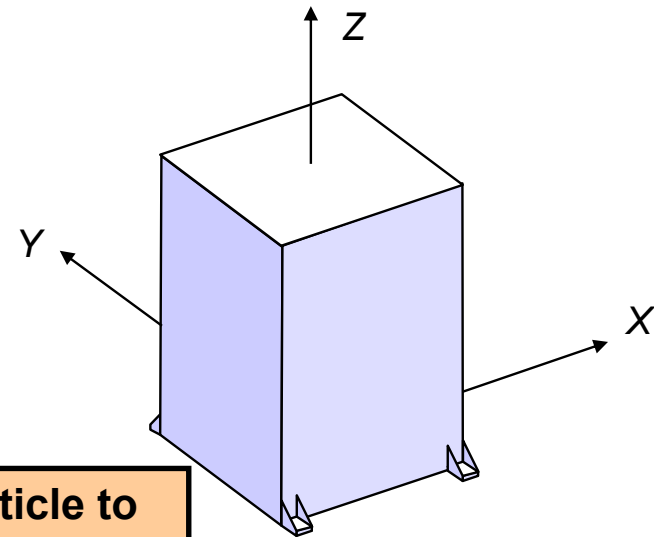
Example Problem 4-a, continued: Option 2

RSS the loads:

$$\sqrt{(8.5)^2 + (8.5)^2} = 12.0 g$$

Multiply the resultant by 1.25:

$$12.0(1.25) = 15.0 g$$



**In what direction(s) should we load the test article to 15 g in order to ensure an adequate strength test?
Would it be adequate to test separately in X, Y, and Z?**

Answer: No. To ensure an adequate strength test at this acceleration, we would have to build a special fixture that enables orientation of the test article such that the resultant test accelerations are in all directions of the resultant design-load vectors.

continued

Example Problem 4-a, continued: Option 3

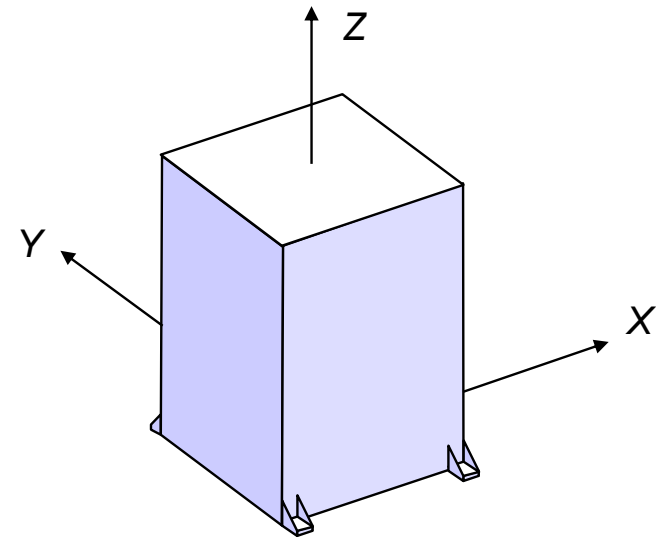
Add the accelerations:

$$8.5 + 8.5 = 17.0g$$

Multiply the sum by the 1.25 test factor:

$$17.0(1.25) = 21.3g$$

Test to this level in each of the X, Y, and Z axes.



Such an approach would adequately load the structure.

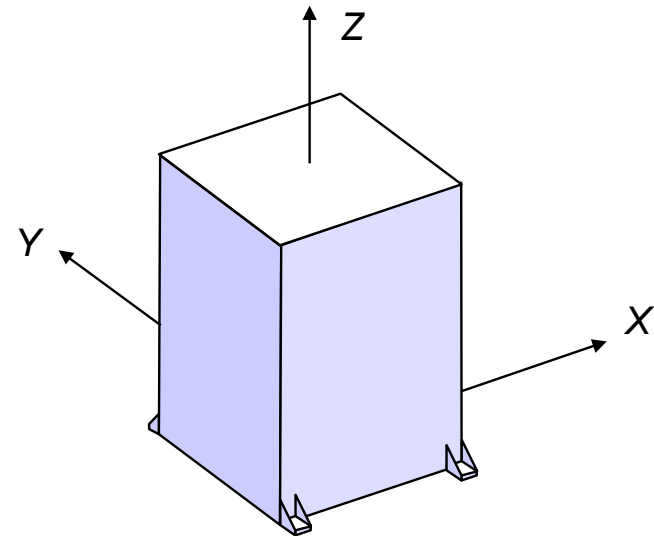
But most of the structure would be excessively loaded.

This option may be acceptable if weight is not critical and the structure is designed for the test loads rather than the specified loads.

continued

Example Problem 4-a, conclusion: Option 4

- Identify all the structure's potentially critical modes of failure targeted by the test along with the **indicator parameters**—such as stress at a particular location or bolt load—that best relate to those failure modes.
- Calculate values for the indicator parameters for the specified limit loads. These values, when multiplied by the test factor, become the **target loads** for designing the test.
- Identify the test axis that would most efficiently achieve the target load for each of the indicator parameters.
- Derive the acceleration required in each axis to collectively achieve all the target loads, using the same analysis method or math model as was used in Step b, above (apples to apples). Remember to account for the 1 g bias from gravity, if significant.

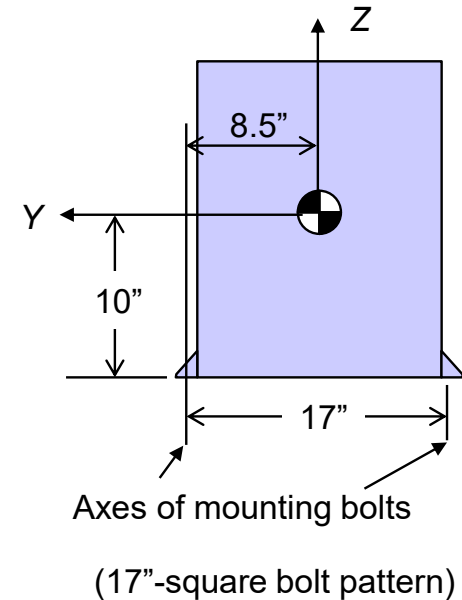


This is an example of how to design a test to meet its objectives.

Class Problem 4-1

Given:

- The test article is mounted to the test fixture with four bolts, one at each corner of a 17" square pattern.
- The CG is 10" above the plane of the bolt pattern and centered on the Z axis.
- The test article (flight article) weighs 150 lb.
- The specified quasi-static limit loads are $\pm 8.5\text{ g}$ in the X axis simultaneous with 8.5 g in either Y or Z.*
- Sine burst tests will be run in the X, Y, and Z directions.
- The structure must be tested to 1.25 times limit.
- Assume tension in the mounting bolts is the only indicator parameter being tracked to ensure adequate test loads.



Problem statement:

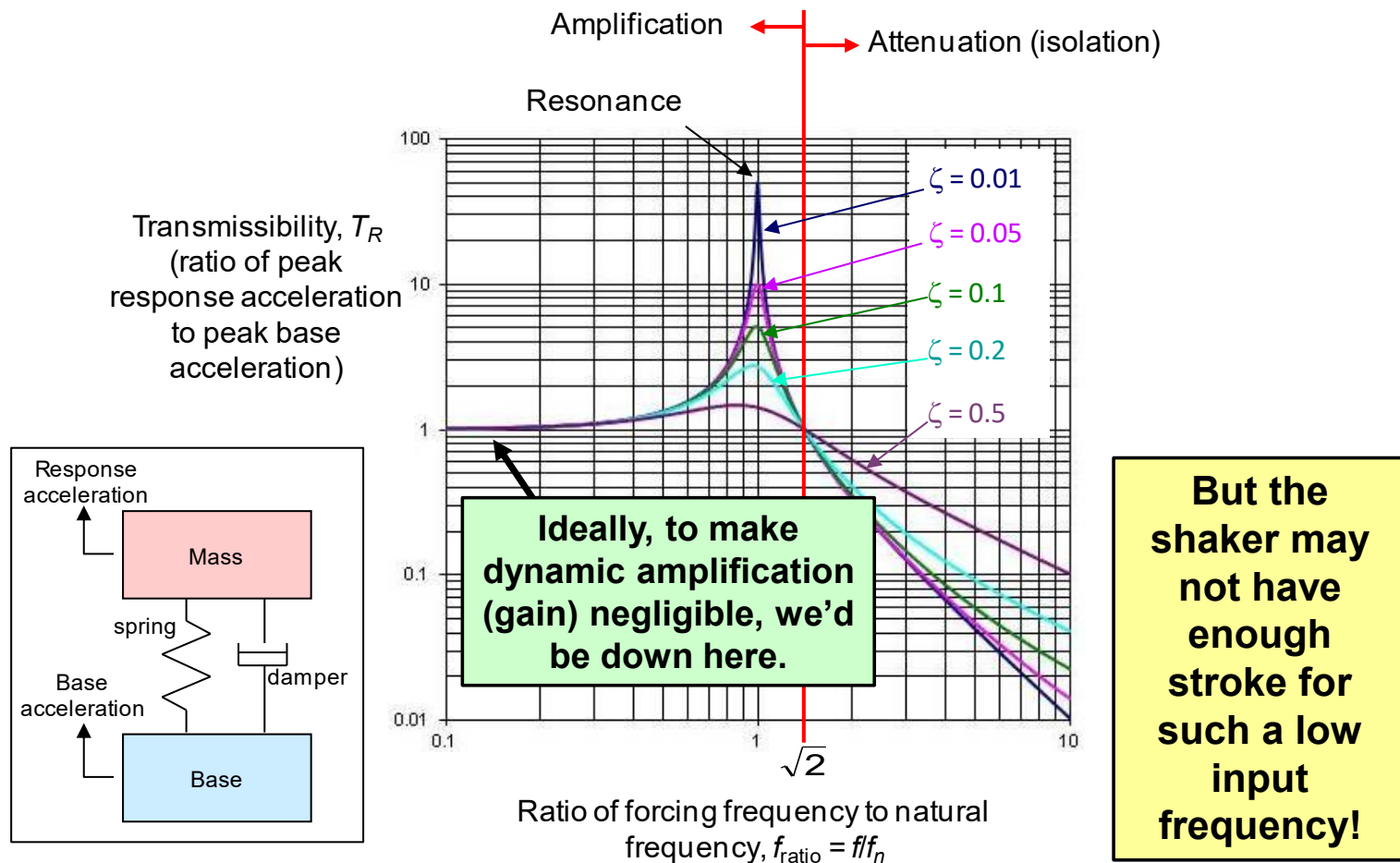
Using Option 4 on the previous page, select the test axis (X, Y, or Z) that would most efficiently load the mounting bolts in tension, then derive the required acceleration for sine burst testing in that axis to ensure the bolts are adequately tested in tension.

*See the appendix of this section for the case in which the thrust-axis load acts simultaneously with a load in any lateral axis of the LV (any axis within the spacecraft Y-Z plane).

Process for Deriving the Target Acceleration in Each Test Axis (Option 4 from Example Problem 4-a)

Step	What we did in Class Problem 4-1
a. Identify all the potentially critical modes of failure in the structure, along with the indicator parameters that relate to them.	We used bolt tensile load (discounting preload) as an indicator parameter for certain failure modes that could occur in the joint. To ensure you fully test the structure, identify indicator parameters at multiple locations within the structure, corresponding to different potential failure modes.
b. Calculate values for the indicator parameters for the specified limit loads. These values, when multiplied by the test factor, become the target loads for designing the test.	We calculated 938 lb as the target bolt load for test, including the 1.25 test factor.
c. Identify the test axis that would most efficiently achieve the target load for each of the indicator parameters.	We identified either of the lateral axes, X or Y. A simple check shows that loading in these axes is more severe for the bolts than loading in the Z axis.
d. Derive the acceleration required in each axis to achieve all the target loads. Remember to account for the 1 g bias from gravity, if significant.	We derived an acceleration of 21.3 g, including the 1.25 test factor but ignoring the 1 g bias in Z. To negate the 1 g effects, with a target bolt tensile load of 938 lb, we calculated a required acceleration in X or Y of 22.1 g.

Selecting the Sine Burst Frequency



The Relationship Between Displacement and Acceleration for Sinusoidal Translation of an Undamped Linear System

Recall from Sec. 1:

Displacement, $\delta = x(t) = \delta_{\max} \sin(\omega t)$ (Eq. 1-5)

Velocity, $V = \dot{x}(t) = \omega \delta_{\max} \cos(\omega t)$ (Eq. 1-6)

Acceleration, $A = \ddot{x}(t) = -\omega^2 \delta_{\max} \sin(\omega t)$ (Eq. 1-7)

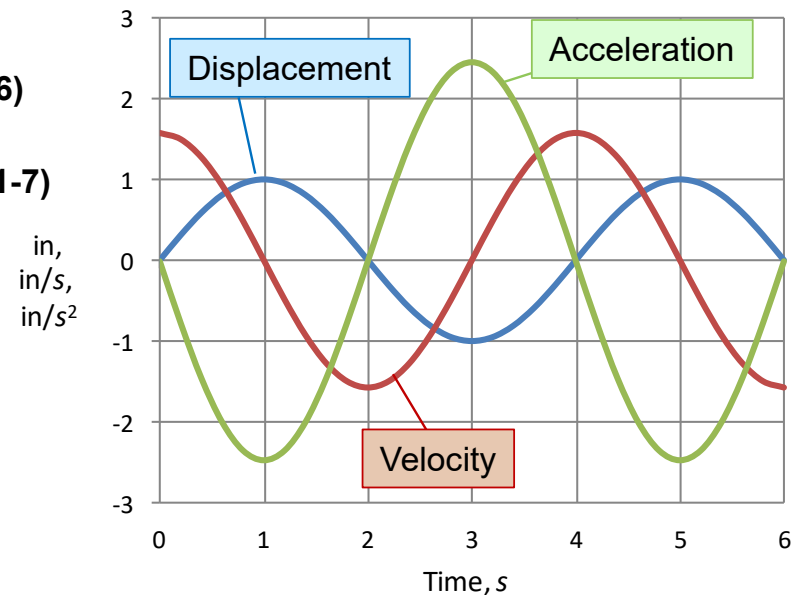
where

$\delta_{\max}$ is peak displacement

t is time

ω is frequency in radians per second

Example: 1" peak displacement at 0.25 Hz



Thus, the maximum acceleration, $A_{\max} = \omega^2 \delta_{\max} = (2\pi f)^2 \delta_{\max}$ (Eq. 4-1)

frequency in Hz

Input Acceleration is Limited by Shaker Stroke

To provide some margin, we should count on only about 80%* of the theoretical or advertised stroke. Thus, from Eq. 4-1, ...

$$\text{Test frequency, } f \geq \frac{1}{2\pi} \sqrt{\frac{A_{\max}}{0.8\delta_{\max}}} \quad (\text{Eq. 4-2})$$

*Check with lab personnel

- Example:**
- Shaker stroke, $S = 2''$, peak to peak (fairly common in the U.S.)
 - $\delta_{\max} = S/2 = 1''$
 - Required input acceleration, $A_{\max} = 15 g$ $\Leftarrow g = 386 \text{ in/s}^2$
 - The minimum frequency for the test is

$$f \geq \frac{1}{2\pi} \sqrt{\frac{15(386 \text{ in/s}^2)}{0.8(1.0 \text{ in})}} = 13.5 \text{ Hz}$$

Round up to an integer: **$f = 14 \text{ Hz}$ test frequency**

Selecting the Sine Burst Frequency for a Shaker With a 2-inch Stroke, Peak to Peak

Theoretical maximum acceleration

Recommended allowable acceleration (80% of theoretical)—unless the test-lab personnel recommend something different

f, Hz	A_{max}, g	A_{allow}, g
5	2.56	2.05
6	3.68	2.95
7	5.01	4.01
8	6.55	5.24
9	8.28	6.63
10	10.2	8.18
11	12.4	9.90
12	14.7	11.8
13	17.3	13.8
14	20.0	16.0
15	23.0	18.4
16	26.2	20.9
17	29.6	23.6
18	33.1	26.5
19	36.9	29.5
20	40.9	32.7

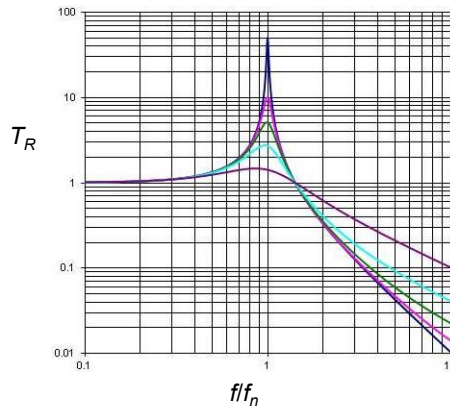
Example from previous page:

Required input acceleration = 15 g

The test should be run at a frequency no lower than 14 Hz

If the shaker you will be using has a different stroke, scale these accelerations accordingly. Peak acceleration at a given frequency is proportional to stroke.

Accounting for Dynamic Amplification (Gain)



For $Q \geq 7.5$, to 3 significant figures:

$f_{\text{ratio}} = f/f_n$	T_R
0.60	1.56
0.50	1.33
0.40	1.19
0.35	1.14
0.30	1.10
0.25	1.07
0.20	1.04
0.15	1.02
0.10	1.01

$$T_R = \frac{1 + \left(\frac{2\zeta f}{f_n} \right)^2}{\sqrt{\left[1 - \left(\frac{f}{f_n} \right)^2 \right]^2 + \left(\frac{2\zeta f}{f_n} \right)^2}} \quad (\text{Ref. Eq. 1-14})$$

These values apply to a mass on a spring but can be used to estimate dynamic gain at a spacecraft's fundamental frequency.

Example: 10 g sine burst test at 15 Hz, for a structure with a fundamental frequency of 30 Hz and a Q of 20:

$$f_{\text{ratio}} = \frac{15}{30} = 0.5$$

Expected average response acceleration, A_{max} :

$$A_{\text{max}} \approx 1.33(10) = 13.3g$$

Accounting for Dynamic Amplification (continued)

When the dynamic gain is significant, as it is for the example on the previous page, we should decrease input acceleration to avoid overtesting the structure.

- However, for this example, we can't simply reduce the input to $10/1.33 = 7.5$ g unless our structure is truly a mass on a spring.
- Actual response acceleration will vary, depending on the mode shape.
- We can usually meet the objective of verifying strength if the required base force and moment (loads at the mounting interface) are achieved.
 - We addressed methods of determining base loads in Sec. 2 of this course.
 - Tune the input environment based on lower-level test runs as needed to achieve the required test loads in the full-level run.

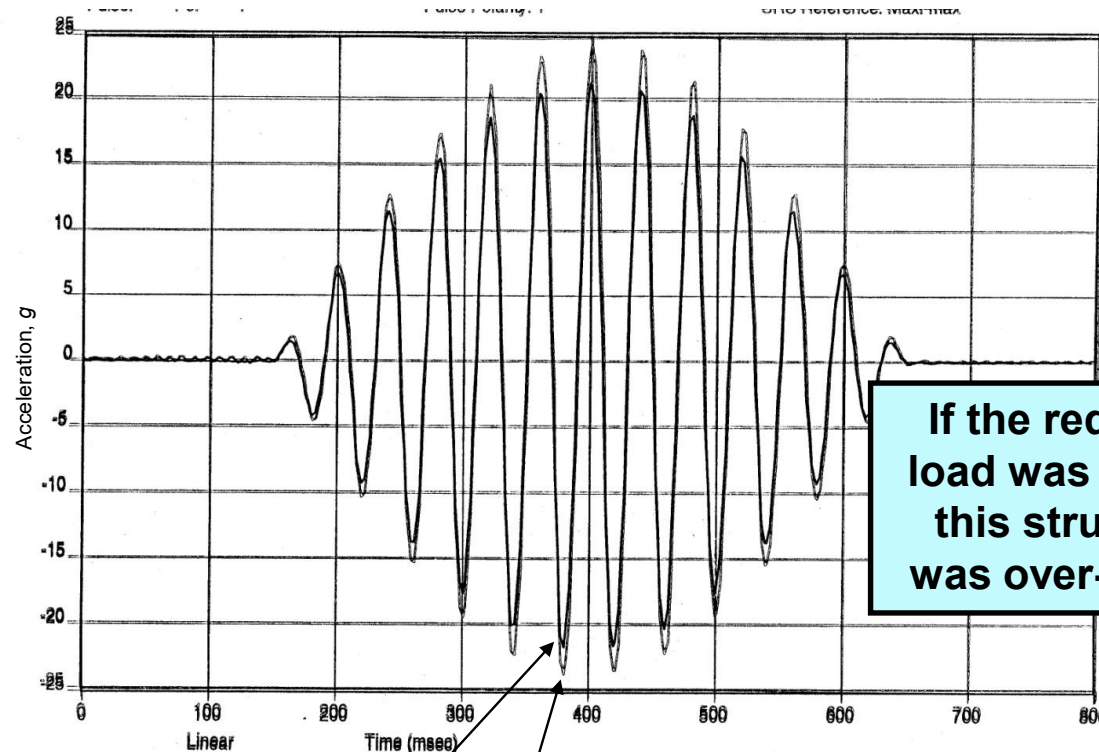
Example of Dynamic Response During a Sine Burst Test

Response of the test article overlaid onto the acceleration of the shaker

This was a lateral test, with response measured at top of the top of the test article

25-Hz input

100-Hz
fundamental
lateral frequency
(rocking mode)

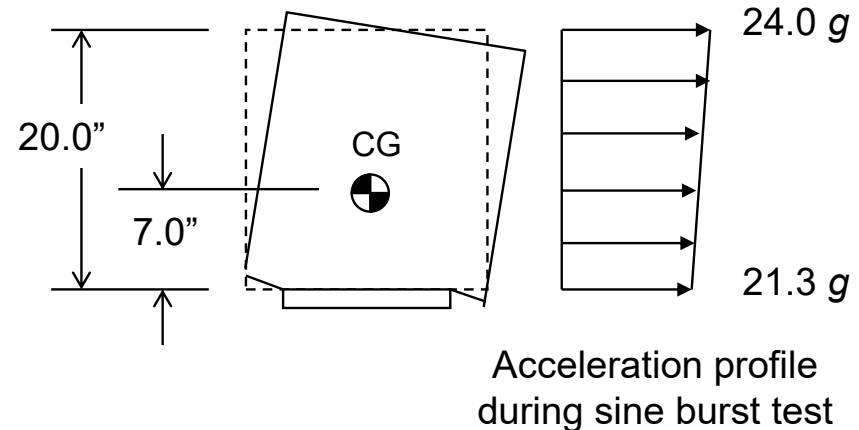


21.3-g peak input

24.0-g peak response (13% greater than the input)
(not experienced by all of the mass)

Class Problem 4-2: Deriving an Equivalent Sine Burst Level

- The 100-Hz mode corresponding to the plot on the previous page is rocking of a rigid body on a base spring.
- The test article weighs 150 lb, with CG shown at right.
- Mass moment of inertia at CG, $J = 30.0 \text{ lb-s}^2\text{-in.}$



Problem statement: Given that the objective of this test is to verify strength of this structure at its mounting interface, derive the peak input acceleration for a 25-Hz sine burst that causes the same interface (base) moment as does a 21.3 g quasi-static load.

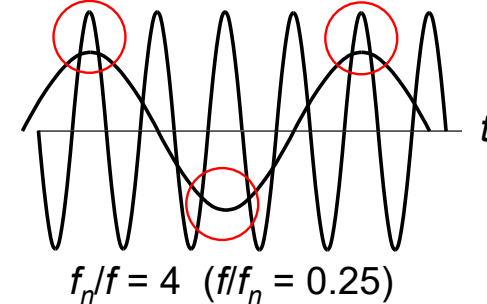
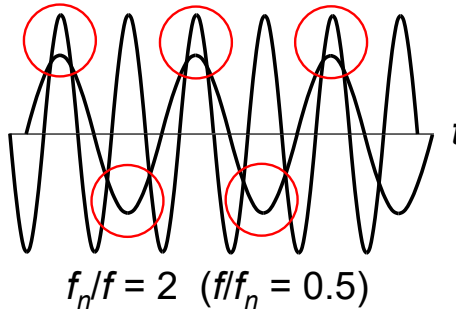
Assume all of the mass rotates as a rigid body according to the top and bottom accelerations shown above.

Use Equations 2-1 through 2-3 at the end of Sec. 2.

Potential Problem with Sine Burst Testing: Avoid Natural Frequency Being an Odd Multiple of the Input Frequency

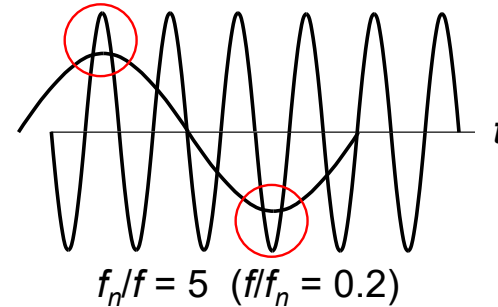
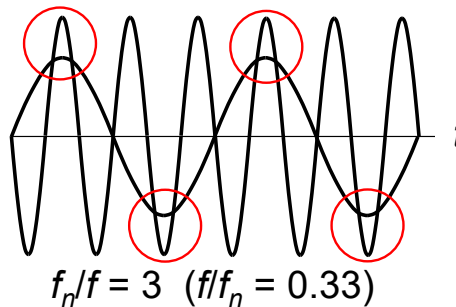
Even integer multiples: →

- No concern
- Peaks do not align



Odd integer multiples: →

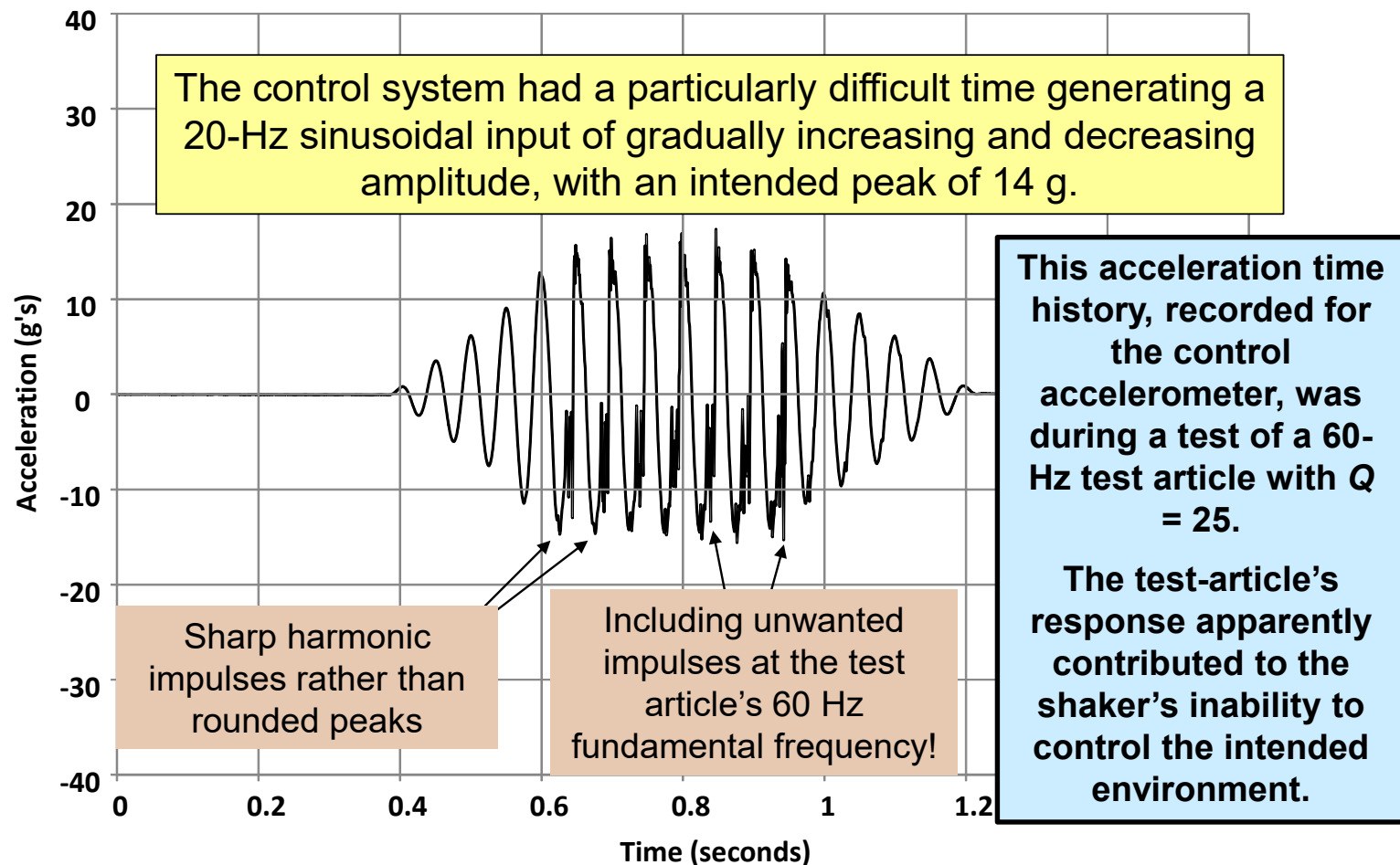
- Avoid
- Peaks align



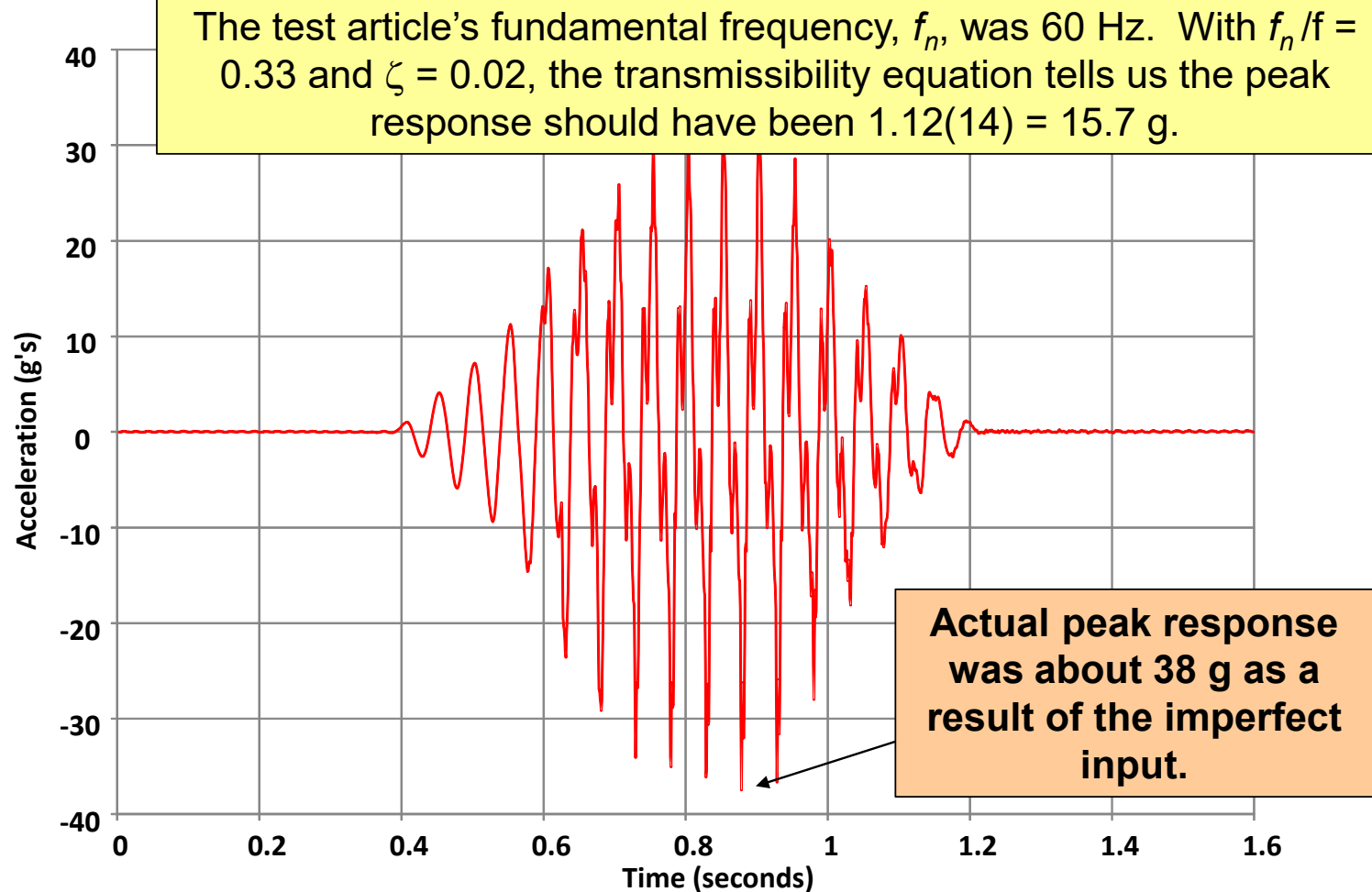
The sine burst input from the shaker will not be a perfect sine wave. As a result, the test article may see high response loads if its natural frequency is an odd multiple of the input frequency.

In addition, if f_n is an odd multiple of f , the test article's response is more likely to prevent the control system from generating a sinusoidal input.

Example (case history): Imperfect Sine Burst Input

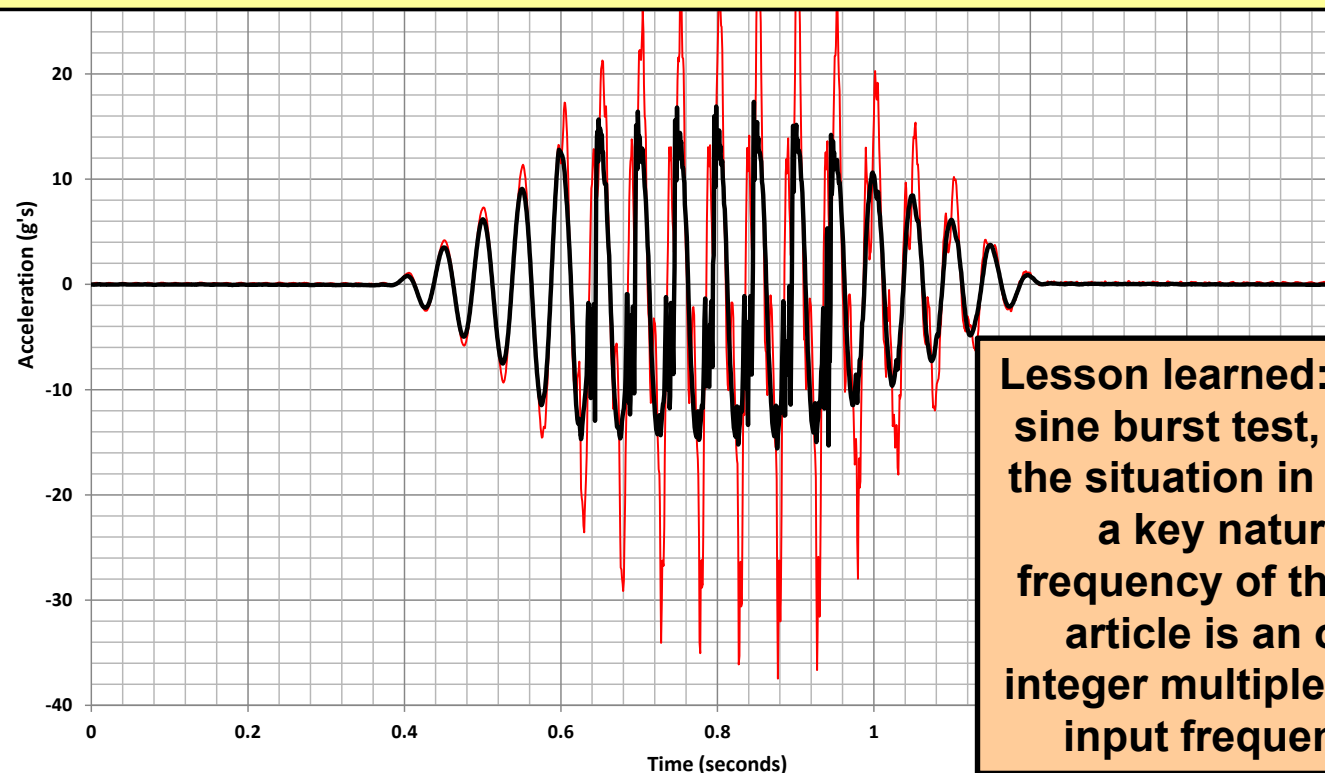


Example (continued):
How the Test Article Responded to the Imperfect Sine Burst Input



Example (continued): Superimposed Input and Response

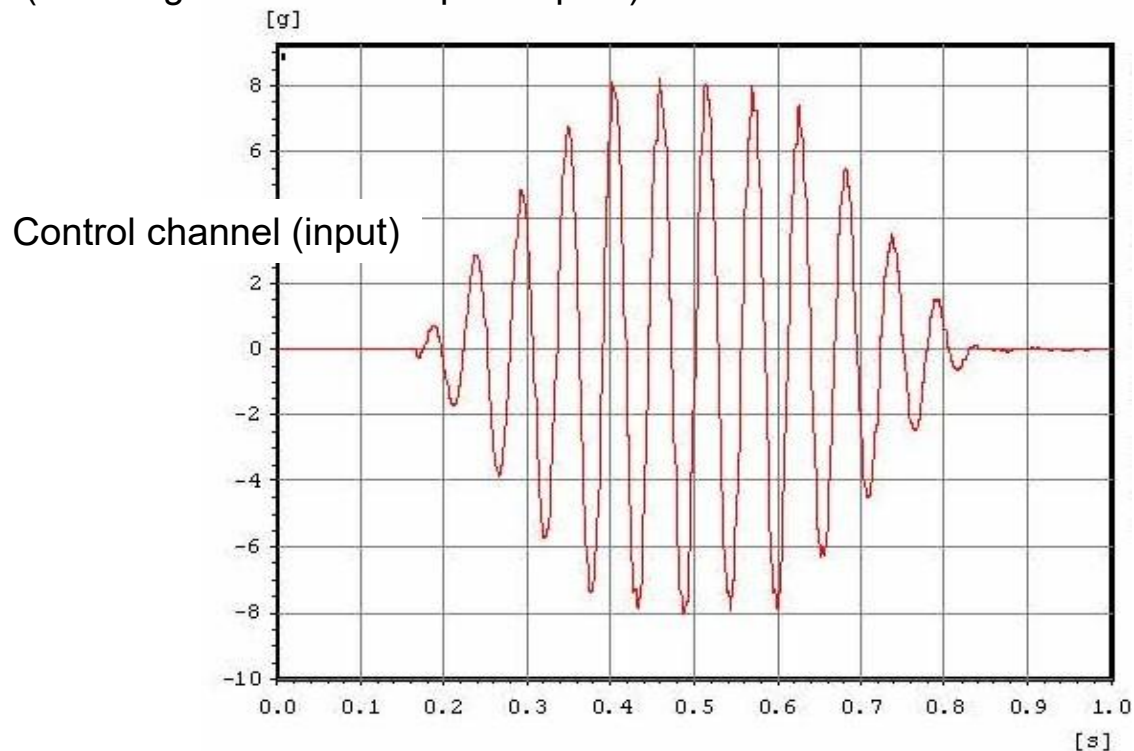
The shaker's inability to properly control the input sine wave, given 60-Hz response of the test article, may have been at least partly the result of electrical noise being at 60 Hz.



Lesson learned: For a sine burst test, avoid the situation in which a key natural frequency of the test article is an odd integer multiple of the input frequency.

Same Problem in Another Test to a Lesser Extent

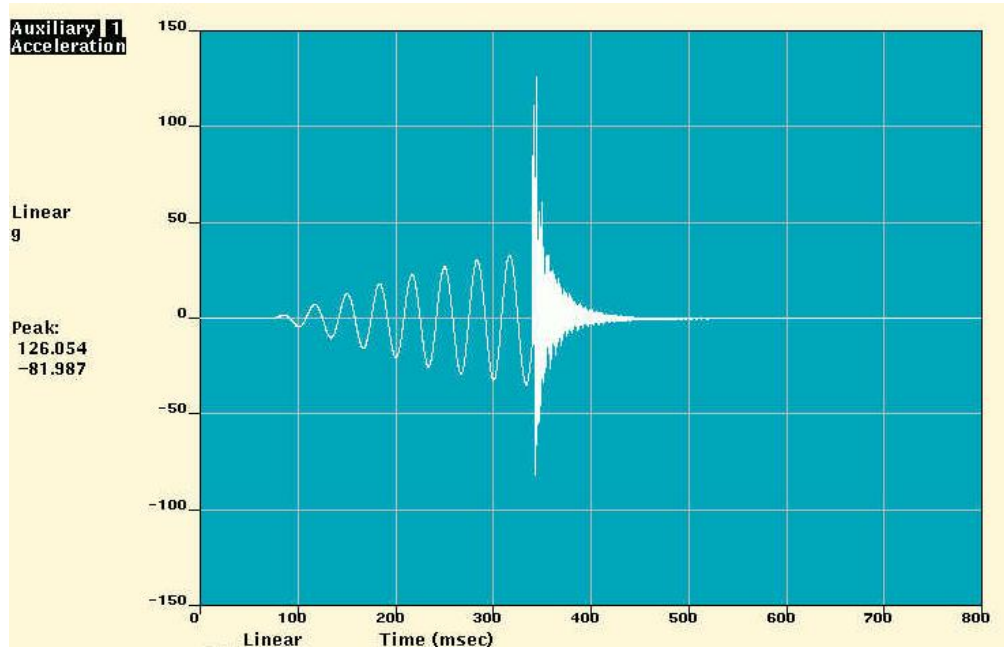
- Sine burst test of a large telescope: 8.1 g full level at 18 Hz, vertical axis
- Input deviated from a sine wave, with harmonic frequencies; did not occur at -6 dB
- Secondary mirror saw 13.4 g response, with pronounced peaks at 126 Hz (7 times 18)
(I no longer have the response plot.)



Again the problem was a response mode at an odd multiple of the input frequency.

This was for a different test lab than the previous case history.

Problem Encountered During Sine Burst Testing of the FalconSat-2 Engineering Model (EM)



Intended to be a 35-g sine burst test

- This plot shows response of the top of the EM during a lateral sine burst test.
- The shaker's amplifier failed during sine burst testing and imparted an impulse (or sudden stop), which excited the EM's rocking mode.
- The top of the EM saw a peak acceleration of 126 g.
- Fortunately it was not flight hardware!

Recommendations Regarding Sine Burst Testing

As evidenced by the case histories shared herein, sine burst testing is risky. Thus,

- If you plan to do a sine burst test on a relatively large (say, >100 lb) or expensive item, don't do it on the fully integrated flight assembly.
 - Remember: The objective of sine burst testing usually is to verify strength of the primary structure only. Strength of most equipment and secondary structures is verified by random vibration testing.
- Instead, do the sine burst test prior to system integration on the primary structure with mass simulators representing the missing items.
- Or test the primary structure with static loads or on a centrifuge prior to system integration.

If the first chance to do a strength test is on the shaker for the integrated flight article, consider a low-frequency sine vibe test instead of a sine burst test. Example: 12 to 15 Hz sweep for a 50 Hz test article.

Ask the test-lab personnel if such a test is easier to control with their control system.

Key Points of This Section

- If it's questionable whether a sine burst test will meet its objectives—strength verification—given the structure's potential failure modes, consider a different test, such as static loads or centrifuge.
- Look for clues in the response data that something in the structure is behaving nonlinearly, which could imply yielding or impending failure.
 - Make sure either the total base loads are at or above the required levels,
 - or each relatively high-mass part of the test article sees the required acceleration.
- Design the test to ensure, apples to apples, that the structure is loaded at least as severely as the specified limit loads multiplied by the test factor.
- The shaker's stroke limits how low the frequency can be for a sine burst test to achieve the required acceleration.
- When applicable, reduce the input acceleration to account for significant dynamic amplification.
- Avoid a test frequency for which the fundamental frequency of the test article is an odd integer multiple.
- Avoid sine burst tests on fully integrated spacecraft; test the structure first, with mass simulators for components.

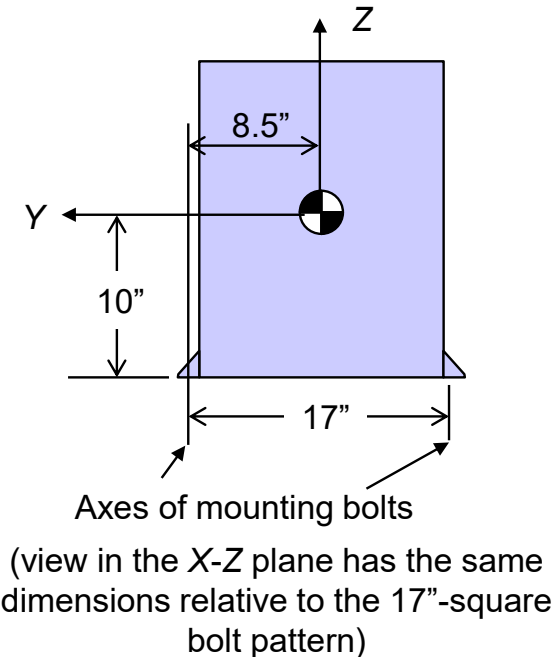
Appendix to Sec. 4

Example Problem 4-b: A variation of Class Problem 4-1

Appendix to Sec. 4: Example Problem 4-b

Given:

- The test article is mounted to the test fixture with four bolts, one at each corner of a 17" square pattern.
- The CG is 10" above the plane of the bolt pattern and centered on the Z axis.
- The test article (flight article) weighs 150 lb.
- The specified quasi-static limit loads are +/- 8.5 g in the X axis simultaneous with 8.5 g in any axis in the Y-Z plane.
- Sine burst tests will be run in the X, Y, and Z directions.
- The structure must be tested to 1.25 times limit.
- Assume tension in the mounting bolts is the only indicator parameter being tracked to ensure adequate test loads.



Problem statement: Rework Class Problem 4-1 with the above change in wording regarding the specified limit loads.

Determine the sine-burst acceleration in either X or Y that would adequately load the mounting bolts in tension.

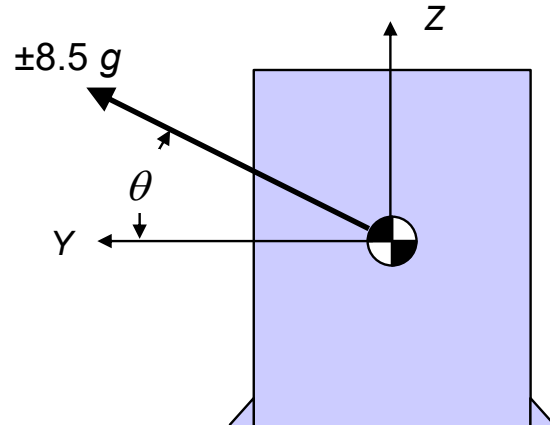
continued

Appendix to Sec. 4: Example Problem 4-b, continued

From the solution to Class Problem 4-1, the target bolt tensile load is calculated by ...

$$P_{t\text{-target}} = 1.25(44.12n_x + 44.12n_y + 37.50n_z)$$

Compute bolt load for load cases with 8.5 g in X and 8.5 g acting at varying angles in the Y-Z plane:



$$n_y = 8.5 \cos \theta$$

$$n_z = 8.5 \sin \theta$$

θ , deg	Limit load factors			$P_{t\text{-target}}$, lb
	n_x	n_y	n_z	
0.0	8.50	8.50	0.00	938
5.0	8.50	8.47	0.74	970
10.0	8.50	8.37	1.48	1000
15.0	8.50	8.21	2.20	1025
20.0	8.50	7.99	2.91	1046
25.0	8.50	7.70	3.59	1062
30.0	8.50	7.36	4.25	1074
35.0	8.50	6.96	4.88	1081
40.0	8.50	6.51	5.46	1084
45.0	8.50	6.01	6.01	1082
50.0	8.50	5.46	6.51	1075
55.0	8.50	4.88	6.96	1064
60.0	8.50	4.25	7.36	1048
65.0	8.50	3.59	7.70	1028
70.0	8.50	2.91	7.99	1004
75.0	8.50	2.20	8.21	975
80.0	8.50	1.48	8.37	943
85.0	8.50	0.74	8.47	907
90.0	8.50	0.00	8.50	867

← Target bolt load for test

continued

Appendix to Sec. 4: Example Problem 4-b, continued

Derive the test acceleration required in the X or Y axis to achieve the target bolt load:

$$a = \frac{1084 + 37.5}{44.12} = 25.4 \text{ g}$$

to overcome the -37.5 lb
gravitational load

continued

Appendix to Sec. 4: A Shortcut for Example Problem 4-b

When load factors are specified in this manner (“anywhere within the Y-Z plane”), we can take a conservative shortcut, as we did with Option 3 for Example 4-3: add the accelerations (or load factors).

A conservative assumption in this case is that the load acts at a 45° angle, bisecting the Y and Z axes ($\theta = 45^\circ$). For Example 4-7,

$$n_y = 8.5 \cos(45^\circ) = 6.01$$

$$n_z = 8.5 \sin(45^\circ) = 6.01$$

$$\text{Load factor sum} = n_x + n_y + n_z = 8.5 + 6.01 + 6.01 = 20.52$$

Test to $1.25 \times 20.52 + 1 = 26.7 \text{ g}$ separately in each of the X, Y, and Z axes.

to counteract the effects of gravity

The maximum bolt load will be from the X and Y tests:

$$P_t = 44.12(26.7) - 37.50 = 1140 \text{ lb}$$

gravitational load

vs. 1084 lb target
load derived earlier

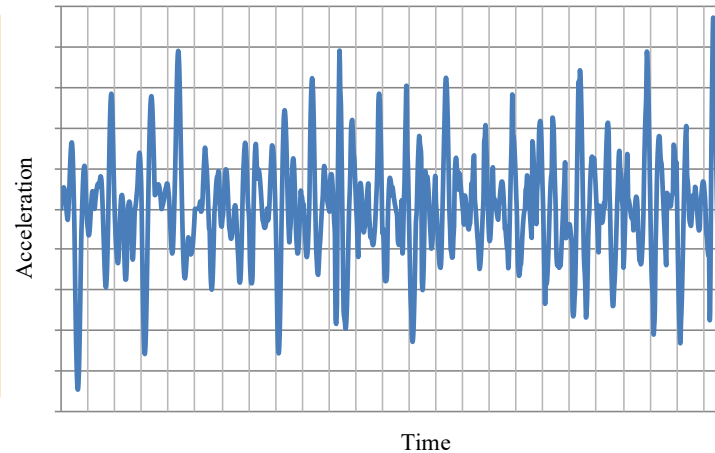
5. Random Vibration Testing

- Introduction and Objectives
- Acceleration Power Spectral Density:
Understanding g^2/Hz
- Root-Mean-Square (RMS) Acceleration
- Understanding Probability for Random Vibration
- Peak Load for Random Vibration
- Decibels
- How Test Environments Are Derived
- Government Standards
- Use and Limitations of GEVS Environments
- Standard Test-Control Tolerances
- Data Resolution
- Interpreting Test Data
- Appendix: An Example of Using Miles' Equation
To Estimate Random Vibration Response

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Introduction to Random Vibration

Random vibration consists of multiple frequencies of acceleration, each with random amplitude, acting simultaneously.



Example random combination of waves of different frequency

In the space industry, we traditionally quantify random vibration over the frequency range of 20 – 2000 Hz.

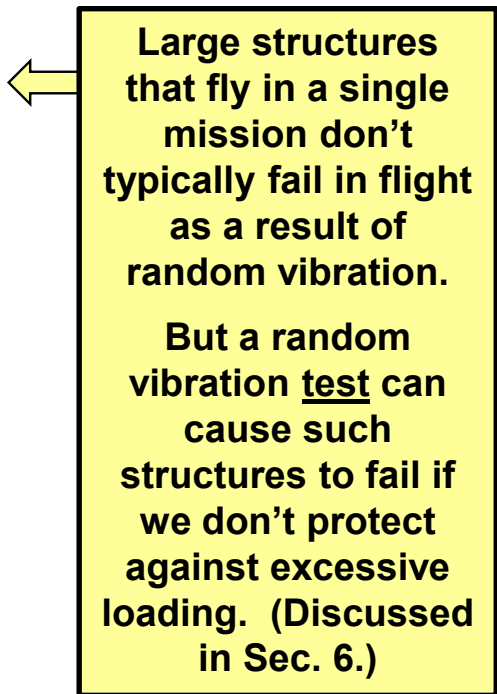
- During launch, random vibration is generated mostly by response of panels and shells to random acoustic pressure, but also by engine vibration and random aerodynamic pressure (turbulence).
 - The launch-vehicle and spacecraft structures transmit vibration to mounted components.
- Acoustic levels peak at liftoff or during the transonic phase, with total duration of significant levels typically less than 15 seconds.
- Random vibration from turbulence (< 100 Hz) can last longer, perhaps 30 seconds.

