



Structural Test Design and Interpretation

3-Day Course for Aerospace Engineers

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Disclaimer

Our intent is to provide dependable information and recommendations regarding structural test design and interpretation, but we cannot guarantee their accuracy.

Structural Test Design and Interpretation

3-day Course for Aerospace Engineers

- Objectives:** Improve your understanding of how to ...
- identify or define test objectives
 - design (or recognize) a test that satisfies the objectives while avoiding unnecessary risk of over-testing
 - technically justify notching and force limiting for vibration tests
 - establish pass/fail and success criteria
 - design test instrumentation and interpret test data
 - write a good test plan and a good test report

Audience: All engineers involved in ensuring that flight vehicles and their equipment and payloads can withstand mission environments and function as needed

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My Bio

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

Technical papers and full sets of course materials are available for free download at <https://instarengineering.com/resources.html>

Agenda

Introduction

1. Overview of Structural Testing
2. Designing 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 Test of a Large Spacecraft Telescope

Appendix: Test Documentation, Reviews, and Controls
(not covered in class)

Summary/Wrap-up

Detailed Contents

1. Overview of Structural Testing
 - Reasons for doing a structural test
 - Structural requirements and environments
 - Some definitions and a review of mechanics
 - The building-blocks verification process
 - Verification logic flows
 - Categories and types of structural tests
 - Selecting the right type of strength test
 - Pass/fail and success criteria
 - Taking responsibility for verification
2. Designing a Test
 - Designing a test
 - Test-article configuration
 - Boundary conditions
 - Ensuring adequacy of a strength test
 - A key difference between a qualification test and a proof test
 - Instrumentation
 - Preparing to interpret test data
 - Appendix: example problem 2-d, a variation of class problem 2-1

Detailed Contents (continued)

3. Loads Testing of Small Specimens
 - Applications and objectives
 - Common loading systems
 - Test standards
 - Case history: designing a small-specimen test to substantiate NASA-STD-5020 criteria for bolt analysis
4. Static Loads Testing of Large Assemblies
 - Introduction to static loads testing
 - Government standards for factors of safety and test factors
 - Special considerations
 - Introducing and controlling loads
 - Developing the load cases
 - Be sure to design the right test!

Detailed Contents (continued)

5. Testing on an Electrodynamic Shaker
 - Introduction and typical configuration
 - Limitations of testing on a shaker
 - Fixture design
 - Deriving loads from measured accelerations
 - Sine sweep testing
 - Deriving damping from test data
 - Sine burst testing
 - Understanding random vibration
 - Random vibration testing

Detailed Contents (continued)

6. Notching and Force Limiting

- List of references for this section
- What is notching and its technical justification
- The dynamic absorber (impedance) effect
- Three methods of notching relative 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 NASA's 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

Detailed Contents (continued)

7. Overview of Other Types of Structural Tests
 - Modal survey testing and correlation of the finite element model
 - Centrifuge testing
 - Acoustic testing
 - Shock testing
8. Case History: Structural Testing of a Large Spacecraft Telescope
 - Overview
 - Originally planned structural test program
 - Problem statement
 - Revised approach
 - Telescope testing details
 - Test anomalies
 - Lessons learned and conclusions

Appendix: Test Documentation, Reviews, and Controls
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 Structural Testing

- Reasons for Doing a Structural Test
- Structural Requirements and Environments
- Some Definitions and a Review of Mechanics
- The Building-blocks Verification Process
- Verification Logic Flows
- Categories and Types of Structural Tests
- Selecting the Right Type of Strength Test
- Pass/Fail and Success Criteria
- Taking Responsibility for Verification

Some of the pages in this section are from Instar's course "Spacecraft Structures, from Concept to Launch" (SCS). Some of the figures and tables are modified versions of ones from the book *Spacecraft Structures and Mechanisms: From Concept to Launch* (SSAM) (Ref. 1).

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What Are Some Likely Reasons for Doing a Structural Test?

- ~~It's a requirement imposed by customer (or launch service provider).~~
 - 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 (e.g., why your customer is requiring the test).
- Verify compliance with a true requirement, such as “shall withstand mission environments without jeopardizing the mission.”
 - Or verify compliance for a required characteristic, such as stiffness or modal frequencies.
- Acquire information.
 - To obtain needed properties for materials or joints
 - To help develop manufacturing processes
 - To improve a design
 - To correlate a math model

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

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

Requirements Stem From Functions and Constraints

Functions

- What the product must do or be able to do

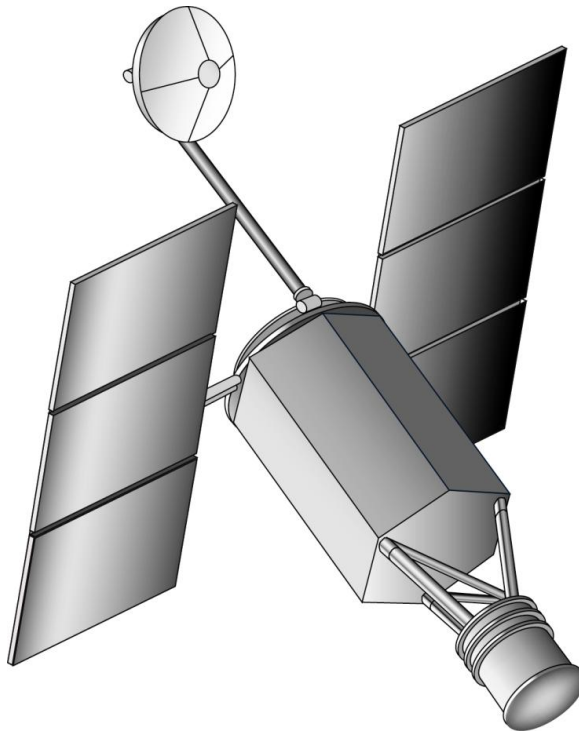
Constraints:

- Limitations, driven mostly by ...
 - interfaces
 - allocations from higher-level requirements

So what are some examples of structural functions and constraints?

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

Common Structural Functions



The structure may serve nonstructural functions as well, such as conducting electricity and transferring or isolating from heat.

- Physically support something
- Attach two or more items
- Protect sensitive components or people from environments, such as vibration or radiation
- Maintain alignment of sensors, antennas, and moving parts, as needed for system performance, considering ...
 - permanent displacement (e.g., from launch),
 - temporary displacement (from temperature changes in orbit),
 - and **jitter**: on-orbit vibration from operating equipment

From my course “Spacecraft Structures, from Concept to Launch” (SCS)

Typical Constraints

- **Dynamic envelope**—the physical space the item must stay within while deflecting under service loads
 - **Static envelope**—a smaller envelope derived from the dynamic envelope and an allocation for deformation under loads
- Mass properties (mass, center of mass (or CG), moments of inertia)
- Mechanical interface
 - Bolt pattern
 - Flatness
- Constraints driven by the need for access (e.g., for installation and removal)
- **Fundamental frequency** (lowest natural frequency for a structure)
- Environments
- Programmatic constraints such as cost and schedule

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

Flight Vehicles Must Function Despite Exposure to Life-cycle Environments

- Corrosive environments on the ground
- Gravitational loading
- Steady-state acceleration
- Transient loads, e.g., impulses
- Pressure, vacuum
- Aerodynamic turbulence
- Aerodynamic heating
- Vibration
- Shock
- Thermal radiation
- Charged-particle radiation
- Electromagnetic interference
- Magnetic fields
- Atomic oxygen



F-15 (courtesy U. S. Air Force)

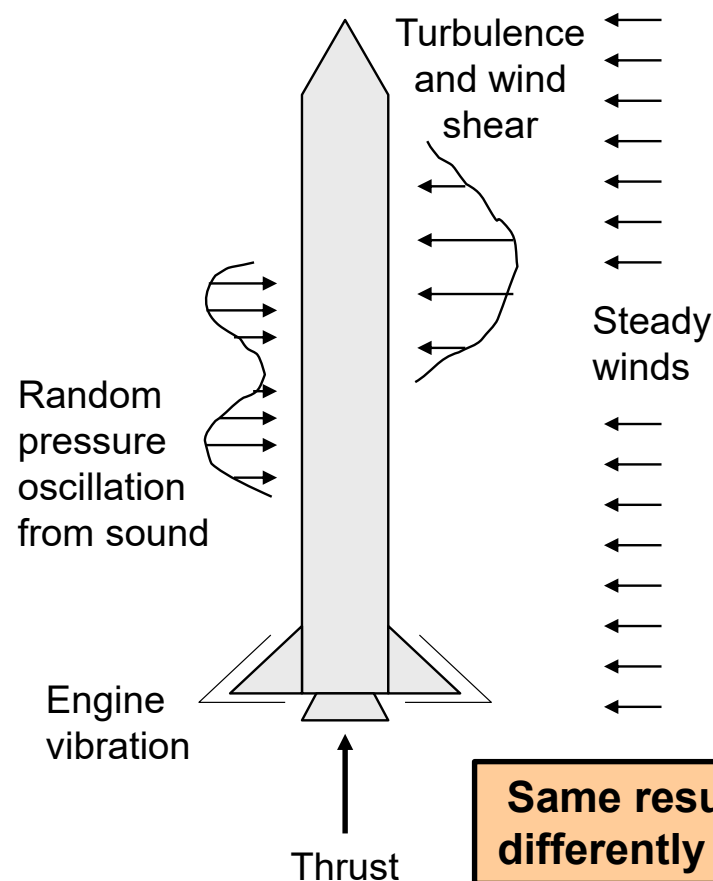


Atlas V (courtesy U.S. Air Force)

Environments are constraints under which the product must function.

An important structural function can be to protect something or someone from environments.

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)
 - 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**)

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 structural tests, 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:

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

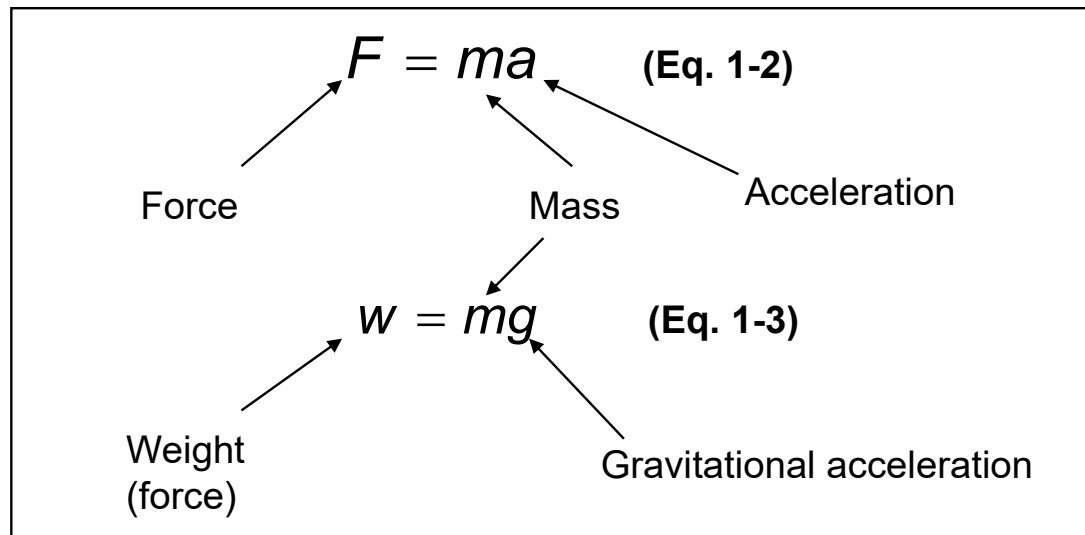
where a = acceleration (in/s² or m/s²)
 n = load factor
 g = gravitational acceleration (in/s² or m/s²)

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.

$$F = ma$$

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

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

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

Internal force $\Rightarrow F_i = nw$ (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.

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

$$a = 5g$$

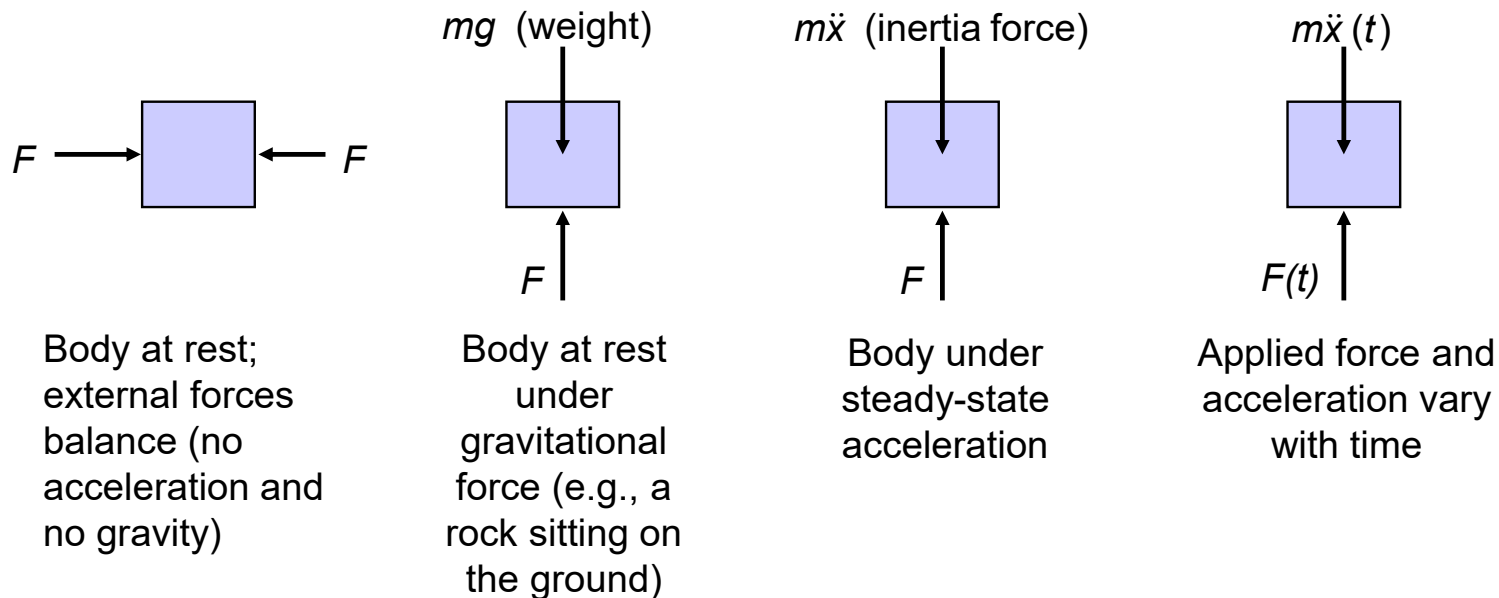
$$n = -5$$

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

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

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

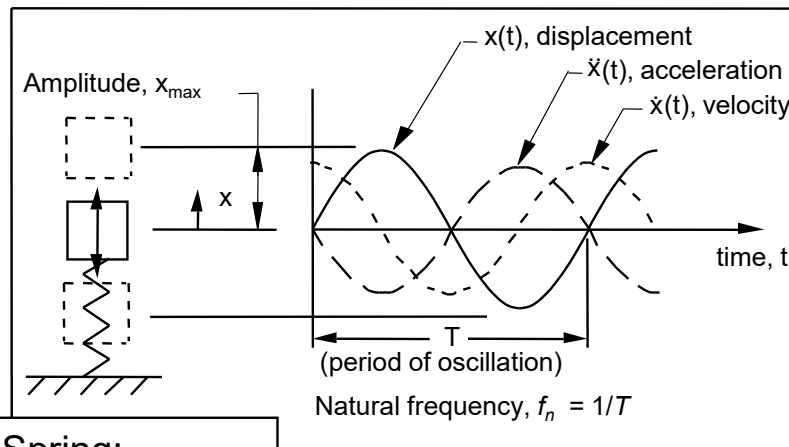
Note: If the first mode for a spacecraft is bending of a small antenna, that mode is considered the fundamental mode of the antenna but not of the spacecraft.

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.



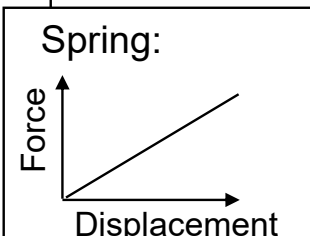
$$\text{Displacement, } x(t) = x_{\max} \sin(\omega_n t) \quad (\text{Eq. 1-5})$$

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

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

Note: Acceleration is proportional to displacement.

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



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 experience with 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:

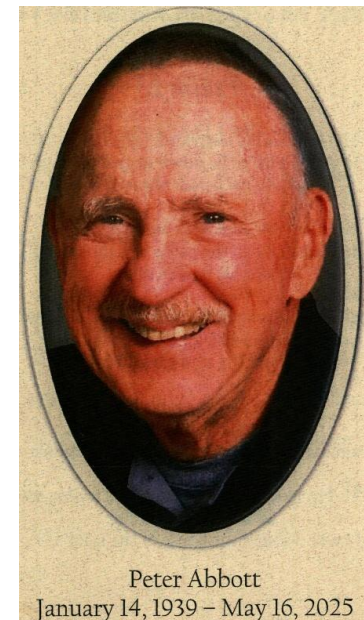
$$m\ddot{x}(t) + c\dot{x}(t) + kx(t) = F(t) \quad (\text{Eq. 2-13})$$

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. 1) Sec. 7.2.)



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

What Does “Verification” Mean in the Space Industry?

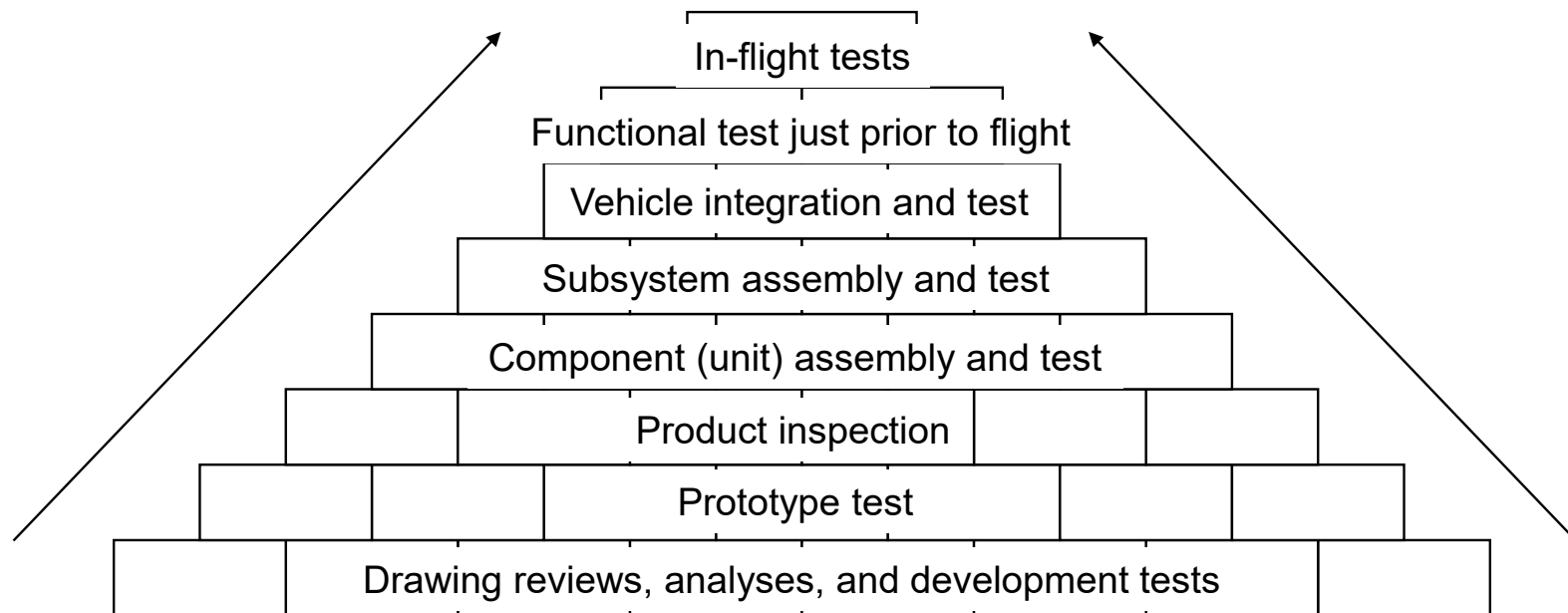
In absence of a standard, accepted definition, here’s mine for space programs:

Verification means establishing confidence before launch that the product or system will satisfy its requirements.

- “before launch”—For hardware, verification takes place while we can still change something; after launch it’s usually too late.
- “establishing confidence”—not proof
 - We can show our product satisfies defined, measurable criteria.
 - But we can’t prove the mission will be successful—and trying to do so would be prohibitively expensive.
- “will satisfy its requirements”—This means we have to understand the product’s requirements before attempting to verify compliance.

Structural testing plays important roles in the verification process, particularly regarding the need to function as required either during or after exposure to environments.

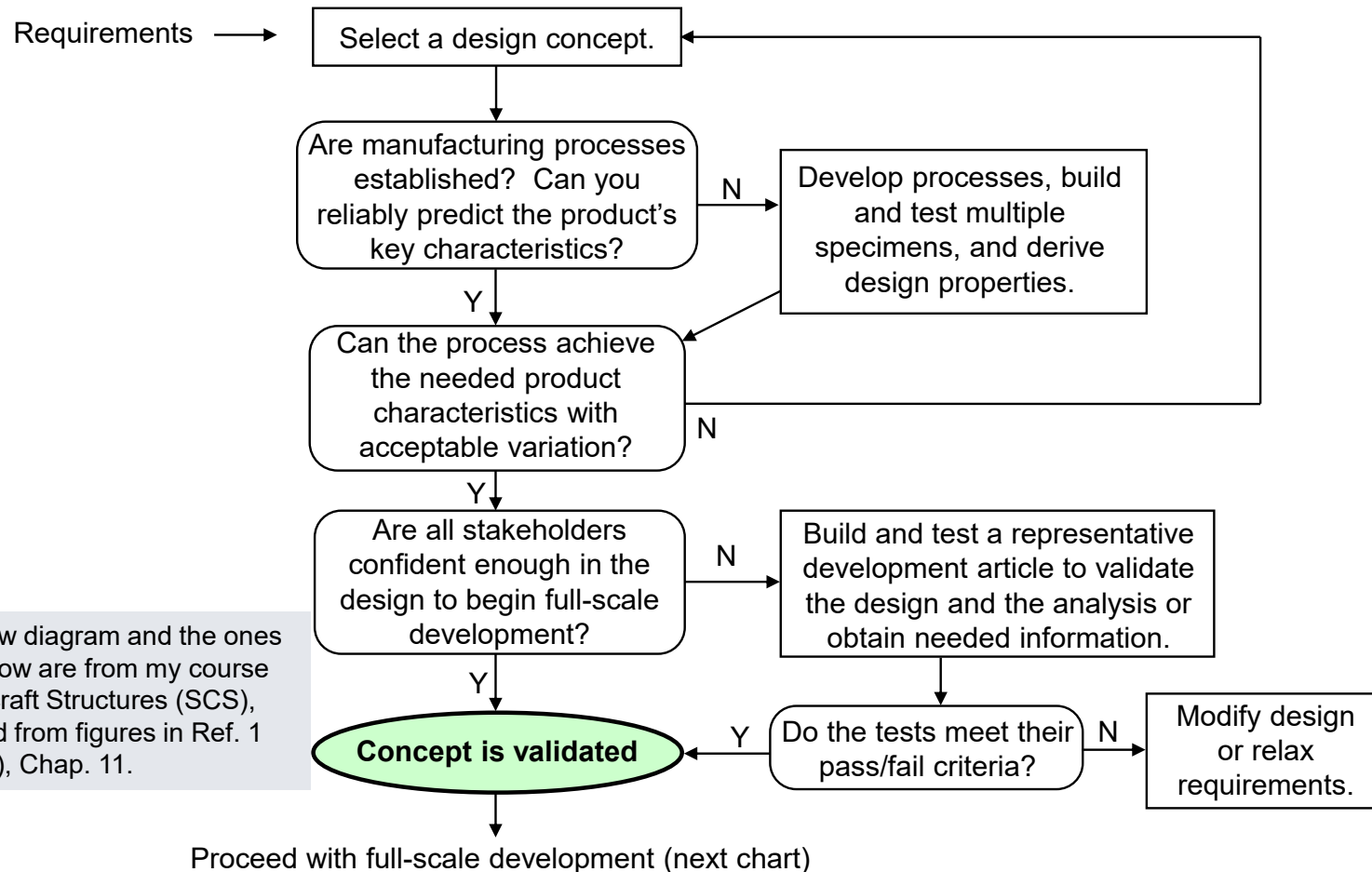
Verification Follows a Building-blocks Process



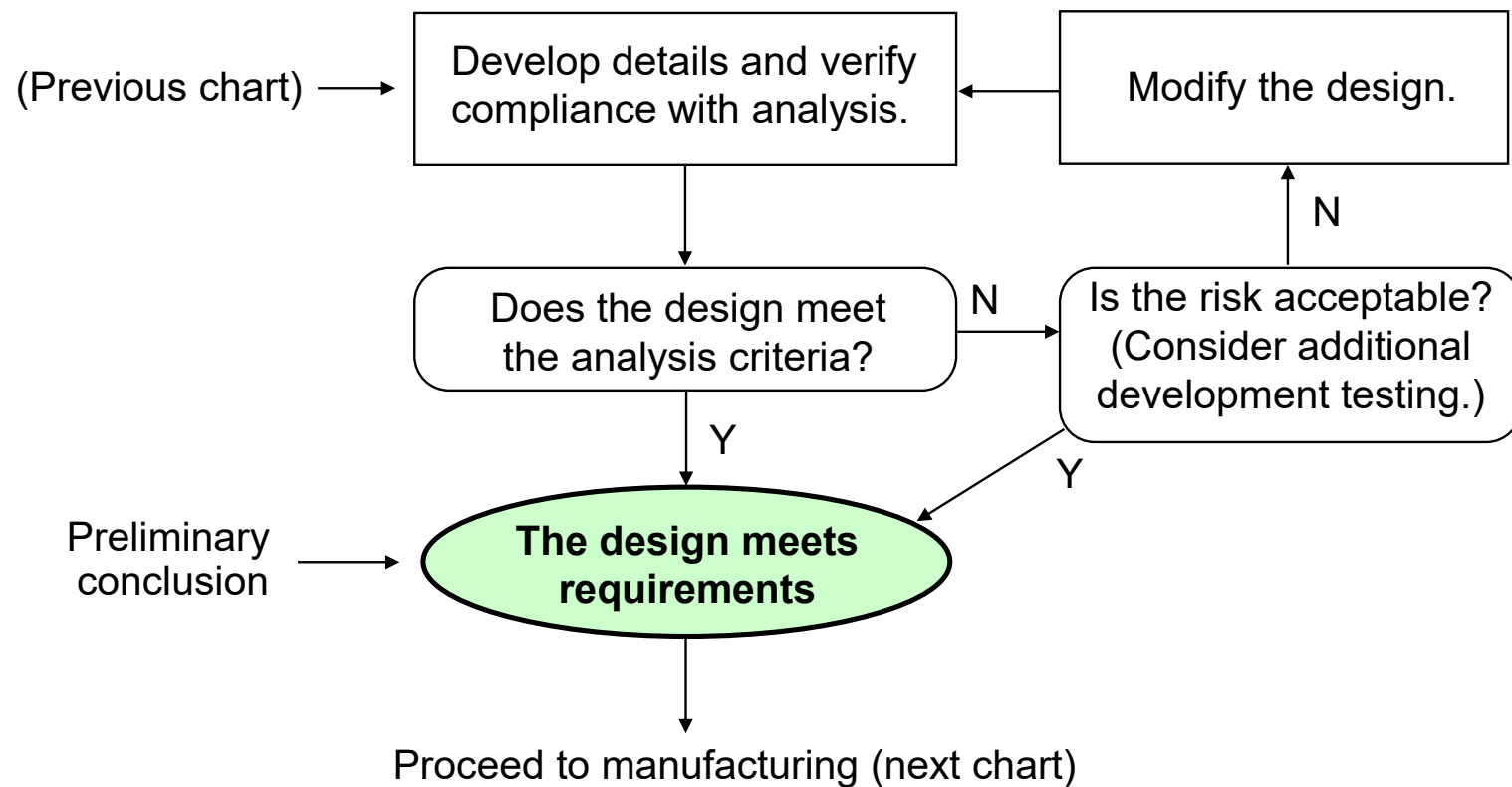
Objective: Root out problems when they are least costly to find

When planning your verification program, don't defer risk to a later test if a low-level test is capable of detecting a potential defect. System-level tests should be for verifying interfaces and subsystem interactions.

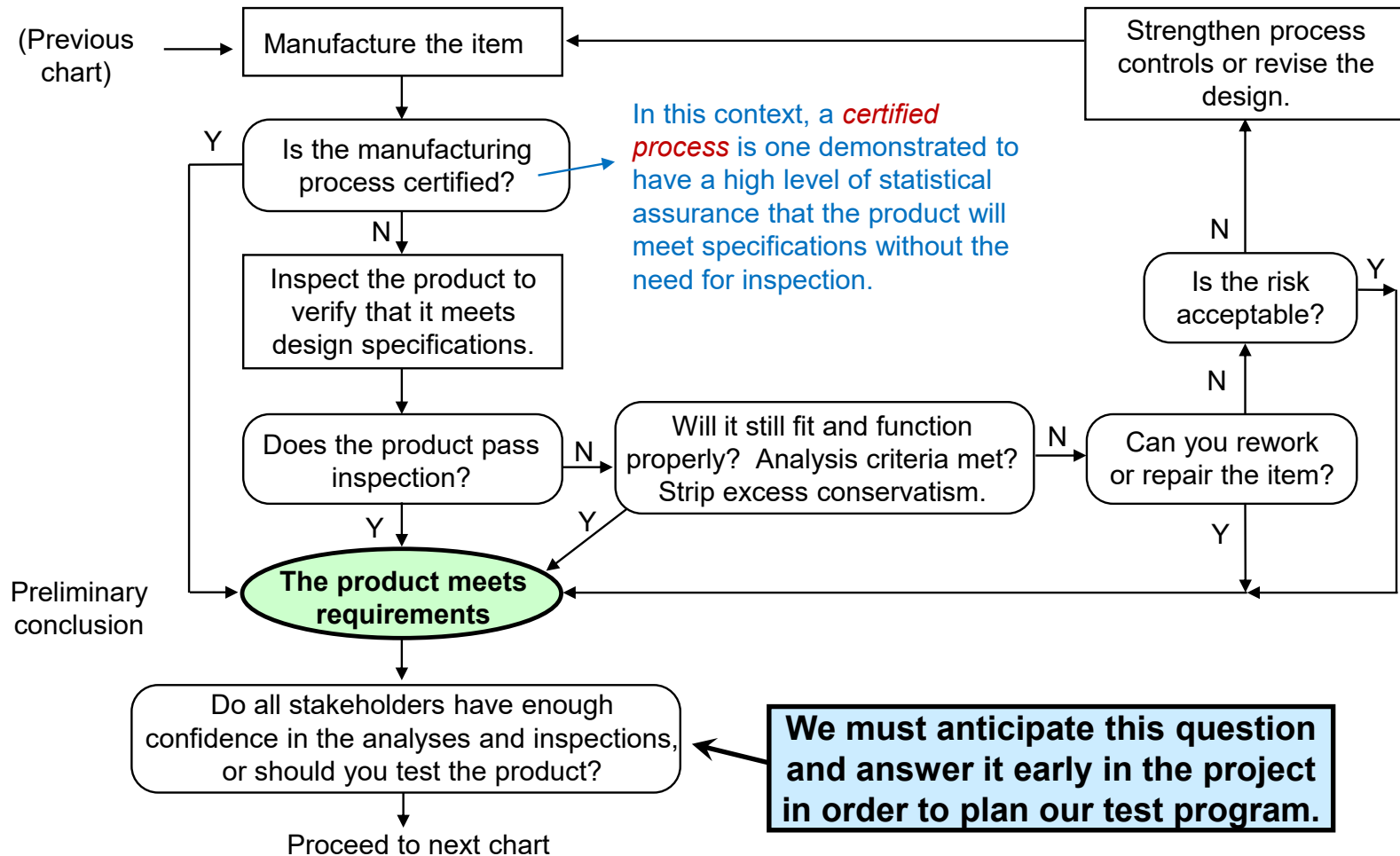
Verification Logic Flow during Conceptual Design



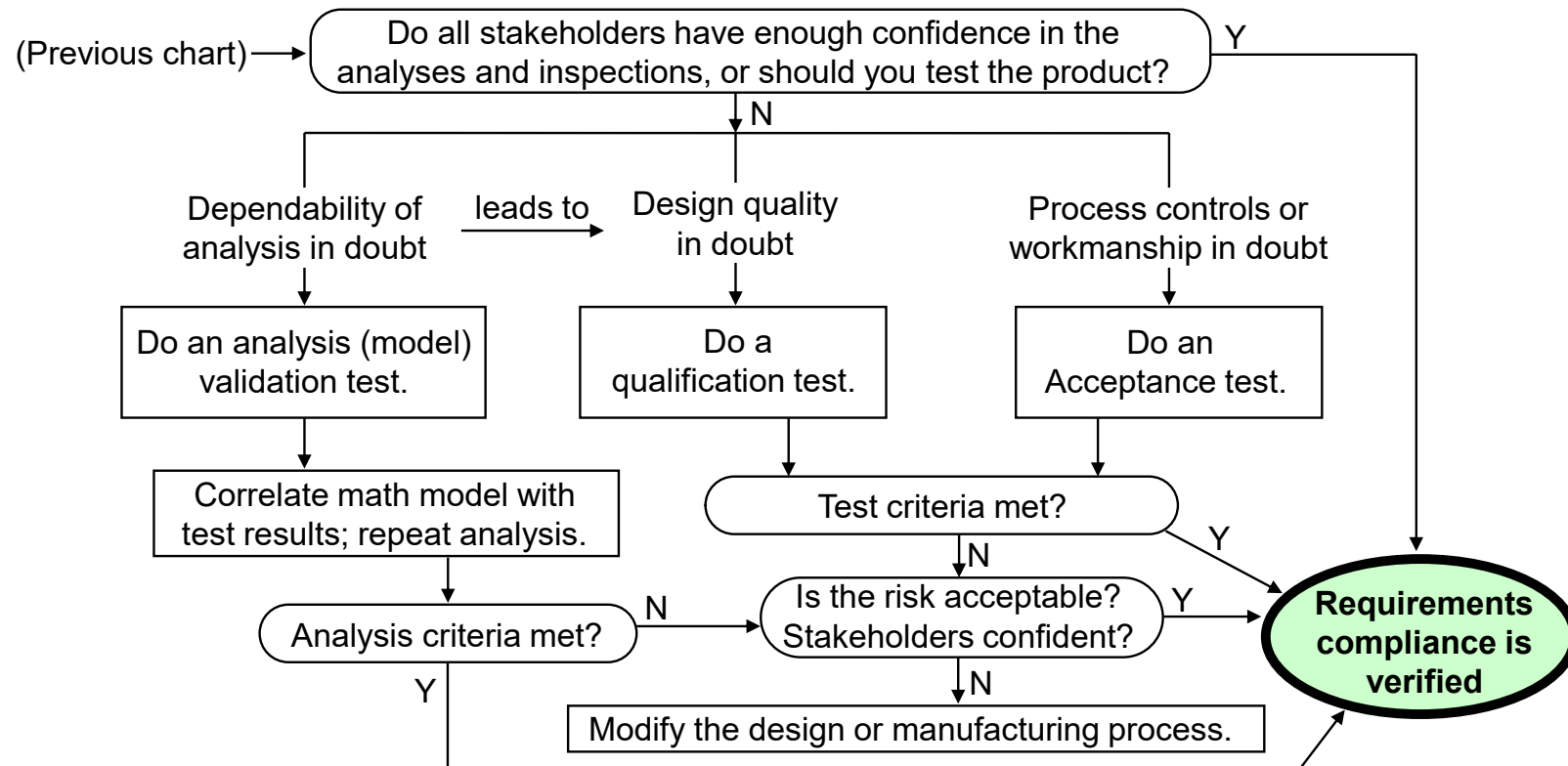
Verification Logic Flow: Full-Scale Development, Detail Design



Verification Logic Flow: Manufacturing



Verification Logic Flow: Test and Final Verification



There are many routes to verification.

Make sure you understand the reason for doing a test!

Test Classification

Test Class	Test Article	Specific Objectives	Examples
Development	Something built to a design that is representative of but not identical to the flight design (e.g., engineering model)	Examples: Acquire data for use in developing manufacturing processes, deriving material properties for design, substantiating methods of analysis, or becoming confident in a design.	Strength testing of bonded-joint specimens
Qualification (a.k.a. prototype)	Dedicated hardware not intended for flight but built to the flight design	Establish confidence in a design by testing a single build to loads or environments that are more severe than what the flight hardware will ever see.	Static loads, random vibration, acoustics
Acceptance (referred to as a proof test for strength testing)	Flight hardware (each build) of a design that has been qualification (prototype) tested	Establish confidence in a particular build, given variation in materials and manufacturing processes, by testing to the maximum expected loads or environments (or somewhat higher).	Static loads, random vibration, acoustics
Protoflight (NASA) or Flight-proof (Space Force)	Flight hardware (each build), when there is no qualification (prototype) test.	Same as for acceptance testing, but test environments are more severe to compensate for absence of qualification testing.	Static loads, random vibration, acoustics
Protoqualification (Space Force programs)	Flight hardware (first build only), when there is no qualification test; subsequent builds are acceptance tested.	Combined objectives of qualification and acceptance testing.	Static loads, random vibration, acoustics
Analysis-validation (or model-validation)	Varies	Obtain data that will be used to validate or correlate a math model or a method of analysis.	Modal survey (a.k.a. mode survey)

Note that, for strength (e.g., static loads) testing of structures made solely of well-controlled metals, with well-controlled manufacturing processes (as evidenced by statistical data), proof testing of each build is not typically expected or required as long as a single build passes a qualification or protoqualification test.

General Types of Structural Tests

Category	Description and Purpose (Structural)	Examples	Comments
Environmental test (controlled environment)	Put the test article through simulated mission environments to verify requirements for strength, life, or general mechanical integrity (soldered wires, connectors, etc.).	• Vibration testing without notching* • Acoustic • Shock	The test meets objectives as long as the right environment and boundary conditions are used.
Controlled-load test (or controlled-response test)	Put the test article under conditions that will achieve target accelerations, external or internal loads, or temperatures. Verifies strength or life. Relies more heavily on analysis than does true environmental testing.	• Static loads • Cyclic loads • Centrifuge • Vibration testing with notching • Sine burst • Pressure	The test is only as good as the analysis used to derive the test loads.
Other	Verify requirements for structural characteristics, or obtain data.	• Static stiffness • Low-level sine sweep • Modal survey	The test meets objectives if all needed data are collected and validated.

***Notching** means reducing the input environment at and near a natural frequency of the test article in order to avoid unrealistically high response acceleration. Addressed Sec. 6 herein.

Common Types of Controlled-Load Tests

Type	Objective	Description	Applications
Static loads (discussed in Secs. 3 & 4)	Verify strength or determine stiffness	Apply discrete loads at a slow rate of change so that there will be no dynamic response.	Small specimens, such as joints; large structures that will be significantly loaded during the mission by excited low- frequency modes of vibration
Sine burst (Sec. 5)	Verify strength	See next page	Small or mid-size structures; discussed later in this section
Centrifuge (Sec. 7)	Verify strength	Apply inertia loads by spinning the test article on long arm.	Compact structures that will be loaded during the mission mostly by uniform acceleration, when sine burst testing is impractical or inadequate
Cyclic loads (Sec. 3)	Verify fatigue life, or acquire fatigue data	Apply cyclic loads of controlled magnitudes.	Structures for which fatigue life is of concern, such as multi-mission hardware

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.)

Do We Really Need a Dedicated Strength Test?

For an item that will be on a shaker, why can't we use the random vibration test as a strength test for the 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

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, incurring significant risk of failure.

- 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 items weighing less than 50 lb or so.

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

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.

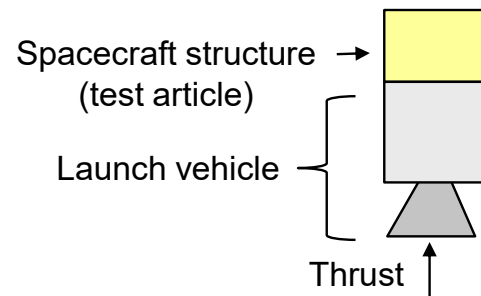
When we can't over-design the primary structure to withstand the uncertain loads associated with random vibe or sine vibe testing, we're left with the more traditional options for strength testing :

Static loads, centrifuge, or sine burst.

Let's compare them ...

In Sec. 5, 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.

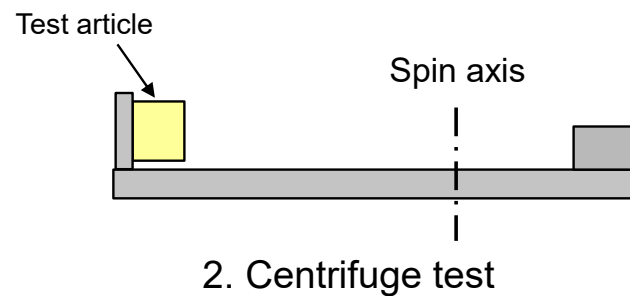
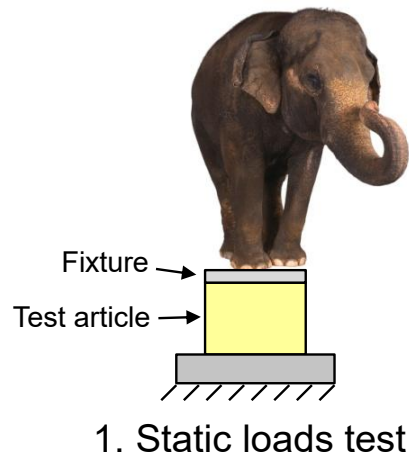
Options for Strength Testing



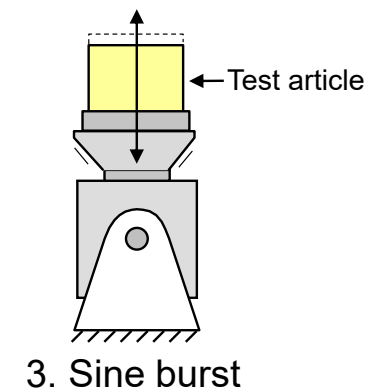
We want to test a spacecraft's primary structure to verify that it has enough strength to withstand the design ultimate load without rupture or collapse.

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

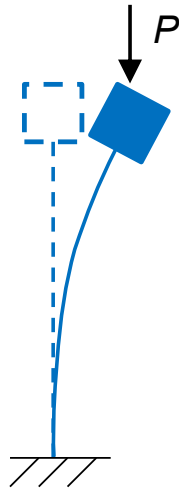
Type of test	Advantages and applications	Disadvantages and limitations
Static loads	Can simulate inertia loading from response of one or modes of vibration combined with steady-state loads. Often best for primary structures of large vehicles.	Application of concentrated loads can be more stressful than 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 Sec. 2).	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 in Sec. 5).

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

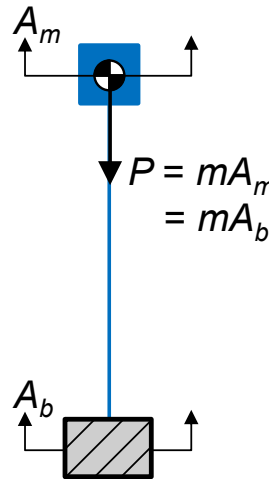
Example, Limitations of Sine Burst for Strength Verification: Column Buckling



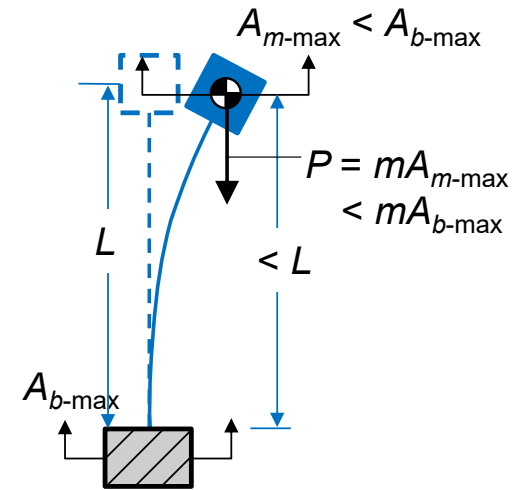
a. Static load, P , applied to mass, m .



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



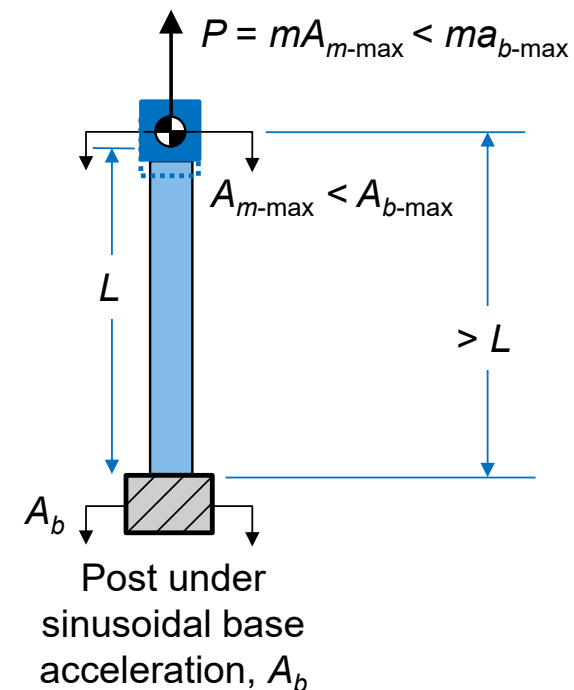
c. Column under sinusoidal base acceleration, A_b . If the column is rigid, $A_m = A_b$.



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.

Another Example, 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 and negligible 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 flight vehicles and 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 smaller flight structures, a higher percentage of total flight load is dynamic.

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

For test effectiveness, 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, which we'll discuss in Sec. 5.

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.

Taking Responsibility for Verification

As stated previously, **verification** in the space industry means making yourself and other stakeholders confident that your flight vehicle or other product will meet its requirements.

Spacecraft structural testing is most often 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 (LSP). **The LSP has no reason to care whether its payloads function properly in orbit!**

Verification is the responsibility of the product developer, not the customer or the LSP.

Again, the reason you're doing a 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 from This Section

- Two main reasons (objectives) for doing a test:
 - Verify compliance with a true requirement
 - Acquire information
- Start the process of designing a test by clearly defining the objectives, related to the requirements being verified or the information needed.
- Use the logic in the flow diagrams presented in this section when planning the verification program.
- Static loads and centrifuge tests are more effective at finding strength deficiencies in relatively large structures than vibration tests, especially when the critical modes of failure are ductile or are some form of instability.
- Clearly define the pass/fail or success criteria before the test, and be ready to investigate violations of the criteria quickly during the test.
- 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. Designing a Test

- Designing a Test
- Test-article Configuration
- Boundary Conditions
- Ensuring Adequacy of a Strength Test
- A Key Difference between a Qualification Test and a Proof Test
- Instrumentation
- Preparing to Interpret Test Data
- Appendix: Example Problem 2-d, A Variation of Class Problem 2-1

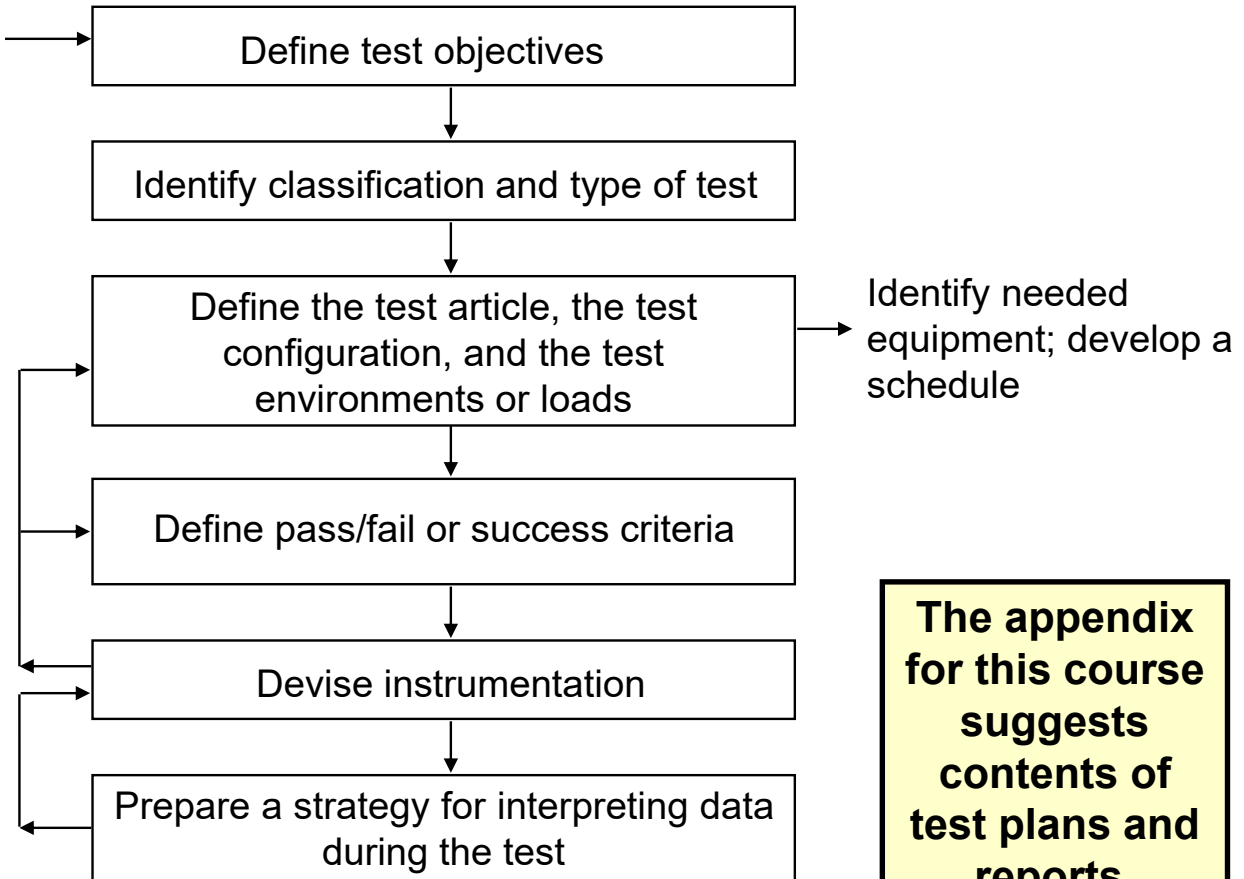
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Designing a Test—a High-level view

Always start with a clear definition of the objectives!

Coordinate the test with a test plan.

Iterate:
Develop and distribute a preliminary plan in order to scope the test and keep stakeholders informed, then revise the plan as the test design evolves.



The appendix for this course suggests contents of test plans and reports.

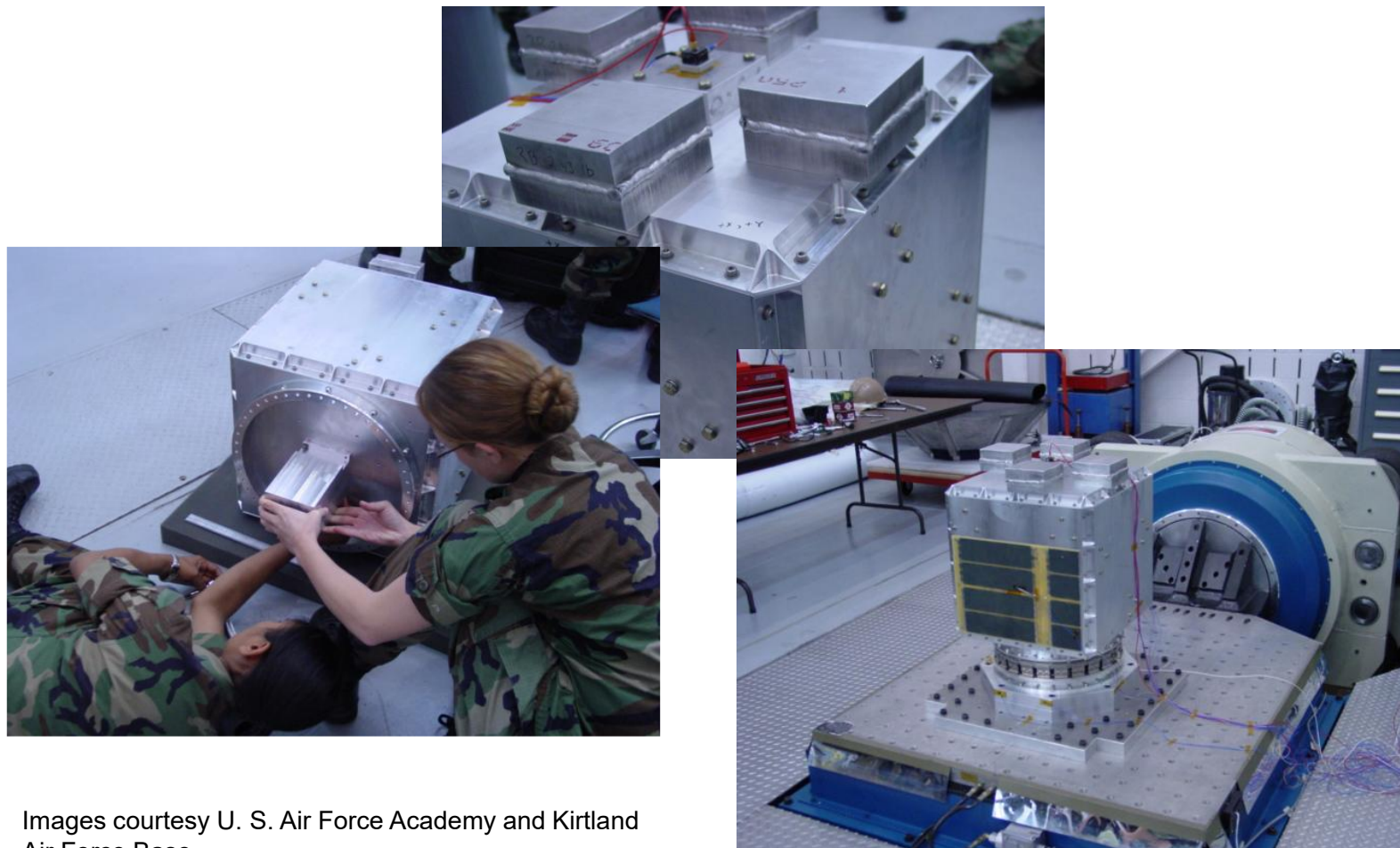
Test-article Configuration

The configuration of the test article may not be the same as the flight configuration.

- To reduce risk, testing of the primary structure often takes place before vehicle integration.
 - The fully integrated assembly then is tested for vibration.
- 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.

**Plan to generate a separate finite element model for the test configuration.
Start with the model used for loads analysis, making only the changes
needed to reflect configuration differences.**

FalconSAT-3 Engineering Model with Mass Simulators



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

Ensure Proper Boundary Conditions

Potential Problems	How You Can Avoid Them
Test fixtures can over-constrain the structure in a strength test. – Improper load distribution – Artificially enforce alignment	• Mount the test article on a structure that is of the same design as the flight structure that will support it. • Or design and build a stiffness simulator to support the test article.
Test fixtures can contaminate test data. – Unwanted flexibility or damping can affect the test article's stiffness, natural frequencies, or damping, and may even invalidate the test.	• Design stiff fixtures; confirm with analysis. • Measure and account for fixture characteristics.
Configuration of test article and test cell can affect the environment (e.g., acoustics).	• In an acoustic test, position multiple control microphones approximately 2 ft from the test article surface.

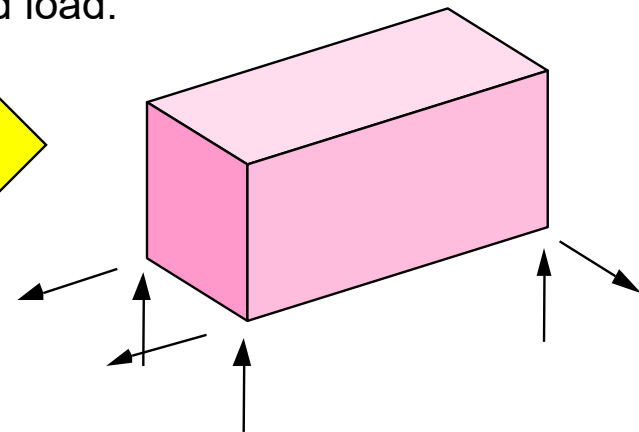
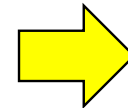
The right approach depends on the test and its objectives.

Structures with Statically Indeterminate Mounting Interfaces Can Be Difficult to Test Adequately

A **statically determinate** (**kinematic**) mounting interface has six well-selected constrained degrees of freedom (DOFs), just enough to satisfy the six equations of static equilibrium for any applied load.

If this box has any more constrained DOFs than shown, its structural loading is complicated in two ways:

- Under applied loads, the interface loads (**reactions**) depend on the stiffness of the box relative to the stiffness of the box's mounting structure.
- The interface is loaded when the support structure distorts for any reason (e.g., from structural loads or temperature change).

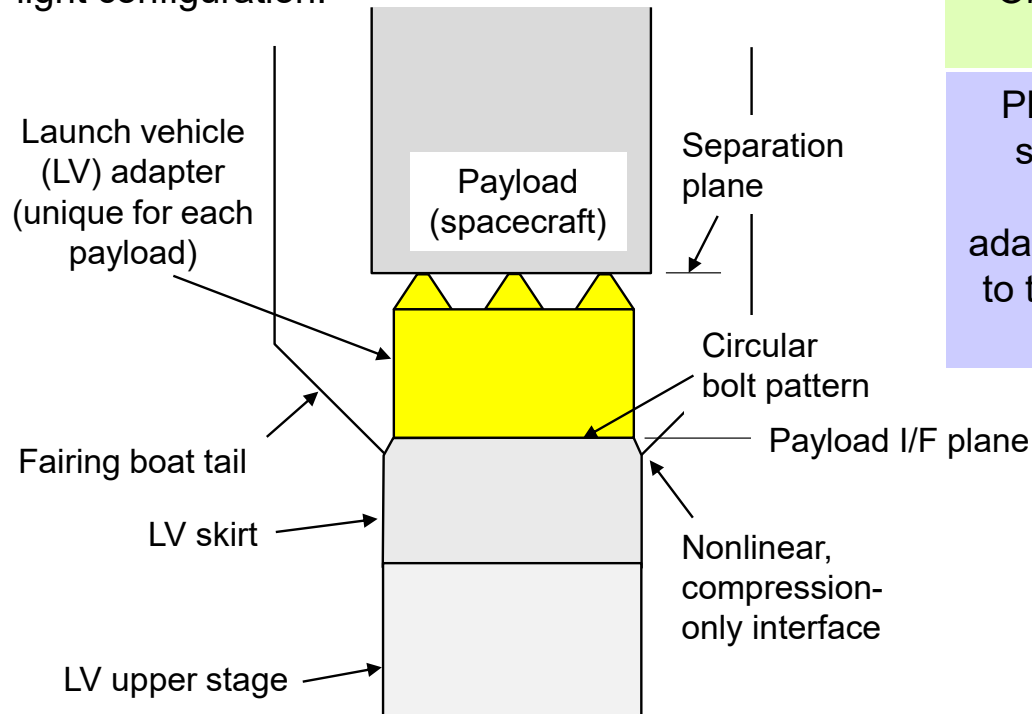


Example of a statically determinate interface, with six constrained DOFs

So how do you design a good test for a structure with a statically indeterminate mounting interface?

Example Problem 2-a: Designing a Static Loads Test for a Launch Vehicle Adapter

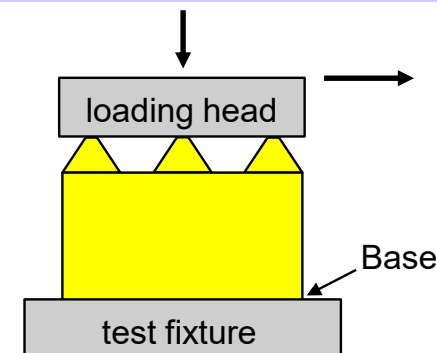
Flight configuration:



During launch, loads from the fairing come into the LV skirt just 2" from the LV adapter, thus distorting the adapter's mounting interface.

Objective: Verify strength of the payload adapter.

Plan: Bolt the adapter to a rigid support fixture, attach another fixture (loading head) to the adapter, and then apply static loads to the loading head to achieve the required loads at the base.



Problem: The test fixture overconstrains the base.

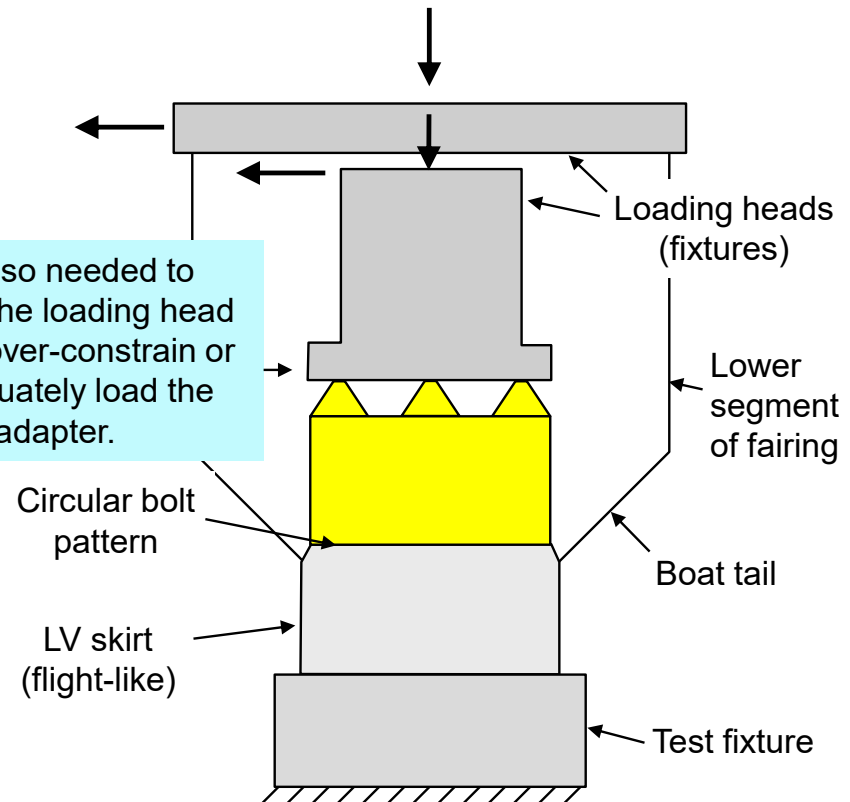
How would you test this adapter?

Example Problem 2-a: Designing a Static Loads Test for a Launch Vehicle Adapter, continued

Here's what we did in 1989:

- Mount LV skirt to rigid fixture
- Mount LV adapter on top of LV skirt
- Attach lower segment of payload fairing with boat tail
- Mount rigid loading heads to tops of adapter and fairing segment
- Apply loads to both loading heads at different relative clocking angles

We also needed to ensure the loading head did not over-constrain or inadequately load the adapter.



For any given structure, use analysis to determine the test configuration that will achieve the test objectives.

Basic Process for Ensuring Adequate Design of a Strength Test

1. Make the overall test environments or loads representative of the maximum expected flight conditions.
 - Key indicators: total forces and moments at major interfaces and section cuts
2. Perform an apples-to-apples comparison:
 - Identify potentially critical areas or failure modes in the structure.
 - In each critical area, show that the internal loads or stresses predicted for the test are at least as high as those predicted for the limit (maximum expected) flight loads.
 - Use the same math models (revised only to reflect configuration differences) and assumptions for making the above comparison.
3. Apply the specified test factor to the test loads.

Example Problem 2-b: Deriving the Target Acceleration for a Sine Burst

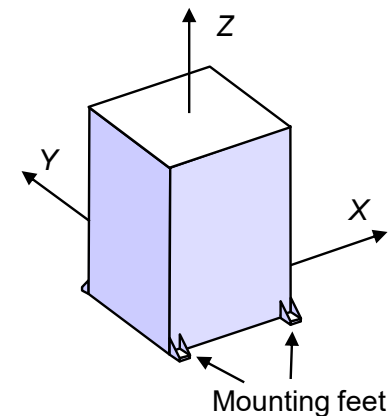
You're planning a sine burst test for an item that will be bolted to the primary structure of a host vehicle.

- Square bolt pattern with four mounting bolts, one at each corner

The specified limit quasi-static loads are

- $\pm 8.5 g$ in each of any two orthogonal axes acting simultaneously.

Must test to 1.25 times the limit loads to verify strength.



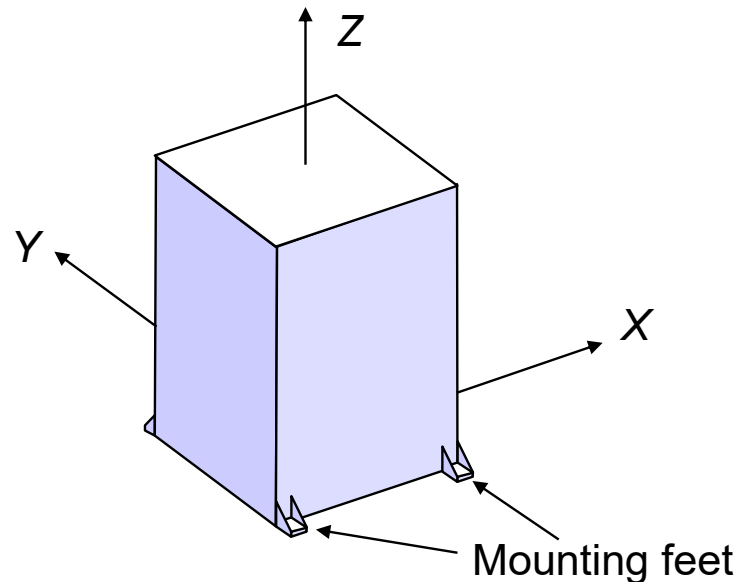
Problem statement:

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

continued

Example Problem 2-b, 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

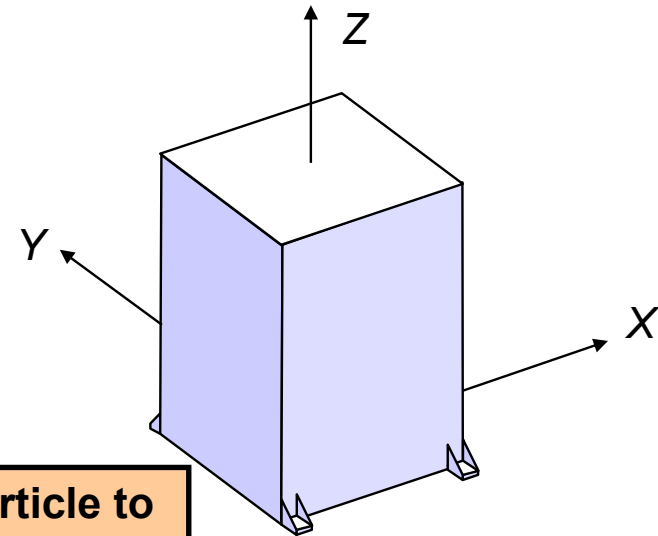
Example Problem 2-b, continued: Option 2

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 2-b, continued: Option 3

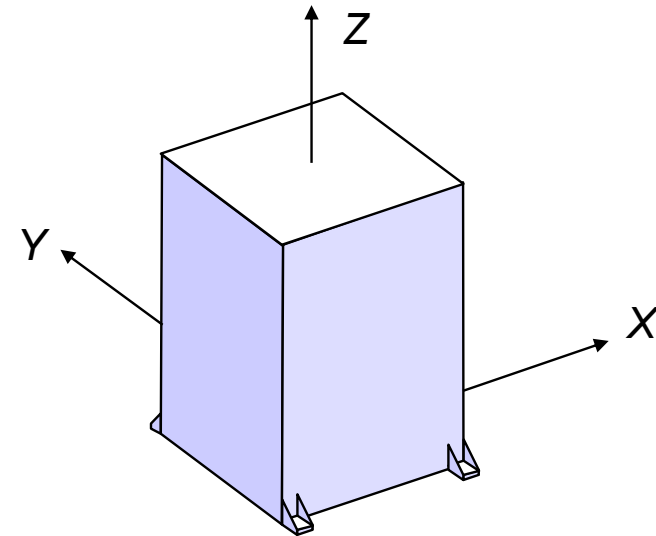
Add the accelerations:

$$8.5 + 8.5 = 17.0 g$$

Multiply the sum by the 1.25 test factor:

$$17.0(1.25) = 21.3 g$$

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



Such an approach would adequately load the structure.

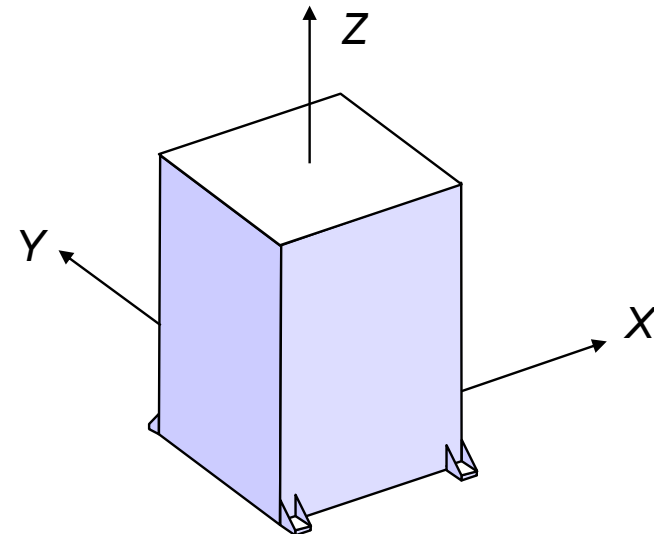
But most of the structure would be excessively loaded.

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

continued

Example Problem 2-b, 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.



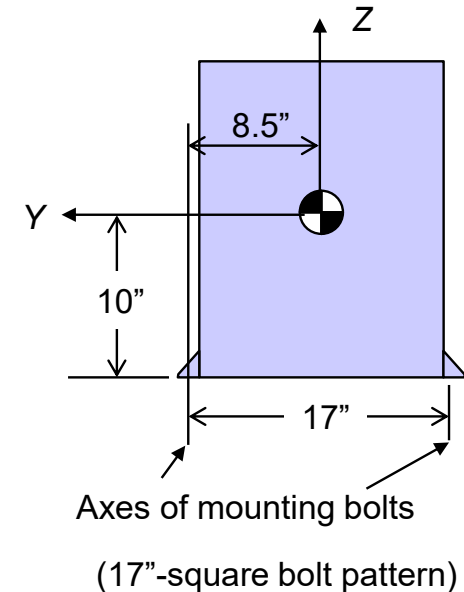
This is an example of how to design a test to meet its objectives.

Class Problem 2-1: Designing a Sine Burst Test

(in continuation of Example Problem 2-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 limit loads are $\pm 8.5g$ in each of any two axes acting simultaneously.*
- 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.



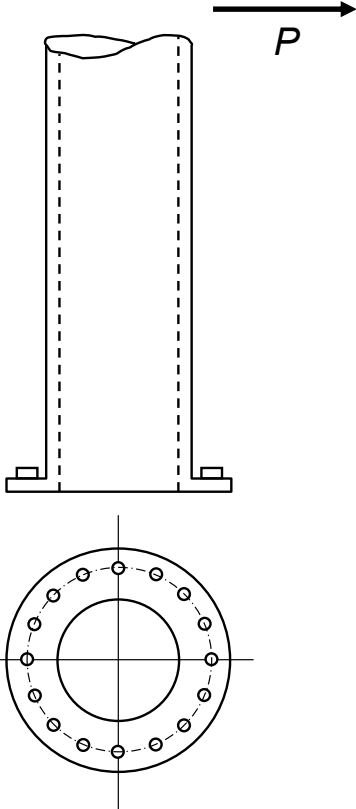
Problem statement:

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 specified limit loads are $\pm 8.5g$ in the X axis combined with $8.5g$ in any axis within the Y-Z plane.

A Key Difference Between a Qualification Test and a Proof Test

Example Problem 2-c: A cantilevered cylinder (flight hardware), bolted at its base, can be equally loaded in any lateral direction during flight. Devise a strength test for the cylinder and its mounting bolts.

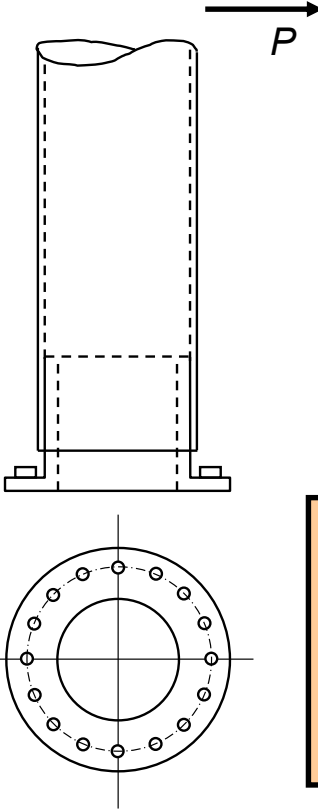


Situation #1:

The cylinder is machined 6061-T6 aluminum alloy.

We apply a load P , as shown, in any lateral direction to achieve the required base shear and moment.

We're done.



Situation #2:

The cylinder is made of carbon-fiber composite and is bonded to a metal fitting.

Can we test this joint with a single load case, as we did in Situation #1?

Understanding Proof Testing

A proof test is an acceptance test, intended to verify strength of each build, given defects introduced by random variables in manufacturing processes and workmanship.

- Applicable for materials and joints that historically have wide variation in strength, such as brittle materials, composites, bonded joints, and potted inserts
 - Parts made from 3-D printing are in this category until we acquire enough data that shows little variation in strength between specimens.
- For structures made solely of well-controlled metals (strength controlled by specifications) and manufacturing processes, testing only one of each type of joint—a qualification test—is typically sufficient (test load includes a qualification test factor).

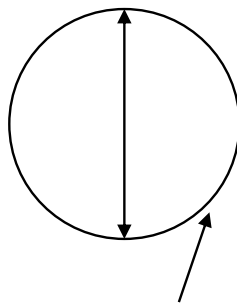
To meet the objectives of a proof test, we must test each potentially critical region of each part and each joint.

The same logic applies for all acceptance tests.

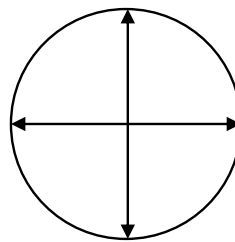
For example, an acceptance test for random vibration should be done in all three axes, even if the test article is symmetric.

Proof Testing of a Bonded Joint

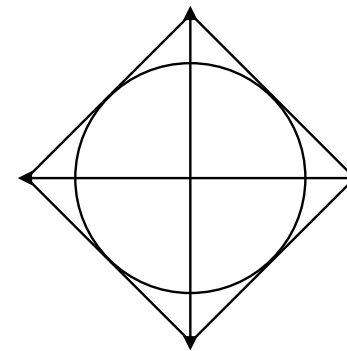
Returning to Example Problem 2-c:



At this location, testing in the direction shown would not adequately stress the bond



Neither would two test cases, 90° apart.



If we want to test the entire bondline with two +/- load cases, we would have to increase the applied load by 41%.

If we plan to test this way, we need to design the bonded joint to withstand this increased load.

Recognize That No Test Will Be Perfect

- Criterion for a good strength test: Show by analysis that all critical areas of the structure are stressed by the test conditions at least as severely as they would be under flight limit loads multiplied by the applicable test factor.
 - Qualification test: Only one of each identical part or design aspect must be tested.
 - Acceptance test: Each critical part must be tested, regardless of symmetry.
- With proper planning up front, you'll easily be able to meet this criterion for perhaps 95% of the structure.
- Even with proper planning, though, meeting the criterion for the remaining 5% or so may greatly impact the program.
 - Higher test cost, more time in test, risk of failure from overtesting elsewhere, weight impacts as a result of designing the structure to pass the test

**As is normally the case, failure to satisfy a criterion does not necessarily mean failure to meet a requirement (or that the structure will fail in service).
It simply means increased risk.**

Elevate violations of test-design criteria to management, and make sure all stakeholders have a vote.

Guidelines for Effective Instrumentation

Start by defining the objectives for instrumentation: Provide information for ...

- controlling or tuning the test environment
- assessing the health of the test article
- determining whether the test was successful
- correlating a math model or an analysis method
- developing or improving a design or a manufacturing process

Then discuss the objectives with an instrumentation expert to devise a strategy.

- Identify the type of information that will best meet the objectives (displacement, stress, force, acceleration)
- Define the needed accuracy
- Find an instrument, a location, and a process for obtaining what you need
- Make sure you can measure any important influence of test fixtures

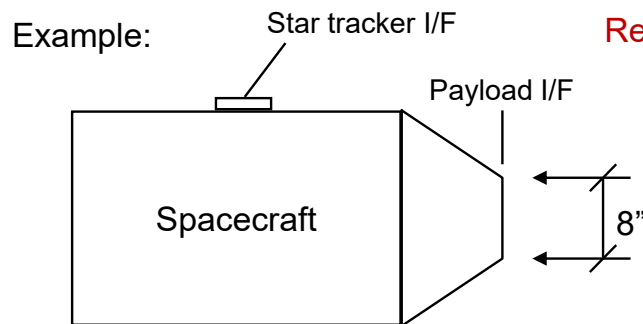
Don't specify instrumentation for a test until you've

(a) defined the objectives and needed accuracy and

(b) researched the capabilities and limitations of available instruments and data-acquisition systems with the aid of an expert.

Effective Instrumentation when Alignment Is Critical

Identify the deformation in your structure that is considered detrimental, and then make sure you have the instrumentation needed to detect it—or do a post-test system performance test that can detect detrimental deformation. The pass/fail criteria for the test should be based on a meaningful assessment.



Requirement: The structure shall withstand launch environments without suffering permanent change in alignment of the payload interface relative to the star-tracker interface greater than ± 1 arc minute.

- Plan:
- Apply loads to the payload interface
 - Measure rotation of the payload interface with two displacement gages 8" apart

Convert the requirement to residual displacements at the gage locations:

$$1 \text{ minute} \left(\frac{1 \text{ degree}}{60 \text{ minutes}} \right) \left(\frac{2\pi \text{ radians}}{360 \text{ degrees}} \right) = 0.00029 \text{ radians}$$

$$\text{Displacement} = 0.00029 \text{ radians} (8"/2) = 0.0012"$$

Pass/fail criterion

Thus, the gages and the data-acquisition system must have accuracy and resolution of 0.0001" (10% of the pass/fail value) or better for this plan to be valid.

Post-test optical measurements or performance testing may be necessary.

The Difference between Accuracy and Resolution

Resolution: the smallest change that can be measured.

- The term “resolution” also is used in reference to the frequency increments for data processing in a random vibration test

For an instrument such as a strain gage, you can get better resolution by increasing the gain (ratio of output voltage to input voltage).

Accuracy: how close a measurement is to the true value

- Affected by calibration and electrical noise

Dependable measurements require sufficient resolution and accuracy.

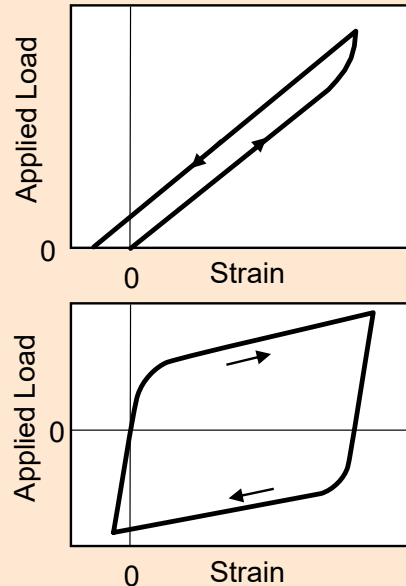
Strain Gages (aka Gauges): Use them Intelligently, and Know What You're Looking For

A **strain gage** is a small device whose electrical resistance varies linearly with length change.

Measures local material strain, which can be hard to interpret.

- Local yielding is not considered failure unless it is detrimental (e.g., deformation).
- For ductile metals, we have allowable stresses, not allowable strains, so, as an indicator of impending rupture, strain is more useful for brittle materials than ductile materials.

What do these strain plots tell you?



Good uses of strain gages:

- to determine internal load distribution
 - especially when calibrated (load-strain relationship)
 - when the gage is away from discontinuities
- to recognize and prevent potential failure
 - when we can relate strain to failure
- to detect anomalous behavior

Questionable uses of strain gages:

- to identify local yielding near discontinuities
- to use as pass/fail criteria for the test

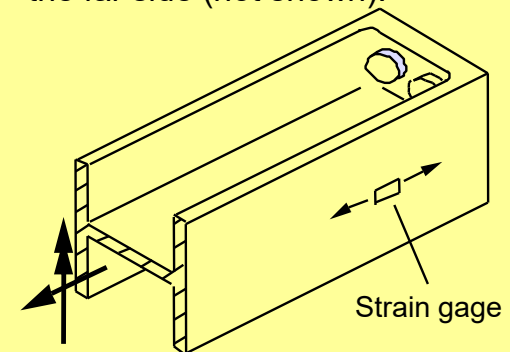
Displacement is often a better indicator of impending structural failure than strain.

Guidelines for Where to Put Strain Gages

- Be sure to think through your objectives:
 - For detecting impending rupture, put the gage in a location that will give you information you know how to interpret.
 - Usually not in a region affected by stress concentrations if the material is ductile
 - For determining load distribution,
 - Again keep the gage away from stress concentrations (e.g., joints)—put it in a location with a predictable relationship between load and strain.
 - Calibrate the gage with known load if practical.
- Use a rosette (measures strain in 3 directions) if you want to know the full state of plane stress.
- Stay away from areas significantly affected by friction.
- If strain data is critical and the gage is hard to access during test, install a redundant gage for backup (same for critical accelerometers and displacement gages).

Example:

If we want to know axial force and bending moment in this beam, we could position a single-axis strain gage as shown along with a second gage in the same location on the far side (not shown).



Nonlinear strain readings may indicate onset of yielding in the bolted joint.

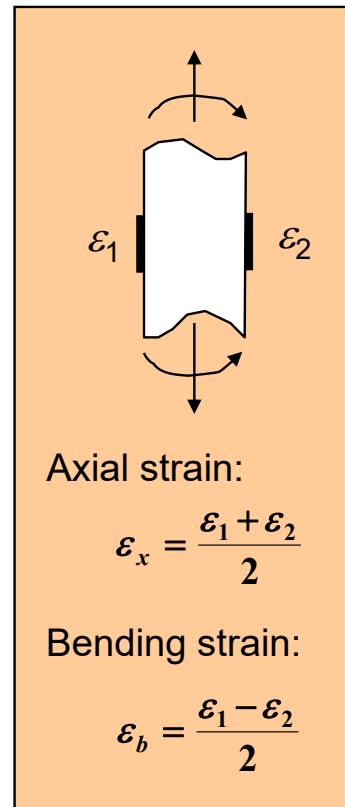
Strain gages can be used for dynamic (e.g., vibration) tests as well as static tests.

Bridging Strain Gages

Sometimes it's best to bridge two gages into a single channel.

- to measure axial strain only
- to measure bending strain only

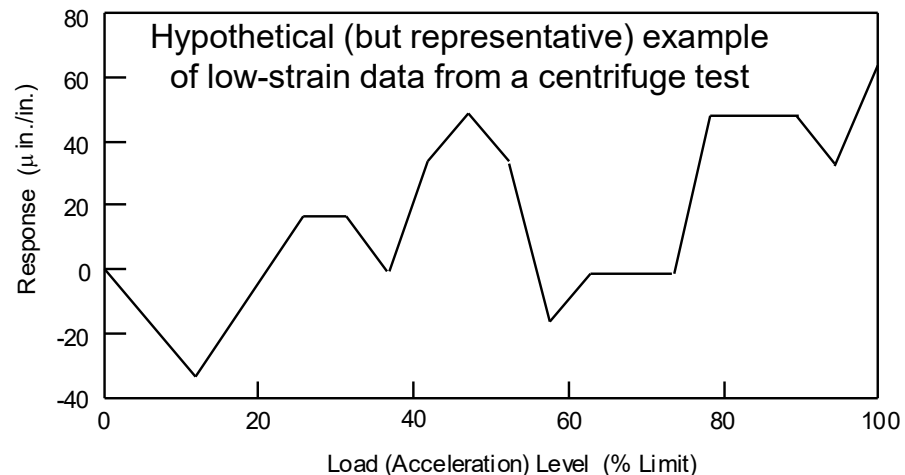
Giving such requirements to the instrumentation expert reduces the amount of post-processing.



The electrical bridges effectively do this to the data, so all you see are ϵ_x and ϵ_b .

Things that Can Affect Accuracy of a Strain Measurement

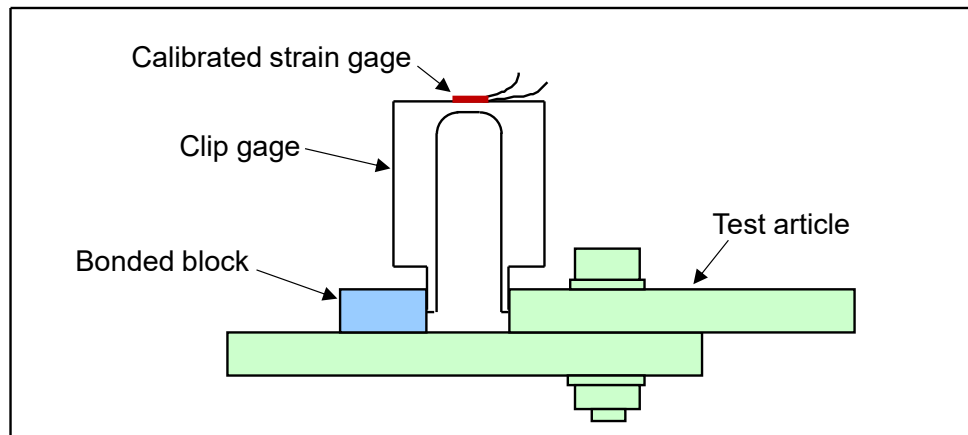
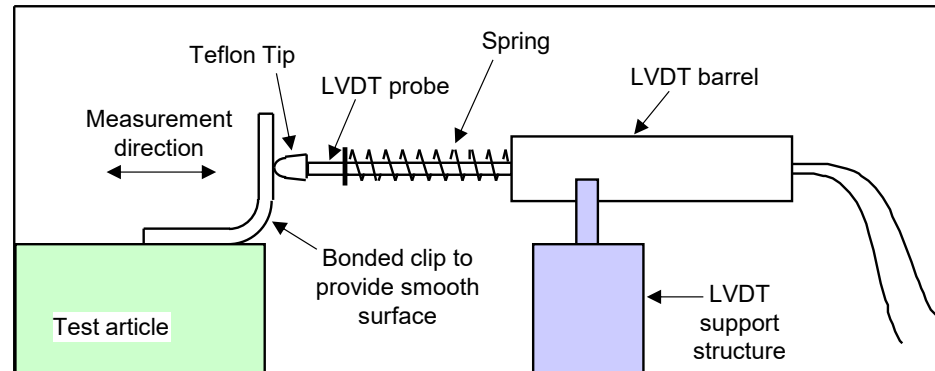
- Random **electrical noise** (interference in the electrical signal)
 - To reduce the noise, use low-resistance, shielded cables.
 - To filter the noise, computer-based acquisition systems calculate an average of many electrical signals, but some noise will remain.
 - Data can be particularly noisy when using spinning electrical contacts (slip rings), such as in a centrifuge test.
- Temperature changes
 - Magnified by intermittent sunlight, heating, or air conditioning
 - Match the coefficient of thermal expansion for the gage with that for the metal it will be bonded to.
- Manufacturing tolerance on part thickness can cause prediction error.
 - Calibrate the gage, if you can.
 - Or measure thickness.



Displacement Gages

The most common type is the **linear variable differential transformer (LVDT)**.

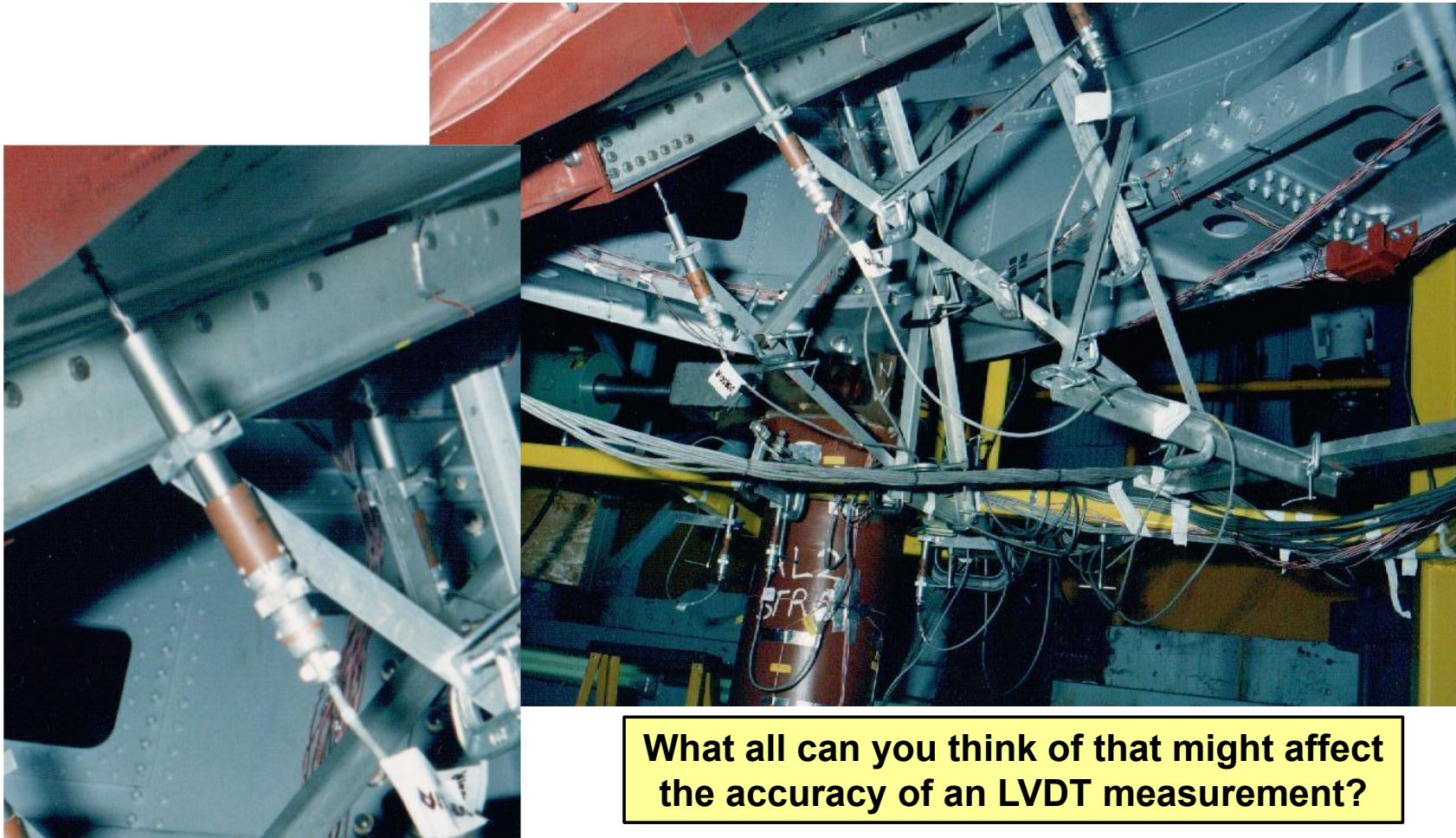
- As the probe extends or retracts, output voltage changes.
- This relationship is linear within the range of the instrument.
- Teflon tip used to reduce friction from lateral motion.



Other types:

- Clip gages: greater accuracy for small deflection (often custom made and calibrated)
- Potentiometer-based displacement transducers: rotary types referred to as “string pots”—can be used for large deflections

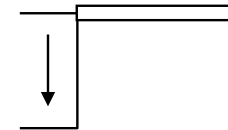
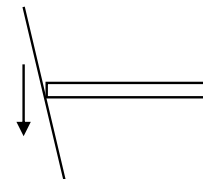
Example LVDT Arrangement



What all can you think of that might affect the accuracy of an LVDT measurement?

Many Things Affect LVDT Accuracy

- LVDT size and calibration range combined with resolution of acquisition system
 - Smaller provides more accuracy but can be harder to set up.
- Stiffness of support structure
 - If the structure supporting the LVDT deflects under compression of the probe spring, the LVDT will measure less deflection than there is.
- Perpendicularity of probe to measurement surface
 - If the probe is not perpendicular, lateral motion will cause it to move in and out.
- Alignment of LVDT with axis of interest
- Surface roughness
 - Measurement affected by lateral motion
- Distance from probe tip to edge of measurement surface
 - Another concern associated with lateral motion: The measurement surface may slide out from under the probe.



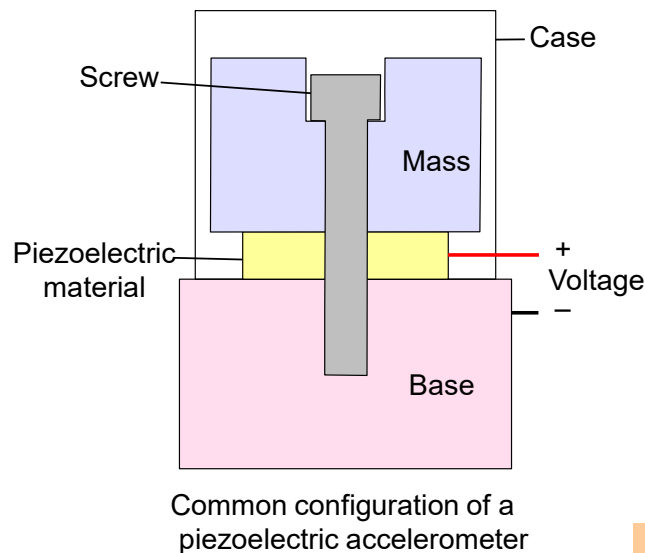
**Before the test, check each instrument to ensure you'll get good data.
And then check again during and after the test.**

Accelerometers

An **accelerometer** is an instrument that measures acceleration indirectly from deformation, which in turn relates to inertia force that resists acceleration.

For vibration testing at high frequencies (> 500 Hz), **piezoelectric accelerometers** are more accurate than other types and are thus most commonly used.

- They incorporate materials that produce a voltage when they deform dynamically (**piezoelectric materials**)—not useful for measuring steady-state acceleration



Kistler triaxial accelerometer
32 grams (1.1 ounce)



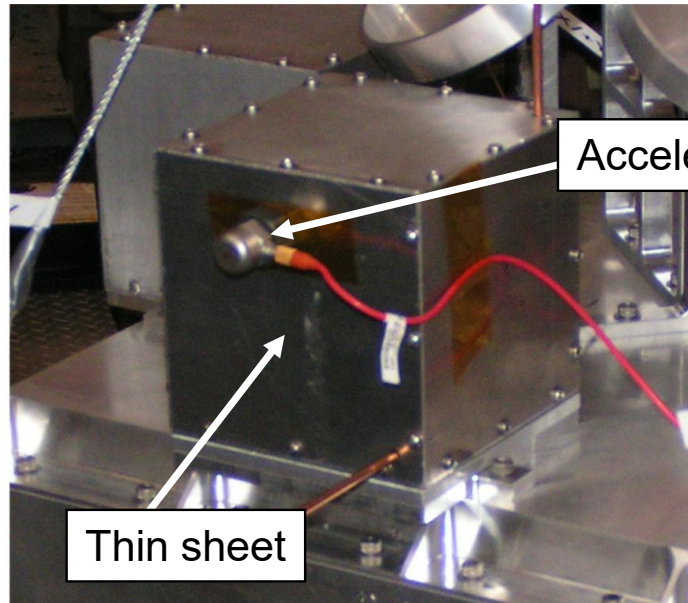
Brüel & Kjaer teardrop
(single axis)
accelerometer, 1 gram
(0.035 ounce)

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

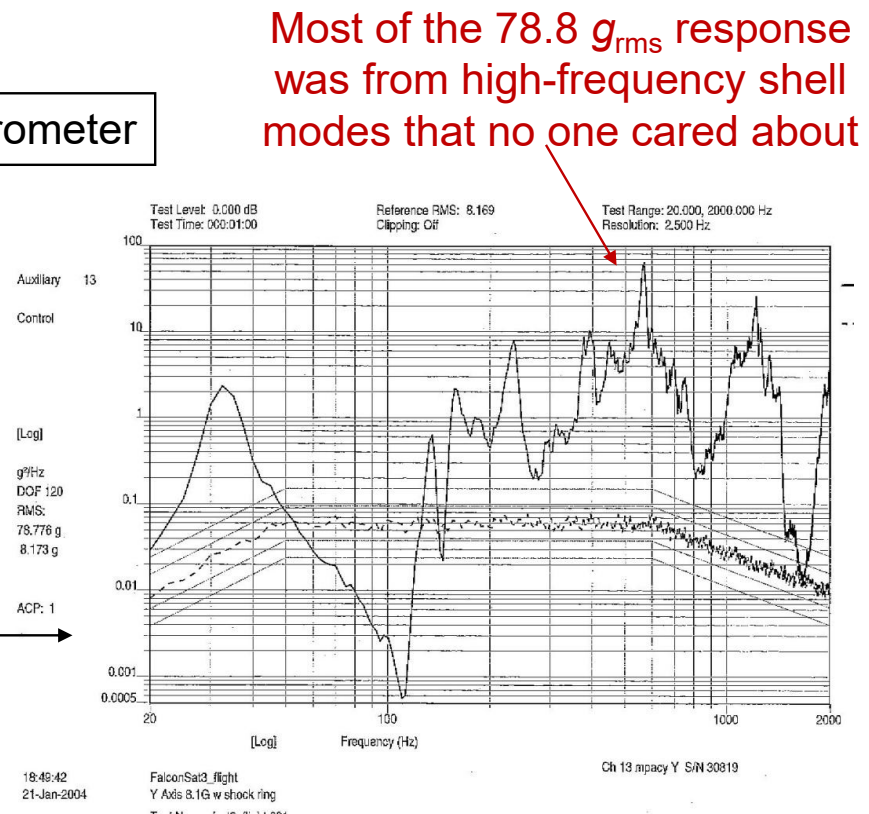
Guidelines for Where To Put Accelerometers

- 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.
- 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.
- For convenience, align the accelerometer's axes with the loading directions or the axes of the test article's coordinate system.

Example of How Local Shell Modes Contaminate Data



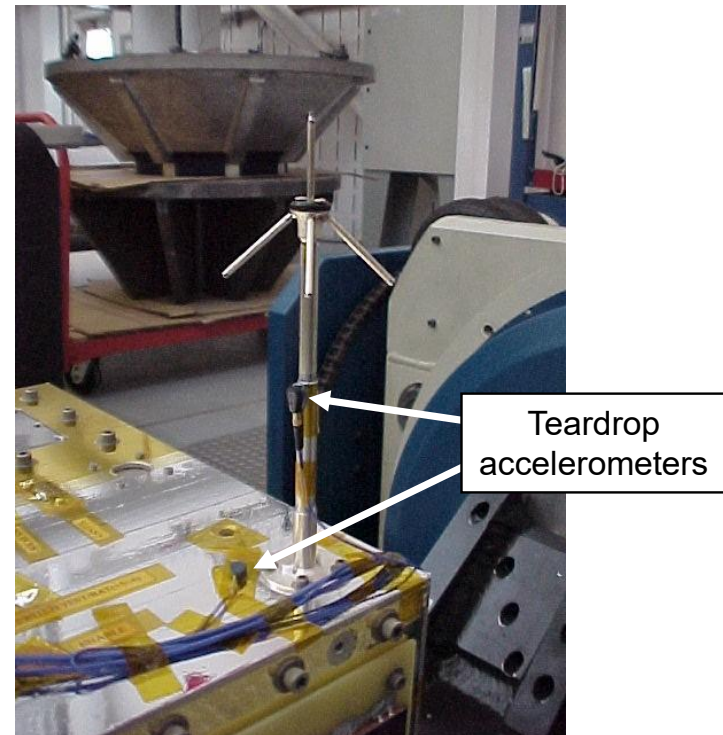
Response data from
random vibration test



Don't put accelerometers on thin sheets!

Example: Accelerometer Use

- During random vibration testing, this small antenna suffered a fatigue crack in the soldered joint between nested tubes.
- The small accelerometers shown were used to
 - determine the antenna's fundamental frequency
 - and characterize the base environment so that a new antenna could be designed and then tested separately.
- The base accelerometer shown at right was for acceleration normal to the surface of the satellite's top plate.
 - Lateral acceleration also was measured (not shown).
- However, to devise the proper test for the redesigned antenna, it also was important to understand angular acceleration at the base.
 - Instead of measuring angular acceleration, which is more challenging, a finite-element model was used to help design the subsequent test.



Small (5-ounce) antenna on top of FalconSAT-2
(courtesy U.S. Air Force Academy)

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,

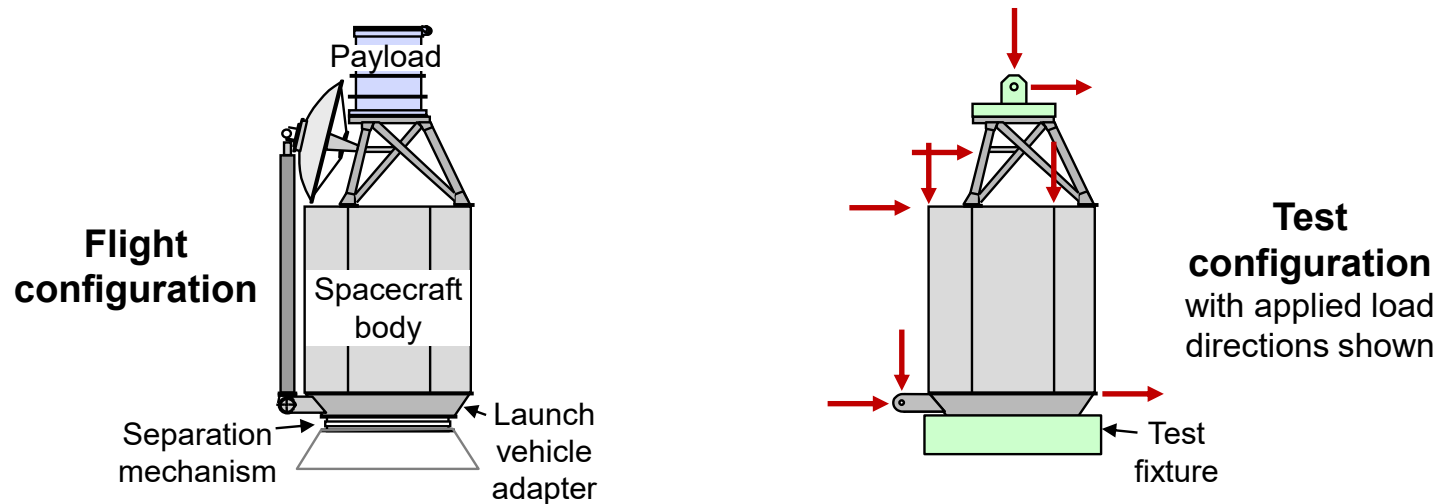
- **Insist that your customer approve the plan.**
- **For a complex test, plan to conduct a test design review.**

Hypothetical Example: FireSat Static Loads Test

The company under contract to develop and produce three FireSat spacecraft of identical design is planning to verify strength of the primary structure by doing a static loads test on the first-built flight article.