Objectives and Susceptibility

Objective of random vibration testing: Verify that the hardware will be able to withstand random vibration during flight and still function properly.

Failures that tend to occur from flight random vibration:

- Material rupture—usually for relatively small items with high (> 100 Hz) fundamental frequencies
 - Small instruments, avionics, PC boards, soldered joints
 - Primary structures of very small satellites, such as CubeSats
- Loosening of threaded fasteners (loss of preload)—may or may not be considered “failure”, depending on the consequences—and fasteners backing out completely
- Unseating of electrical connectors and flipping phase of mechanical relays



Large structures that fly in a single mission don't typically fail in flight as a result of random vibration.

But a random vibration test can cause such structures to fail if we don't protect against excessive loading. (Discussed in Sec. 6.)

Material Rupture as a Result of Random Vibration

Rupture of brittle materials is a result of the peak stress occurring during an event; this is considered a strength failure (aka **single-passage failure**).

When a ductile material ruptures under exposure to random vibration, the failure is usually from the accumulation of loading cycles (fatigue).

A main objective of qualification random vibration testing is to verify (or build confidence in) fatigue life.

The duration of exposure to the environment is thus important.

Acceleration Power Spectral Density (PSD)

- By definition, random acceleration is not predictable at any point in time, so we don't use a time history of acceleration to define the environment.
- Instead, we define the environment by its frequency content with PSD.
- Starting with a representative time history, we break the full frequency range into bands and calculate the PSD for each band, including only the frequency content of acceleration within that band.
- For a given frequency band, with f as the center, the PSD is the mean-square acceleration within that frequency band divided by the bandwidth, Δf .

$$\text{PSD, } W(f) = \frac{1}{\Delta f} \left[\overbrace{\frac{1}{T} \int_0^T a^2 dt}^{\text{mean square}} \right]$$

(Eq. 5-1)

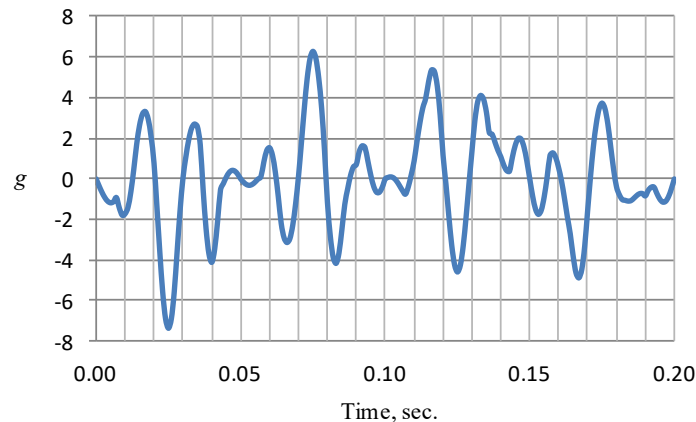
where a is the acceleration within the frequency band, and T is the duration of the sample time history

Power spectral density can be used for processing of any random signal that varies with time. The mean-square value of any random signal is referred to as the **power** of that signal.

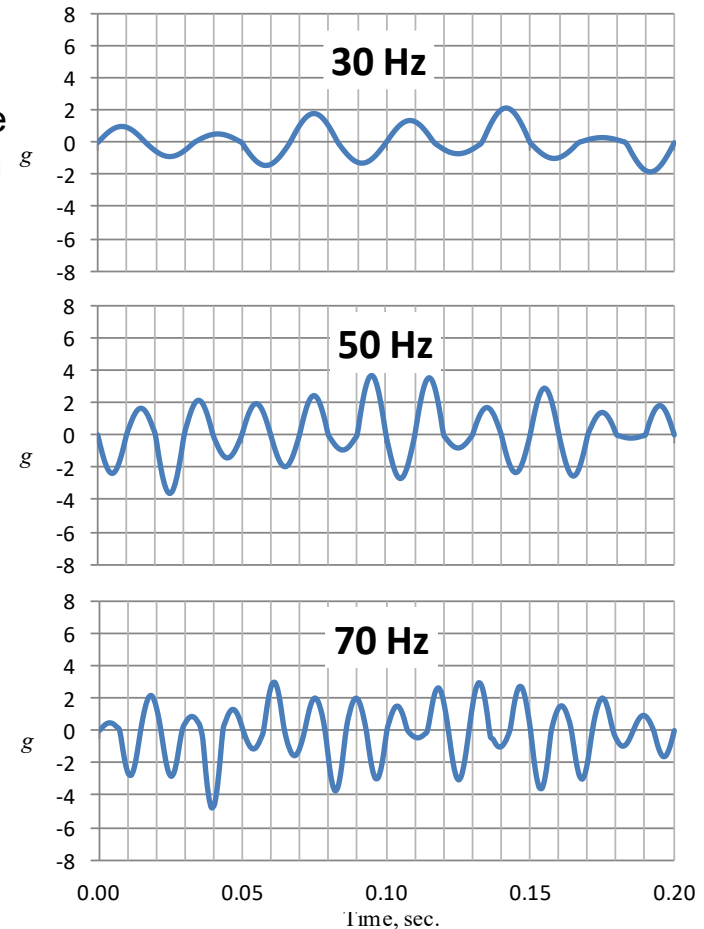
Example Problem 5-a: Understanding Random Vibration

Problem statement:

Derive a PSD from the acceleration time history shown below, recorded over a 0.2-sec interval:



We break the signal down into three sine functions with random amplitude

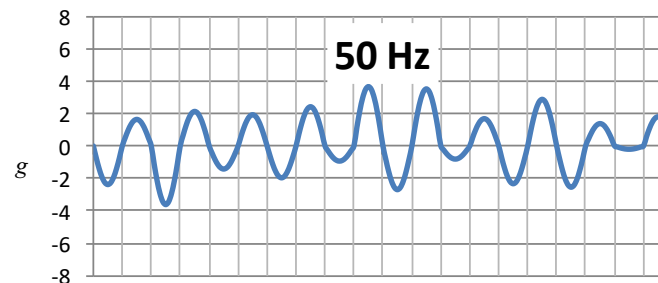


Here, the frequency resolution (bandwidth) is 20 Hz, and the bands are 20 – 40 Hz, 40 – 60 Hz, and 60 – 80 Hz.

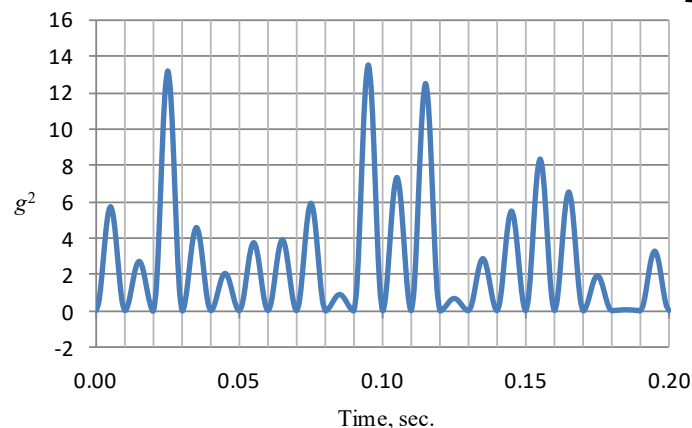
continued

Example Problem 5-a, continued

We'd like to characterize each random sine function, but calculating the average acceleration is of no use; we know the average is zero.



Squaring the acceleration makes all values positive:



We calculate the average squared (mean square) acceleration by taking the area under the curve and dividing it by the duration, T (0.20 sec):

$$\bar{a}^2 = \frac{1}{T} \int_0^T a^2 dt \quad (\text{Eq. 5-2})$$

We then divide this value by the frequency bandwidth (20 Hz in this example) to get the PSD in g^2/Hz .

continued

Example Problem 5-a, continued: Calculating PSD and RMS

Overall acceleration (g)					30-Hz acceleration			50-Hz acceleration			70-Hz acceleration		
t (sec)	a	a ₁	a ₁ ²	a ₁ ² dt	a ₂	a ₂ ²	a ₂ ² dt	a ₃	a ₃ ²	a ₃ ² dt			
0.000	0.00	0.00	0.00	0.00000	0.00	0.00	0.00000	0.00	0.00	0.00000			
0.001	-0.34	0.19	0.04	0.00004	-0.74	0.54	0.00054	0.21	0.04	0.00004			
0.002	-0.66	0.37	0.14	0.00014	-1.40	1.97	0.00197	0.37	0.14	0.00014			
0.003	-0.92	0.54	0.30	0.00030	-1.93	3.73	0.00373	0.47	0.22	0.00022			
...	...	...	...	...	...	...	...	...	...	...			
...	...	...	...	...	...	...	...	...	...	...			
0.197	-1.10	-0.98	0.95	0.00095	1.46	2.13	0.00213	-1.59	2.51	0.00251			
0.198	-0.87	-0.67	0.45	0.00045	1.06	1.13	0.00113	-1.26	1.59	0.00159			
0.199	-0.48	-0.34	0.12	0.00012	0.56	0.31	0.00031	-0.70	0.49	0.00049			
0.200	0.00	0.00	0.00	0.00000	0.00	0.00	0.00000	0.00	0.00	0.00000			
Sum (area under the curve), g ² -sec				0.17108			0.52371			0.58094			
Mean square (sum ÷ 0.2 sec), g ²				0.86			2.62			2.90			
PSD, (mean square ÷ 20 Hz), g ² /Hz				0.043			0.131			0.145			

Overall, the mean square acceleration is $0.86 + 2.62 + 2.90 = 6.38 \text{ g}^2$

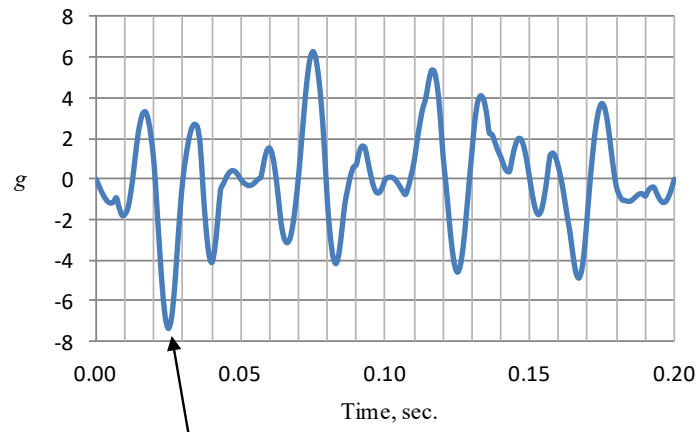
The square root of this value is the **root-mean-square (RMS)** acceleration: 2.53 g, commonly referred to as 2.53 g_{rms} (also written as G-rms, gRMS, g rms, etc.)

The RMS acceleration is the standard deviation, σ , of the random acceleration.

continued

Example Problem 5-a, conclusion: Peak Acceleration

From the original time history, we see the absolute peak acceleration is 7.35 g.



Absolute peak

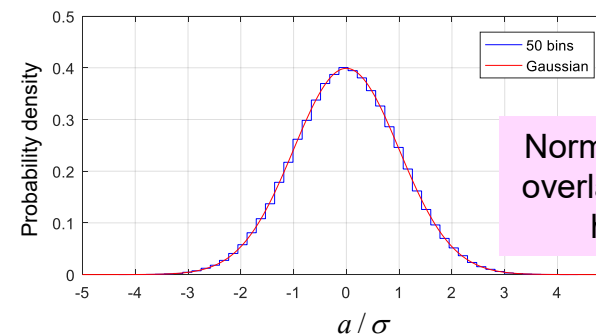
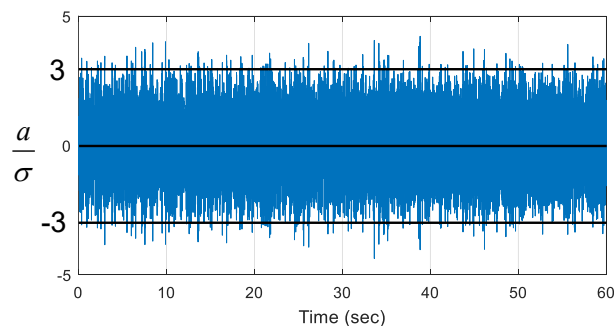
- With the standard deviation, $\sigma = 2.53$ g, the 7.35g peak is a 2.9σ value.
- This peak occurred in just 0.20 sec.
- If we sample a longer time span—say, the full 60 sec of a test—we would expect the peak value to be considerably higher.

It's often assumed that the peak load is 3σ , but the actual peak load during a random vibration event is usually well higher than that.

Probability Distributions for Random Vibration: The Normal Distribution for Instantaneous Signal Level

For random vibration, if we consider acceleration, a , at any point in time as a random variable, that variable tends to follow a normal (Gaussian) distribution.

When we calculate a root-mean-square (RMS) acceleration, that value is the standard deviation, σ , of the random acceleration at any point in time.

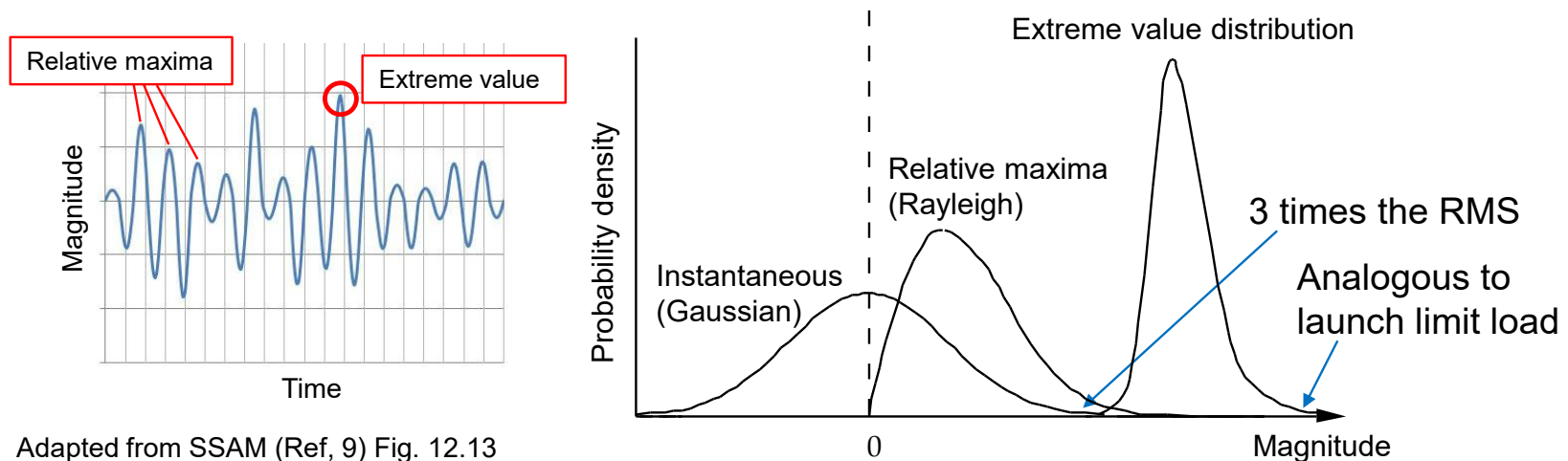


Normal distribution
overlaid on top of a
histogram

At any particular *instant*, there is a 99.74% (3σ) probability that the acceleration is contained within $\pm 3\sigma$, but over a long duration, such as 60 seconds, there will be many absolute peaks of acceleration that exceed $\pm 3\sigma$.

This slide contributed by John MacCoun

Understanding Probability for Random Vibration



Adapted from SSAM (Ref, 9) Fig. 12.13

- The **normal distribution** represents the acceleration at any point in time. The root-mean-square (RMS) value associated with the response PSD is the standard deviation of the distribution represented by Curve A.
- The **Rayleigh distribution** represents the distribution of relative maxima for narrow-band (single frequency) random vibration (Ref. 27, Sec. 21-10).
- The **extreme value distribution** represents the distribution of the highest level that occurs over a given time period (Ref. 27, Sec. 21-11).

When we say a launch limit load (e.g., for coupled loads analysis) is 3σ , we're aiming for 99.87% probability on the peak load during launch (extreme value distribution).

We often see 4.5σ and 5σ peaks in test, not just in theory—even when using “3-sigma clipping”

Reference 10 debunks the idea that 3σ clipping protects a test article from seeing responses greater than 3σ .

It is the **drive signal** (the electrical signal from the controller to the amplifier) that is clipped.

After explaining 3σ clipping of the drive signal, Ref. 10 poses the questions ...

“What happens when we actually drive a shaker with a clipped Drive? Does the output of the amplifier reflect the limiting? Does an accelerometer sitting on the shaker table detect a clipped Control signal?”

The authors of Ref. 10 then report on a test they conducted:

In the test, “the amplifier’s input is sharply limited to $\pm 3\sigma$. Note that the amplifier’s output also reflects this limiting. However, the clipped voltage applied to the shaker does not result in a limited Control signal.” The data “clearly shows the table acceleration experiencing $\pm 4\sigma$, essentially the same range as without Drive clipping.”

“So it is most unlikely that clipping will ... do anything to protect your delicate DUT” (device under test).

Is 3 Sigma (3 times the RMS value) for Random Vibration an Appropriate Limit Load for Stress Analysis?

The assumption of 3σ as the peak load for use in analysis is commonly made and accepted in the space industry.

- Why? Most likely because of not understanding the difference between the instantaneous (Gaussian) distribution and the extreme value distribution.

And use of this assumption has been successful for the most part. Why? Because of other compensating conservatism:

- Factor of safety and A-basis allowable stresses for strength analysis
- Damping usually higher than assumed, especially at full test levels
- Using the calculated RMS response acceleration as a quasi-static load in analysis (especially conservative when the RMS value is calculated from 20 – 2000 Hz)
- Linear-elastic analysis for ductile failure modes
 - Yielding absorbs energy and can reduce the peak response acceleration.
- Single-mission hardware often not seeing enough loading cycles to cause fatigue failure

But we must be cautious in counting on past success with the 3-sigma assumption, especially if we remove some of the above conservatism.

Suggested Sigma Level for Use in Design for Random Vibration

For strength analysis, with a factor of safety, ...

- Brittle failure (composites, ceramics, glass): Use the maximum level predicted by the Rayleigh distribution. (See Sec. 5 of Ref. 36, SDA course book.)
- Ductile failure: Use of 3σ as the limit load is probably adequate when making other assumptions that are conservative and using an ultimate factor of safety—but fatigue analysis is more appropriate.
 - Exception: Treat any failure that exhibits little plastic deformation, such as stripping of bolt threads, as brittle failure, and use the appropriate sigma level, per the Rayleigh distribution.

For fatigue or damage-tolerance (crack-growth) analysis, without a factor of safety, ...

- Use the maximum sigma level predicted by the Rayleigh distribution ...
- unless you are convinced there is enough compensating conservatism in the analysis to truncate the analysis at 3σ .

Decibels

- When a PSD is increased or decreased, we refer to that level change in decibels.
- A **decibel** (dB) is the logarithm of a ratio.
- At a given frequency, when going from an initial PSD, W_1 , to a final PSD, W_2 , the decibel change is

$$N_{dB} = 10 \left[\log_{10} \left(\frac{W_2}{W_1} \right) \right] \quad (\text{Eq. 5-3})$$

- Conversely, the ratio by which a PSD increases for a given decibel change is approximately

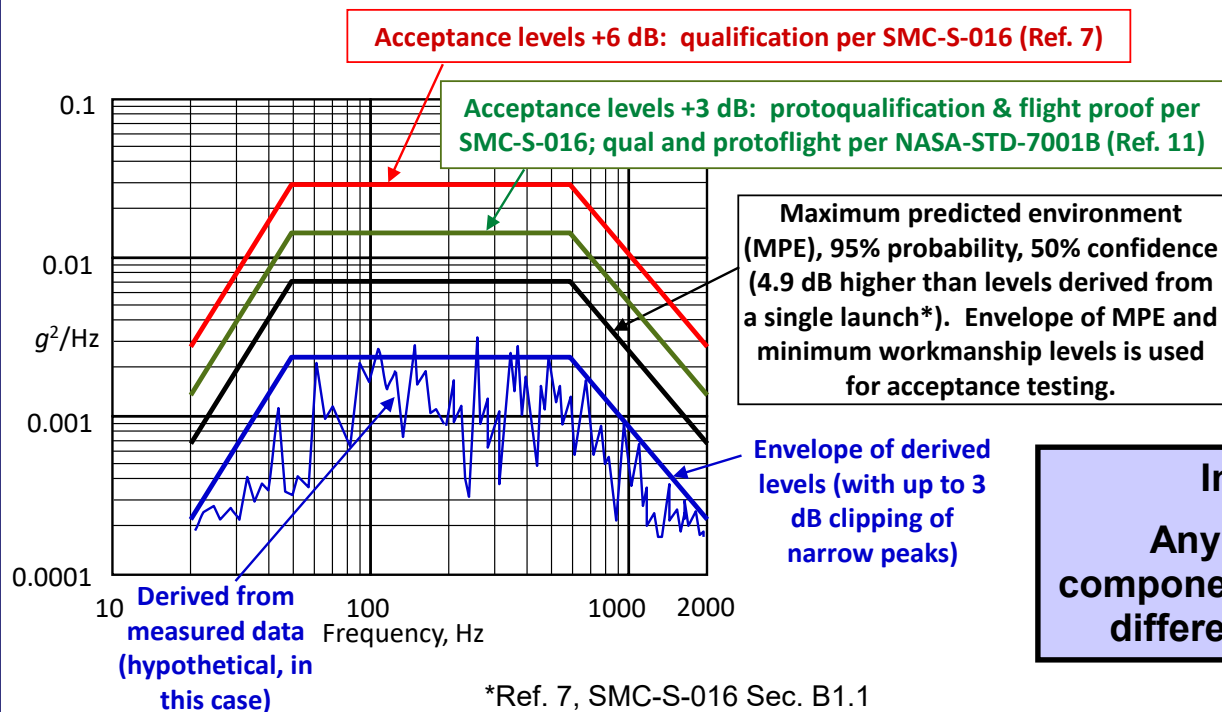
$$\frac{W_2}{W_1} \approx 2^{(N_{dB}/3)} \quad (\text{Eq. 5-4})$$

Example: For a 6 dB increase, the g^2/Hz goes up approximately by a factor of four, which means acceleration doubles.

How Test Environments Are Derived from Launch Data

For easy control during test—and to account for uncertainty—we smooth out the PSD plot to envelop any significant peaks. Then we increase the levels to obtain 95% probability and 50% confidence, and add margin for test.

Example:

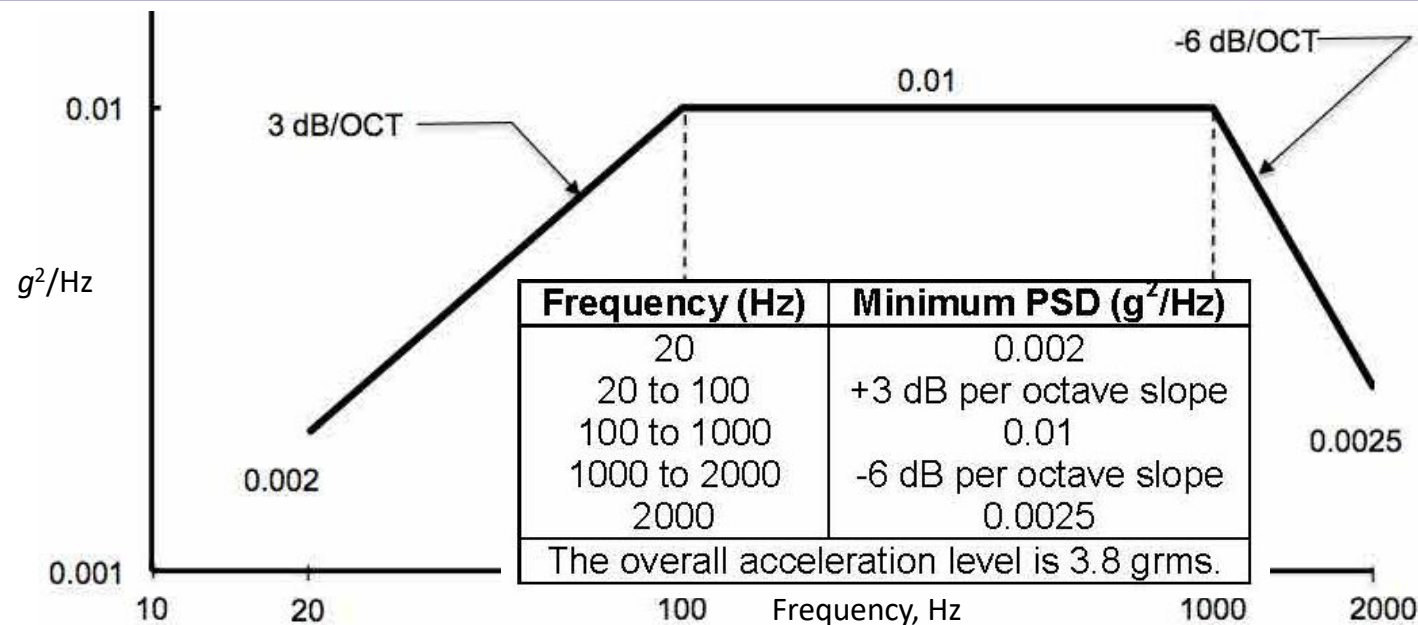


The RMS (root mean square) acceleration is the square root of the area under the PSD curve and is equal to the standard deviation of random acceleration.

Input $\neq$ Response

Any point on a vibrating component will have its own PSD, different than the input PSD.

Minimum Workmanship Levels for SmallSats Up To 400 lb per Ref. 7, SMC-S-016



(Reformatted SMC-S-016 (September 2014) Fig. 8.3.7-1)

GSFC-STD-7000B (Ref. 12) and NASA-STD-7001B (Ref. 11) specify minimum workmanship levels for components but not for full spacecraft.

A test to **minimum workmanship levels** is intended to ensure the hardware is adequately screened for defects related to variation in manufacturing processes and workmanship. Even if the mission MPE is quite low within certain frequency bands, the hardware will experience random vibration during ground processing, handling, and transportation.

Standards for Random Vibration Testing at Vehicle Level for Space and Launch Hardware with a Single Launch

Testing to be done in each of three orthogonal axes

Class of Test	SMC-S-016 (Ref. 7)		NASA-STD-7001B (Ref. 11)	
	Level	Duration	Level	Duration
Qualification	+6 dB	3 minutes	+3 dB	2 minutes
Protoqual or protoflight	+3 dB	2 minutes	+3 dB	1 minute
Acceptance	+0 dB	1 minute	+0 dB	1 minute

- Notes:
- Level specified is above acceptance levels, which are the higher of the maximum predicted (flight) environments (MPE) and minimum workmanship levels.
 - + 6 dB represents a factor of 2 on acceleration (a factor of 4 on g^2/Hz).
 - Durations are per axis.
 - The US Military traditionally does not put large spacecraft on shakers; SMS-S-016 says the above criteria apply typically for small spacecraft (<400 lb weight, 180 kg mass).
 - SMC-S-016 test durations shown above are the most commonly selected options. SMC-S-016 provides other options as well.

Generalized Random Vibration Environments for Spacecraft Components from GEVS

Spacecraft components up to 50 lb:

Frequency (Hz)	ASD Level (g^2/Hz)	
	Qualification	Acceptance
20	0.026	0.013
20-50	+6 dB/oct	+6 dB/oct
50-800	0.16	0.08
800-2000	-6 dB/oct	-6 dB/oct
2000	0.026	0.013
Overall	14.1 G_{rms}	10.0 G_{rms}

The acceleration spectral density level may be reduced for components weighing more than 22.7-kg (50 lb) according to:

	Weight in kg	Weight in lb	
dB reduction	$= 10 \log(W/22.7)$	$10 \log(W/50)$	
ASD(50-800 Hz)	$= 0.16 \cdot (22.7/W)$	$0.16 \cdot (50/W)$	for protoflight
ASD(50-800 Hz)	$= 0.08 \cdot (22.7/W)$	$0.08 \cdot (50/W)$	for accept

Where W = component weight.

The slopes shall be maintained at + and - 6dB/oct for components weighing up to 59-kg (130-lb). Above that weight, the slopes shall be adjusted to maintain an ASD level of 0.01 g^2/Hz at 20 and 2000 Hz.

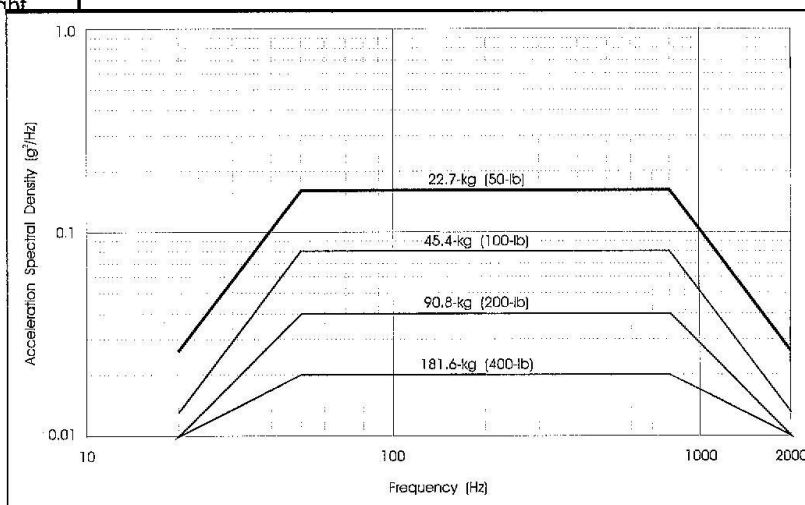
For components weighing over 182-kg (400-lb), the test specification will be maintained at the level for 182-kg (400 pounds).

Source: Ref. 12, GSFC-STD-7000B (GEVS).
Based on acoustic response analysis and
acoustic test results for large spacecraft.

Often used for LargeSat
components and entire SmallSats
when no better data are available

But we invite trouble if we use these
levels for SmallSat components
(addressed in Sec. 8 herein).

Qualification levels



Standard Test-Control Tolerances

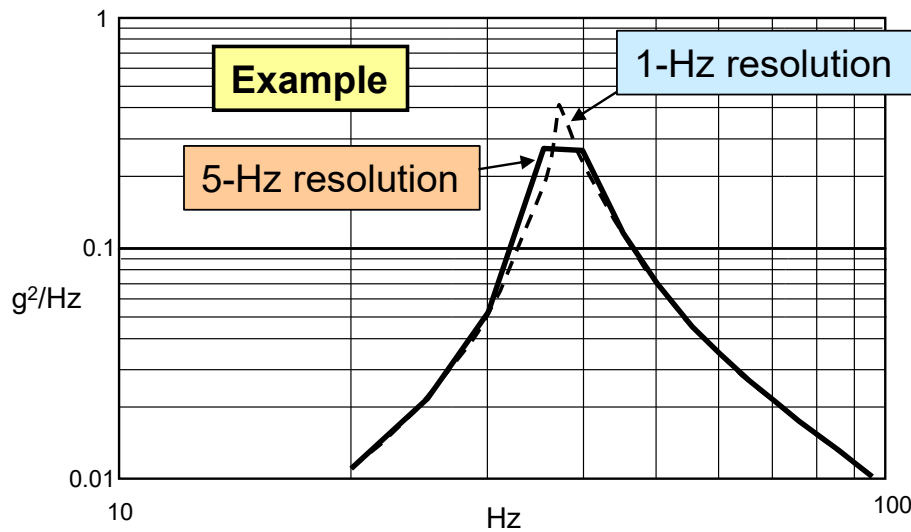
Frequency Range	Test Parameter	NASA-STD-7001B (Ref. 11)	SMC-S-016 (Ref. 7)
20 - 100 Hz	Power spectral density	± 3 dB	± 1.5 dB
	Maximum bandwidth for data resolution	25 Hz	10 Hz
100 - 1000 Hz	Power spectral density	± 3 dB	± 1.5 dB
	Maximum bandwidth for data resolution	25 Hz	10% of mid-band freq.
1000 - 2000 Hz	Power spectral density	± 3 dB	± 3 dB
	Maximum bandwidth for data resolution	25 Hz	100 Hz
20 - 2000 Hz	Overall level, g-rms	$\pm 10\%$	± 1 dB ($\pm 12\%$)

**The above standards for data resolution are outdated, in my opinion.
Today's controllers can do much better—and should.**

Effect of Data Resolution

Data resolution indicates the number of calculated PSD values (frequency bands). Increasing resolution means decreasing bandwidth of data processing.

The higher the resolution, the more accurately the test environment is controlled to the specification and the more accurately we'll be able to reduce the response data to useful information.



Truncated response can lead to considerable error, especially for low-frequency modes*.

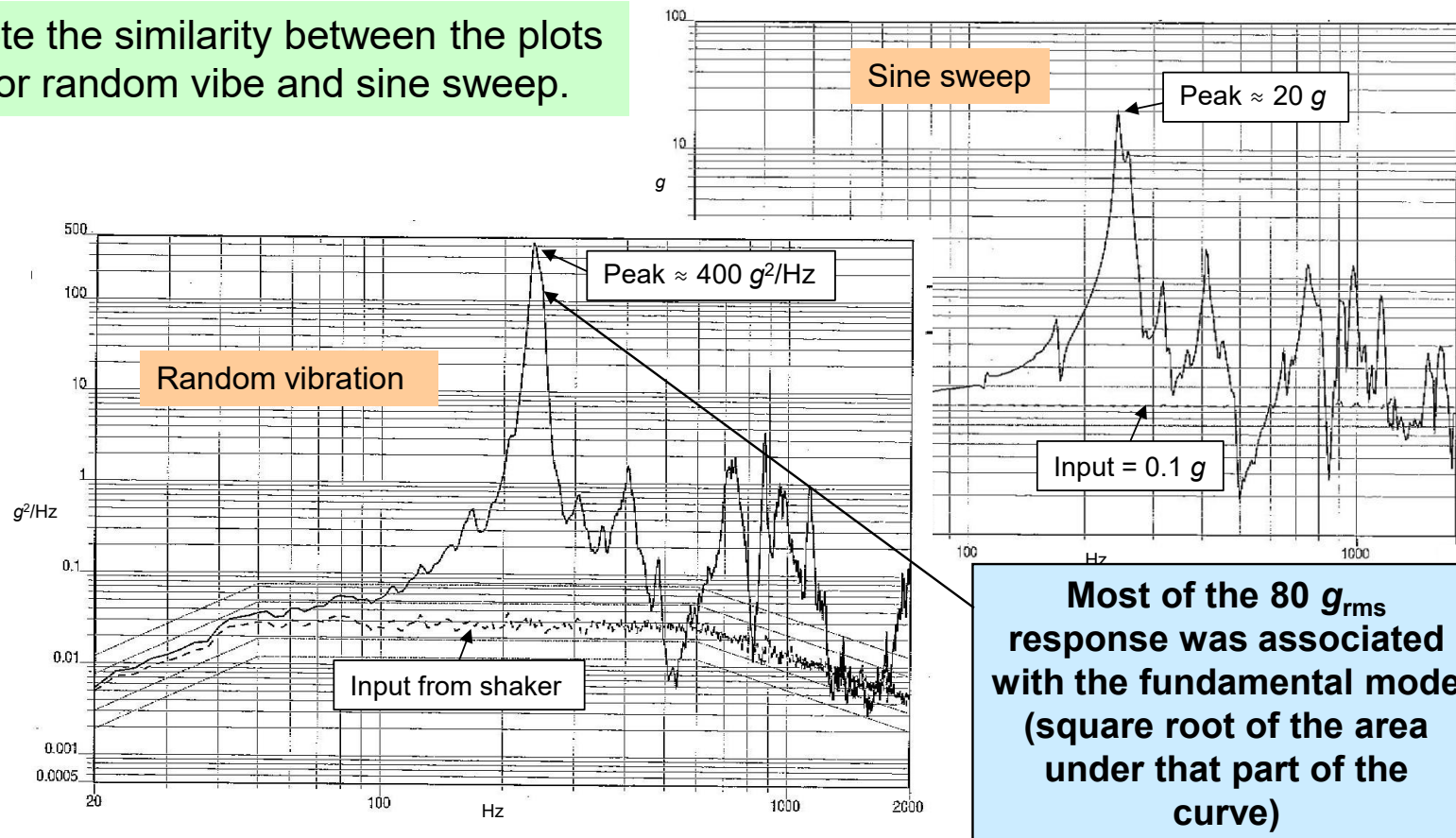
*What is important to accuracy is the ratio of that bandwidth to the test article's natural frequency of interest.

But higher resolution means more processing time, thus exposing the hardware longer to vibration as the controller brings the environment within tolerances.

Use the highest resolution that still allows the control system to perform responsively. I usually request 1.25 Hz, which works well with most shaker control systems.

Example: Random Vibe Response vs. Sine Sweep Response

Note the similarity between the plots for random vibe and sine sweep.



Relationship Between Sine Vibe Response and Random Vibe Response

For any data channel, the ratio of response to input, when computed as a function of frequency, is a **transfer function** for that channel.

If response is acceleration, same as input, we refer to this ratio as the **gain**.

For a linear system, at frequency f ...

$$R_{rv}(f) = I_{rv}(f) \left[\frac{R_{sv}(f)}{I_{sv}(f)} \right]^2 \quad (\text{Eq. 5-5})$$

random vibe response, g^2/Hz sine vibe response, g

random vibe input, g^2/Hz sine vibe input, g

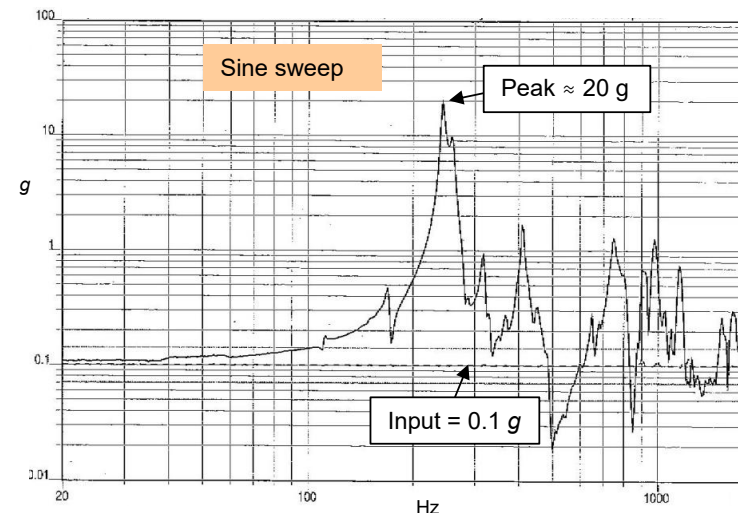
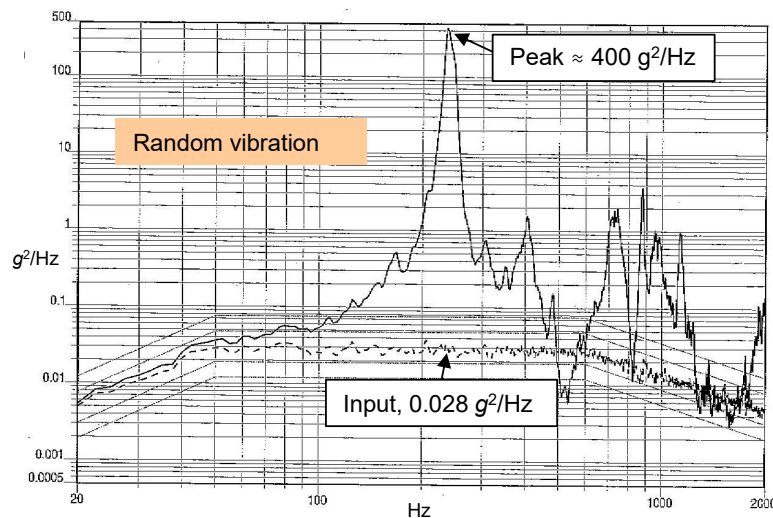
Instead of doing a dedicated low-level sine sweep, you can do a low-level random vibe test at a low, constant level* across the frequency spectrum, and derive the transfer function by rearranging the above equation.

*Referred to as **white noise**

Back to the Previous Example

If response had been linear—same frequency, mode shape, and damping for random vibrate and sine sweep—from Eq. 5-5 we would expect the peak, p , in the random vibrate test to be ...

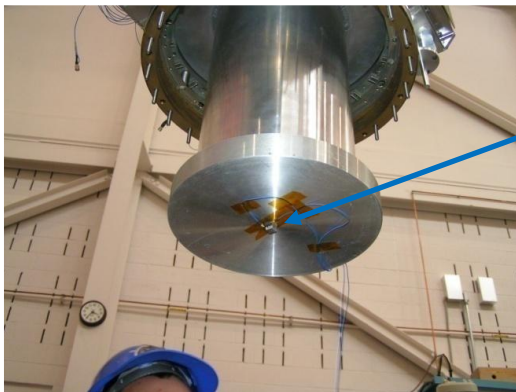
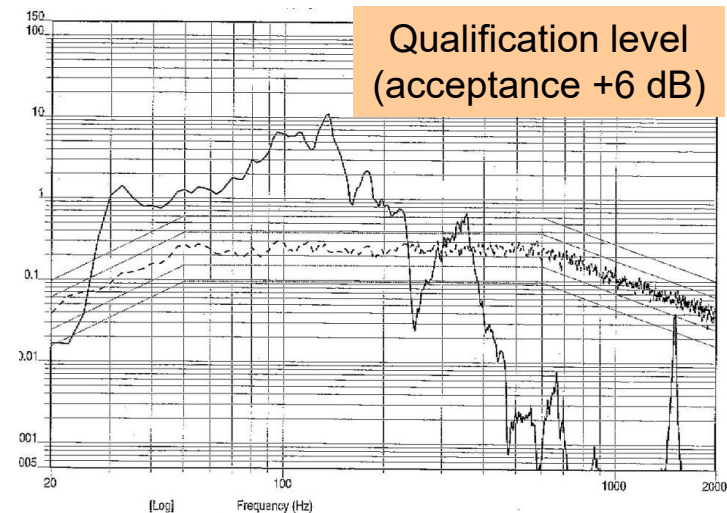
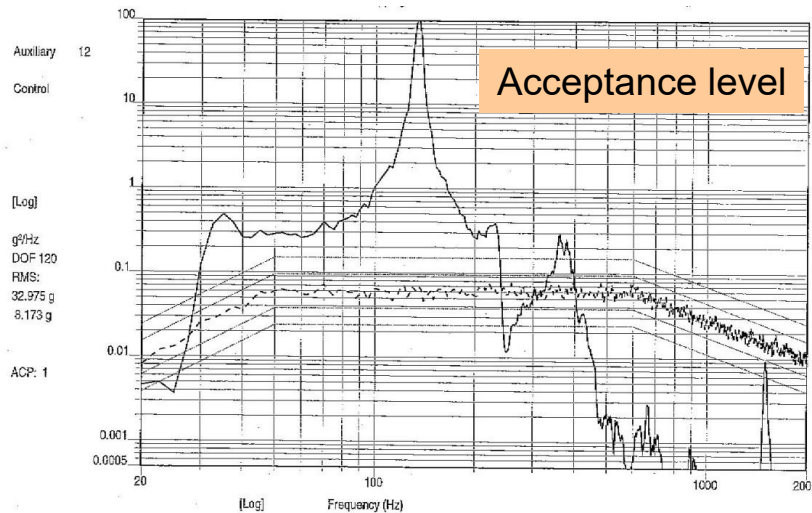
$$p = 0.028 \text{ g}^2/\text{Hz} \left(\frac{20 \text{ g}}{0.1 \text{ g}} \right)^2 = 1120 \text{ g}^2/\text{Hz}$$



The actual peak was only about $400 \text{ g}^2/\text{Hz}$, mostly because of increased damping at the higher level of excitation in the random vibration test:

Nonlinearity also tends to cause natural frequencies to drop at higher levels of excitation.

This structure (or the measurement) did not stay linear



Lateral acceleration was measured at the tip of a cantilevered, aluminum mass simulator that was attached with 12 bolts.

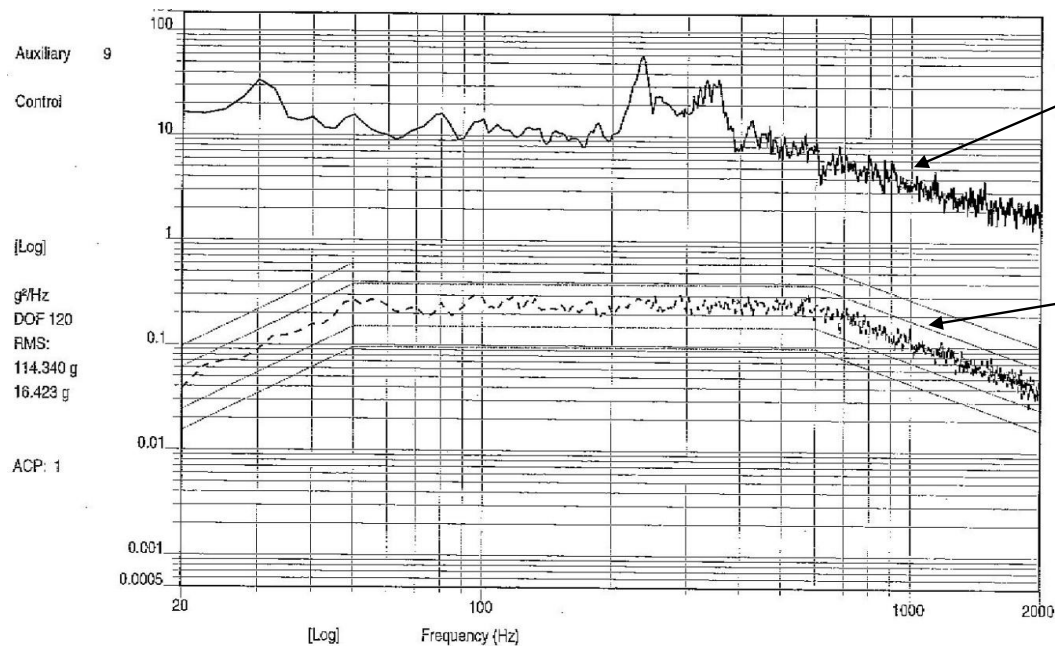
What do you think caused this nonlinearity?

An investigation is needed:

- Repeat sine sweep, compare data
- Physically inspect the test article
- Check bolt torques
- Check for accelerometer saturation*

*Sensor **saturation** occurs when the signal being measured exceeds the dynamic range of the sensor.

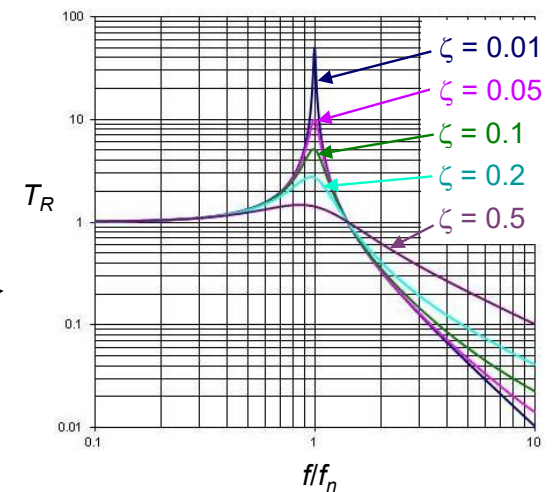
How Would You Interpret This Random-vibration Data?



Response acceleration
(114 g-rms)

Shaker input acceleration
(16.4 g-rms)

Hint: Remember the transmissibility curve. →

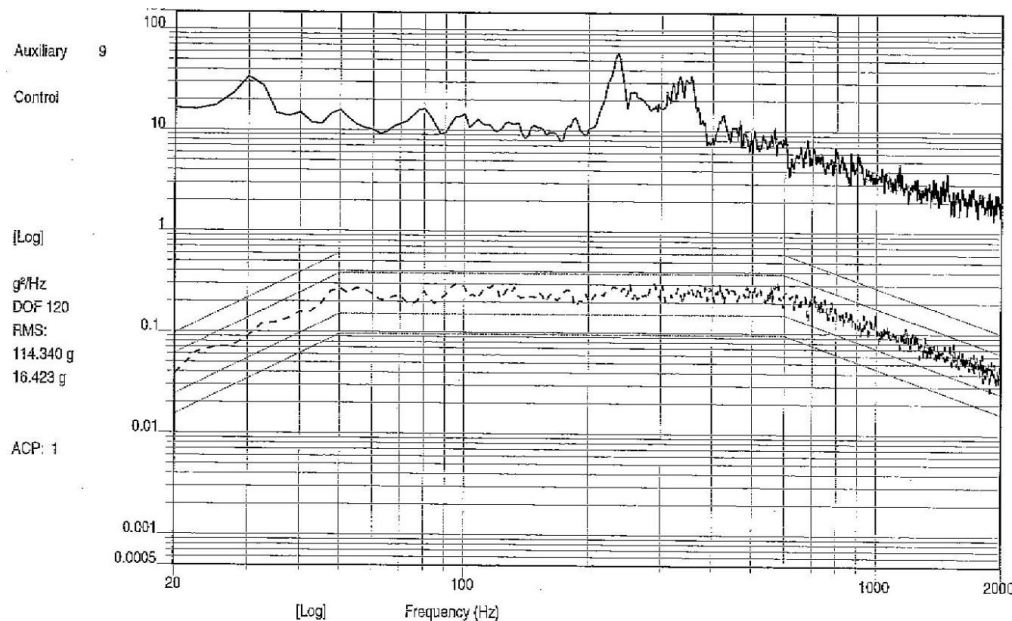


Answer to Question on Previous Page: Something is Wrong with the Data

The test article can't possibly respond this way.

In-axis response PSD would be slightly above the input at 20 Hz, with the response ramping up to the test article's fundamental frequency.

At frequencies above the fundamental frequency, the response plot should cross above and below the input up to 2000 Hz, with isolation likely at high frequencies.



When confronted with response data that make no sense, stop the test and investigate.

We'll look at another example of interpreting random vibration test data in the case history presented in Sec. 8.

Key Points of This Section

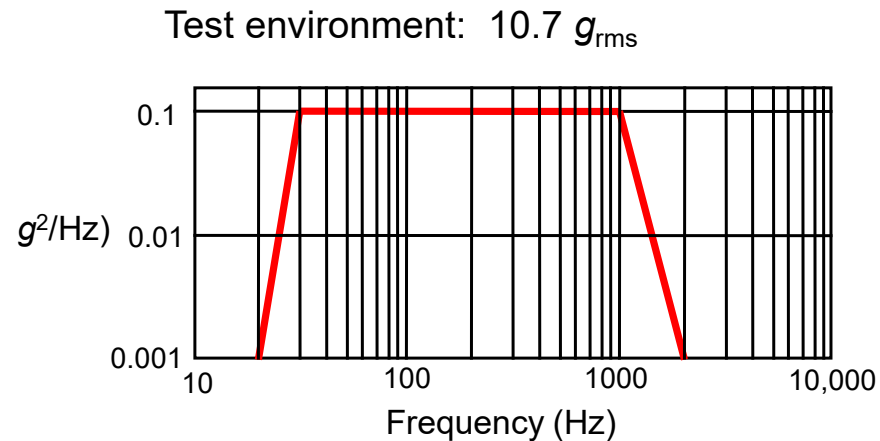
- Most failures caused by random vibration are fatigue failures.
 - Subject your spacecraft and its components to no longer duration of test exposure than is necessary.
- Peak acceleration usually will be well higher than 3σ (3 times RMS).
 - often 4 to 5 times RMS
 - If you're designing only to 3σ , make sure the analysis has other compensating conservatism
- If the test article has no modes below 2000 Hz, it will move nearly as a rigid body with the shaker and have an RMS acceleration nearly equal to the RMS of the input PSD.
 - Otherwise, its modes of vibration will respond to the random excitation, and the RMS will be different at each location within the test article.
- Identify and investigate anomalous test data before tearing down the test setup.
- Brush up on probability and statistics so that you can better understand random vibration and how test environments are derived.

Appendix to Section 5

Example Problem 5-b: Using Miles' Equation To Estimate Random Vibration Response

Example Problem 5-b: Predicted Response to Base-Driven Random Vibration

An electronics box will be subjected to a single-axis random vibration test.



Ref. 9, SSAM, Example 5.3

Problem statement: Derive a limit load for sizing the housing and the mounting bolts.

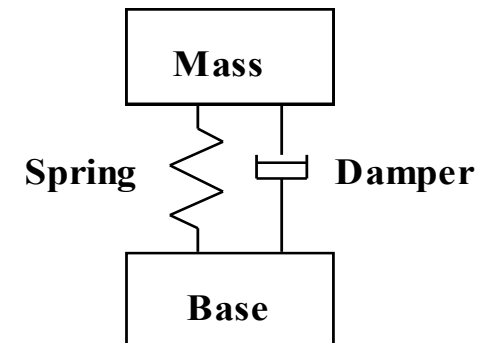
continued

Example Problem 5-b, continued

Solution:

Idealize the structure as a base-driven single-DOF system.