Design factors of safety: 1.1 yield and 1.25 ultimate

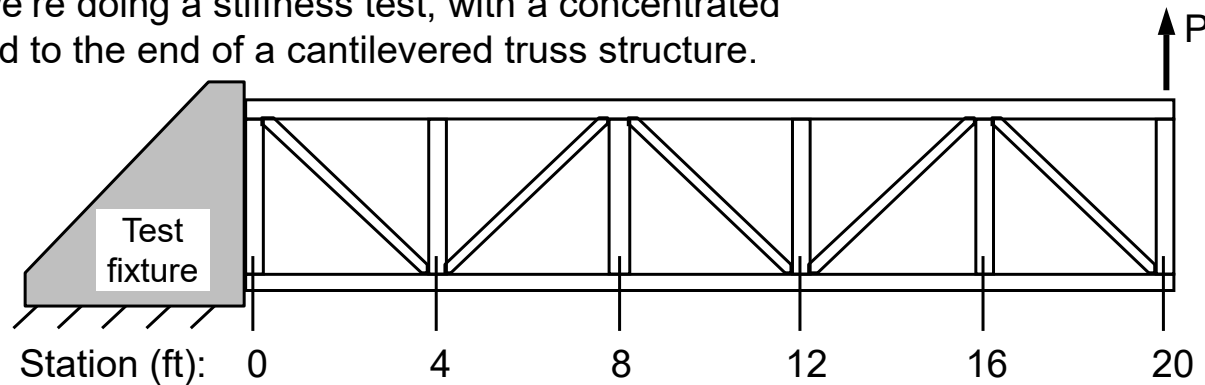
Pass/fail criteria for the test: No detrimental permanent deformation at 1.1 times limit load
No rupture or collapse at 1.25 times limit load



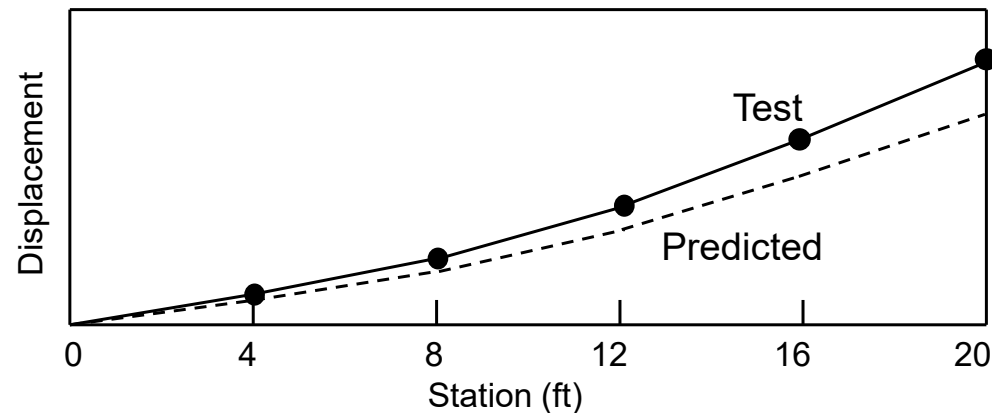
**You are the customer preparing to attend the test design review.
What do you want to know?**

How Would You Interpret This Data?

Let's say we're doing a stiffness test, with a concentrated load applied to the end of a cantilevered truss structure.

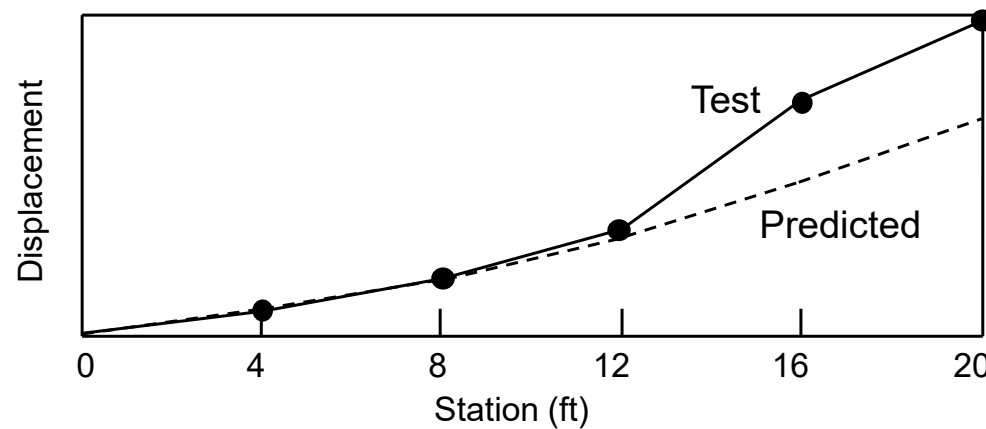
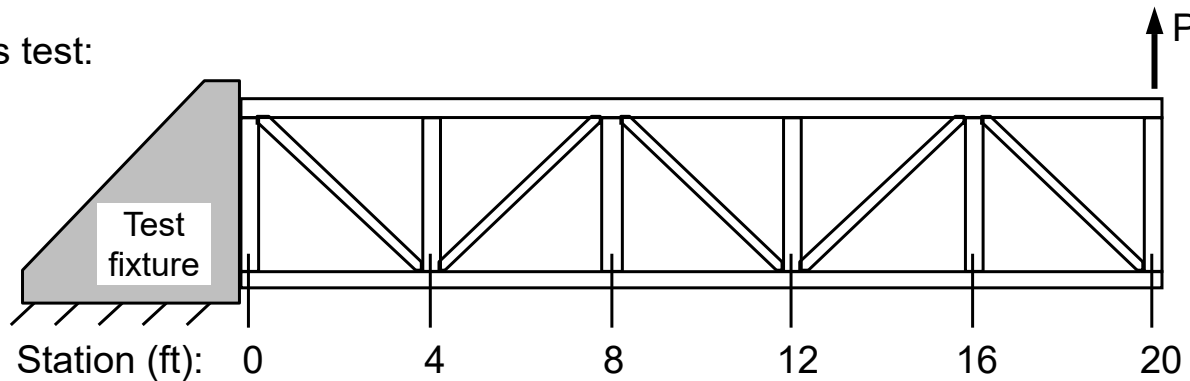


We measure displacement at every frame location and plot it vs. station (distance from mounting interface).



What Does This Data Tell You?

Stiffness test:



Make Sure the Test Data are Reasonable and Investigate Anomalies before Dismantling the Test Setup!

Possible causes of bad data:

- Misaligned or faulty instruments
- Displacement gages mounted on flexible support structure
- Inadequately bonded strain gages
- Accelerometers mounted on locally flexible structure
- Improper calibration factors used to convert electrical output to meaningful data
- Incorrect applied loads or environments because of improper computer input or calibration factors
- Improper data processing
- Unknown effects of test fixtures

Make sure instrumentation is verified prior to test (e.g., use a gage block to check displacement gages, use a calibrated resistor for checking load cells, etc.), but recognize that such verification is not foolproof.

If you are the responsible engineer for a test whose success depends on acquired data, it is crucial for you to ensure you have accurate data and have adequately investigated any anomalies before allowing the test setup to be torn down!

Come prepared to interpret data within minutes of seeing it!

Key Points of This Section

- Design a test as you would design any other product:
 - systematically, starting with objectives
 - by thinking ahead
- Plan the test thoroughly and communicate it with a written test plan.
- Make sure test fixtures don't over-constrain the test article.
- Establish criteria for recognizing whether a planned test will be adequate.
 - Show the test loads will be sufficient with an apples-to-apples comparison.
 - When doing an acceptance or proof test, plan to test each axis, each member, each joint, and each potted insert, as applicable.
- Establish pass/fail or success criteria related to objectives and the requirements being verified.
- Design effective instrumentation to satisfy objectives and enable assessment of pass/fail criteria.
- Come prepared to interpret data and make decisions quickly.
- Elevate any violations of pass/fail or success criteria to management and customers, and recommend a course of action.

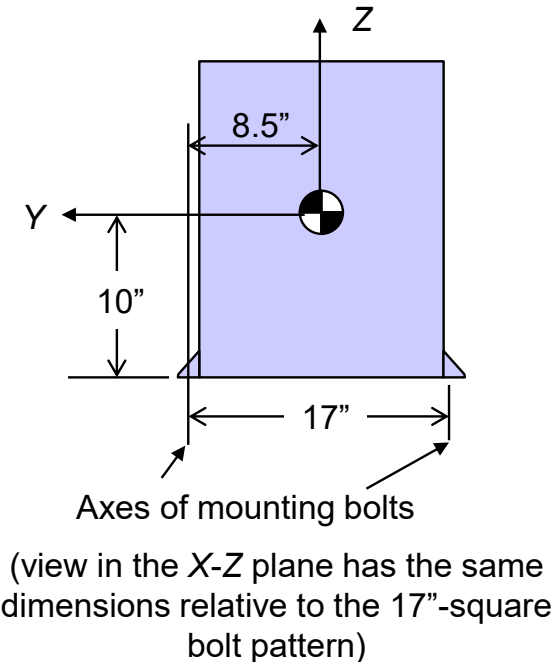
Appendix to Sec. 2

Example 2-d: A Variation of Class Problem 2-1

Appendix to Sec. 2: Example 2-d

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 2-1 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.

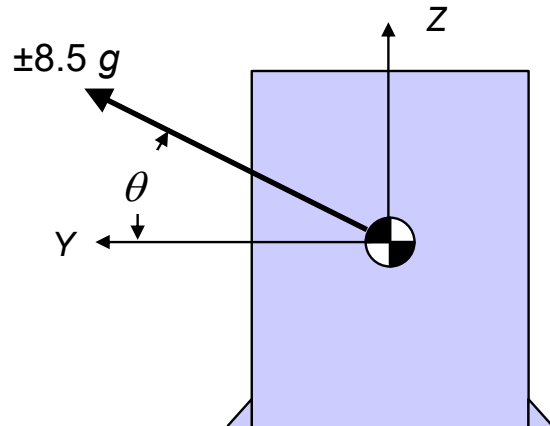
continued

Appendix to Sec. 2: Example 2-d, 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. 2: Example 2-d, conclusion

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

Appendix to Sec. 2: A Shortcut for Example 2-d

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 2-b: 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 2-h,

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

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

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

Test to $1.25 \times 20.52 + 1 = 26.7$ 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

3. Loads Testing of Small Specimens

- Applications and Objectives
- Common Loading Systems
- Test Standards
- Case History: Designing a Test to Substantiate NASA-STD-5020 Criteria for Bolt Analysis

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What Are Some Likely Reasons for Loads Testing of Small Specimens?



Typical Applications and Objectives

I ➡

- Support design
 - Establish allowable stresses for new materials and allowable loads for new joints
 - Provide fatigue data (S/N curves)
 - Provide other needed mechanical properties (e.g., elastic modulus or coefficient of thermal expansion)
 - Derive empirical methods of structural analysis
 - Identify features that would improve the design

I ➡

- Support manufacturing process development
 - Strength testing of bonded or welded joints to confirm the manufacturing process can achieve the needed properties or to support statistical process control

V ➡

- Ensure quality of procured materials and fasteners or fabricated parts
 - Test a randomly selected sample or “tag ends” (trimmed excess from a panel)

I ➡

- Identify causes of failures

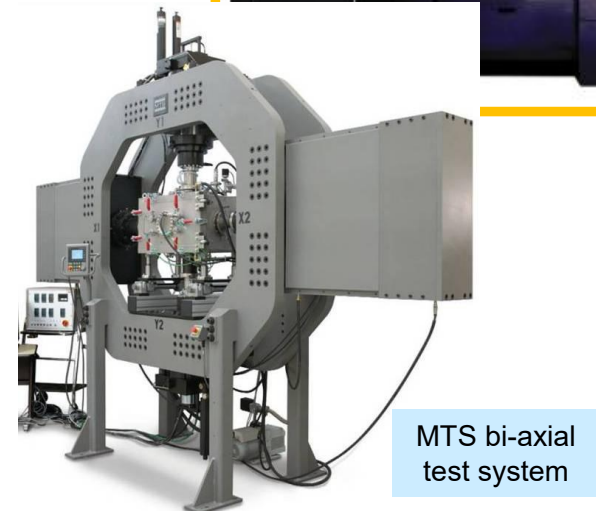
V ➡

- Proof test flight hardware
 - Subassemblies with bonded joints, prior to integration with a larger assembly

— I = obtain information; V = verify compliance

Common Loading Systems for Small Test Items

- Most load systems are single-axis machines.
 - Electro-mechanical (single or double ball screw)—preferred for strength testing
 - Servo-hydraulic (hydraulic cylinder)—more practical for fatigue testing
 - Servo-electric (smaller systems with lower load capacity)
- Bi-axial test machines are sometimes used.
- With an electro-mechanical test machine, load is applied by controlling displacement.
 - This is why stress-strain curves are plotted with strain on the horizontal axis.
 - Rates are adjustable.
- Testing of large structures normally requires custom test configuration, with load control rather than displacement control. (see Sec. 4)



Why is it important to standardize test methods for materials, fasteners, and joints?

Answer: To ensure that ...

- the results are meaningful
- two organizations testing the same thing will get the same results
 - other than from inherent variation of material properties (random defects)
- users (e.g., analysts) know how to apply the test results
- there is no unnecessary expense (re-engineering)

Standards Organizations Related to Structural and Materials Testing

- ASTM International (originally American Society for Testing and Materials)
 - Maintains over 12,000 standards
- AIA – Aerospace Industries Association
 - e.g., NASM1312-8, Fastener Test Methods – Tensile Strength
- SAE International – Society of Automotive Engineers, Aerospace
 - e.g., USCAR10-1, Torque-Tension Testing of Threaded Fasteners
- ASME – American Society of Mechanical Engineers
 - e.g., BPVC-IIID - 2007 BPVC Section II-Materials-Part D-Properties-(Customary)
- ANSI – American National Standards Institute
 - e.g., ANSI N14.5-1997, Radioactive Materials - Leakage Tests on Packages for Shipment
- AWS – American Welding Society
 - e.g., B4.0 2007, Standard Methods for Mechanical Testing of Welds
- ISO – International Organization for Standardization
 - e.g., ISO 3800, Threaded Fasteners—Axial Load Fatigue Testing
- IFI – Industrial Fastener Institute
 - e.g., IFI-135, Mechanical Testing of Blind rivets

Test Standards: Examples

- Tension, ASTM E 8 (Ref. 2)
- Compression, ASTM E 9 (Ref. 3)
- Fracture toughness, ASTM E-399 (Ref. 4)



**Still, it's not practical to standardize every test that may be needed.
Sometimes there is a need to design a unique test.**

Case History: Designing a Test—Background and Problem Statement

NASA's standard (1989 to 2012) for assessing strength of preloaded bolts, NSTS-08307, was not substantiated by test and was thought by many to be overly complex and penalizing, driving cost and weight unnecessarily, requiring ...

- The ultimate margin of safety for bolt tension to include the effects of preload
- A positive margin of safety on bolt yield
- Analysis for shear loading to include interaction with preload

NASA' Engineering and Safety Center (NESC) tasked a team in 2007 to develop a new standard for threaded fastening systems (for many reasons in addition to the above).

- The team consisted of civil servants, NASA contractors, researchers, and consultants.

The NESC team decided to design a test program to answer the following questions for joints using A-286 fasteners and aluminum fittings (joint members):

- Can preload reduce the ultimate tensile or shear strength of a bolted joint? Or can we ignore preload in ultimate-strength analysis?
- Is bolt yielding detrimental for a typical joint?

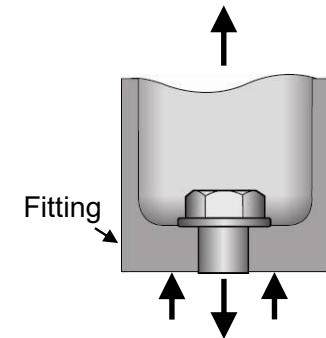
There were many other objectives of this project. The case history presented here addresses only the above questions and includes only the design of the tensile test.

The product, NASA-STD-5020 (Ref. 7), was approved and released in March 2012.

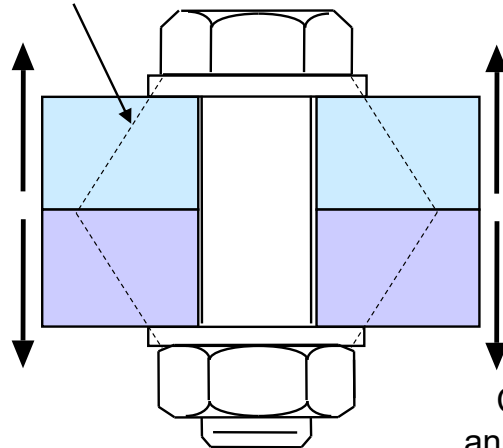
Background Info: How a Preloaded Joint Carries Applied Tensile Load

Applied tensile load is carried across a preloaded joint (prior to gapping) two ways: (1) additional load in the bolt, on top of the preload, and (2) relieved compression at the mating surface between the parts.

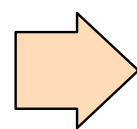
The division of load depends on bolt stiffness (K_b), clamp stiffness (K_c), and the load-introduction factor (n).



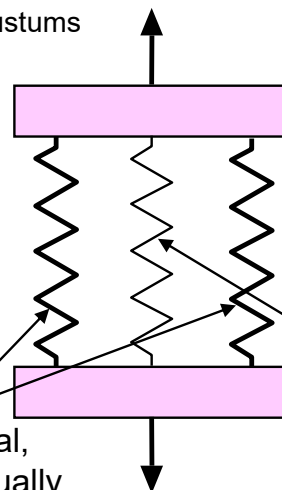
K_c commonly estimated as the stiffness of back-to-back frustums



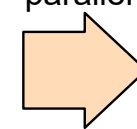
idealized as



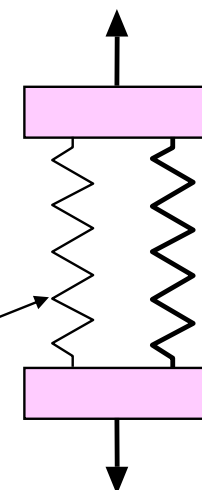
Clamped material, annular spring (usually stiffer than the bolt)



more simply, two springs in parallel

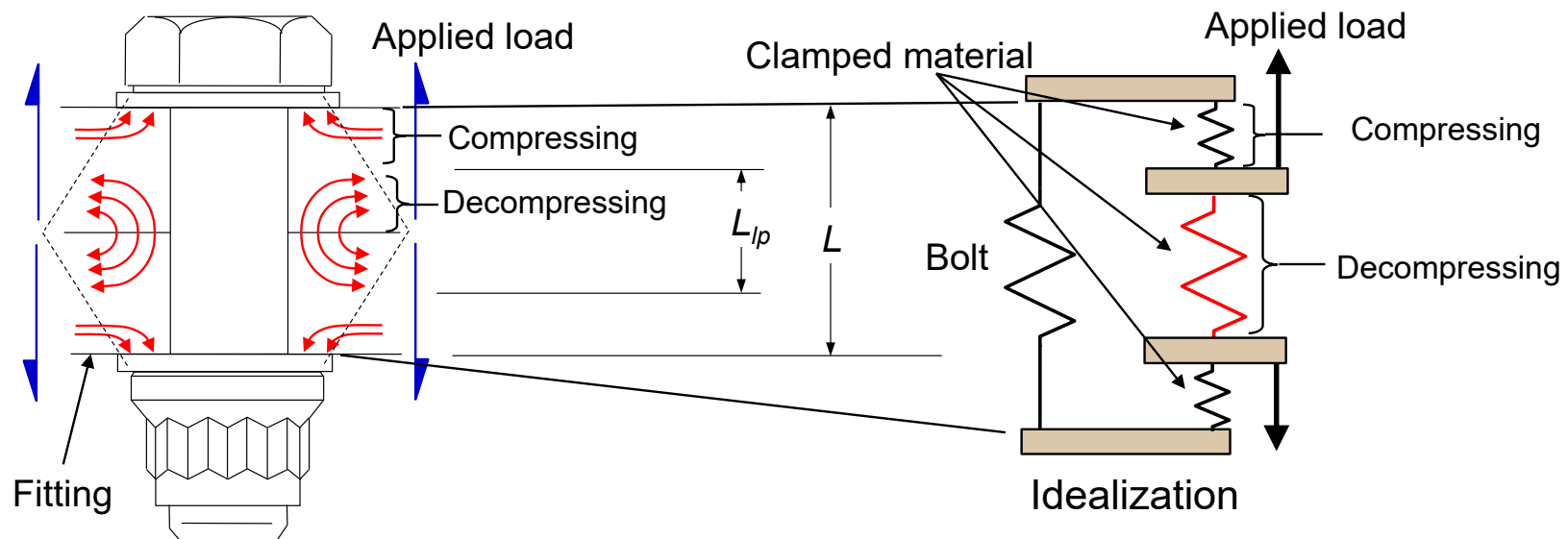


Bolt



From my course "Design and Analysis of Bolted Joints" (DABJ)

The Load-Introduction Factor Adjusts the Model to Account for the Actual Mechanics



From my DABJ course

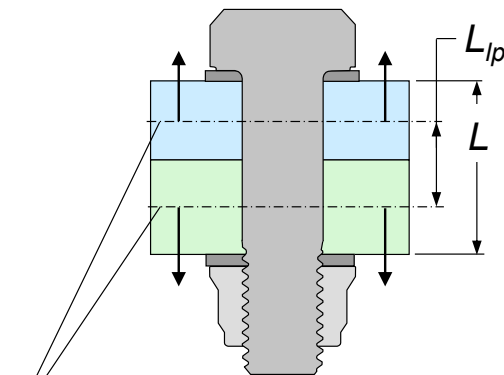
Total bolt load (linear approximation, prior to gapping):

$$P_{tb} = P_p + n\phi P_t$$

P_p ← Preload
 n ← Load-introduction factor ($n = L_{ip}/L = 0.5$ for most joints)
 ϕ ← $\phi = \frac{K_b}{K_b + K_c}$
 P_t ← applied load
 K_b ← Bolt stiffness
 K_c ← Stiffness of clamped material

n is increased by the presence of a spacer, such as a shim, between load-bearing parts

Estimating the Load-Introduction Factor

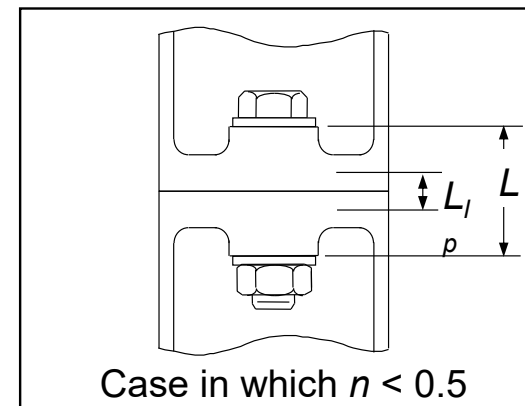
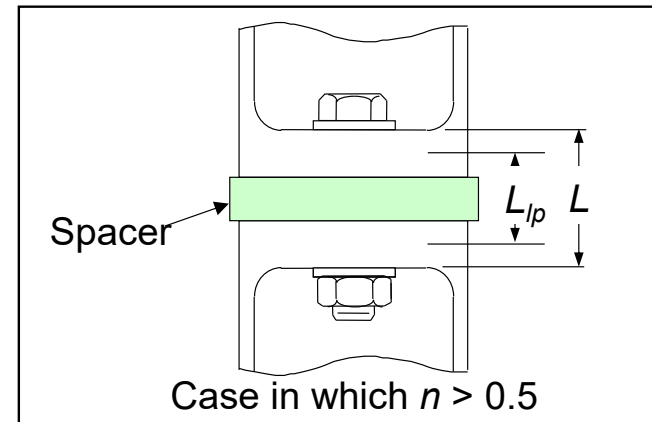
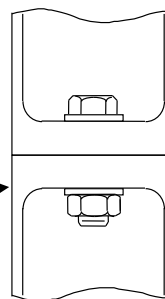


Adapted from
NASA-STD-5020B,
Fig. 3

Planes at which load is introduced
(assumed to be half way through pad
thicknesses of load-carrying parts)

Load-introduction factor, $n = \frac{L_{lp}}{L}$

For a typical joint without
spacers, $n = 0.5$



For a joint under applied tensile loading, a higher load-introduction factor leads to higher total load in the bolt prior to gapping.

Case History, continued: Tensile-Test Objectives

1. Acquire data needed to improve understanding of how preload affects the strength of bolted joints, particularly those using A-286 bolts and aluminum fittings.
 - Test out the theory that a ductile bolt will yield before rupture and thus cause a preloaded joint (with aluminum joint members) to separate prior to rupture.
 - If so, the only load in the bolt at rupture is the applied load, so there is no need to include preload in ultimate-strength analysis for tensile loading.
2. Acquire data to help determine whether or when bolt yielding is detrimental.
 - Test the theory that, if a preloaded bolt yields under an applied tensile load that does not cause the joint to gap, and the applied load is then removed, the residual preload will be less than it was originally, but ...
 - the preload loss will be a relatively low percentage, with no impact for most joints,
 - and, because of strain hardening, subsequent applied loads of the same magnitude will not cause additional loss in preload.
 - Determine whether yielding prevents bolt removal or affects the torque-preload relationship during reinstallation.

Case History: Constraints

- There is no standard test that would meet the objectives.
- The NESC team was severely constrained by budget, schedule, and fastener availability.
- The team was able to find two suitable types of fasteners in NASA storage (quantity: about 30 of each type, half of which were allocated for shear-joint testing):
 - #10 (0.190"-dia), 160-ksi A-286 (allowable tensile yield strength of 120 ksi), of mixed lots (no traceability); length = 1.75"
 - Specified minimum ultimate tensile strength (allowable load), $P_{tu\text{-allow}} = 3200 \text{ lb}$
 - Calculated allowable tensile yield load, $P_{ty\text{-allow}} = (120/160)(3200) = 2400 \text{ lb}$
 - 3/8"-dia, 180-ksi A-286 (minimum tensile yield strength of 150 ksi), of a single traceable lot; length = 2.328"
 - $P_{tu\text{-allow}} = 17,100 \text{ lb}$
 - $P_{ty\text{-allow}} = (150/180)(17,100) = 14,300 \text{ lb}$
 - Washers and strength-compatible nuts also were available.

Class Exercise

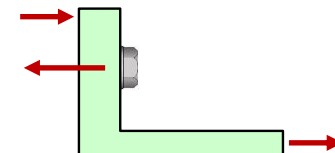
Brainstorm ideas for a test that meets the objective stated below, given the noted constraints. Consider ...

- **Configuration**
- **Instrumentation**
- **Test procedure**

Objective: Test out the theory that a joint with aluminum joint members and an A-286 bolt, with no nonmetallic materials between the joint members, would gap under applied tensile load before the bolt breaks.

- If this theory is right, then preload does not reduce the applied load that such a joint can carry, which means the margin of safety for ultimate tensile strength should be based on applied load only.

Strategy: Design a joint (with a single bolt) that would cause the bolt, prior to joint gapping, to see a higher-than-normal percentage of applied tensile load on top of preload. Don't design a joint that introduces prying, as shown at right, because that would introduce another variable (effects of prying).



Don't design this type of joint

Intentionally left blank

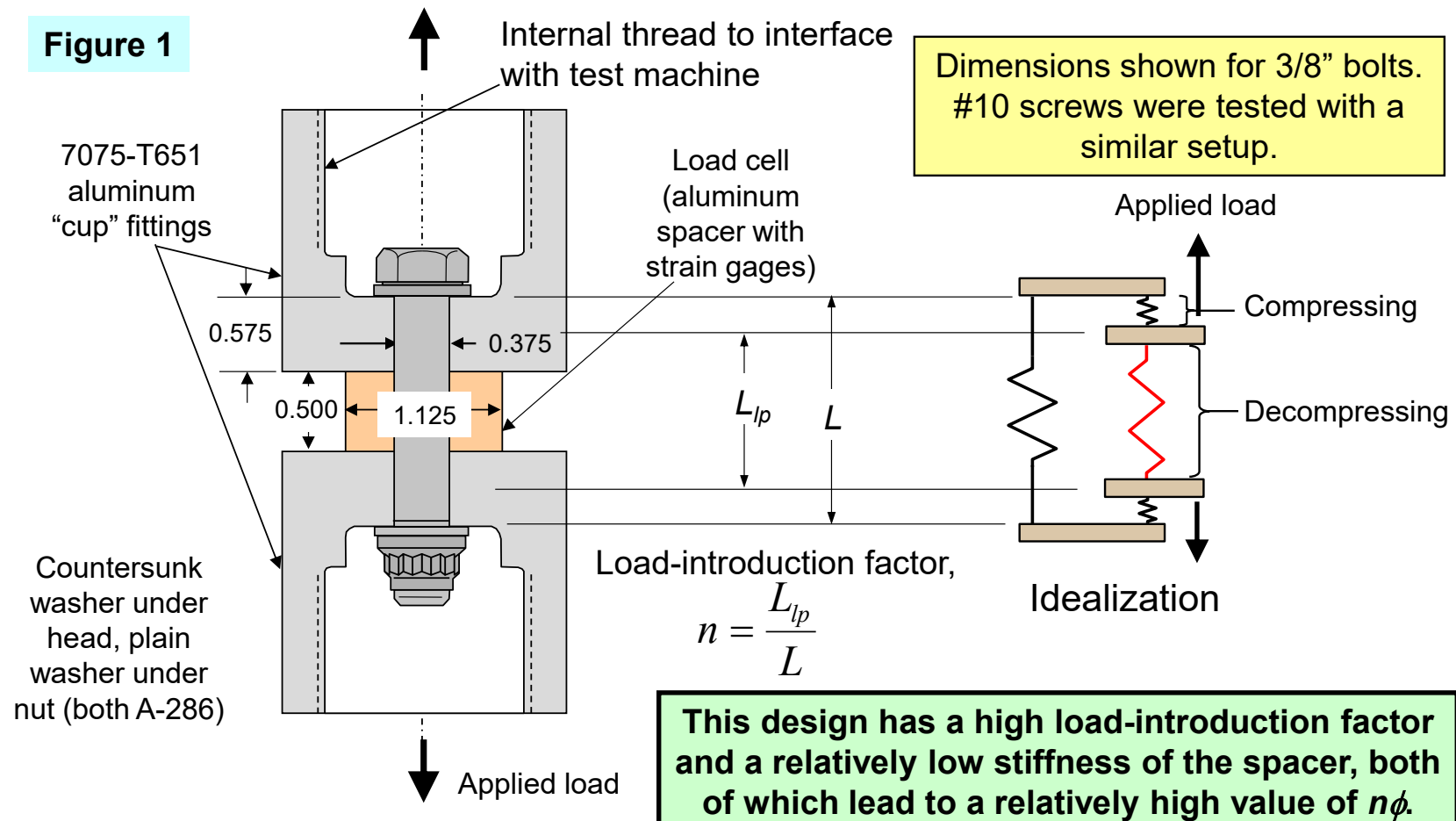
Case History: Actual Test Concept

- A-286 bolts
- Aluminum fittings
- Two sizes and types of bolts tested:
 - #10 (0.190" dia) (NAS1351N3-28, 160-ksi A-286)
 - 3/8" dia (NAS1956C28, 180-ksi A-286)
- Tested to rupture
 - 5 with no preload
 - 5 with high preload (at or above yield)
- Objective: A test configuration in which, when the joint is preloaded, the bolt load increases by a high percentage of applied load
 - So that conclusions can be safely applied to other joint designs using these materials
- Meeting this objective requires a joint in which clamp stiffness (K_c) is relatively low and the load-introduction factor (n) is high.
 - Without using non-metallic clamped materials

Test documented
in Ref. 8

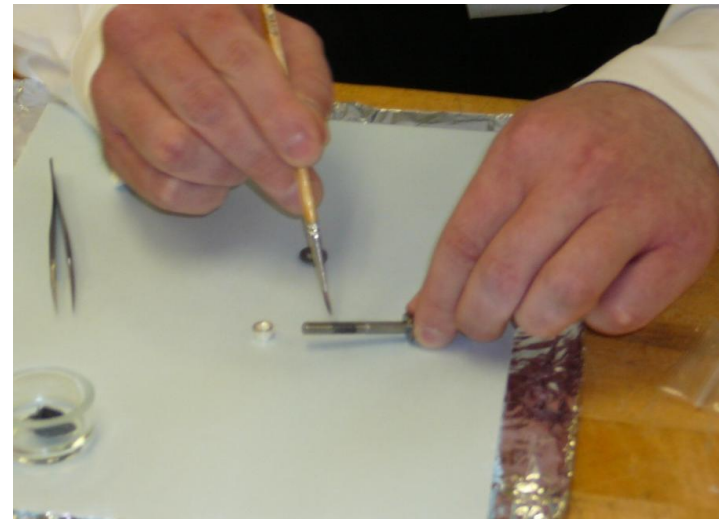
Case History: Test Fixtures and Assembled Configuration

Figure 1



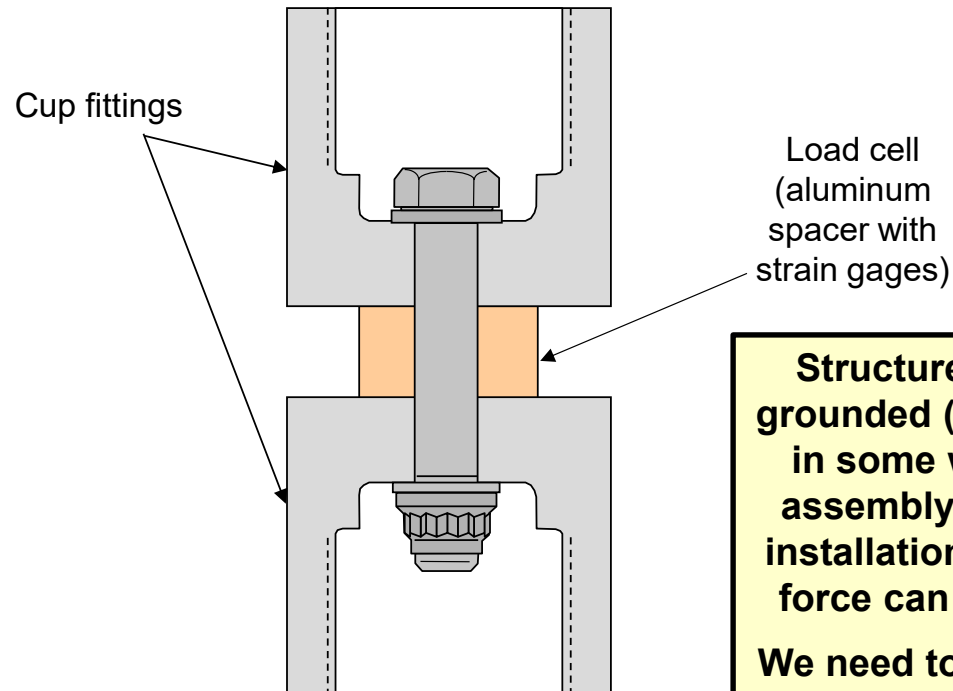
Case History: Assembly Process

- Prior to each assembly in the configuration shown in Fig. 1, the fasteners were lubricated
 - to reduce friction (thus increasing the achievable preload)
 - and to reduce variation in preload for given applied torque
- Lubrication process:
 - Lubricant: Braycote 602 grease
 - Application: using small brush
 - Surfaces lubricated:
 - Bolt threads
 - Both sides of washer under nut
- Torque applied to nut



Assembly

Given the assembly shown below, how would you torque the nut against the bolt head with hand wrenches?



Structures must be grounded (constrained) in some way during assembly so that the installation torque and force can be reacted. We need to ground this joint as well.

Assembly Required a Vice



Torquing a #10 fastener.

For the 3/8" bolts, we had to anchor
the base of the vise.

Fortunately, we thought through the
assembly process ahead of time
and designed the cup fittings with
flats to interface with a vice.



Case History: Test Matrix

Test number	1 - 5	6 - 10	11 - 15	16 - 20
Bolt diameter (in)	0.190	0.190	0.375	0.375
Preload	0	100% yield	0	100% yield
Quantity	5	5	5	5

Success criterion:

The test is considered successful if all needed data are obtained.

The 3/8" bolts were all from the same procurement lot, whereas the #10s were from NASA Goddard's inventory, without lot traceability. (Test results indicated multiple lots for #10s because of distinct groupings in strength, whereas the 3/8" bolts had little variation in strength.)

In addition, the #10s were tested with low-strength washers that yielded during test, whereas the 3/8" bolts were tested with high-strength A-286 washers.

As a result, the test results with the #10s were more varied and less dependable, and the test team gave more credence to the results of testing the 3/8" bolts.

Case History—General Procedure: Tests 1 – 5 (#10) and 11 – 15 (3/8)

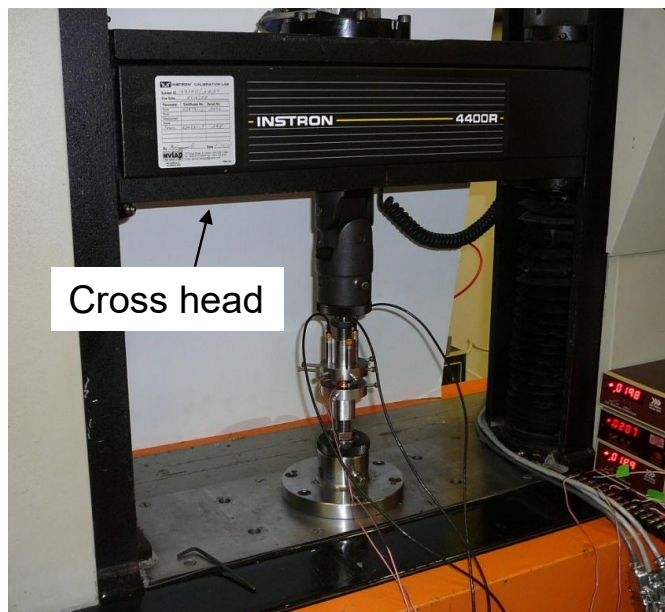
1. Pull each of 5 bolts to failure, plotting load vs. displacement as measured by the test machine.
2. Determine the bolt's yield strength from the load-displacement plot using the Johnson's 2/3 slope method*, per NASM1312-8.
 - *This step was dropped for testing the 3/8" bolts after recognizing during testing of the #10s that determining the yield strength was not needed to meet the test objectives.*
3. Compute the sample mean ultimate strength for tests 1 – 5 and 11 - 15.
 - *Results:*
 - *#10 screws: 3690 lb (vs. 3200 lb specified minimum strength)*
 - *3/8" bolts: 19,800 lb (vs. 17,100 lb specified minimum strength)*

*See backup charts at the end of this section.

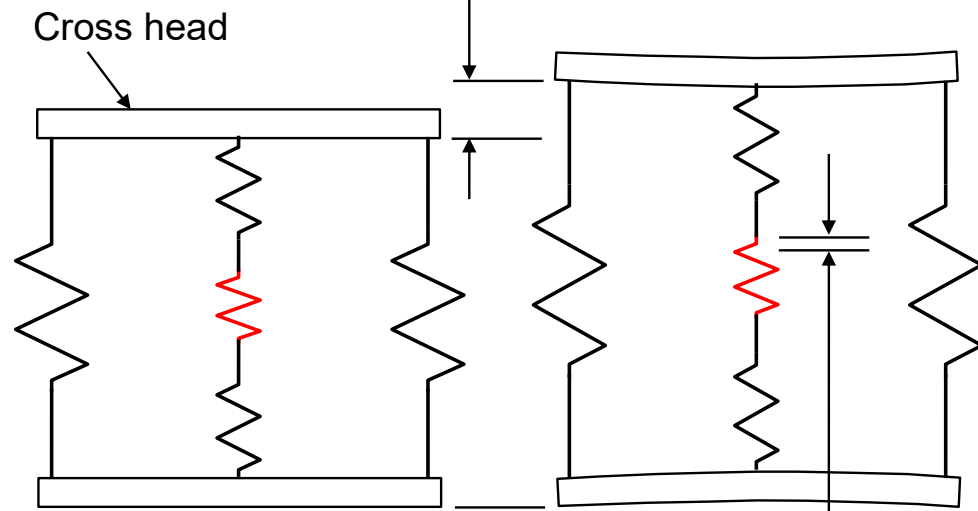
Case History: Measuring Displacement

As noted earlier, a mechanical test machine controls displacement and measures load, thus allowing more meaningful plots of load vs. displacement.

However, the displacement that you see as “output” from the test machine is the total displacement of the cross head relative to the base.



This is what the machine controls and records

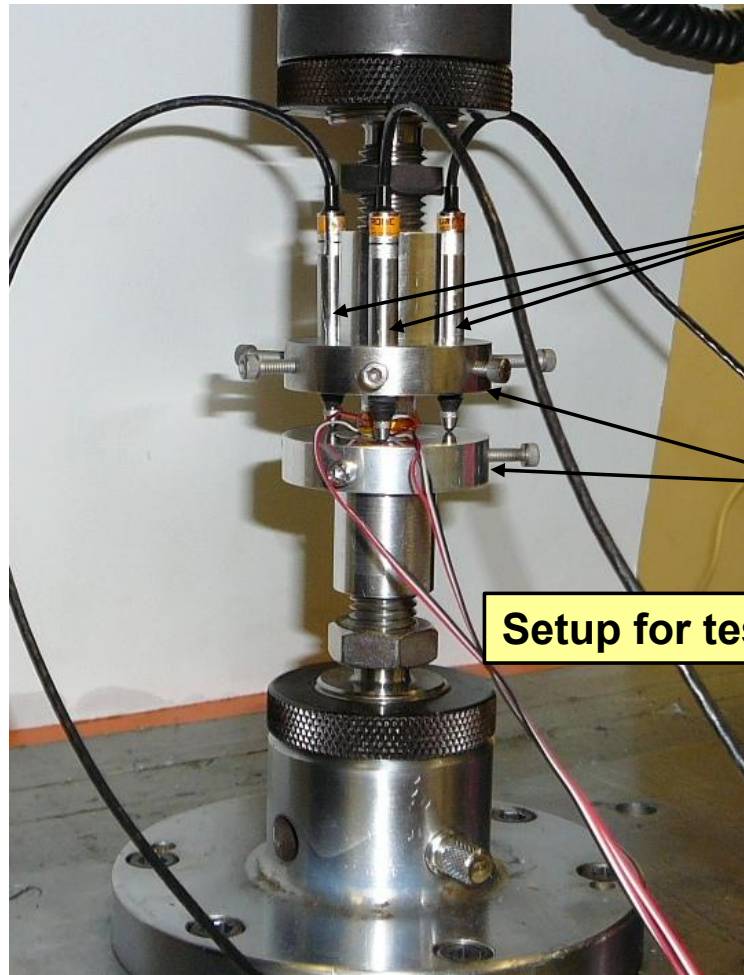


This is what you want to measure

See backup charts for an example of backing out unwanted displacement from test data.

Case History: Displacement Gages for Measuring Local Displacement for the Joint Shown in Fig. 1

Figure 2



LVDTs used to measure displacement across the joint

Fixtures for mounting LVDTs

Setup for tests 6 – 10

(LVDTs not used for tests 16 – 20;
load cell used to indicate gapping)

Case History—General Procedure: Tests 16 – 20

(after working out the bugs in the original procedure used in tests 6 – 10)

1. Calibrate the load cell to convert average strain reading to pounds (lb) compressive force.
2. Assemble the joint shown in Figure 1. Torque the nut until the load cell indicates a preload equal to 65% of the fastener's allowable yield load (not the actual measured yield strength). (Several NASA centers aim for this value as a nominal preload.) Compute the nut factor, K , which relates effective torque, T (equal to total applied torque minus any running torque from the locking feature), to the initial preload, P_{pi} (subject to potential relaxation and thermal effects):

$$K = \frac{T}{DP_{pi}} \quad \text{where } D = \text{bolt diameter}$$

3. Remove the nut, relubricate, reinstall, and again torque nut per step 2, above. Purpose: Confirm that the nut factor is not significantly changed by reuse.
 - *Results: 0% to 13.5% change in K exhibited by fasteners tested.*

Steps 2 and 3 established a baseline for later comparison (when determining whether yielding significantly affects the torque-tension relationship).

(continued)

Case History—General Procedure: Tests 16 – 20 (continued)

4. Torque the nut until the load cell indicates a preload higher than the fastener's allowable yield load of 14,300 lb (83.3% of the specified minimum ultimate strength for 3/8" bolts).
5. Apply the maximum design yield load, and record the load-cell reading.
 - *Maximum design yield load was defined as specified minimum ultimate strength multiplied by the ratio of 1.25/1.4 (yield factor of safety divided by ultimate factor of safety for NASA's protoflight test approach) = $(1.25/1.4)(17,100) = 15,268$ lb for 3/8" bolts.*
6. Remove applied load. Record the residual preload and determine the percentage of preload that was lost as a result of fastener yielding.
 - *Results for 3/8" bolts: loss ranged from 18% to 21%.*
7. Reapply the maximum design yield load, and then remove it. Record the residual preload and determine the percentage of preload that was lost as a result of fastener yielding.
 - *Results for 3/8" bolts: additional loss ranged from 0% to 1.5% (insignificant).*
 - *Conclusion: Test confirms the theory that, because of strain hardening, a second application of the load that first yielded the fastener will not cause significant additional loss in preload.*

(continued)

Case History—General Procedure: Tests 16 – 20 (continued)

8. Disassemble the joint, then reassemble in the same configuration, using the same hardware. Relubricate, and retorque the nut until the load cell indicates a preload higher than the fastener's allowable yield load. Based on comparison with the results of steps 2 and 3, draw conclusions as to whether yielding has significantly affected the torque-preload relationship.
 - *This step was to help determine whether fastener yielding is detrimental, considering that fasteners are sometimes removed (e.g., after structural testing of a flight assembly) and then reinstalled.*
 - *If yielding significantly affects the torque-tension relationship, the NESC team may have to either prohibit yielding, as verified by analysis, or prohibit reuse of fasteners that may have yielded.*
 - *Test results: less than 10% change in nut factor, which was about the same variation as was found between installations in Step 3.*

Case History—General Procedure: Tests 16 – 20 (continued)

9. Apply tensile load until failure occurs.
 - *Question to be answered here: Does such a high preload reduce the joint's ultimate tensile strength?*
10. Compute the sample mean ultimate strength, and compare it with the value obtained in step 3 of test sets 1 – 5 and 11 – 15 (fasteners pulled to failure without preload).
 - *Results*
 - *#10 screws: 3600 lb (vs. 3690 lb for fasteners tested without preload)*
 - *3/8" bolts: 19,980 lb (vs. 19,810 lb for fasteners tested without preload)*
 - *Conclusion: Insignificant change*

Note: The above procedure is a simplified version of the actual procedure used during test.

Case History: Test Summary and Conclusions

- Preload did not reduce the ultimate tensile strength of the joints tested.
- Bolt yielding during load application caused an average loss in preload of 19% for the 3/8" bolts.
 - Bolts preloaded to the allowable tensile yield load (83.3% of rated ultimate strength)
 - Applied load = 89% of rated (specified minimum) ultimate strength
- A second application of the same load caused negligible additional preload loss.
- Bolt yielding had insignificant affect on the nut factor during reinstallation.

Tentative* conclusions:

For a typical joint with A-286 bolts and aluminum fittings,

- **Assessment of ultimate tensile strength should be based on applied load only, ignoring preload.**
- **Bolt yielding is not detrimental if the associated preload loss is acceptable (and if there is no other design-specific requirement that would not be met as a result of yielding).**

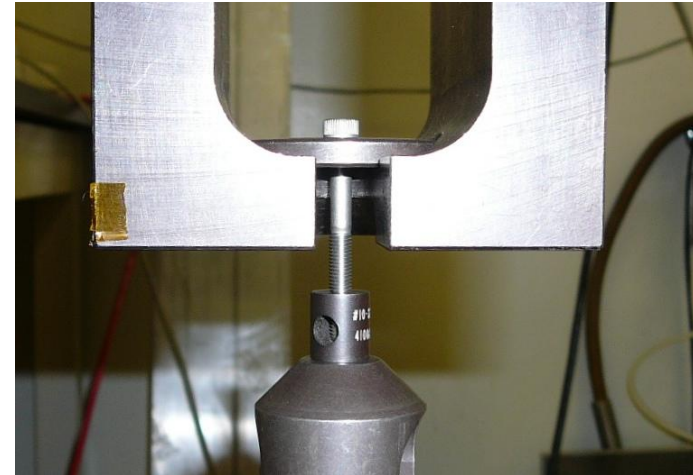
*The NESC team decided additional study was warranted given the test limitations. This study resulted in identified design constraints for which the above conclusions apply. See NASA-STD-5020.

Backup Pages

Example of Backing Out Displacement of the Test Machine

Case History: Example of Estimating Local Displacement

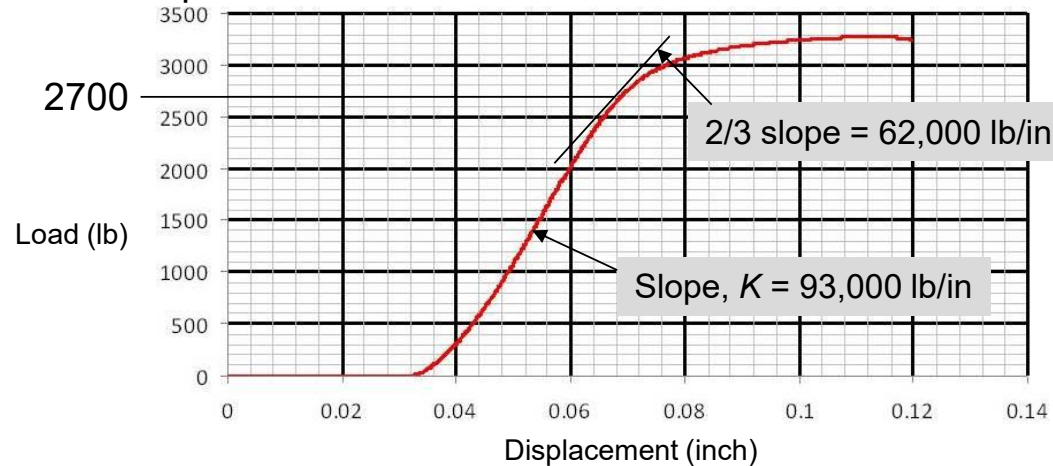
- The figure at right shows the test setup for tensile testing of a #10 socket-head cap screw (tests 1 – 5).
- Measuring local displacement in this configuration was not practical.
- Instead, the test engineers ...
 - estimated the elastic stiffness of the bolt, $K_b = AE/L$ (= 483,000 lb/in for the #10 fasteners),
 - subtracted the calculated length change of the bolt from the total displacement recorded in the linear region of the data and derived an approximate stiffness of the test machine,
 - and then backed out the machine displacement from the total recorded displacement to get local displacement only.
- From the resulting load-deflection plot (with machine displacement removed), the bolt's yield strength was estimated using Johnson's 2/3-slope method.



Note: For tests 11 – 15 (3/8" bolts), the bolts were tested to failure in the configuration shown in Figure 1 so that any local yielding under the washers in the aluminum parts would take place, strain-hardening the material.

Comparison of Data Plots (Test #2, lowest strength of all fasteners tested)

As output from the test machine:



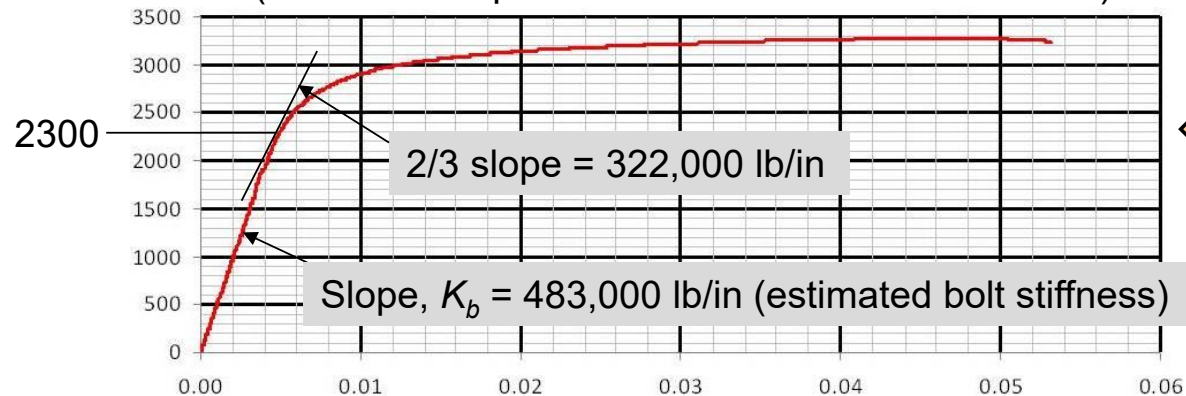
Estimate machine stiffness, K_m :

$$K_m = \left[\frac{1}{K} - \frac{1}{K_b} \right]^{-1}$$

$$= \left[\frac{1}{93,000} - \frac{1}{483,000} \right]^{-1}$$

$$= 115,000 \text{ lb/in}$$

Reduced data (estimated displacement of test machine removed):



Assume (1) that the initial nonlinearity is in the test machine and (2) that, once past the initial nonlinearity, K_m is constant throughout the test. Subtract the calculated machine displacement from the raw data to get the plot at left.

4. Static Loads Testing of Large Assemblies for Strength Verification

- Introduction to Static Loads Testing
- Government Standards for Factors of Safety and Test Factors
- Special Considerations
- Introducing and Controlling Loads
- Developing the Load Cases
- Be Sure to Design the Right Test!

This section includes contributions from Tim Ring, formerly at Instar Engineering

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Introduction to Static Loads Testing

- When to consider:
 - Large structures that will be loaded in ways other than uniform inertia loading or that would not be adequately tested on a shaker
 - Example: when buckling is of concern
 - Simple structures that can be tested with one or two concentrated applied loads
 - When you have an available test facility configured for static testing
- Introducing loads
 - **Loading heads** (a.k.a. **load heads**): test fixtures installed in place of the structure's payloads or equipment for the purpose of receiving and distributing concentrated loads
 - **Load lines**: assembled combinations of ...
 - (usually) Load actuators (e.g., hydraulic jacks)
 - Load sensors (e.g., load cells, which use calibrated strain gages)
 - Straps, rods, cables, or other devices for spanning distances
 - Attachment fittings
 - **Load control**: method or system for controlling loads
- Instrumentation and data acquisition—discussed in Sec. 2

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. 9)	AIAA S-110-2005 (Ref. 10) combined with SMC-S-016 (Ref. 11), derived from the inactive DOD-HDBK-343 (Ref. 12)		

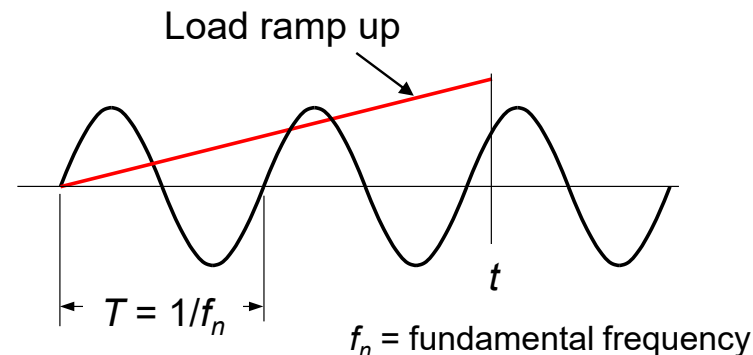
The structure should be tested to the specified or derived limit load multiplied by the applicable test factor.

*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.

Special Considerations for Static Loads Testing

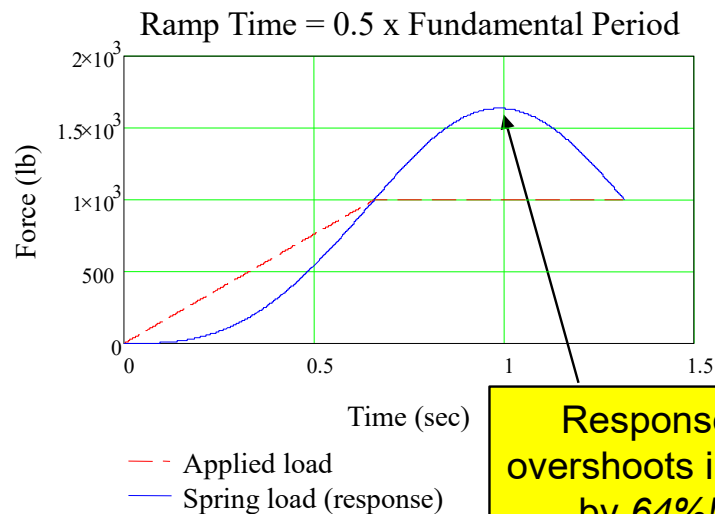
- Ensuring test fixtures, including loading heads, don't over-constrain an interface that is being tested (see Sec. 2)
- Devising appropriate static load cases
 - Can be challenging when simulating dynamic loads
 - See later discussion in this section
- Accommodating large deflections to ensure that any angle change in load vectors is acceptable
- Ensuring safety provisions are in place to prevent overloading
- Ensuring the rate of load application is slow enough to avoid significant dynamic response of the test article, which leads to an over-test

Relative to the test article's fundamental period, T , what should be the applied load's ramp-up time, t , in order to ensure negligible dynamic response?

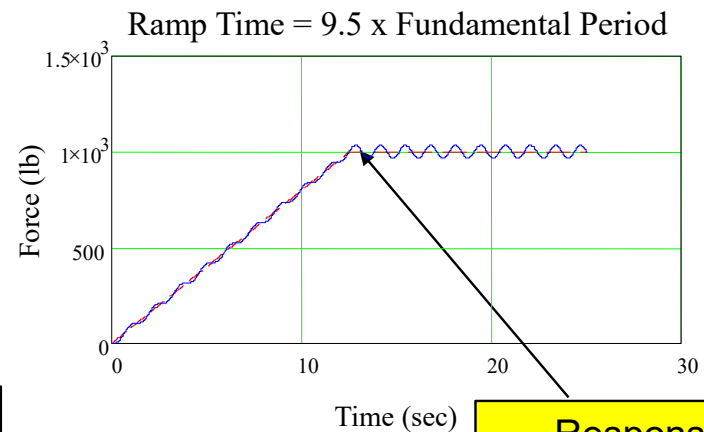


What Loading Rate Constitutes a “Static” Test?

Consider the following response of an undamped, 0.75-Hz mass-spring system to different ramp (and hold) time intervals:



Response
overshoots input
by 64%!



Response
overshoots input
by 3%

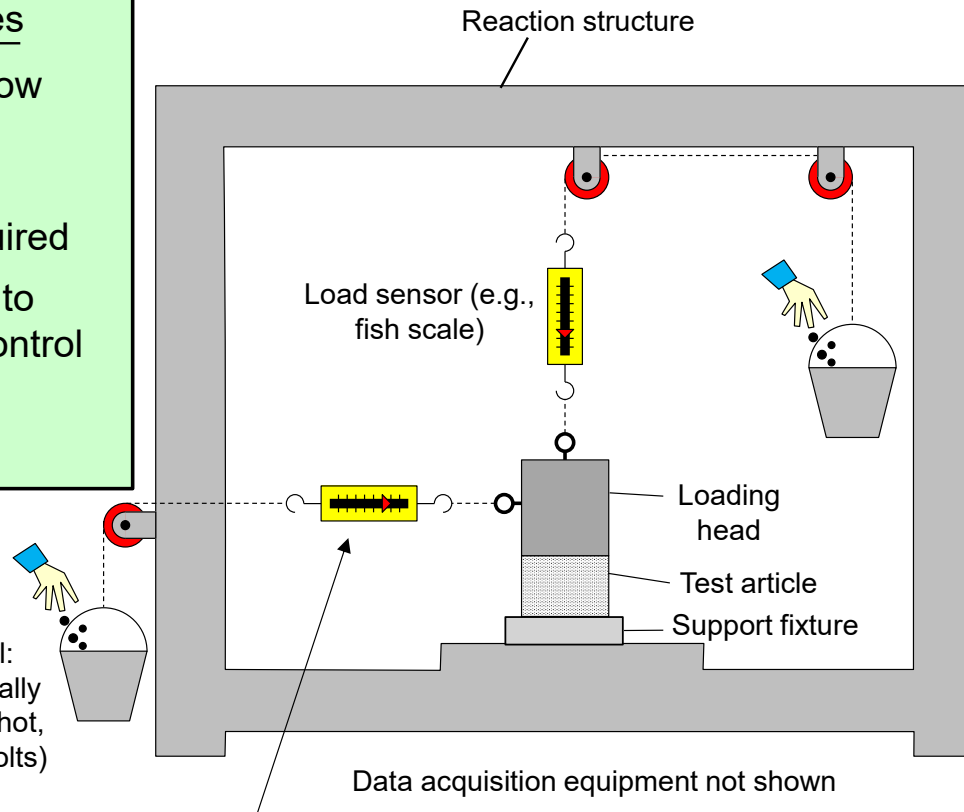
Recommendation: Use a ramp-up time for applied load of at least 10 times the fundamental period.

Loads Can Be Applied with Dead Weights

Advantages

- Simple and low cost
- No special facilities required
- May be able to accurately control low levels of applied load

Load control:
weights manually
added (steel shot,
sand, rocks, bolts)



Disadvantages and Limitations

- Not practical for high loads or when simultaneous loading in many load lines is required
- Can't apply compressive loads
- Stick/slip of pulleys (friction)
- Cumbersome to unload

Tip: Because of variable friction loss in a pulley, put the load sensor between the test article and the pulley.

Applying Loads with Hydraulic Jacks

- A calibrated load cell is placed in series with each hydraulic jack.
- Load introduced by each hydraulic jack is controlled by pressure.
 - This can be done manually with a hand pump to achieve the desired load.
 - Limits the number of load lines, and loads won't increase simultaneously
 - More versatile and safer for the test article: control pressure (hence load) with a computerized control system.
 - Loads are more accurately controlled, especially when using multiple load lines.
 - The control system can “dump” pressure throughout the hydraulic system when the load reading in any line spikes or drops outside of a specified tolerance band.
 - Sudden spikes or drops can result from sudden displacement of the test article, which in turn can be caused by joint slip or local structural failure.
 - May be able to avoid catastrophic structural failure when using specified dump tolerances.
- Loads can be applied in compression (if the load line is stable) as well as tension.
- The hydraulic pressure in a jack, when multiplied by the applicable surface area in the jack, can be used as a sanity check for the load-sensor reading.

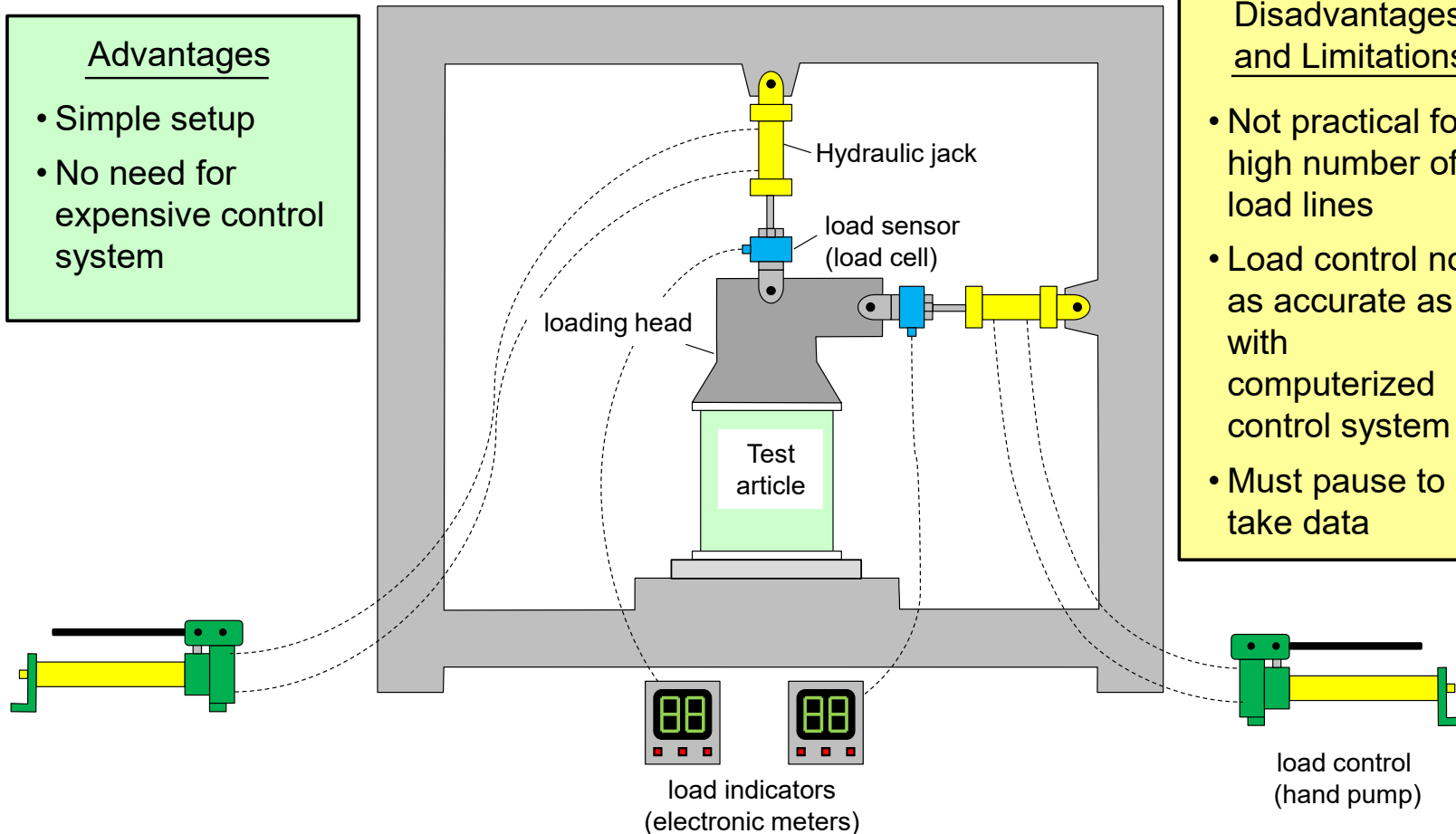
Disadvantages of Using Hydraulic Jacks

- You probably won't be able to accurately control low-magnitude loads.
 - If you want to apply 20 lb, for example, you may be able to get better accuracy by slowly adding weights to a suspended bucket (see earlier page).
- Load lines can be heavy.
 - Weight adds load to the test article.
- It's likely that hydraulic fluid will leak onto the test article.

Example Setup for Manual Control of Hydraulic Jacks

Advantages

- Simple setup
- No need for expensive control system



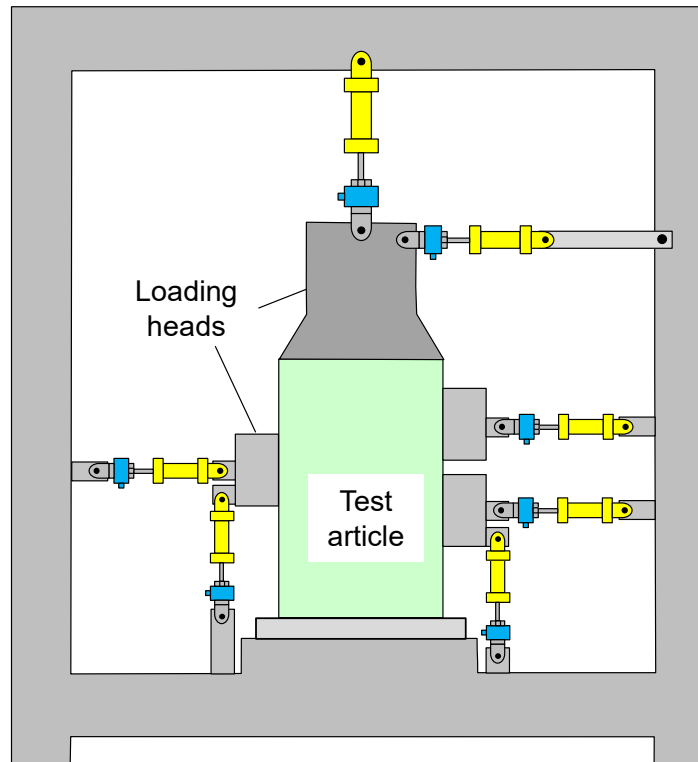
Disadvantages and Limitations

- Not practical for high number of load lines
- Load control not as accurate as with computerized control system
- Must pause to take data

Computerized Load Control with Hydraulic Jacks

Advantages

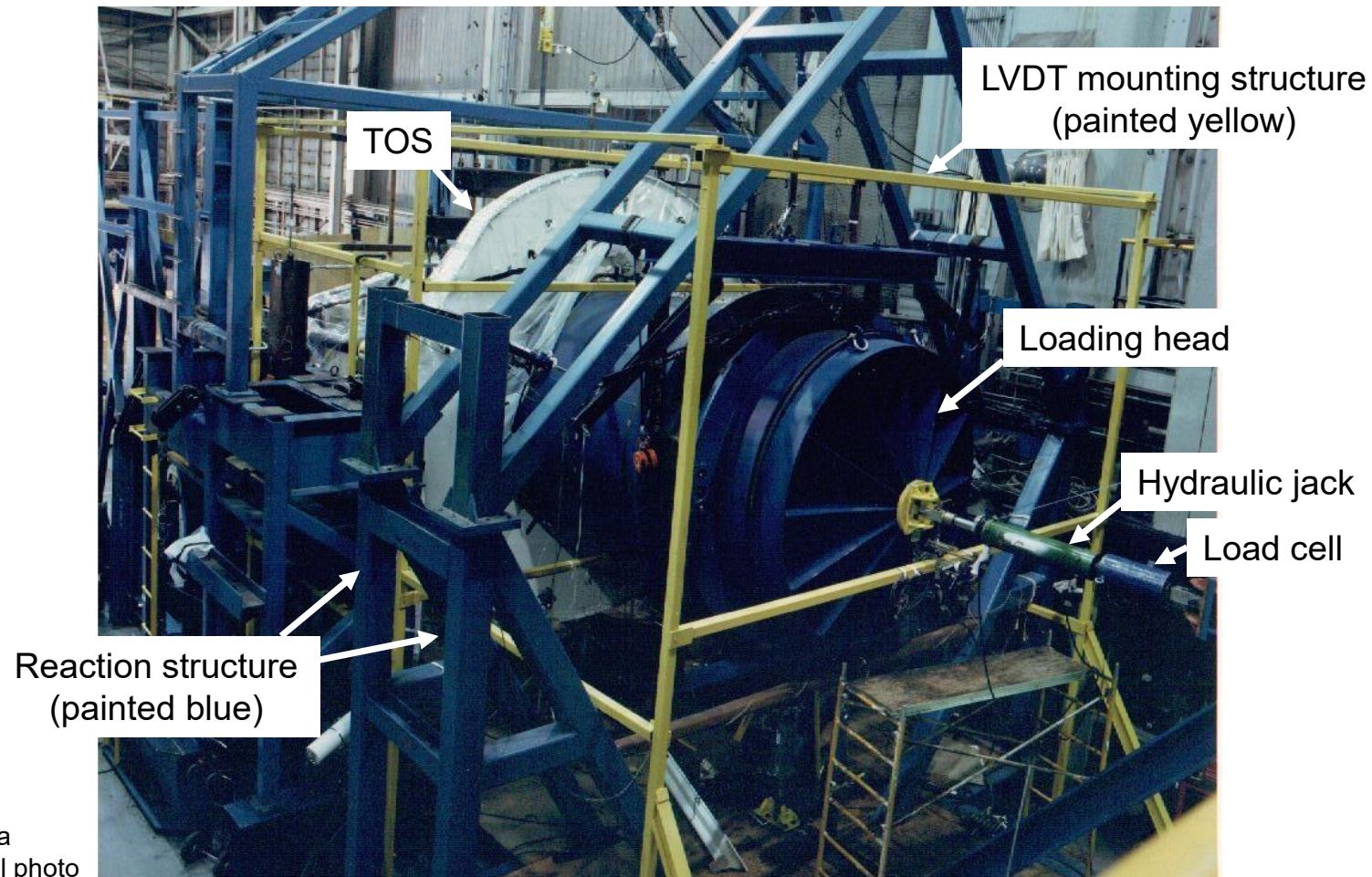
- Can easily accommodate many load lines, with simultaneous loading
- Near-continuous data acquisition
- Can program safety provisions such as “hold” tolerances (perhaps $\pm 3\%$ of load magnitude) and “dump” tolerances (e.g., $\pm 6\%$)



Disadvantages and Limitations

- Requires expensive equipment that needs periodic maintenance
- As with any use of computers, we have to check the program and input data (easy to become complacent)

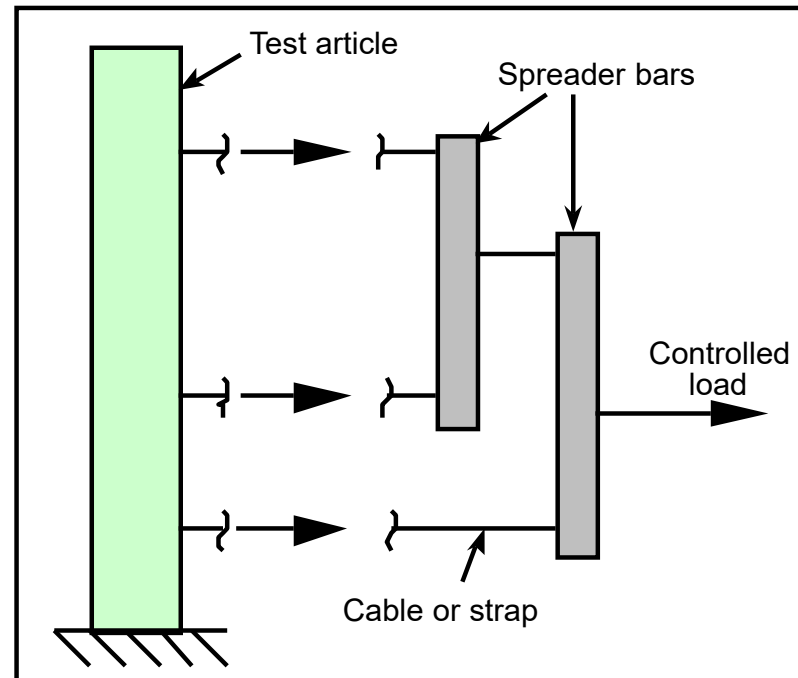
Transfer Orbit Stage in Static Loads Test



Martin
Marietta
archival photo

Adding Load Lines with Whiffletrees

When simulating inertia loads, introducing small loads at many locations makes for a better test than applying one large load.



A **whiffletree** (a.k.a. **whippletree**) is a statically determinate assembly of spreader bars and cables that divides a single applied load into multiple loads.

Image from [http://en.wikipedia.org/wiki/Whippletree_\(mechanism\)](http://en.wikipedia.org/wiki/Whippletree_(mechanism))

Ensuring a Static Loads Test Adequately Verifies Strength

How do we ensure the static loads test we are designing will adequately load the structure to meet the objective of verifying strength?

- Review the basic process from Sec. 2:
 - Make the overall loads representative of the maximum expected flight conditions.
 - Show by analysis that the limit test conditions will stress the critical areas at least as severely as will the flight conditions.
 - Using the same basic finite element model (FEM) as is used for loads analysis
 - Multiply the limit test loads by the specified test factor.
- This process is most easily followed with a simple, compact structure loaded in flight mostly by uniform acceleration.
- It's more difficult to ensure an adequate static test for a complex, redundant structure that will be significantly stressed in flight by response of its own modes of vibration.
- In either case, the finite element model may not accurately predict load distribution and stresses.

What If Loads Do Not Distribute As Predicted?

Let's say you calibrate a strain gage to measure the axial load in a particular member in a statically indeterminate structure (redundant members and load paths).

- The limit load from flight loads analysis for that member is 3920 lb.
- The test must be done to 1.25 times limit, or 4900 lb.
- For the test loads, applied to the overall structure, your FEM (same as used for the flight loads analysis) predicts a load in that member of 4900 lb.
- But, in test, the calibrated strain gage shows the actual member load is only 4500 lb.

Has this structural member been adequately tested?

One way to respond to the above situation: Increase the applied loads until the member's axial load reaches 4900 lb.

Do you agree with this strategy?

The problem with this approach: The test FEM does not accurately predict load distribution. But neither does the model used for flight loads analysis, assuming the test FEM is essentially the same.

Thus, 4900 lb is the wrong target load.

Ideally, we'd correlate the model with the test data and then repeat the loads analysis.

But program schedules normally don't allow us to delay the test for this purpose.

So How Do We Ensure a Static Test Adequately Verifies Strength?

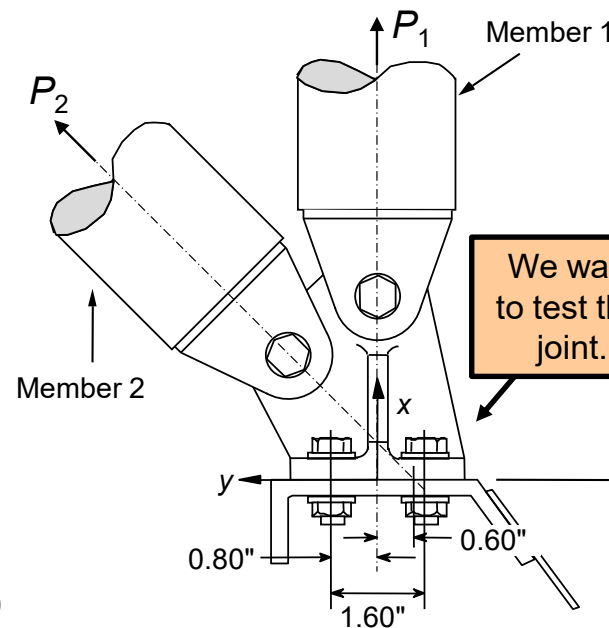
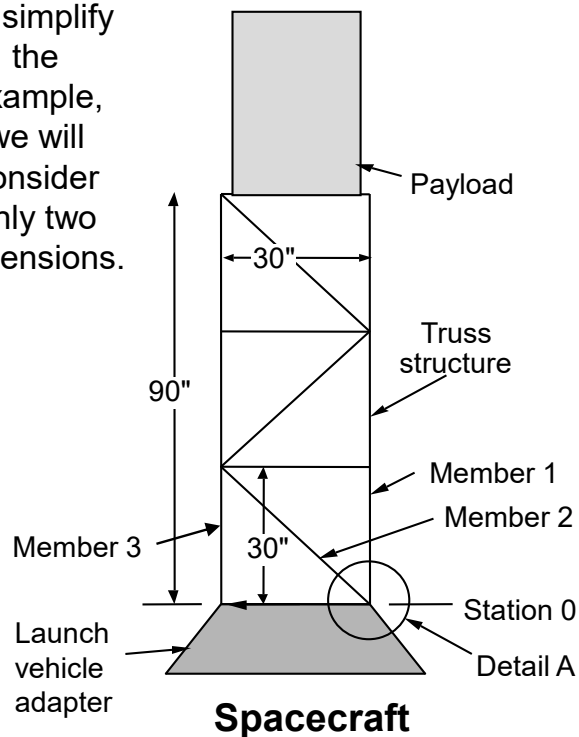
- The key is to combine two strategies:
 - Make sure the load cases envelop—and provide a realistic simulation of—the overall flight loads (e.g., total loads at major interfaces) so that any errors in predicted load distribution are consistent between flight loads analysis and test loads analysis.
 - As demonstrated for designing a sine burst test in Sec. 2: Using consistent FEMs or other methods of predicting load distribution, show the critical areas will be stressed at least as severely as they are by the design loads.
- An effective static load case is often a simulation of the distribution of inertia loading experienced from quasi-static loads combined with one or more excited mode shapes.
- With this apples-to-apples approach, we establish a practical, high level of confidence.
- I can't think of a better approach, given the impracticality of delaying the test for flight loads analysis with a model that's correlated for load distribution.

With the above approach, the test is considered adequate regardless of actual strain readings.

Example Problem 4-a: Designing a Static Load Case

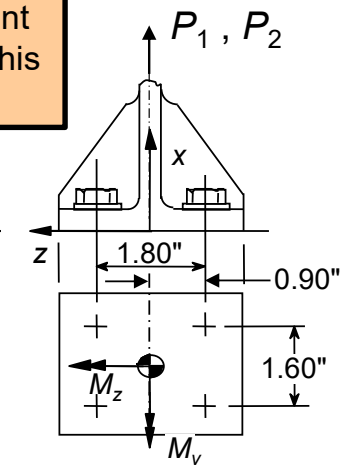
Objective: Design a static load case for this spacecraft truss structure that will adequately test the 4-bolt joint in Detail A.

To simplify the example, we will consider only two dimensions.



We want to test this joint.

Detail A



Projected views

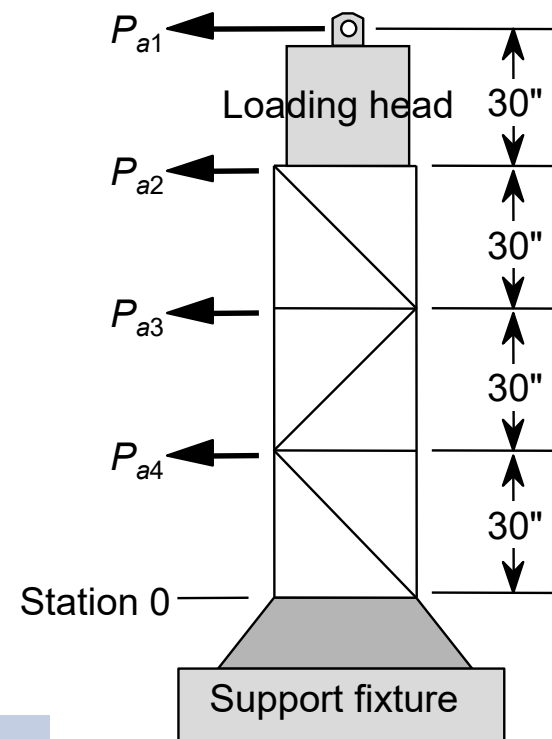
continued

Example Problem 4-a, Continued

We want to design a static load case that verifies strength for the joint shown in Detail A on the previous page.

Assumptions:

- Design loads (test targets) are based on coupled loads analysis (CLA) for flight events.
- Possible locations for applied loads are as shown at right.



Note: The process demonstrated in this example can be used also when starting with specified quasi-static loads rather than CLA results.

continued

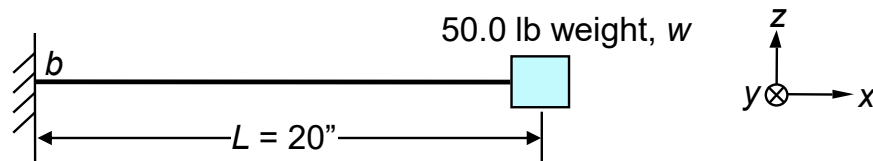
Example Problem 4-a, Continued: Process for Designing Static Load Cases

1. Identify all critical potential failure modes and key overall interface or section-cut loads along with associated indicator load parameters (e.g., bolt load). If those parameters are not calculated directly in the loads analysis, generate a **load transformation matrix (LTM)**, which is a set of linear equations relating internal loads to external loads or dynamic responses, in matrix form.
2. Calculate target loads for the indicator parameters, based on the specified design loads or by flight loads analysis.
3. Identify the driving load conditions (e.g., uniform acceleration, response of one or more modes of vibration). Dedicate a test case to each mode or loading condition.
4. For each load case, define preliminary applied loads that are consistent with the selected driving loading conditions, achieving target total forces and moments at major interfaces.
5. Modify the FEM of the spacecraft used in flight loads analysis to match the test configuration. Generate an LTM that relates the indicator parameters to loads applied at the loading locations.
6. Use the LTM to assess the initial load cases from Step 4: Are all target loads achieved? Revise, scrub, or add load cases, as needed. Identify permissible amounts of over-test for certain parameters to ensure all failure modes are tested.

For Example 4-a, we'll use the above process, but we'll generate only one load case. continued

Let's leave Example Problem 4-a for a moment:
Example Problem 4-b: Generating a Load Transformation Matrix

The cantilevered beam shown below is subject to dynamic base acceleration. We want to calculate bending moments and axial force at point b in terms of response accelerations of the concentrated mass. Assume the beam's mass is negligible.



$$\text{Mass, } m = \frac{w}{g} = \frac{50 \text{ lb}}{386 \text{ in/s}^2} = 0.130 \text{ lb} \cdot \text{s}^2 / \text{in}$$

(for use when the units of acceleration are in/s²)

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

(Continued)

Example Problem 4-b, Continued

Write equations for base loads in terms of mass accelerations:

Recalling that inertia force has a sign opposite that of the acceleration,

$$F_{bx} = m(-a_x) = -0.130a_x$$

$$M_{by} = -Lm(-a_z) = -20.0(-0.130)a_z = 2.60a_z$$

$$M_{bz} = Lm(-a_y) = 20.0(-0.130)a_y = -2.60a_y$$

where F_{bx} = axial force (+ implies tension) (lb)
 M_{by} = moment at point b about y axis (in-lb)
 M_{bz} = moment at point b about z axis (in-lb)
 a_x, a_y, a_z = accelerations in x, y , and z (in/s²)

In matrix form,

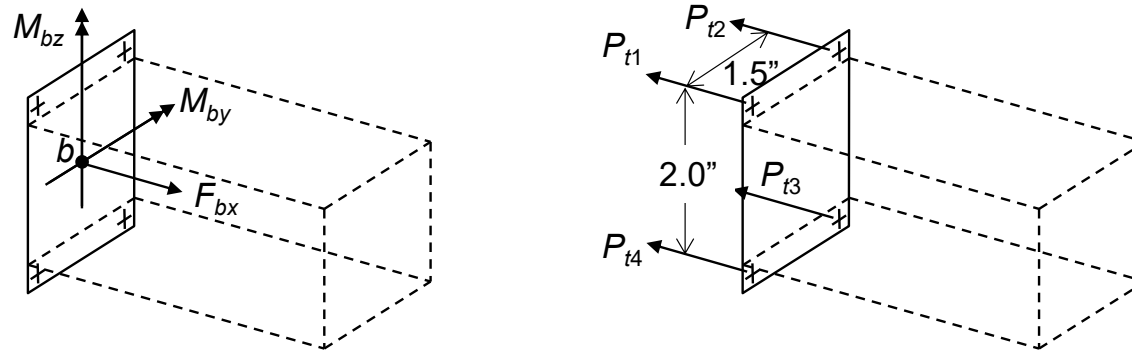
$$\begin{Bmatrix} F_{bx} \\ M_{by} \\ M_{bz} \end{Bmatrix} = \underbrace{\begin{bmatrix} -0.130 & 0 & 0 \\ 0 & 0 & 2.60 \\ 0 & -2.60 & 0 \end{bmatrix}}_{\text{LTM}} \begin{Bmatrix} a_x \\ a_y \\ a_z \end{Bmatrix}$$

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

(Continued)

Example Problem 4-b, Continued

Now let's generate an LTM that will calculate bolt loads at the base of the cantilevered beam directly from accelerations:



$$\begin{aligned} \text{Bolt tensile loads (lb): } P_{ti} &= \frac{F_{bx}}{4} \pm \frac{M_{by}}{2(2.0)} \pm \frac{M_{bz}}{2(1.5)} \\ &= 0.250F_{bx} \pm 0.250M_{by} \pm 0.333M_{bz} \end{aligned}$$

In matrix form,

$$\begin{Bmatrix} P_{t1} \\ P_{t2} \\ P_{t3} \\ P_{t4} \end{Bmatrix} = \begin{bmatrix} 0.250 & 0.250 & 0.333 \\ 0.250 & 0.250 & -0.333 \\ 0.250 & -0.250 & -0.333 \\ 0.250 & -0.250 & 0.333 \end{bmatrix} \begin{Bmatrix} F_x \\ M_{ay} \\ M_{az} \end{Bmatrix}$$

(Continued)

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

Example Problem 4-b, Continued

Now combine equations from the previous two pages:

$$P_{ti} = 0.250F_{bx} \pm 0.250M_{by} \pm 0.333M_{bz}$$

$$= 0.250(-0.130a_x) \pm 0.250(2.60a_z) \pm 0.333(-2.60a_y) = -0.0325a_x \pm 0.866a_y \pm 0.650a_z$$

With matrix multiplication,

$$\begin{Bmatrix} P_{t1} \\ P_{t2} \\ P_{t3} \\ P_{t4} \end{Bmatrix} = \begin{bmatrix} 0.250 & 0.250 & 0.333 \\ 0.250 & 0.250 & -0.333 \\ 0.250 & -0.250 & -0.333 \\ 0.250 & -0.250 & 0.333 \end{bmatrix} \begin{bmatrix} -0.130 & 0 & 0 \\ 0 & 0 & 2.60 \\ 0 & -2.60 & 0 \end{bmatrix} \begin{Bmatrix} a_x \\ a_y \\ a_z \end{Bmatrix} = \begin{bmatrix} -0.0325 & -0.866 & 0.650 \\ -0.0325 & 0.866 & 0.650 \\ -0.0325 & 0.866 & -0.650 \\ -0.0325 & -0.866 & -0.650 \end{bmatrix} \begin{Bmatrix} a_x \\ a_y \\ a_z \end{Bmatrix}$$

Integrated LTM

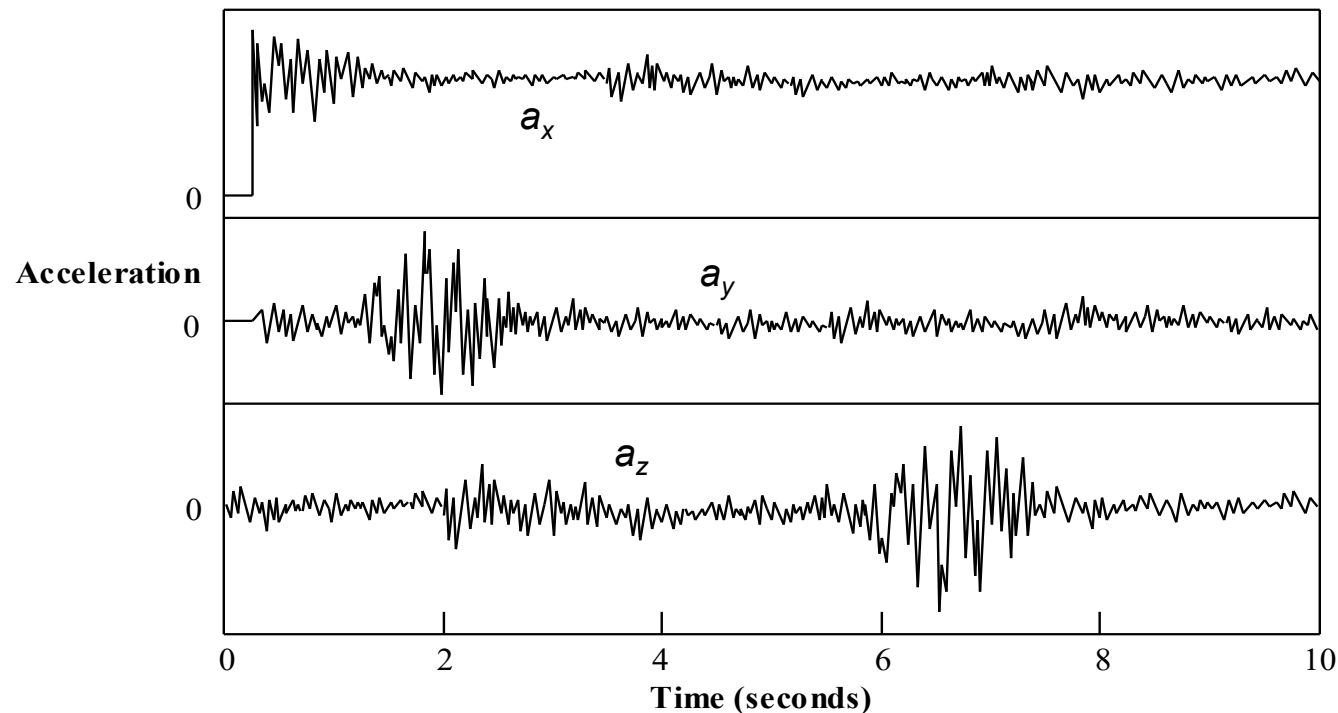
We now have a set of equations that we can use in the flight loads analysis to calculate bolt loads directly—without having to include the FEM detail needed to model bolts individually.

To design static test load cases, we can similarly develop an LTM for bolt loads or stresses in terms of applied forces rather than accelerations.

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

Hypothetical Acceleration Time Histories for Example Problem 4-b

Assume our cantilevered beam in Example 4-b has the following response time histories (x, y, and z directions) over a 10-second interval at lift-off:



Note that the responses don't peak at the same time.

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

Hypothetical Loads for Example Problem 4-b

Response Parameter	Time, t (sec.):	0.312	2.235	7.164	Maximum
Accelerations (in/s ²):					
a_x		2290	562	1430	2290
a_y		1890	7690	1940	7690
a_z		1980	2460	9840	9840
Axial force at base, F_{bx} (lb)		298	73	186	298
Moments at base (in·lb):					
M_{by}		5150	9780	25,600	25,600
M_{bz}		4910	20,000	6320	20,000
Peak bolt load (lb) calculated from the maximum base loads		—	—	—	13,120 ←
Peak bolt load (lb) calculated directly with integrated LTM		3000	8280	8120	8280 ←

(Absolute values)

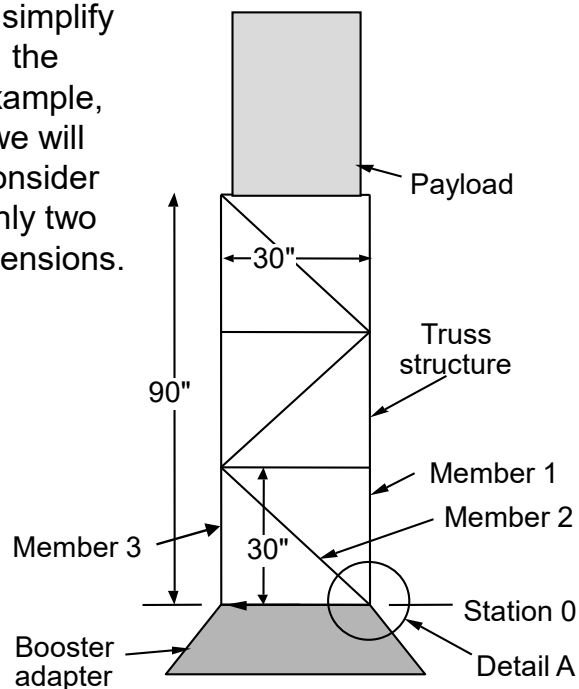
A 37% reduction in bolt load when computed time-consistently!

This example is hypothetical but intended to be representative

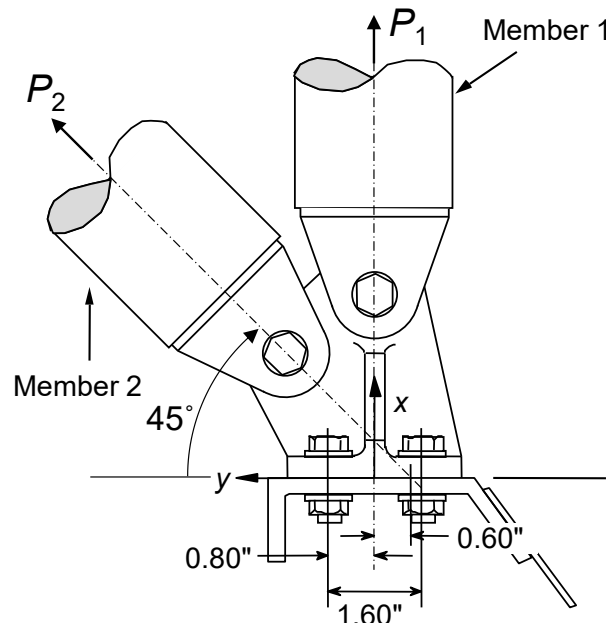
Using the same math model and same LTM, we would design a static load case that is predicted to cause a peak bolt load of 8280 lb times the applicable test factor.

Back to Example Problem 4-a: Designing a Static Load Case

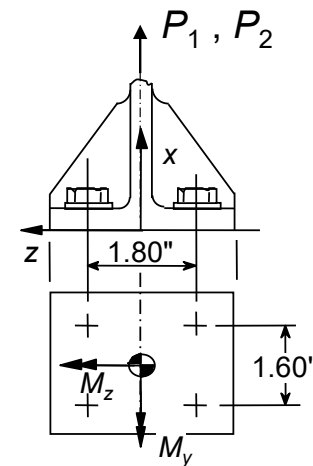
To simplify the example, we will consider only two dimensions.



a. Stowed spacecraft



b. Detail A



Problem statement: Design a static load case that will hit target loads for (a) tension in member 1 and (b) bolt tension, to 1.25 times limit.

Example adapted from Ref. 13.

Example Problem 4-a (continued)

Step 1: Identify all critical potential failure modes and associated indicator parameters. If those parameters are not calculated directly in the loads analysis, generate an LTM relating the parameters to responses calculated in the loads analysis.

In this example, let's use two indicator parameters: tensile load in Member 1 (P_1) and bolt tensile load (P_t).

a. Write equations for transforming member loads to the bolt-pattern centroid:

$$\text{Tension, } P_x = P_1 + P_2 \sin 45^\circ = P_1 + 0.707P_2$$

$$\text{Moment, } M_z = 0.60P_2 \sin 45^\circ = 0.424P_2$$

Note: We're ignoring the small difference between 45° and the actual angle of member 2 when accounting for the 0.6" offset.

b. Develop an equation for peak bolt tension in terms of the centroidal loads:

$$\text{Bolt tensile load, } P_t = \left[\frac{P_x}{4} \pm \frac{M_z}{2(1.60)} \right] = 0.250P_x \pm 0.313M_z$$

c. Combine equations to determine bolt tension in terms of member loads:

$$\begin{aligned} P_t &= 0.250(P_1 + 0.707P_2) \pm 0.313(0.424P_2) \\ &= 0.250P_1 + 0.309P_2 \quad \Leftarrow (\text{max. bolt load}) \end{aligned} \quad (\text{Continued})$$

Example Problem 4-a (continued)

Step 1 (continued):

- d. To ensure the test is representative of the flight loading conditions, let's also write equations for total shear force, V_y , and moment, M_x , at Station 0:

$$V_y = 0.707P_2$$

$$M_x = -15.0(P_1 + 0.707P_2) + 15.0P_3$$

$$= -15.0P_1 - 10.6P_2 + 15.0P_3$$

Note: To simplify the example, we are ignoring axial loading for this spacecraft.

- e. Now let's compile the equations into an LTM for the four parameters of interest:

$$\begin{Bmatrix} P_1 \\ P_t \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0.250 & 0.309 & 0 \\ 0 & 0.707 & 0 \\ -15.0 & -10.6 & 15.0 \end{bmatrix} \begin{Bmatrix} P_1 \\ P_2 \\ P_3 \end{Bmatrix}$$

- f. Finally, we multiply this matrix by the LTM (generated by FEA) that relates the three member loads to parameters computed in the coupled loads analysis, just as we did for the cantilevered-beam example.

For this example, we'll refer to V_y as **base force** and M_x as **base moment**. Together, they are the **base loads**.

Probably the most important thing to do in order to ensure an adequate strength test is to achieve base loads that envelop the flight limit base loads times the test factor.

Example Problem 4-a (continued)

Step 2: Calculate target loads for the indicator parameters, based on the specified design loads or by dynamic loads analysis.

Let's assume for this problem that the LV organization does the coupled loads analysis, using the LTM generated in Step 1 to calculate the limit (maximum and minimum) loads of interest.

Hypothetical results:

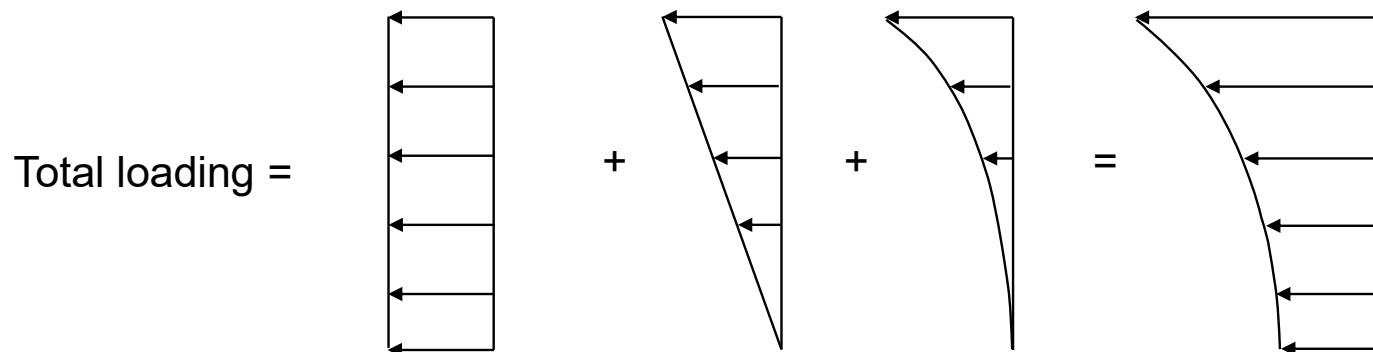
<u>Parameter</u>	<u>Limit Load</u>	Includes 1.25 test factor		
		<u>Test Target</u>	<u>Allowable Load</u>	<u>Margin of Safety</u>
P_1 (lb)	+ 10,700	+ 13,400	14,300	+0.07
P_t (lb)	+ 5120	+ 6400	6670	+0.04
V_y (lb)	± 6480	± 8100	N/A	N/A
M_x (in-lb)	± 492,000	± 615,000	N/A	N/A

Hypothetical, for this example

Example Problem 4-a (continued)

Step 3: Identify the driving load conditions.

- During the launch event of interest, let's assume there will be some amount of uniform steady-state lateral loading from wind and some dynamic load.
- Hypothetically, the fundamental bending frequency of the spacecraft is 16 Hz and the second bending mode is 88 Hz.
- Let's also say, for this example, the truncation frequency for CLA was 50 Hz, so the second bending mode was not included.
- Thus, the lateral loading for our spacecraft is some combination of steady-state loading, dynamic angular acceleration as a rigid body, and excitation of the fundamental bending mode.