- We are assuming that the housing and mounting bolts are stressed mostly by the component's fundamental vibration mode, in which most of the mass moves in the same direction.
- Let's say we predict a fundamental frequency, f_n , of 43.6 Hz.
- We will assume the damping for this mode is 5% of critical (a reasonable estimate for an electronics box).



continued

Example Problem 5-b, continued

We will use a frequency-domain solution process: **Miles' equation**, an approximation based on the assumption of **white noise** input (constant PSD from zero to infinity Hz).

RMS response acceleration, $a_{rms} \cong \sqrt{\frac{\pi f_n W}{4\zeta}} \cong \sqrt{\frac{\pi Q f_n W}{2}}$ (Eq. 5-6)

Input PSD at the natural frequency f_n

Damping ratio

$$= \sqrt{\frac{\pi(43.6)(0.100)}{4(0.05)}} = 8.28 g_{rms}$$

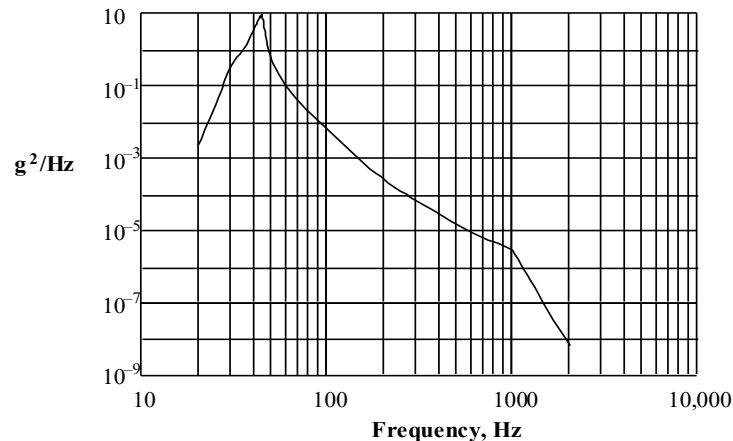
This is the standard deviation of the random response.

Note: The response would be $18.5g_{rms}$ if damping were 1% rather than 5%.

continued

Example Problem 5-b: Check the Results

Here is the computed response PSD based on the Fourier transform, which is the exact solution if the model is accurate:



From numerical integration of this curve, the RMS response acceleration is

$$\ddot{x}_{1_{rms}} = \sqrt{\int_0^{\infty} W_{\ddot{x}_1}(f) df} = 8.07 g_{rms}$$

vs. $8.28 g_{rms}$ from Miles' equation

Miles' equation works pretty well when ...

- there is a single dominant response mode;
- the specified PSD is relatively constant near the item's fundamental frequency;
- the specified PSD does not exceed, at any frequency, the value used in the equation;
- and damping < 10% of critical.

Remember: This is the standard deviation, σ , of random response, subject to a multiplier of at least 3. See earlier recommendations on appropriate limit loads for analysis.

6. Notching and Force Limiting

- Introduction to Notching and Its Technical Justification
- The Dynamic Absorber Effect
- Three Methods of Notching Related to Base Force
- STEP-4 Case History: How Notching Without Technical Rationale Led to Mission Failure
- Using Force Gages to Measure Base Force and Moment
- Force Limiting with the Semi-Empirical Method
 - Understanding the Semi-Empirical Method
 - Common Methods for Deriving C^2
 - Examples
- Response Limiting to Simulate Force Limiting
- Manual Notching to Simulate Force Limiting
- Other Technical Rationale for Notching
- Appendix 6A: The Simple TDFS Method of Deriving Force Limits
- Appendix 6B: “An Example that Raises Several Issues Associated with Force Limiting,” presented at the 2023 SCLV workshop

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Introduction to Notching

Notching means reducing the input levels at selected natural frequencies of the test article in order to avoid excessive loading.

- Applicable to random vibration testing and sine vibrate testing

Sounds like cheating, doesn't it?

That's exactly what I thought in 1992 when I first came across the subject of notching in a paper on vibration testing of the TOPEX/Poseidon spacecraft (VTSS Ref. 13, STDI Ref. 24):

"The inputs in the sine tests were notched to limit the responses at a number of critical positions to limits based primarily on structural capability."

Definitely cheating, I thought.

It's not much of a test if you reduce the input levels to ensure the response doesn't exceed the predicted capability!

It was several years later that I started to understand that notching can be technically justified ...

The Underlying Technical Justification for Notching

Testing on a shaker is not truly representative of the vibration that a small satellite or a component experiences in flight.

- Energy imparted by the shaker is limited only by the force rating of the shaker,
 - whereas the vibration energy during launch is usually much more limited.
- In test, acceleration is introduced uniformly across the mounting interface,
 - which is not the case for flight vibration, especially at high frequencies.

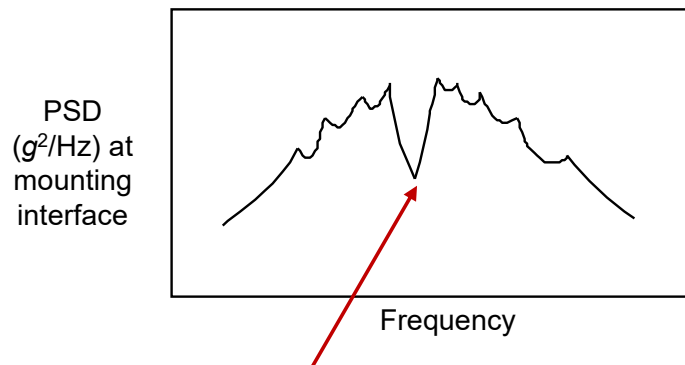
If we mount a single-degree-of-freedom system (SDFS) to a vibrating structure that has limited energy, some of the energy moves into the SDFS, especially at frequencies near and equal to the SDFS's natural frequency. The result is a natural reduction (notch) in acceleration at the base of the SDFS at that frequency.

- The same happens when we mount a spacecraft to a vibrating launch vehicle.
- Or a component to a vibrating spacecraft, missile, or aircraft.

Notching is technically justified when (1) it is done in a manner that simulates the notches that occur naturally in flight or (2) we otherwise know the test environment is excessive at certain frequencies.

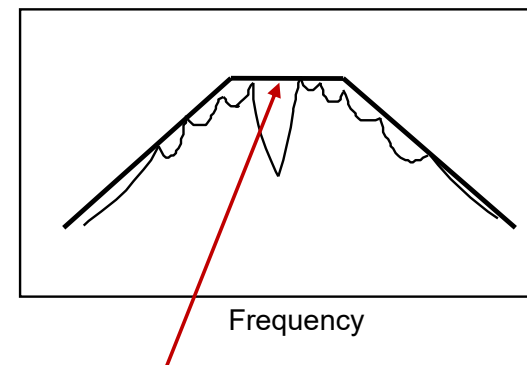
Notching Occurs Naturally in Flight

Hypothetical example of what happens in flight:



Vibration energy is limited, so at the mounted structure's fundamental frequency, some of the energy transfers from the host structure to the mounted structure, thus causing a reduction (notch) in the interface acceleration.

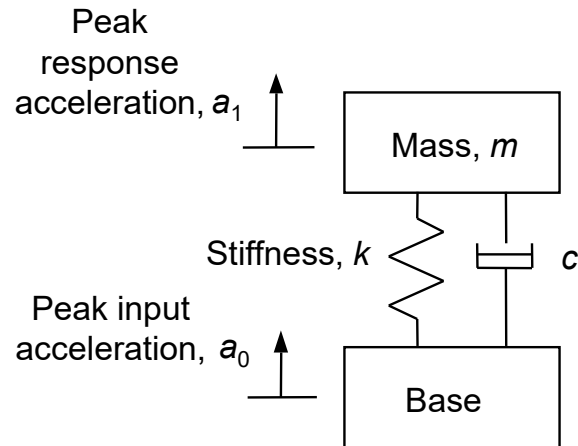
Then we draw a smooth envelope for test:



Forcing the shaker to achieve this acceleration at the mounted structure's fundamental frequency would excessively load the test article.

Notching is (or should be) an attempt to make the test more flight-like.

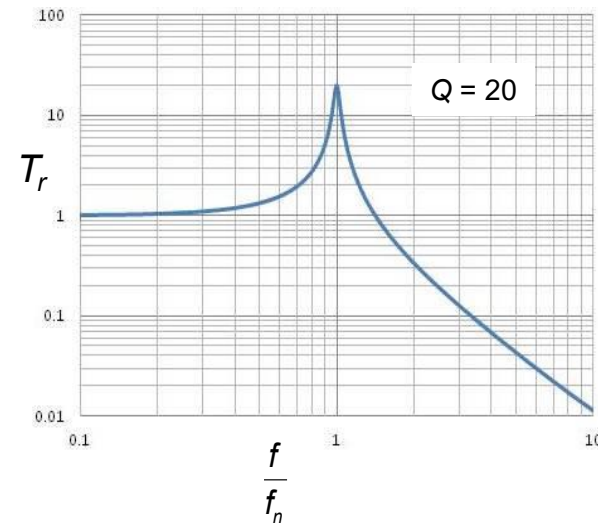
Review: Transmissibility for a Sinusoidally Base-driven Mass on a Spring (Single-Degree-of-Freedom System (SDFS))



➡
Sinusoidal base input at varying frequency f yields the transmissibility curve

$$\text{Transmissibility, } T_R = \frac{a_1}{a_0} = \sqrt{\frac{1 + \left(\frac{2\zeta f}{f_n}\right)^2}{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \left(\frac{2\zeta f}{f_n}\right)^2}}$$

Transmissibility curve:



(VTSS Eq. 1-14, STDI Eq. 5-1)

f = frequency of sinusoidal input
 f_n = natural frequency of grounded SDFS
 c = damping factor
 Q = quality factor $\approx 1/(2\zeta)$, where ζ is the damping ratio

Apparent Mass

The **apparent mass**, m_A , of a system is the frequency-dependent ratio of peak sinusoidal **base force** (force at mounting interface) to peak sinusoidal base acceleration, representing the system's **impedance** (resistance to sinusoidal base motion).

For a base-driven SDFS, let $m = \text{mass (lb or N)}$ ←

$a_0 = \text{base acceleration (g)}$

$a_1 = \text{acceleration of the mass (g)}$

$F = \text{spring force (base force) (lb or N)}$

As explained earlier, when acceleration is in g's, weight in force units (lb or N) is used for mass.

Given that transmissibility, T_r , is the ratio a_1/a_0 , as a function of frequency ...

$$F(f) = ma_1(f) = mT_r(f)a_0(f)$$

So the apparent mass for an SDFS is

$$m_A(f) = \frac{F(f)}{a_0(f)} = \frac{mT_r(f)a_0(f)}{a_0(f)} = mT_r(f) \quad (\text{Eq. 6-1a}) \quad (\text{units: lb or N})$$

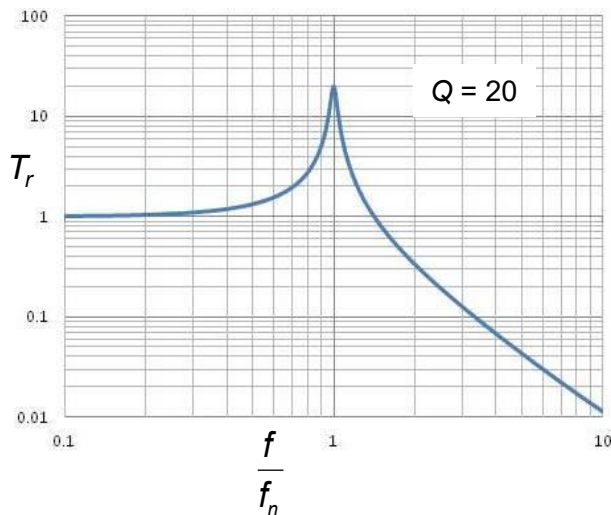
For any item, such as a test article, the apparent mass is

$$m_A(f) = \frac{F(f)}{a_0(f)} \quad (\text{Eq. 6-1b})$$

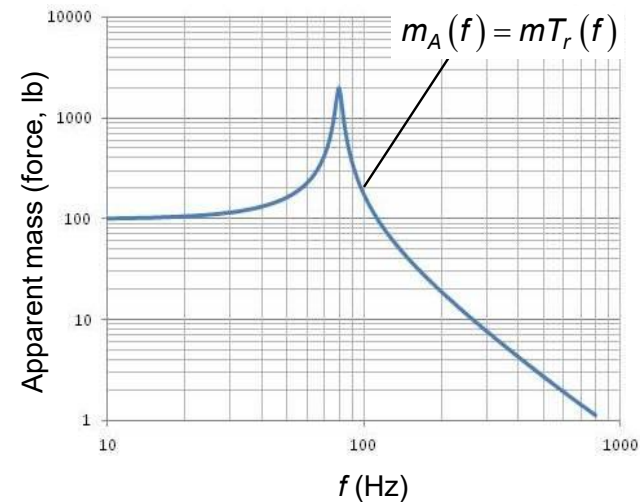
To get force units for m_A , we're actually dividing F by the unitless load factor. Alternatively, you can think of the units for apparent mass as being lb/g instead of lb. But it's traditional to use lb (or N) for m_A .

Apparent Mass for a Base-driven SDFS

Let's set the mass m to 100 lb, and the natural frequency, f_n , when grounded at the base of the spring to 80 Hz, with a 1 g sinusoidal input acceleration from 10 to 800 Hz:



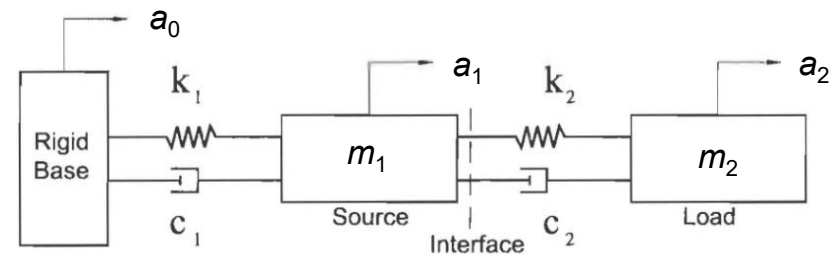
Multiplying the transmissibility function, $T_r(f)$, by mass yields the apparent mass, $m_A(f)$, which is the spring (base) force caused by 1 g sinusoidal base acceleration.



Now Consider a Two-Degree-of-Freedom System (TDFS)

For sinusoidal or random acceleration input at the base of a TDFS:

The response of the load—hence the force at the interface between the load and the source—is highest when the natural frequency of the load by itself, f_{n2} , is the same as the natural frequency of the source by itself, f_{n1} (when the load is not present).



$$f_{n2} = f_{n1} = \frac{1}{2\pi} \sqrt{\frac{k_1}{m_1}} = \frac{1}{2\pi} \sqrt{\frac{k_2}{m_2}}$$

For this case, the two natural frequencies of the coupled system are

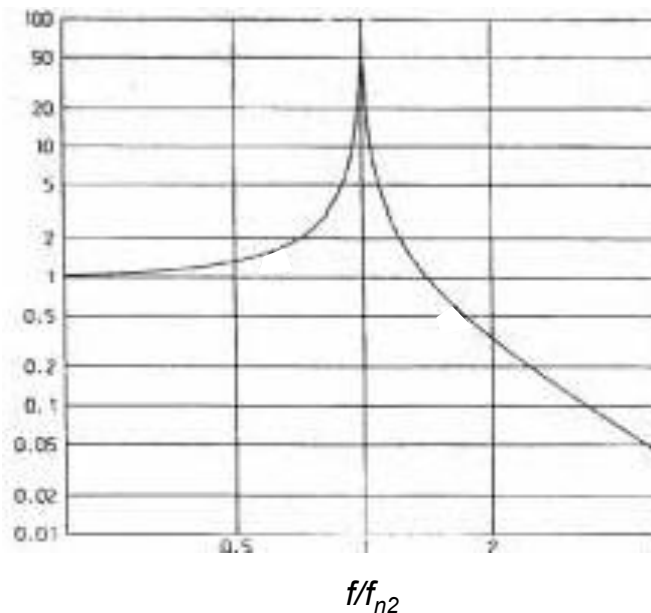
$$f_{\text{sys-1}}, f_{\text{sys-2}} = f_{n2} \sqrt{1 + \frac{m_2 / m_1}{2} \pm \sqrt{m_2 / m_1 + \frac{(m_2 / m_1)^2}{4}}} \quad (\text{Eq. 6-2})$$

(VTSS Ref. 14, STDI Ref. 25: Hartog)

Example: The Dynamic Absorber Effect

Start with an SDFS (“load”) with mass $m_2 = 1$ lb, spring rate k_2 , natural frequency f_{n2} , and $Q = 50$. If we drive it at its base with 1 g sinusoidal acceleration at varying frequency f , we get the apparent mass, equal to the spring force (base force):

Apparent
mass of the
load, m_{A2} ,
lb

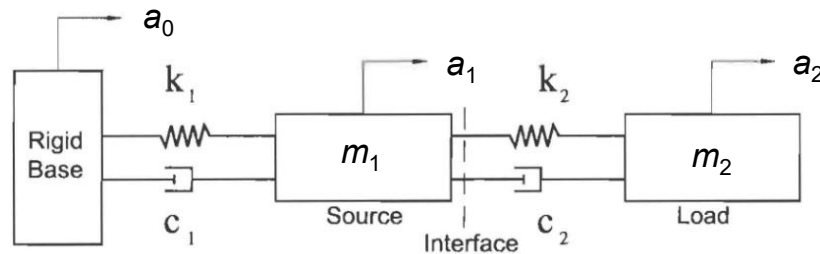


This example is from VTSS Ref. 15, STDI Ref. 26, “Force Limited Vibration Testing Monograph”, based on VTSS Ref. 14, STDI Ref. 25 (Hartog)

Continued

Example, Dynamic Absorber Effect (continued)

Now we create a TDFS by mounting the SDFS (load) onto another SDFS (source), with $m_1 = m_2$, $k_1 = k_2$, and $Q_1 = Q_2 = 50$:

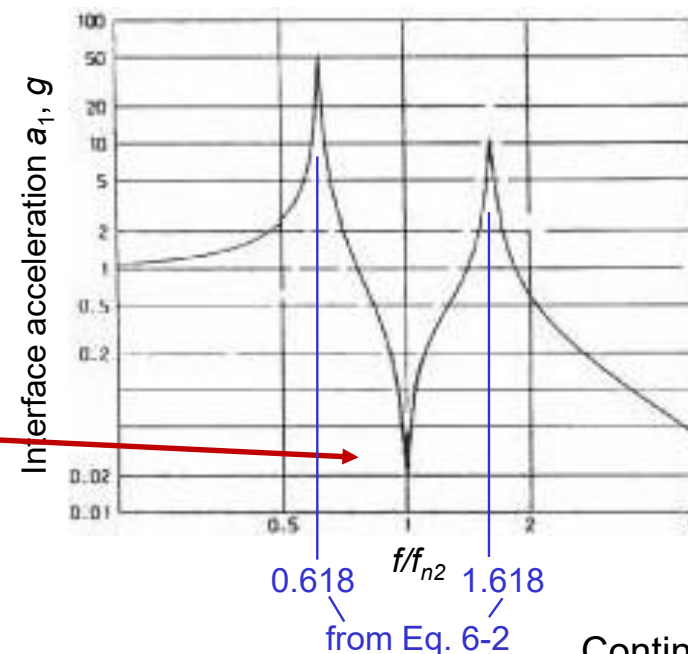


Note the notch (valley) at $f = f_{n2}$, the natural frequency of the load by itself.

This notch is the **antiresonance**, which is often referred to as a **zero** because the interface acceleration would be zero if there were no damping. This zero exists at f_{n2} even if m_2 is much less than m_1 and if f_{n1} is different than f_{n2} !

With damping, notch depth is $1/Q$.

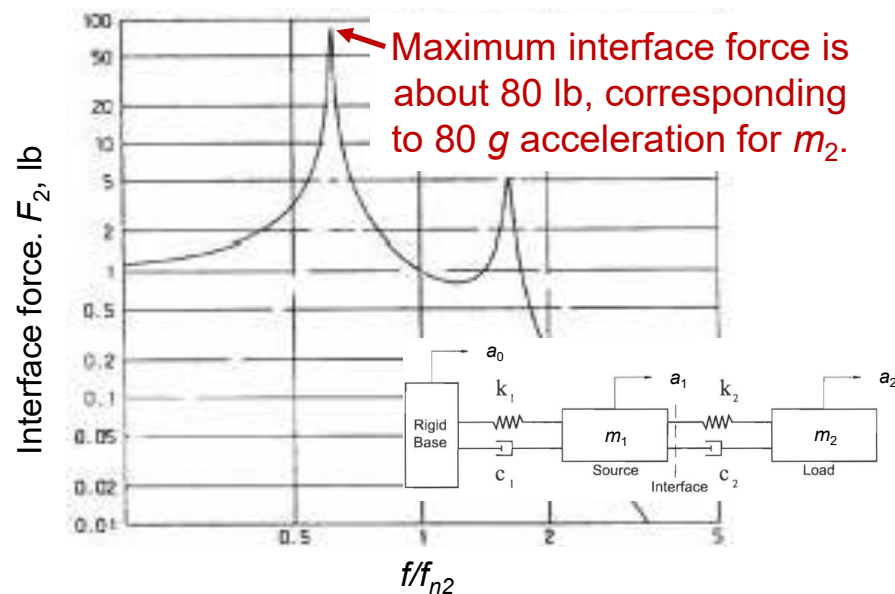
The rigid base imparts 1 g acceleration at varying frequency, resulting in the following plot of interface acceleration (acceleration of m_1 , which is the base of the k_2 spring):



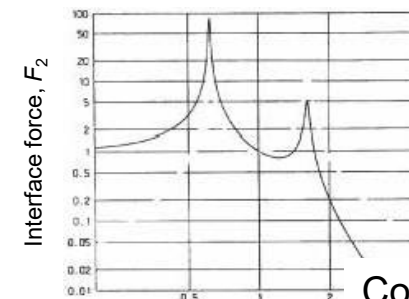
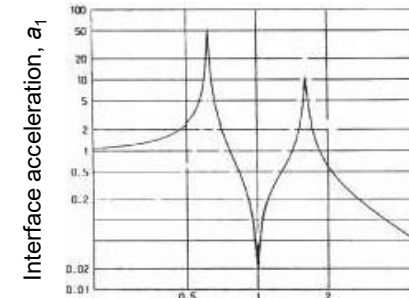
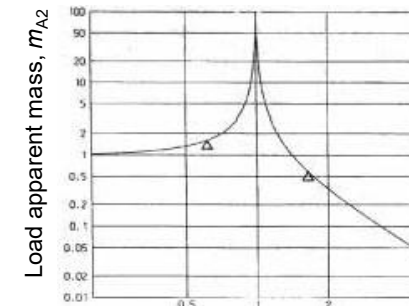
Continued

Example, Dynamic Absorber Effect (continued)

Here's the interface force, F_2 , (force in spring k_2) for the TDFS with 1 g sinusoidal base acceleration:

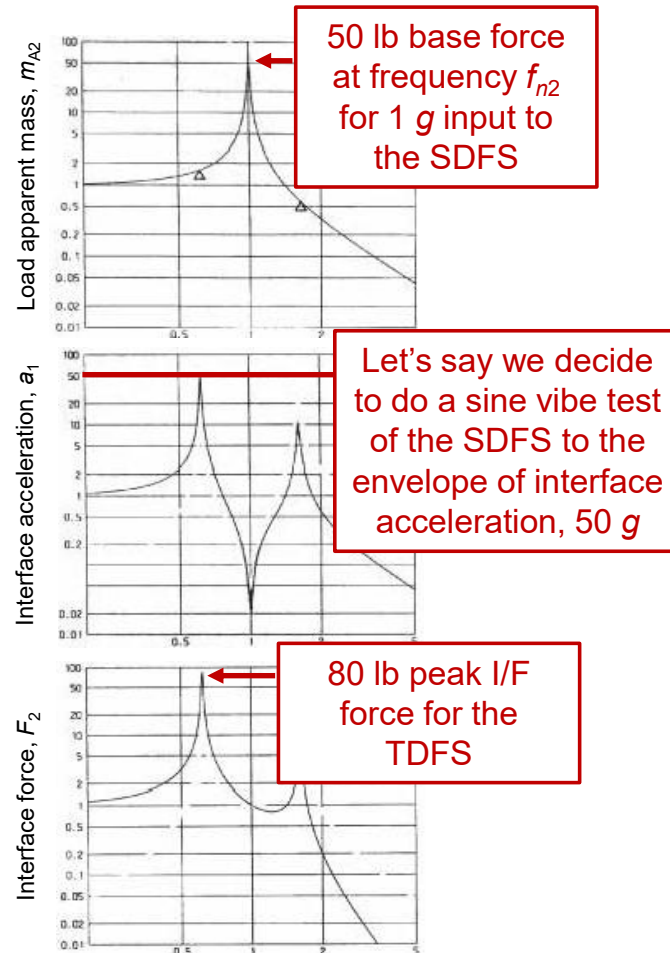


$$F_2(f) = m_{A2}(f)a_1(f) \quad (\text{Eq. 6-3})$$



Continued

Example, Dynamic Absorber Effect: Summary



Without notching, the peak base force in the SDFS test will be 50 g times 50 lb/g = 2500 lb at frequency f_{n2} ...
... vs. 80 lb for the TDFS!

Three Methods of Notching Related to Base Force

Force limiting

- Controlling the shaker by force, with force sensors at the base of the test article, instead of acceleration within any frequency band in which the measured base force otherwise would exceed a prescribed limit.

Response limiting

- Controlling the shaker with a response accelerometer mounted to the test article instead of base acceleration within any frequency band in which the measured response otherwise would exceed a prescribed limit.

Manual notching

- Reduce input levels near a key modal frequency, creating a smooth notch that the shaker can control, as needed to keep either a response acceleration or the predicted base force from exceeding a prescribed limit.

Regardless of the method, notching should be used only with valid technical justification that the test adequately envelops the flight environment.

Wanting to keep component responses during system-level testing from exceeding predicted capability or environments to which the components were tested is NOT valid technical rationale!

Example of How Notching Without Technical Rationale Led to Mission Failure

STEP-4/Pegasus case history (1997)

VTSS Ref. 16, STDI Ref. 27

- Following launch on a Pegasus XL, no contact could be made with the Space Test Experiments Platform (STEP)-4 spacecraft.
 - A DOD Space Test Program (STP) spacecraft
 - Weight: 870 lb (395 kg mass)
- Evidence shows the Pegasus placed STEP-4 in the proper orbit and separated cleanly from it.
- The failure was attributed to overloading during the captive-carry part of flight (Pegasus held by an L-1011 aircraft).
- The test program was deemed inadequate.



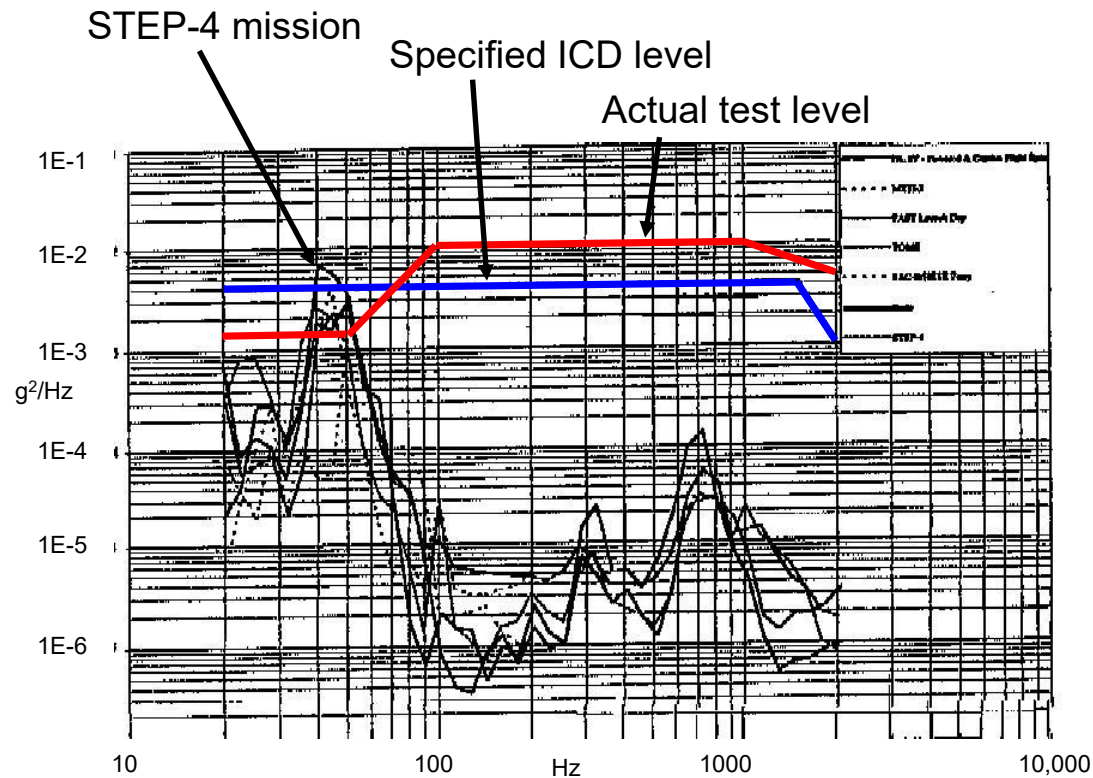
(Continued)

Background to the STEP-4 Failure (same reference)

- After several spacecraft components failed in test, component random vibe environments were reduced “because the space vehicle manufacturer felt the testing levels were unnecessarily high.”
 - Reductions made at low frequency (20 to 500 Hz).
- The vehicle was tested to acceptance (maximum expected) levels only (no margin).
- Vehicle random-vibe environments were reduced at low frequency to keep component accelerations below previous test levels.
- STEP-4 had a key mode of vibration near 40 Hz.
- The 40-Hz mode was clearly excited by lateral input in test.
- During captive carry, Pegasus XL introduces high levels of lateral vibration at the payload interface between 40 and 50 Hz.
- There was no coupled loads analysis for the captive-carry configuration. (CLA is not normally done for aircraft.)

Conclusion: STEP-4 had not been adequately tested in the 40-50 Hz range.

STEP-4 Failure (continued)



Comparison of flight data (captive carry) with test levels

Summary:

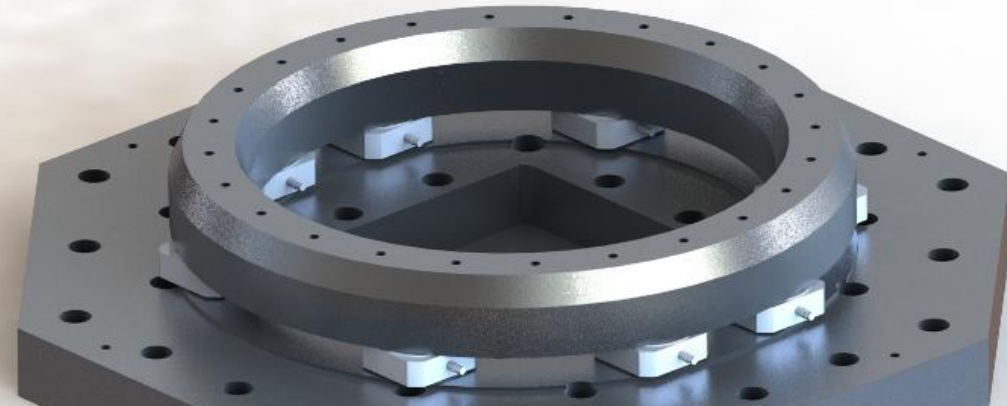
- Captive carry excites a 40-50 Hz mode in Pegasus XL.
- The spacecraft developer apparently was not aware of this excitation.
- Notching apparently was without technical rationale.

Make sure you understand the flight environment and how your spacecraft will respond before notching the test environment!

Using Force Sensors to Measure Base Force and Moment in Test

- Force sensors can be sandwiched between the test article and its mounting structure or between two fixtures.
 - I refer to the latter assembly as a **force-limiting fixture (FLF)**.
- Putting force sensors directly under the test article is preferable when practical (depends on number and spacing of mounting bolts).
- Use a junction box to sum forces and moments.

Example: FLF for ESPA-class small satellites (15"-dia. I/F bolt cir.)



The sensors should be the piezoelectric ring-shaped types, with the mounting bolt passing through and preloading the ring. These gages are very stiff.

Measuring base force and moment helps justify notching even if the test will not be directly force limited.

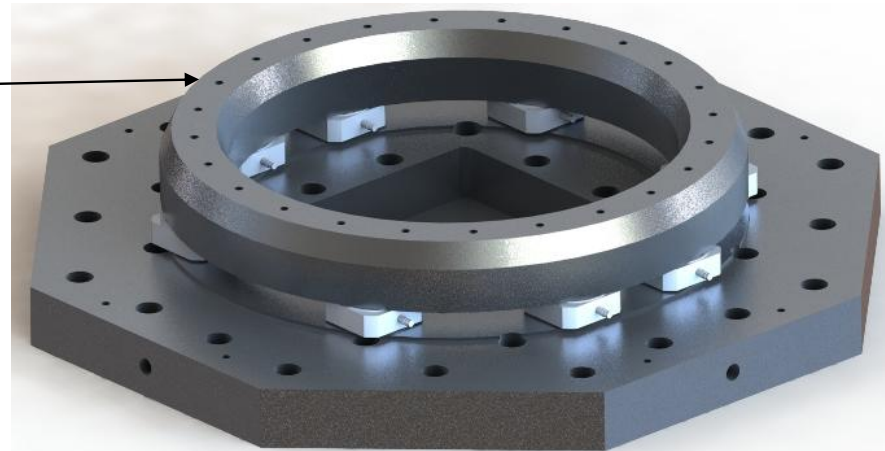


Triaxial force sensor from PCB Piezotronics

Considerations in Designing a Force-Limiting Fixture

- As is usually the case for test fixtures, the FLF as an assembly should be stiff enough not to significantly affect the test article's modes of vibration.
- The mass of the upper fixture should be less than about 10% of the test article's mass so that it doesn't significantly affect the measured force.
 - Even then, that mass can contribute most of the measured force at high frequencies, where the test-article mass may be isolated. Accordingly, when using an FLF, plan not to force limit above 1000 Hz or so (depends on the test article and the FLF design).
- See Sec. 4 of NASA-HDBK-7004C (VTSS Ref. 17, STDI Ref. 28) for additional guidance.

Upper fixture (ring above force sensors)



Force Limiting for Random Vibration

Main step: Derive a justifiable **force limit** as a function of frequency.

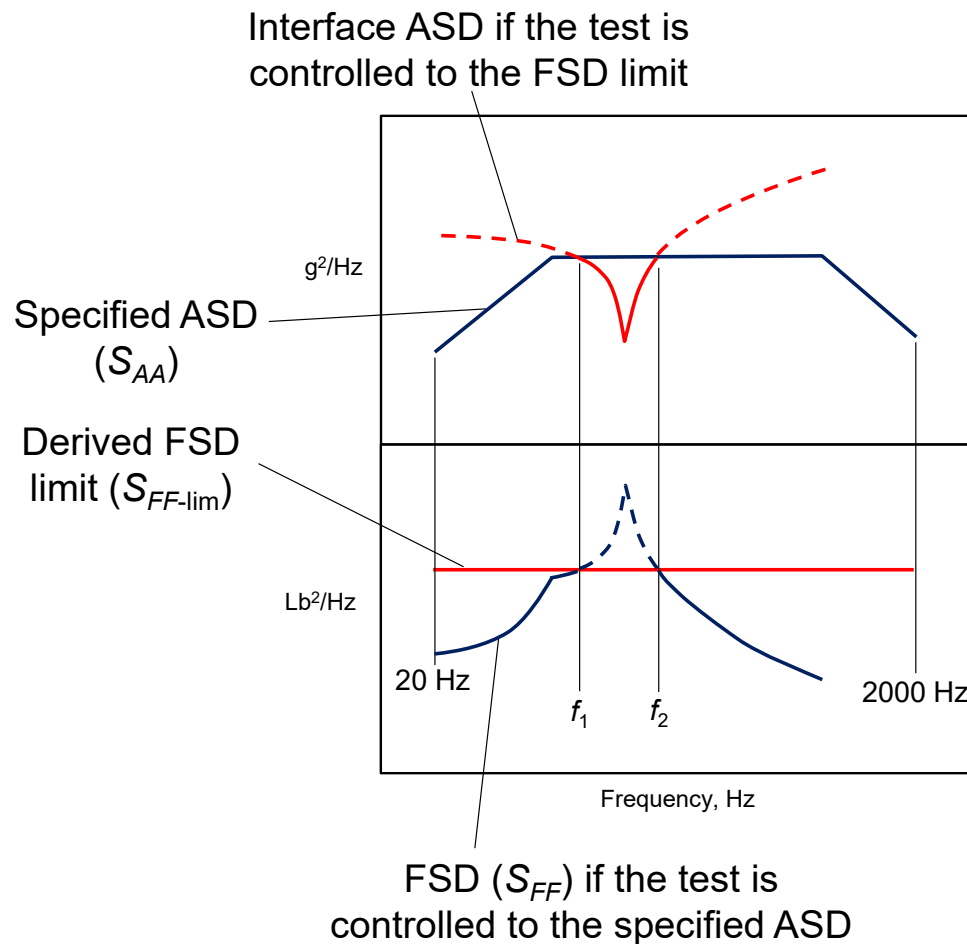
- In terms of a force PSD (referred to herein as **force spectral density, FSD** (S_{FF})), in units Lb^2/Hz or N^2/Hz , at the base of the test article
- Let's call this force limit an **FSD limit** ($S_{FF\text{-lim}}$), and in test it applies at the I/F with the test fixture.

The process is similar for a sine vibrate test, but the force limit is in terms of force (lb or N).

The FSD limit should be based on the specified acceleration PSD (a.k.a. **ASD**) (S_{AA}) and the relative impedance of the test article and the flight host structure

- Recall: impedance is resistance to sinusoidal base motion, as quantified by apparent mass.

What the Shaker's Control System Does in a Force-limited Random Vibration Test



The control system software automatically switches between ASD and FSD, as needed to ensure neither the FSD limit nor the specified ASD is exceeded.

- From 20 Hz to f_1 , the shaker will be controlled to the specified ASD.
- From f_1 to f_2 , the shaker will be controlled to the FSD limit.
- Above f_2 , the shaker will again be controlled to the specified ASD.

NASA's Semi-Empirical Method for Force Limits

The **semi-empirical method** from NASA-HDBK-7004C is the most commonly used formalism (rather than an actual method) for force limits.

For sinusoidal vibration, the force limit, $F_{\text{lim}}(f)$, is ...

$$\text{For } f < f_b, \quad F_{\text{lim}}(f) = Cma_0(f) \quad (\text{Eq. 6-4})$$

$$\text{For } f \geq f_b, \quad F_{\text{lim}}(f) = Cm \left(\frac{f_b}{f} \right)^n a_0(f) \quad (\text{Eq. 6-5})$$

For random vibration, the force limit is expressed as an FSD limit, $S_{FF-\text{lim}}(f)$:

$$\text{For } f < f_b, \quad S_{FF-\text{lim}}(f) = C^2 m^2 S_{AA-0}(f) \quad (\text{Eq. 6-6})$$

$$\text{For } f \geq f_b, \quad S_{FF-\text{lim}}(f) = C^2 m^2 \left(\frac{f_b}{f} \right)^{2n} S_{AA-0}(f) \quad (\text{Eq. 6-7})$$

where f = frequency

f_b = break frequency (see next page)

$a_0(f)$ = base sinusoidal acceleration (g) at frequency f

m = total mass of the test article

C and n = constants without units

$S_{FF-\text{lim}}(f)$ = base FSD limit (lb^2/Hz) at frequency f

$S_{AA-0}(f)$ = the base random vibration ASD (g^2/Hz) at frequency f

Understanding the Semi-Empirical Method

Recall Eq. 6-1: Apparent mass, $m_A(f) = \frac{F(f)}{a_0(f)} = mT_r(f)$ for a mass-spring system (SDFS)

Base force
Base acceleration
Mass
Transmissibility

Recognizing that Q is the peak value of the transmissibility function, for a sinusoidally base-driven SDFS the maximum base force is

$$F_{\max} = F(f_n) = Qma_0(f_n) \quad (\text{Eq. 6-8})$$

Base acceleration at the natural frequency of the SDFS

In Eq. 6-4 for defining a force limit, C replaces Q to account for the dynamic absorber effect:

$$F_{\text{lim}}(f) = Cma_0(f)$$

C is essentially a knockdown factor.

Continued

Understanding the Semi-Empirical Method (continued)

For random vibration, the force limit is expressed as an FSD limit, $S_{FF-lim}(f)$:

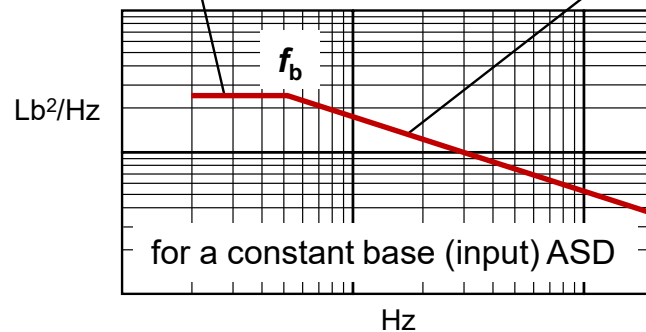
Up to the break frequency,

$$S_{FF-lim}(f) = C^2 m^2 S_{AA-0}(f)$$

S_{AA-0} is the base ASD

Above the break frequency,

$$S_{FF-lim}(f) = C^2 m^2 \left(\frac{f_b}{f} \right)^{2n} S_{AA-0}(f)$$



The break frequency, f_b , is the frequency above which the apparent mass of the test article “rolls off” (each peak lower than the previous one).

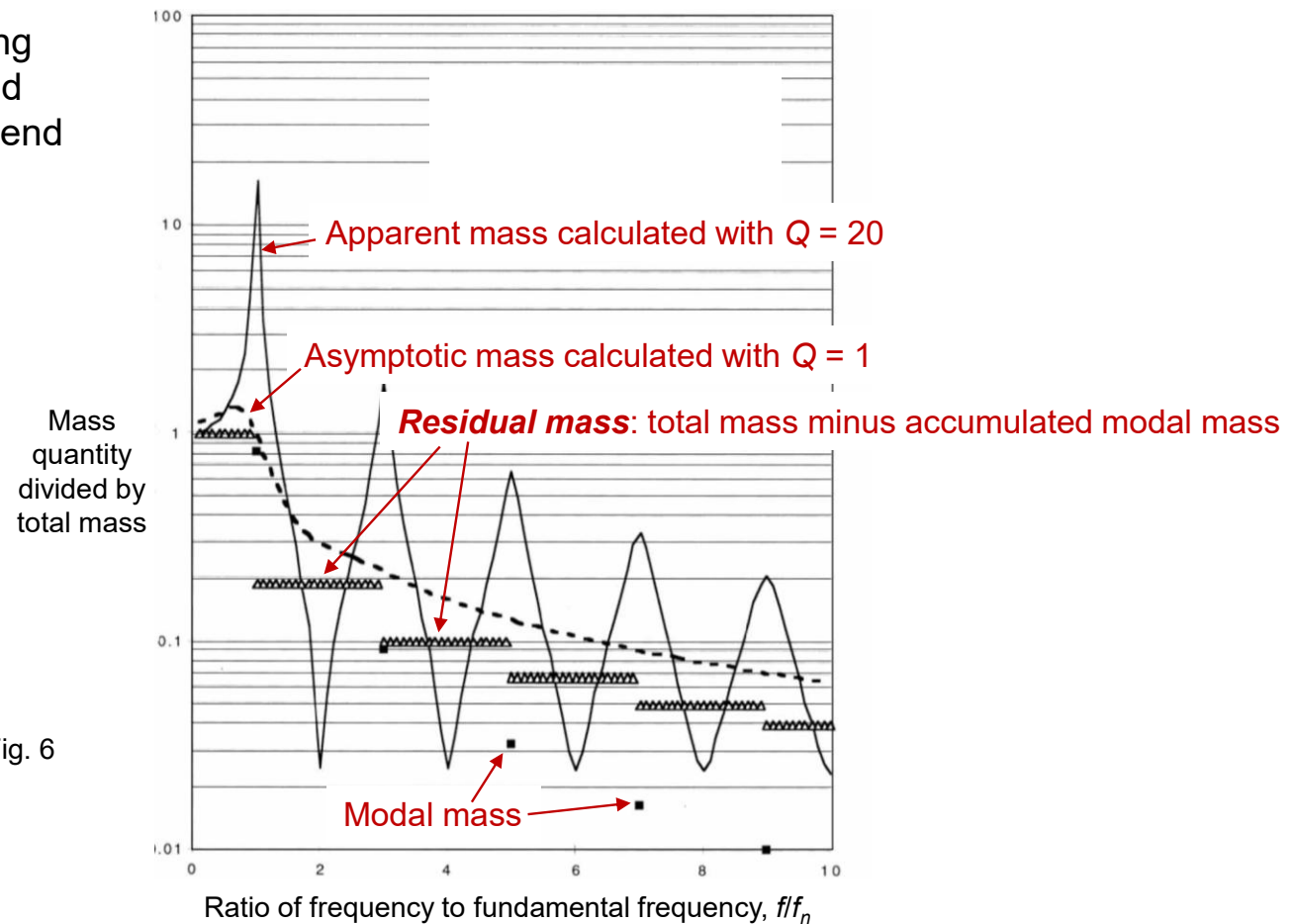
- Often set equal to the fundamental frequency of the test article.
- But, if the apparent mass (or response FSD) shows two or more nearly equal peaks, the break frequency should be the highest frequency of those peaks.

$2n$ sets the slope of the rolloff.

The value for n should be set to match the rolloff in the **asymptotic mass**, which is a smooth-line average of peaks and valleys in the apparent mass plot

Example of Asymptotic Mass

Longitudinally vibrating
rod excited at one end
and free at the other end



Adapted from
NASA-HDBK-7004C, Fig. 6

Example: Base-driven SDFS

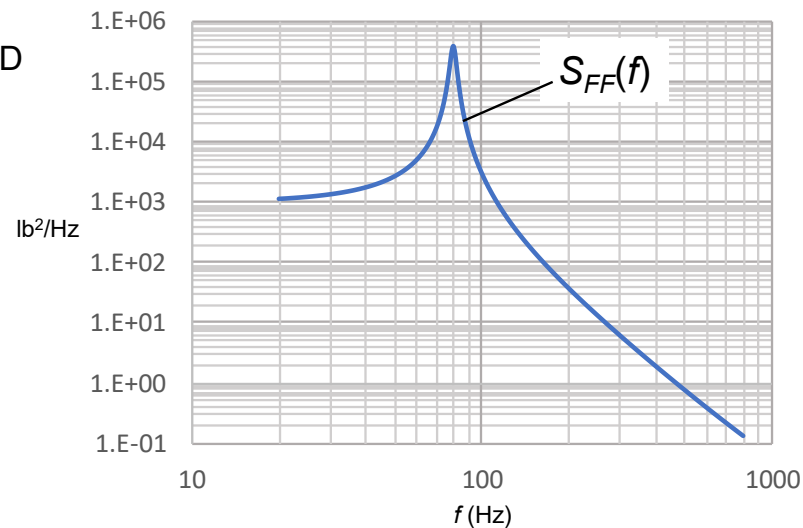
Semi-Empirical Force Limits for Random Vibration

SDFS (mass $m = 100$ lb, natural frequency $f_n = 80$ Hz, and quality factor $Q = 20$), base driven with random vibration, $S_{AA}(f) = 0.1 \text{ g}^2/\text{Hz}$ from 20 to 800 Hz.

The response of the mass causes a force spectral density (FSD), $S_{FF}(f)$, in the spring:

$$S_{FF}(f) = m_A^2(f) S_{AA-0}(f) \quad (\text{Eq. 6-9})$$

Base FSD Apparent mass (Eq. 6-1) Base ASD



continued

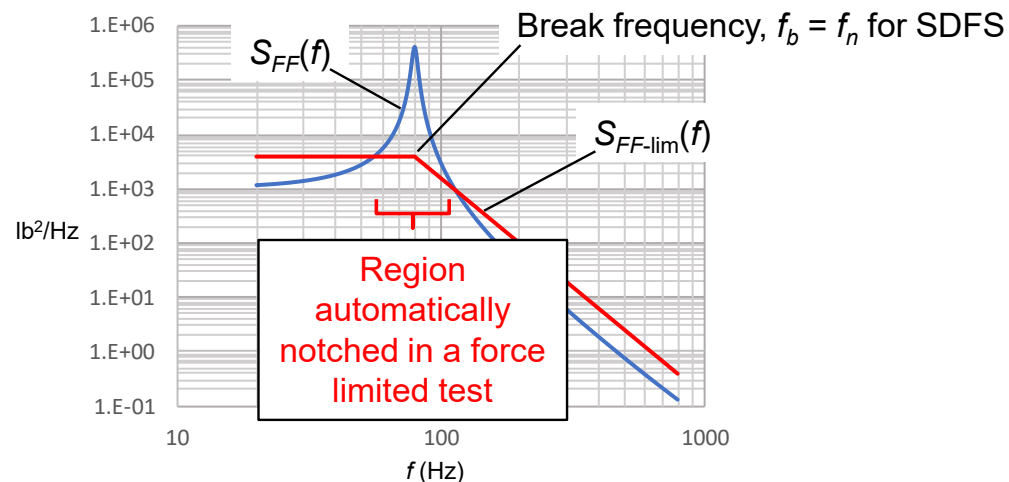
Example, Base-driven SDFS, continued: Semi-Empirical Force Limits for Random Vibration

We calculate an FSD limit using Eqs. 6-6 and 6-7:

$$\text{For } f < f_b, \quad S_{FF-\text{lim}}(f) = C^2 m^2 S_{AA-0}(f)$$

$$\text{For } f \geq f_b, \quad S_{FF-\text{lim}}(f) = C^2 m^2 \left(\frac{f_b}{f} \right)^{2n} S_{AA-0}(f)$$

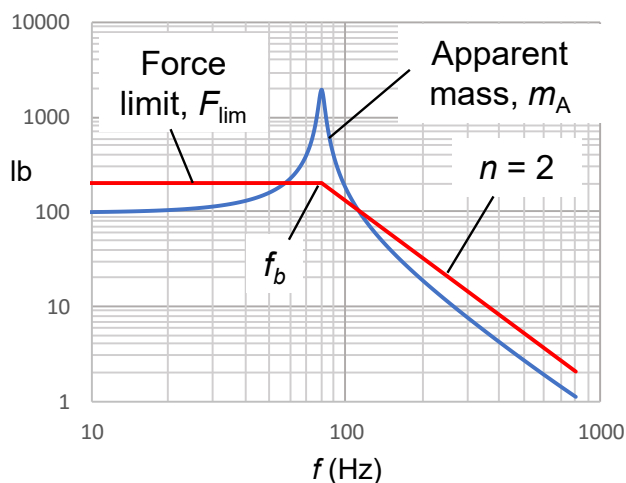
For arbitrarily (for this example) selected $C = 2$ ($C^2 = 4$) and $n = 2$ ($2n = 4$), here's what the FSD limit looks like for a base ASD, $S_{AA-0}(f)$, of $0.1 \text{ g}^2/\text{Hz}$ from 20 to 800 Hz:



continued

Example, Base-driven SDFS, conclusion:
Comparison of Force Limits with $C = 2$ ($C^2 = 4$)

1g sinusoidal input
(Eqs. 6-4 and 6-5)



0.1 g^2 /Hz random vibration input
(Eqs. 6-6 and 6-7)

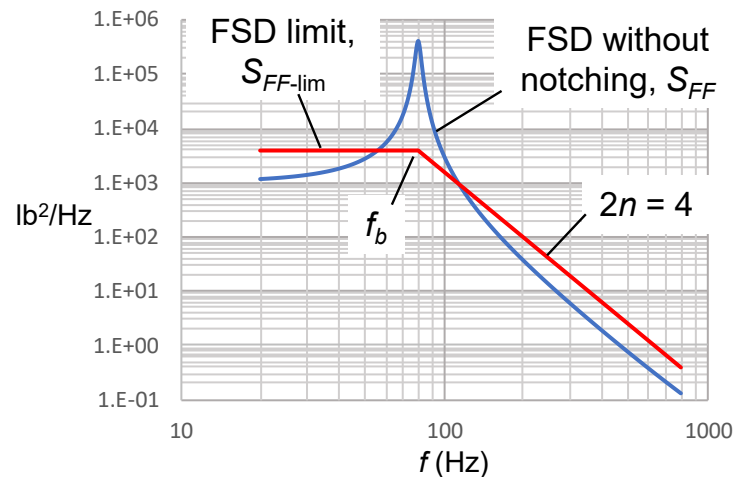


Table 6-1: Common Methods for Deriving C^2

Method	Comments
1. Assume a value based on studies that show C^2 is usually between 2 (for large spacecraft) and 5 (for relatively lightweight items) (VTSS Refs. 18 & 19, STDI Refs. 29 & 30, which draw from multiple other references).	Technical justification is not as solid as for other methods, and there is risk of under-testing lightweight items. C^2 can be greater than 10 (VTSS Refs. 18 & 19, STDI Refs. 29 & 30).
2. Simple two-degree-of-freedom system (TDFS) method (described in VTSS Refs. 15 & 17, STDI Refs. 26 & 28; overview and example in Appendix 6A herein)	Not usually applicable for full spacecraft because the method requires frequency-dependent impedance of the mounting structure, which is not often available from launch vehicle (LV) organizations.
3. Complex TDFS method (VTSS Ref. 15, STDI Ref. 26)--intended as an improvement to the simple TDFS method	Also requires impedance of mounting structure, so again not normally used for full spacecraft.
4. Derive a value from measured or predicted acceleration and force at the interface between the item in question and its mounting structure when the coupled system is subjected to an environment that is representative of the flight vibration environment.	Also requires impedance of mounting structure, so again not normally used for full spacecraft. It can be of use, though, for separate testing of a spacecraft instrument or other component (example included in Appendix 6B herein).
5. Set the value as needed to make the maximum base force or moment approximately equal to--but not exceeding--the flight limit load (maximum expected load during the mission). (See upcoming examples.)	Often the most practical method of deriving force limits for full spacecraft when a model of the LV mounting structure is not available. Requires that the limit load encompass all flight environments (discussed in an upcoming slide). Can lead to excessive notching, as addressed in upcoming Example Problem 6-a and Appendix 6B.