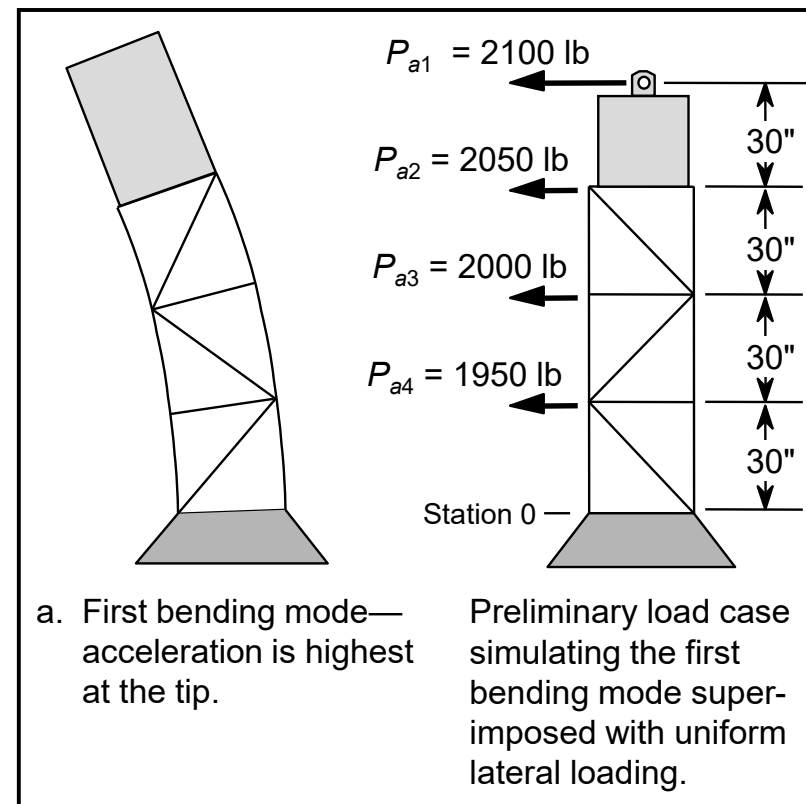
Example Problem 4-a (continued)

Step 4: Define preliminary applied loads that are consistent with the driving loading conditions.

Possible loading locations for the test are shown at right.

The load magnitudes shown here were chosen to achieve the overall shear force target of 8100 lb and the overall moment target of 615,000 in-lb while also trying to stay consistent with the mode shape (higher inertia load at top).

Precision is not necessary in this step. We simply want to establish a representative, initial load case.



Example Problem 4-a (continued)

Step 5: Generate an LTM that relates the indicator parameters to loads applied at the loading locations.

- a. Derive equations for loads in members 1 – 3 in terms of the four applied loads:

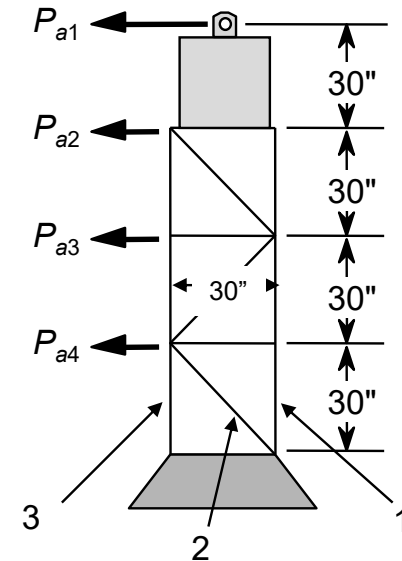
This is a statically determinate truss, so based on statics alone we calculate the force in member 1, P_1 , as

$$P_1 = 3.00P_{a1} + 2.00P_{a2} + 1.00P_{a3}$$

We can likewise calculate the loads in members 2 and 3.

Then we combine the equations into a matrix:

$$\begin{Bmatrix} P_1 \\ P_2 \\ P_3 \end{Bmatrix} = \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0 \\ 1.41 & 1.41 & 1.41 & 1.41 \\ -4.00 & -3.00 & -2.00 & -1.00 \end{bmatrix} \begin{Bmatrix} P_{a1} \\ P_{a2} \\ P_{a3} \\ P_{a4} \end{Bmatrix}$$



Example Problem 4-a (continued)

Step 5 (continued):

We now have two sets of equations:

$$\begin{Bmatrix} P_1 \\ P_t \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0.250 & 0.309 & 0 \\ 0 & 0.707 & 0 \\ -15.0 & -10.6 & 15.0 \end{bmatrix} \begin{Bmatrix} P_1 \\ P_2 \\ P_3 \end{Bmatrix} \quad \text{and} \quad \begin{Bmatrix} P_1 \\ P_2 \\ P_3 \end{Bmatrix} = \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0 \\ 1.41 & 1.41 & 1.41 & 1.41 \\ -4.00 & -3.00 & -2.00 & -1.00 \end{bmatrix} \begin{Bmatrix} P_{a1} \\ P_{a2} \\ P_{a3} \\ P_{a4} \end{Bmatrix}$$

Combining them, we generate a matrix relating the critical parameters to the applied loads:

$$\begin{Bmatrix} P_1 \\ P_t \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0.250 & 0.309 & 0 \\ 0 & 0.707 & 0 \\ -15.0 & -10.6 & 15.0 \end{bmatrix} \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0 \\ 1.41 & 1.41 & 1.41 & 1.41 \\ -4.00 & -3.00 & -2.00 & -1.00 \end{bmatrix} \begin{Bmatrix} P_{a1} \\ P_{a2} \\ P_{a3} \\ P_{a4} \end{Bmatrix}$$

$$\begin{Bmatrix} P_1 \\ P_t \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0. \\ 1.19 & 0.936 & 0.686 & 0.436 \\ 1.00 & 1.00 & 1.00 & 1.00 \\ -120. & -90.0 & -60.0 & -30.0 \end{bmatrix} \begin{Bmatrix} P_{a1} \\ P_{a2} \\ P_{a3} \\ P_{a4} \end{Bmatrix}$$

continued

Example Problem 4-a (continued)

Step 6: Use the LTM to assess the preliminary load case from Step 4. Modify the load case, if necessary to achieve the target internal loads while maintaining positive margins of safety.

$$\begin{Bmatrix} P_1 \\ P_t \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0. \\ 1.19 & 0.936 & 0.686 & 0.436 \\ 1.00 & 1.00 & 1.00 & 1.00 \\ -120. & -90.0 & -60.0 & -30.0 \end{bmatrix} \begin{Bmatrix} 2100 \\ 2050 \\ 2000 \\ 1950 \end{Bmatrix} = \begin{Bmatrix} 12,400 \\ 6640 \\ 8100 \\ -615,000 \end{Bmatrix}$$

Parameter	Predicted Load	Target Load	Ratio	Allowable Ratio
P_1 (lb)	+ 12,400	+ 13,400	0.93	1.07
P_t (lb)	+ 6640	+ 6400	1.04	1.04
V_y (lb)	+ 8100	± 8100	1.00	N/A
M_x (in-lb)	- 615,000	± 615,000	1.00	N/A

Conclusions:

The preliminary load case under-tests Member 1 and slightly over-tests the bolt.

Example Problem 4-a (continued)

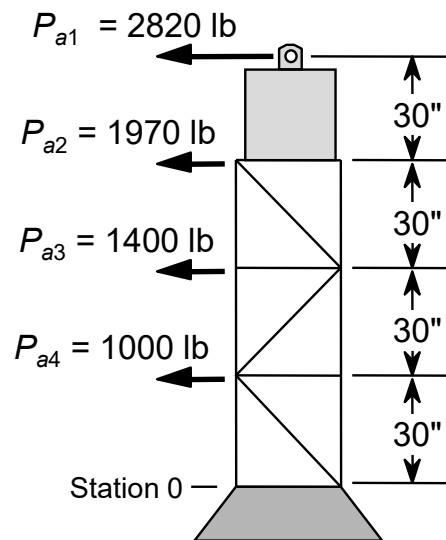
Modify the load case:

The matrix on the previous page has no inverse. The indicator parameters are not independent variables, and there is no load case that would hit all four targets exactly.

⇒ Base force apparently does not peak at the same time as base moment in the CLA.

We could confirm this by post-processing the CLA response time histories.

Let's aim only at overall moment, axial load in member 1, and bolt tension. By iteration, we find the following load case satisfies our criteria without over testing:



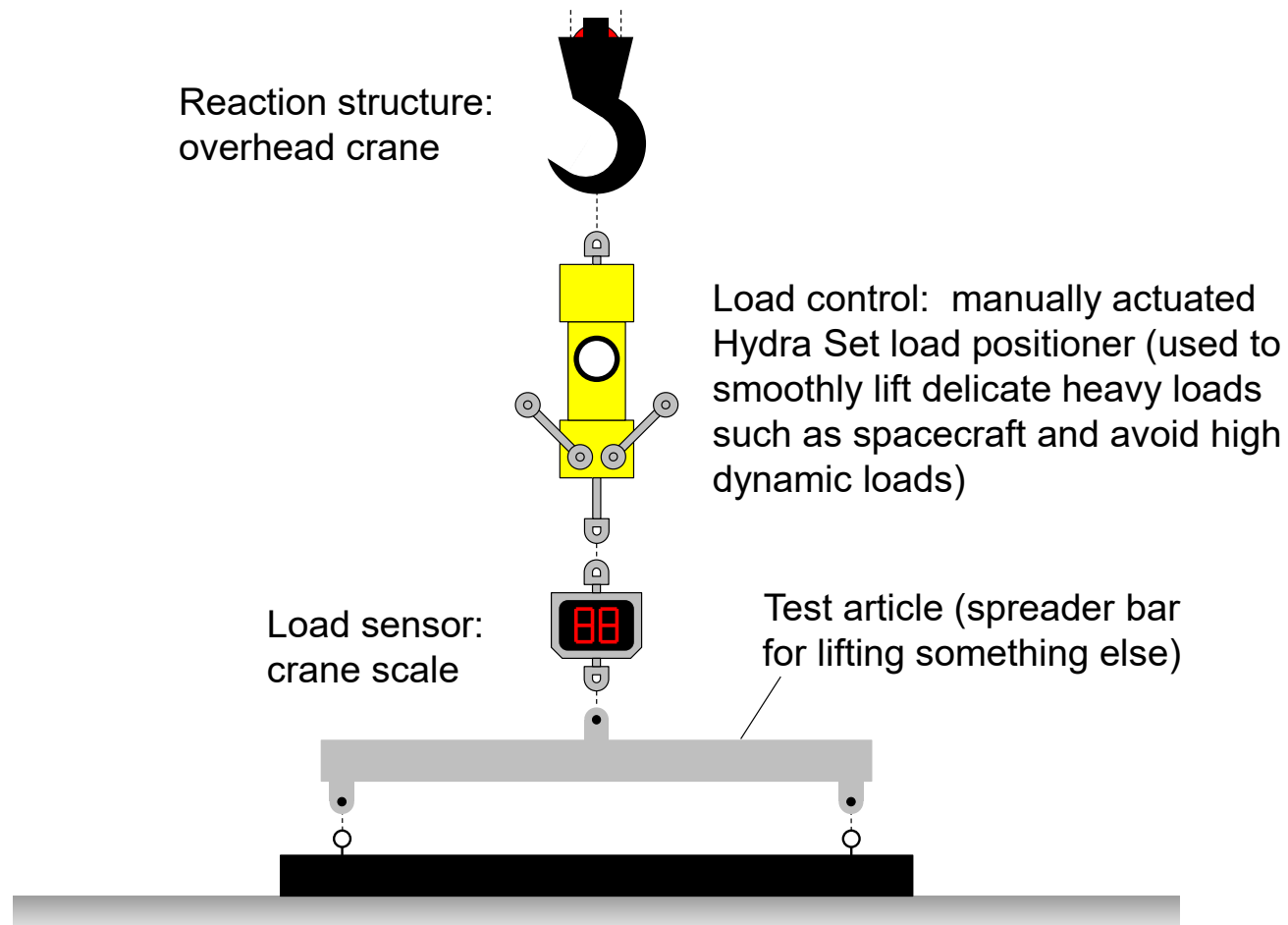
Parameter	Predicted Load	Target Load	Ratio	Allowable Ratio
P_1 (lb)	+ 13,400	+ 13,400	1.00	1.07
P_t (lb)	+ 6420	+ 6400	1.00	1.04
V_y (lb)	+ 7010	± 8100	0.87	N/A
M_x (in-lb)	- 614,000	± 615,000	1.00	N/A

In an actual test, we should either add another case that hits the target base force with reduced moment or accept the over test associated with hitting or exceeding all targets in one case.

Other Uses of Output-Transformation Matrices

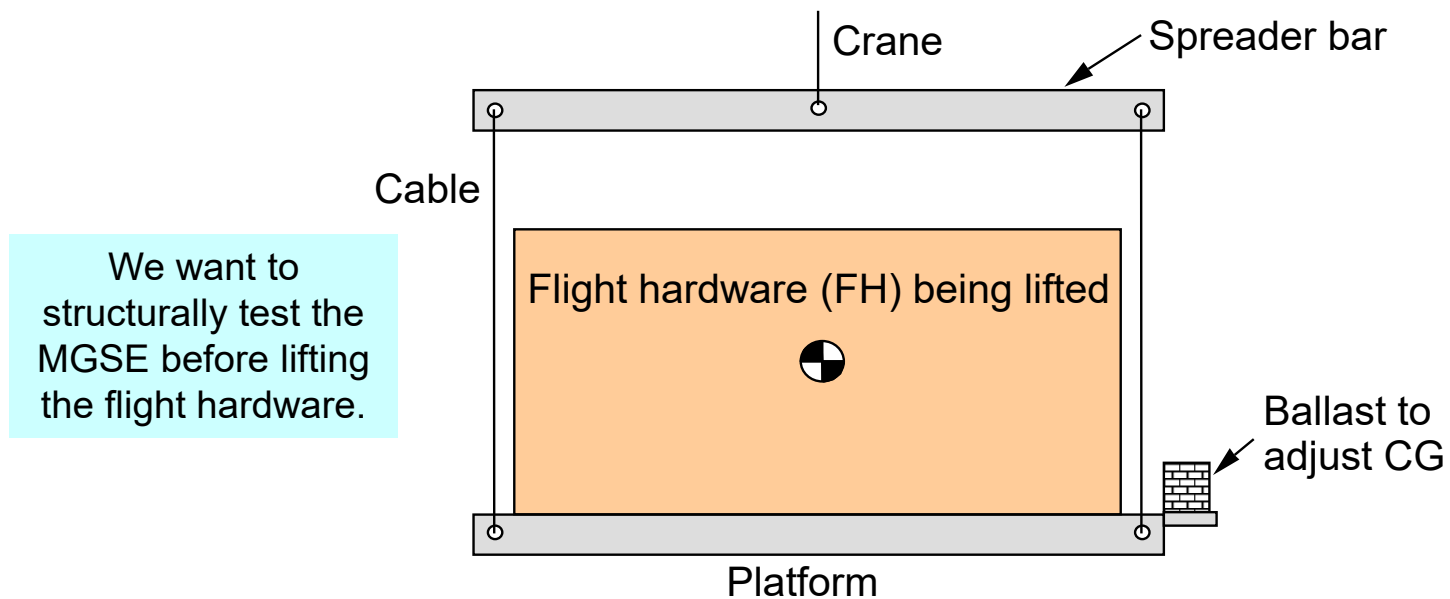
- A load-transformation matrix is a form of output-transformation matrix (OTM), a.k.a. data recovery matrix (DRM).
- OTMs can be generated from any collection of linear equations.
 - The same equations you may already be generating for spreadsheets.
- Examples:
 - Strain or displacement predictions: equations relating strain or displacement to unit applied loads for each direction in each potential load location.
 - With an OTM for predicted strains and displacements, you can revise load cases the day before test, and new data predictions will be generated in time for test start.
 - Reducing strain data: equations relating meaningful information, such as member loads, to measured strain.

Another Test Approach: Using a Crane and a Hydra Set[®]



Be Sure to Design the Right Test!

Example, mechanical ground support equipment (MGSE): lifting fixture/sling assembly



- Test approach:
1. Load a dead weight on the platform (>2 times the FH weight).
 2. Use a crane and Hydra Set, as shown on the previous page, to apply a vertical load of 2 times the FH weight (per MGSE proof-test criteria).

What's wrong with this test approach?

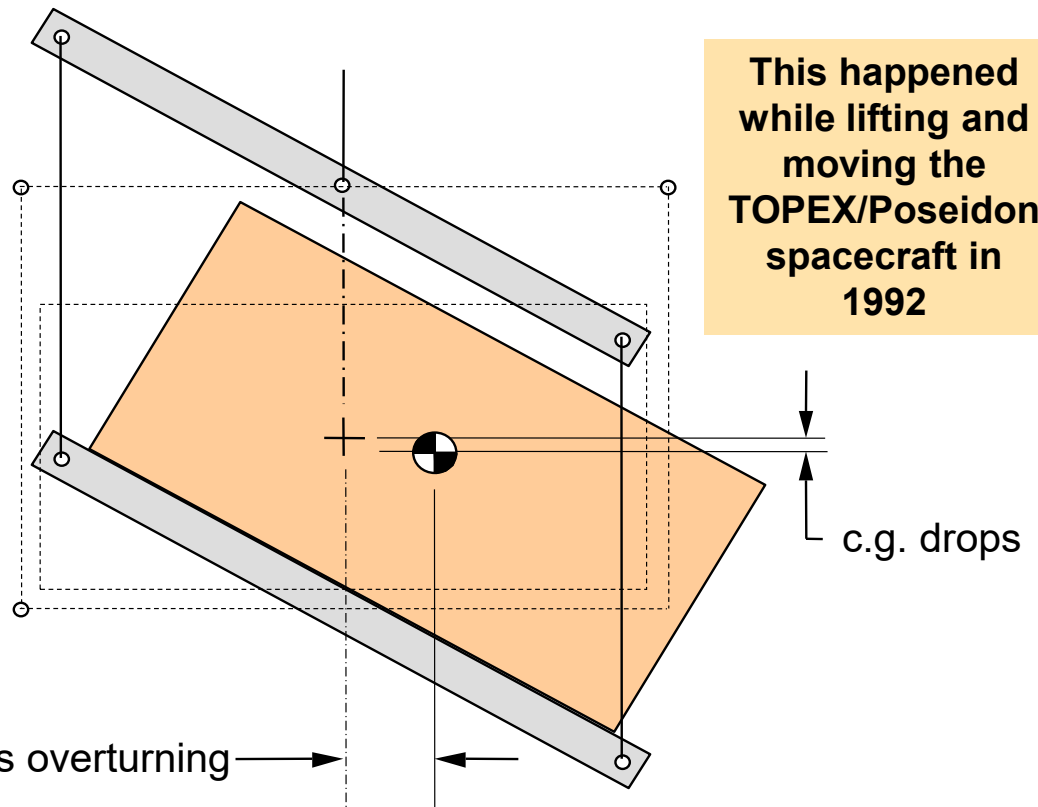
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Problem with Test Approach

**The sling assembly is unstable.
And the proposed test won't detect the deficiency!**

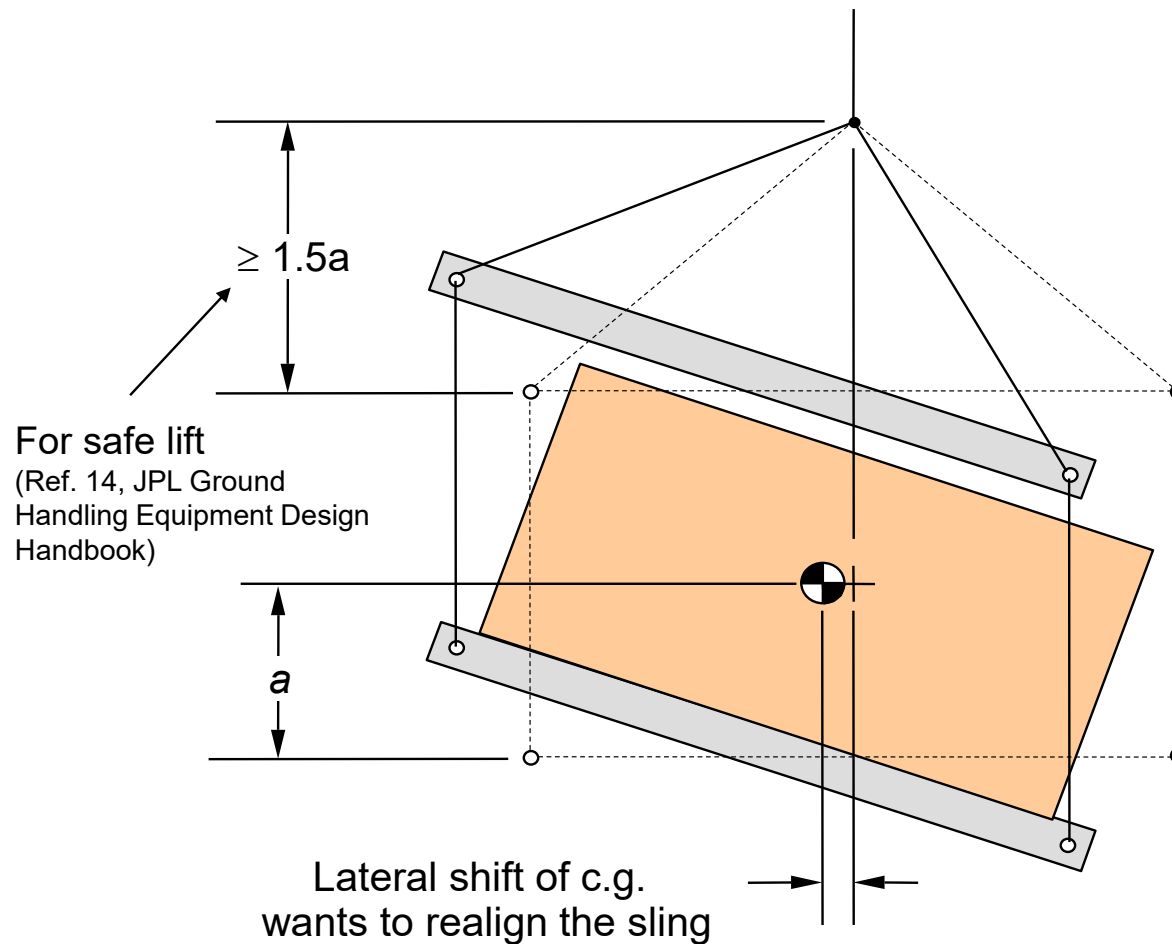
We've all heard the adage "test as you fly!"
With MGSE, test as you plan to use it!

The right test here would have been to lift and move a mass that has the same CG as the spacecraft and weighs some amount more (perhaps 1.5 times).



Lateral shift of c.g. causes overturning moment on spreader bar

Criterion for Safe Lift with a Sling Assembly



Key Points from This Section

- For tests needing multiple simultaneously applied loads, try to find a test facility that has an automated loads control system.
- Follow a sound process for designing load cases.
 - Identify potential failure modes and corresponding indicator (load) parameters.
 - Define target loads for the indicator parameters.
 - Make the test representative of the actual flight loads, achieving target overall loads at major interfaces and section cuts.
 - Tune the load cases to hit the target loads analytically, with an apples-to-apples approach.
- Make use of load transformation matrices both to calculate the target loads and to design the load cases.
- Test ground support structures the way they'll be used.

5. Testing on an Electrodynamic Shaker

- Introduction and Typical Configuration
- Limitations of Testing on a Shaker
- Fixture Design
- Deriving Loads from Measured Accelerations
- Sine Sweep Testing
- Deriving Damping from Test Data
- Sine Burst Testing
- Understanding Random Vibration
- Random Vibration Testing

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Introduction

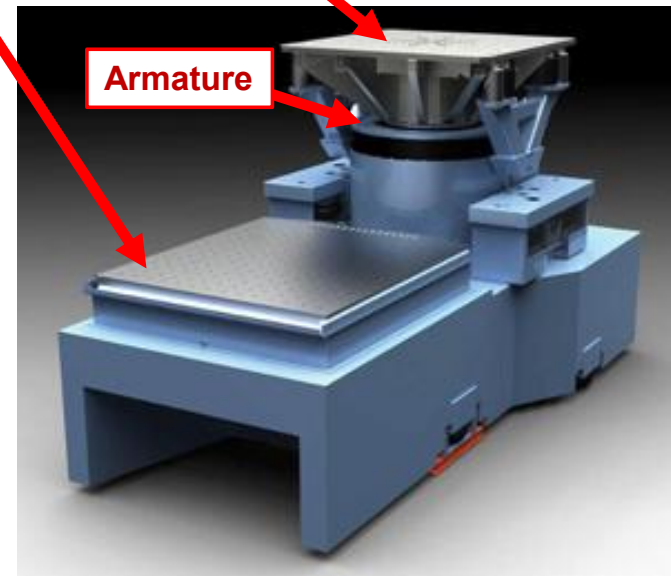
An **electrodynamic shaker** is a device that imparts force that is proportional to electric current that passes through a coil of wire suspended in a magnetic field.

Force causes acceleration, which 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)

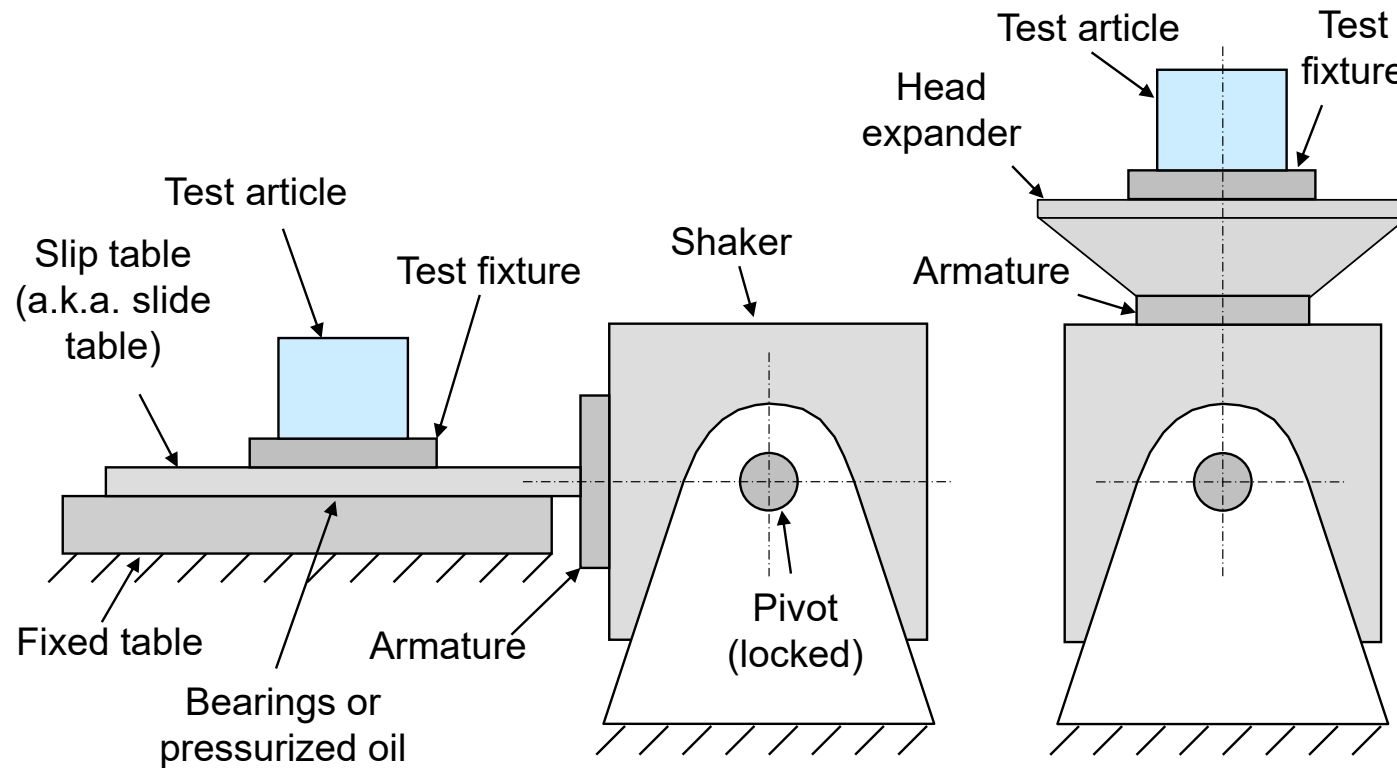
Acceleration is controlled by use of a **control channel**.

- Typically the average of two or more accelerometers placed at the base of the test article.
- Closed-loop system: Current is automatically adjusted until the control channel measures the desired acceleration.

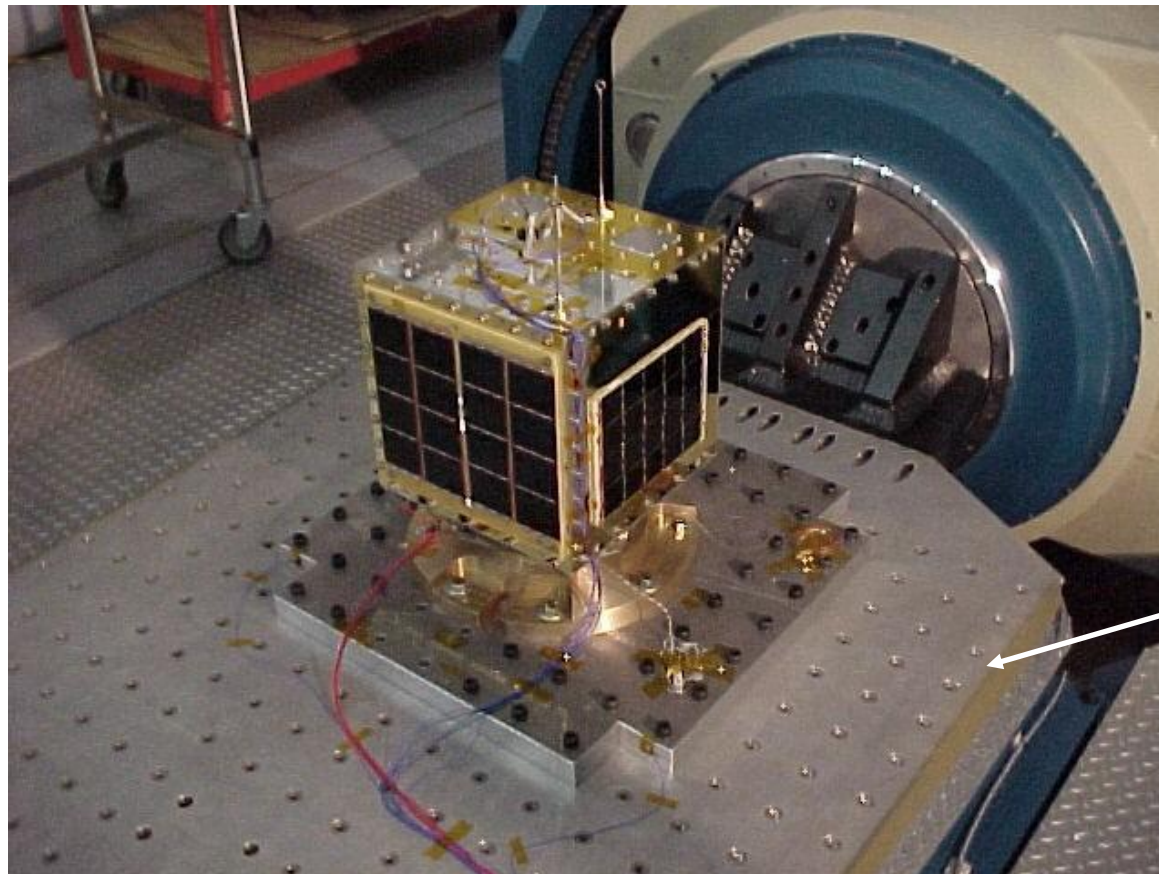


Dayton T. Brown (DTB) T5500-3 shaker

Typical Configuration for a Vibration Test



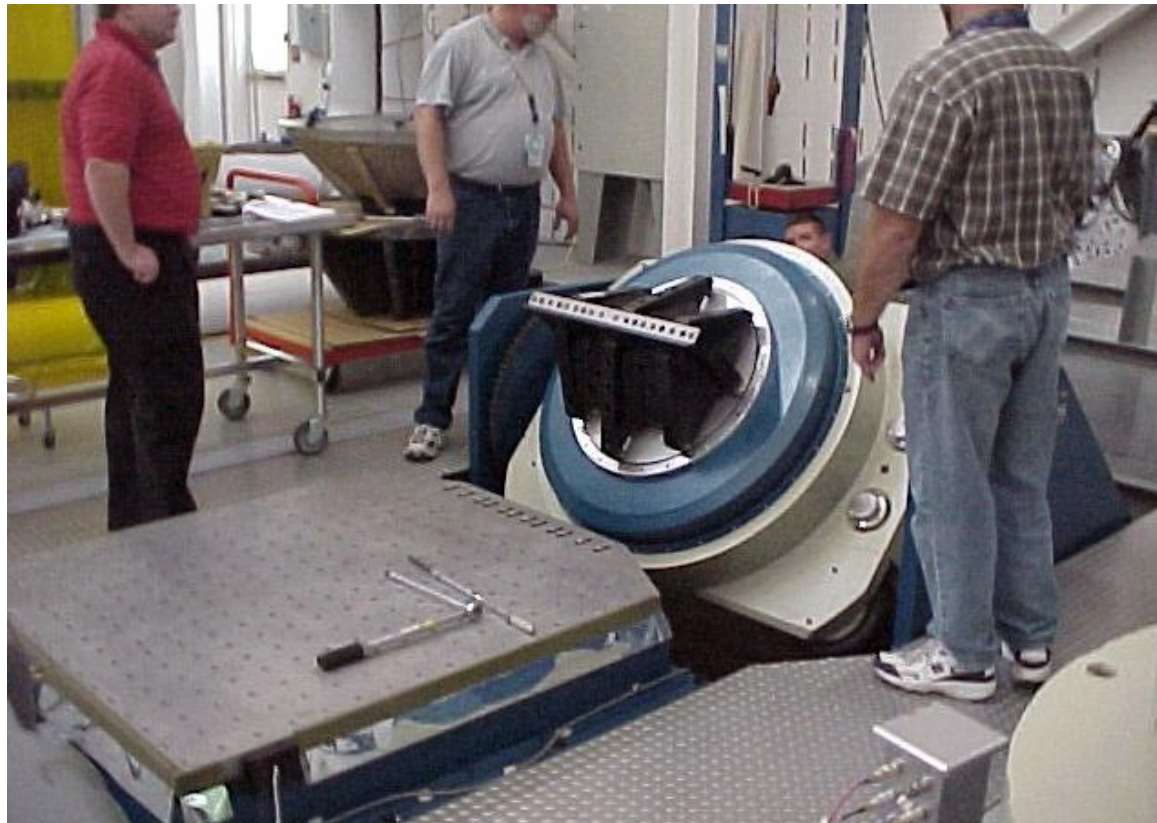
FalconSat-2 Ready for Vibration Testing



Slip table

Courtesy U.S. Air Force Academy and Kirtland Air Force Base

Shaker Being Rotated



Attaching FalconSat-2 for Vertical-axis Testing

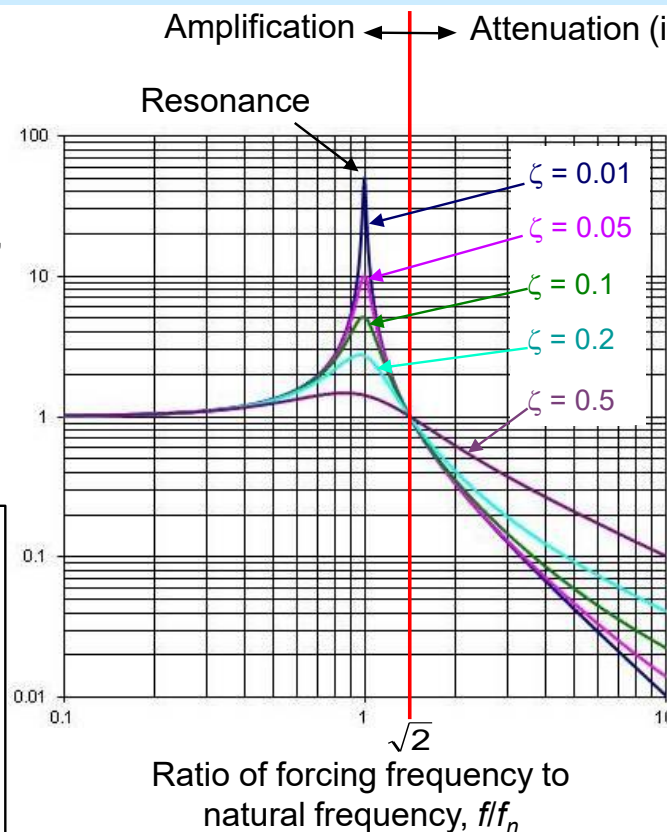
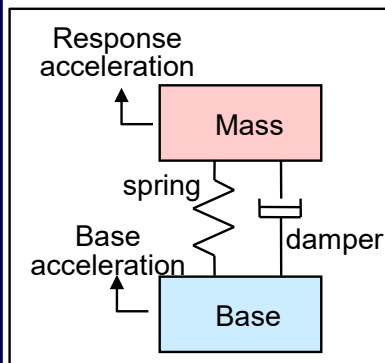


Head expander

Response of a Mass on a Spring to Base-driven Sinusoidal Vibration: Transmissibility

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

Transmissibility, TR
(ratio of peak response acceleration to peak base acceleration)



$$TR = \sqrt{\frac{1 + \left(\frac{2\zeta f}{f_n}\right)^2}{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \left(\frac{2\zeta f}{f_n}\right)^2}}$$

(Eq. 5-1)

where

f = Forcing frequency in Hz

f_n = System's natural frequency in Hz

ζ = Damping ratio

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

= response ratio (**gain**) at resonance for a mass on a spring

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.

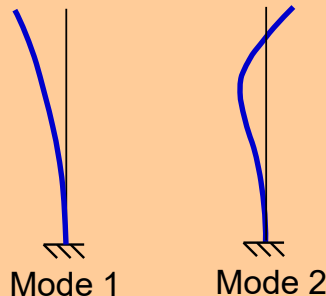
Stroke—the maximum range of displacement for the armature

- Limits the acceleration that can be achieved at a given frequency
- We'll address later in this section how stroke limits sine burst testing.

Limitations of 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 (most common, although multi-axis shakers exist)
- Translational input only; no angular (rotational) acceleration (most common)
- 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 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!

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. 15 – 17)

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.

Test Fixture Design

The **test fixture** 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 ...

- 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
- yet no heavier than it needs to be
 - The shaker has a force rating, and the fixture moves with the test article ($F = ma$), so mass of the test fixture limits the peak achievable acceleration.
- and to provide a flight-like interface
 - to ensure proper modal response and interface load distribution

These objectives often conflict.

To have the desired stiffness, a fixture may be quite heavy.

In addition, a rigid fixture doesn't provide a flight-like interface.

Let's look at some options for resolving this conflict ...

Resolving Conflicting Objectives for Test-fixture Design

Option 1: Mount the test article on the flight mounting structure—best if ...

- a. the flight mounting structure is available and will fit on the shaker
- b. and you can define the vibration environment at the base of the flight mounting structure

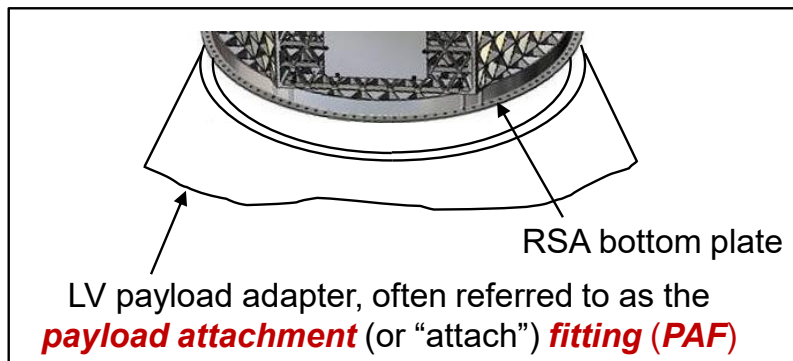
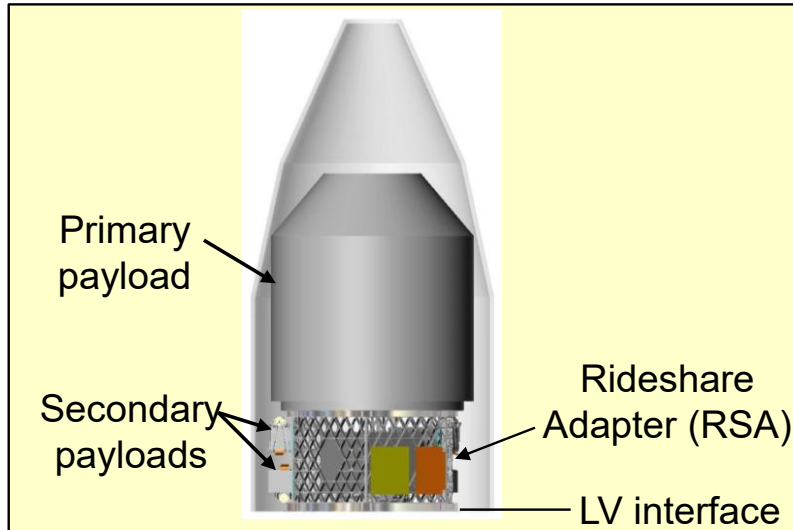
Option 2: Mount the test article on a structure that simulates the geometry and stiffness of the flight mounting structure—preferred when item 1b, above, is true and ...

- a. the actual flight mounting structure is not available
- b. and designing and building a simulator is practical

Option 3: Mount the test article directly on a rigid fixture that provides flight-like interface geometry—when the above options are not practical.

- A rigid fixture introduces the environment uniformly, as specified.
- Flight-like interface geometry avoids over-constraint for modal response (see upcoming example).
- But the interface probably won't be adequately tested because of excessive mounting stiffness.
 - May require structural verification by analysis alone

Example Problem 5-a: Designing a Test Fixture



Use of Rideshare Adapter images courtesy DNet Engineering

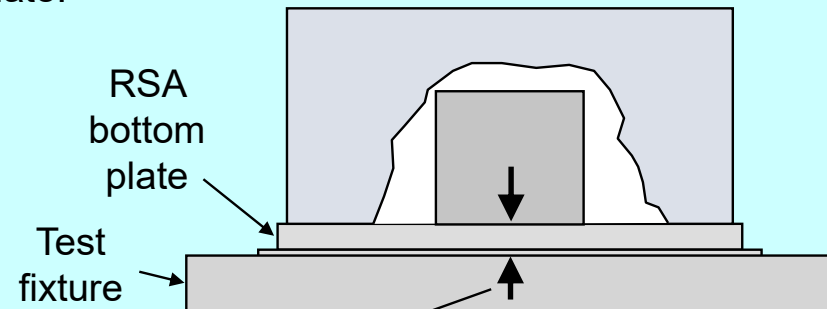
- For launch, the RSA will attach to a ring flange on the LV adapter, with 60 bolts on a 38.81" circle.
- Inside the RSA, mounted to a separation mechanism on the isogrid bottom plate, is a small spacecraft.
- We want to put a mass-loaded RSA (total weight of 650 lb) on a shaker for vibration testing.
 - Main objectives: Verify the RSA's strength and overall structural integrity with sine burst and random vibration testing.
- The shaker's slip table and head expander have threaded inserts lying on a 6" grid.
- The LV adapter is not available (and too big in diameter for the shaker).

Let's look at some considerations in designing the test fixture for mounting the RSA ...

Example Problem 5-a Continued: Considerations in Designing the Test Fixture

- High stiffness
- Uniform stiffness
 - To avoid putting more load in certain bolts
- Flight-like boundary conditions $\Rightarrow$
 - But using a flight-like conical adapter under the RSA is not practical (too large in diameter at the base for the shaker)
- Weight
 - Easier to handle if lighter
 - Will affect the peak acceleration based on the shaker's force rating
- How to pick it up and handle it

Problem: If we test the RSA mounted to a solid surface (fixture), the fixture will provide non-flight-like boundary conditions for the RSA's bottom plate.



A compressive load on the bottom plate from the internally mounted payload is reacted directly by the fixture. The fixture prevents the plate from bending as it would during launch.

We'll have to address this problem in order to ensure a good test.

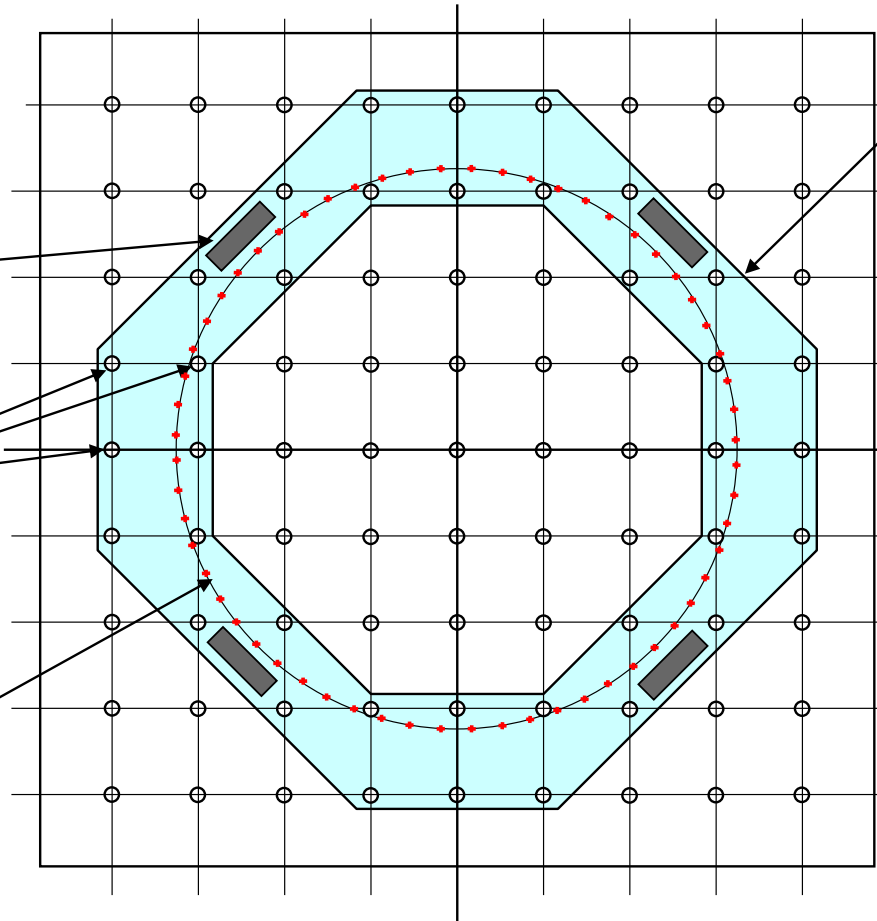
Example Problem 5-a Continued: Design of the Test Fixture for the RSA

6" grid for threaded inserts in the slip table and the head expander

Bolt-on handles for crane straps or hands

Counterbored through holes matching the 6" grid

38.81" diameter circle of 60 evenly spaced $\frac{1}{4}$ " bolts for the RSA



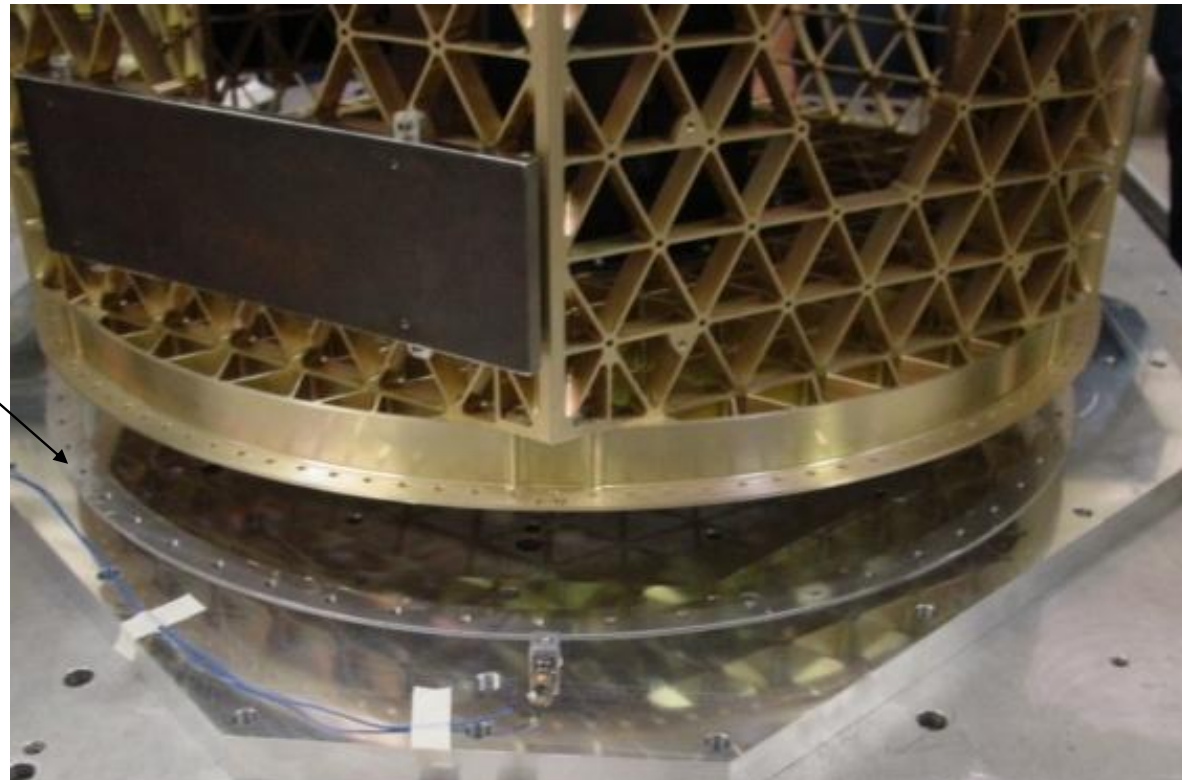
Fixture
(6061-T651
aluminum,
1.40" thick,
120 lbs)

**Still doesn't
provide the
right
boundary
conditions for
the RSA,
though**

Example Problem 5-a Continued: Making the RSA Mounting Interface More Flight-like

Our solution: Add a spacer ring cut from sheet metal to raise the RSA and allow contact only on the ring.

Another option was to machine a proud (raised) ring surface into the test fixture.



Use of a spacer ring gave more linear, more flight-like interface, but it did not compensate for the absence of the PAF. Properly testing the interface, particularly the effects on the PAF, requires a static loads test with the RSA on the PAF.

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%!