Of these methods, only 1 and 5 are usually available for force limiting a full spacecraft.

Common Strategy: Method 5 from Table 6-1, A Practical Shortcut to Deriving C^2

Methods 2 – 4 in Table 6-1 all require impedance data—either from testing or from finite element models (FEMs)—for the “load” and “source”, e.g., component and spacecraft to which the component will be mounted.

When a FEM is not available for the mounting structure (such as the launch vehicle), a practical option is to set the value of C^2 as needed to keep the maximum* base force or moment from exceeding a specified or derived limit load (maximum expected load for the mission). (VTSS Refs. 18 & 19, STDI Refs. 29 & 30)

- Usually corresponding to a specified limit quasi-static load.
- Intended to envelop the full effects of launch loads.

This approach is technically justified as long as ...

- the limit load accounts for all load contributors (see next page) ...
- and resulting notches are not excessive as a result of compensating for conservatism in test levels at other frequencies, e.g., from the test levels being an envelope of peaks and valleys. (See the example in Appendix 6B).

* “Maximum” usually means 3σ , but there’s a reasonable argument (although one that’s harder to sell) that 4 to 5σ should be used instead, resulting in deeper notches, especially for testing items made of brittle materials.

Setting C^2 to Ensure the Limit Load is Not Exceeded in Test

When deriving C^2 based on a specified or derived limit load rather than on relative impedance, **we have to be sure the limit load accounts for all contributors to flight loads.**

Despite what many think, the results of traditional coupled loads analysis (CLA) are **NOT** the maximum expected (limit) loads for a SmallSat or its components.

CLA does not include the full effects of flight random vibration.

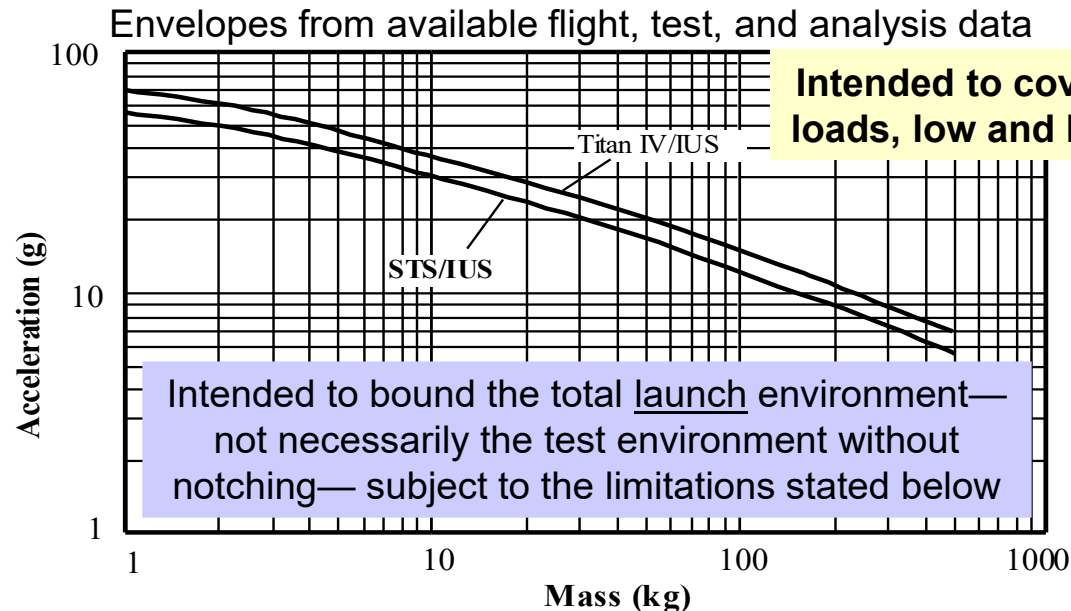
- CLA includes modes of vibration only up to the **truncation frequency**, which is often 50 Hz but depends on the launch services provider (LSP).
 - Modes above that frequency are truncated from the coupled math model.
- Thus, the effects of random vibration introduced at the LV interface above the truncation frequency are not included. These loads can be significant for SmallSats.

Most likely, the quasi-static loads specified by the LSP for your SmallSat are the full limit loads for the primary structure, including the effects of random vibration.

- But this may not be the case. Many new LVs are being developed, with LSPs learning on the job.
- Don't make assumptions; ask the LSP if the specified quasi-static loads include the effects of random vibration.

For SmallSat instruments and other components, the most practical way to derive limit loads is by use of an approved mass-acceleration curve (next page).

Example Mass Acceleration Curves (MACs) for Limit Loads



MACs have been developed for other LVs, but some are company proprietary.

If you can't get an approved MAC for your program, try to get approval for using the higher (Titan IV) curve shown here, which is the highest I've seen.

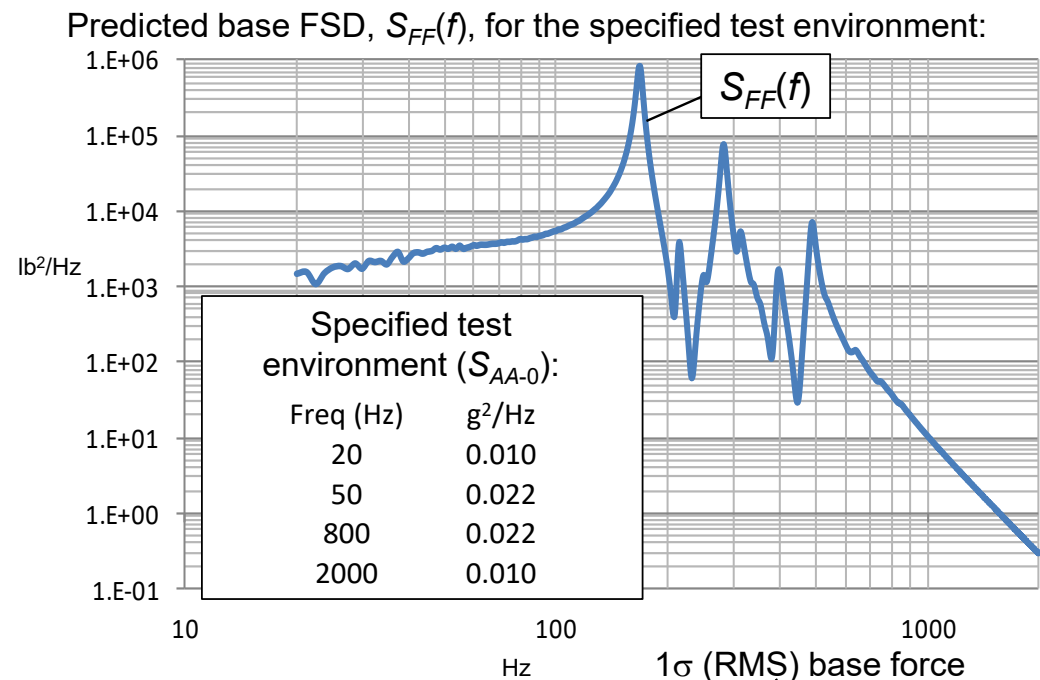
- Instructions:
- Applies in any direction, one axis at a time
 - In the thrust axis, add 3.9g for T-IV and 1.5g for STS (quasi-static)
- Limitations:
- For “lumped-mass” (compact) components and subsystems only
 - For appendage-like components having a statically determinate mount
 - For preliminary design only (although some programs use a MAC for final design and test as well)

VTSS Ref. 25, STD I Ref. 34: original MAC paper at JPL.

Example Problem 6-a: Force Limiting with Method 5

Problem statement: Design a force (FSD) limit so that the 3σ base force doesn't exceed the specified or derived limit load.

- Test-article weight = 362 lb (1610 N) ($m = 362$ lb)
- Specified limit quasi-static load is 8.5 g
 - Making the limit base force = $8.5(362) = 3077$ lb
- RMS base force for the response shown at right is 3160 lb
 - Making the 3σ force = 9480 lb
- Thus, force limiting is required to prevent an unrealistically severe test of the primary structure.



Target RMS base force is $3077/3 = 1026$ lb

3σ base force

continued

Example Problem 6-a, continued: Deriving f_b and n from the Apparent Mass

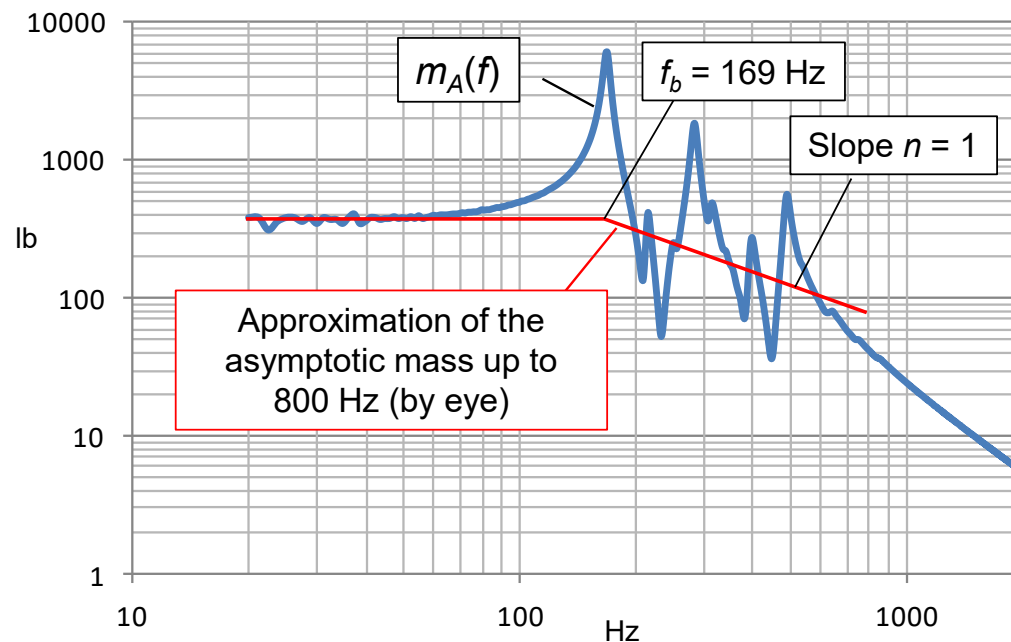
Recall Eq. 6-9:

$$S_{FF}(f) = m_A^2(f) S_{AA-0}(f)$$

Base FSD Apparent mass (Eq. 6-1) Input (base) ASD

So the apparent mass is

$$m_A(f) = \sqrt{\frac{S_{FF}(f)}{S_{AA-0}(f)}} \quad (\text{Eq. 6-10})$$



We've now derived values for f_b and n .

Next we need to decide on C^2 .

continued

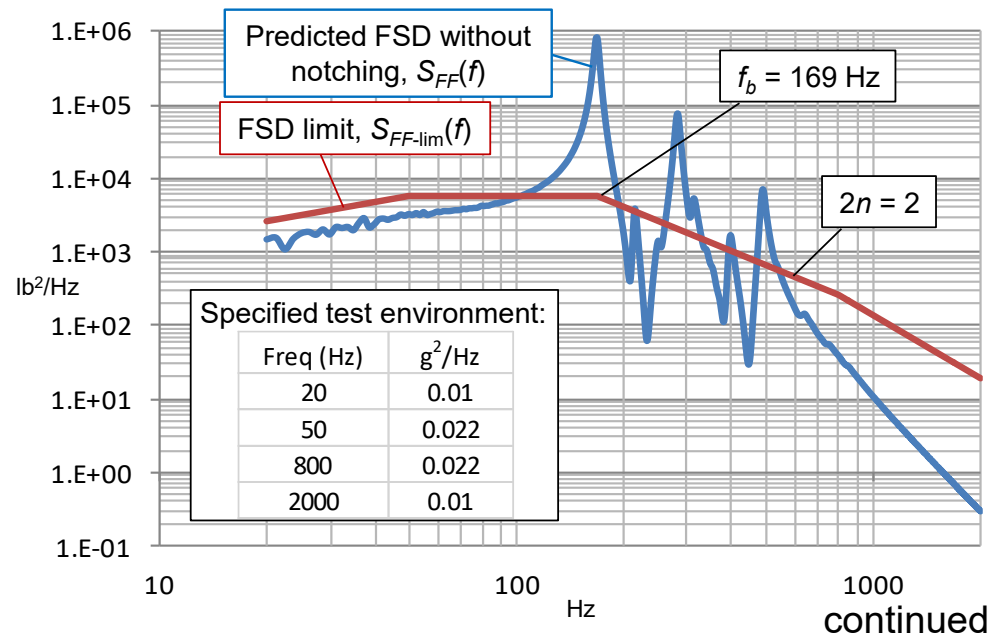
Example Problem 6-a, continued: Deriving an Appropriate Value for C^2

1. Put the base-force data ($S_{FF}(f)$ vs. frequency) into a spreadsheet set up to do the following.
2. Calculate an FSD limit with Eqs. 6-6 and 6-7 for the derived values of f_b and n , along with an assumed value of C^2 (usually between 2 and 5 but can be higher for low-mass items).
3. Truncate the $S_{FF}(f)$ data where needed to keep the FSD from exceeding the FSD limit, and numerically integrate to find the RMS force (square root of area under curve).
4. Iterate values of C^2 until the RMS force approximately equals the target of 1026 lb.

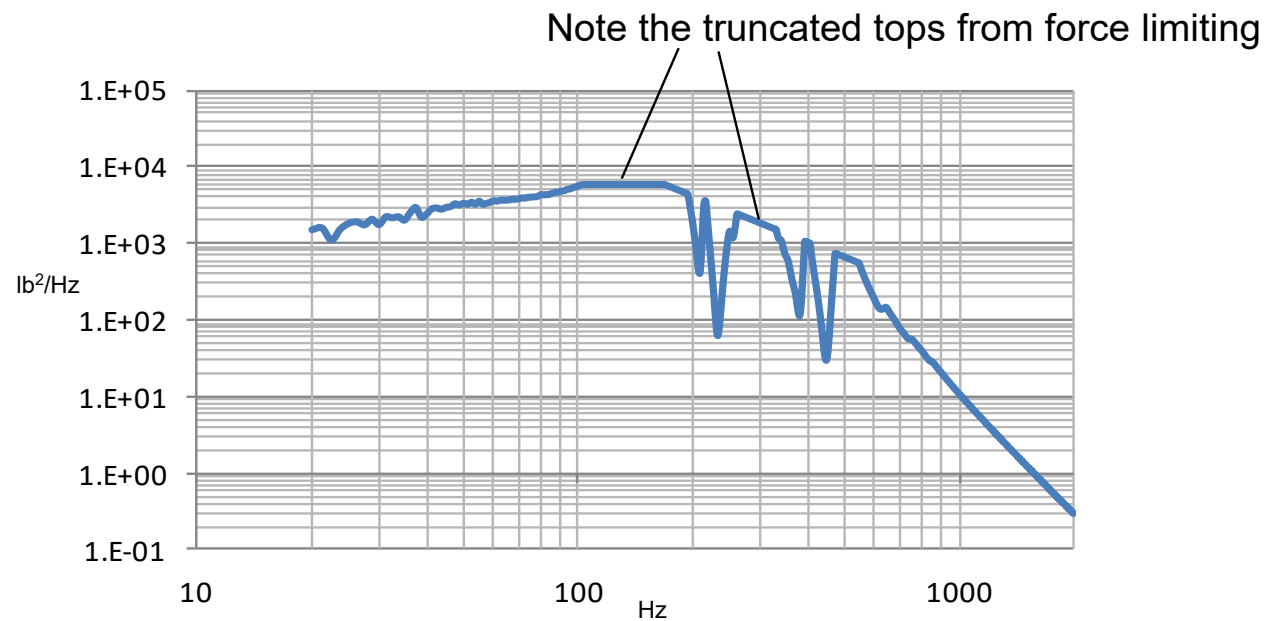
Results of iteration:
 $C^2 = 2$, yielding the force-limit curve shown at right.

We typically derive a pretest FSD limit for each test axis, using this process.

Then we run a low-level (e.g., -12 dB) test and modify the force limit with this process for the actual measured base force.

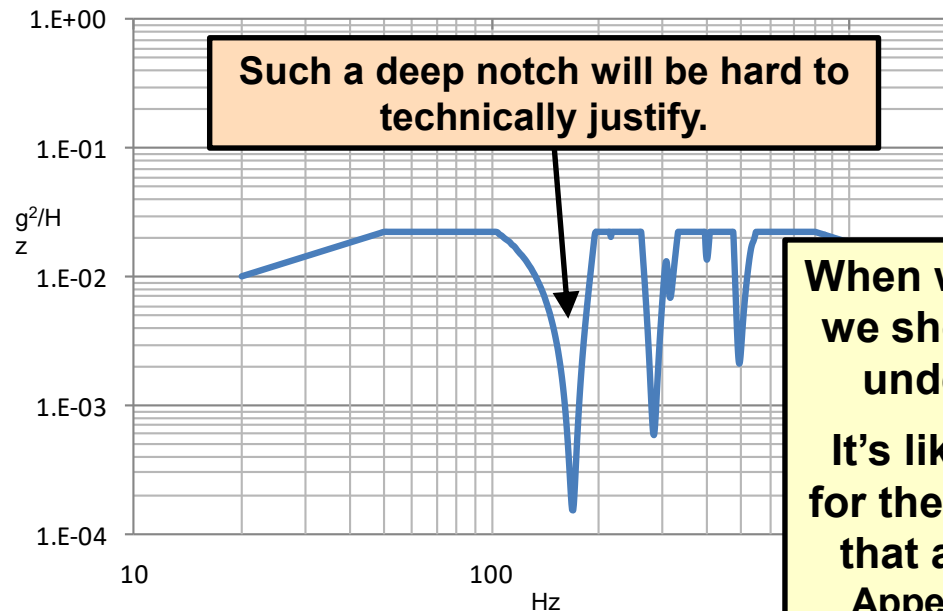


Example Problem 6-a, continued: Base FSD After Force Limiting



continued

Example Problem 6-a Conclusion: Predicted Notches in the ASD with Force Limiting



Notch depth, per VTSS Eq. 5-3,
STDI Eq. 5-10:

$$N_{dB} = 10 \left[\log_{10} \left(\frac{0.022}{0.00015} \right) \right] = 21.7 \text{ dB}$$

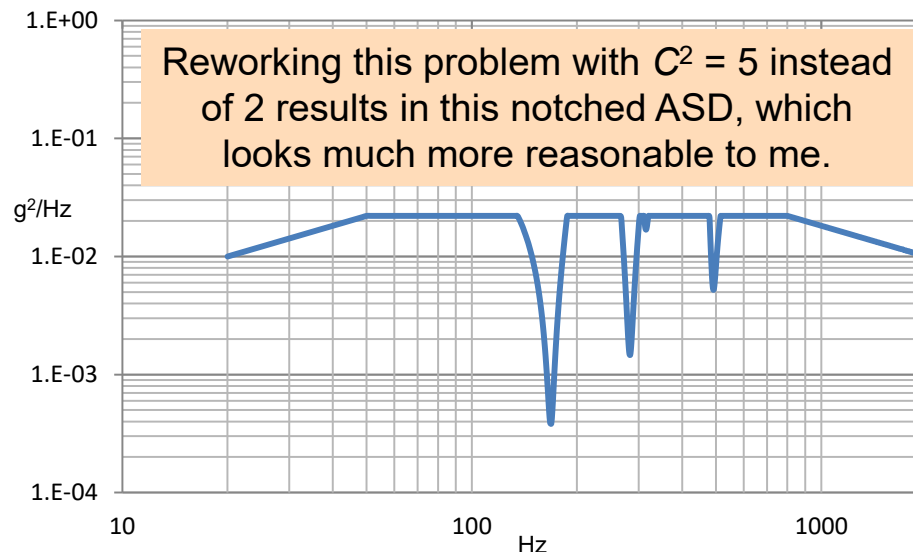
When we end up with such a deep notch,
we should ask ourselves if we might be
undertesting between 100 – 200 Hz.

It's likely that we're overcompensating
for the rigid-body response to test levels
that are excessive below 100 Hz. (See
Appendix 6B for an example, with further
discussion.)

It might be wise to use a higher value for C^2 and accept an overtest of the
primary structure in order to ensure the SmallSat is adequately tested
above 100 Hz.

Be sure to design for this load, though!

Staying with Example Problem 6-a: An Option Accepting an Overtest with $C^2 = 5$ Instead of 2



Notch depth:

$$N_{dB} = 10 \left[\log_{10} \left(\frac{0.022}{0.00038} \right) \right] = 17.6 \text{ dB}$$

which still might be hard to justify.

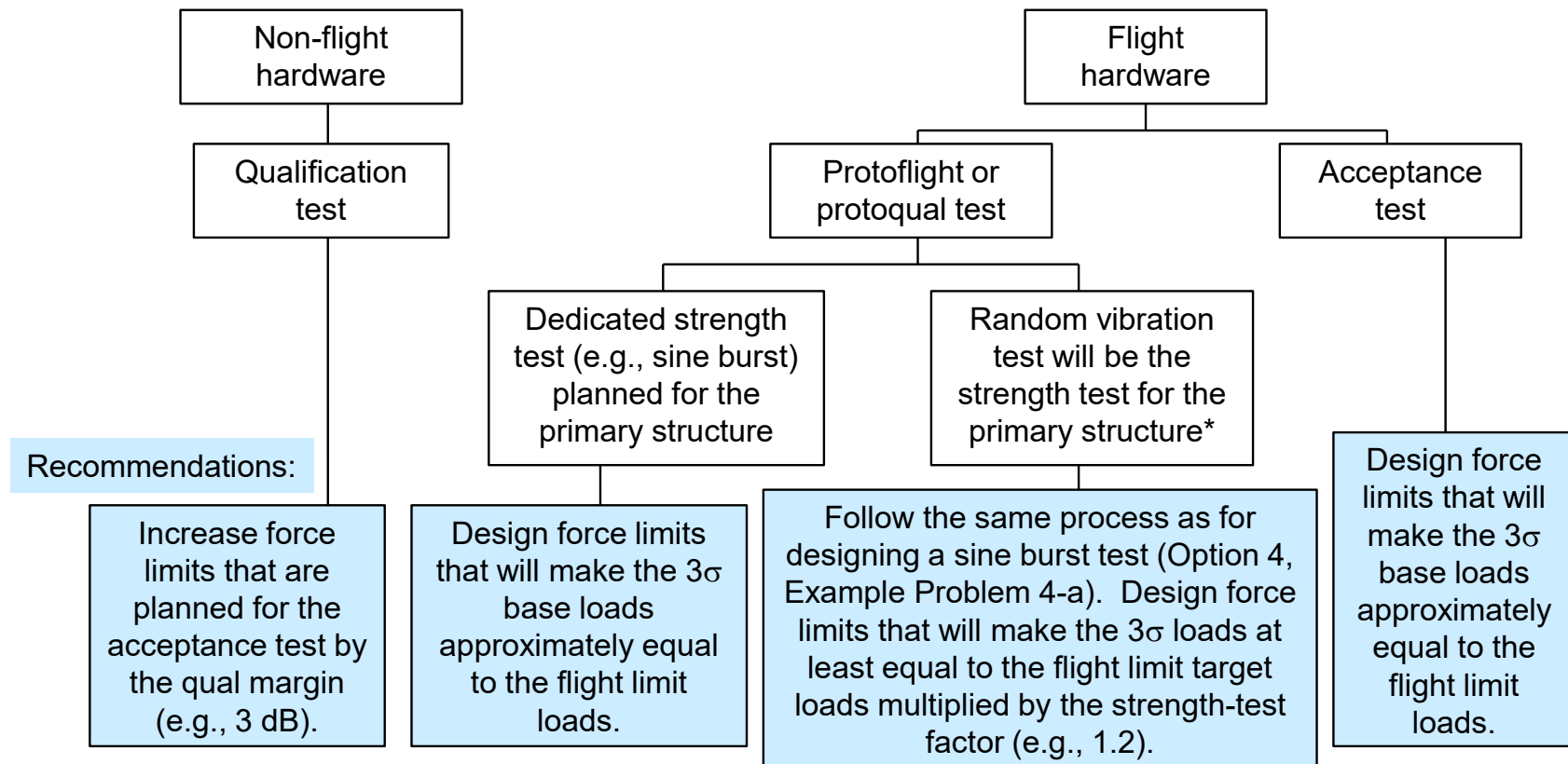
But the RMS base force is now 1390 lb, which is 36% higher than the target value of 1026 lb.

We would have to design the primary structure for a design load (limit load times factor of safety) that is 36% higher than required for the mission.

Alternative:

We could investigate the idea of breaking the test into two tests, one from 20 – 100 Hz (mostly rigid body, here) and the other from 100 – 2000 Hz.
(See related example in Appendix 6B to this section.)

My Thoughts on Appropriate Levels when Force Limiting for Random Vibration

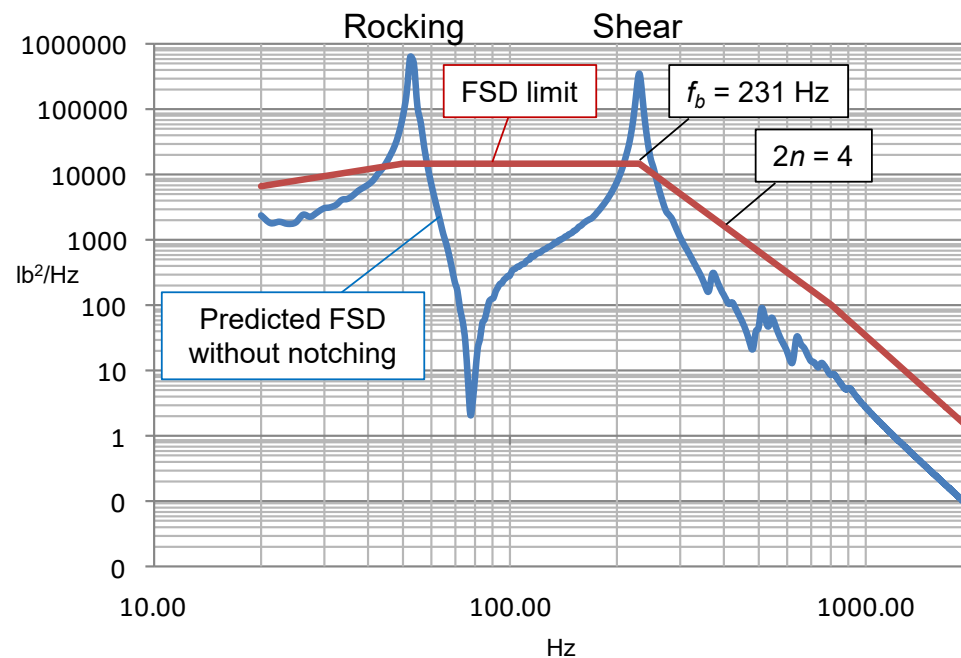


* As discussed in Sec. 4, the random vibration test should not be the strength test for the primary structure unless mass is low enough that the structure can be designed to have the needed strength and fatigue life for the planned test, accounting for uncertainty in response loads.

Example Problem 6-b: Force Limiting

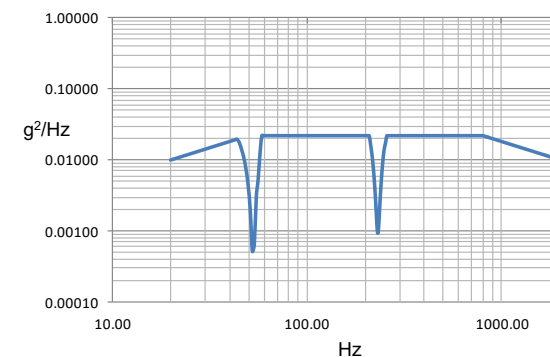
In this example, the 362-lb test article has two modes that have high effective mass in the lateral direction: a rocking mode and a shear mode (shearing of the separation mechanism).

As in Example Problem 6-a, the target RMS base force is 1027 lb.



In this case, we set the break frequency equal to the frequency of the second mode.

Resulting notches in ASD



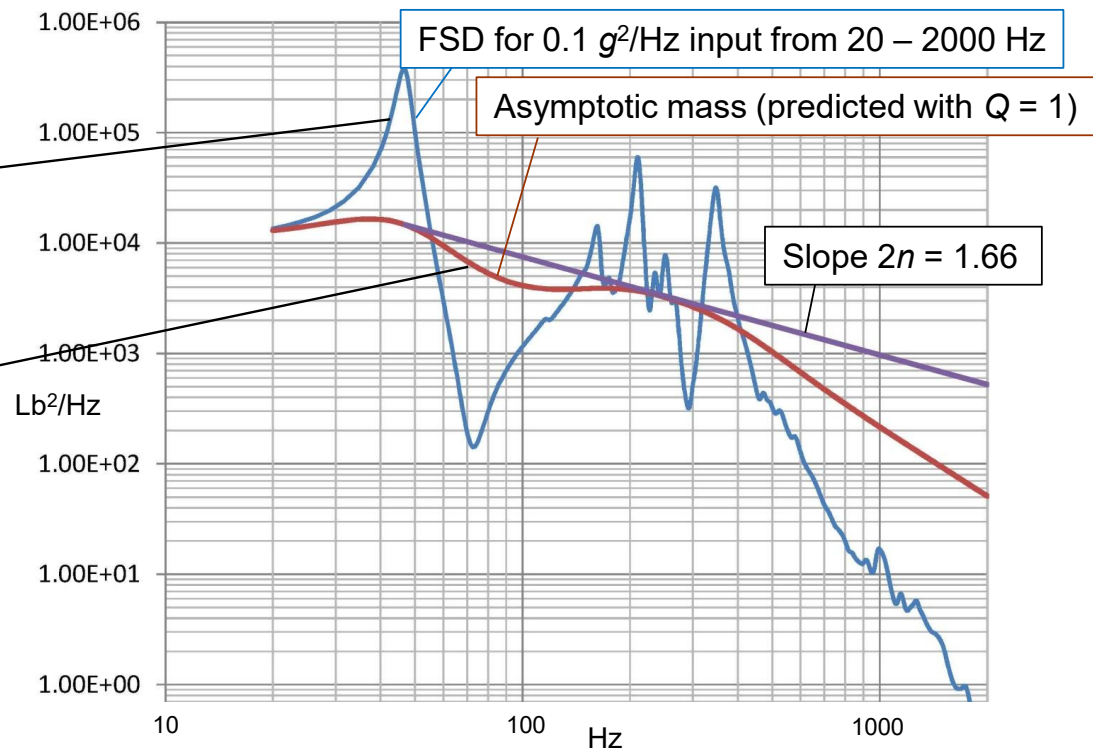
Example Problem 6-c: Calculating the Asymptotic Mass

In this example, we calculated the FSD two ways, both for constant $0.1 \text{ g}^2/\text{Hz}$ input:

- Once with the assumption of the actual estimated damping for this structure
- And once with the assumption of $Q = 1$ (no dynamic gain).

The second plot is analogous to the asymptotic mass.

- Even though the units are lb^2/Hz rather than lb



We then set the value of $2n = 1.66$, which just touches the asymptotic mass. Using a higher value of $2n$ may lead to excessive notching between 150 – 400 Hz.

Adjusting the Force Limit During Test

- f_b , break frequency:
 - Move it to match the applicable modal frequency found in a low-level sine-sweep or low-level white-noise random test.
- $2n$, rolloff slope:
 - Check by eye that the pretest value matches fairly well the average rolloff in the apparent mass. If not, adjust accordingly.
- C^2 :
 - Run a force-limited random vibration test to -12 dB, with a force-limit function based on f_b and $2n$, as modified above and the pretest C^2 value. The FSD limit also should be scaled down 12 dB.
 - If the proper C^2 value is used, the RMS base force should be approximately 25% of the target RMS value at full test levels.
 - If not, estimate the amount C^2 should be raised or lowered in order to approximately hit the target, and then use the modified value in the next-higher-level test (-9 dB).
- Assess the constants for the above considerations at each incrementally higher test level.

Simulating Force Limiting without Use of Force Sensors

If you test without force gages, you can still use force limiting methodology.

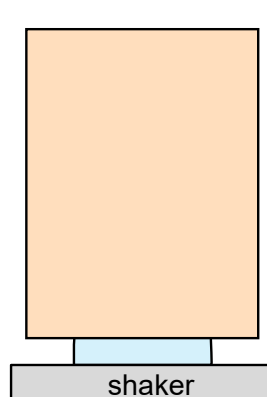
You can use either response limiting or manual notching to simulate force limiting, with the difference being that you have to trust your finite element model to predict the base force or moment.

Let's look at an example of response limiting used in this fashion; then we'll do the same for manual notching ...

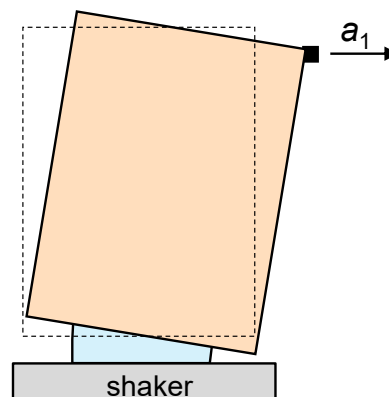
Example Problem 6-d: Response Limiting

As an example of response limiting that simulates force limiting, consider the test article shown below.

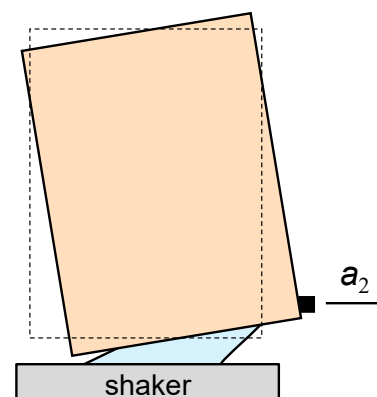
- Two modes with high effective mass for lateral testing: rocking and base shear. (Example Problem 6-b is for a small satellite with modes such as these.)
- Two accelerometer locations measuring accelerations a_1 and a_2 .
- Within the frequency range in which response is dominated by the first mode, there's a strong relationship between base force and a_1 ; for the second mode, base force relates more to a_2 .
- We can calculate transfer functions from the FEM.



a. Undeformed



b. First mode: rocking on separation mechanism



c. Second mode: shear of separation mechanism

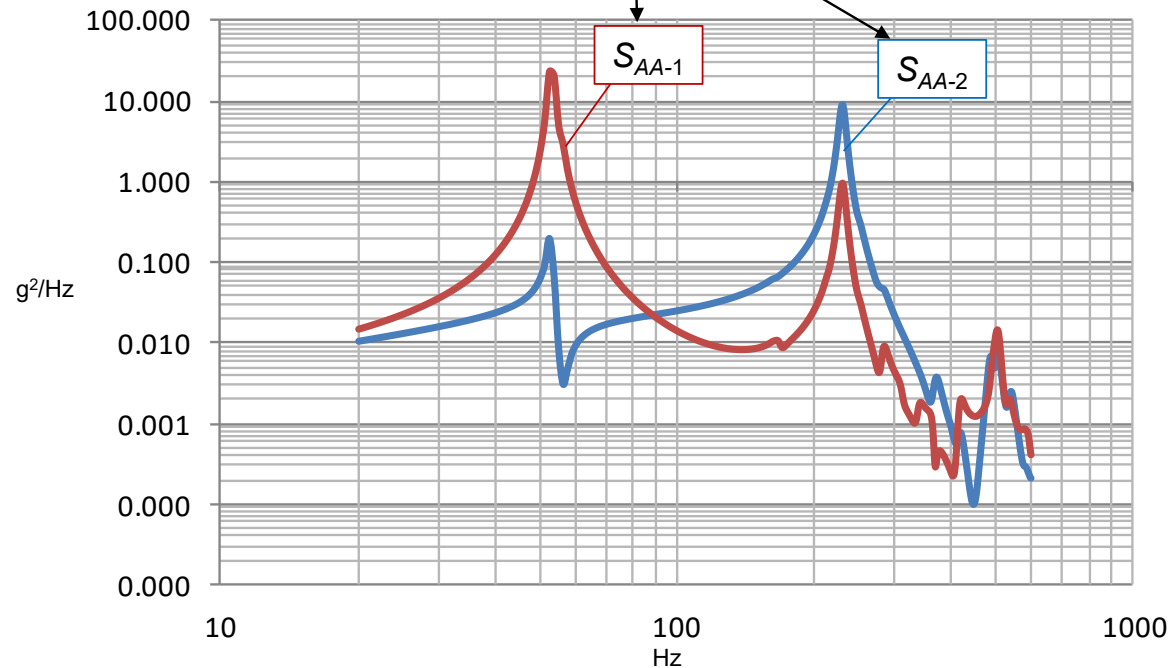
If we have high confidence in the FEM for these two modes, we can derive response limits for these two accelerometers to simulate force limiting.

continued

Example Problem 6-d, continued: Predicted Random Vibration Responses (ASDs S_{AA-1} and S_{AA-2})

(based on the same FEM and test environment as for Example Problem 6-b)

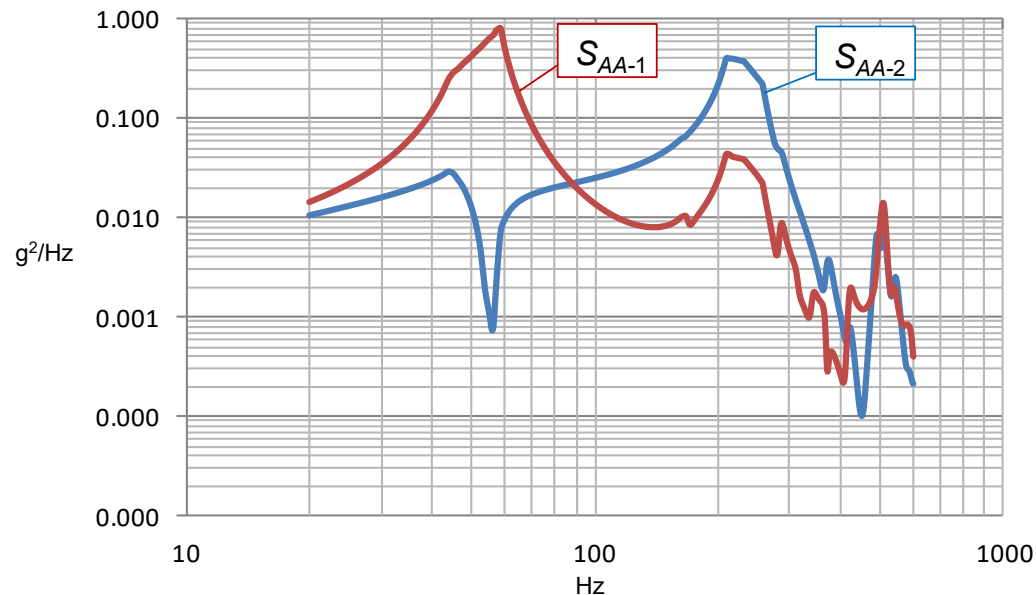
Corresponding to accelerations a_1 and a_2 , respectively, from previous page



continued

Example Problem 6-d, continued: Predicted Random Vibration Responses with Force Limiting

With the FSD limit derived in Example Problem 6-b, and the corresponding notched input ASD, here's how the predicted response accelerations change.



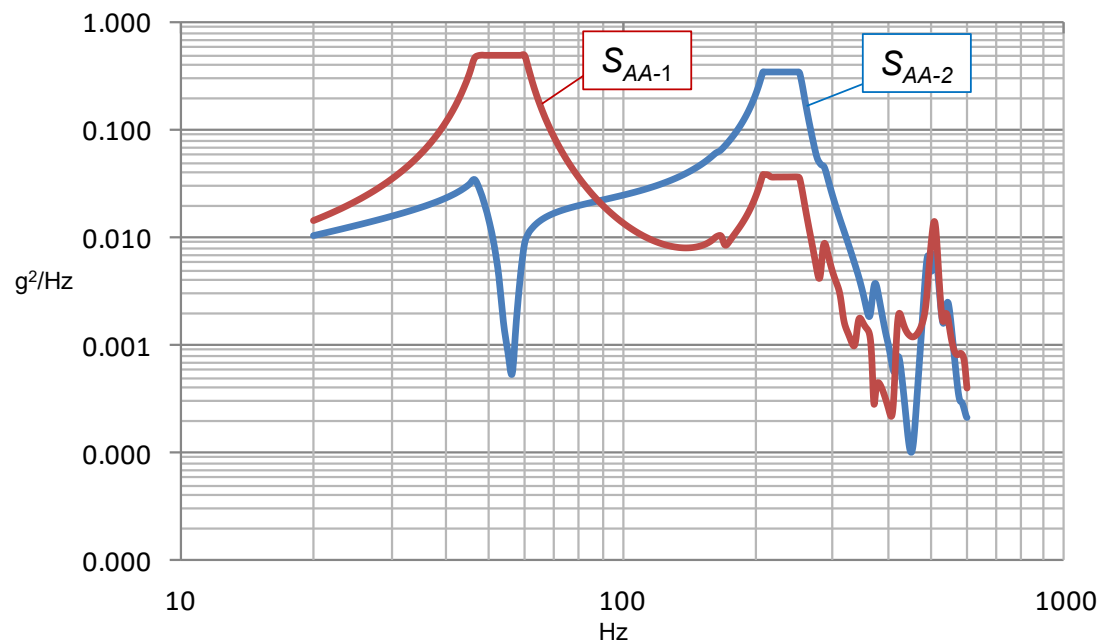
If base force and acceleration— a_1 for the first mode and a_2 for the second—were truly proportional within the target frequency bands, the peaks for S_{AA-1} and S_{AA-2} would be truncated with the same shapes as the FSD peaks, or nearly flat, as they would be for true force limiting (Example Problem 6-b).

This is not the case, but it's close enough to being true for us to make an approximation that will allow response limiting to simulate force limiting.

continued

Example Problem 6-d, continued: Predicted Random Vibration Responses with Response Limiting

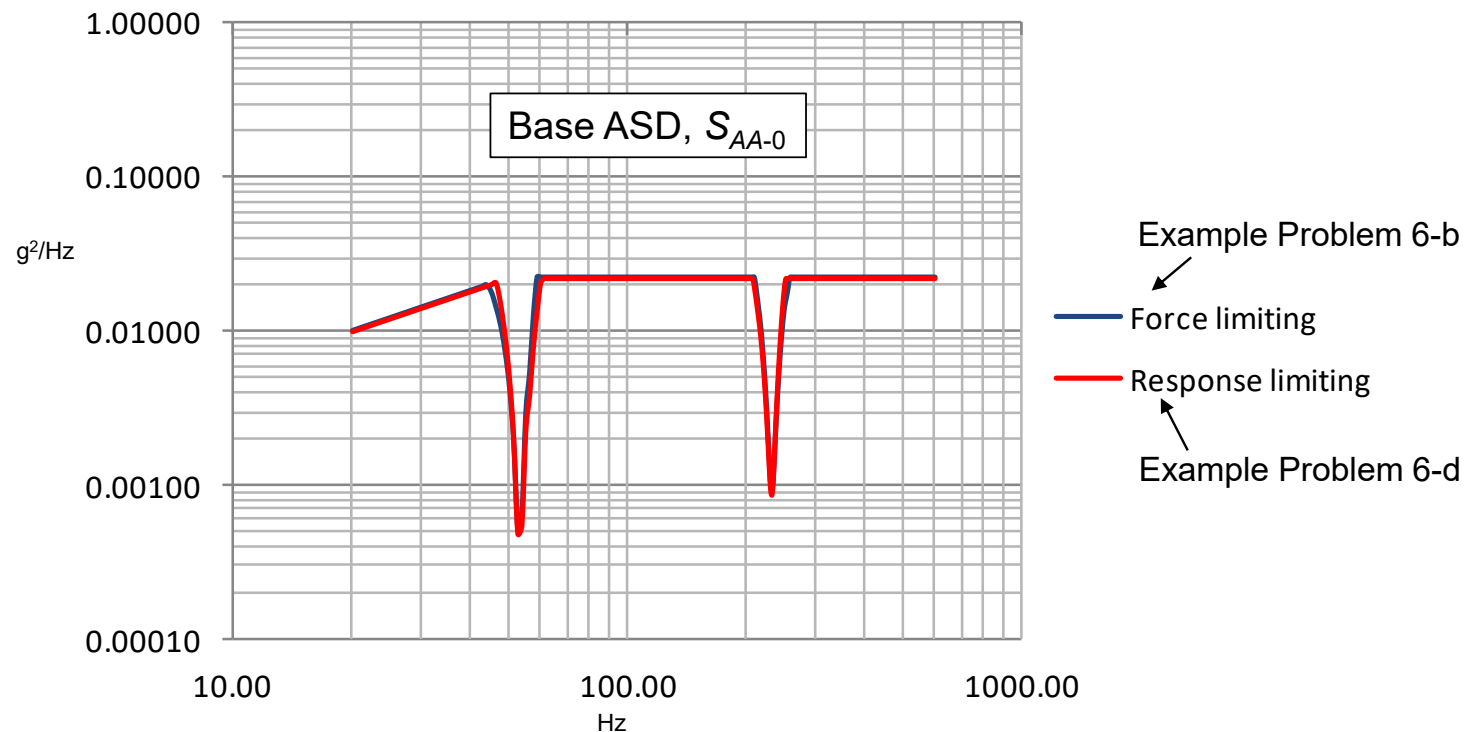
Flattening the peaks with response limits of $0.50 \text{ g}^2/\text{Hz}$ for S_{AA-1} between 20 – 100 Hz and $0.35 \text{ g}^2/\text{Hz}$ for S_{AA-2} between 150 – 300 Hz:



I derived these response limits by trial and error with a spreadsheet to get the same predicted RMS base force as I got with force limiting (Example Problem 6-b).

continued

Example 6-d Conclusion: Comparing Notches, Response Limiting and Force Limiting

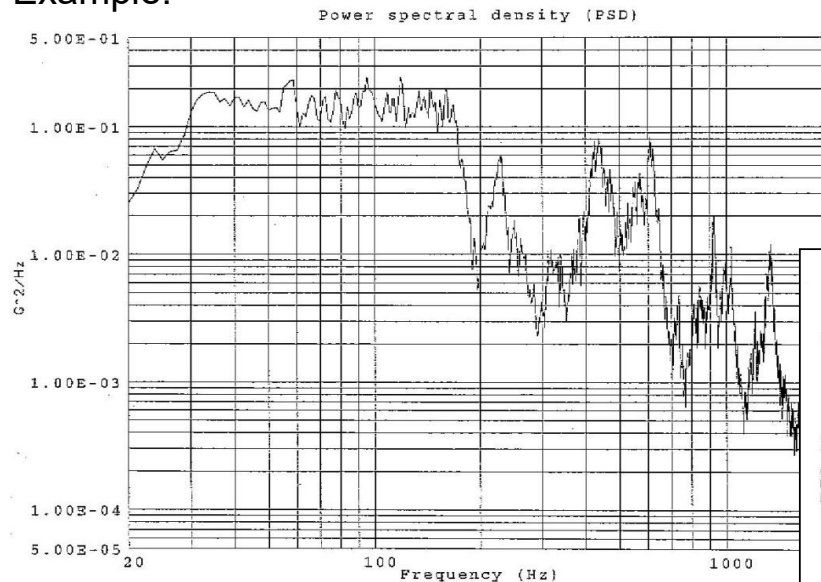


Conclusion: If our model is accurate, the notches are nearly the same as we'd get with true force limiting.

Because model accuracy falls off for higher-order modes, we typically can use this approach with confidence only for the first mode or two.

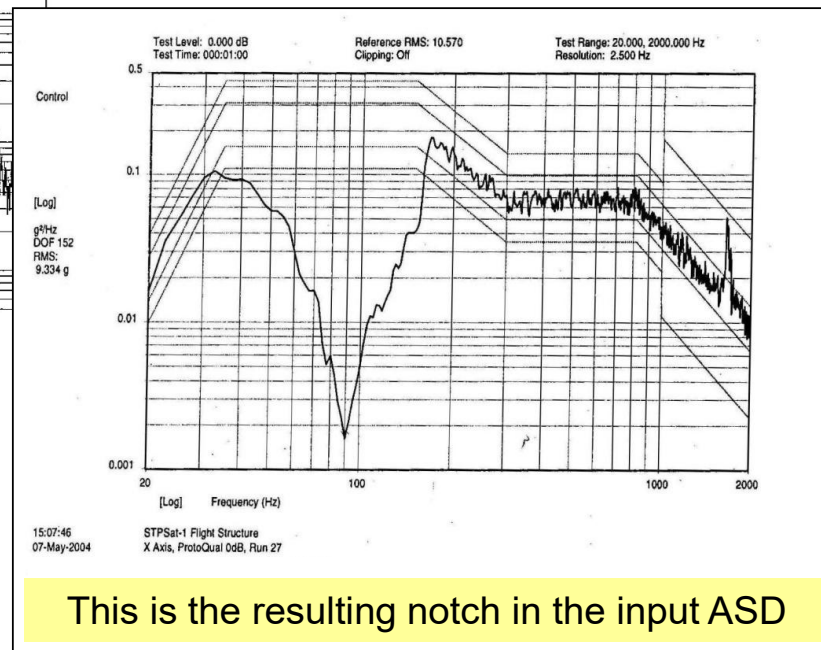
Response Limiting Requires Confidence in How the Response Acceleration Relates to Base Force

Example:



To avoid excessive loading for a 90-Hz mode, the shaker was controlled between 30 and 170 Hz by an accelerometer at the test article's CG. But acceleration at the CG was amplified by a local mode.

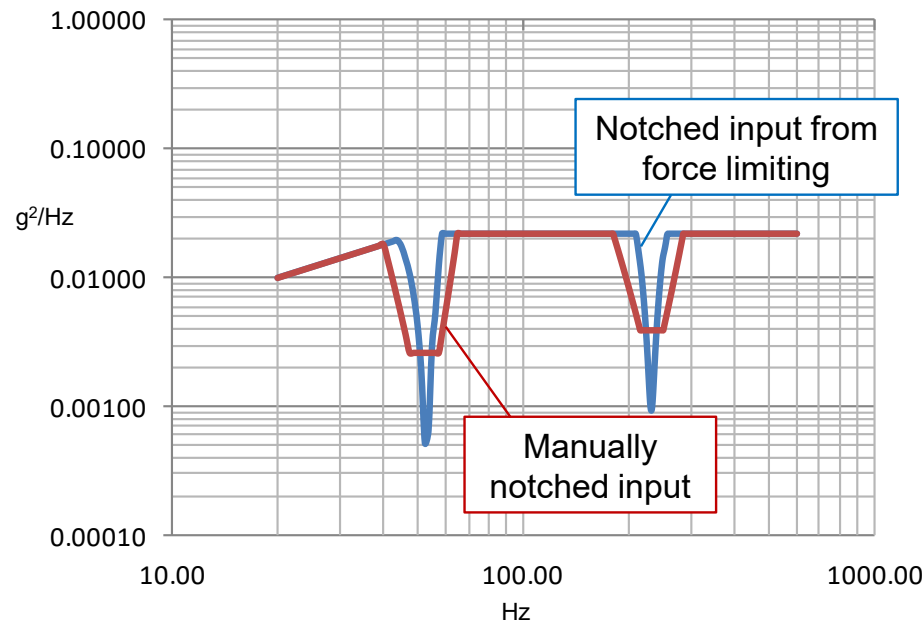
In this case, the response channel did not relate well to base force, and notching was excessive.



This is the resulting notch in the input ASD

Manual Notching to Simulate Force Limiting

- The notches shown below were derived to simulate force limiting for the test article and environment in Example Problem 6-b. These notches result in approximately the same base force.
 - The notched ASD is entered into the control system in place of the specified ASD.
- Manual notches can't be as steep, narrow, and deep as a notch created by force limiting or response limiting, or the shaker won't be able to control them.
- So we have to widen the notches to compensate.

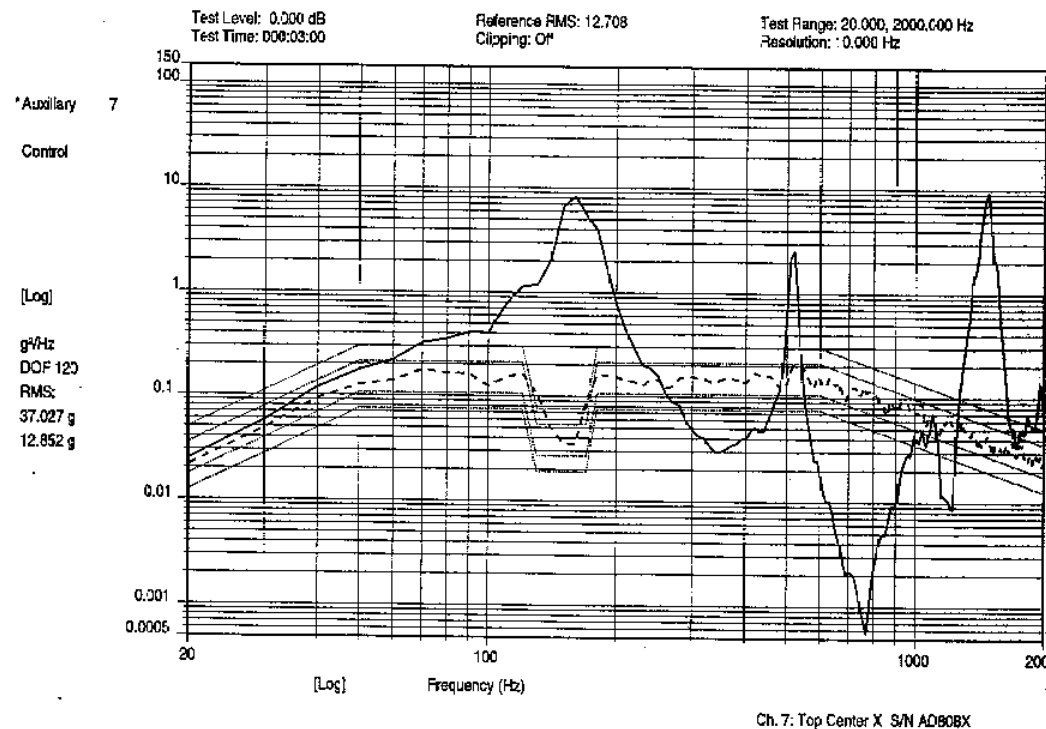


To generate these notches, I put the model-predicted data into a spreadsheet and assumed response scales linearly with input.

During test we did the same thing but with actual test data in place of model data. We moved up in level (3 dB increments), adjusting the notches as needed before going to the next level.

Notching Can Also Be Justified by “Moment Limiting”

FalconSAT-2 Qualification Test



A NASA contractor tailored this notch to keep the predicted 3σ base moment from exceeding the flight limit moment.

Other Technical Rationale for Notching

The examples in this section have all related to base force or moment.

You also can technically justify notching by use of response limiting based on keeping the maximum (3σ or other statistical level as appropriate) acceleration for a system component (e.g., satellite instrument) from exceeding a clearly conservative flight limit (maximum expected) acceleration derived from ...

- an approved mass-acceleration curve (MAC).
- or loads analysis that accounts for steady-state loads, transient loads, and random vibration (e.g., CLA results combined with vibroacoustic analysis results).
 - Accelerations should be calculated and test measured at the same locations and over the same frequency ranges;
 - and the random portion of flight limit loads should be calculated at the same sigma level as the test loads.

See next slide for important notes related to the above.

Other Technical Rationale for Notching—Important Notes

With the approaches described on the previous slide, we're trying to ensure a component's equivalent quasi-static load (EQSL), as quantified by total force at the interface between the component and its host structure, does not exceed the component's specified or predicted limit EQSL.

We should be able to predict a component's EQSL for the system-level vibration test with FEA, but during test we probably won't be able to actually measure interface force.

Thus, the limitations of response limiting and manual notching addressed earlier in this section—for force at the base of the test article (e.g., satellite), when base force is not actually measured in test—apply for the component as well:

- The FEM won't be accurate for higher-order modes, so in test we should limit the component's measured response acceleration only for the first mode (possibly the second as well) that shows amplified acceleration for that component.
- We would not be able to technically justify response limiting over the full spectrum of 20 – 2000 Hz.

Key Points of This Section

- For any method and extent of notching, make sure you have technical rationale.
 - and make sure all stakeholders buy in before you do the test!
- The most common method of justifying notching is use of force-limiting principles.
 - Even when not measuring base force directly with force sensors.
- When force sensors are not present, response limit or manually notch only for the modes for which you can confidently relate measured acceleration to base force or interface force, as applicable.
 - Typically the first response mode, and second mode only if the FEM is correlated for that mode
- When using a specified or calculated limit load to substantiate notching, make sure the limit load encompasses the effects of all contributors during launch.
- Don't notch excessively at some frequencies to compensate for excessive test levels at other frequencies.



*This section applies to both VTSS—Vibration Testing of Small Satellites
and STDI—Structural Test Design and Interpretation*



Appendix 6A

The Simple TDFS Method of Deriving Force Limits

Appendix 6A: Simple TDFS Method of Deriving C^2 (Method 2 from Table 6-1)

Recall Eq. 6-9 for an SDFS: $S_{FF}(f) = m_{App}^2(f) S_{AA-0}(f)$

m_A peaks at the SDFS's natural frequency, f_{n2} :

$$m_{App-max} = m_{App}(f_{n2}) = Qm_2 \quad (\text{Eq. 6-11})$$

S_{FF} = base FSD
 m_A = apparent mass
 S_{AA-0} = base ASD
 f = frequency
 f_{n2} = natural frequency
 m_2 = SDFS mass
 Q = quality factor

Thus, for white noise random vibration input (S_{AA-0} constant over the frequency spectrum), the maximum base FSD for the SDFS is

$$S_{FF-max} = S_{FF}(f_{n2}) = Q^2 m_2^2 S_{AA-0} \quad (\text{Eq. 6-12})$$

We can rearrange Eq. 6-12 to solve for Q^2 when we know S_{FF-max} for white noise input:

$$Q^2 = \frac{1}{m_2^2} \left(\frac{S_{FF-max}}{S_{AA-0}} \right) \quad (\text{Eq. 6-13})$$

continued

Appendix 6A: Simple TDFS Method of Deriving C^2 , continued

Repeating Eq. 6-13,

$$Q^2 = \frac{1}{m_2^2} \left(\frac{S_{FF-\max}}{S_{AA-0}} \right)$$

Now we assemble a TDFS, with the former SDFS now referred to as the “load”.

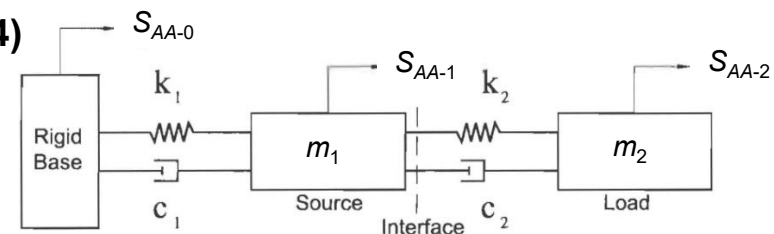
$S_{FF-\max}$ = maximum base FSD
 S_{AA-0} = base ASD (white noise)
 f = frequency
 f_{n2} = natural frequency of the load by itself
 m_2 = SDFS (load) mass
 Q = quality factor

To account for the dynamic absorber effect in a TDFS, as stated previously, we replace Q with C to define a force limit for testing the load by itself. Recognizing that, for white noise random input, when $f_{n2} = f_{n1}$, the interface FSD, S_{FF-2} , peaks at the TDFS’s fundamental frequency, $f_{\text{sys-1}}$,

$$C^2 = \frac{1}{m_2^2} \left[\frac{S_{FF-2}(f_{\text{sys-1}})}{S_{AA-1}(f_{\text{sys-1}})} \right] \quad (\text{Eq. 6-14})$$

Interface FSD at $f = f_{\text{sys-1}}$

Interface ASD at $f = f_{\text{sys-1}}$



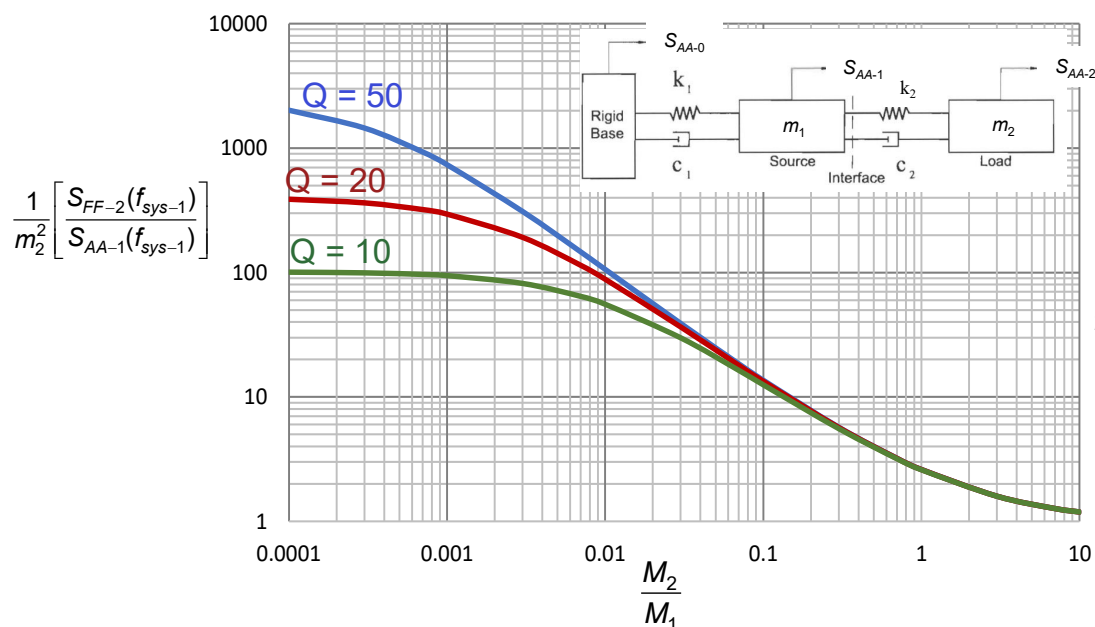
continued

Appendix 6A: Simple TDFS Method, continued

For the case in which $f_{n1} = f_{n2}$ (worst case for the load),

$$C^2 = \frac{1}{m_2^2} \left[\frac{S_{FF-2}(f_{sys-1})}{S_{AA-1}(f_{sys-1})} \right] = \frac{1 + \frac{r^2}{Q_2^2}}{\left[1 - r^2\right]^2 + \frac{r^2}{Q_2^2}} \quad (\text{Eq. 6-15})$$

(VTSS Ref. 15,
STD1 Ref. 26)



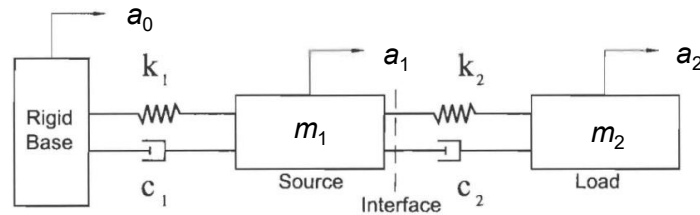
f_{n1} = natural frequency of the source as an SDFS
 f_{n2} = natural frequency of the load as an SDFS
 $S_{FF-2}(f_{sys-1})$ = maximum interface FSD (at f_{sys-1})
 $S_{AA-1}(f_{sys-1})$ = interface ASD at f_{sys-1}
 Q = quality factor
 $r = \frac{f_{sys-1}}{f_{n2}}$
 f_{sys-1} = first natural frequency of TDFS

$$f_{sys-1} = f_{n2} \sqrt{1 + \frac{m_2 / m_1}{2} - \sqrt{m_2 / m_1 + \frac{(m_2 / m_1)^2}{4}}}$$

(from Eq. 6-2)

Appendix 6A, continued: Example: Simple TDFS Method

From Example 6-1, with 1 g sinusoidal base input,



$$\left. \begin{array}{l} m_1 = m_2 = 1 \text{ lb} \\ k_1 = k_2 \\ c_1 = c_2 \text{ (} Q_1 = Q_2 = 50\text{)} \\ f_{\text{sys-1}} = 0.618 f_{n2}, f_{\text{sys-2}} = 1.618 f_{n2} \end{array} \right\} \Rightarrow f_{n1} = f_{n2}$$

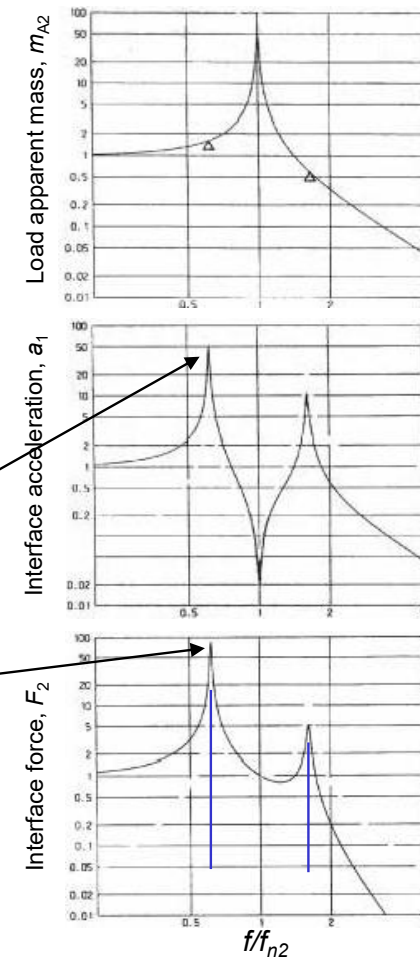
For this system driven with unit white noise
random vibration (1 g²/Hz base input),

$$S_{AA-1}(f_{\text{sys-1}}) = (50)^2 = 2500 \text{ g}^2/\text{Hz}$$

$$S_{FF-2}(f_{\text{sys-1}}) = (80)^2 = 6400 \text{ lb}^2/\text{Hz}$$

$$\frac{1}{m_2^2} \left[\frac{S_{FF-2}(f_{\text{sys-1}})}{S_{AA-1}(f_{\text{sys-1}})} \right] = \frac{1}{1} \left[\frac{6400}{2500} \right] = 2.56$$

which agrees with Eq. 6-15 and the plot on the previous page



Appendix 6A, continued: Two Options for Use of the Simple TDFS Method

1. Use the semi-empirical method, with C^2 calculated with Eq. 6-15, with m_1 and m_2 equal to the asymptotic mass of the source and the load, respectively, at the load's natural frequency, f_{n2} .
2. Use the simple TDFS method directly to define the FSD limit, as follows (not using the Semi-Empirical Method):
 - a. Break the frequency spectrum into one-third octave bands.
 - b. Within each band, calculate an FSD limit, S_{FF-lim} , equal to the maximum interface FSD, $S_{FF-2}(f_{sys-1})$ (as calculated by rearranging Eq. 6-15) and setting m_1 and m_2 equal to the residual masses of the source and load, respectively.

Recall: residual mass at a given frequency is total mass minus the accumulated modal mass up to that frequency.

- c. Draw smooth line segments through the one-third-octave values to generate S_{FF-lim} as a function of frequency over the full spectrum.

Both approaches to using the simple TDFS method are approximate.

The complex TDFS method (Method 3 in Table 6-1), per VTSS Refs. 15 and 17 (STDI Refs. 26 and 28), can be an improvement but is still approximate.

Per VTSS Ref. 19 (STDI Ref. 30), the complex TDFS method usually over-specifies the force limit, leading to over testing, whereas the simple TDFS method can lead to under testing.

Appendix 6B

An Example that Raises Several Issues Associated with Force Limiting

Tom Sarafin
Instar Engineering and Consulting, Inc.

as presented at the
Spacecraft and Launch Vehicle Dynamic Environments Workshop
June 27 – 29, 2023

Note: Equation numbers and some of the variables are different here than in the rest of Sec. 6.

Introduction

The **semi-empirical method** from NASA-HDBK-7004C is the most commonly used formalism (rather than an actual method) for force limits for vibration testing. For random vibration, the force limit—in the form of a force spectral density (FSD) limit with units lb^2/Hz or N^2/Hz —is defined as a function of frequency with two equations:

$$\text{For } f < f_b, \quad S_{FF-\text{lim}}(f) = C^2 M_0^2 S_{AA-0}(f) \quad \text{Eq. 1}$$

$$\text{For } f \geq f_b, \quad S_{FF-\text{lim}}(f) = C^2 M_0^2 \left(\frac{f_b}{f} \right)^{2n} S_{AA-0}(f) \quad \text{Eq. 2}$$

where f = frequency

f_b = break frequency (usually the fundamental frequency of the test article)

M_0 = total mass of the test article (in weight units, lb or N, when acceleration is in g units)

C and n = constants without units

$S_{FF-\text{lim}}(f)$ = base FSD limit (lb^2/Hz) at frequency f

$S_{AA-0}(f)$ = the base random vibration acceleration spectral density (ASD) (g^2/Hz) at frequency f

Deriving a technically defensible value for C^2 can be challenging.

The Study

A practical example for deriving C^2 for testing of a spacecraft payload from base-drive analysis for the full spacecraft (bus-payload coupled system) was presented in the following paper:

Soucy, Y, V. Dharanipathi, and R. Sedaghati. “Investigation of Force-Limited Vibration for Reduction of Overtesting”, published in Journal of Spacecraft and Rockets, Vol. 43, No. 4, July-August 2006.

Great example, but because the objectives of that paper were different—investigate the expected range of C^2 values—than for my study, the paper carried the example only to the point of deriving C^2 .

I decided to continue the example to see ...

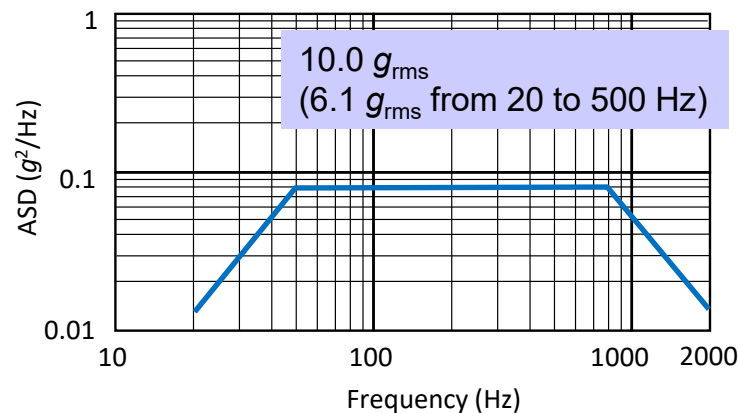
- what the FSD limit and notched base ASD would look like,
- how much the RMS base force would be reduced by force limiting,
- and how the RMS base force compares to the bus-payload interface force from the SC analysis.

This study raised some issues that can arise when designing a force-limited test.

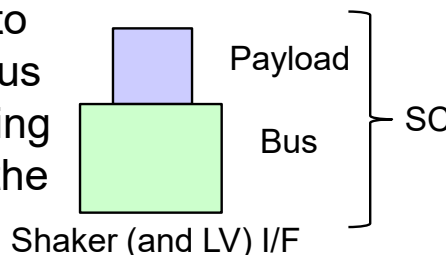
I ended the study by investigating several options for addressing these issues.

Problem Statement (expanded for this study)

The test article is a 34.6 lb payload for a small spacecraft (SC). The integrated SC, including the payload, has a total weight of 96.1 lb and will be tested in the vertical axis to the random vibration environment defined below.



The payload is mounted to the top panel of the SC bus structure with four mounting bolts and is centered on the top panel.



Problem statement:

With an assumed Q of 20, derive a test environment and an FSD limit for separate testing of the payload, based on predicted responses when the SC is tested without force limiting.

The problem is worked up to 500 Hz only.*

*The original example in the referenced paper went to 2000 Hz. I didn't have access to the FEMs, so I manually digitized data to match up to 500 Hz in a spreadsheet so that I could continue the example.

Step 1: Calculate the Apparent Mass for the Payload

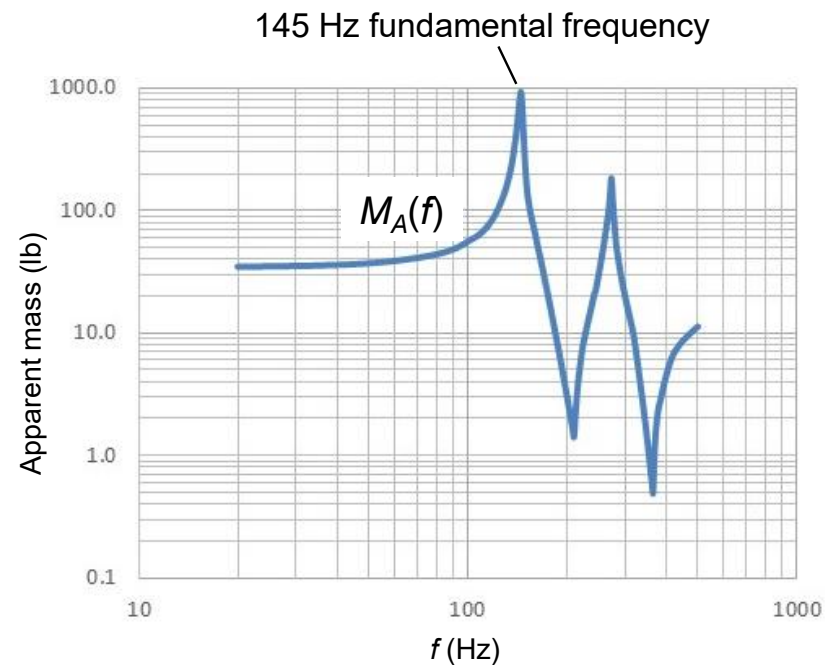
To determine the payload's apparent mass, $M_A(f)$, we base drive the payload FEM with any arbitrary ASD, $S_{AA-0}(f)$, and calculate the base FSD, $S_{FF}(f)$, assuming $Q = 20$ for this example.

The **apparent mass** is the payload's frequency-varying impedance, equal to the base force caused by unit- g sinusoidal base motion:

$$M_A(f) = \sqrt{\frac{S_{FF}(f)}{S_{AA-0}(f)}}$$

Base ASD

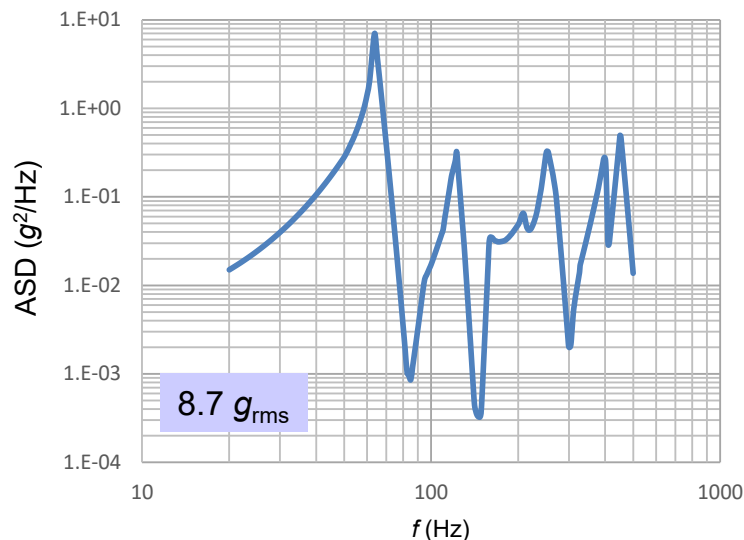
Base FSD



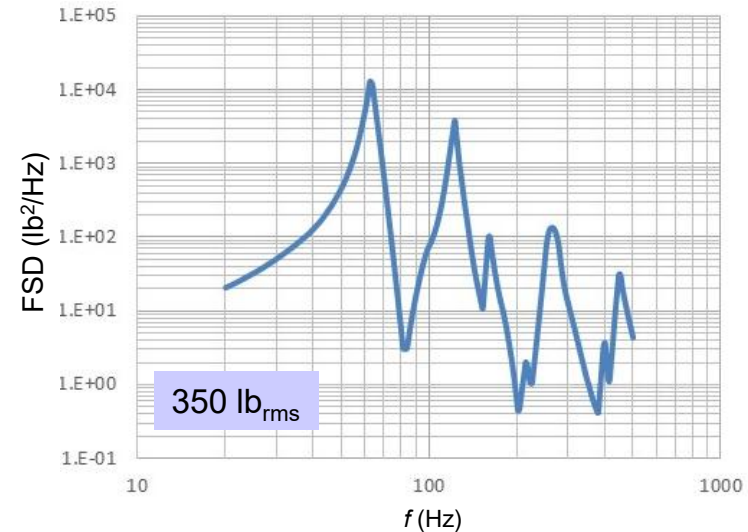
Step 2: Calculate Interface ASD and FSD for the SC Test

We base drive the coupled SC model with the specified test environment, again with $Q = 20$.

Calculated interface (bus-payload) ASD, S_{AA-1} ,
(average level at the 4 interface bolt locations)



Calculated interface FSD, S_{FF-1} ,
(total force between bus and payload)

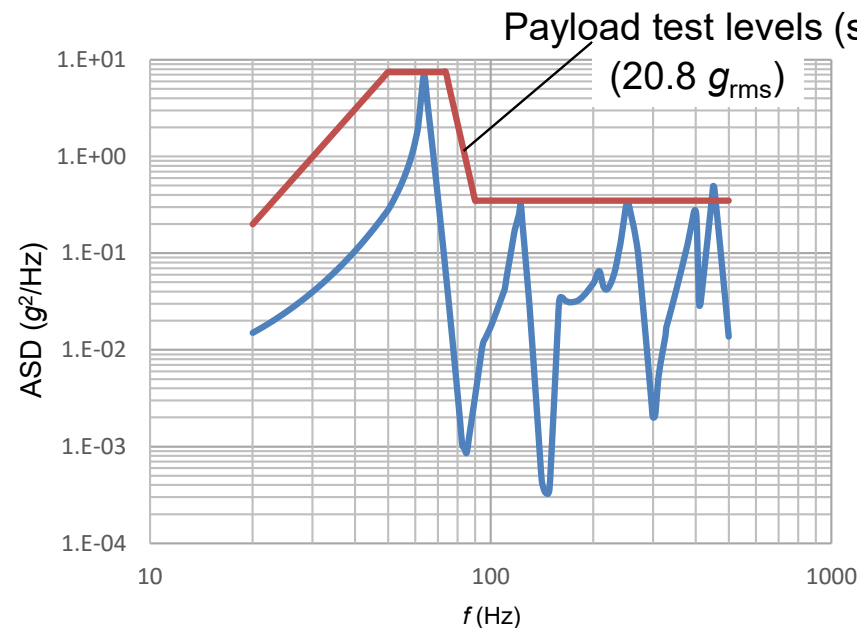


To sum FSDs at multiple interface nodes in the FEM, $FSD_{total} = \left[\sum \sqrt{FSD_{individual}} \right]^2$

However, as noted in the referenced paper, this equation applies only if the individual forces are in phase, which may not be the case for high-order modes.

Step 3: Derive Payload Test Levels

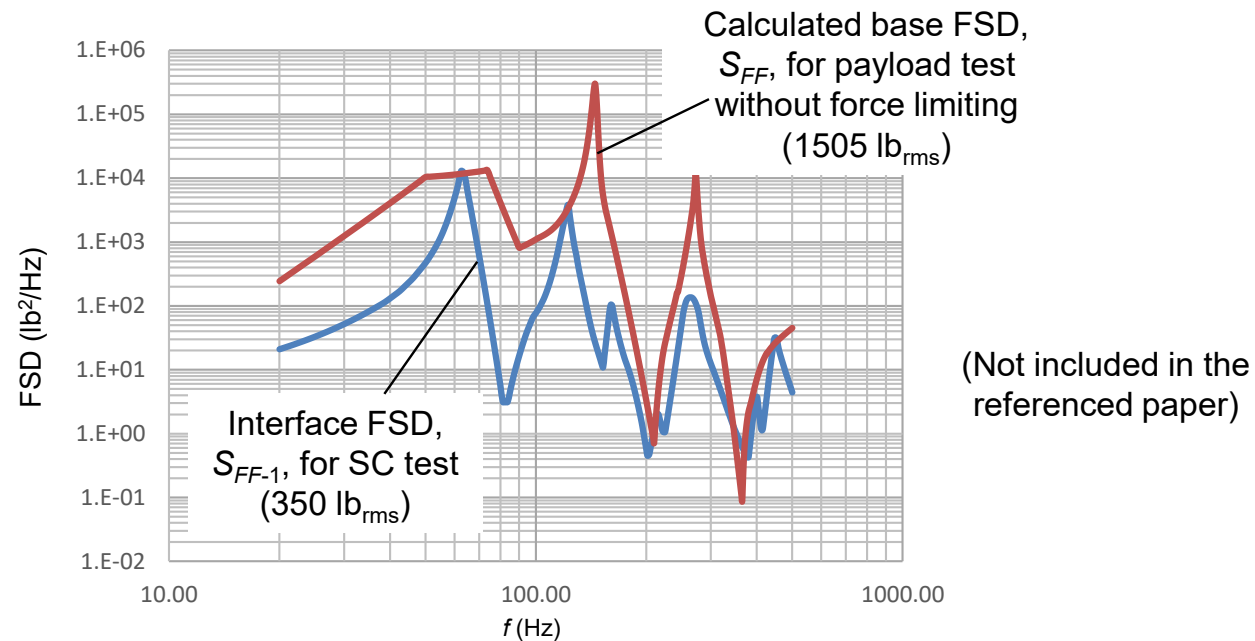
Based on the calculated interface ASD for the SC test, we can derive a test environment for the payload (from the referenced paper):



Note: In the referenced paper, the envelope shown here was not the point. The only values of importance to that study were those near the payload's fundamental frequency (145 Hz).

Let's see what would happen if we test the payload to this environment without force limiting ...

Step 4: Predict Payload Response to Derived Test Levels



The base of the payload would be over tested by a ratio of $1505/350 = 4.3$

We should force-limit the payload test.

Step 5: Derive a Value for C^2

When we know the interface acceleration and the interface force between two bodies for a given system-level (SC) input, we can derive C^2 from Eq. 1:

$$\text{For } f < f_b, \quad S_{FF-\text{lim}}(f) = C^2 M_0^2 S_{AA-0}(f)$$

Substituting S_{AA-1} for S_{AA-0} , and S_{FF-1} for $S_{FF-\text{lim}}$, and rearranging ...

$$C^2 = \frac{S_{FF-1}(f_{\text{max}-f})}{M_0^2 \cdot S_{AA-1}(f_{\text{max}-a})}$$

where

M_0 is the mass of the test article (34.6 lb for this example)

S_{AA-1} is the bus-payload interface ASD, analogous to the base ASD S_{AA-0} for the payload alone

$S_{FF-\text{lim}}$ is the FSD limit for testing the payload separately

S_{FF-1} is the bus-payload interface FSD, analogous to the base FSD S_{FF} for the payload alone

$f_{\text{max}-a}$ is the frequency of the higher of the two S_{AA} peaks on each side of the antiresonance (zero) associated with the payload's fundamental frequency

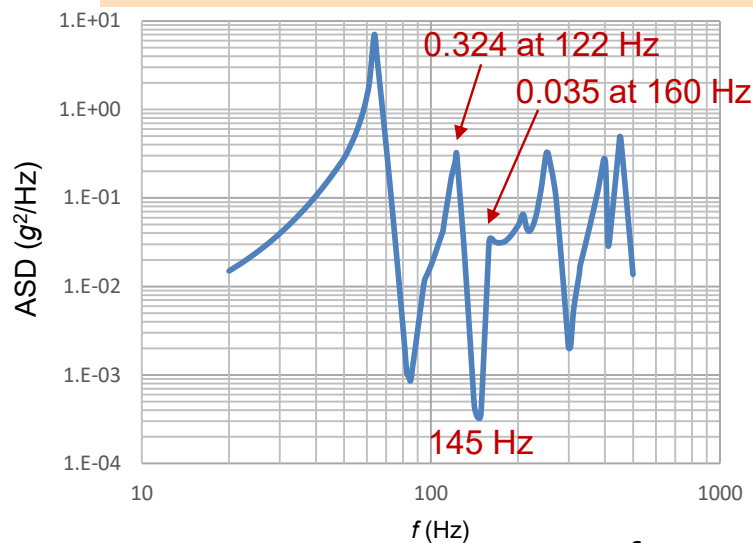
$f_{\text{max}-f}$ is the frequency of the higher of the two S_{FF} peaks on each side of the antiresonance

Usually, $f_{\text{max}-f} = f_{\text{max}-a}$

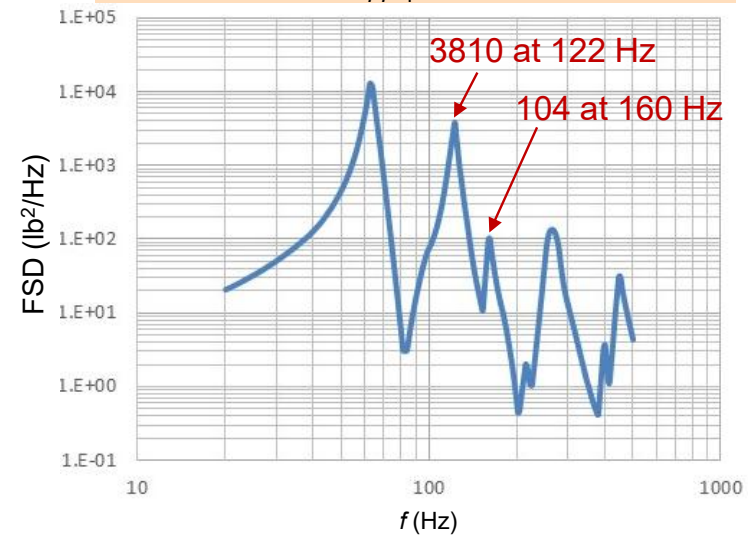
Let's do this for our example ...

Step 5: Derive a Value for C^2 (cont'd)

Interface ASD, S_{AA-1} , for base driven SC



Interface FSD, S_{FF-1} , for base driven SC

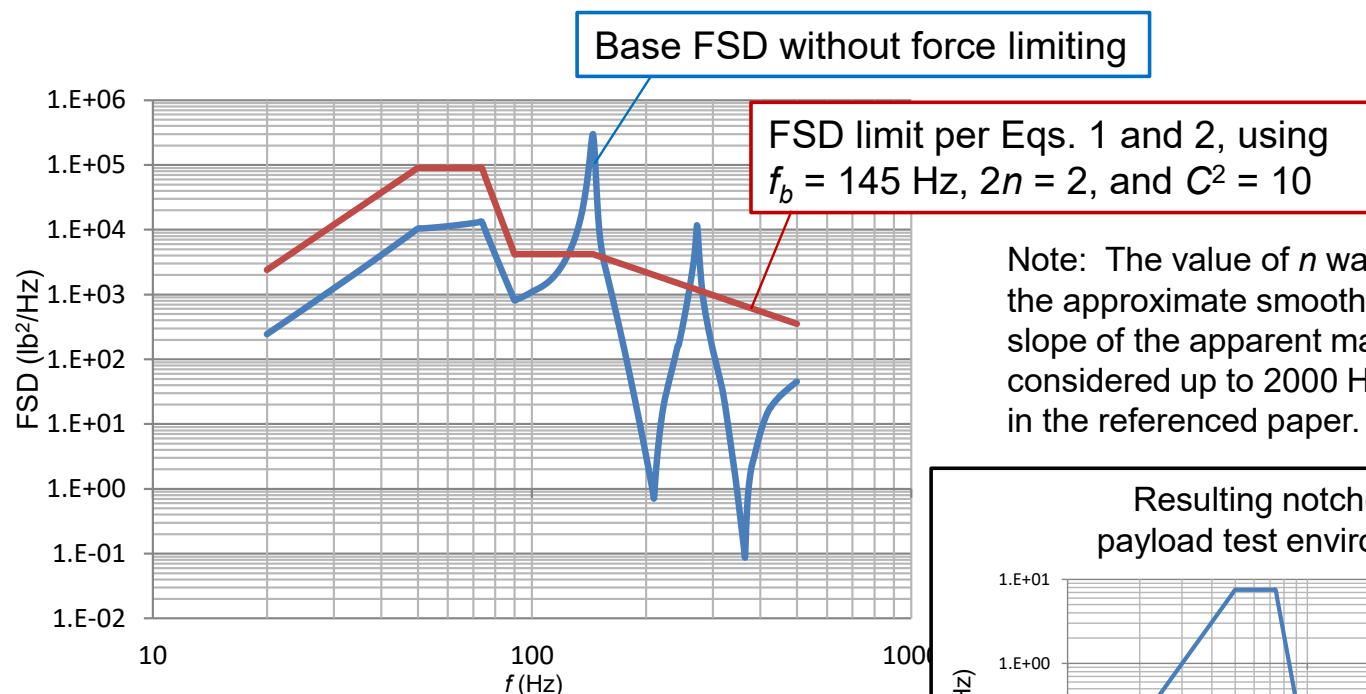


$$f_{\max-f} = f_{\max-a} = 122 \text{ Hz}$$

$$C^2 = \frac{S_{FF-1}(f_{\max-f})}{M_0^2 \cdot S_{AA-1}(f_{\max-a})} = \frac{3810}{(34.6)^2 (0.324)} = 10$$

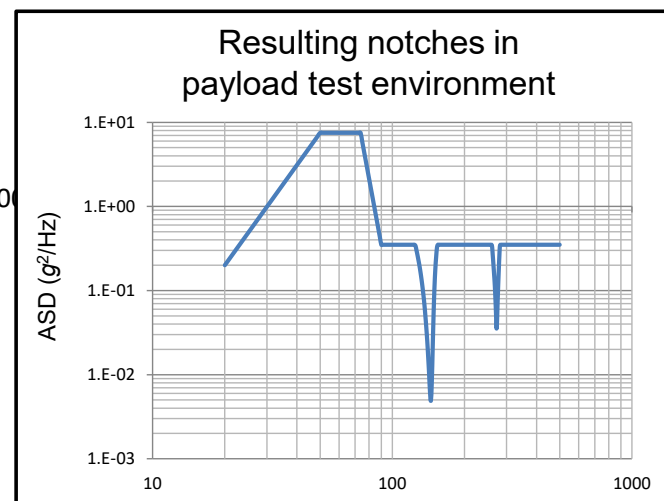
This is about as far as the referenced paper carried the example.
We'll continue it here...

Step 6: Define the FSD Limit



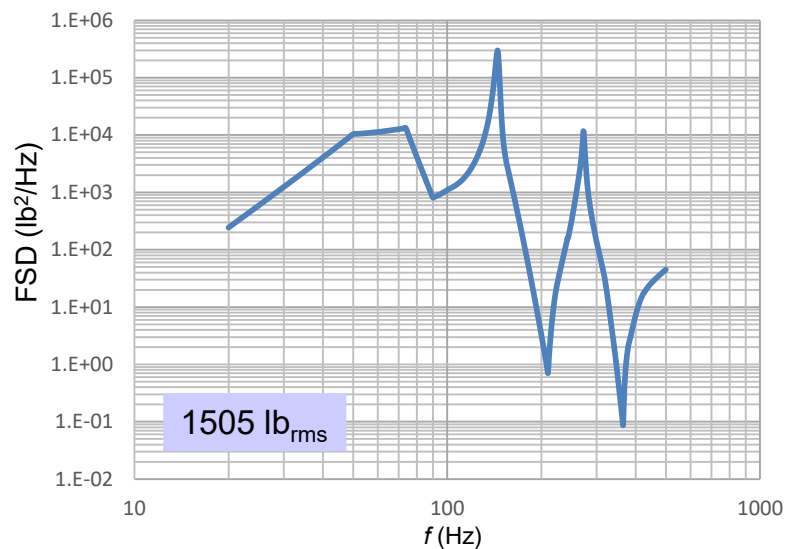
Note: The value of n was based on the approximate smooth-line average slope of the apparent mass when considered up to 2000 Hz, as shown in the referenced paper.

The FSD limit at the payload's fundamental frequency is 3810 lb²/Hz, same as the peak value near that frequency for the coupled bus-payload system.

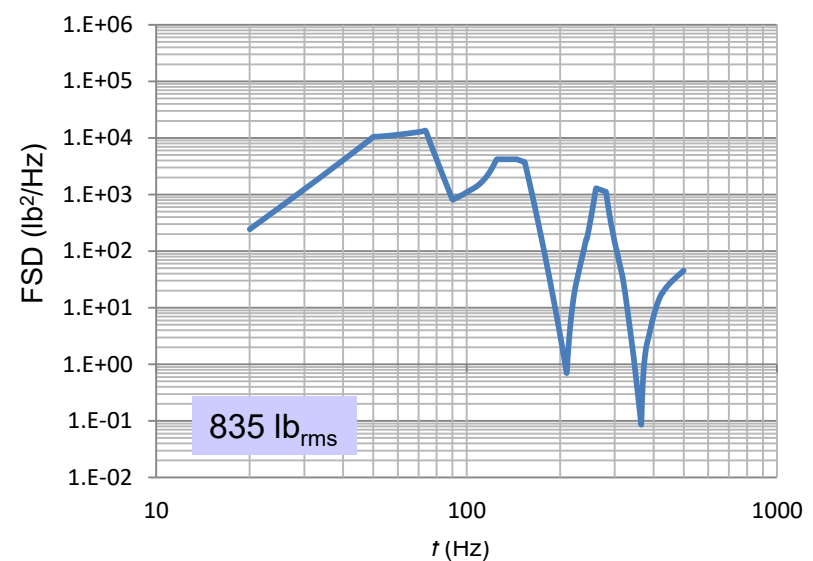


Step 7: Assess the Effects of Force Limiting

Payload-base FSD without force limiting



Payload-base FSD with force limiting



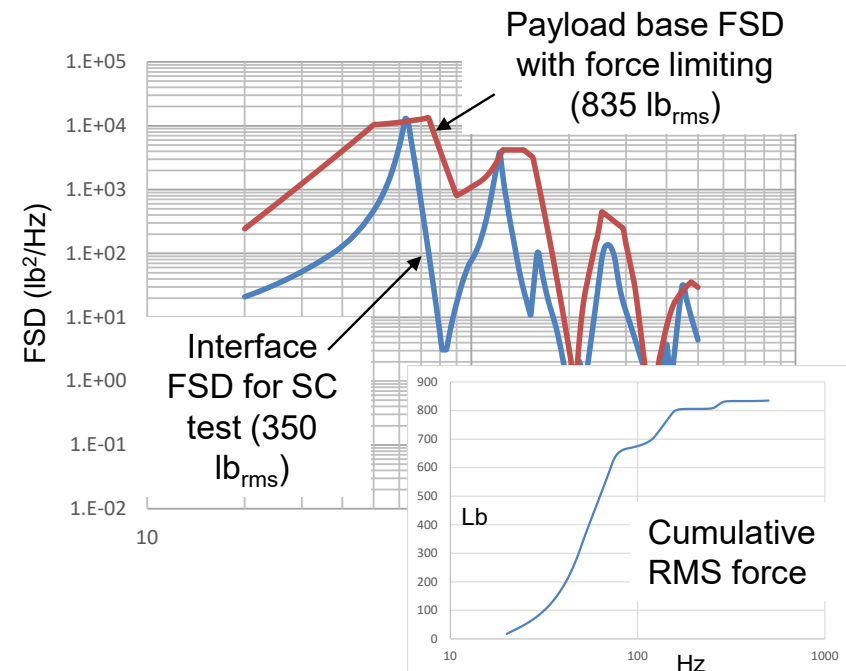
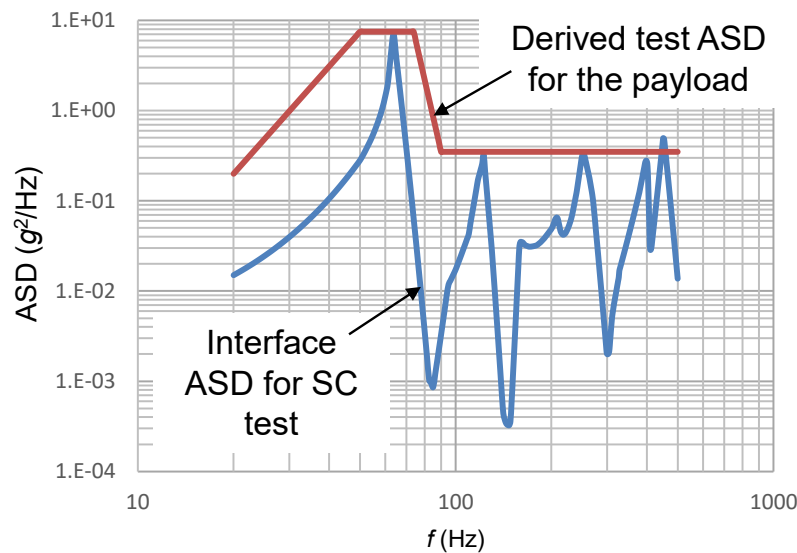
The RMS base force has dropped by about 45%.

But it's still well more than twice the 350 lb_{rms} interface force for the SC test!

What went wrong in this example problem?

Two Reasons for the High Payload Base Force

1. The derived payload test environment is an envelope of peaks and thus is considerably more severe for the payload than the SC test.
2. Most of the RMS base force is below 100 Hz, where there is no force limiting.



Let's look at some options for addressing this problem ...

Option 1: Accept the Over Test

We could simply accept the higher load during the payload test ($835 \text{ lb}_{\text{rms}}$).

But, for this example, the payload weight is only 34.6 lb, and the 3σ load is $3 \times 835 = 2505 \text{ lb}$.

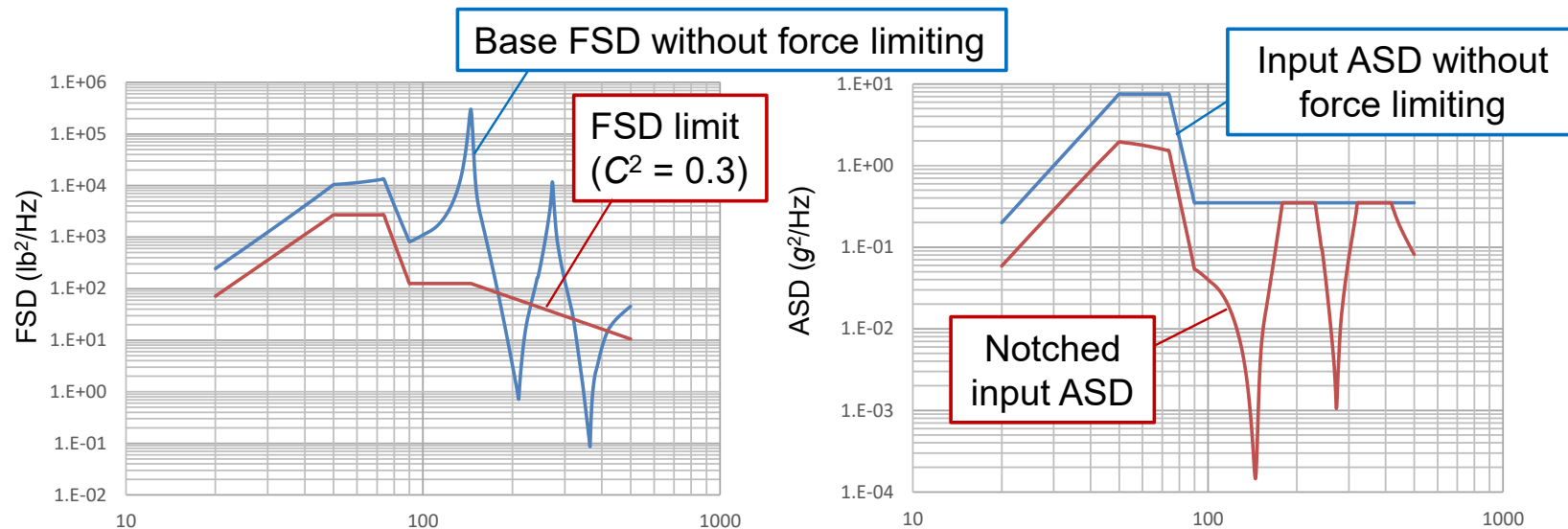
This means we would have to design the payload's primary structure to a limit quasi-static equivalent load of at least $2505/34.6 = \mathbf{72 \text{ g!}}$

And $\mathbf{121 \text{ g}}$ if we choose to design to a 5σ load!

Option 2

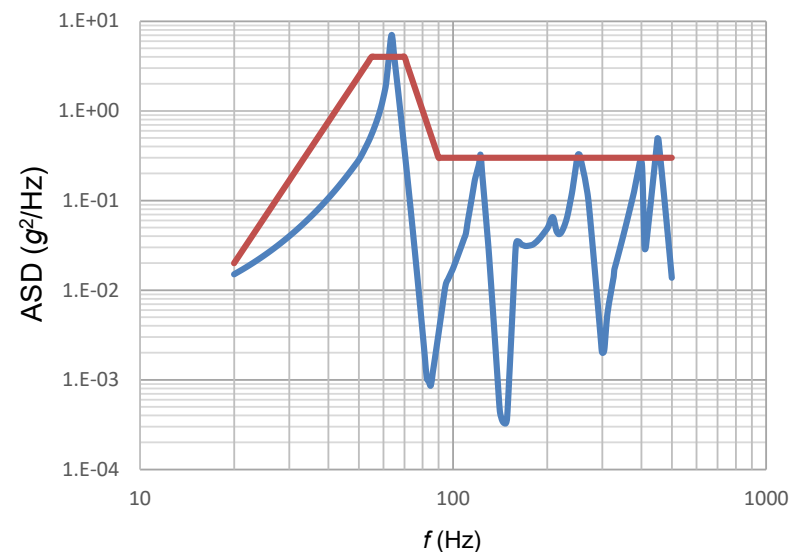
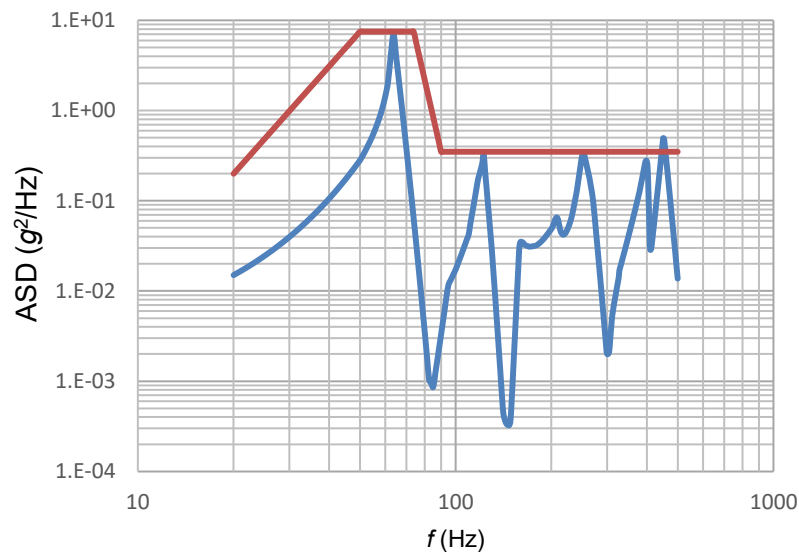
What if we set C^2 equal to the value that makes the RMS base force for the payload test equal the RMS interface force for the SC test?

The RMS interface force for the SC test (not force limited) is 350 lb. To keep from exceeding 350 lb in the payload test, we'd have to use a **C^2 of 0.3**, with these results:



This is not a technically justifiable option. By compensating for the conservatism from the test envelope and trying to protect the primary structure, we would undertest the payload in broad frequency ranges.

Option 3: Derive a Less Conservative Payload Test Environment

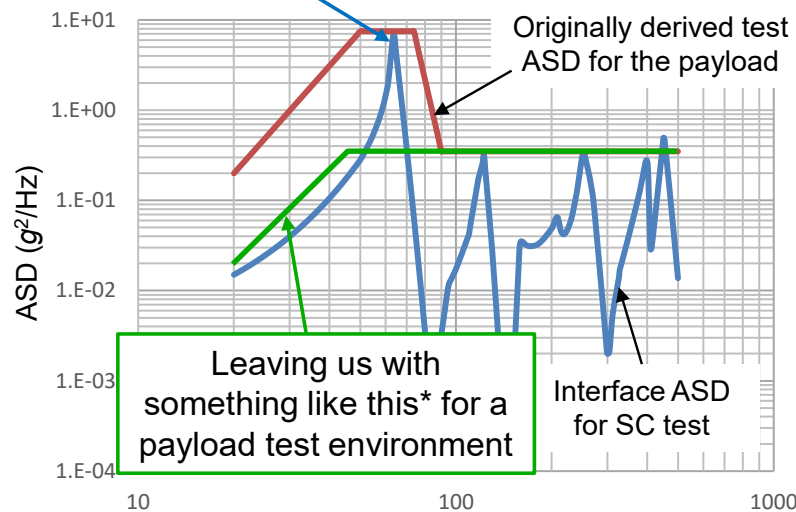


With the less-conservative enveloping shown above right, the base force for the force-limited ($C^2 = 10$) payload test drops from 835 lb_{rms} to 638 lb_{rms} (vs. 350 lb_{rms} interface force for the SC test).