Designing Mass Simulators

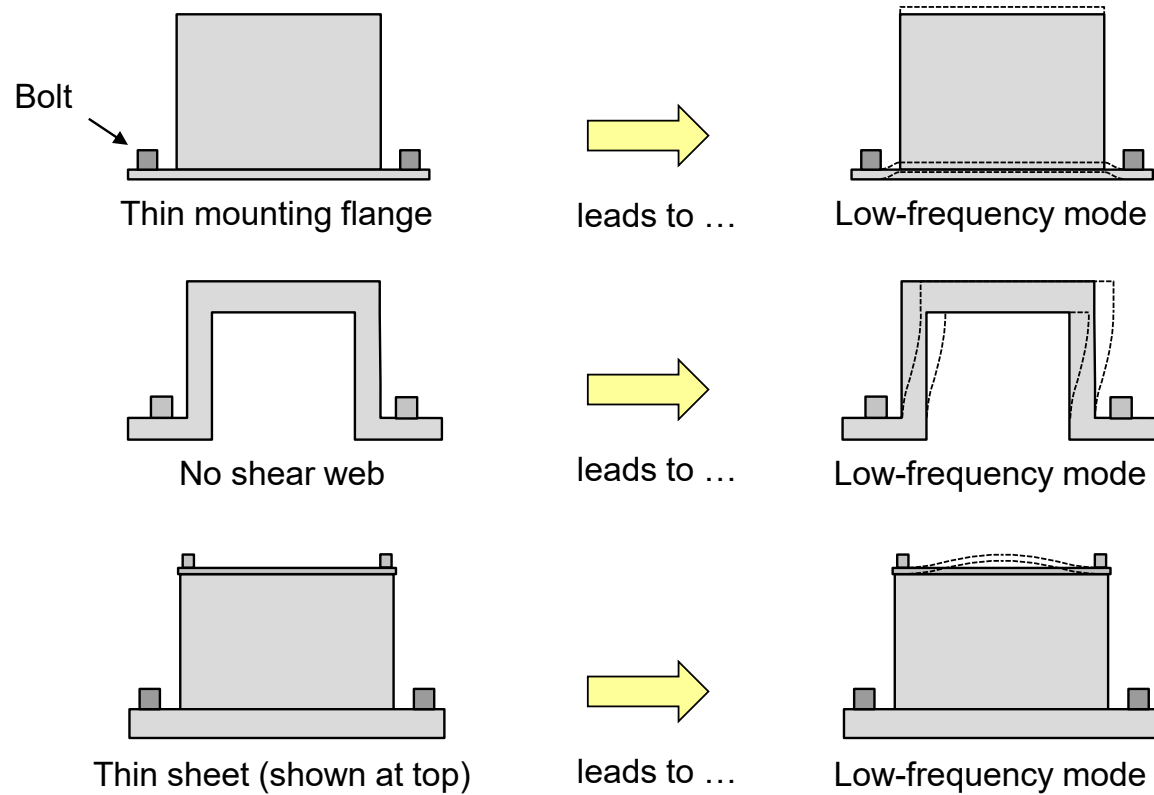
In most cases, we want mass simulators to add mass without otherwise affecting the test article's key modes of vibration.

- 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 the Rideshare Adapter (RSA).

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



- 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.

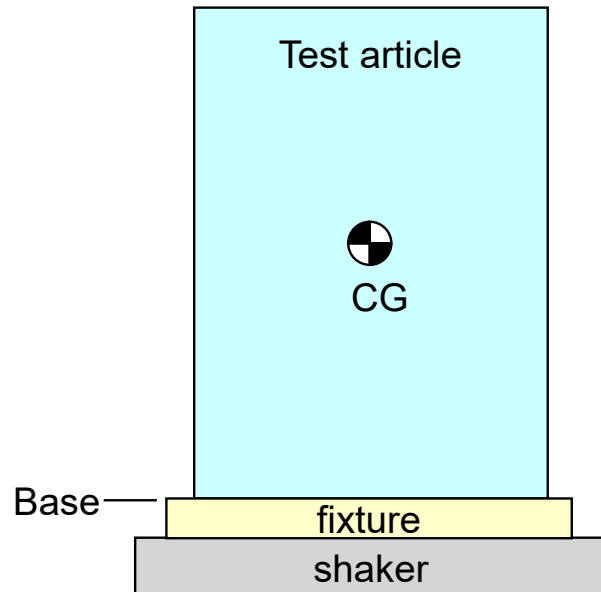
Example: Part of the structure is loaded significantly by modal response of a missing component.

- 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.

Example Problem 5-b: Determining Base Force and Moment during Test

Hypothetical situation:



Your flight structure 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 ...

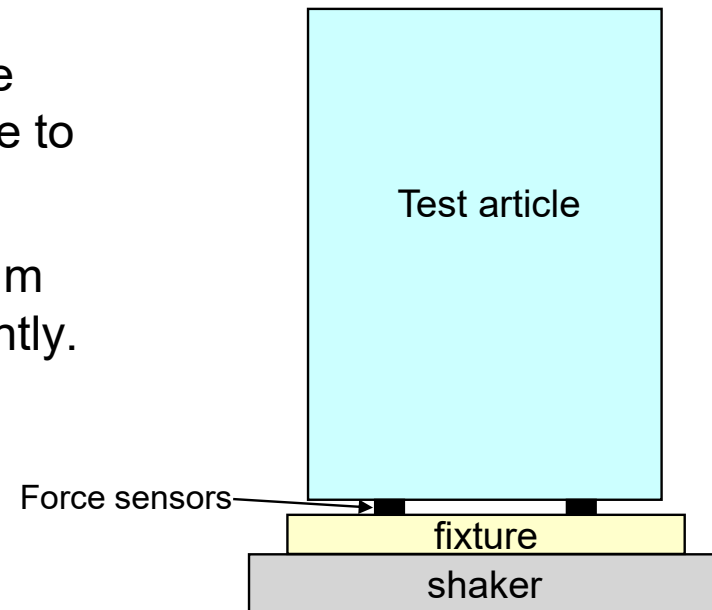
continued

Example Problem 5-b, 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 5-b, continued:
Option 2: Put an Accelerometer at the CG

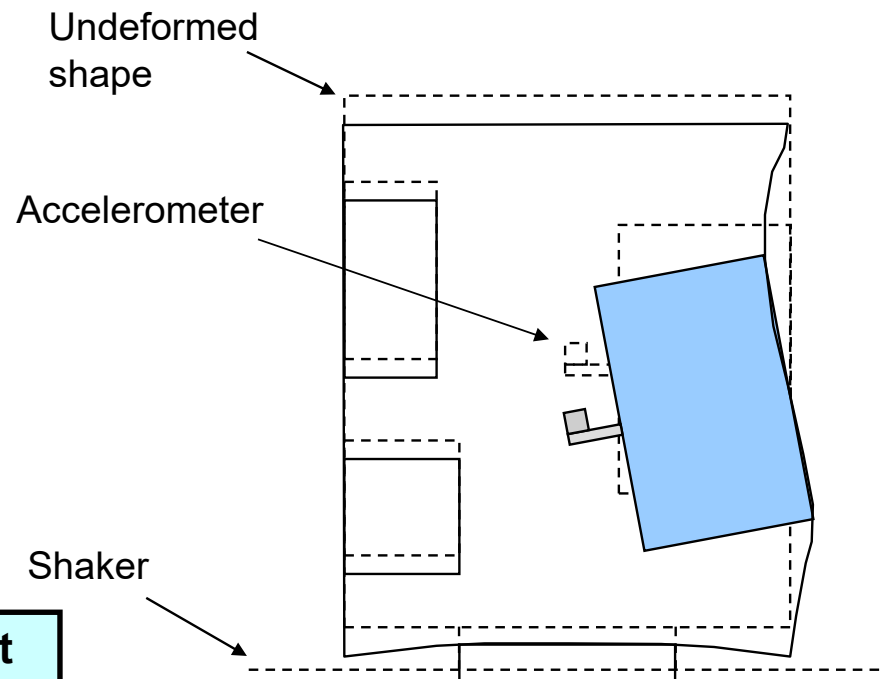
We place an accelerometer at the test article's CG by cantilevering a mounting platform off a nearby component that mounts to a side wall.

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!

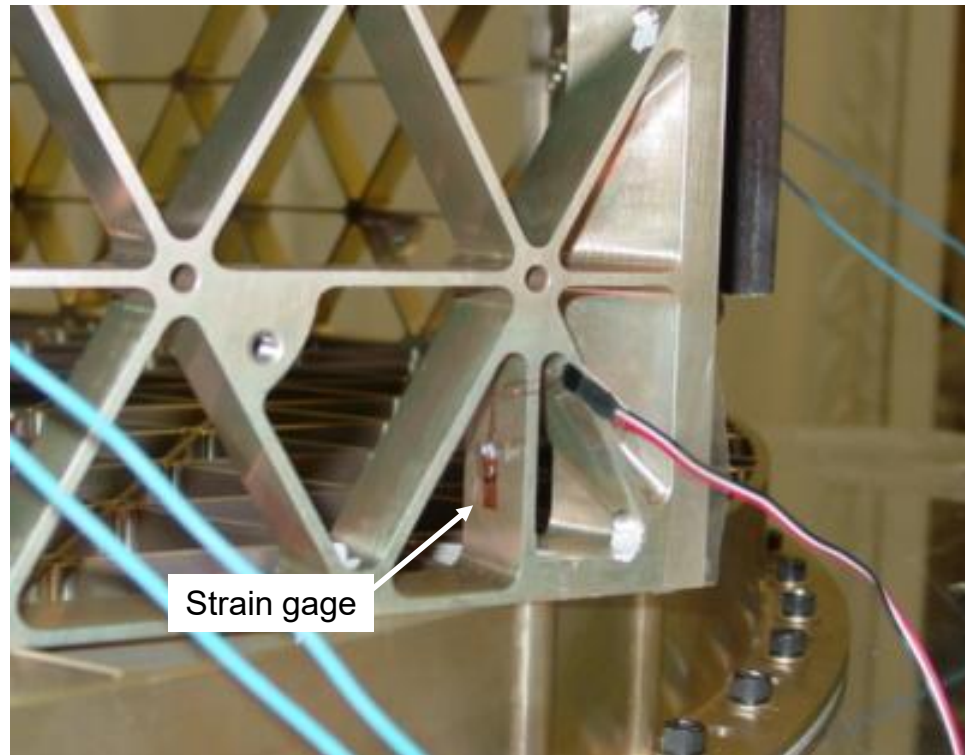


continued

Example Problem 5-b, continued: Option 3: Use Strain Gages

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.

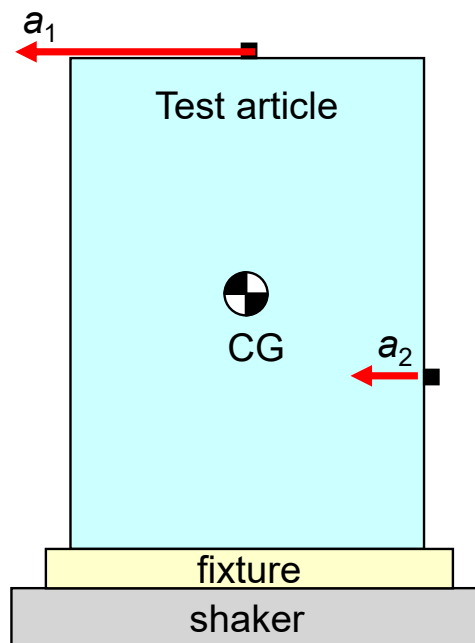
The calibration applied only for low-frequency measurements.



Strain gage

continued

Example Problem 5-b continued:
Option 4: Use FEM-derived Transfer Functions

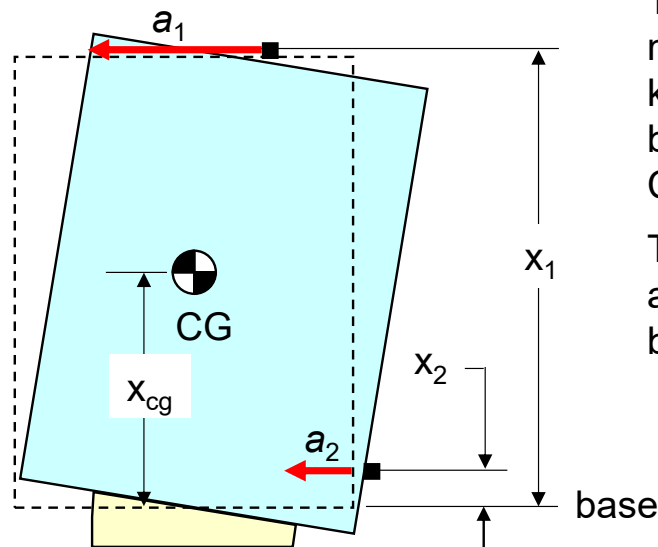


With the FEM, calculate the frequency-dependent transfer functions relating base loads to measured acceleration at one or more locations.

- 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 5-b, 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 .

$$a_{cg} = a_2 + (a_1 - a_2) \frac{x_{cg} - x_2}{x_1 - x_2} \quad (\text{Eq. 5-2})$$

With a_1 and a_2 in g units,

$$R = \frac{(a_1 - a_2)g}{x_1 - x_2} \quad (\text{Eq. 5-3})$$

(units: rad/s^2)

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

continued

Example Problem 5-b, conclusion:
Option 4 Without Use of FEM Transfer Functions: Rocking Mode, continued

$$\text{Base moment, } M \approx ma_{cg}x_{cg} + RJ \quad (\text{Eq. 5-4})$$

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.

Typical Objectives of Low-level Sine Sweeps

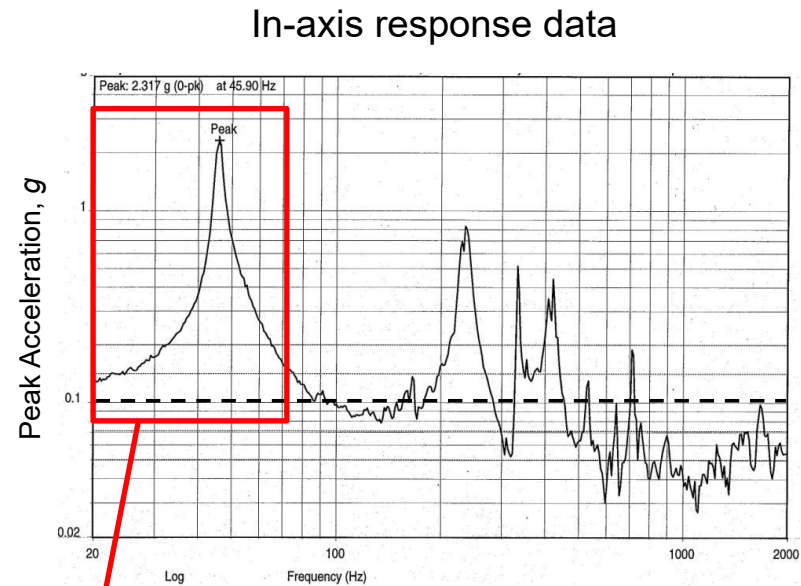
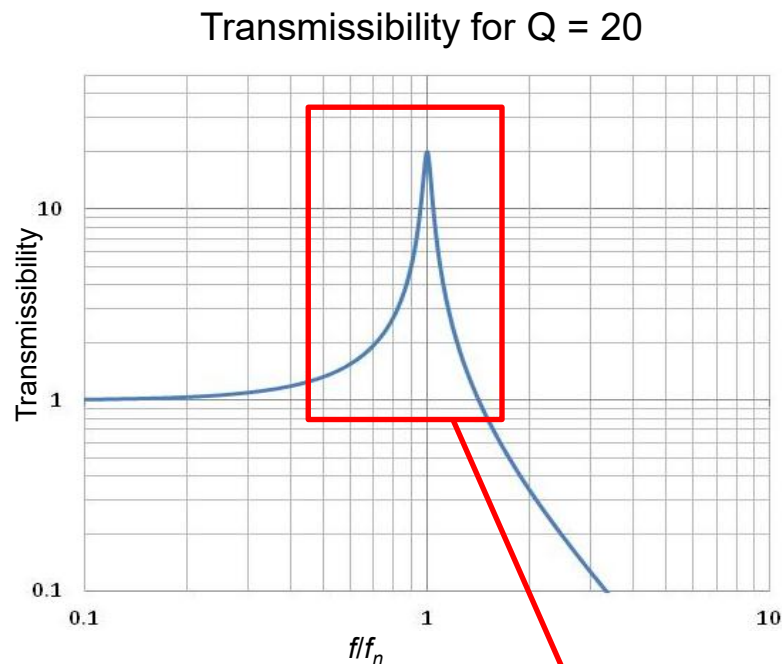
- Determine natural frequencies.
- Acquire information to help you understand mode shapes.
 - Compare relative response accelerations at various locations.
- Determine modal damping.
- Allow an assessment of structural integrity as part of the pass/fail criteria for high-level tests (e.g., random vibration 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)
 - 0.1 g suggested, and maybe lower yet for large test articles
 - You want to avoid unnecessary fatigue damage
- 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

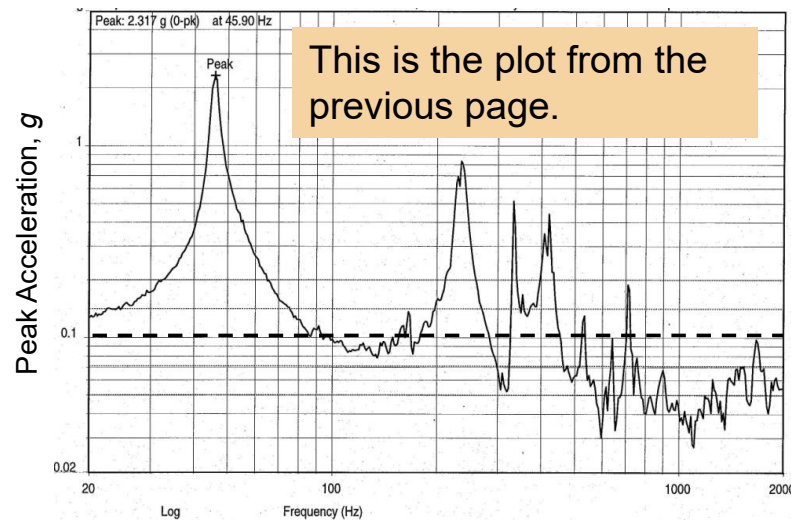
An **octave** is a doubling of frequency.

Comparing Sine Sweep Data with the Transmissibility Plot



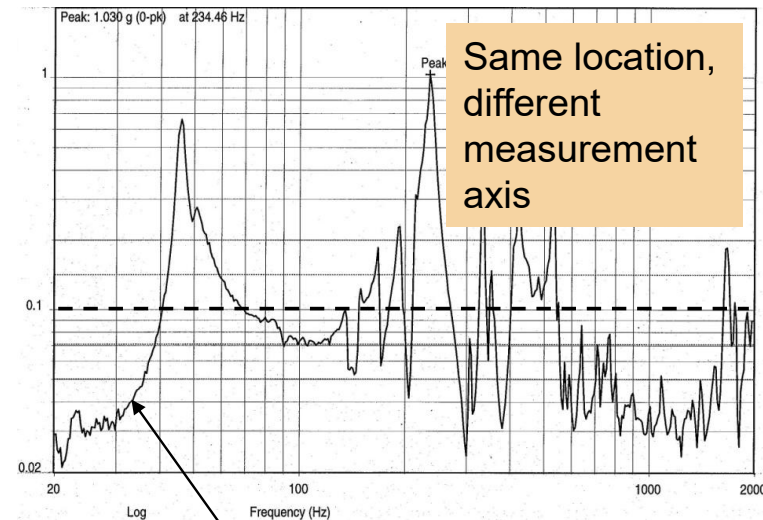
Note the similarity between the response and transmissibility plots in the red-boxed regions.

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.

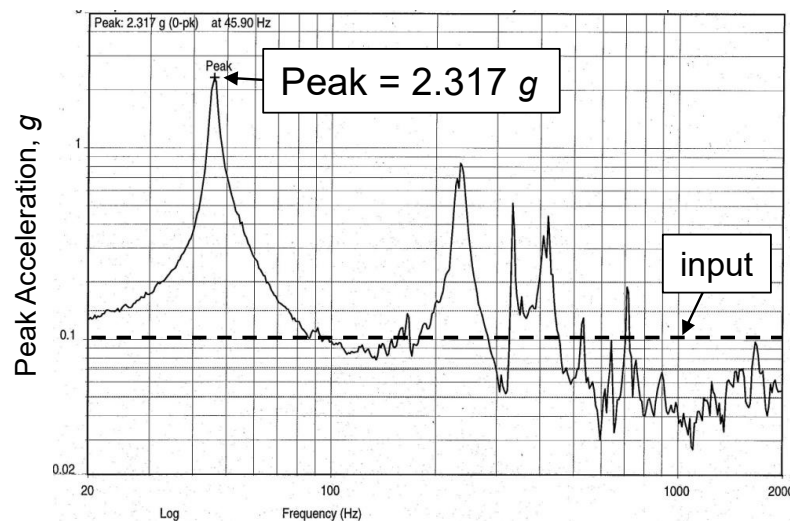
Note: In-axis response can be lower than input at 20 Hz only if the test article has a mode of vibration < 20 Hz.

Deriving Damping from Sine Sweep Data

Let's say we want to derive damping for the fundamental mode from the plot shown below.

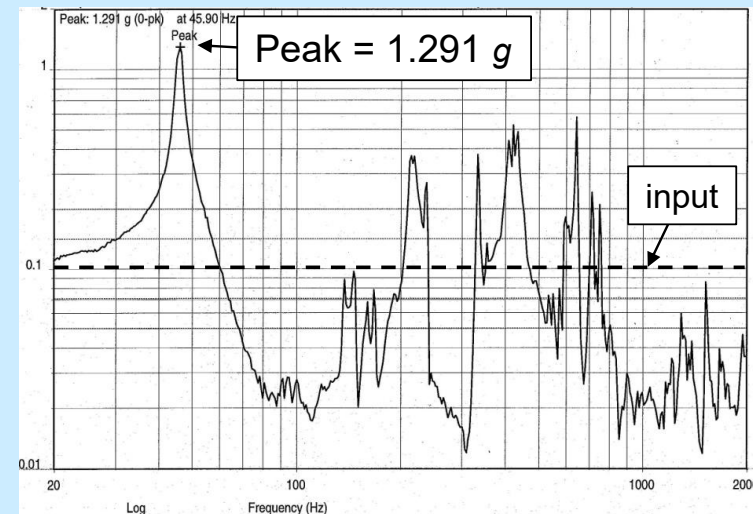
We might think that $Q = 2.317/0.1 \approx 23$

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 5-c: 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,

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

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

$$Q \approx \frac{f_n}{\Delta f_{hp}} = \frac{45.9}{2.5} = 18 \quad (\text{Eq. 5-5})$$

where

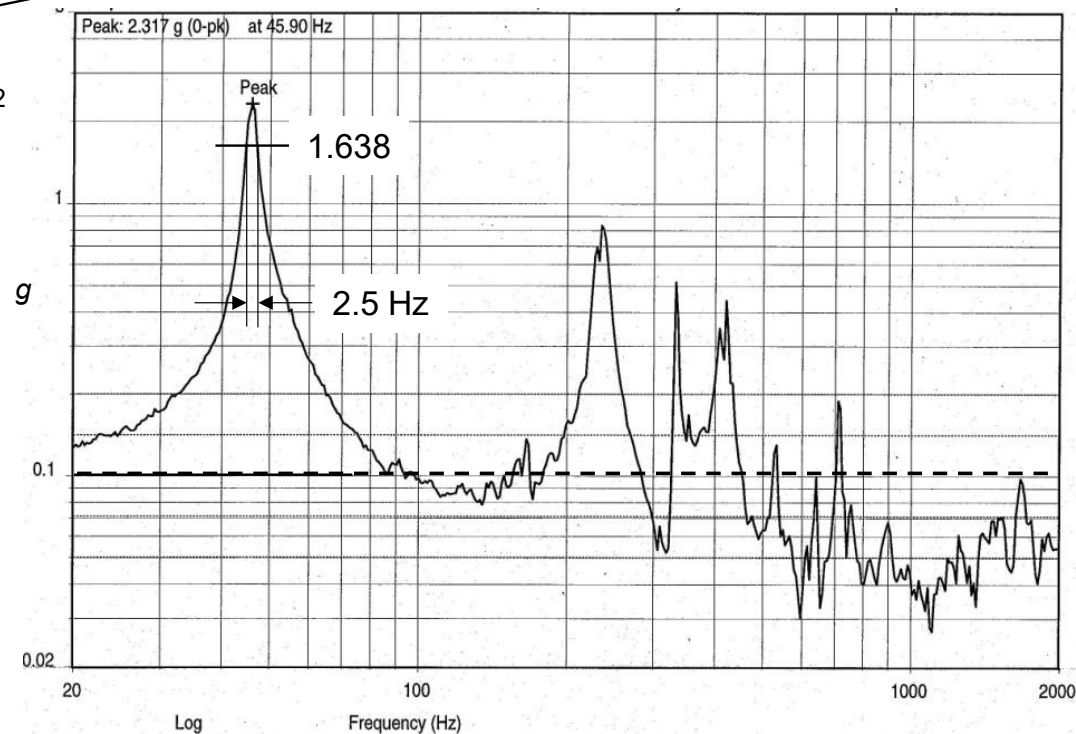
f_n = natural frequency

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

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

for the fundamental mode.

Power in this sense is acceleration squared.



continued

Example Problem 5-c, continued: Half-Power Method of Deriving Damping

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

Maximum acceleration,

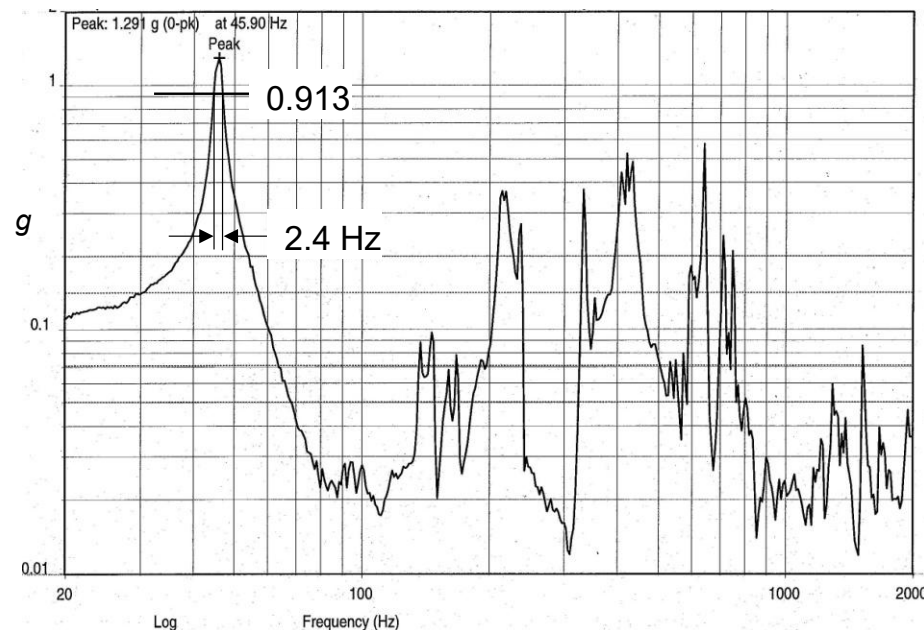
$$a_{\max} = 1.291 \text{ g}$$

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).

Note: The half-power method of determining damping becomes less accurate for higher damping. Reference 18, NASA/TM-2016-218227 suggests using this method only when $\zeta \leq 5\%$ of critical.

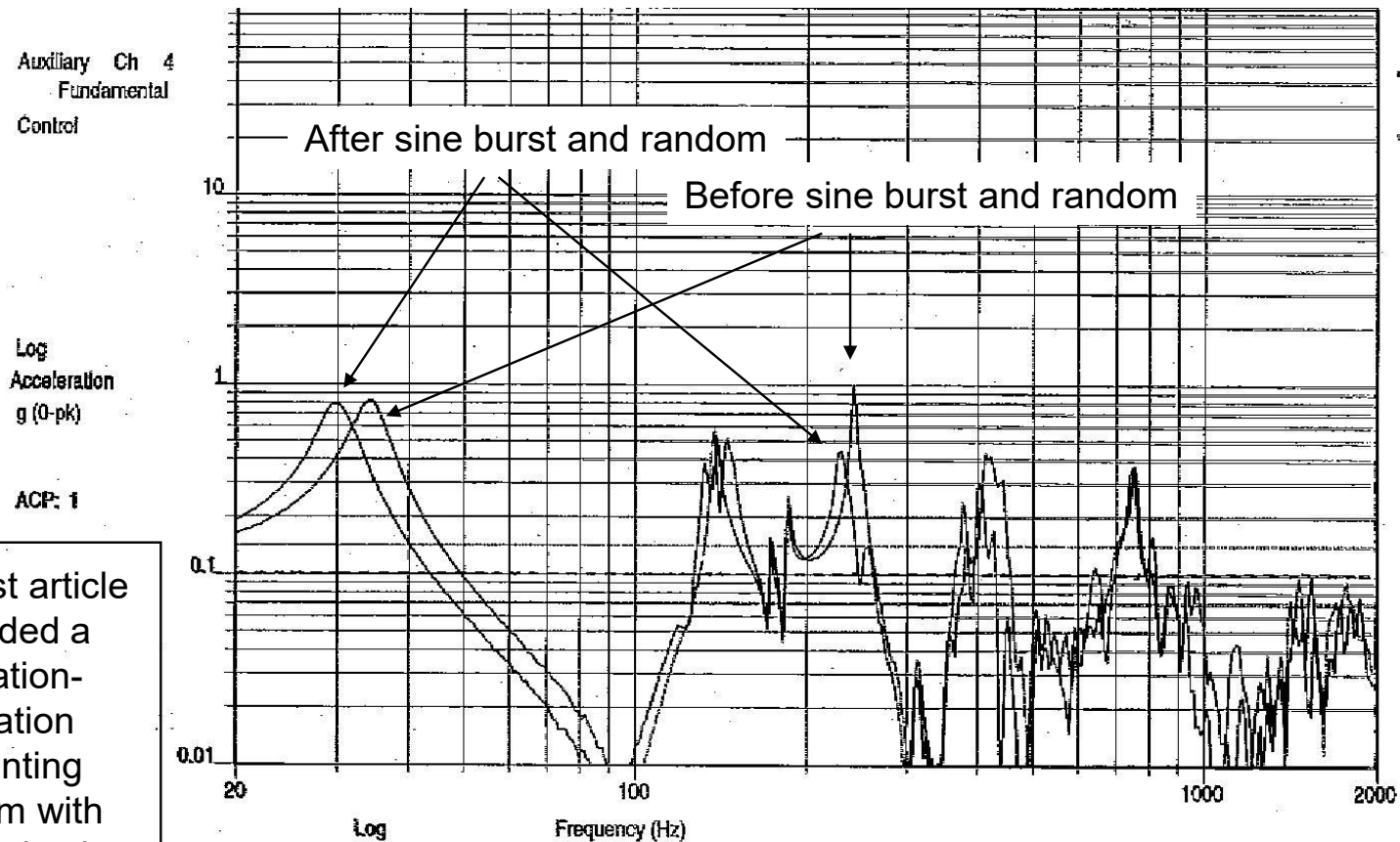
Criteria Related to Low-level Sine Sweeps

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

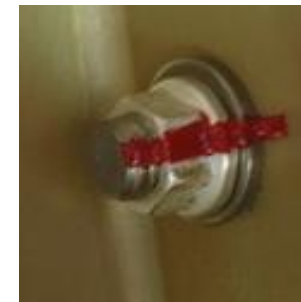


The test article included a vibration-isolation mounting system with visco-elastic materials.

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
 - Check bolt torque after the test, retorque if necessary, and 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



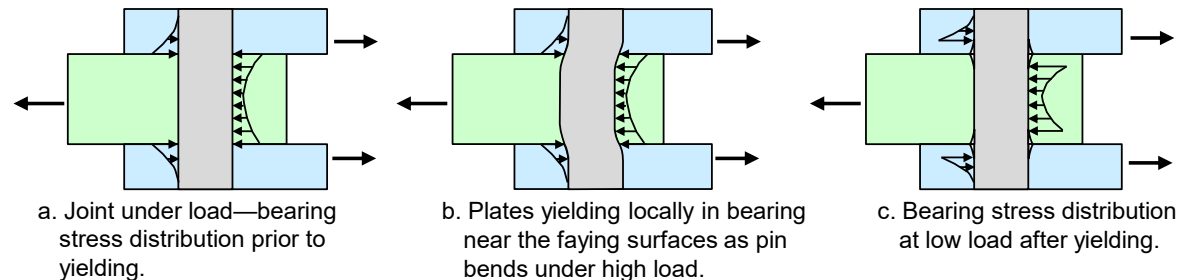
Torque striping to detect nut rotation in a vibration test

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.

Lug-and-clevis
example:



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.

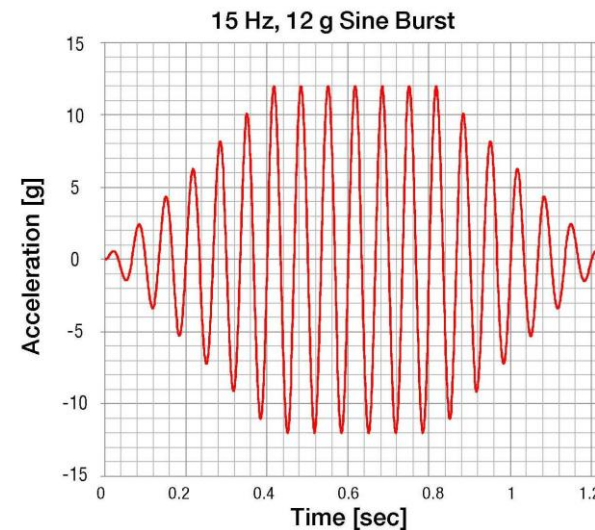
If the structure must maintain alignment of component interfaces, you'll need to find some way of detecting detrimental deformation other than low-level sine sweep comparisons, such as performance testing for alignment.

Sine Burst Testing

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** is the central core (body) structure of the spacecraft and the structure interfacing with the launch vehicle.



NASA wants to see at least 5 cycles at full load (Ref. 19). To minimize fatigue damage, don't go much beyond that.

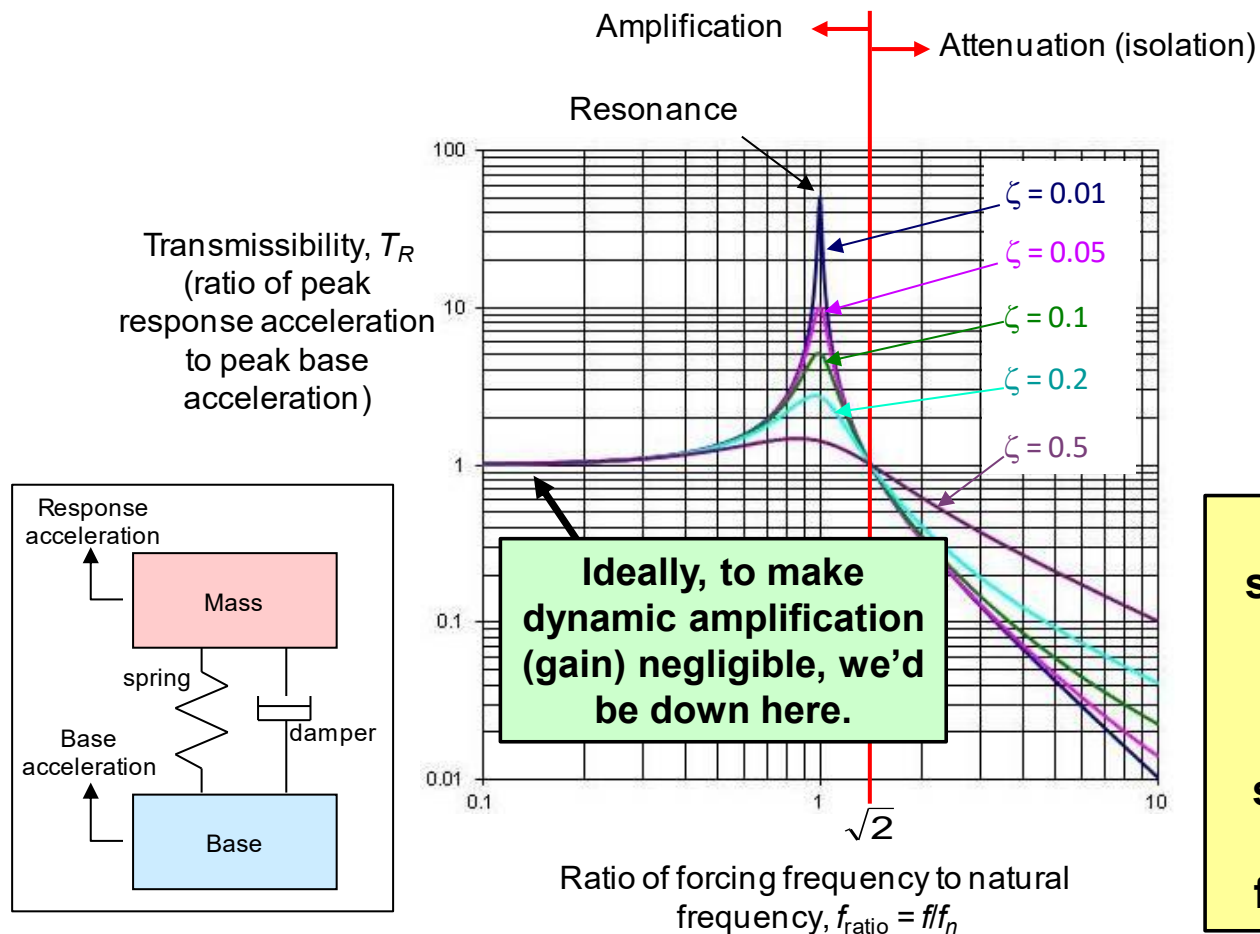
Designing a Sine Burst Test

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.

Addressed in Sec. 2
(Example 2-b and
Class Problem 2-1)

Addressed in the
following pages

Selecting the Sine Burst Frequency



But the shaker may not have enough stroke for such a low input 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)

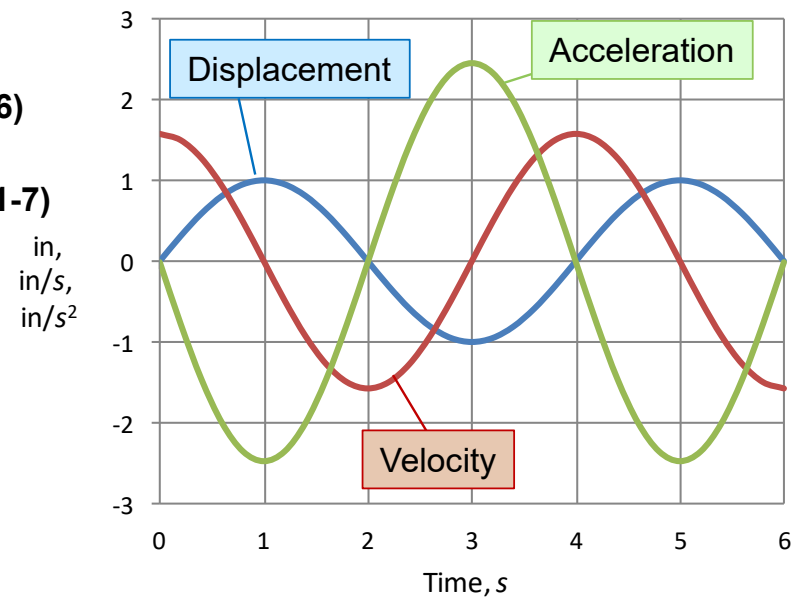
where

$\delta_{\max}$ 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. 5-6)

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. 5-6, ...

$$\text{Test frequency, } f \geq \frac{1}{2\pi} \sqrt{\frac{A_{\max}}{0.8\delta_{\max}}} \quad (\text{Eq. 5-7})$$

*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

$f, \text{ Hz}$	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

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

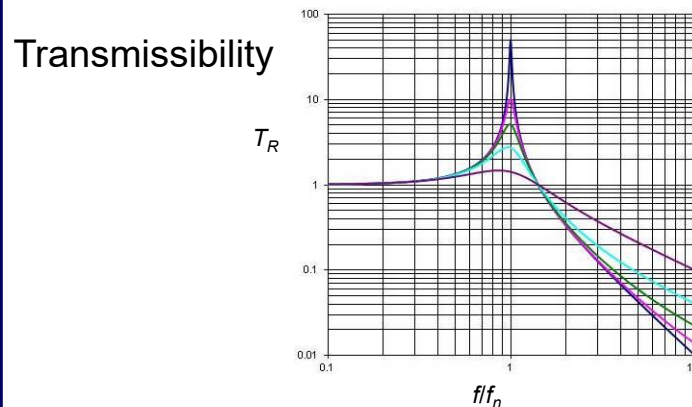
From the example from the previous page:

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

$$T_R = \frac{1 + \left(\frac{2\zeta f}{f_n}\right)^2}{\sqrt{\left[1 - \left(\frac{f}{f_n}\right)^2\right]^2 + \left(\frac{2\zeta f}{f_n}\right)^2}} \quad (\text{Ref. Eq. 5-1})$$

These values apply to a mass on a spring but can be used to estimate dynamic gain at a test article'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.

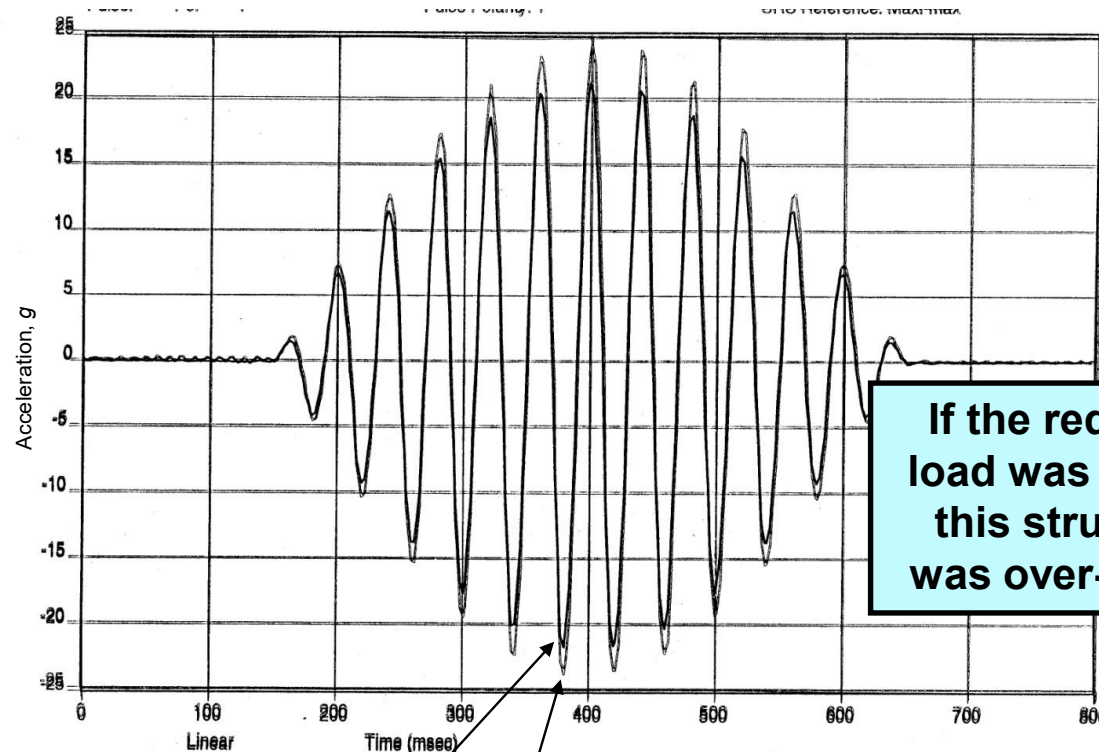
- However, for the example on the previous page, we can't simply reduce the input to $10/1.33 = 7.5$ g unless our structure is truly a mass on a spring.
- Actual response acceleration will vary, depending on the mode shape.
- We can usually meet the objective of verifying strength if the required base 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)



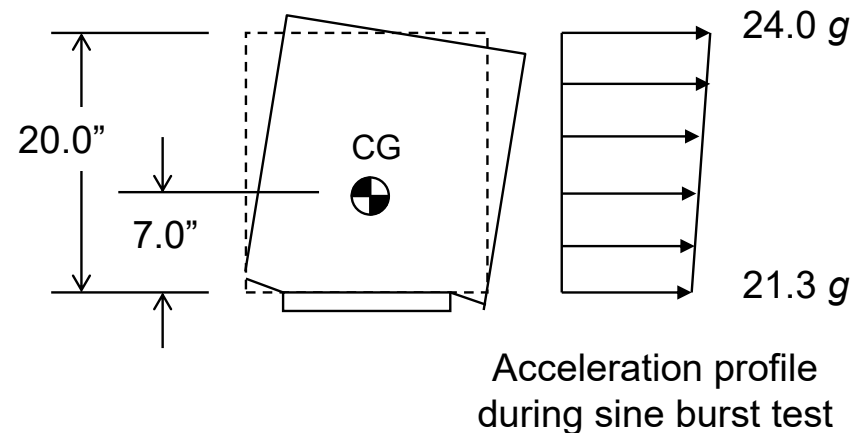
21.3-g peak input

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

If the required load was 21.3 g, this structure was over-tested

Class Problem 5-1: 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, $J = 30.0 \text{ lb-s}^2\text{-in.}$

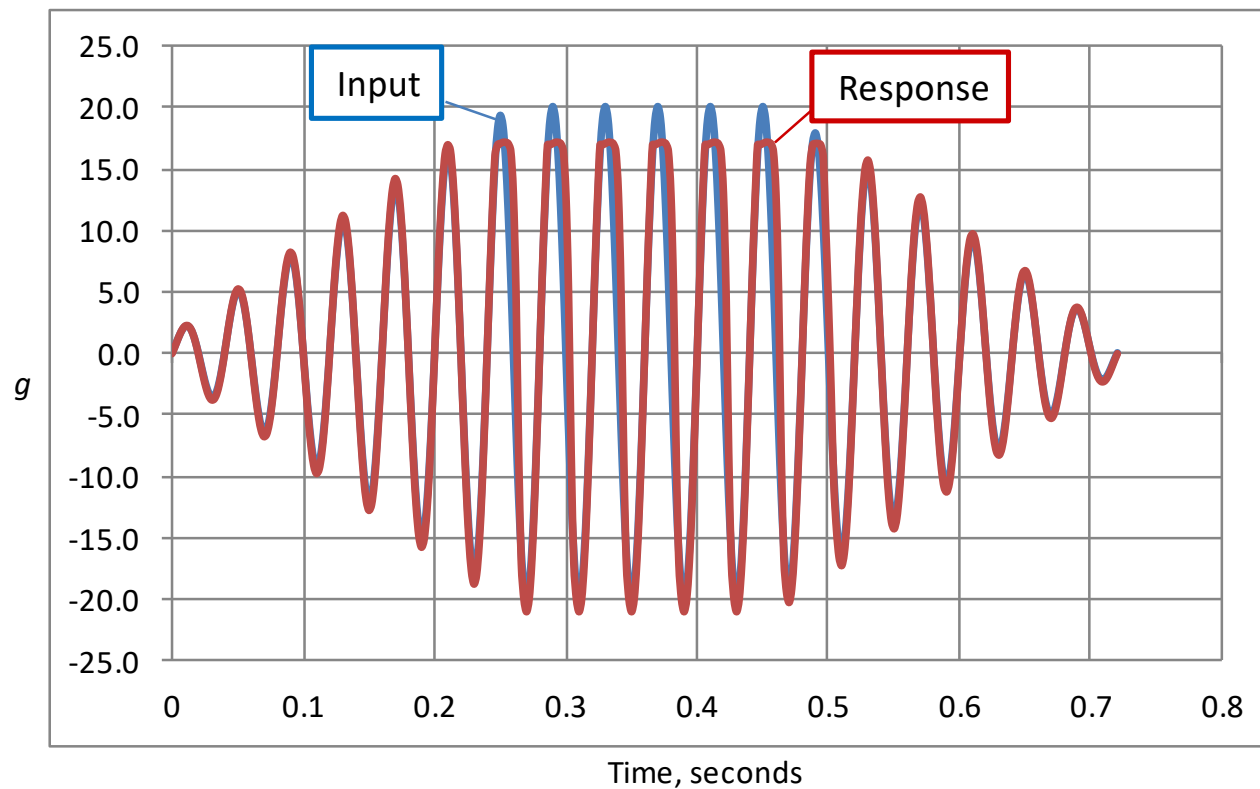


Problem statement: Given that the objective of this test is to verify strength of this structure at its mounting interface, derive the peak input acceleration for a 25-Hz sine burst that causes the same interface (base) moment as does a 21.3 g quasi-static load.

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

Use Equations 5-2 through 5-4.

How Would You Interpret This Hypothetical Sine Burst Test Data?



Answer: Something in the structure may be buckling under compression, thus preventing the response location from seeing the full input acceleration.

Special Considerations Related to Sine Burst Test Effectiveness

- As discussed in Sec. 1, 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.

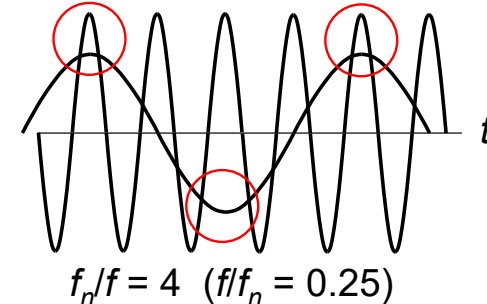
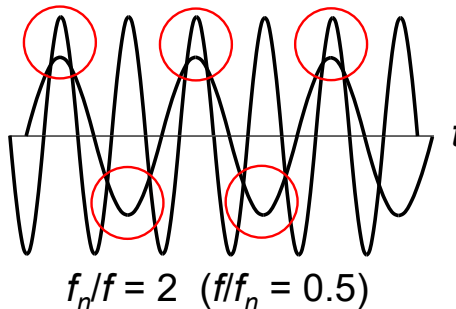
If the above criterion, as applicable, isn’t met, try increasing input levels to see if the target responses are achieved.