Helps, but doesn't solve the problem

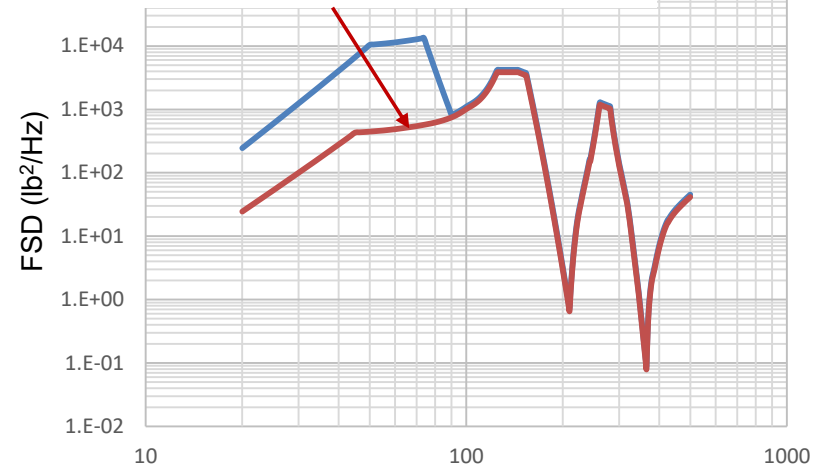
Option 4: Convince our Customer to Force Limit the SC Test, Also

If the SC test is force limited, this peak in the interface ASD goes way down.



*I do not have the data needed to derive force limits for the SC test in this example.

Resulting drop in base FSD for the force-limited payload test when using the same C^2 value of 10 as before (base force reduced from 835 lb_{rms} to 508 lb_{rms})



Base force in the force-limited payload test drops from 835 lb_{rms} to 508 lb_{rms}, but it's still well higher than the 350 lb_{rms} interface force from the non-force-limited SC test. And the interface force would be much lower than 350 lb_{rms} if the SC test is force limited. So using $C^2 = 10$ still results in a significant overtest of the payload's primary structure.

Option 5: Break the Payload Test into Two Tests in combination with Option 3, less conservative enveloping for payload levels

Assuming the SC test is not force limited, break the payload test environment from Option 3 (less conservative enveloping) into low- and high-frequency environments, and test separately to them:

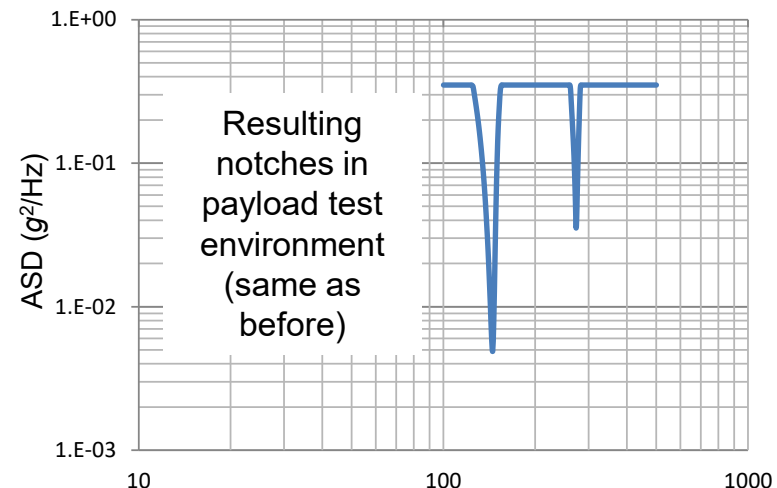
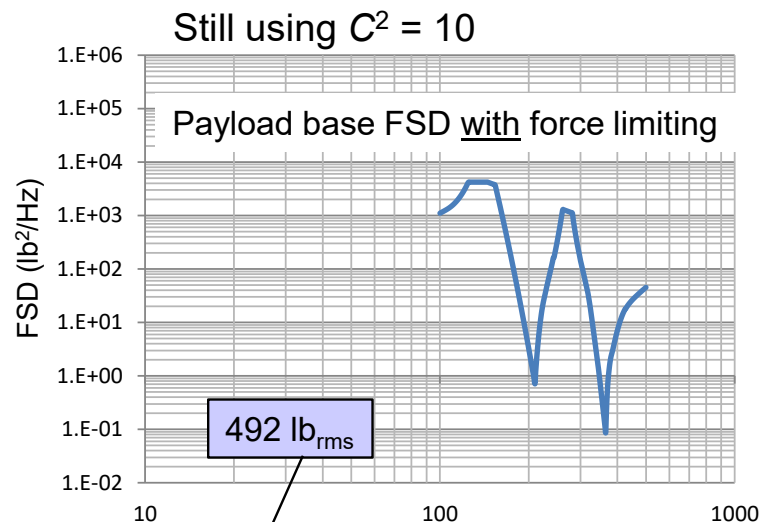
1. Low frequency, from 20 – 100 Hz without force limiting (base force is 448 lb_{rms})
2. High frequency, from 100 – 500 Hz with force limiting (base force is 492 lb_{rms})

Rationale: With a fundamental frequency of 145 Hz, the payload will respond mostly as a rigid body below 100 Hz, which would not contribute much to loading of payload components.

We still want to do the 20 – 100 Hz test because there may be other small components, not included in the FEM, that are susceptible in that frequency range.

If we find from a low-level run that there is a significant mode near 100 Hz, the low- and high-frequency tests should be overlapped with this strategy, such as 20 – 110 Hz and 90 – 2000 Hz.

Results of Force Limiting a 100 – 500 Hz Test



vs. 350 lb_{rms} for the non-force-limited SC test and 835 lb_{rms} when force limiting a 20 – 500 Hz payload test

Still exceeds the interface force for the SC test by 41%.

But, if the SC test will not be force limited, this option (#5) may be best for balancing the risk of over testing the primary structure with that of under testing items in the payload.



Summarizing the Options

Option	Interface force in SC test, lb_{rms}	Base force in payload test, lb_{rms}	Comments
1. Accept the over test	350	835	72 g equivalent quasi-static load at 3 sigma
2. Set C^2 as needed to match the RMS interface force	350	350	Unacceptable; significant undertest within broad frequency ranges
3. Derive less conservative payload test levels	350	638	A step in the right direction
4. Force limit the SC test	< 350	508	Looks like the best option, but base force is still considerably higher than for the SC test
5. Break payload test into low- and high-frequency tests (combined with Option 3)	350	492	May be best if the SC test is not force limited

Conclusions from This Example

**There is no single “right” way to implement force limiting.
We often must use judgment in finding the best path forward.**

Certainly, in situations like this one, it is more flight-like for both the SC test and the payload test to be force limited.

But anytime we derive a test environment by enveloping peaks, as is usually done, the test will be more stressful than flight.

The objective of force limiting is to account for impedance difference in test, not to compensate for the effects of enveloping.

A key point:

We must decide on the test strategy early so that we can design the bus and the payload for the test loads!

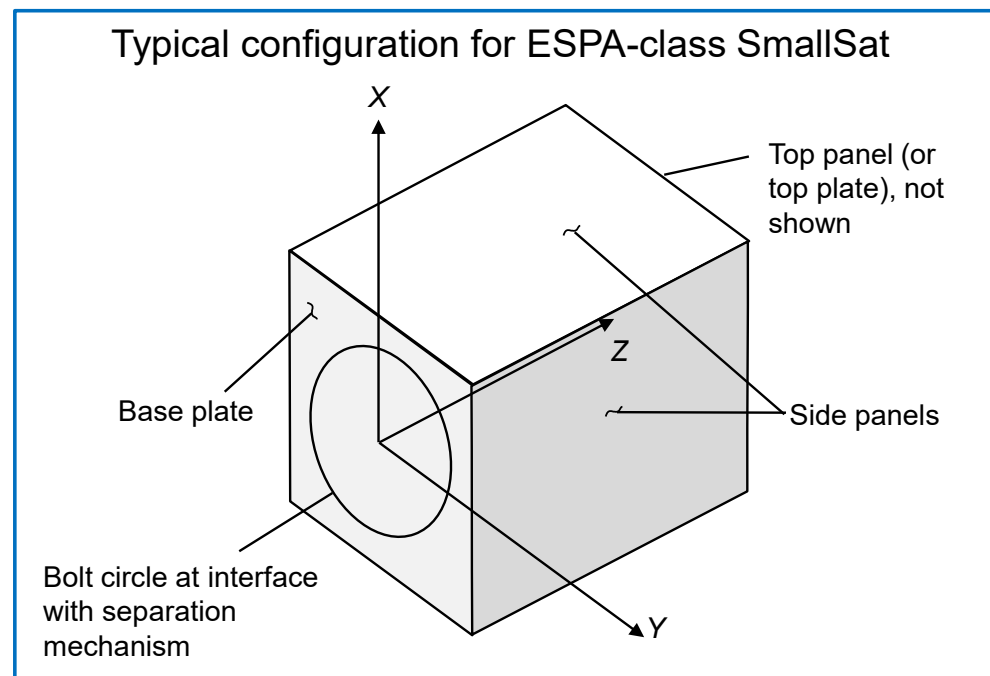
7. Designing a SmallSat To Pass the Vibration Test

- Introduction
- General Strategies
- Making Load Paths Direct
- Separating Modal Frequencies
- Panels Used to Mount Components and Mounting Strategies
- Understanding How the Primary Structure Carries Loads
- Bolted Joints
- Avoiding Component Failure during SmallSat Testing
- Using Appropriate Design Loads

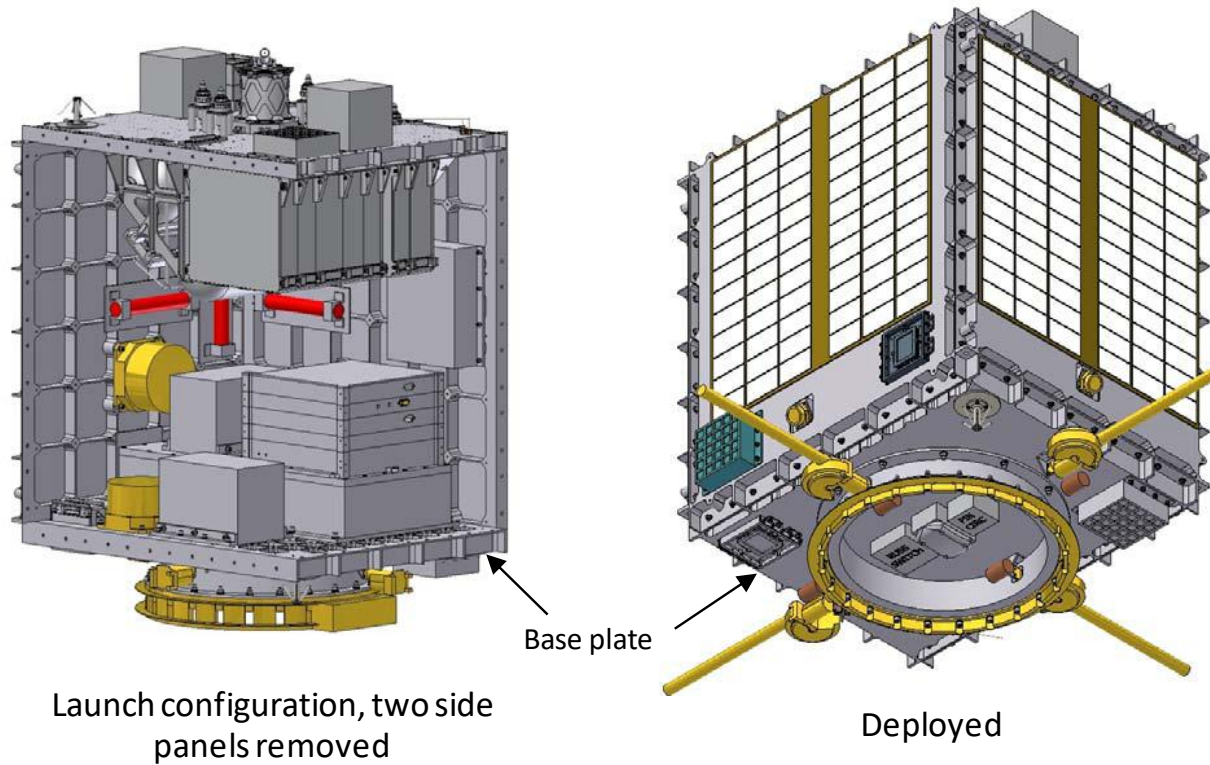
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Introduction

- In a basic sense, a SmallSat consists of structure and components (things the structure supports).
- The main structure, often referred to as the **body structure**, **bus structure**, or **primary structure**, typically consists of base plate, top panel, and side panels.
- The body structure can have a cross section (X-Y plane) that is square, rectangular, hexagonal, octagonal, or other shape.
- Components mount to the panels or to interior decks or columns.
- Some components are stacked.



Example: FalconSat-5 Configuration



(Images from Ref. 20)

Common Failures 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 soldered joints
- Threaded fasteners loosening or backing out completely
- Detrimental permanent deformation (e.g., misalignment that impacts performance) from materials yielding or joints slipping
- Unseating of electrical connectors

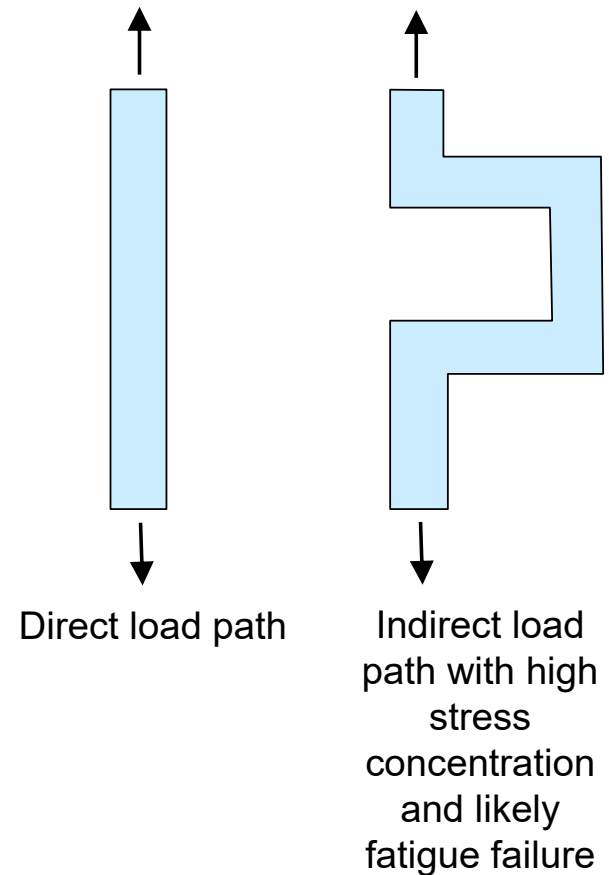
When a ductile material ruptures in a random vibration test, it's usually a **fatigue failure** (failure caused by crack formation and growth under cyclic loading, usually in a region of high stress concentration) rather than a **strength failure** (inability to withstand the highest load).

Accordingly, avoid sudden changes in cross section (e.g., sharp fillets), which lead to high stress concentrations, in parts that are highly loaded.

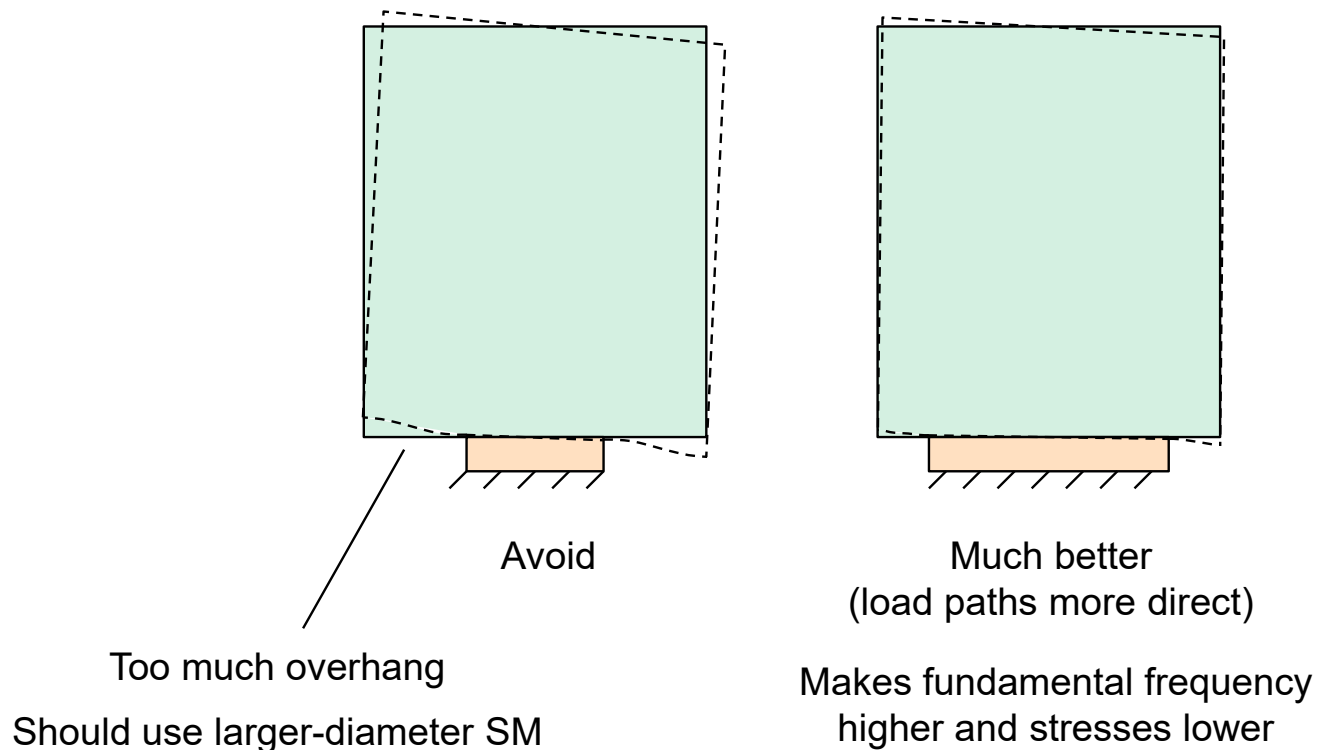
And avoid high cyclic stresses in parts that have unavoidable sudden changes in cross section, e.g., adequately preload bolted joints.

General Strategies

- Design direct load paths.
- If weight is not critical, don't finesse the structure: make it a brick. ("BrickSat")
- Make key test decisions—such as derivation of sine burst loads and whether to notch the random vibration test—before releasing the structural design so that you can design the structure for the test loads.



Load Path Example:
Match the SmallSat Body to the Separation Mechanism



Nearly any design decision that increases natural frequencies (other than changing materials) will also reduce stresses in the materials and lead to a better chance of withstanding the test.

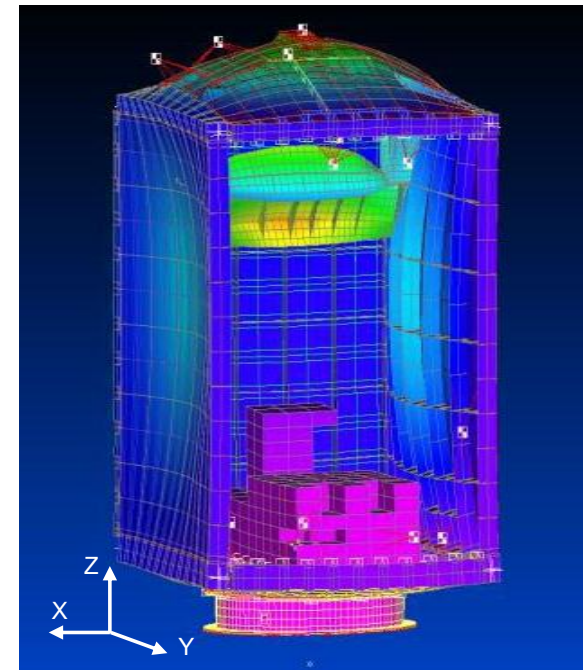
Separating Modal Frequencies

Modal coupling can lead to high panel loads and component loads.

Example shown at right: The top panel is bending as part of the SmallSat's fundamental axial mode—a common situation.

Strategy: Separate the modal frequencies to reduce loads. Aim for about 2-to-1 separation.

1. Calculate the fundamental frequency of the SmallSat with the top panel rigidized but mass loaded.
2. Design the mass-loaded panel so that, when grounded at the panel edges, it has a fundamental frequency either > 2 times or < 0.5 times the frequency calculated in Step 1.



Top panel mode coupled with the SmallSat's fundamental axial mode

We'll look at an example of a modal coupling problem in Sec. 8, FalconSat-2 case study.

Panels Used To Mount Components

Panels that mount components should be made of either sandwich construction or machined plates with ribs in a pattern such as isogrid (equilateral triangles) or orthogrid (rectangles).

This includes the base plate, the side panels, the top plate, and any internal component-mounting decks.



Orthogrid panels

Do not mount components on sheet metal. Doing so will cause multiple low-frequency modes and may lead to fatigue failure of the sheet metal.

Selecting Component Locations and Mounting Schemes

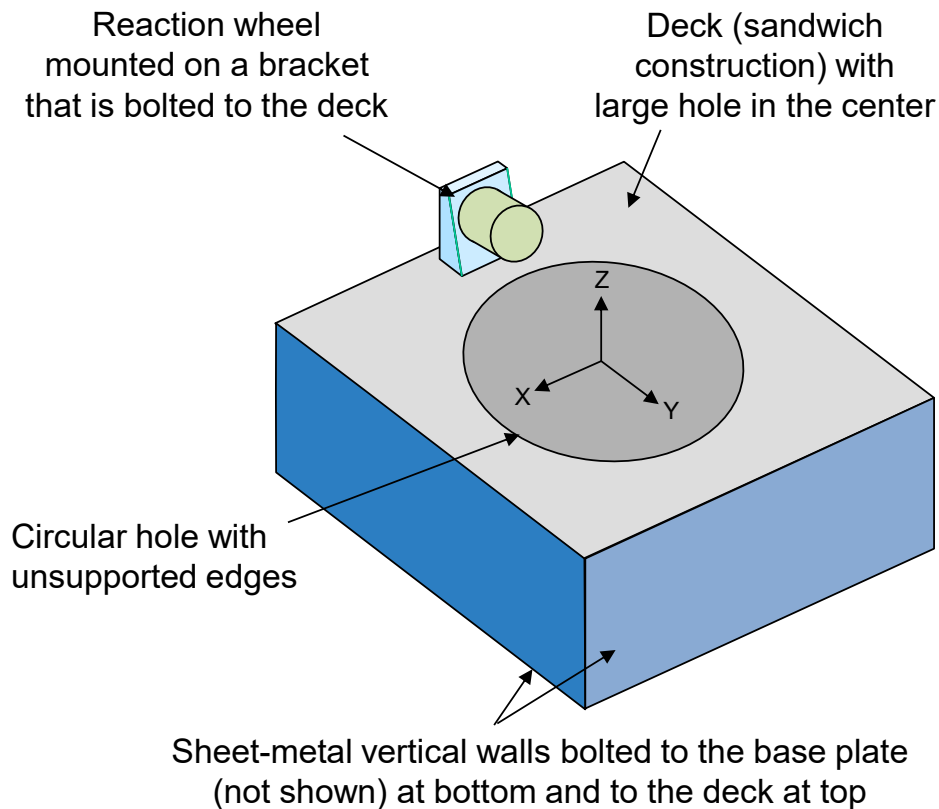
Guideline: Put heavy items aft, on or near the base plate.

- Lowers CG, increases fundamental lateral frequency, reduces base moment
- Can't always follow this guideline; instruments are often put on the top panel to provide the best field of view and get them away from the source of separation shock

To better allow separation of modal frequencies, mount components close to panel edges rather than the panel center.

- Increases the first bending frequency of the loaded panel, reduces panel stress, and reduces component acceleration
- But make sure the panel edge is constrained by the mating panels against free rotation. See example on next page.

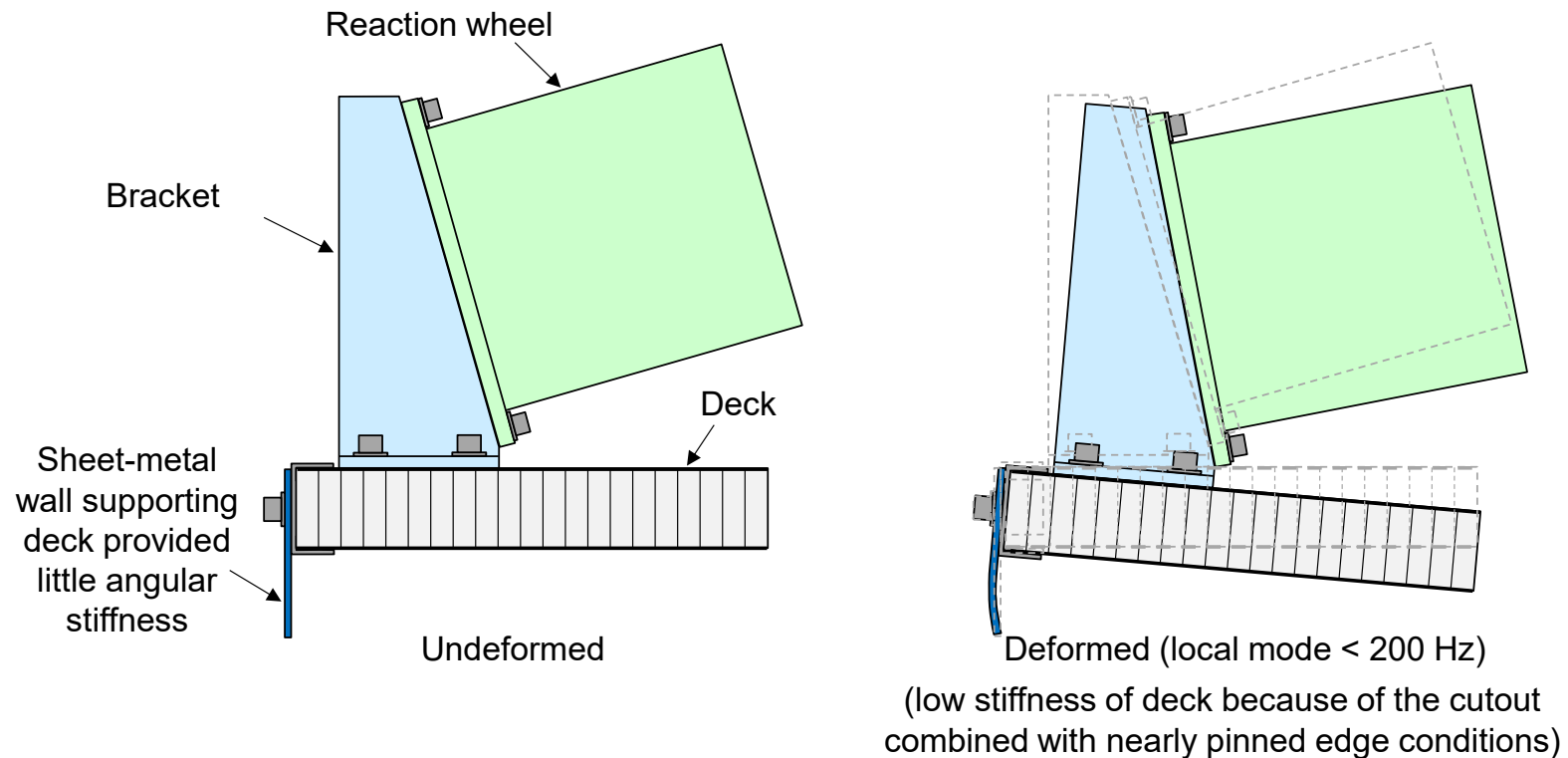
Example: Mounting of a Reaction Wheel



See next page for the problems encountered as a result of this method of mounting.

continued

Example, continued: High Response Loads during Spacecraft Random Vibration Test

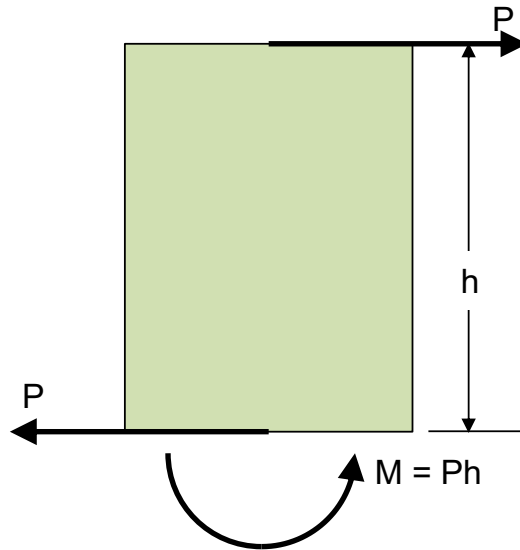


Response levels were much higher than the GEVS (Ref. 12) $14.1 g_{rms}$ qualification environment used for testing the reaction wheel.

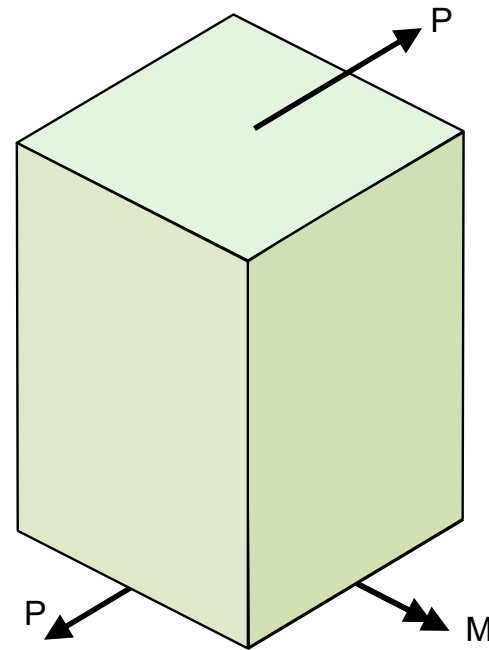
Resolving this problem cost a great deal of money and schedule time.

Lateral Loading Typically Drives the Design of the Primary Structure

Inertia loading during launch is a distributed load, but it's easier to understand the load paths if we envision concentrated loads.

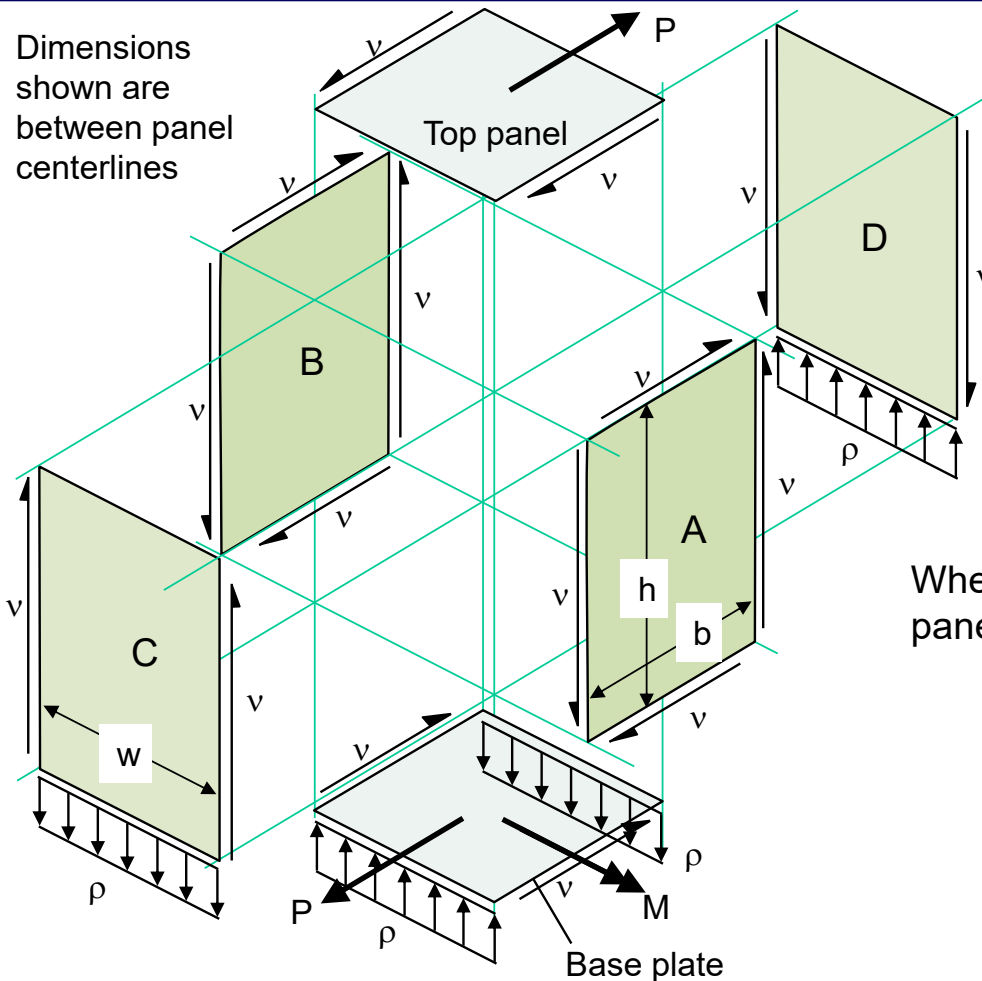


a. 2-dimensional view



b. 3-dimensional view

Understanding How the Primary Structure Carries Loads



Not shown for simplicity is the moment carried by panels A and B.

The main roles of panels A and B are to carry lateral load and to shear loads into panels C and D so that C and D can react the moment.

When ignoring the moment carried by panels A and B, the running loads are

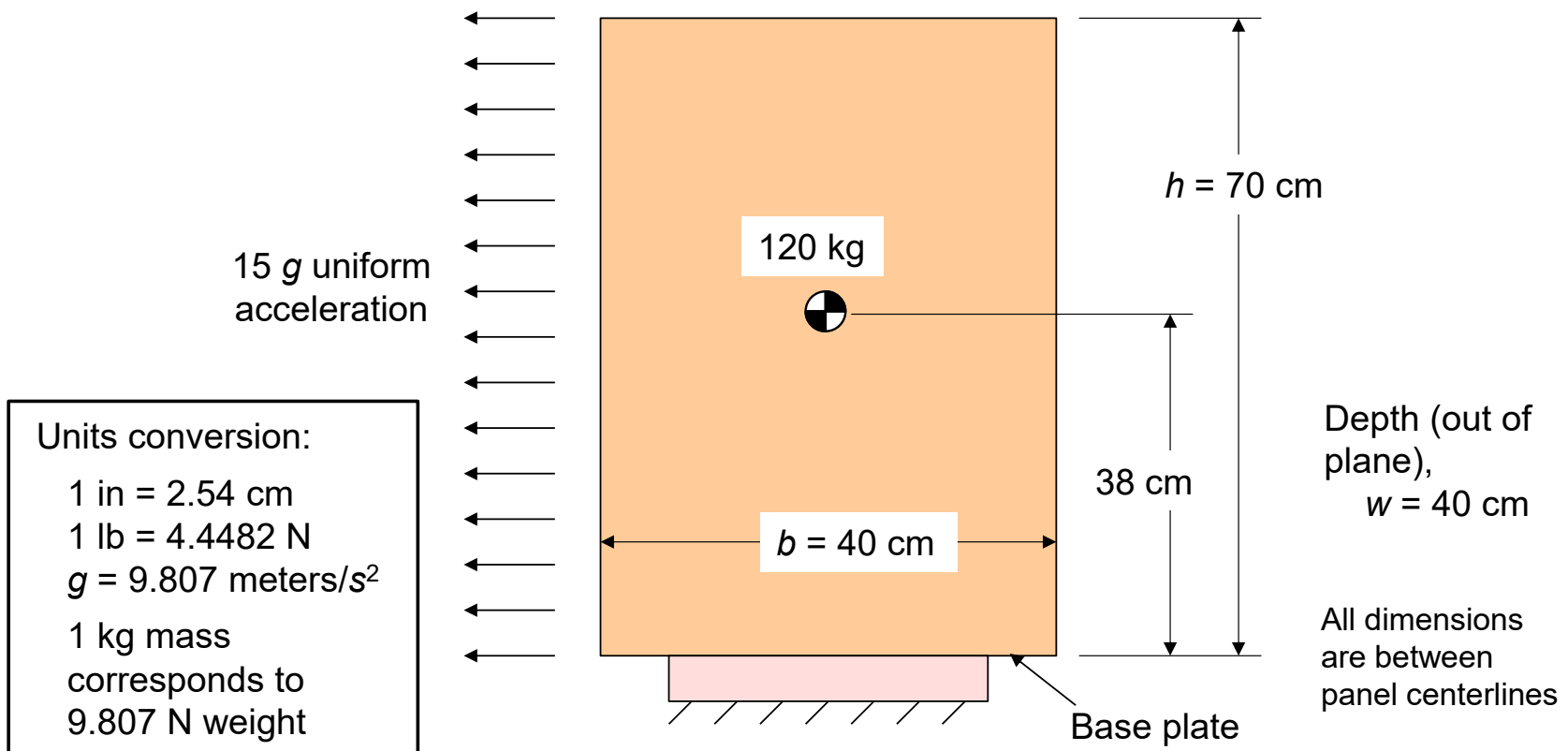
$$v = \frac{P}{2b} \quad (\text{Eq. 7-1})$$

$$\rho = \frac{2vh}{w} \quad (\text{Eq. 7-2})$$

Units: force/length, such as lb/in

Class Problem 7-1: Estimating Running Loads on Panel Edges

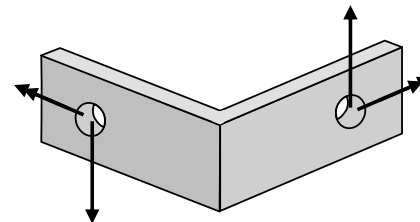
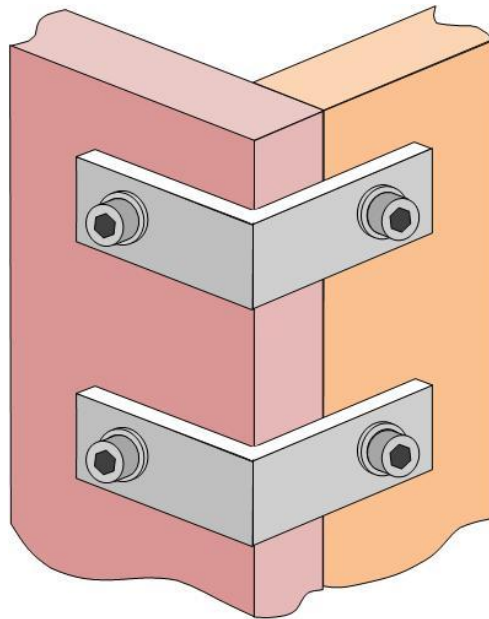
Problem statement: For a quasi-static load of 15 g , estimate running loads v and ρ , as defined on the previous page, in units N/cm and lb/in.



Bolted Joints: Attaching Panels in the Primary Structure

Start by reviewing how the primary structure carries loads, then make sure the attachments can efficiently carry the loads between panels.

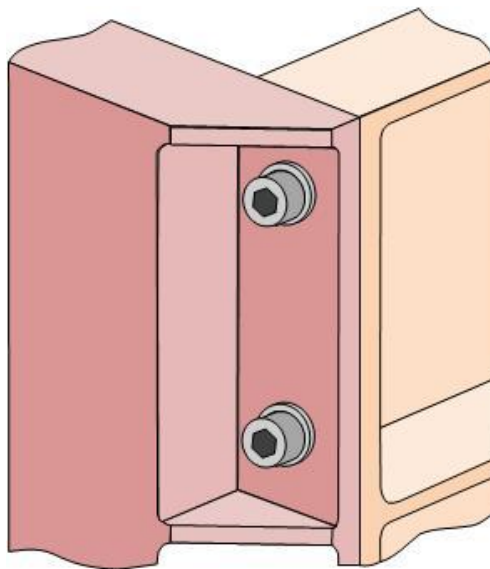
Example of what **NOT** to do:



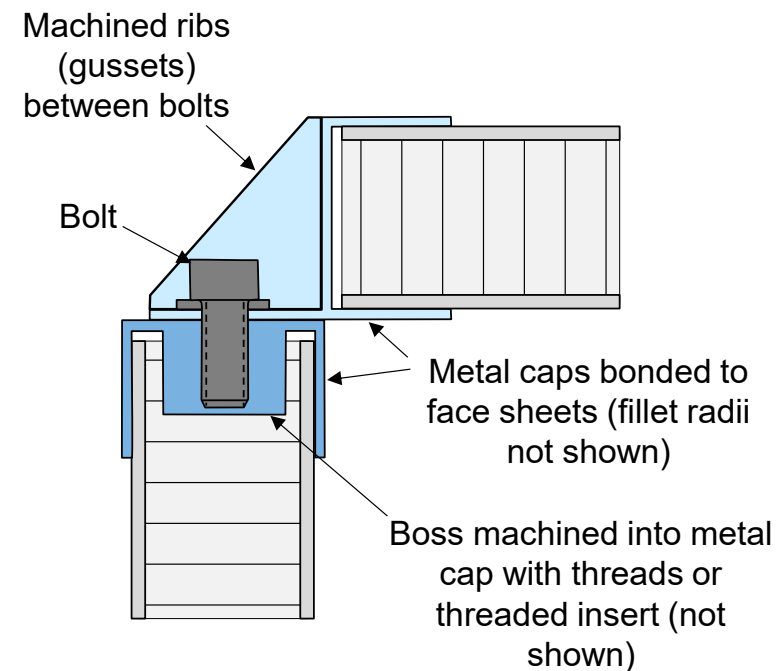
Free-body diagram of clip
when transferring shear
loads between panels

Each leg of the clip will be in torsion and will pry on the bolt head, resulting in low strength and low stiffness.

Bolted Joints: Better Ways to Attach Panels



Machined metal panels



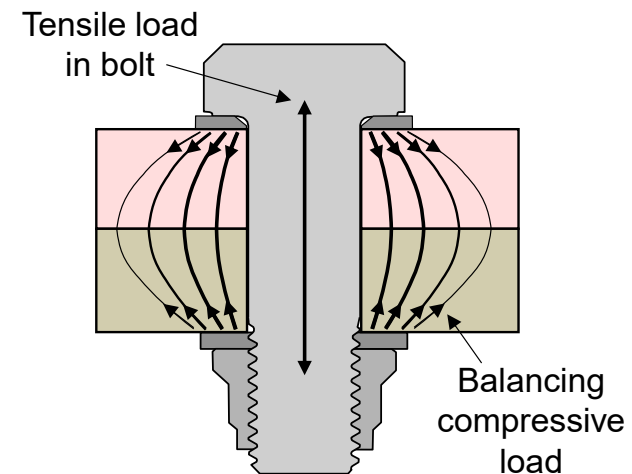
Sandwich panels

Bolted Joints: The Importance of Preload

The most important aspect of a bolted joint is **preload**—the tensile load in the bolt developed during installation, balanced by the equal compressive (“clamp”) load between mating parts.

It’s the clamp load that’s important!

- Makes joints stiff
- Keeps bolts and joint members from failing in fatigue
 - Stress in bolts and joint members doesn’t change much under applied loads.
- Prevents joints from slipping back and forth in clearance holes
 - Cyclic slip makes structural behavior hard to predict, causes loss of preload, and causes bending stress in the bolt that can lead to fatigue failure



In design, we want to ensure preload is high enough to prevent (a) gapping (total loss of clamp load) and (b) slipping within clearance holes under the limit loads for the mission—and the limit loads for test as well.

Adapted from my course “Design and Analysis of Bolted Joints” (DABJ), Ref. 21

For a Given Bolt Diameter and Materials, Preload Tends to be Nearly Proportional to Torque

$$T = KDP_{pi} \quad \text{or} \quad P_{pi} = T/KD$$

T = effective torque = total torque minus running (resisting) torque from locking feature

D = bolt nominal diameter

P_{pi} = initial preload (subject to relaxation and thermal effects)

K = nut factor, which accounts for friction, geometry, and other variables; derived from test data

With typical material combinations for fastening hardware, the nut factor, K , when installed without lubrication is usually between 0.20 and 0.30 but can be higher than 0.50.

With lubrication, K tends to be between 0.1 and 0.25 depending on lubricant and lubrication process.

- We can get higher preload with lubrication. High K leads to high torsional stress in the bolt, which limits the achievable preload.
- Acceptable lubricants for space use: molybdenum-disulphide dry film and Braycote 601EF and 602EF grease

For a given applied torque, preload is highly uncertain and will vary considerably between bolts.

Adapted from DABJ (Ref. 21)

Recommendations for Achieving the Desired Preload

- Best when practical: Do your own torque-preload tests to determine the range of nut factor for the joints you're designing.
- If you don't have the budget or ability to do your own tests, research available torque-preload test data. Then, to improve accuracy of predicted preload for your design, use the exact same ...
 - Bolts
 - Nuts or inserts
 - Washers
 - Lubricant
 - Lubrication process
- If you can't find test data that apply to your design, do a simple test to determine how highly you can torque your fasteners:
 - Torque a few fasteners to failure, using the same combination of materials, fastening hardware, and lubrication as for the flight design.
 - Then, for flight fasteners, back off a safe amount (perhaps 40%) from the failure torque.

Bolted Joints: Fastener Selection

Use high-strength bolts for structural applications.

- Example: The most commonly used fastener in the space industry is the socket head cap screw, controlled by specifications such as NAS1351 (fine thread) and NAS1352 (coarse thread).
 - Use the A-286 (“heat resistant steel”) version (160 ksi ultimate tensile strength) for any fastener that can see significant applied load
 - The “corrosion resistant steel” version has very low strength, especially yield strength (~30 ksi)

Use larger-diameter bolts than you think you need.

- If you have room: no smaller than #10 (0.190” dia.) or M5 (metric) for primary structure of SmallSats weighing > 100 lb
- An A-286 NAS1351 #10 screw has a tensile load rating of 3200 lb
- More than you may need—but will result in fewer problems; usually doesn’t add much weight

Bolted Joints:
Example of Load Capacity Relative to Preload

We should be able to preload a #10 A-286 bolt to at least 1000 lb (and as high as about 2400 lb), given variation in the nut factor. So, with an assumed 0.2 coefficient of friction* between mating parts (joint members), each fastener location can carry $0.2(1000) = 200$ lb in shear without slipping.

In Class Problem 7-1, we estimated running loads of 68.4 lb/in (transverse along edges) and 239 lb/in (normal to bottom edges of side panels, in plane). If we space the bolts two inches apart, the bolt loads are

$$2(68.4) = 137 \text{ lb} \Rightarrow \text{No slip}$$

$$2(239) = 478 \text{ lb} \Rightarrow \text{Slip, so orient the bolts to be in line with the load (tension rather than shear)}$$

*The coefficient of friction (CoF) between metallic joint members is usually higher than 0.2, but NASA-STD-5020B (Ref. 22) says not to count on 0.2 except for cleaned, bare-metal surfaces; otherwise, the CoF can be as low as 0.1.

Bolted Joints: Nuts and Threaded Inserts

Use nuts and threaded inserts that are at least as strong as the bolts.

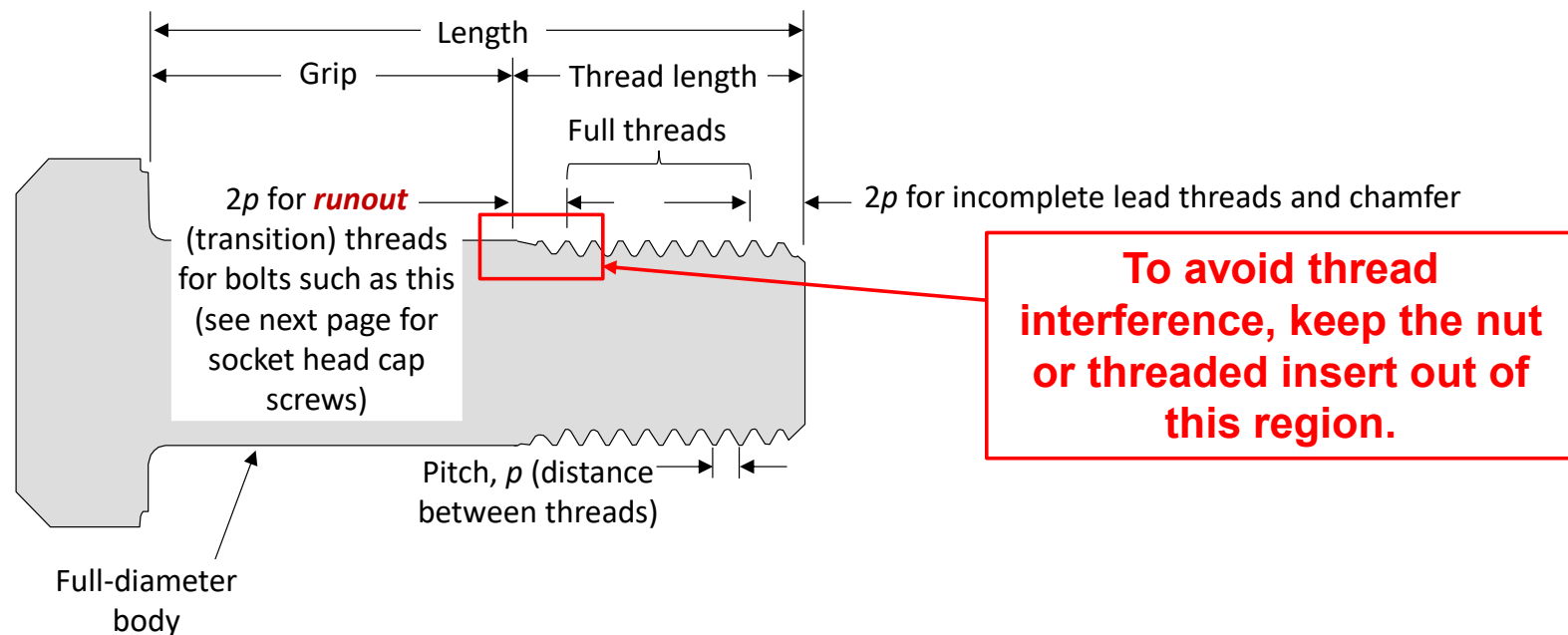
- Check the nut specification, which should include a minimum strength as a requirement to the supplier.
- The strength of a Heli-Coil® is limited by the shear strength of the material in which the insert is installed (“parent material”).
 - Heli-Coils are sold in 0.5D length increments, where D is the bolt’s nominal diameter. Longer Heli-Coils provide greater strength.
 - Example: For a Heli-Coil installed in 6061-T6 aluminum, you’ll need a 2D length—fully engaged in the parent material—to have the same strength as an A-286 160-ksi fastener.
- The strength of a Keensert® is limited not only by the parent material’s shear strength but also by the insert’s load rating, as defined in the insert specification.
 - Keenserts are sold as separate versions: Lightweight, Heavy Duty, and Extra Heavy Duty



A Heli-Coil is a threaded insert made of a coiled wire

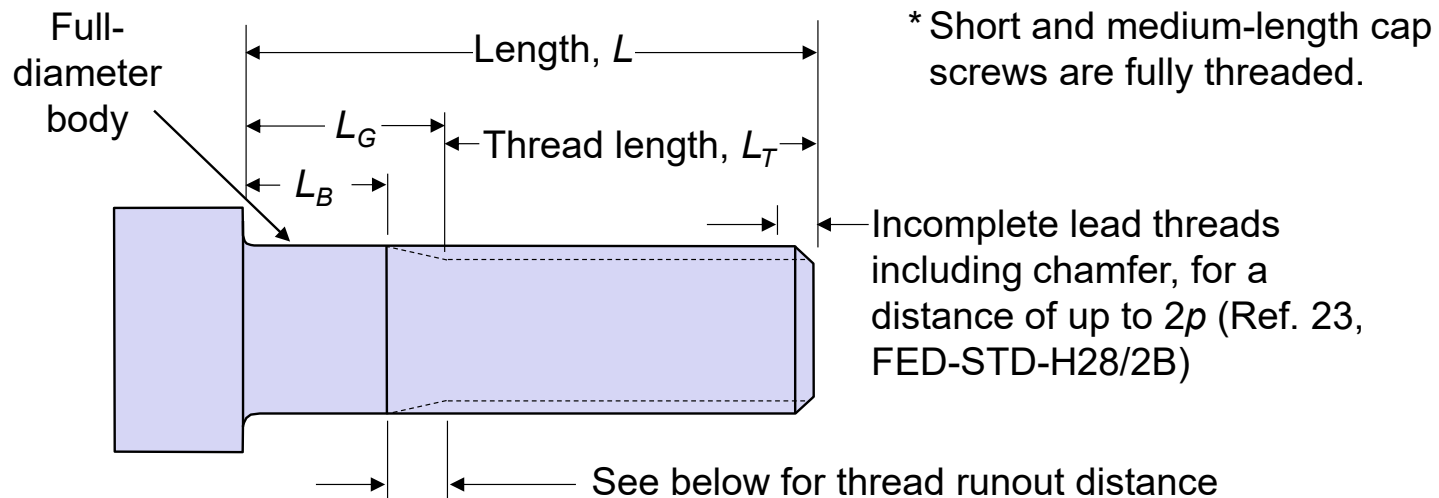
A Keensert is a solid bushing with internal and external threads

Bolted Joints: Avoid Interference that Can Prevent Preload



Adapted from DABJ (Ref. 21)

Dimensions for Relatively Long* Socket Head Cap Screws



L_B is minimum distance from under-head surface to the end of the full-diameter body
 L_G is maximum distance from under-head surface to the first complete thread

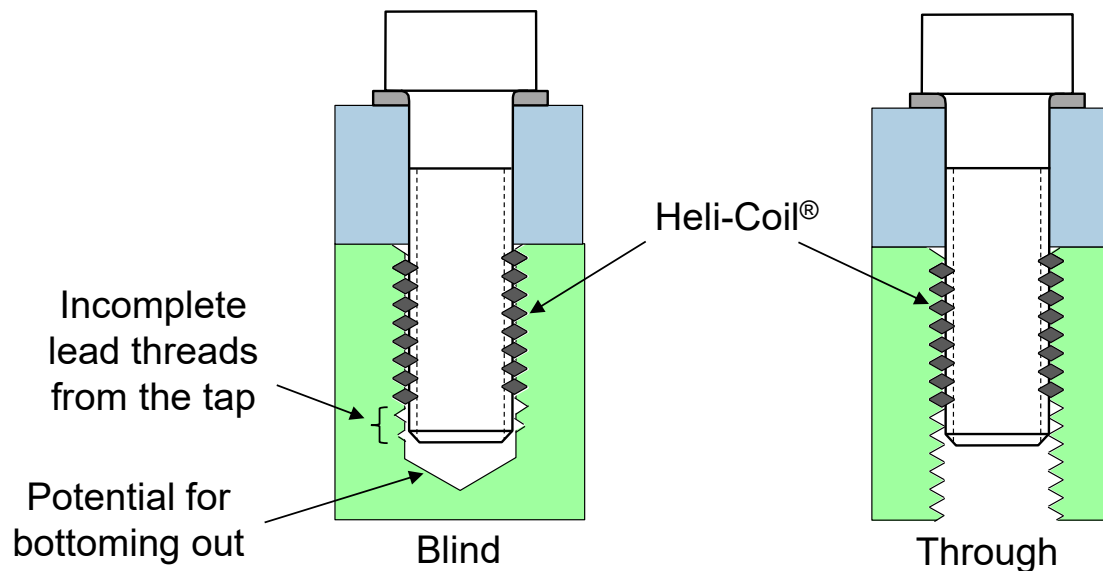
Per ASME B18.3 (Ref. 24), Table 1C, which provides dimensions L_G and L_B , Note 1:

$$L_G - L_B = 5 \text{ times the pitch of the coarse thread (UNRC) for the applicable bolt diameter}$$

Adapted from DABJ (Ref. 21)

Bolted Joints: Fixed Fasteners (with Tapped Holes or Threaded Inserts)

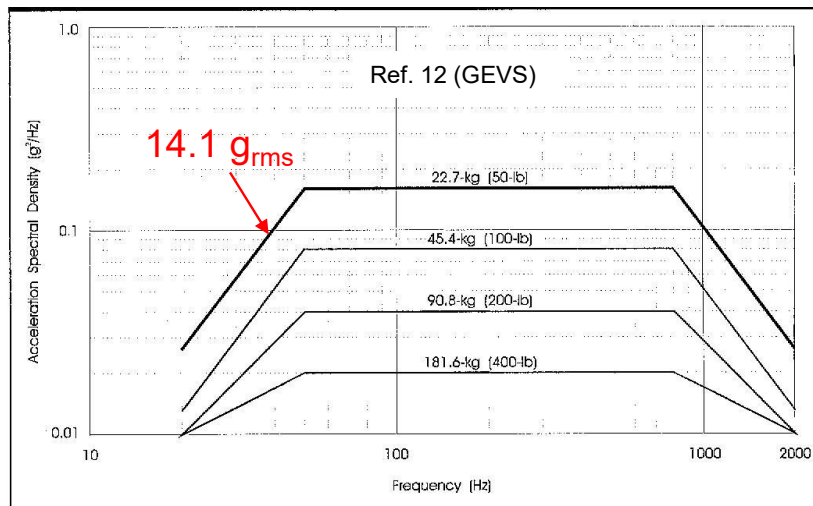
If a blind hole is not necessary, use a through hole to avoid potential interference with the tap's incomplete lead threads or the bottom of the hole.