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

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

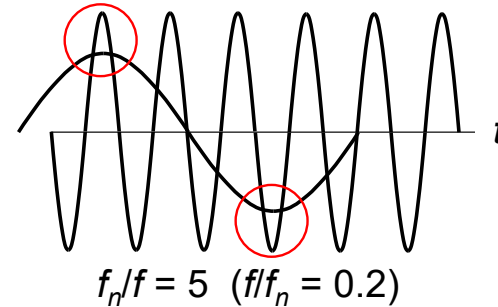
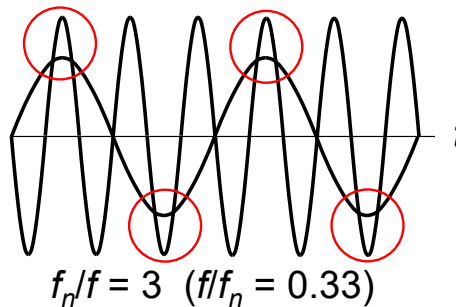
Even integer multiples: →

- No concern
- Peaks do not align



Odd integer multiples: →

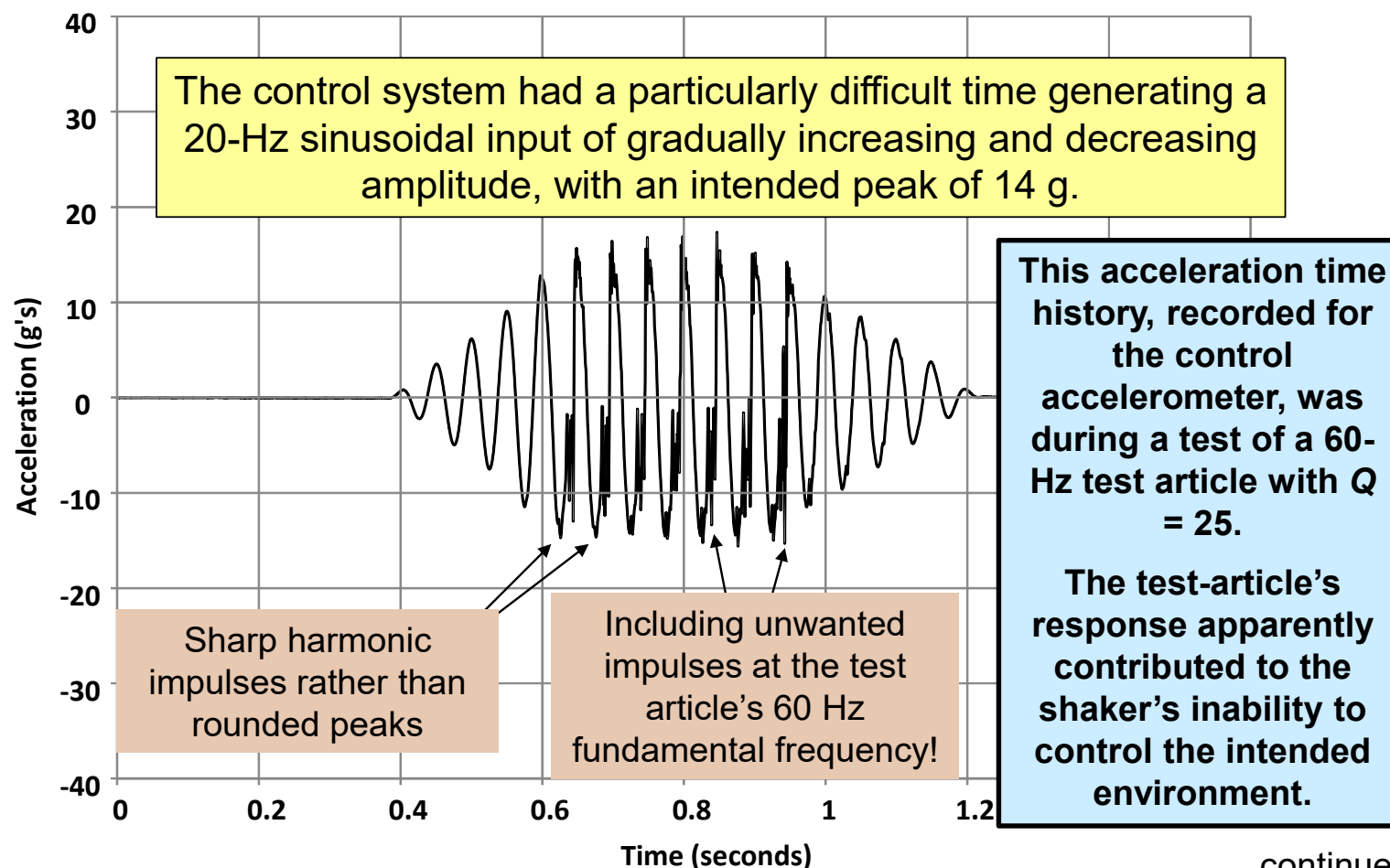
- Avoid
- Peaks align



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.

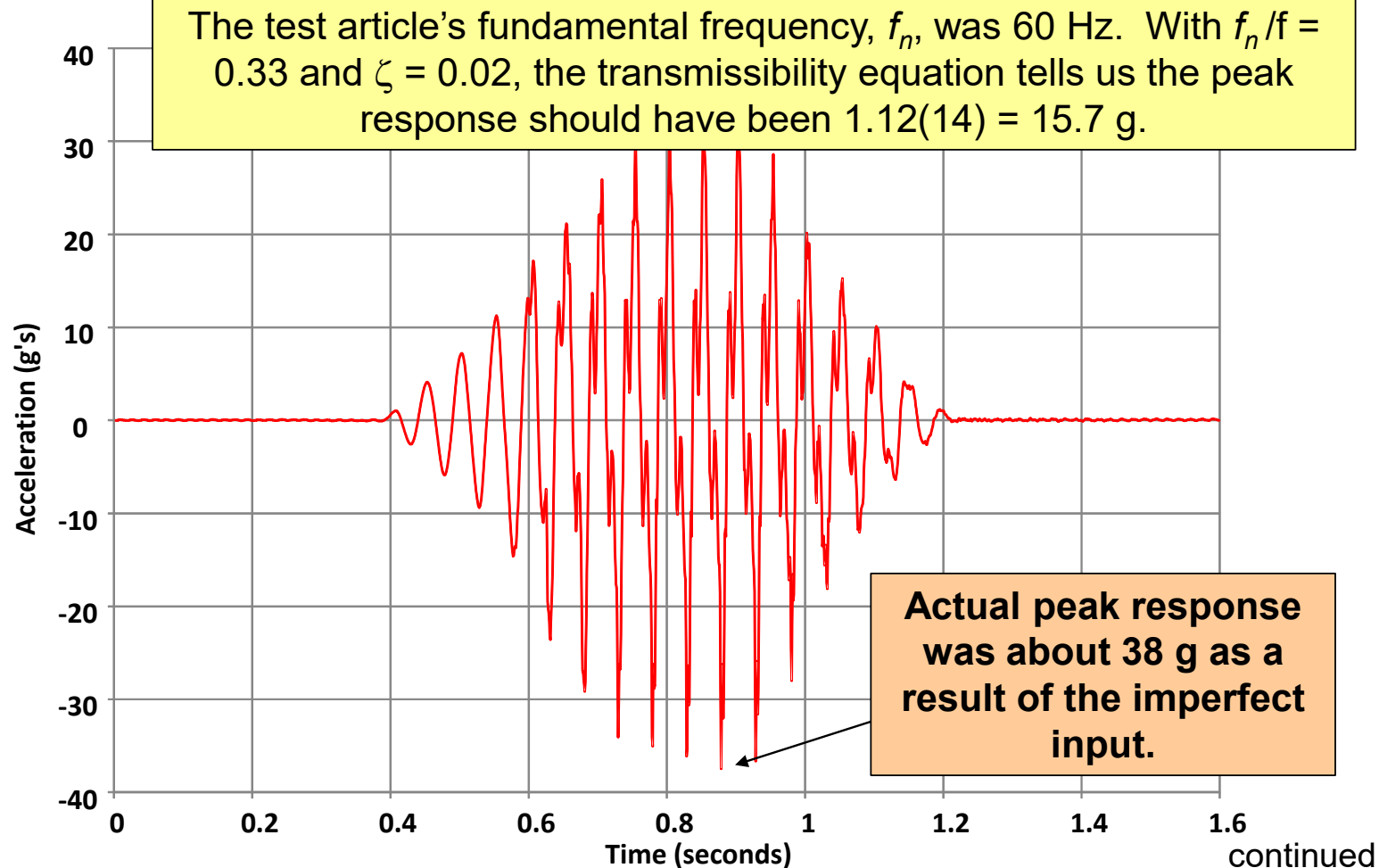
In addition, if f_n is an odd multiple of f , the test article's response is more likely to prevent the control system from generating a sinusoidal input.

Example from an Actual Case History: Imperfect Sine Burst Input



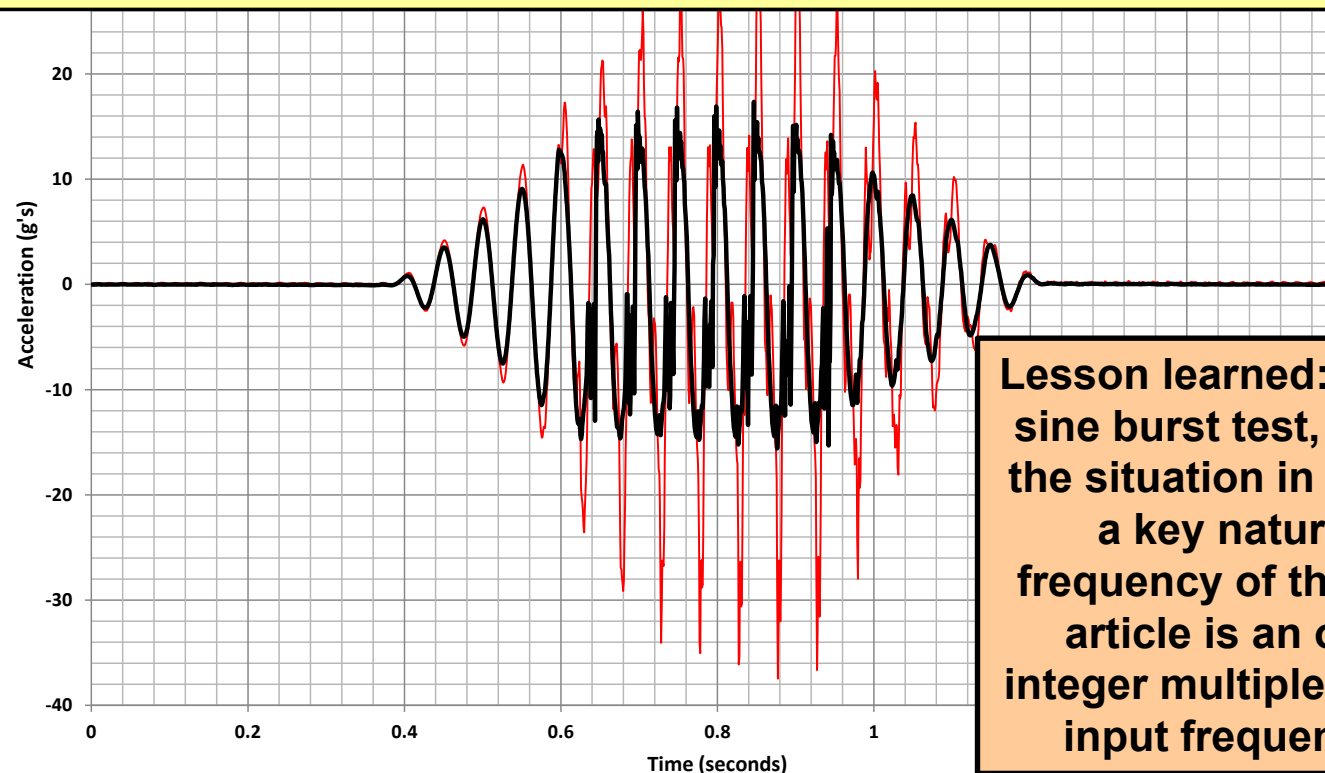
continued

Example from an Actual Case History (continued): How the Test Article Responded to the Imperfect Sine Burst Input



Example from an Actual Case History (continued): Superimposed Input and Response

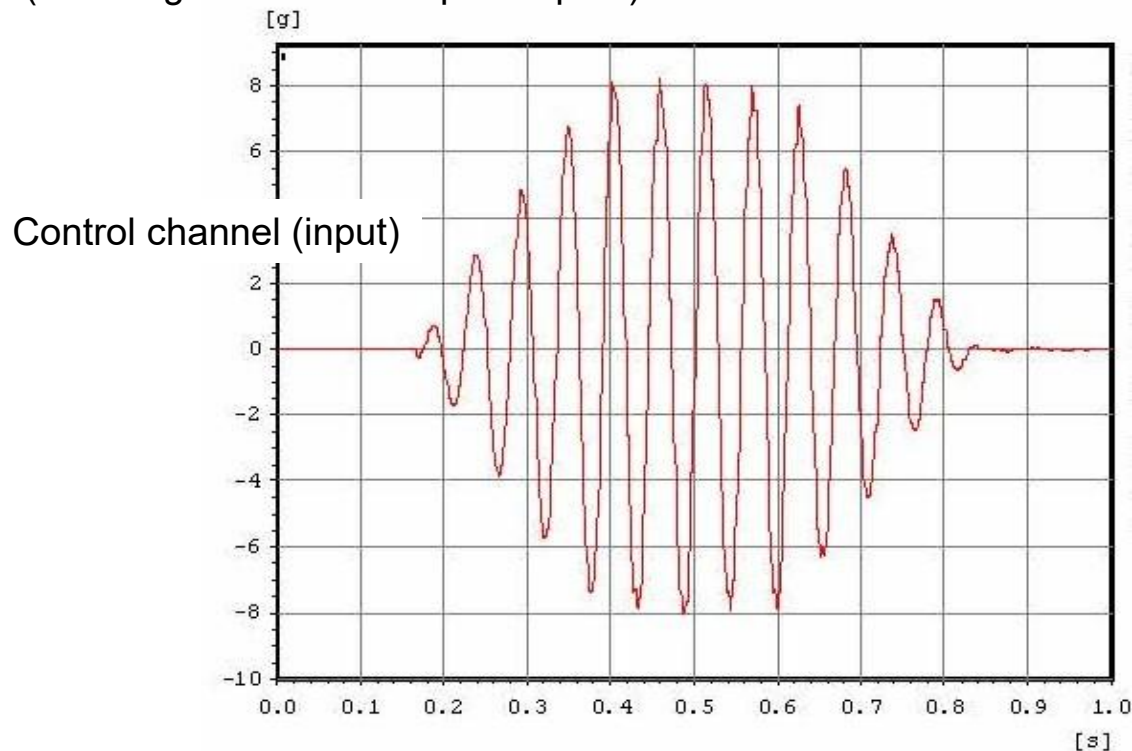
The shaker's inability to properly control the input sine wave, given 60-Hz response of the test article, may have been at least partly the result of electrical noise being at 60 Hz.



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

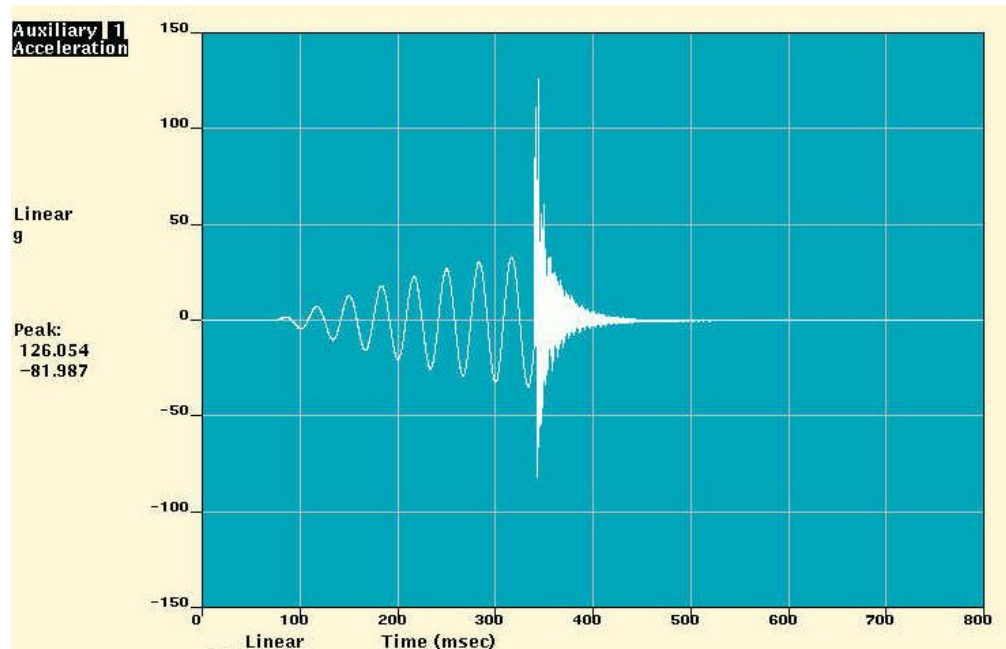
- Sine burst test of a large telescope: 8.1 g full level at 18 Hz, vertical axis
- Input deviated from a sine wave, with harmonic frequencies; did not occur at -6 dB
- Secondary mirror saw 13.4 g response, with pronounced peaks at 126 Hz (7 times 18)
(I no longer have the response plot.)



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,

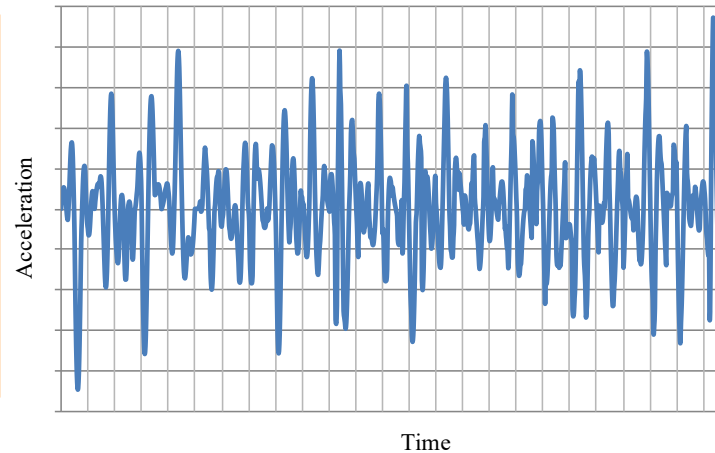
- If you plan to do a sine burst test on a relatively large (say, >100 lb) or expensive item, don't do it on the fully integrated flight assembly.
 - Remember: The objective of sine burst testing usually is to verify strength of the primary structure only. Strength of most equipment and secondary structures 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.

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.
- Lower frequency buffet from turbulence can last longer, perhaps 30 seconds.

Random Vibration Testing: 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 random vibration:

- Unseating of electrical connectors and flipping phase of mechanical relays
- Loosening of threaded fasteners (loss of preload)—may or may not be considered “failure”, depending on the consequences —and fasteners backing out completely
- 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

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 load (strength failure)

Rupture of ductile materials is usually from the accumulation of loading cycles (fatigue)

When a ductile material fails during random vibration testing, it's usually a fatigue failure.

A main objective of qualification 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 .

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

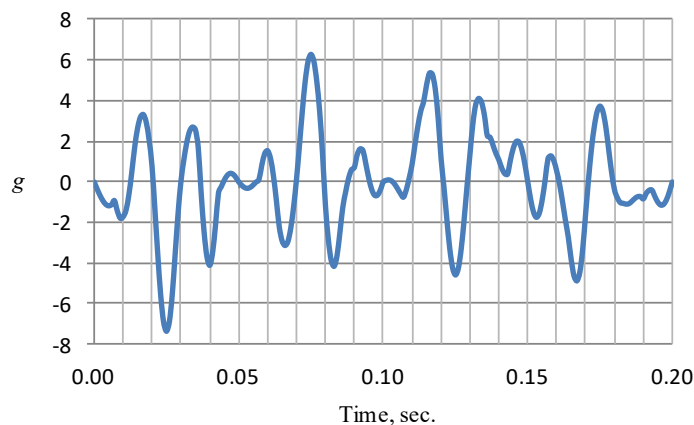
(Eq. 5-8)

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

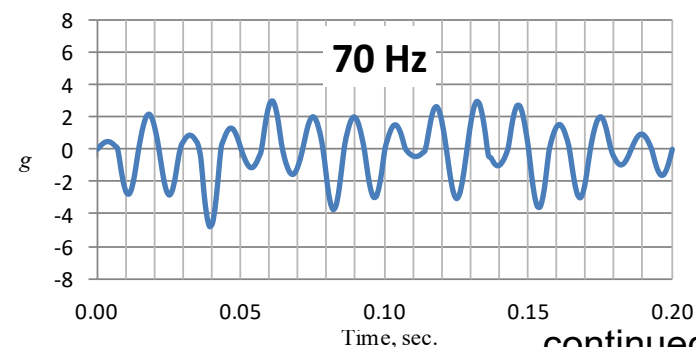
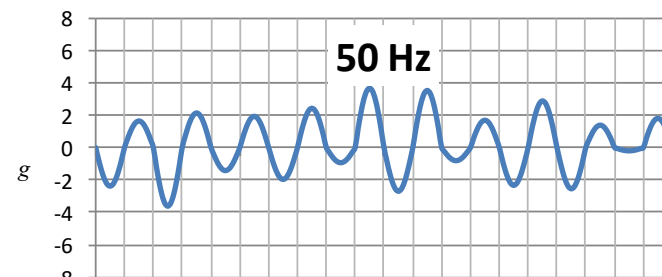
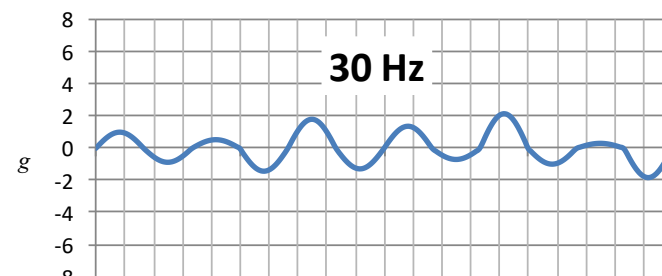
Power spectral density can be used for processing of any random signal that varies with time. The mean-square value of any random signal is referred to as the **power** of that signal.

Example Problem 5-d: Understanding Random Vibration

Say we record acceleration over a 0.2-sec interval:



Then we break the signal down into three sine functions with random amplitude

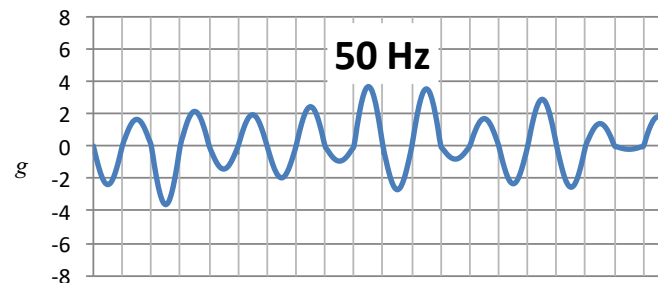


Here, the frequency resolution (bandwidth) is 20 Hz, and the bands are 20 – 40 Hz, 40 – 60 Hz, and 60 – 80 Hz.

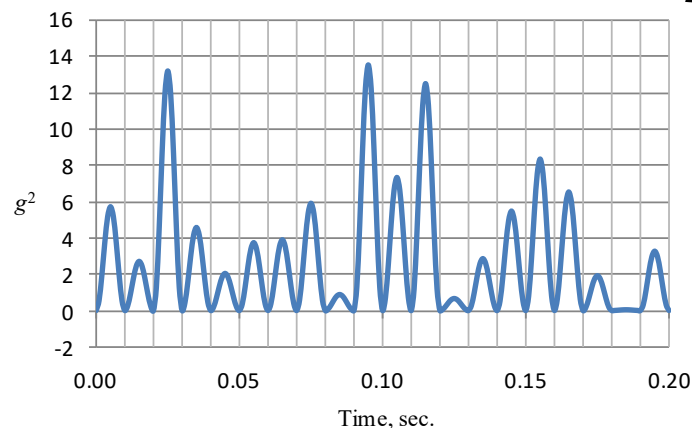
continued

Example Problem 5-d, 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):

$$\bar{a}^2 = \frac{1}{T} \int_0^T a^2 dt \quad (\text{Eq. 5-9})$$

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-d, continued: Calculating PSD and RMS

t (sec)	Overall acceleration (g)	30-Hz acceleration			50-Hz acceleration			70-Hz acceleration		
	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 $0.86 + 2.62 + 2.90 = 6.38 \text{ g}^2$

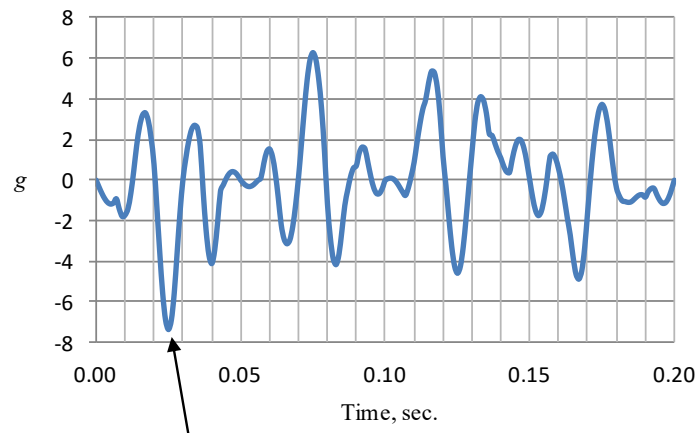
The square root of this value is the **root-mean-square (RMS)** acceleration: 2.53 g, commonly referred to as 2.53 g_{rms} (also written as G-rms, gRMS, g rms, etc.)

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

continued

Example Problem 5-d, conclusion: Peak Acceleration

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



Absolute peak

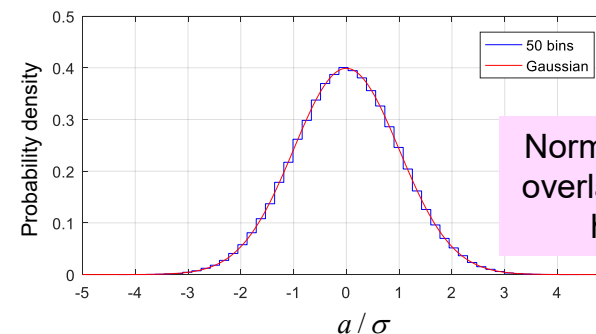
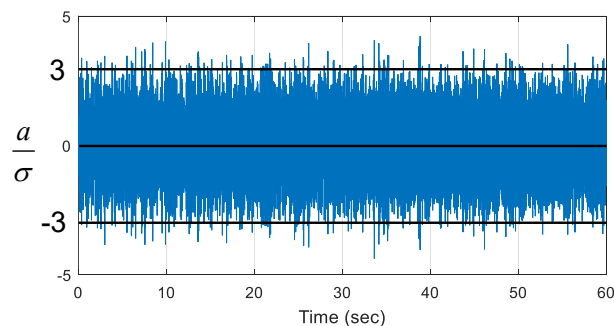
- With the standard deviation, $\sigma = 2.53$ g, the 7.35g peak is a 2.9σ value.
- This peak occurred in just 0.20 sec.
- If we sample a longer time span—say, the full 60 sec of a test—we would expect the peak value to be considerably higher.

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.

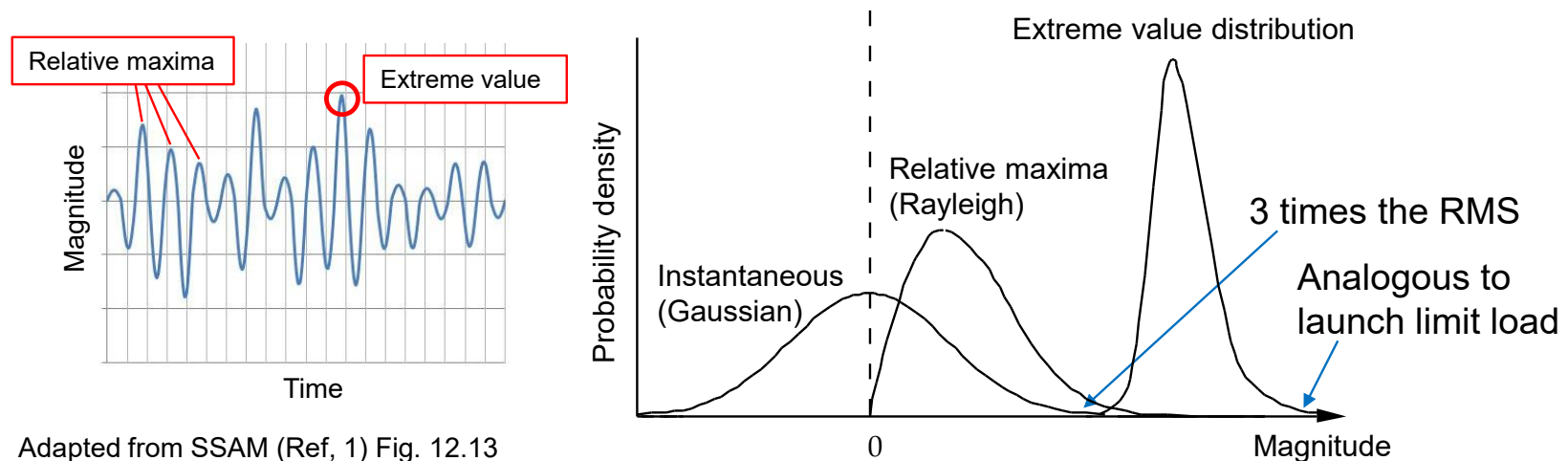


Normal distribution
overlaid on top of a
histogram

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



- 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. 20, Sec. 21-10).
- The **extreme-value distribution** represents the distribution of the highest level that occurs over a given time period (Ref. 20, 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).

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

We often see 4.5σ and 5σ peaks in test, not just from theory—even when using “3-sigma clipping” (Ref. 10).

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

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

The assumption of 3 sigma as the peak load for random vibration has been successful for the most part 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.

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

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

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

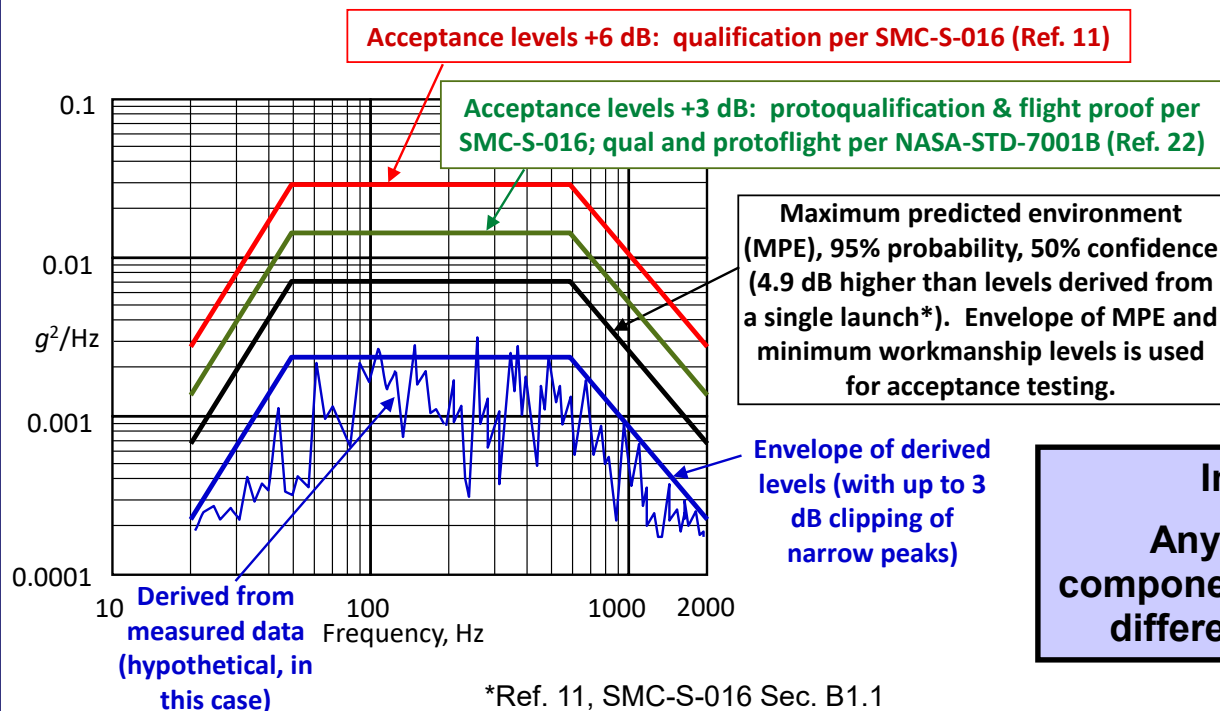
$$\frac{W_2}{W_1} \approx 2^{(N_{dB}/3)} \quad (\text{Eq. 5-11})$$

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.

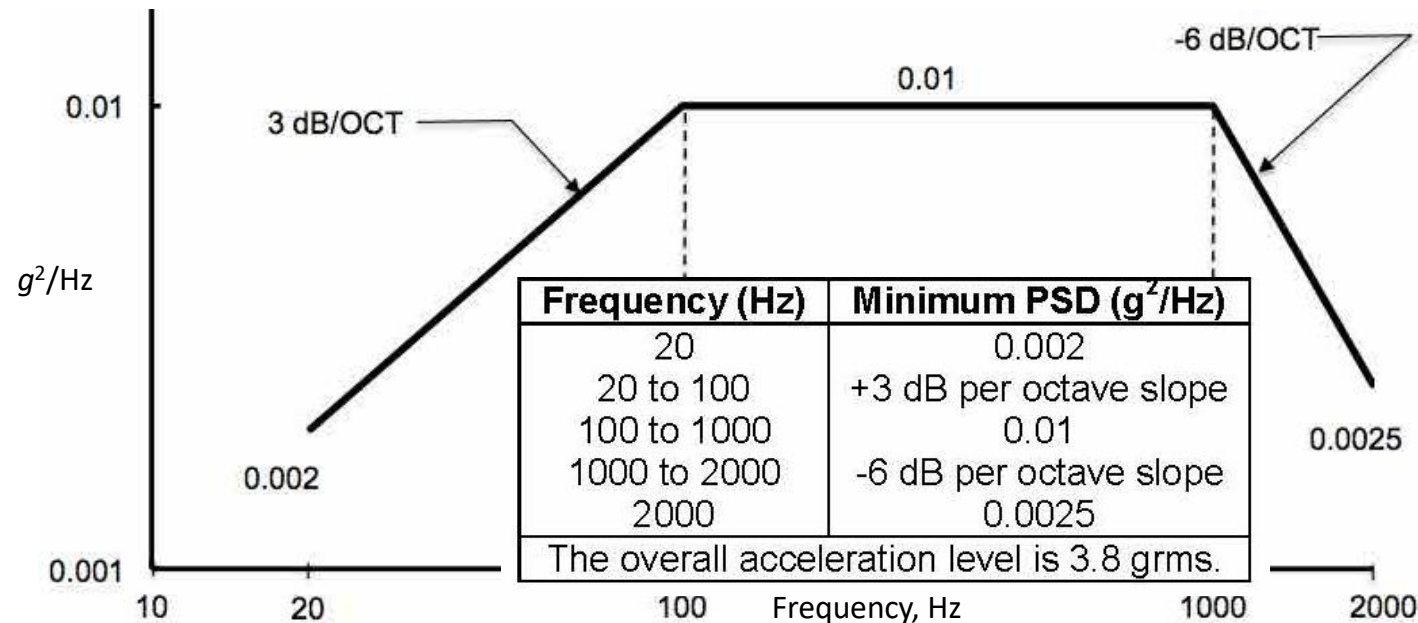
Example:



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. 11, SMC-S-016



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

GSFC-STD-7000B (GEVS, Ref. 23) and NASA-STD-7001B (Ref. 22) 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. 11)		NASA-STD-7001B (Ref. 22)	
	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.

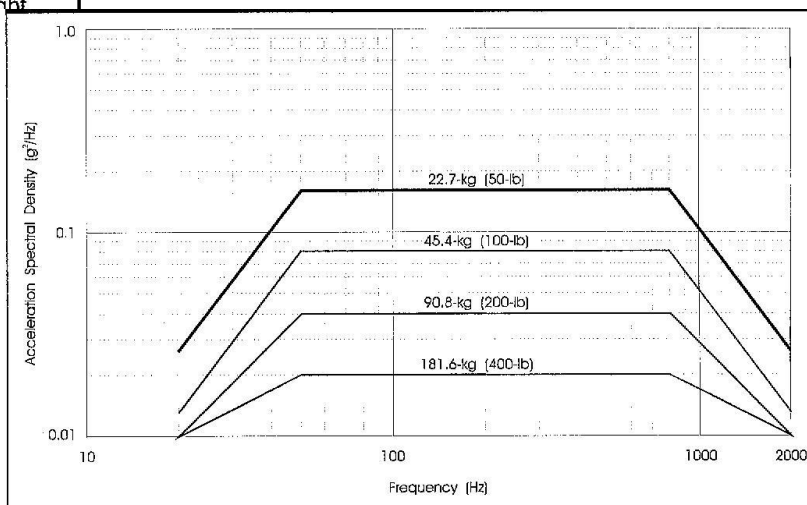
For components weighing over 182-kg (400-lb), the test specification will be maintained at the level for 182-kg (400 pounds).

Source: Ref. 23, 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

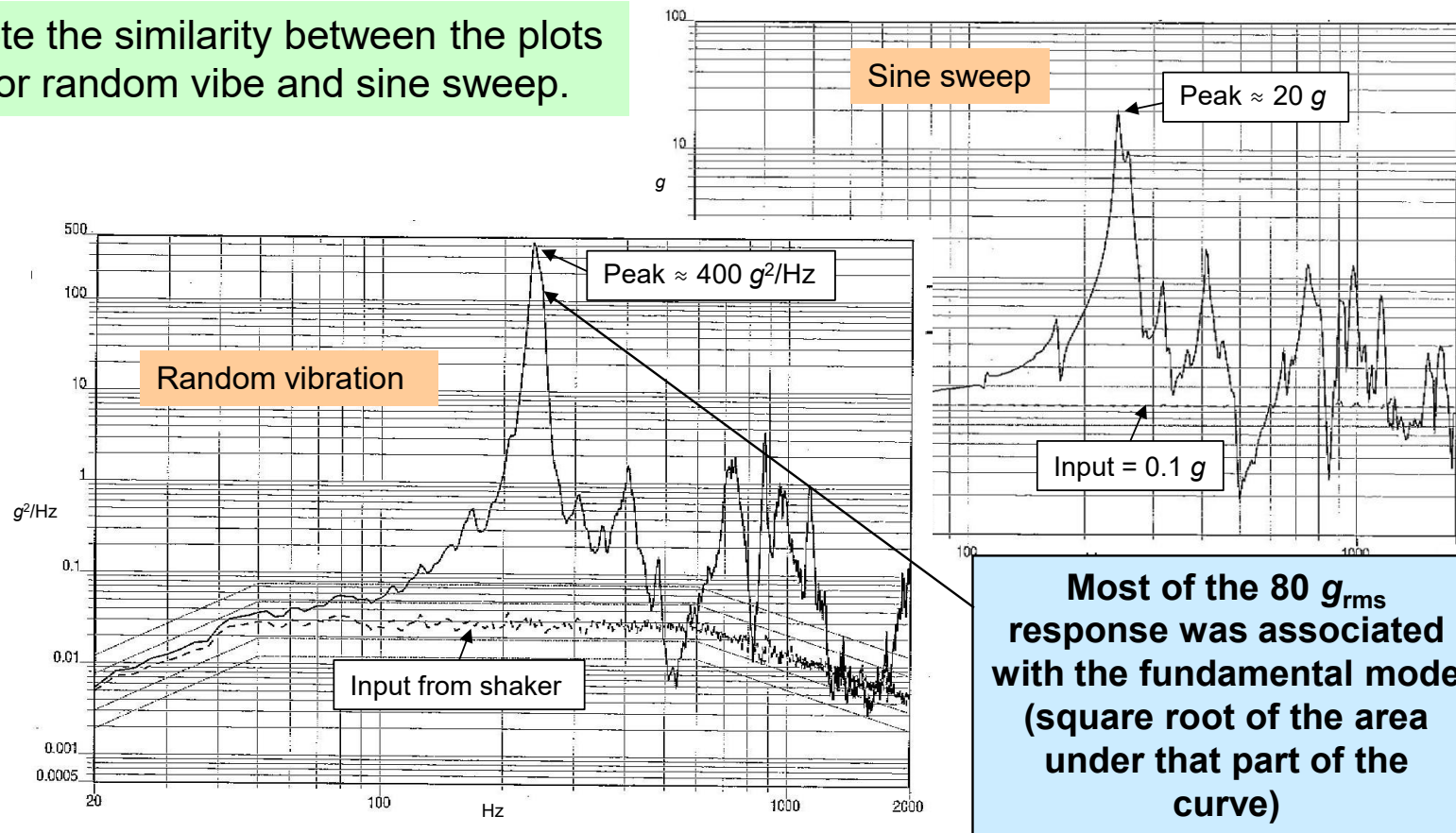
But we invite trouble if we use these levels for SmallSat components.

Qualification levels



Example: Random Vibe Response vs. Sine Sweep Response

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



continued

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 ...

$$R_{rv}(f) = I_{rv}(f) \left[\frac{R_{sv}(f)}{I_{sv}(f)} \right]^2 \quad (\text{Eq. 5-12})$$

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

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

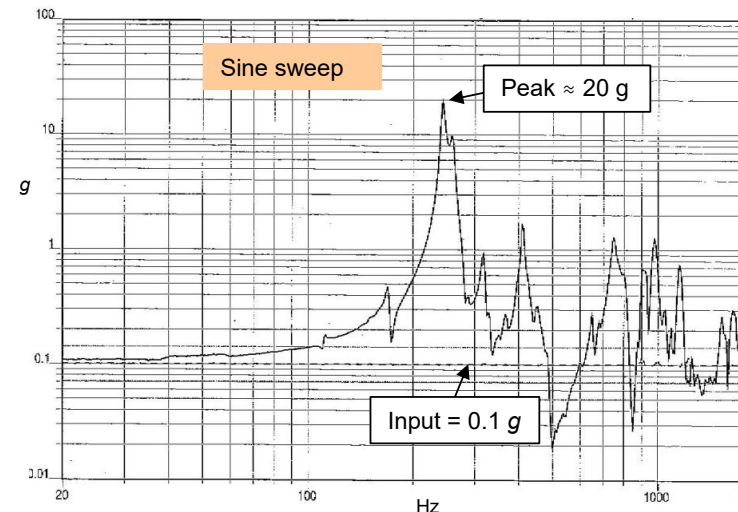
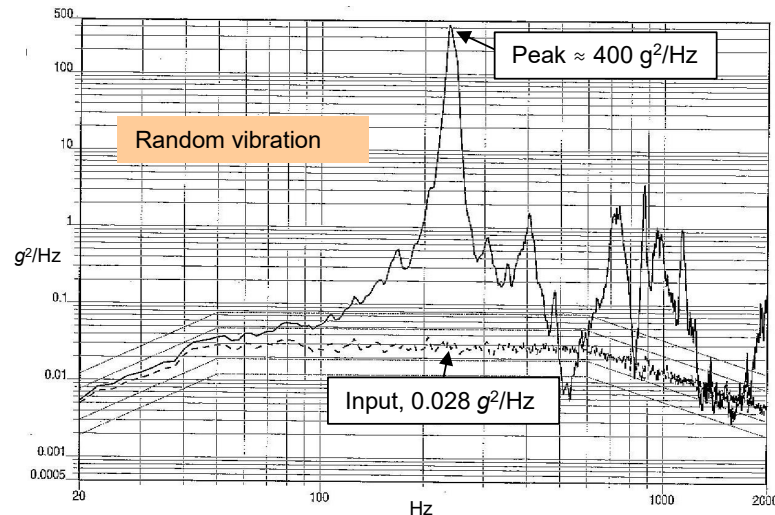
Instead of doing a dedicated low-level sine sweep, you can do a low-level random vibe test at a low, constant level* across the frequency spectrum, and derive the transfer function by rearranging the above equation.

*Referred to as **white noise**

Back to the Previous Example

If response had been linear, from Eq. 5-12 we would expect the peak, p , in the random vibrate test to be ...

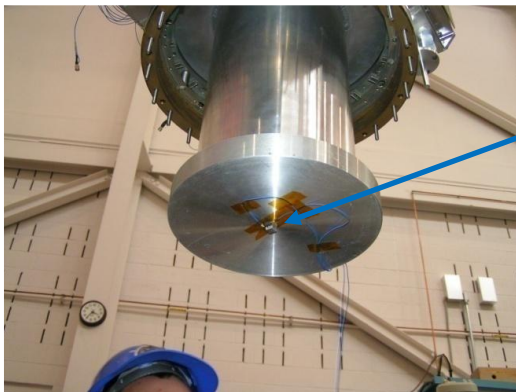
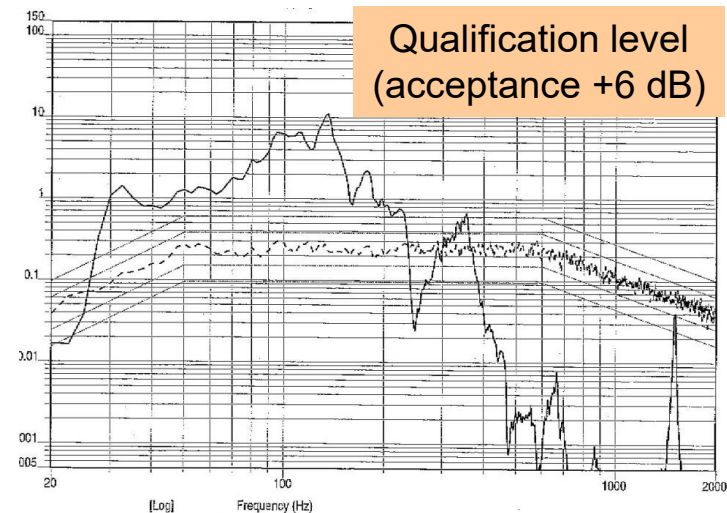
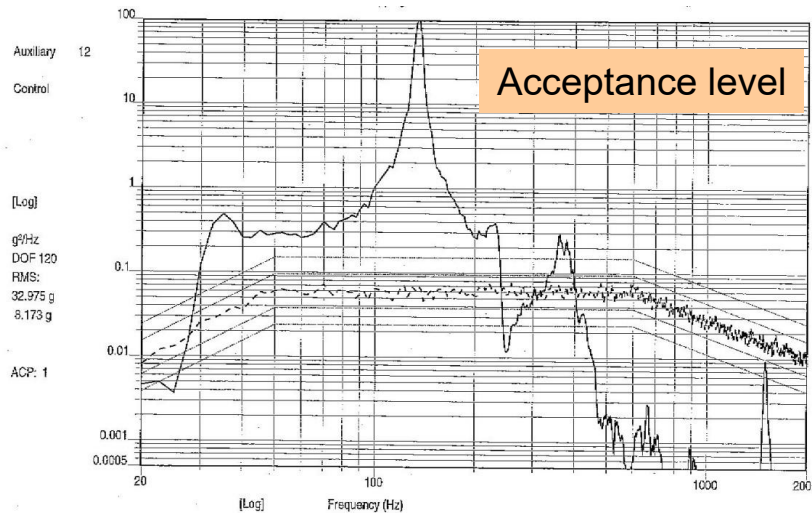
$$p = 0.028 g^2/\text{Hz} \left(\frac{20 g}{0.1 g} \right)^2 = 1120 g^2/\text{Hz}$$



The actual peak was only about $400 g^2/\text{Hz}$ 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.

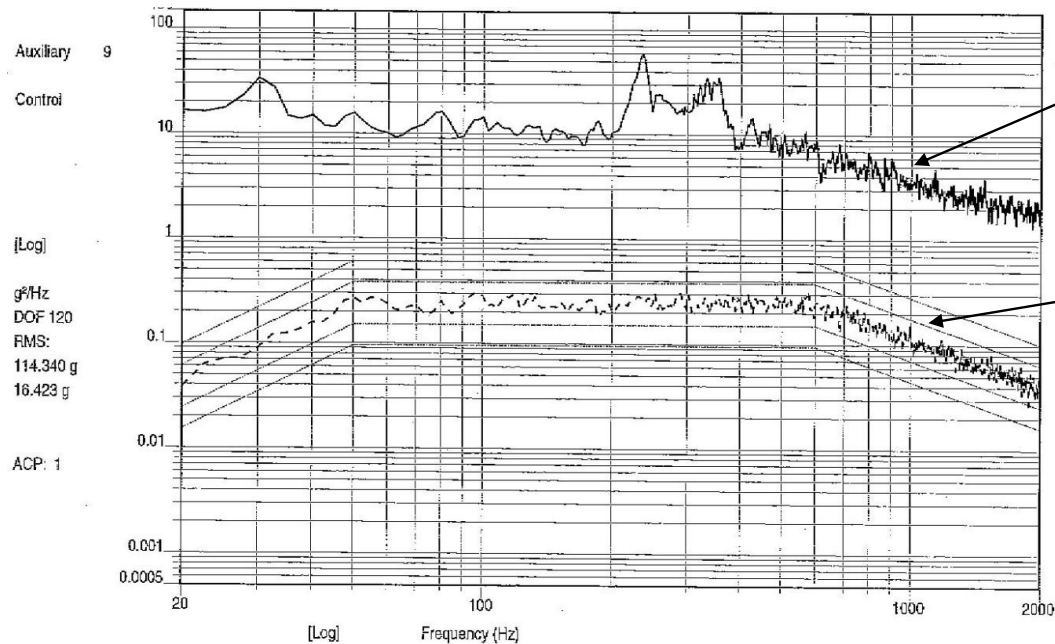
What do you think caused this nonlinearity?

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.

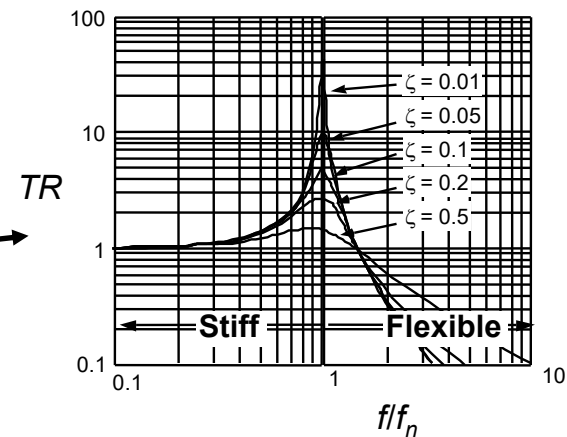
How Would You Interpret This Random-vibration Data?



Response acceleration
(114 g_{rms})

Shaker input acceleration
(16.4 g_{rms})

Hint: Remember the transmissibility curve.

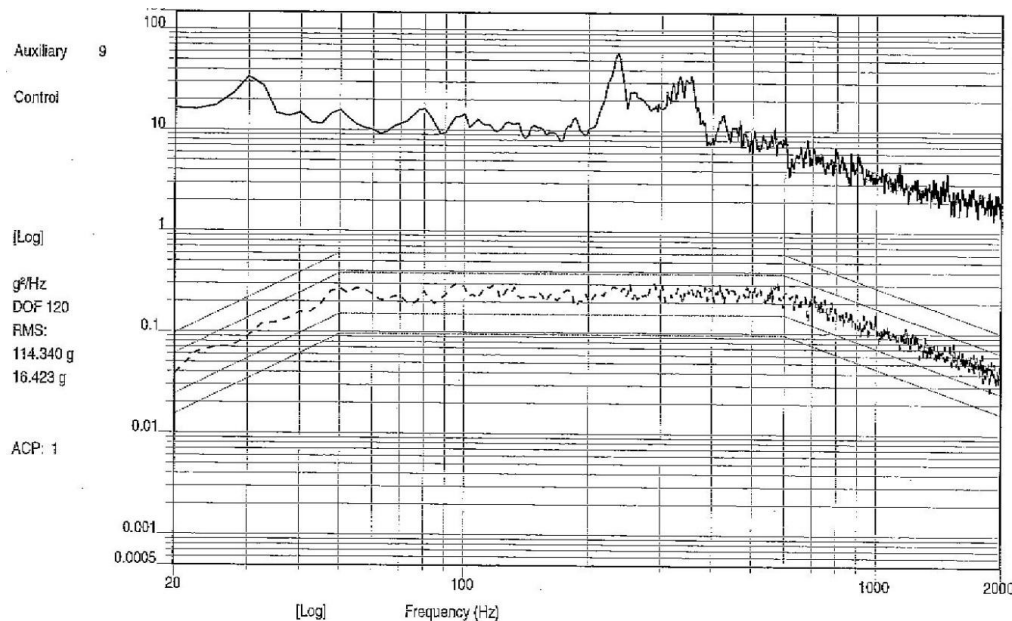


Answer to Question on Previous Page: Something is Wrong with the Data

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.

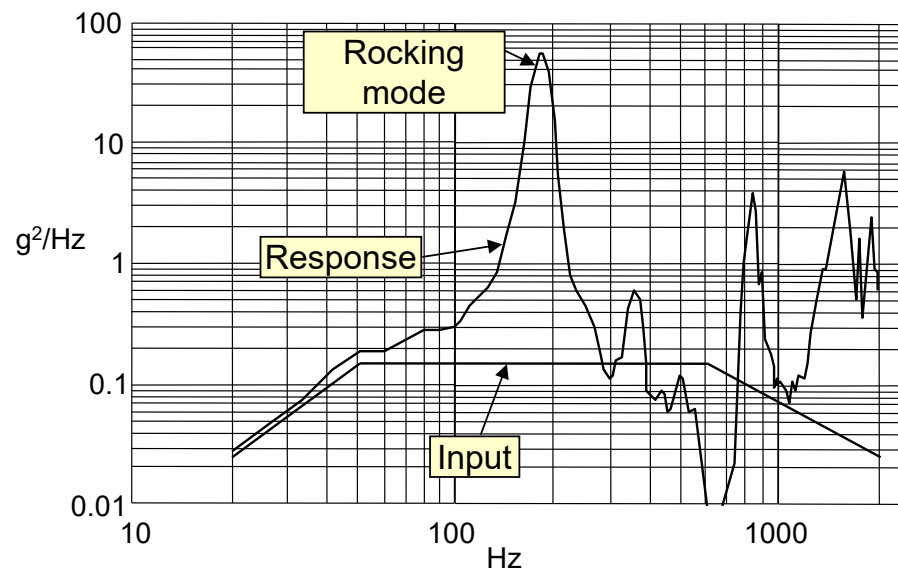
At frequencies above the fundamental frequency, the response plot should cross above and below the input up to 2000 Hz, with isolation likely at high frequencies.