For additional information on bolted joints, download the PDF file for my DABJ course at <https://instarengineering.com/resources.html>

Avoiding Problems with Components during SmallSat Random Vibration Tests

- Goal: Make sure components are tested adequately prior to integration with the SmallSat.
- Common problem during SmallSat system-level testing: Exceeding levels used for qualification or acceptance testing at the component level.
- Many SmallSat components are qualification tested to the 14.1 g_{rms} GEVS levels (shown below) for spacecraft components weighing up to 50 lb.

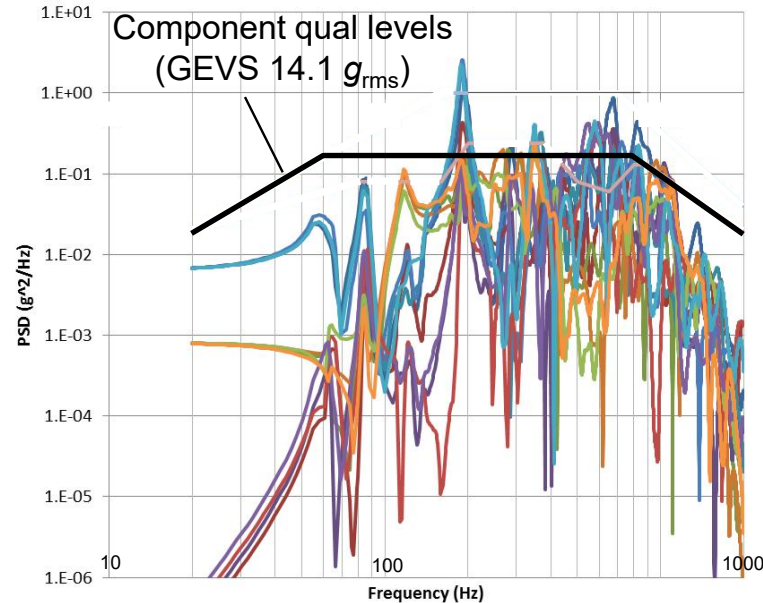


- But the 14.1 g_{rms} GEVS levels **do not** apply to SmallSat components.
- When testing a SmallSat for random vibration, components often see environments that are significantly more severe than the 14.1 g_{rms} GEVS levels in certain frequency ranges.
 - Thus exceeding component test levels

GEVS levels, above, were derived from acoustic testing and analysis of large satellites and thus do not apply for components of SmallSats that will be tested on a shaker!

Example: Exceeding GEVS in Vehicle-Level Test

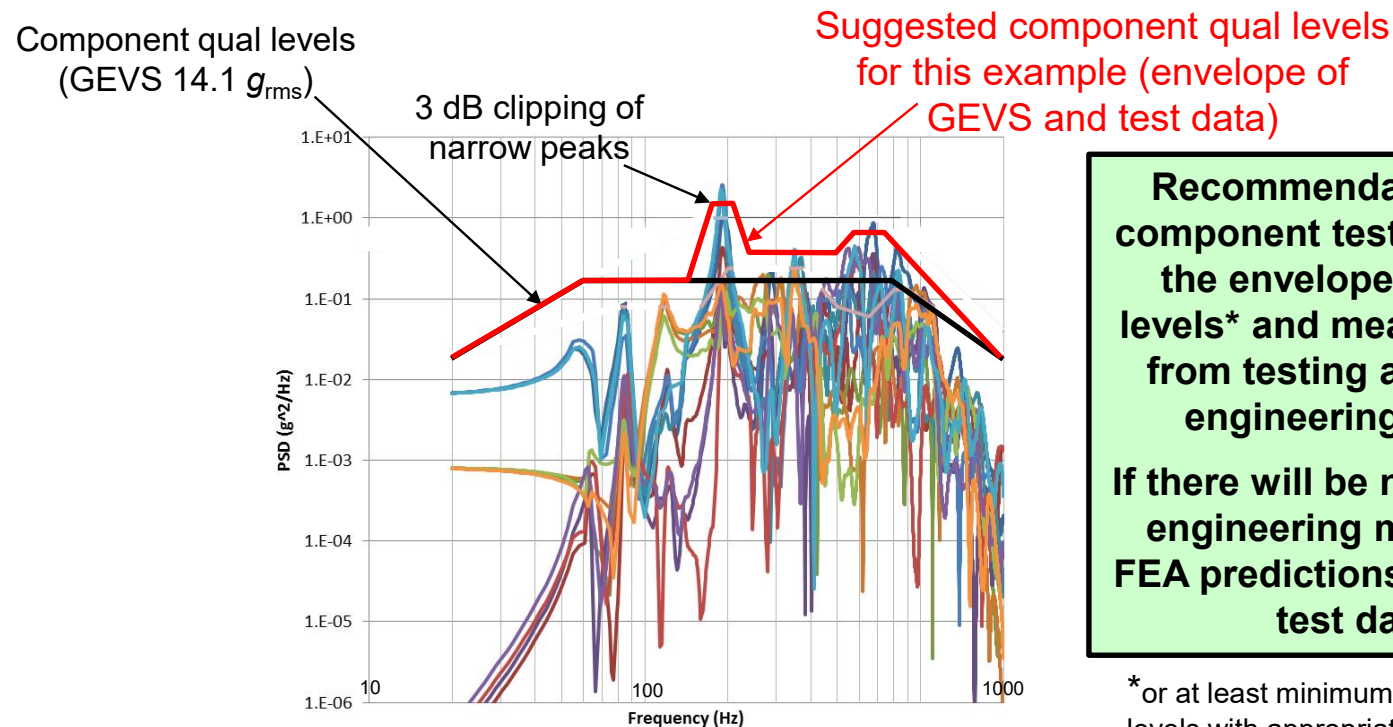
Responses during a force-limited SmallSat-level test at the mounting interfaces for reaction wheels that weighed < 5 lb



SmallSat components should be qualified to levels higher than GEVS.

continued

Example, continued: Deriving Component Test Levels



See earlier slide on separating modal frequencies in order to reduce component levels during SmallSat testing.

Using Appropriate Design Loads

As discussed in Sec. 6 ...

- Coupled loads analysis accounts only for steady-state and low-frequency dynamic loads, often up to only 50 Hz.
- The quasi-static loads or load factors specified in the LV user guide most likely represent the full limit loads for your SmallSat, accounting for the effects of random vibration—but check to make sure.
- Use of a mass acceleration curve may be the most practical way to derive limit load factors for SmallSat components.
- Once the limit loads are properly defined, a common notching strategy is to reduce test levels at certain frequencies as needed to ensure those loads, when multiplied by any applicable test factor, are not exceeded in test.

Regardless of whether notching or not, make sure you design your SmallSat and its components to loads that are at least as high as those expected in test.

Design for the test loads.

This means you must design the vibration tests before you release the structural designs for the SmallSat and its components.

Key Points from This Section

- Design direct load paths.
- Make the satellite a “BrickSat” when you can.
- Separate modal frequencies.
- When designing joints, make sure you understand how the structure carries loads.
- In potentially critical joints, use high-strength bolts that are bigger than you think you need, and preload them highly.
- Match the strength of the nut or the threaded insert to that of the bolt.
- Design to avoid potential interference with runout threads, which can prevent preload.
- Develop a preliminary test plan before releasing the design.
- Design structures to withstand the test loads, and specify appropriate test environments for SmallSat components.

8. Case Study: FalconSat-2

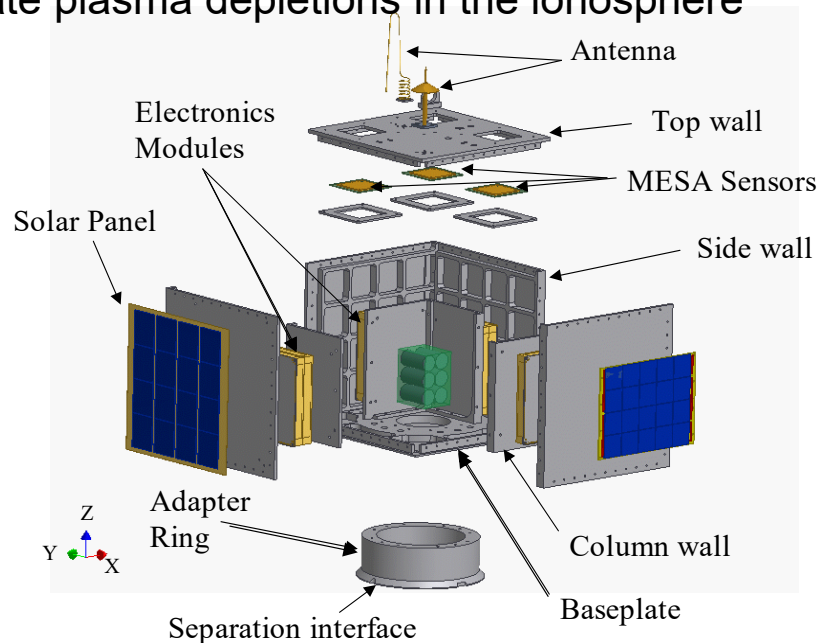
- Introduction to FalconSat-2
- Key Structural Requirements
- Approach to Structural Design and Verification
- Simplifying the Design Loads
- Testing the Engineering Model (EM)
- Deriving Design Loads from EM Test Results
- Designing the Flight Structure
- Qualification and Acceptance Testing
- Launch ... and FalconSat-2 today
- Process Changes for FalconSat-3
- Conclusions

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Case Study: FalconSat-2 (FS-2) at the United States Air Force Academy

- What: A microsat designed by USAFA staff, cadets, and consultants—third in a series, following FalconGold (1996 – 1997) and FalconSat-1 (1998 – 2000)
 - Named after the academy's nickname (and mascot): Falcon
- Why: To learn space by doing space
- Mission: Investigate plasma depletions in the ionosphere

- Cubic structure, 12.5" (0.318 m) per side
- About 43 lbs final flight weight (19.5 kg mass) for all of FS-2, including equipment and adapter ring



- Originally planned to launch in early 2003 aboard the Space Shuttle as part of NASA's Hitchhiker program, launched within a canister
- Was the sole payload on the first launch of SpaceX's Falcon-1 in March 2006 (launch failed)

FalconSAT-2 Program Philosophy

Test early and often

- Experience building and testing meets the program objectives (build good air force officers) better than detailed analysis.
- Testing is usually more dependable than analysis at rooting out problems.
- A building-blocks approach to testing and verification is cost-effective.
 - Use what is learned from testing to improve the design.

Keep everything as simple as possible

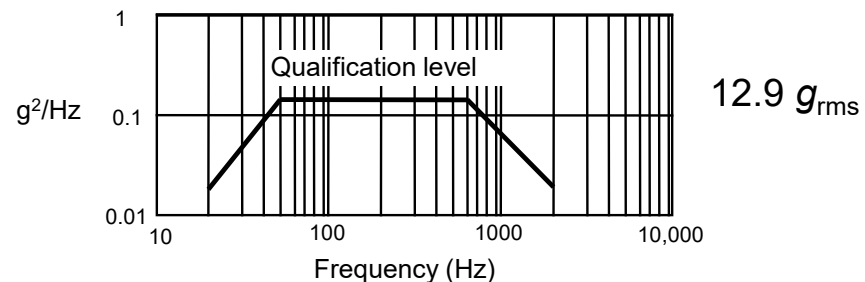
- Weight is not critical—take advantage of this:
 - No composites; parts machined from 6061-T651 plate
 - Direct load paths that can be assessed with simple analysis
 - High structural margins of safety

**The driving requirement was the need to be
safe as a Shuttle payload.**

The driving constraint was cost.

Key Structural Requirements (Hitchhiker Payloads)

- Quasi-static limit accelerations
 - +/- 11 g in X, Y, and Z
 - +/- 85 rad/s² about X, Y, and Z
 - All loads acting simultaneously in any +/- combination
- Factors of safety for structural analysis: 1.4 for ultimate, 1.1 for yield
 - Testing required to 1.25 times the quasi-static limit loads
 - We decided to build an engineering model, based on a conceptual design, and a non-flight qual model, and test both to 1.4 times limit; then test the flight model to 1.0 times limit.
- Random vibration
 - Qual levels, 3 dB above max predicted environment, for 2 min/axis (we did 3 min/axis per military standards)
- Fundamental frequency must be above 35 Hz.
 - By keeping it well above 50 Hz, we didn't have to submit a finite element model for coupled loads analysis.



Approach to Structural Design and Verification

1. Develop a conceptual configuration for FS-2.
2. Design and build a structure for an engineering model (EM).
3. Simplify the specified quasi-static loads to enable single-axis testing on a shaker.
4. Test the EM for low-level sine sweep, random vibration, and sine burst (development testing).
5. Interpret test results and learn from them.
6. Develop a simple finite-element model (not enough time to do this earlier), and correlate it with EM test results for fundamental mode in each axis.
7. Design the flight structure.
 - Selected, limited stress analysis only, given the plan for frequent testing
 - Make everything fail-safe to avoid NDE and crack-growth analysis.
8. Build and test a qualification model (QM).
9. Interpret test results and learn from them; if necessary, modify the design.
10. Build and test the flight model (FM).

Simplifying the Design Loads

- Quasi-static accelerations, as specified:
 - +/- 11g in each of X, Y, and Z axes
 - +/- 85 rad/s² about each of X, Y, and Z axes
 - All loads acting simultaneously in any +/- combination
- 64 load cases!
- We wanted to be able to test the structure with sine bursts on a shaker, one axis at a time
 - No special fixtures required beyond the one needed to test for random vibration in X, Y, and Z
- Goal: Find three uni-axial load cases—plus/minus, as would occur in sine burst testing—that would envelop the effects of the 64 specified load cases.

Using the process described in Sec. 4, I derived an equivalent single-axis load of 23.2 g. I rounded up to 25 g. Thus, the test load for sine-burst testing was $1.4 \times 25 = 35 \text{ g}$.

Used for test, but not for design. Random vibe was more severe. See upcoming pages.



23.2 g passes the sanity check: It's about the same as we get from JPL's Mass Acceleration Curve (Ref. 25) for a 19.5 kg payload for launch on STS/IUS.

Development Program: Engineering Model

- These simplified load cases were used for sine burst testing, but not design
 - Random vibration without notching turned out to be more severe
- An engineering model (EM) was built for development testing without the aid of analysis
 - Little internal capability for stress analysis at USAFA
 - Not enough time for me to contribute much to EM design
 - EM structure designed like a brick
 - Strong enough by eye
 - Way too heavy to fly! (Well, that's not true. But we wanted to teach the cadets there was more to structural engineering than this!)
- EM structural testing, April 2001 (each axis):
 - Low-level sine sweep to determine natural frequencies
 - Random vibration to specified PSD (qual level) without notching
 - $25 \times 1.4 = 35 \text{ g}$ sine burst

Objectives:

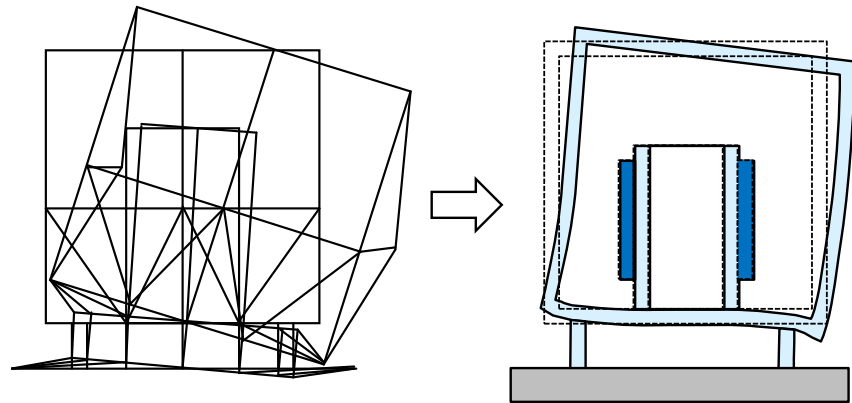
- Give Class of 2001 some test experience
- Learn how to improve the design

Results of EM Sine Sweep (Lateral Axes, X and Y)

Fundamental frequency = 182 Hz lateral

Identified as a rocking mode by examining responses measured at different locations:

- Top of box
- Center of side panel
- Corner of base plate
- Top of internal column

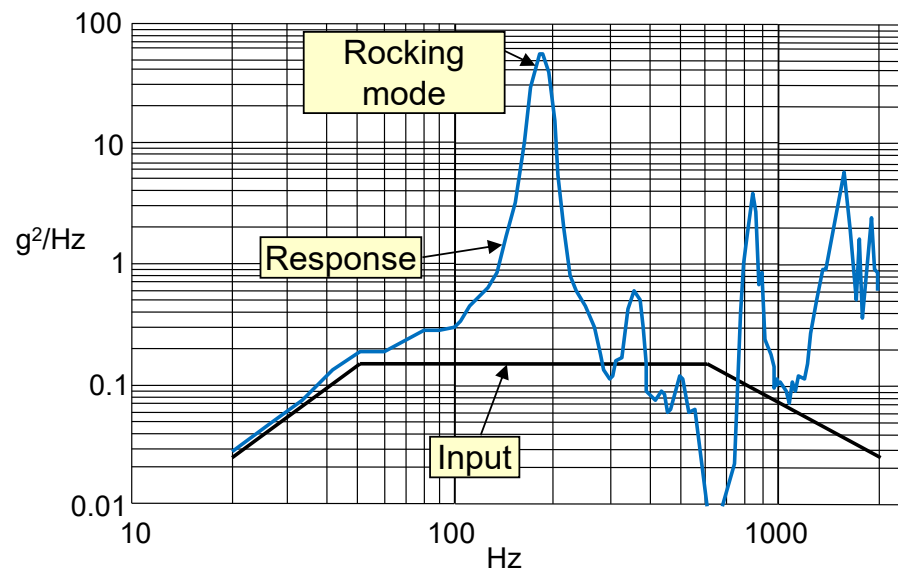


From subsequent correlated
finite-element model

EM Test Results: Lateral Acceleration at the Top of the EM

Results of qualification-level random vibration test in lateral direction:

$54 g_{\text{rms}}$, measured at top of the EM



From this data, I was able to estimate the 3σ moment at the mounting base.

How would you have done this?

Estimating Base Moment from Measured Response Acceleration

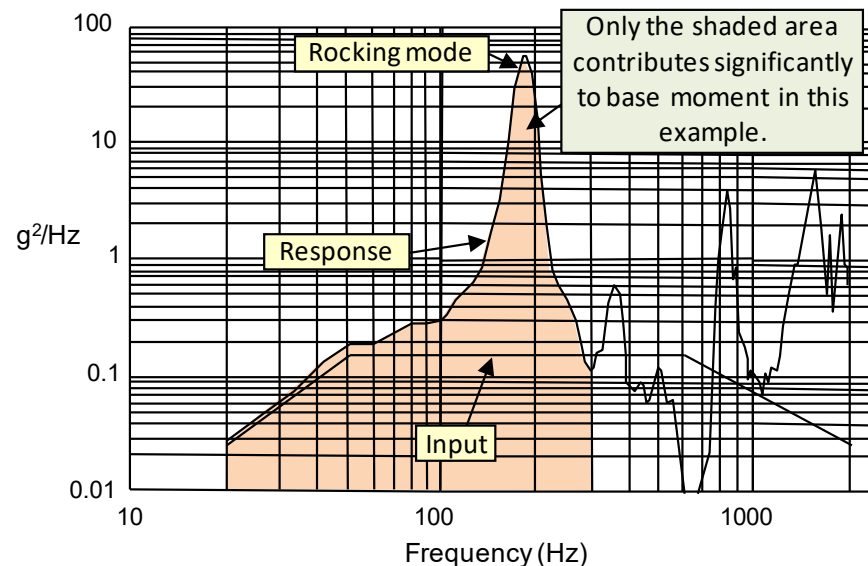
First step: Find the area under and to the left of the big hump in the response PSD associated with the 182-Hz fundamental mode (20 – 300 Hz).

- The square root of this area is the RMS acceleration at the top of the EM associated with the rocking mode, combined with the nearly quasi-static acceleration at low frequency.
- Assume the higher-order modes do not contribute significantly to the base moment (often the case).

My calculation:

41 g_{rms} for the shaded area at right

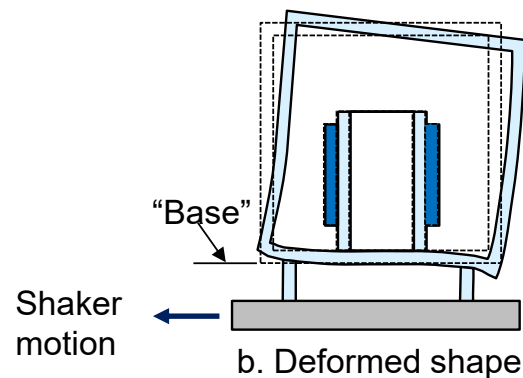
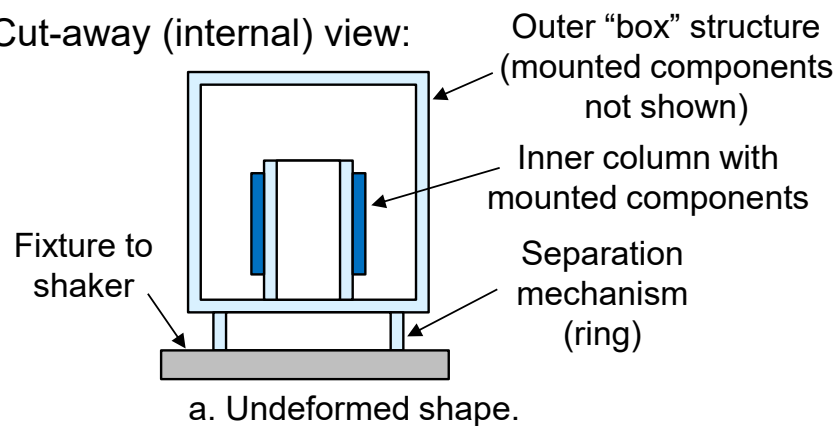
(vs. 54 g_{rms} total)



Estimating Base Moment (continued)

Second step: Understand the mode shape

Cut-away (internal) view:



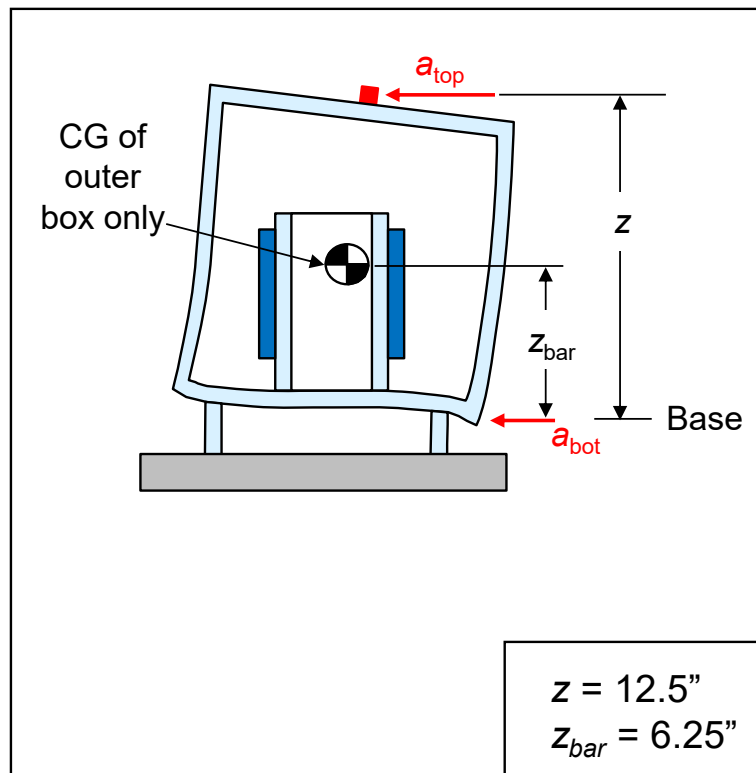
- Test data showed only the outer box mass contributed much to this mode
 - Base plate, four side walls, and top panel
 - Not the column or ring
- The box structure rotates nearly as a rigid body
 - CG translates (moves laterally)
- Mass properties for the box (rotating portion):

Weight, $w = 25 \text{ lb}$

Mass moment of inertia at CG,
 $J = 2.50 \text{ lb} \cdot \text{s}^2 \cdot \text{in} \quad (0.282 \text{ kg} \cdot \text{m}^2)$

Estimating Base Moment (continued)

Third step: Derive rigid-body translational and rotational accelerations



Design criterion: “peak” (limit) load = 3 times the RMS value

– Because failure modes are ductile

By taking the square root of the area under the input PSD between 20 – 300 Hz, we find the base acceleration is about $6 g_{rms}$.

The peak lateral acceleration at the CG is then

$$a_{cg} \approx 3 \left[6 + (41 - 6) \frac{z_{bar}}{z} \right] = 70.5 g \quad (\text{from Eq. 2-1})$$

And the peak rigid-body angular acceleration is

$$R \approx 3(41 - 6)(386) \left(\frac{1}{z} \right) = 3240 \text{ rad/s}^2 \quad (\text{from Eq. 2-2})$$

$$g = 386 \text{ in/s}^2$$

(continued)

Estimating Base Moment (continued)

Fourth step: Estimate the 3σ base moment.

$$\begin{aligned} m_{qrv} &\approx a_{cg} m z_{bar} + R J && \text{(Eq. 2-3)} \\ &= 70.5(25 \text{ lb})(6.25 \text{ in}) + 3240 \text{ rad/s}^2(2.50 \text{ lb} \cdot \text{s}^2 \cdot \text{in}) \\ &= 11,000 + 8100 \\ &= 19,100 \text{ in} \cdot \text{lb} \end{aligned}$$

Recall: We're actually using the unit-less load factor, n , in place of a_{cg} .

The equivalent quasi-static load was about 84 g.

Note: Eq. 2-3, above, applies for rigid-body rocking. It's used as an approximation for the rocking mode of a SmallSat.

Options for Designing the Flight Structure

The high response of the rocking mode was an outcome of an imperfect test, not what would happen during launch.

- Limited energy during launch, but not during test
- Notching at the fundamental frequency would have avoided such over-testing

Options:

1. Try to justify a notching strategy to keep random-vibe response from loading the primary structure more than the 25 g limit quasi-static acceleration (35 g qualification level)
2. Or design the flight structure to withstand the high anticipated response without notching (84 g quasi-static load, qualification level)

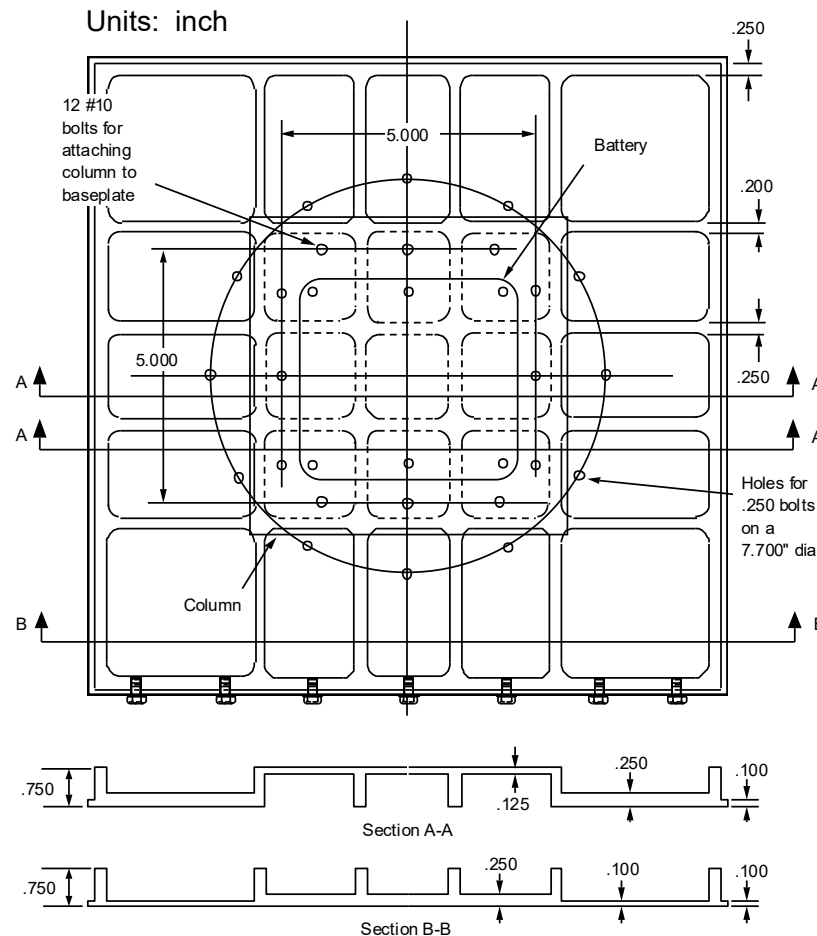
Option 2 won out: Weight was not critical, and there was not enough budget to research and justify a notching strategy. After the design was complete, NASA provided a manual notch in order to reduce risk.

EM Testing Identified Several Objectives for Structural Design

- Reduce weight
 - To reduce structural loads for given acceleration
 - To introduce at least some level of structural engineering to the cadets!
- Reduce the fundamental frequency
 - To reduce acceleration in response to random vibration
 - Do this by making the base plate more flexible in bending
- Increase damping
 - To reduce acceleration in response to random vibration
 - Not much we could do, here, given the constraints of time, budget, and capabilities
 - Fewer bolts should help some

Design of the Flight Structure: The Base Plate, the Critical Part

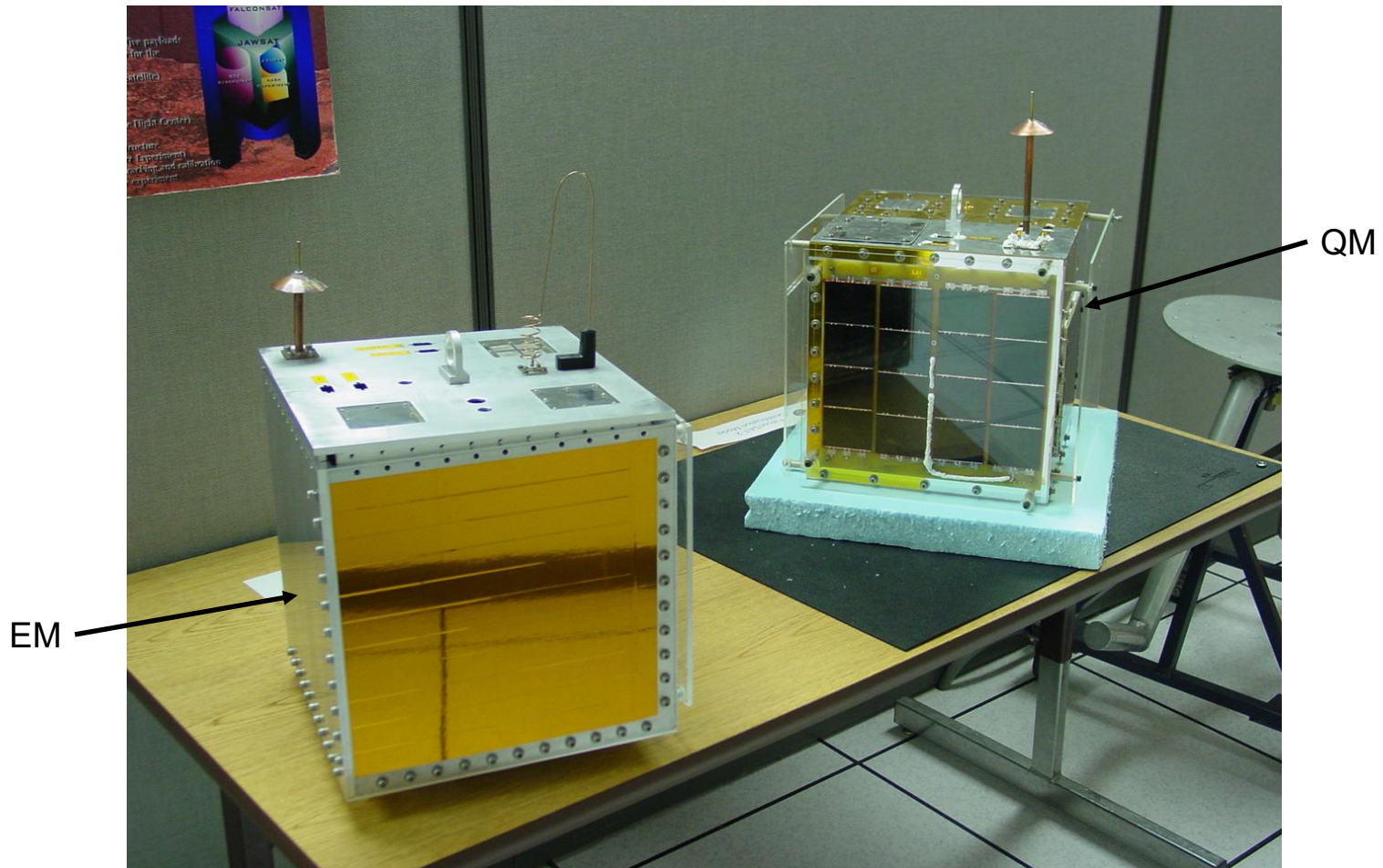
- Orthogrid pattern machined into 0.75" plate, 6061-T651 aluminum alloy
- Fasteners and walls sized by simple methods
- Structural weight for all of FS-2 reduced about 9 lb from EM (but weight growth for nonstructural components nearly made up for that!)



- Fundamental frequency predicted to drop from 182 Hz to about 120 Hz
- Response accelerations predicted to drop by about 37%

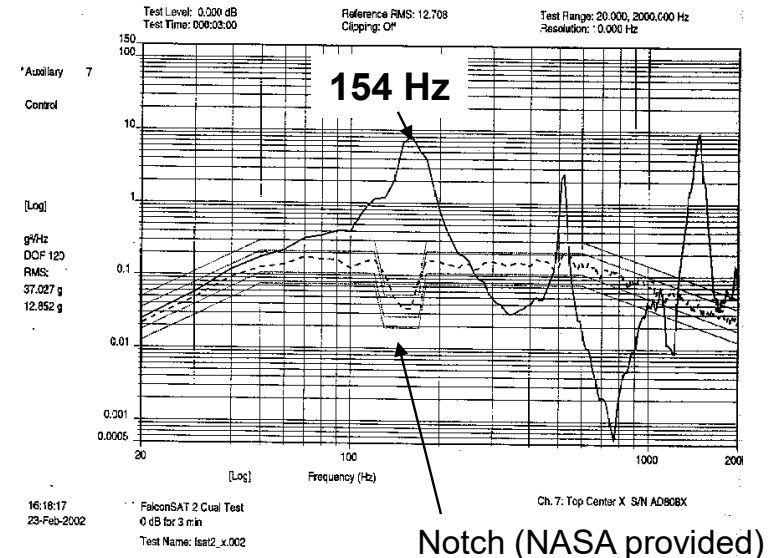
Designed to have load paths and potential failure modes that are simple and predictable

Engineering and Qualification Models



Qualification Test Program

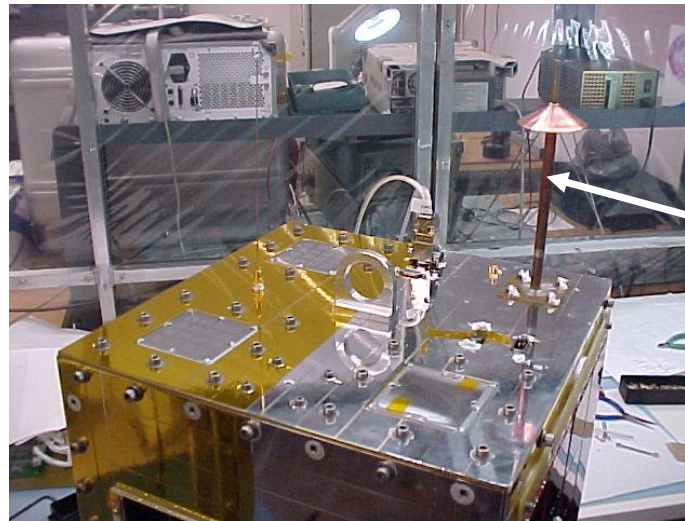
- Qualification testing of dedicated unit (Qual Model, QM)
 - To qualify the design and verify fatigue life
 - Done to a qualification margin above maximum expected levels
 - Random vibration done 3 dB above acceptance for 3 minutes per axis
 - Same as for EM test except the input PSD was notched for lateral testing to prevent excessive loading of the primary structure
- Fundamental rocking frequency = 154 Hz
- Lateral response acceleration at top of box associated only with first-mode rocking = $20 g_{rms}$ at qual levels (notched)
- Recall: This acceleration was $41 g_{rms}$ for EM test (no notch).
- Using the same process as for the EM, we derived new design loads from QM data.



Qualification Test Program (continued)

Testing was successful except in two areas:

- Small S-band antenna (suffered material fatigue failure)
 - After the test, we redesigned the antenna, but without the aid of structural analysis. (Becomes an important point)
- The separation switches bottomed out and prevented full seating at the separation interface (Marmon clamp).
 - Required modification to the center of the base plate.

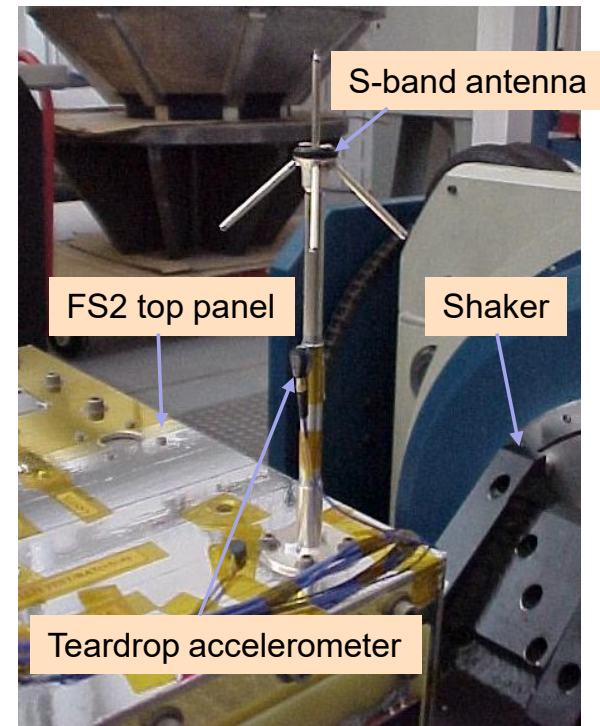


S-band antenna

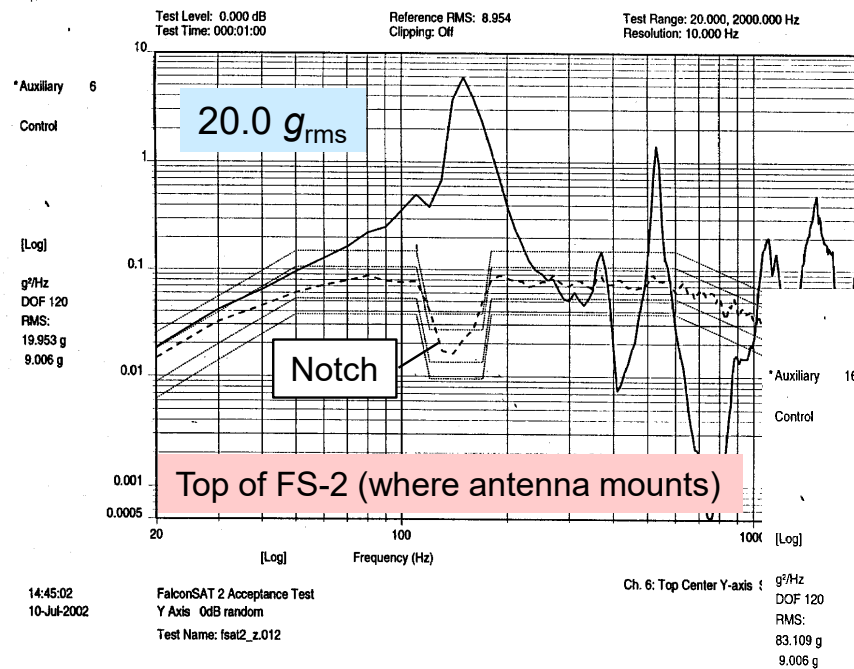
Acceptance Testing of Flight Model (FM)

- To confirm process control and workmanship for the FM
- Tested to acceptance levels, 3 dB below qual
- Completed in July 2002
- Successful other than S-band antenna—again!
 - Fatigue crack
 - Antenna had been redesigned without supporting structural analysis.

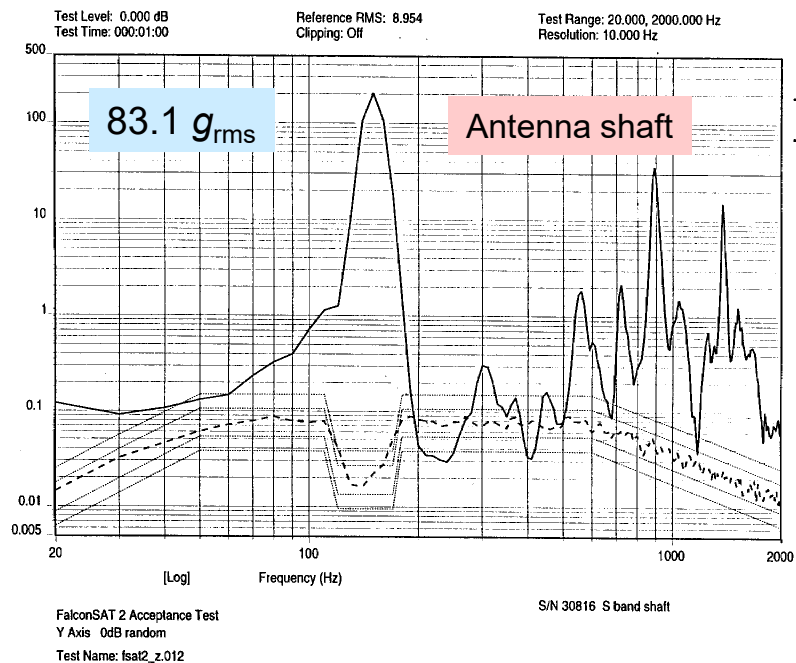
The antenna's fundamental bending mode had coupled with the spacecraft's rocking mode, resulting in extremely high acceleration.



Responses Measured During the FM Test

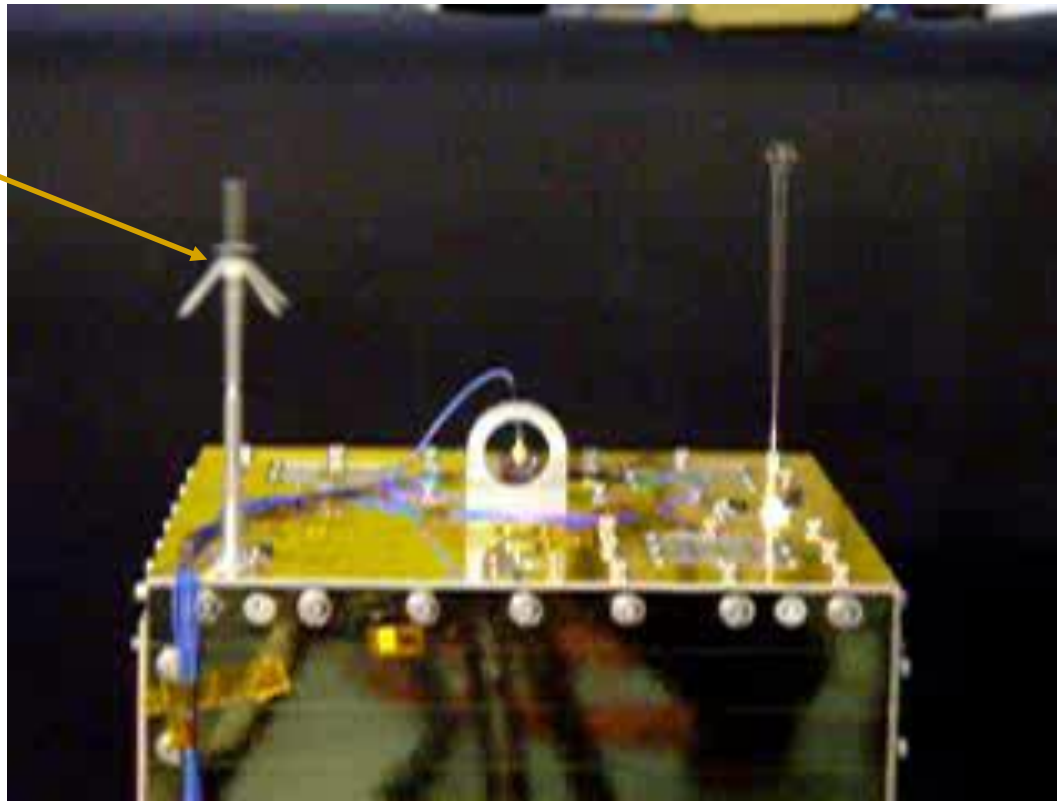


The antenna's first bending mode coupled with FS-2's rocking mode!



FM Acceptance Vibration Test

S-band antenna



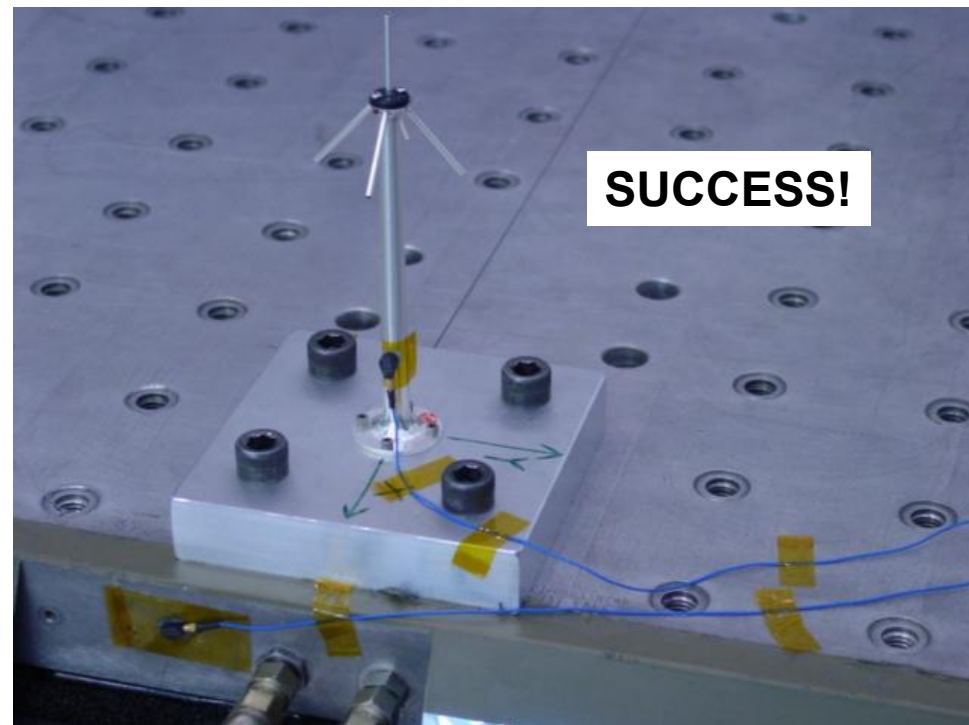
S-band Antenna Resolution

Redesigned the antenna after the FM test, this time with supporting analysis.

- One piece shaft made of brass, turned on a lathe
- Raised natural frequency to about 280 Hz to avoid modal coupling

Tested the antenna separately for qualification and acceptance.

- Using an environment derived from combination of analysis and measured response at the top of FS-2 during the FM test



Final design being tested separately for vibration

Launch of FalconSat-2

- FS-2 was launched on March 24, 2006.
 - First launch of the SpaceX Falcon-1
 - Launched from Omelek Island, part of the Kwajalein Atoll, home of the U.S. Army's Reagan Test Site in the Central Pacific
- Shortly after liftoff, the launch vehicle failed.
 - Fuel leak in main engine from corroded propellant line
- The LV fell to Earth, landing in shallow water just off shore of Omelek.
 - Causing an explosion
- FS-2 was discovered inside a storage shed on Omelek.
 - Having blown free from the LV at impact
 - Crashed through the roof of the storage shed
 - Landed near its shipping container, exhibiting the homing instincts of a true Falcon!

Landing of FalconSAT-2



FalconSat-2 now resides in the academy's astronautics museum.

Process Changes for FalconSat-3

- Keep the same general structural concept (machined orthogrid side panels), with expanded dimensions (18" cube).
- Use a free-of-charge vibration-isolation mounting system to reduce the effects of random vibration.
 - CSA Engineering's "Shock Ring" (AFRL wanted someone to demonstrate use of the Shock Ring, so AFRL provided it at no charge.)
- Develop a finite element model before EM testing, and predict response to random vibration before EM testing.
 - Use sensitivity analysis to identify desired stiffness characteristics for the Shock Ring and for key components (e.g., gravity-gradient boom).
 - Identify potential problems with random vibration, and work with the launch-integration contractor to obtain more realistic environments.
- Continue to use a building-blocks approach to verification.
 - EM test
 - QM test
 - FM test

Conclusions from the FalconSat-2 Case Study

- Testing can play an important, cost-effective role in the development of space-flight structures.
 - Building-blocks approach
 - Does not have to be expensive
 - Finds problems that analysis won't always find
- Low-cost development testing can be particularly valuable.
 - Helps avoid potential problems
- Keys:
 - Taking advantage of weight margin and making the structure robust
 - Using analysis to steer the design and development testing to steer the analysis
 - Readily available facilities for manufacturing and testing
 - Keeping things as simple as possible

Appendix

Test Documentation, Reviews, and Controls

- Managing the Process
- Test Plans and Procedures
- Getting Buy-in Before the Test
- Reviews and Controls
- Flash Report
- Test Report

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Managing the Test Process with Documents, Reviews, and Controls

When we decide to do a test, we want to do it right the first time.

- Objectives must be clear and specific.
- The test must be well designed to ensure the objectives are met.
 - With defined criteria for determining whether a test design is adequate, such as for sine burst testing and force limiting.
- All stakeholders must buy in.
 - Approved exceptions to the above criteria documented.
- All anomalies (surprises, failures, or initially unexplainable results) must be resolved to the satisfaction of all stakeholders.
- The test must be well documented.
 - So we know what we did after our memories fail us
 - So we can re-evaluate the test if key information changes afterwards

In other words, we need quality assurance—and we, the test team, need to provide it!

To ensure quality for the test, we manage the process through a set of documents, reviews, and controls.

Test Documents

- Verification plan
 - Defines the analyses, tests, and top-level criteria that are intended to establish appropriate confidence that a system or subsystem will satisfy its requirements
- Test plan (a.k.a. test specification)
 - Defines scope, objectives, and engineering requirements for a specific test
 - Analogous to an engineering drawing, which defines engineering requirements to the manufacturing group
- Test procedure
 - Step-by-step implementation of the test plan, analogous to a manufacturing procedure
 - Usually written by test-lab personnel, but certain procedural steps that are required to ensure the test meets its objectives should be included in the test plan
- Test report
 - Much of the same general contents as is in the test plan, but as-performed test conditions, results, and conclusions
 - Includes photos, hard data, and reduced data in appendices

Suggested Contents of a Test Plan

- Overview or executive summary
 - One page or less (for those too busy to read the whole plan)
- Scope and objectives
 - What will be tested, and for what purpose? What requirements will be verified?
- Test article definition
 - Description, drawing number, serial number (as applicable)
 - Any differences between the test article and the flight article
- Test facility and equipment
 - Where the test will be performed
 - Needed fixtures, fasteners, tools, camera, instruments, etc., and who will supply each
- Test configuration
 - Figure(s) showing physical configuration of test article, fixture, shaker
- Instrumentation
 - Types of instruments and figures showing locations on test article; label or ID for each data channel

Continued

Suggested Contents of a Test Plan, Continued

- Test types and environments
 - Sine sweep, sine burst, random
 - Details such as sweep rate, data resolution, notching/force limiting
 - Control tolerances
 - Test sequence
- Pass/fail or success criteria
 - How will you judge whether the test article passed the test or whether the test met its objectives?
 - What criteria would trigger a decision to stop the test and investigate an anomaly?
- Special considerations
 - Lifting and handling, safety, cleanliness
- Responsibilities
 - By name, preferably with alternates
- Schedule
- Appendices
 - Derivation of sine-burst levels and notches, force limits, or response limits
 - Documentation of any applicable math models
 - Procedures for key parts of the test (e.g., adjustment of notches or force limits during test)

Iron Out the Test Plan Before Doing the Test!

If you are the customer,

- Don't wait until after the test to decide if it's acceptable!**
- Insist on reviewing the test plan—in detail—before the test is run.**

If you are the one doing the test,

- Insist that your customer approve the plan.**
- For a complex test, plan to conduct a test design review.**

Getting Buy-in Before the Test

If you're the one responsible for the test, you'll have to get buy-in from more than your contractual customer; you have many other "customers" (stakeholders):

- Your supervisor or manager, and other team members
- The launch vehicle contractor/supplier
- The contractor tasked with integrating payloads with the launch vehicle

When writing the test plan, documenting pretest analysis, or generating charts for a test design review, put yourself in the shoes of the above stakeholders: What would you want to know?

- Will the test adequately encompass the effects of launch environments?
- Will the test fixture significantly and adversely affect the test article's modes of vibration, hence the test objectives?
- Will any potential failure, fatigue cracks, detrimental permanent deformation, or loosened fasteners be detected?
- Will anomalous test data be identified and investigated before the test configuration is torn down?
- Will adequate steps be taken to minimize risk associated with over testing?

Asking yourself questions such as these will lead not only to better documentation and more-satisfied customers, but also to a better test!

Reviews and Controls

- Test Design Review, TDR (test plan review)
 - Like any design review: Bring stakeholders up to speed on objectives, approach, configuration, instrumentation, pass/fail criteria, etc.
 - Route draft copies of the plan to stakeholders and solicit comments, and then hold a meeting to walk through the test plan.
 - Objective: Reach concurrence on the test plan. Meeting held far enough before the test to enable adjustments to the test plan.
- Test Readiness Review (TRR)
 - Objective: Reach concurrence that everyone and everything are ready for moving the test article into the test facility and conducting the test. Held just before test start.
 - It's too late now to be arguing about whether the planned test is adequate!
- Post-test, pre-teardown review
 - Ensure all anomalies are identified and resolved before breaking configuration.
 - Many anomalies are the result of something wrong with the configuration (loose bolt, physical interference, fixture issues) or instrumentation (inadequately bonded accelerometer, calibration error).
- Controls: checkpoints, inspections, discrepancy reporting system

Immediately After the Test, Issue a Flash Report

Example from the vibration test of a small-satellite engineering model (EM) (issued one day after completion):

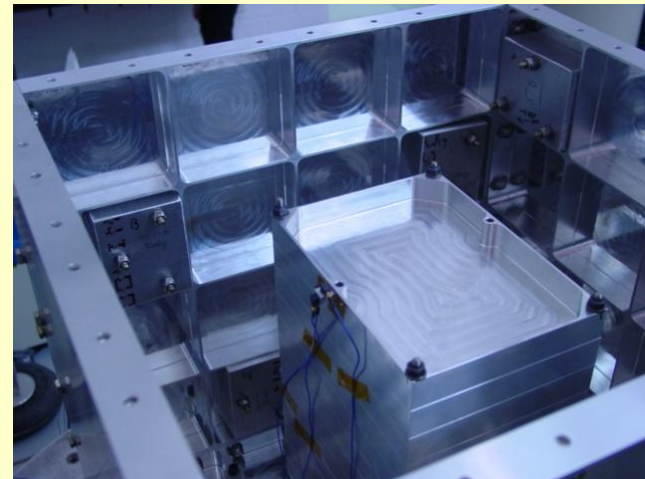
The subject test was performed April 14 and 15, 2003, at Kirtland Air Force Base. The test met all its objectives (see below) and the EM suffered no damage. The random vibration environment used in the test was considerably more severe than we expect to use for qualification testing next year. The maximum predicted environment for launch is presently undefined (and may change after being defined), so we intentionally used a conservative, high environment with the intention of establishing design margin and thus demonstrate our design is robust.

Objectives:

1. Validate the preferred approach to stacking equipment modules ...

Outcome: Objective met. The stack of 12 module simulators withstood all testing with no noticeable change in sine-sweep response.

(Continued)



More from the Example Flash Report

Objective 2: Develop a notching strategy for protecting the spacecraft primary structure from over testing. ...

Outcome: Objective met. We found a process for reducing over-test that is technically justifiable and practical (can be done with little time delay to the test). The process will be documented for future use.

... (more objectives and outcomes) ...

In addition to the above, we uncovered several problems with assembly and learned some lessons, which will be captured in the final report. Some of these lessons will drive design changes to our adapter ring (inadequate access for torque wrench), and some pertain to preparation prior to test.

It's a good idea to issue a flash report such as this, hitting on the key points. Many people will be interested in hearing right away how the test went.

Suggested Contents of a Test Report

Similar in format to a test plan, but now you're documenting the as-run test:

- Executive summary up front (similar to the flash report): Were all objectives met? Did the test article pass the test?
- A review of objectives, configuration, conditions, and pass/fail or success criteria
- Was the test run as planned?
 - Configuration, loading conditions, instrumentation, sequence
 - If not, why?
- Key findings and supporting data
 - Clear conclusions regarding success criteria
- Details, photos, and the bulk of the data in appendices

Include a section on lessons learned.

A Closer Look at Test Reports

Document everything that may be needed to convince someone else that you've done a good test and the data and conclusions can be trusted.

- Photos
 - Configuration at incremental stages of setup
 - Locations of strain gages or accelerometers, and arrangement of displacement gages for each load case
 - Can never have too many photos! Photograph anything that may help answer questions later.
- Calibration records for instruments and checks made for data channels
- Test log and as-run procedure
- Test data and associated checks
- Test anomalies and associated resolution
- Post-test analysis (data reduction)
- Data summary, conclusions, and lessons learned

Any test worth doing is worth documenting well.

Summary

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Key Points from This Course

- For a given test, start with a clear definition of test objectives.
- Define pass/fail or success criteria based on the test objectives.
- Make sure all stakeholders agree on the objectives and criteria before designing the test.
- Make sure all stakeholders agree on the test plan before performing the test.
- Design instrumentation as needed to meet the objectives and assess the pass/fail or success criteria.
- Make sure you have a method other than low-level sine sweep for detecting detrimental permanent deformation if the structure must maintain position or alignment between component interfaces.
- Strengthen your understanding of ...
 - the mechanics of vibration.
 - the limitations of testing on a shaker.

continued

Key Points from This Course

- Establish appropriate boundary conditions, and design fixtures and mass simulators that allow the test to meet its objectives.
- Try to find budget and time for a test of the spacecraft structure prior to vehicle integration, with mass simulators representing components.
 - And a test of an engineering model before committing to the structural design
 - Doesn't have to be expensive; may actually reduce cost by decreasing the engineering labor associated with analysis
- For a strength test, such as sine burst, show by analysis, with an apples-to-apples comparison, that the test loads are at least as severe as the specified limit loads multiplied by the test factor.

continued

Key Points from This Course, continued

- Take advantage of technically justifiable force limiting, even when you don't have the opportunity to use force gages.
- Don't notch—regardless of whether with force limiting, response limiting, or manual notching—without defensible technical rationale and stakeholder buy-in.
- When notching to ensure the flight limit load is not exceeded, ...
 - make sure the limit load is the maximum expected combined load from all sources.
 - don't notch excessively at higher frequencies to compensate for excessive levels at low frequencies.

continued

Key Points from This Course, continued

- Dry run the test before mounting the spacecraft to the shaker unless the shaker and control software have been used recently for similar tests.
- Before disturbing the test configuration, identify and investigate any anomalies and make sure you understand the reason for any significant differences between pre- and post-test sine-sweep results.
 - Come to the test prepared to identify anomalies, e.g., with predicted response.

continued

Key Points from This Course, continued

- Design the SmallSat to pass the test:
 - Design a robust structure with direct load paths.
 - Separate modal frequencies.
 - Pay careful attention to bolted joints:
 - Use larger bolts than you think you need, when you can
 - Match nut or insert strength to that of the bolt.
 - Ensure high preload.
 - Avoid potential interference with runout threads.
 - Make key test decisions before releasing designs of structures and components.
 - Design structures for the test loads.
 - Test SmallSat components to environments higher than GEVS (14.1 g_{rms}) before testing the integrated SmallSat.

continued

Key Points from This Course, continued

**Finally, take responsibility for verification.
Don't expect your customer or launch service provider to
design your vibration test program.
Design it yourself, and then sell it to other stakeholders.**

Solutions to Class Problems

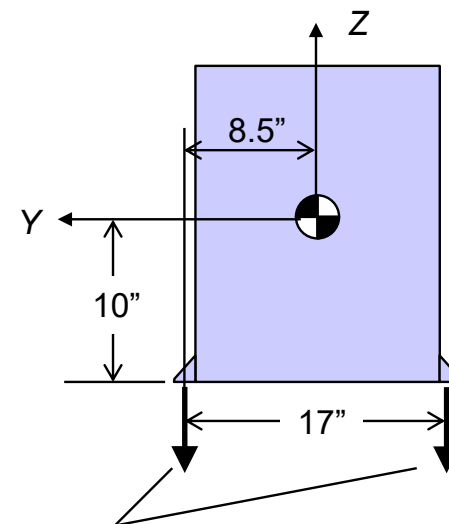
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Solution to Problem 4-1

Use the process defined in Option 4 for Example 4-3:

- a. Identify all the structure's critical modes of failure and associated indicator parameters.

Given: Bolt tensile load, P_t , is the sole indicator parameter for this problem.



Bolt tensile loads

(view in the X-Z plane has the same dimensions relative to the 17"-square bolt pattern)

continued

Solution to Problem 4-1, continued

- b. Calculate the maximum bolt tensile load for the specified design loads. This value becomes the target load for designing the test.

For one-*g* axial (Z-axis) loading, the peak bolt load is

$$P_t = \frac{150}{4} = 37.50 \text{ lb}$$

For one-*g* lateral (X or Y) loading,

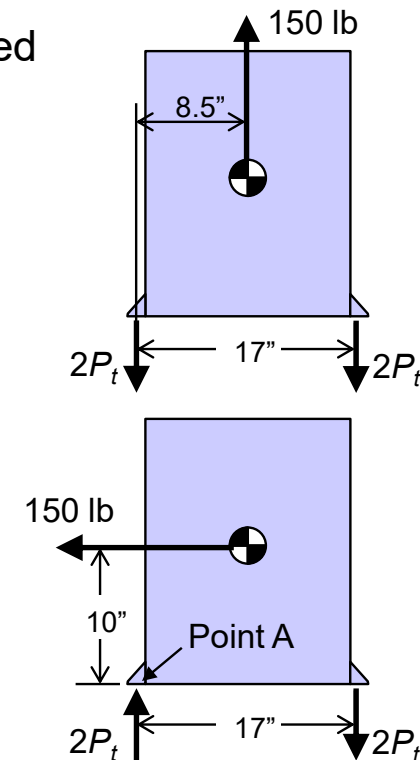
$$\sum M_A = 0 = 10(150) - 17(2P_t)$$

$$\text{So the peak bolt load is } P_t = \frac{10(150)}{2(17)} = 44.12 \text{ lb}$$

Thus, in terms of the limit load factors (n_x , n_y , and n_z) and the 1.25 test factor,

$$\text{Limit bolt load, } P_{t\text{-limit}} = 44.12n_x + 44.12n_y + 37.50n_z$$

$$\text{Target bolt load, } P_{t\text{-target}} = 1.25P_{t\text{-limit}}$$



continued

Solution to Problem 4-1, continued

From the previous page,

$$P_{t\text{-limit}} = 44.12n_x + 44.12n_y + 37.50n_z$$

$$P_{t\text{-target}} = 1.25P_{t\text{-limit}}$$

There are two specified limit load cases:

Case	Limit load factors			Bolt tensile loads, lb	
	n_x	n_y	n_z	$P_{t\text{-limit}}$	$P_{t\text{-target}}$
1	8.5	8.5	0.0	750	938
2	8.5	0.0	8.5	694	867

← Target load for test

continued

Solution to Problem 4-1, continued

- c. Identify the test axis that would most efficiently achieve the target bolt load.
- One g in Z causes 37.50 lb of bolt tension
 - Whereas one g in X or Y causes 44.12 lb of bolt tension
 - Therefore, loading in either lateral axis, X or Y , would most efficiently test the bolts in tension.

continued

Solution to Problem 4-1, conclusion

- d. Derive the acceleration required in the selected axis to achieve the target bolt load.
- From step 2, one g of lateral loading causes 44.12 lb of bolt tensile load, so the quasi-static acceleration in either the X or the Y axis that causes 938 lb of bolt tension is

$$a = \frac{938}{44.12} = 21.3 \text{ } g$$

- However, the test will be conducted with a one- g bias in the Z direction (from gravity), which causes an initial bolt load of -37.50 lb.
- Thus, to load the bolts to the required 938 lb, the acceleration used for sine-burst testing is

$$a = \frac{938 + 37.50}{44.12} = 22.1 \text{ } g$$

Solution to Problem 4-2, Deriving an Equivalent Sine Burst Level

First, let's estimate a reduced input level by use of the table below, from Sec. 4, based on the transmissibility equation, Eq. 1-1:

$f_{\text{ratio}} = f/f_n$	T_R
0.60	1.56
0.50	1.33
0.40	1.19
0.35	1.14
0.30	1.10
0.25	1.07
0.20	1.04
0.15	1.02
0.10	1.01

for $Q \geq 7.5$

Sine burst frequency, $f = 25$ Hz

Fundamental frequency, $f_n = 100$ Hz

Transmissibility, $T_R = 1.07$

Estimated equivalent reduced input acceleration to achieve the same base force and moment as 21.3 g uniform acceleration:

$$a = \frac{21.3}{1.07} = 19.9 \text{ g}$$

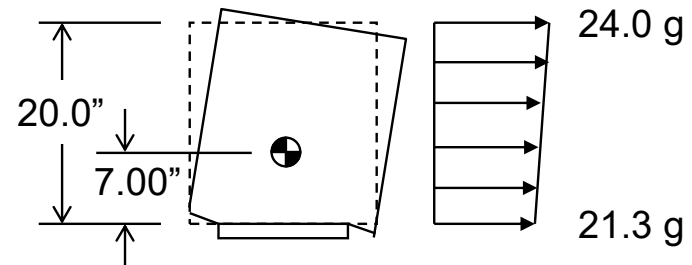
But a rocking mode such as this not the same as a mass on a translational spring, so let's derive an equivalent acceleration using the actual measured response ...

continued

Solution to Problem 4-2, continued

For the acceleration profile shown at right,

Average acceleration, as calculated at CG:



$$a_{cg} = 21.3 + \left(\frac{7.00}{20.0} \right) (24.0 - 21.3) = 22.2 \text{ g} \quad (\text{Eq. 2-1})$$

Angular acceleration:

$$R = \frac{(24.0 - 21.3) (386 \text{ in/s}^2)}{20.0 \text{ in}} = 52.1 \text{ rad / s}^2 \quad (\text{Eq. 2-2})$$

$g = 386 \text{ in/s}^2$

continued

Solution to Problem 4-2, continued

$$\begin{aligned}\text{Moment at base, } M &\approx ma_{cg}x_{cg} + RJ \quad (\text{Eq. 2-3}) \\ &= 150(22.2)(7.00) + 52.1(30.0) \\ &= 23,300 + 1,600 = 24,900 \text{ in-lb}\end{aligned}$$

Target moment at base (21.3 g quasi-static):

$$M_r = 150(21.3)(7.00) = 22,400 \text{ in-lb}$$

Thus, to achieve the target base moment, the input acceleration could be reduced to ...

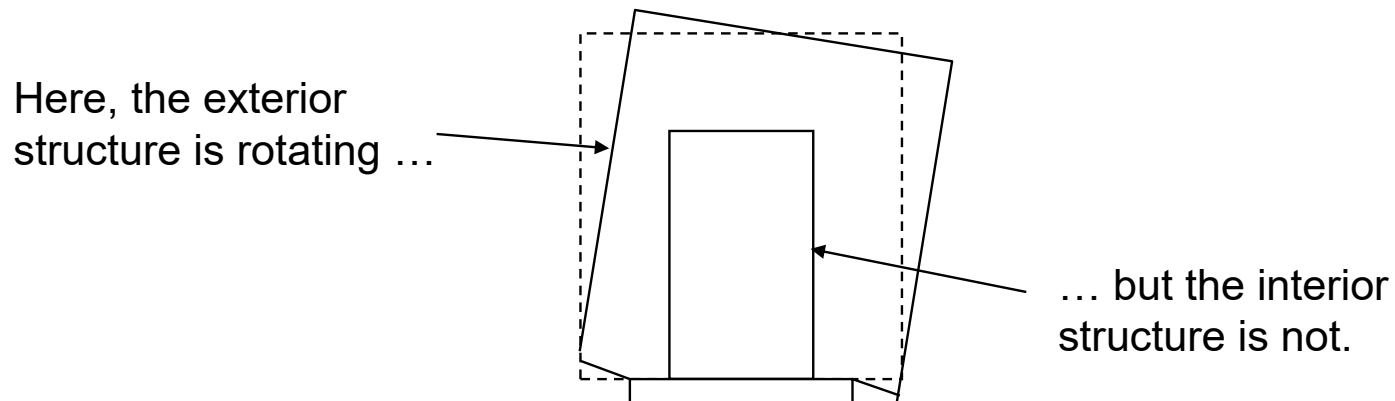
$$a = \left(\frac{22,400}{24,900} \right) (21.3) = 19.1 \text{ g}$$

Note: We can simplify these calculations when acceleration is in g units, just as we did for mass, by using 1/in units for R and lb-in² units for J , recognizing that the g unit cancels.

continued

Solution to Problem 4-2, conclusion

The problem statement said to assume all of the mass rotates as a rigid body. However, this is usually not the case.

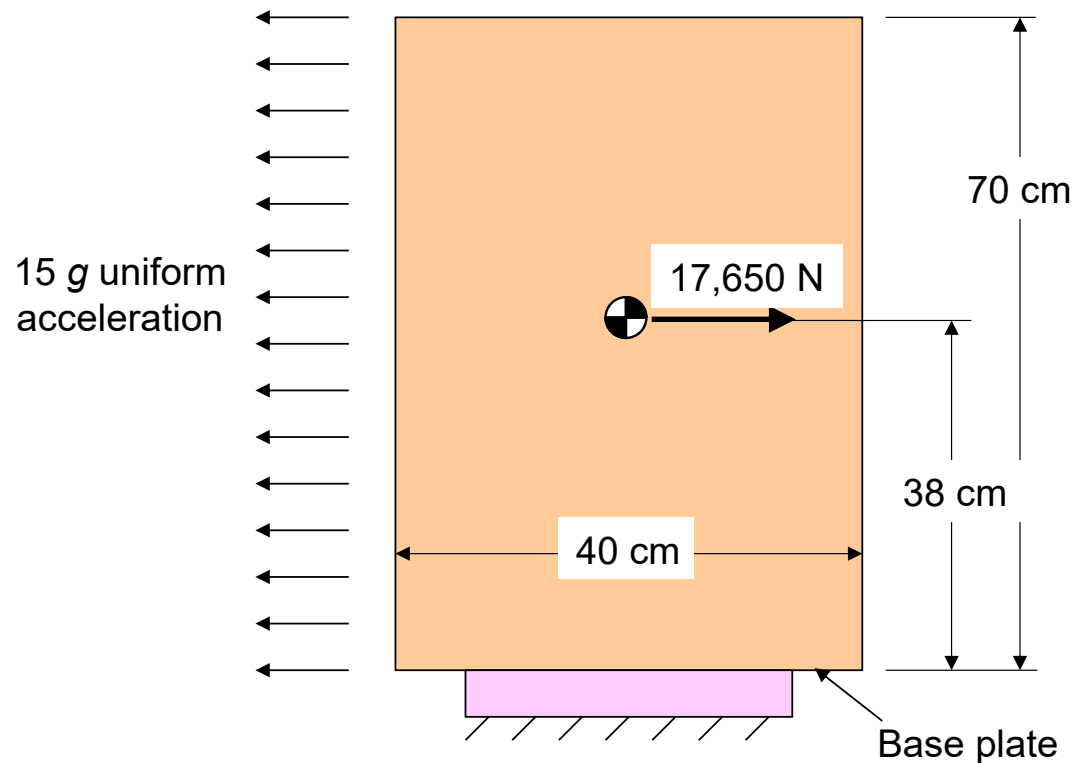


In this case, the approach used on the previous page over-calculates the base moment, and 19.1-g input for sine burst would be inadequate.

You need to understand the mode shape!

Solution to Problem 7-1

1. Calculate the equivalent inertia load at the CG:



$$m = 120 \text{ kg}$$

$$g = 9.807 \text{ m/s}^2$$

$$\text{Weight} = mg$$

$$= 1,177 \text{ N}$$

Equivalent inertia
load at cg:

$$F = 15(1177)$$

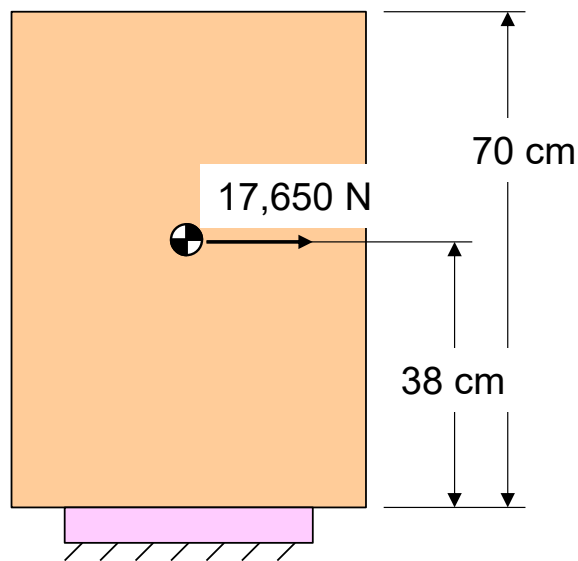
$$= 17,650 \text{ N}$$

Recall: Inertia load is
opposite in direction from
acceleration.

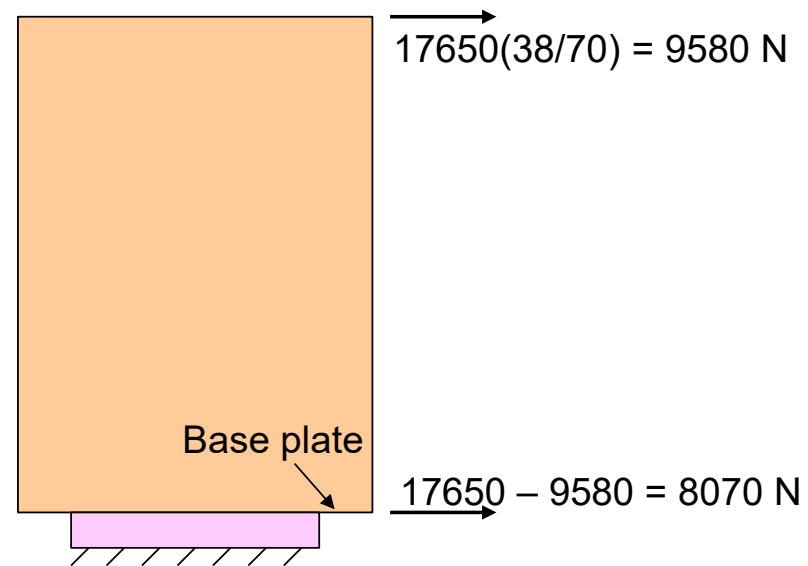
continued

Solution to Problem 7-1, continued

2. Divide the inertia load between top panel and base plate, so that the total force and moment at the base are the same as for the quasi-static load.



a. Inertia load at CG

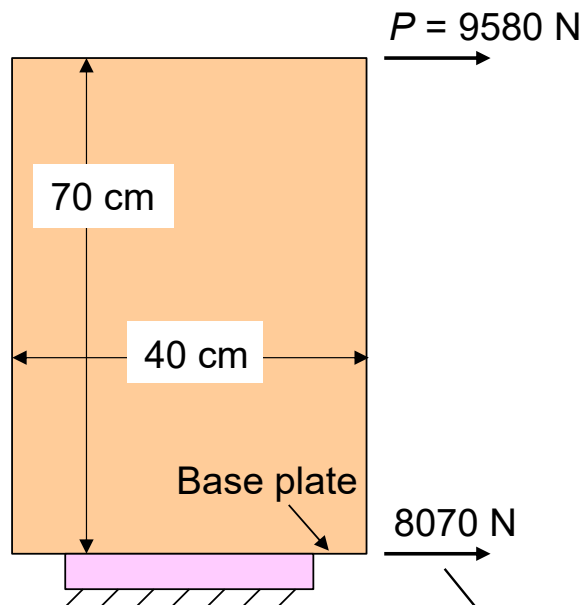


b. Equivalent loads on top and bottom panels

continued

Solution to Problem 7-1 (continued)

3. Estimate running loads on panel edges using Eqs. 7-1 and 7-2.



$$v = \frac{P}{2b} \quad (\text{Eq. 7-1})$$

$$\rho = \frac{2vh}{w} \quad (\text{Eq. 7-2})$$

$$v = \frac{9580}{2(40)} = 119.8 \text{ N/cm} = 68.4 \text{ lb/in}$$

$$\rho = \frac{2(119.8)(70)}{40} = 419 \text{ N/cm} = 239 \text{ lb/in}$$

This force does not cause panel loads

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Last revised April 2026

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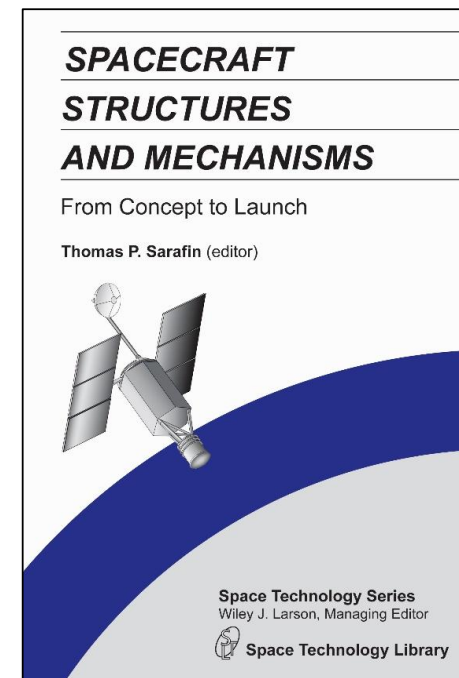
Contact: tom.sarafin@instarengineering.com

Website: aticourses.com

Contact: ati@aticourses.com

SCS—Spacecraft Structures from Concept to Launch

- Presents the structure for a spacecraft as a system
- Objectives: improve your understanding of ...
 - Structural functions, requirements, and environments
 - How structures behave and how they fail
 - How to develop structures that are cost-effective and dependable for space missions
 - Structural verification
- Target audience:
 - Systems engineers
 - Structural and mechanical design engineers
 - Stress analysts
 - Dynamics and loads engineers
 - Anyone else with an engineering degree interested in the subject
- Course length: 3 full days



Book upon which the original
version of this course was based

SCS Topics

1. Overview of spacecraft structures
2. Launch environments and how structures respond
3. Assessing structural integrity: stress analysis
4. Overview of finite element analysis
5. Configuration development and preliminary structural design
6. Improving the loads-cycle process
7. Verification and quality assurance
8. Final verification and risk assessment


Space Mission Structures: From Concept to Launch

Understanding a Shock Response Spectrum (SRS)

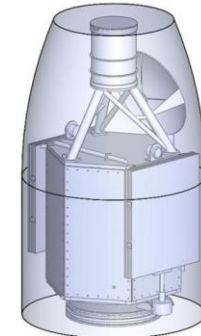
Say we measure a time history of



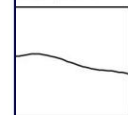
d at its base.
 ing of 5% of
 7 g.
 e compute a


Space Mission Structures: From Concept to Launch

FireSat Solid Model: Stowed Within Static Envelope



This is only one possible configuration for FireSat. On an actual program, we typically develop several optional configurations that are subject to trade studies by the design team.



quency
t for shock.

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“Excellent presentation—a reminder of how much fun engineering can be.”

“This is a great class presented with real-world applications.”

“Many really good examples.”

“Good stuff, and a very clear presentation.”

“I have a much better understanding of structure design issues, especially modeling and its limits. The instructor clearly understands and is effective at communicating this material.”

“Great course!”—former Chief Engineer for USBI

SDA—Structural Design and Analysis for Aerospace Engineers

- This course is a companion or follow-on to Instar's flagship course, SCS—Spacecraft Structures, from Concept to Launch.
 - SCS gives the big picture of spaceflight structures development, with selected details.
 - SDA goes into much more detail on design and analysis of aerospace structures and is not specific to spacecraft. Much of the course applies to aircraft as well.
 - Although we recommend people working in the space industry start with SCS and follow with SDA, SCS is not a prerequisite.
- Target audience:
 - Structural and mechanical design engineers
 - Stress analysts
 - Anyone else with an engineering degree interested in the subject
- Course length: 3 full days

SDA Topics

1. Structural requirements and design criteria
2. Review of statics and dynamics
3. Mechanics of materials
4. Strength analysis
5. Fatigue of metals
6. Structural design


Structural Design and Analysis for Aerospace Engineers

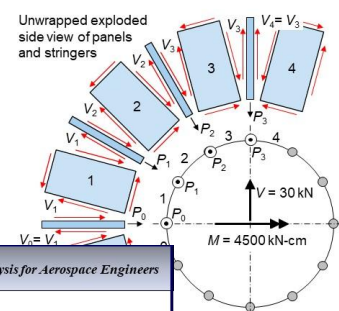
Skin-Stringer Example, continued

Step 2. Calculate the panel shear loads needed to generate the stringer loads at the base, recognizing that the stringer loads on the opposite end of the cylinder are zero.

Start at stringer 3 (top): For equilibrium, ...

$$P_3 = 2V_3 ; V_3 = \frac{7.50}{2} = 3.75 \text{ kN}$$

Now move down:



Unwrapped exploded side view of panels and stringers

continued

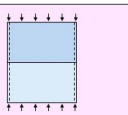

Structural Design and Analysis for Aerospace

Avoid Concentrated Loads on Shells

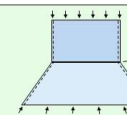
The key to using any shell or plate efficiently is to introduce loads uniformly rather than concentrated loads.

— Regardless of whether the shell is solid skin, sandwich, isogrid, or orthogrid

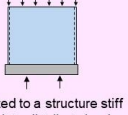
Examples:



Monocoque cylinder mated to another monocoque cylinder



Monocoque cylinder mated to a monocoque conical (frustum) shell



Or mated to a structure stiff enough to distribute loads



Load from bulkhead or ring frame

Load from frustum

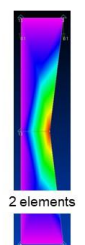
continued


Structural Design and Analysis for Aerospace Engineers

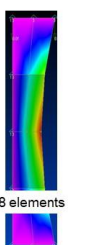
Example 4-b, FEA Results

$a = 10''$
 $b = 2''$
 $t = 0.1''$
 $E = 10^6 \text{ psi}$
 $\nu = 0.33$

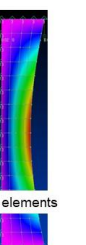
Nx Nastran buckling solution:



2 elements



8 elements



64 elements

Buckling stress, F_{cr} : 11.228 ksi 10.435 ksi 10.064 ksi

vs. 9.38 ksi from the plate buckling equation

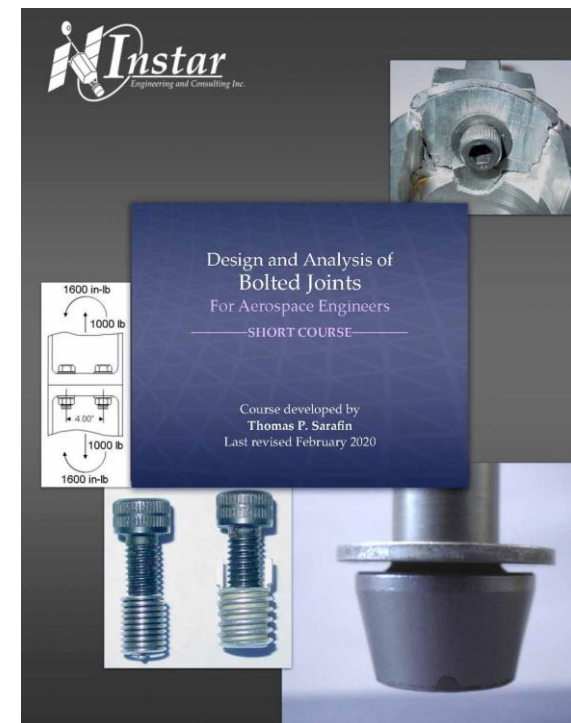
Don't rely solely on FEA!

continued

DABJ—Design and Analysis of Bolted Joints

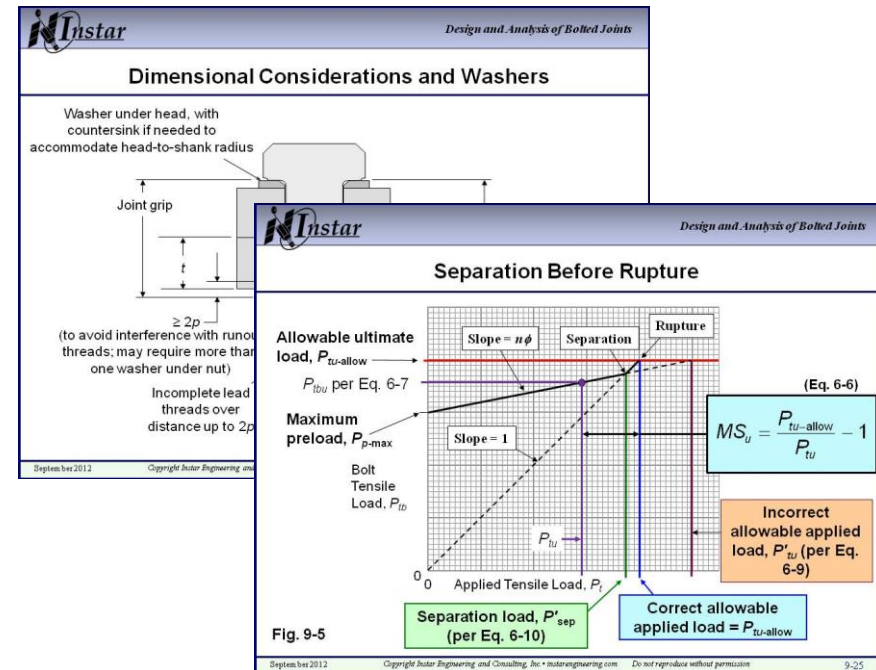
***Includes a close look at NASA-STD-5020 (including Revs. A and B),
“Requirements for Threaded Fastening Systems in Spaceflight Hardware”***

- Objectives:
 - Build an understanding of how bolted joints behave and how they fail
 - Impart effective processes, methods, and standards for design and analysis, drawing on a mix of theory, empirical data, and practical experience
 - Share guidelines, rules of thumb, case histories, and valuable references
- Target audience:
 - Mechanical design engineers
 - Structural analysts
 - Others interested in the subject
- Course length: 3 full days



DABJ Course Topics

1. Overview
2. Screw threads: evolution and important characteristics
3. Developing a concept for the joint
4. Calculating bolt loads when ignoring preload
5. Failure modes and assessment methods
6. Thread stripping and pull-out strength
7. Selecting hardware and detailing the design
8. Mechanics of a preloaded joint under applied tension
9. Fastening system analysis per NASA-STD-5020B
10. Special topics: (a) finite element modeling of bolted joints and (b) deriving design tables for bolt sizing



"Well-researched, well-designed course."

"Interaction between instructor and experienced designers (in class) was priceless."

"Best course I have taken."

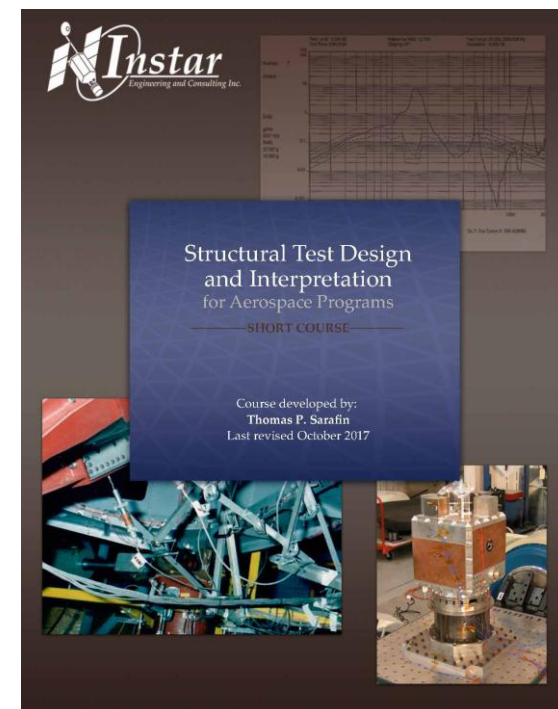
"Really good course. More people need to take this."

"Great course! Lots of lessons learned. The examples made it that much better."

"Strong emphasis on understanding physical principles vs. blindly applying textbook formulas."

STDI—Structural Test Design and Interpretation

- This 3-day course provides a rigorous look at structural testing and its roles in product development and verification for aerospace programs.
- The course starts with a broad view of structural verification throughout product development and the roles of testing, and then covers planning, designing, performing, interpreting, and documenting a test structural requirements and flight environments .
- Objectives are to improve your understanding of how to ...
 - identify and clearly state test objectives
 - design (or recognize) a test that satisfies the identified objectives while minimizing risk
 - establish pass/fail criteria
 - design the instrumentation
 - interpret test data
 - write a good test plan and a good test report
- Target audience: All engineers and managers involved in ensuring that launch vehicles and their payloads are structurally safe and ready to fly
- Course length: 3 full days



STDI Course Topics

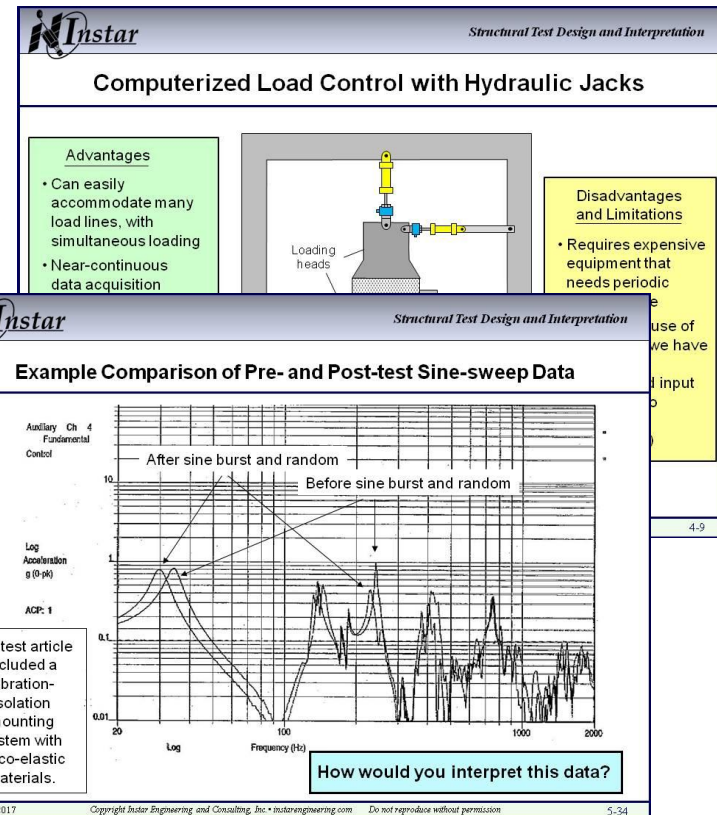
1. Overview of structural testing
2. Designing and documenting a test
3. Loads testing of small specimens
4. Static loads testing of large assemblies
5. Testing on an electrodynamic shaker
6. Notching and force limiting
7. Overview of other types of structural tests
8. Case history: vibration testing of a large spacecraft telescope

"Good job, Tom. These courses are a big help and give us a lot of great fundamental information."

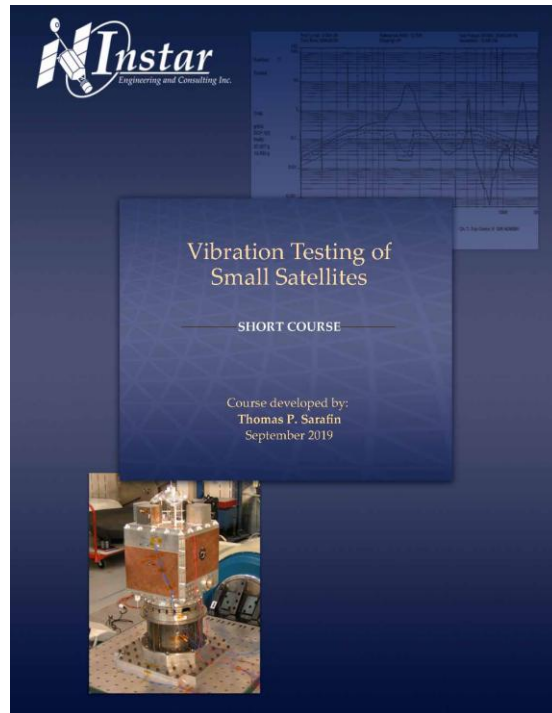
"Tom Sarafin's courses never disappoint. This class offers a well-balanced blend of fundamentals, examples, and lessons learned that any aerospace engineer involved in structural test design and interpretation would benefit from."

"This is a great course."

Can be combined with a one-day computer workshop on notching and force limiting (NFLW)



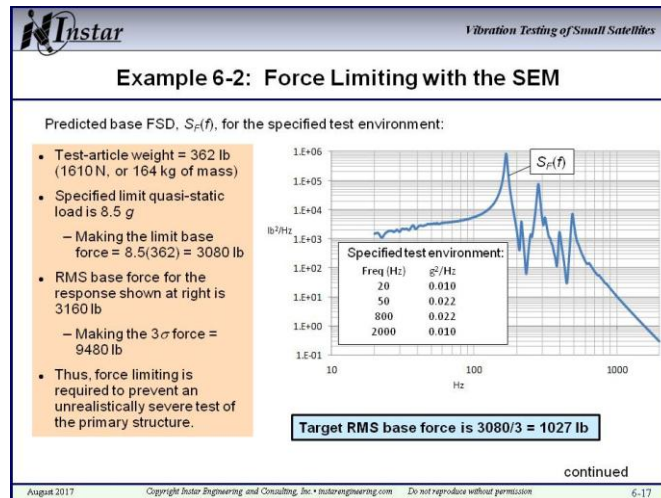
VTSS—Vibration Testing of Small Satellites and NFLW—Notching and Force Limiting Workshop



Can be combined with a one-day computer workshop on notching and force limiting (NFLW)

- This 2-day course provides a tutorial, practical guidance, examples, and recommendations for testing a small satellite on an electrodynamic shaker. Addressed are sine-burst testing, random vibration testing, and low-level diagnostic sine sweeps. Notching, response limiting, and force limiting are addressed in detail, with examples.
- The course is primarily aimed at satellites in the 50 – 500 lb (23 – 230 kg) range, but most of it also applies to CubeSats. Most of the guidance applies to larger satellites as well if they will be tested on a shaker.
- The objectives are to improve your understanding of how to ...
 - establish an effective vibration test program
 - identify and clearly state test objectives
 - design (or recognize) a test that satisfies the objectives while minimizing risk of an over test
 - establish pass/fail criteria and interpret test data
 - write effective test plans and test reports
- Teacher: Tom Sarafin
- Target audience: All engineers and managers involved in ensuring small spacecraft can withstand launch environments

VTSS Course Topics



1. Overview
2. Test configuration, fixtures, and instrumentation
3. Low-level sine sweep testing
4. Sine burst testing
5. Random vibration testing
6. Notching and force limiting
7. Test documentation and reviews
8. Designing a SmallSat to Withstand the Vibration Test

“Anyone involved with the mechanical structure of a spacecraft or test engineering of ensuring a spacecraft survives launch should take this course – very well taught!”

“This course was an outstanding look at the way fundamentals of vibration illuminate the execution details of a good vibration test. Tom’s teaching style and many years of experience make for a rich and entertaining course in what is a very complicated topic.”

“Whether you’re a novice engineer with little to no experience in vibration testing or a more experienced engineer with vibration testing history, you’ll walk away from this class feeling you’ve learned a lot.”

NFLW—Notching and Force Limiting Workshop

NFLW is a one-day computer workshop that is available only as an optional follow-on to Instar's STD1 (Structural Test Design and Interpretation) or VTSS (Vibration Testing of Small Satellites) course, unless the instructor gives permission otherwise.

- The instructor provides in advance several MicroSoft Excel spreadsheets to be used in class. He then guides the class to complete the first spreadsheet by filling in the applicable equations, as covered in STD1 and VTSS. The class then uses that spreadsheet to work an example problem three ways, designing force limits, manual notches, and response limits. The class then similarly works two other examples.
- The objectives of this workshop are for you to “learn by doing”. The goal is for you to be able to design technically justifiable force limits, manual notches, and response limits for future tests.
- Course length: One 6-hour day

TenP—Ten Principles for Successful Space Programs for Leaders, Managers, and Consultants



1. Invest in knowledge and understanding
2. Adopt the right attitude: Put consistent priority on quality and mission success
3. Instill ownership and responsibility
4. Constantly improve teamwork and communication
5. Follow a sound engineering approach
6. Reduce cost and risk simultaneously with good engineering
7. Keep everything as simple as possible
8. Establish an effective quality system that involves everyone
9. Be willing to accept risks, but only those you and other stakeholders truly understand
10. Make sure you—and everyone else—have enough time, resources, and freedom to follow the above principles

- This course addresses the most challenging problem in the space industry: How do we reduce cost and schedule time while also trying to ensure a successful mission?
- After examining the driving issues in space-system development, the instructor introduces ten principles for successful space programs.
- The course then explores key aspects of leadership, management, and engineering within the framework of those principles.
- Objectives:
 - Provide a fresh focus on quality and mission success
 - Build understanding
 - Spur thought
 - Help your program improve efficiency of its organization and processes—from the top level of management on down to how every engineer or technician approaches his or her job
- Target audience: all leaders, managers, supervisors, systems engineers, and consultants involved in procuring, specifying, designing, producing, or testing space vehicles
- Course length: 2 or 3 full days

TenP Topics

Instar Ten Principles for Successful Space Programs

Let's Start by Identifying Common Problems in Space Programs

- Problems with requirements
 - Under specified, over specified, misunderstood, frequently changing
- Problems with designs
 - Difficult to build or test, penalizing to other designs, incompatible materials or fastening hardware, changing after release, mass and power growth
- Process and people problems
 - Decisions based on wrong information, hardware not built to engineering requirements, software not ready on time, test failures that programs can't recover from, not following procedures

Common elements:

- Problems that, in hindsight, could have been avoided
- Problems that seem to occur over and over again

X-33 liquid hydrogen tank test failure, Nov. 1999

NOAA-41 mishap, Sep. 2003

Instar Ten Principles for Successful Space Programs

In the Voyager 2 mission, a cable interference prevented full deployment of the instrument boom

The mission was degraded because the science boom did not fully deploy and engage its damper. The problem was fixed for Voyager 1, which launched two weeks after Voyager 2 in 1977, based on the test shown below on the test vehicle (spare).

High-gain antenna

Interference

Science instruments (not shown)

When the original boom deployment test was done, the high-gain antenna was not present or simulated.

Note: Voyager 1 launched later but passed Voyager 2 in flight

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1. Why Are Space Missions So Challenging?
2. Finding Solutions: Ten Principles
3. Building a Mission Success Culture and an Effective Team
4. Instilling Ownership and Responsibility in Contractors
5. System Development and Requirements Development (3-day course version includes requirements-writing workshop)
6. Reducing Cost and Risk By Design
7. Verification Planning
8. Managing Risk with a Quality System
9. Responsibly Accepting Risk

"This course does a good job of relaying that no process is going to work unless the people implementing it take pride/ownership in making it work."

"Great presentation of how the culture of a space program should be developed."

"All engineers from every field should understand the importance of a quality product as taught by this course."

"Anyone in the space industry or even thinking of being in the space industry needs this class."

"My sincere hope is that the fundamental principles here can take hold to move us forward."