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

Example Problem 5-e: Estimating Base Moment from Random Vibration Response Data

- Accelerometer mounted on top of a SmallSat engineering model (EM)
- Results of qualification-level random vibration test in lateral direction:
 - $54 g_{\text{rms}}$, measured at top of the EM
 - The high response of the rocking mode indicated low damping in the EM structure.



Problem statement:
From this data,
estimate the 3σ
moment at the
mounting base.

**How would you
do this?**

continued

Example Problem 5-e, continued

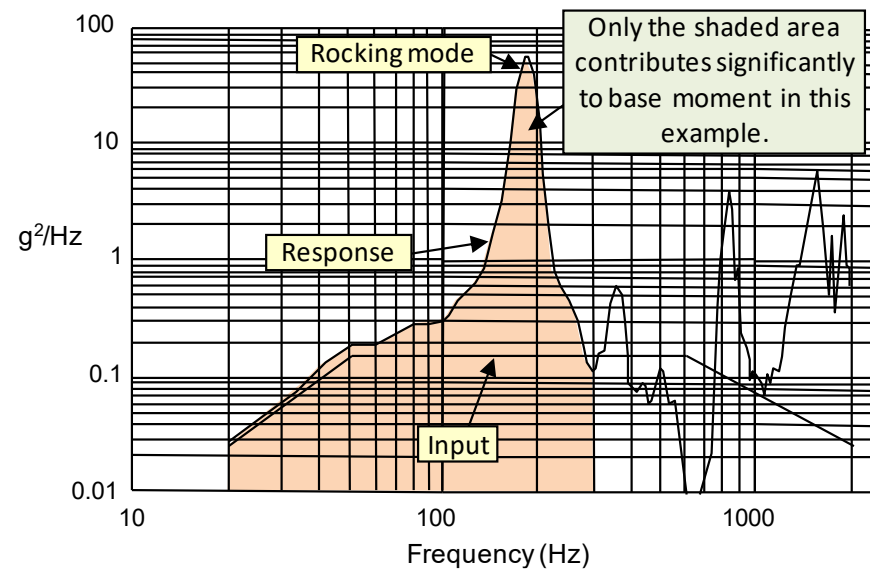
First step: Find the area under the big hump in the response PSD associated with the 182-Hz fundamental mode (20 – 300 Hz).

- 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 contribute insignificantly to the base moment (typically the case).

My estimation:

41 g_{rms} for the shaded area at right

(vs. 54 g_{rms} total)

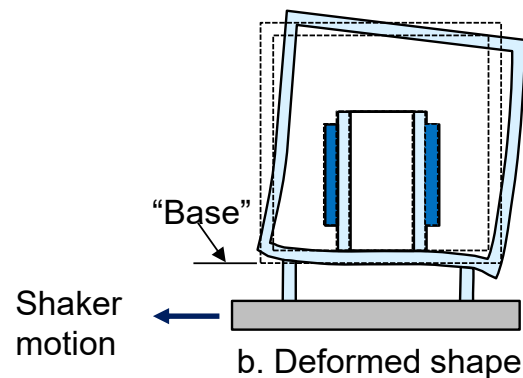
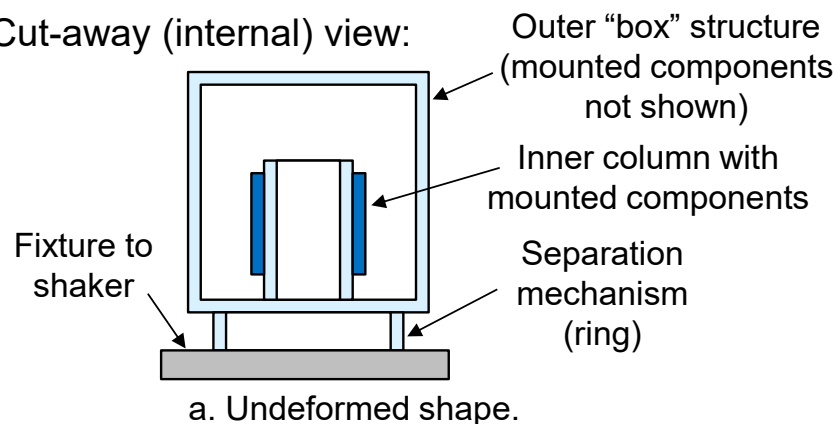


continued

Example Problem 5-e, continued

Second step: Understand the mode shape

Cut-away (internal) view:

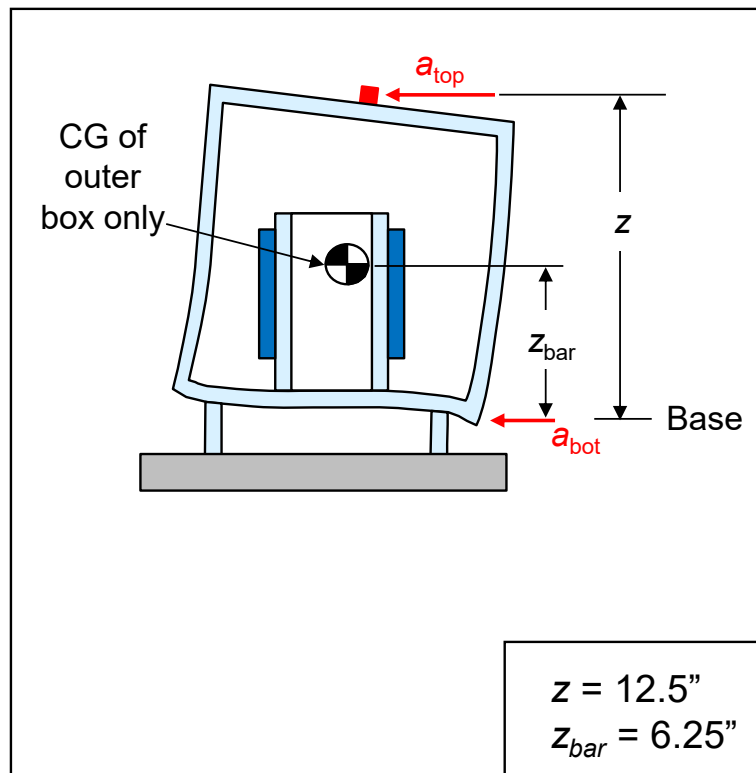


- Pretest predictions and test data showed only the outer box mass contributed much to this mode
 - Base plate, four side walls, and top panel
 - Not the column or ring
- The box structure rotates nearly as a rigid body
 - CG translates (moves laterally)
- Mass properties for the box structure:
 - $m = 0.0648 \text{ lb} \cdot \text{s}^2/\text{in}$ (11.3 kg)
 - (25 lb weight)
 - $J = 2.50 \text{ lb} \cdot \text{s}^2 \cdot \text{in}$ moment of inertia
 - (0.282 $\text{kg} \cdot \text{m}^2$)

continued

Example Problem 5-e, continued

Third step: Derive rigid-body translational and rotational accelerations



Design criterion: “peak” load = 3 times the RMS value

– Because failure modes are ductile

By taking the square root of 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

$$a_{cg} \approx 3 \left[6 + (41 - 6) \frac{z_{bar}}{z} \right] = 70.5 g \quad (\text{Ref. Eq. 5-2})$$

And the peak rigid-body angular acceleration is

$$R \approx 3(41 - 6)(386) \left(\frac{1}{z} \right) = 3240 \text{ rad/s}^2 \quad (\text{Ref. Eq. 5-3})$$

$$g = 386 \text{ in/s}^2$$

continued

Example Problem 5-e, conclusion

Fourth step: Estimate 3σ moment at the box/ring interface

$$\begin{aligned} m_{qrv} &\approx a_{cg} w z_{bar} + RJ \quad (\text{Ref. Eq. 5-4}) \\ &= 70.5(25)(6.25) + 3240(2.50) \\ &= 11,000 + 8100 \\ &= 19,100 \text{ in} \cdot \text{lb} \end{aligned}$$

**The equivalent
quasi-static load
was about 84 g.**

Note: Eq. 5-4, above, applies for rigid-body rocking. It's used here as an approximation for the SmallSat's rocking mode.

Selected Key Points from This Section

- Investigate anomalous test data before disturbing the test configuration.
 - Make sure the data is dependable.
 - Make sure you understand the reason for any significant differences between pre- and post-test sine sweep results.
- Testing on a shaker may not fully verify strength, depending on the failure modes of concern.
 - Pay close attention to the response data to see what it tells you.
- 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.
- Testing on a shaker is risky, so ...
 - 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.
 - Consider alternatives, especially for large structures, and test at the lowest level of assembly that will meet objectives.

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.

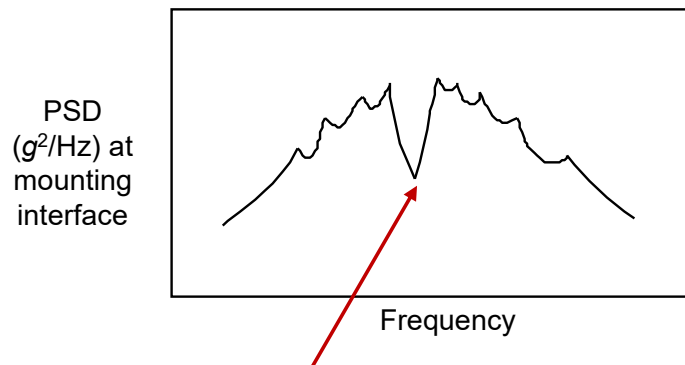
If we mount a single-degree-of-freedom system (SDFS) to a vibrating structure that has limited energy, some of the energy moves into the SDFS, especially at frequencies near and equal to the SDFS's natural frequency. The result is a natural reduction (notch) in acceleration at the base of the SDFS at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

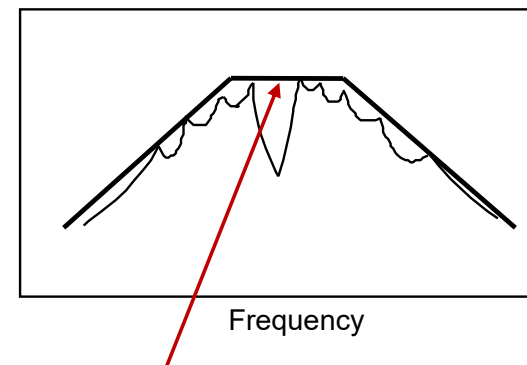
Notching Occurs Naturally in Flight

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.

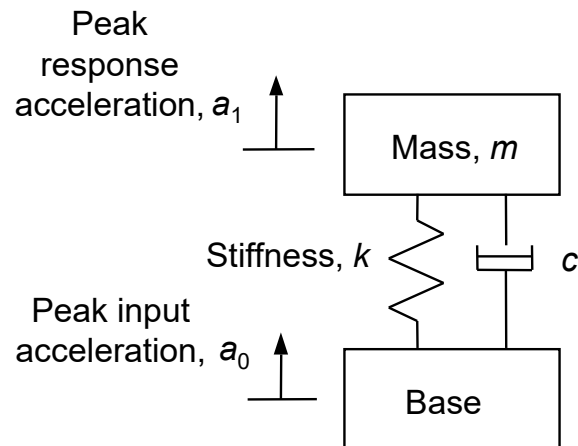
Then we draw a smooth envelope for test:



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.

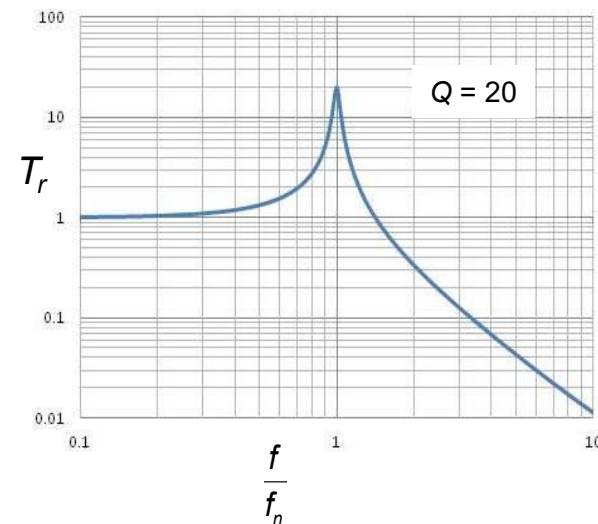
Review: Transmissibility for a Sinusoidally Base-driven Mass on a Spring (Single-Degree-of-Freedom System (SDFS))



➡
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

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

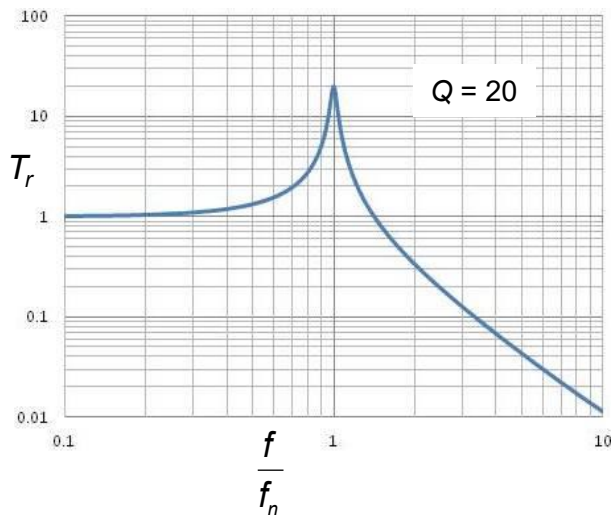
For any item, such as a test article, the apparent mass is

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

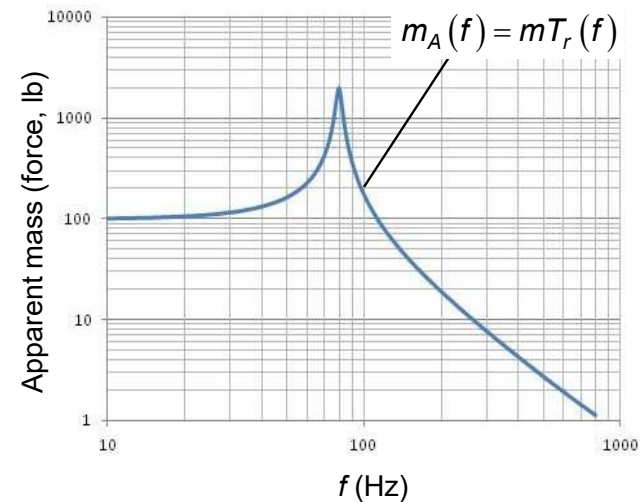
To get force units for m_A , we're actually dividing F by the unitless load factor. Alternatively, you can think of the units for apparent mass as being lb/g instead of lb. But it's traditional to use lb (or N) for m_A .

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:



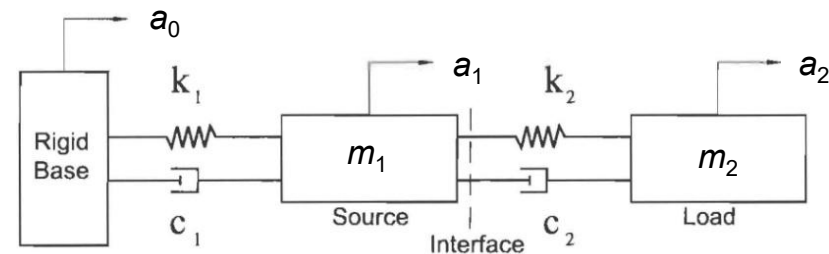
Multiplying the transmissibility function, $T_r(f)$, by mass yields the apparent mass, $m_A(f)$, which is the spring (base) force caused by 1 g sinusoidal base acceleration.



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

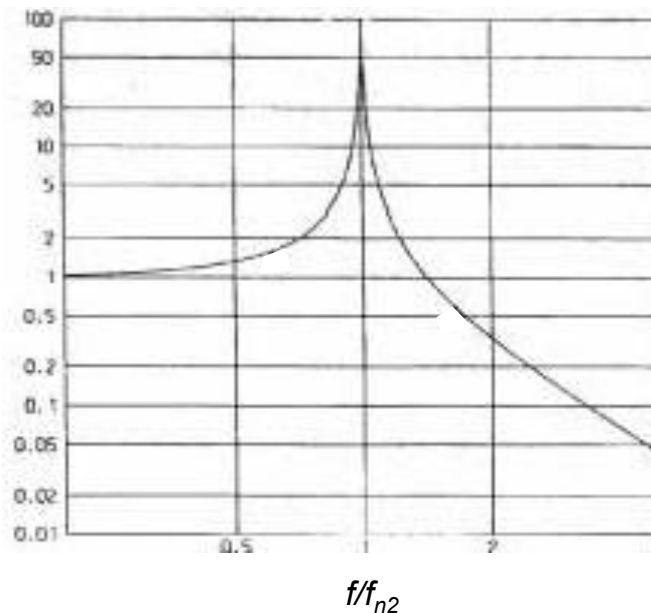
$$f_{\text{sys-1}}, f_{\text{sys-2}} = f_{n2} \sqrt{1 + \frac{m_2 / m_1}{2} \pm \sqrt{m_2 / m_1 + \frac{(m_2 / m_1)^2}{4}}} \quad (\text{Eq. 6-2})$$

(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

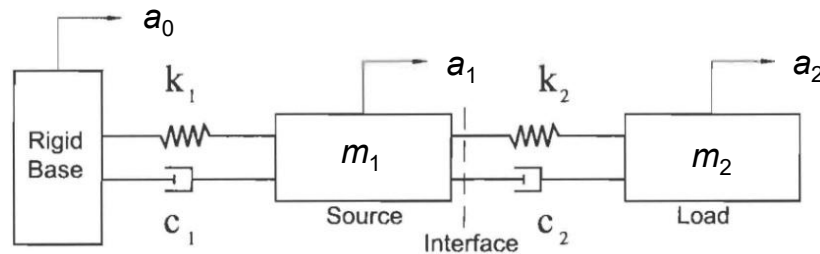


This example is from VTSS Ref. 15, STDI Ref. 26, “Force Limited Vibration Testing Monograph”, based on VTSS Ref. 14, STDI Ref. 25 (Hartog)

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 $Q_1 = Q_2 = 50$:

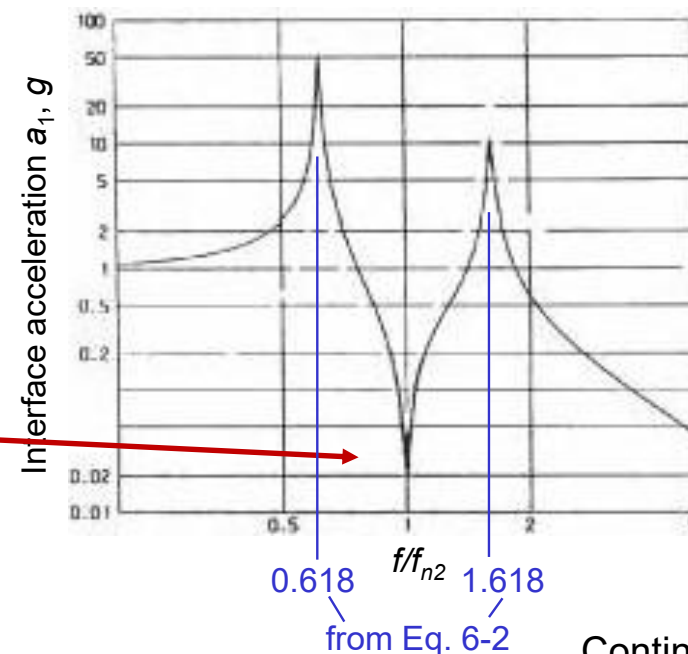


Note the notch (valley) at $f = f_{n2}$, the natural frequency of the load by itself.

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 $1/Q$.

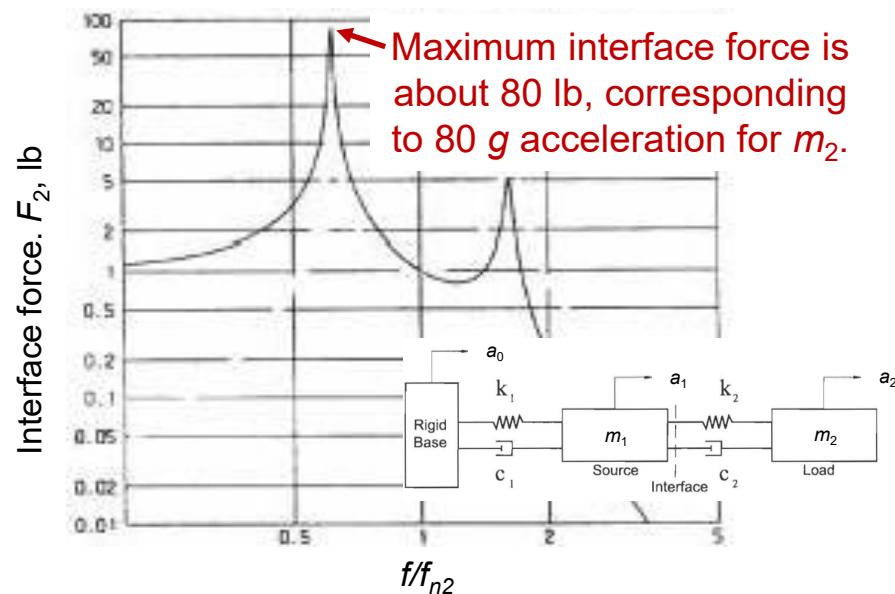
The rigid base imparts 1 g acceleration at varying frequency, resulting in the following plot of interface acceleration (acceleration of m_1 , which is the base of the k_2 spring):



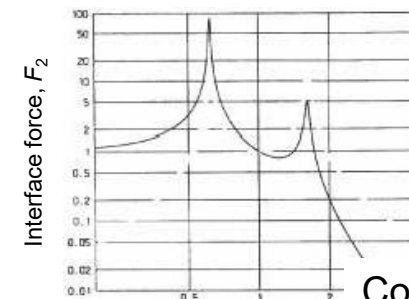
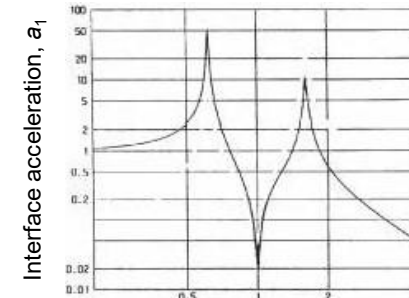
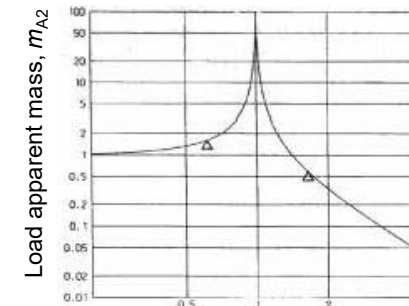
Continued

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:

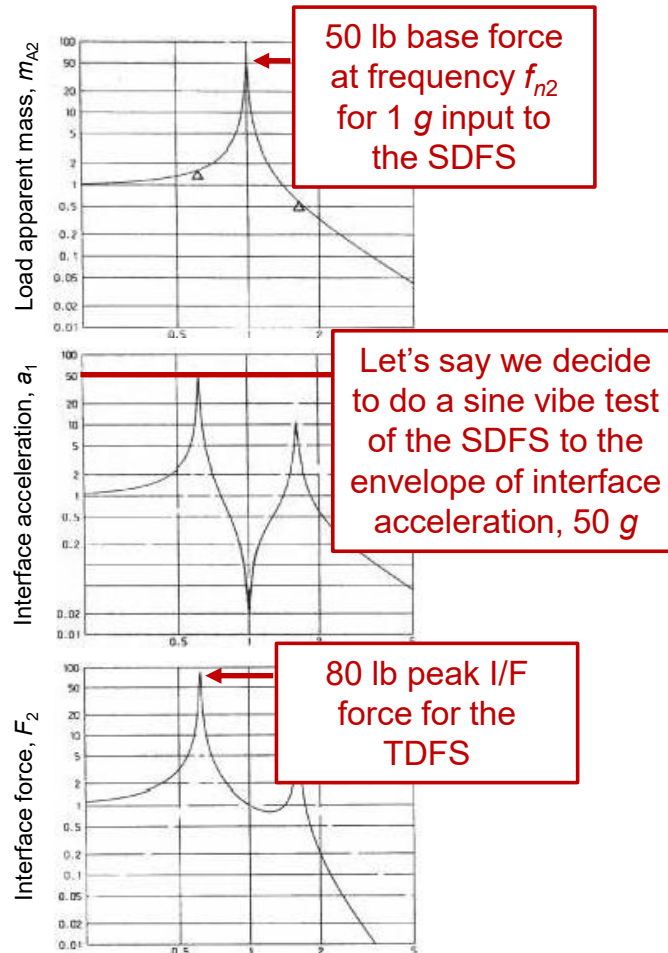


$$F_2(f) = m_{A2}(f)a_1(f) \quad (\text{Eq. 6-3})$$



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.



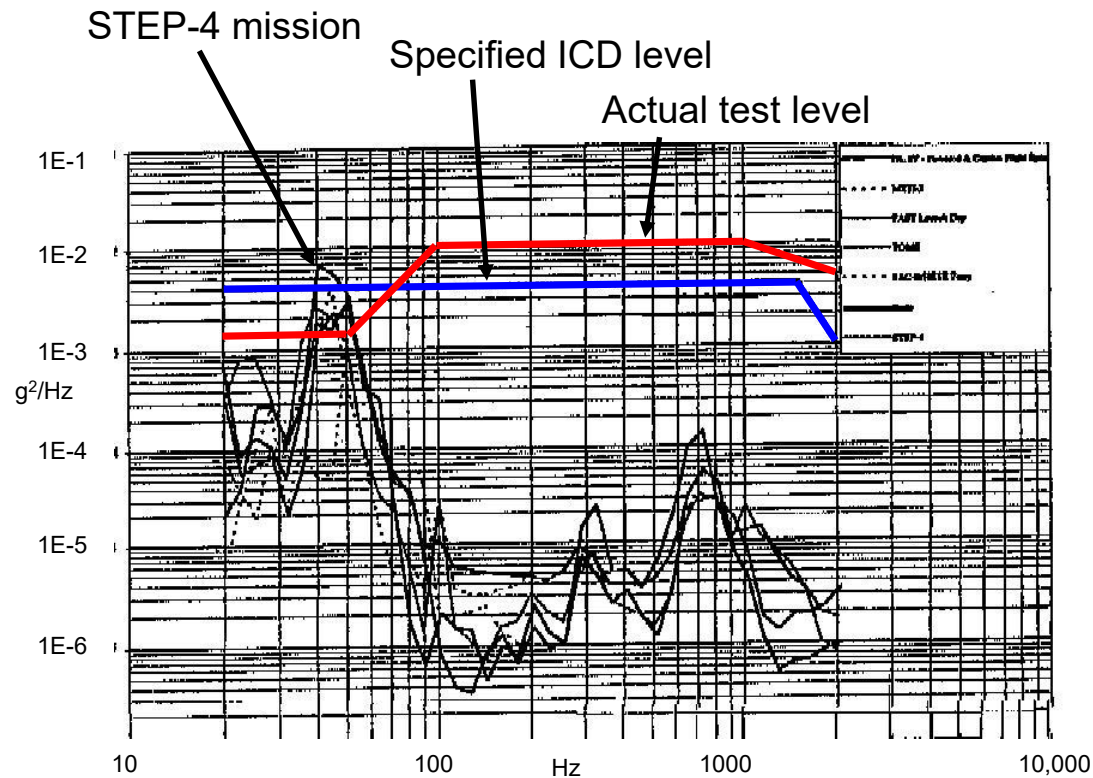
(Continued)

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:

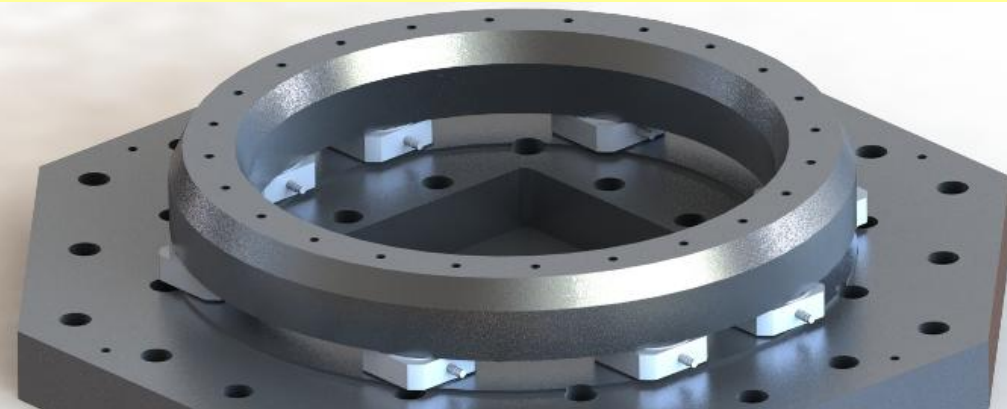
- Captive carry excites a 40-50 Hz mode in Pegasus XL.
- The spacecraft developer apparently was not aware of this excitation.
- Notching apparently was without technical rationale.

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.

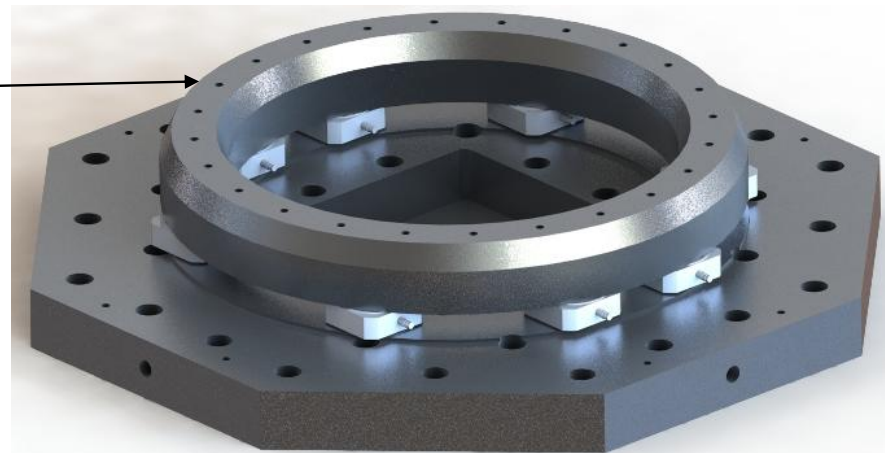


Triaxial force sensor from PCB Piezotronics

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.

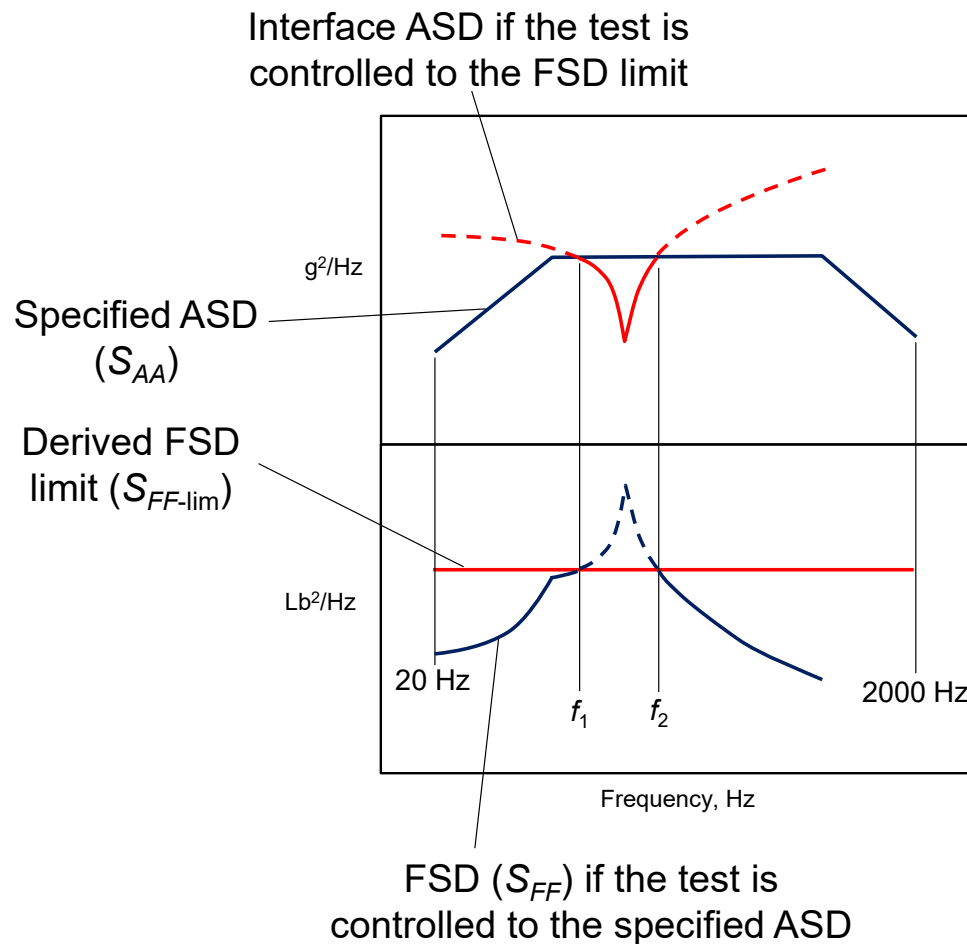
- In terms of a force PSD (referred to herein as **force spectral density, FSD** (S_{FF})), in units Lb^2/Hz or N^2/Hz , at the base of the test article
- Let's call this force limit an **FSD limit** ($S_{FF\text{-lim}}$), and in test it applies at the I/F with the test fixture.

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, $S_{FF-lim}(f)$:

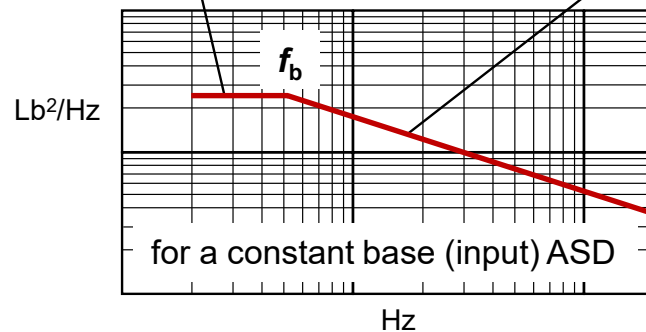
Up to the break frequency,

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

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).

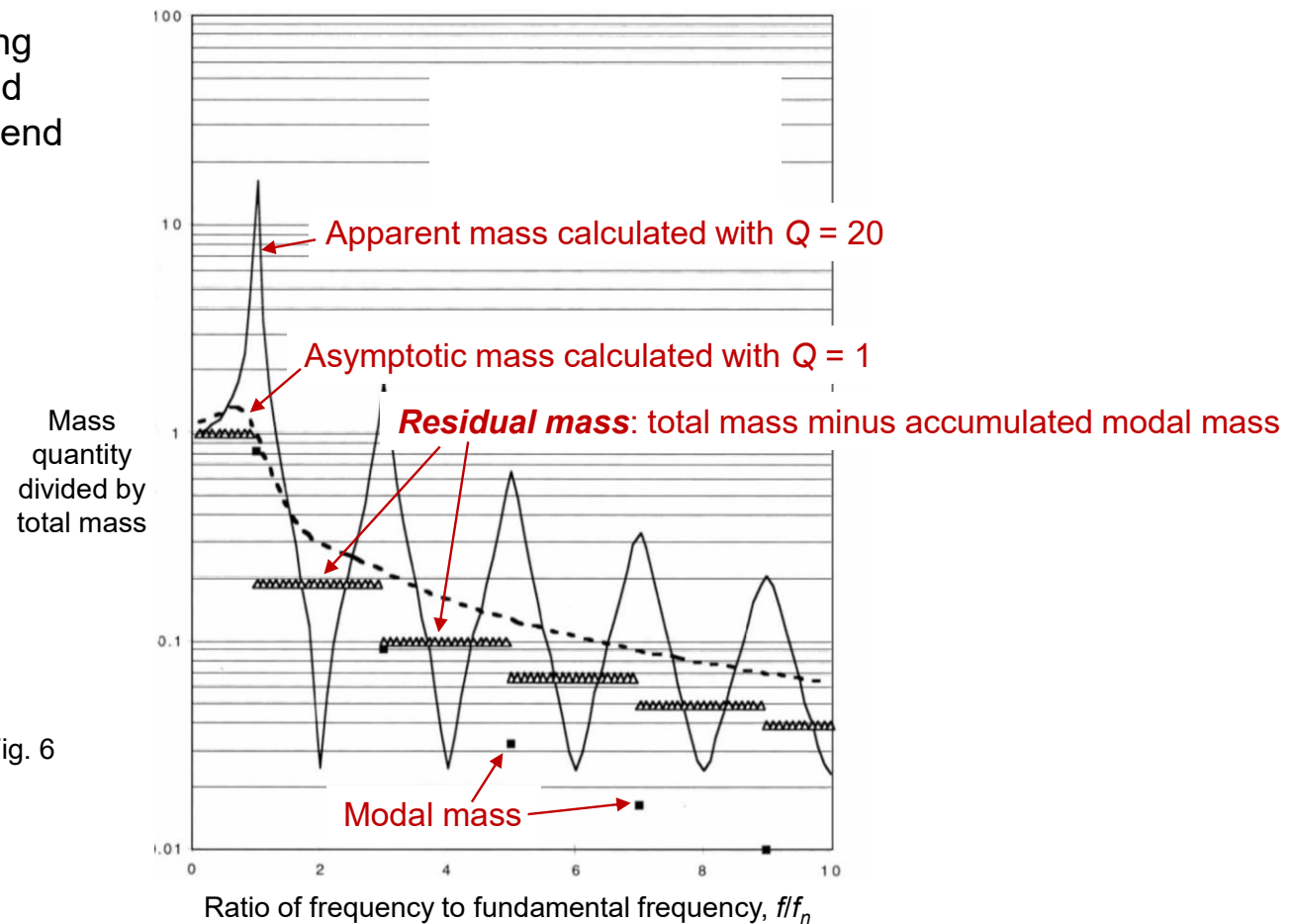
- Often set equal to the fundamental frequency of the test article.
- But, if the apparent mass (or response FSD) shows two or more nearly equal peaks, the break frequency should be the highest frequency of those peaks.

$2n$ 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

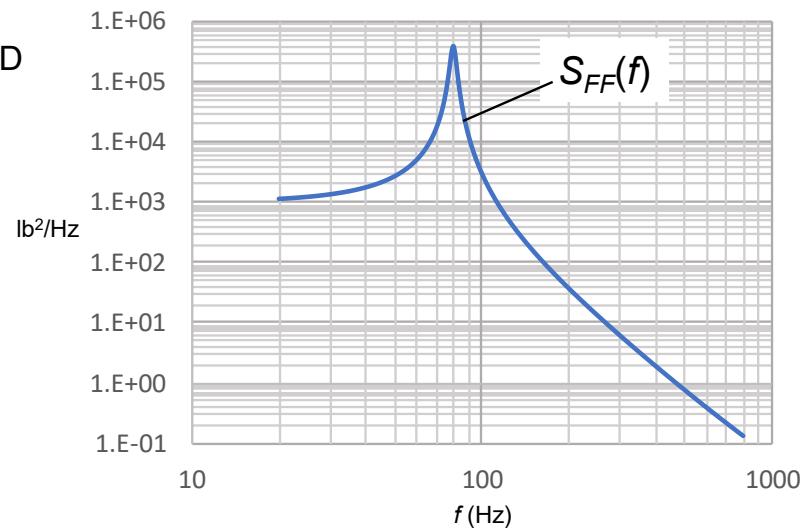
Semi-Empirical Force Limits for Random Vibration

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

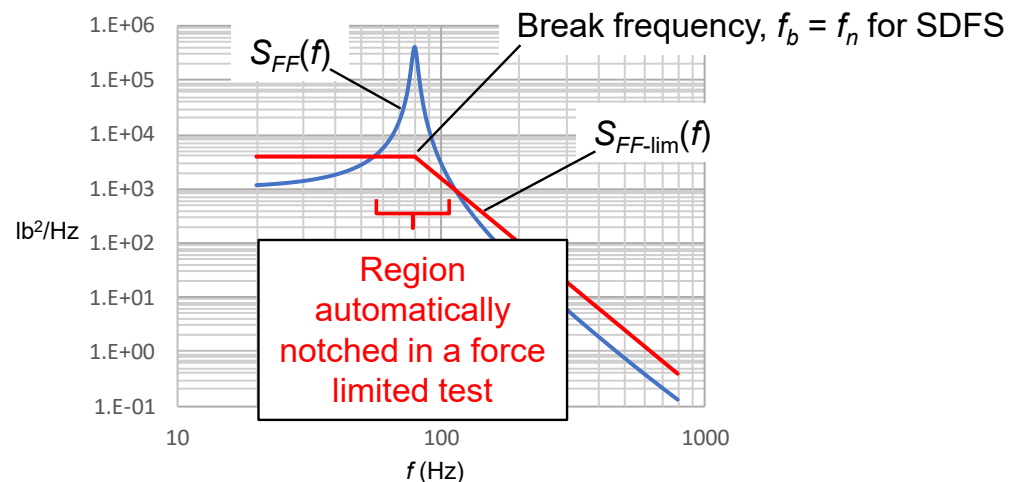
Example, Base-driven SDFS, continued: Semi-Empirical Force Limits for Random Vibration

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

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

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

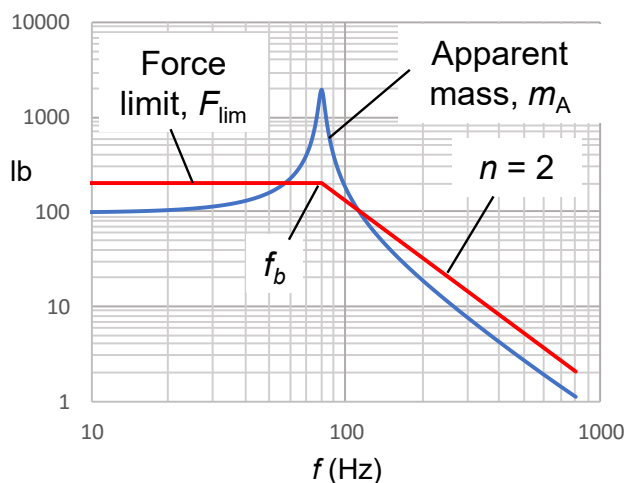
For arbitrarily (for this example) selected $C = 2$ ($C^2 = 4$) and $n = 2$ ($2n = 4$),
here's what the FSD limit looks like for a base ASD, $S_{AA-0}(f)$, of $0.1 \text{ g}^2/\text{Hz}$
from 20 to 800 Hz:



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)

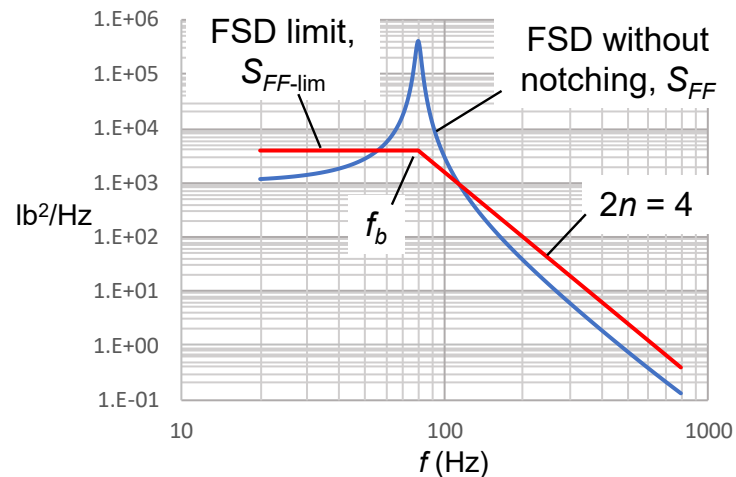


Table 6-1: Common Methods for Deriving C^2

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.

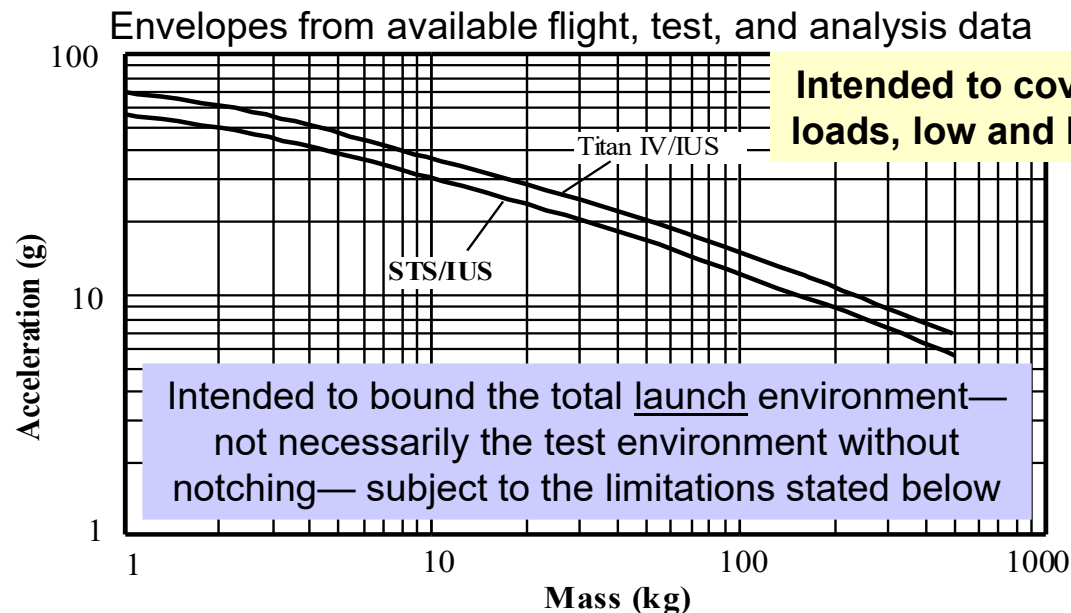
- CLA includes modes of vibration only up to the **truncation frequency**, which is often 50 Hz but depends on the launch services provider (LSP).
 - Modes above that frequency are truncated from the coupled math model.
- Thus, the effects of random vibration introduced at the LV interface above the 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.

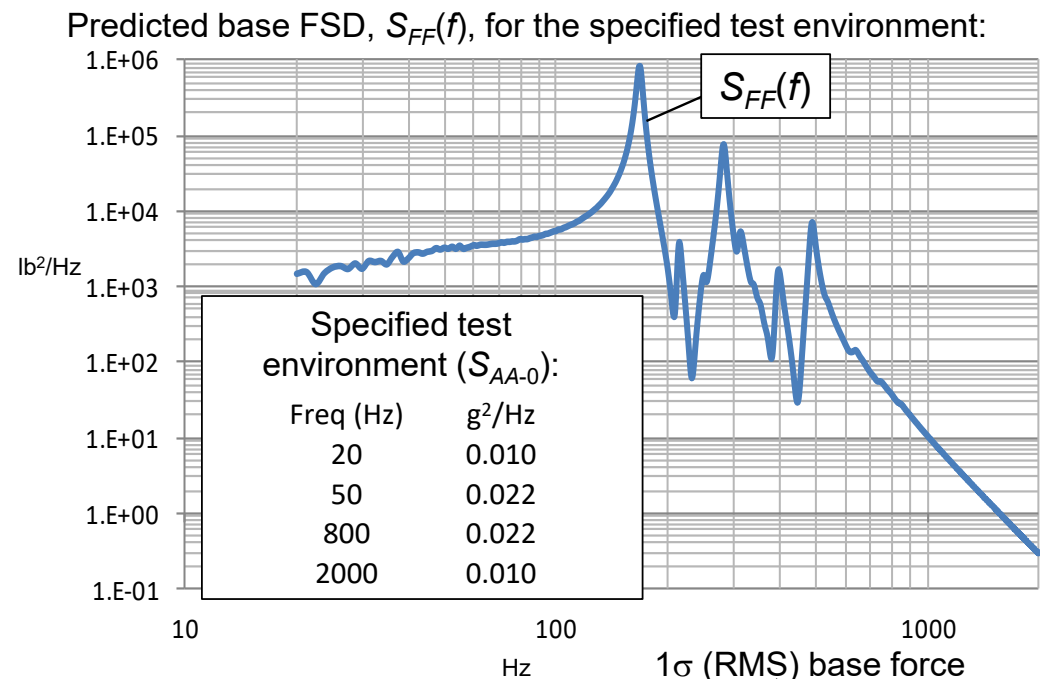
- Instructions:
- Applies in any direction, one axis at a time
 - In the thrust axis, add 3.9g for T-IV and 1.5g for STS (quasi-static)
- Limitations:
- For “lumped-mass” (compact) components and subsystems only
 - For appendage-like components having a statically determinate mount
 - For preliminary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

VTSS Ref. 25, STDI 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

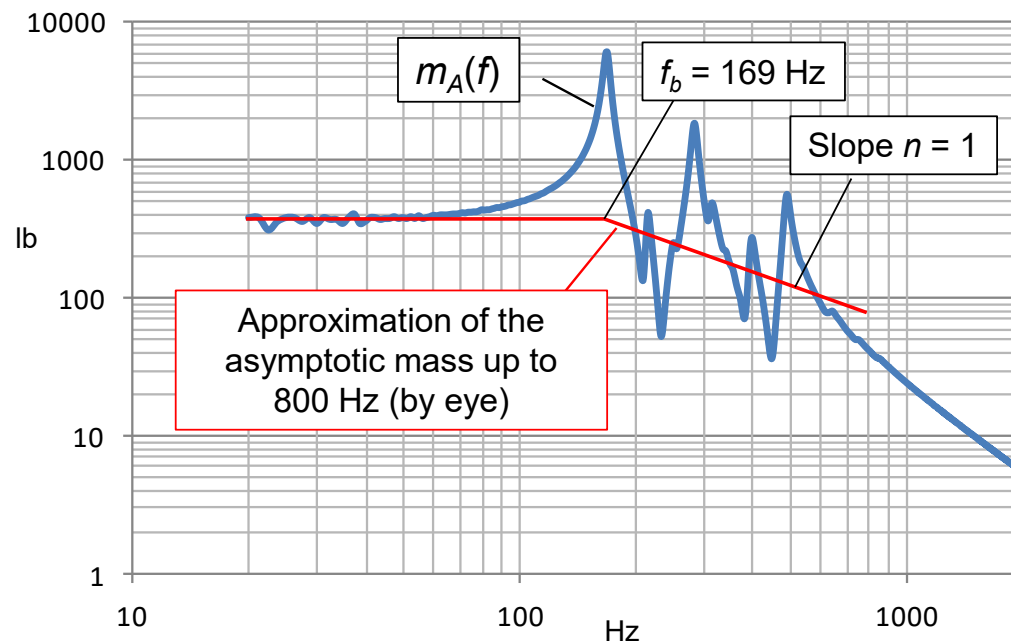
Recall Eq. 6-9:

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

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

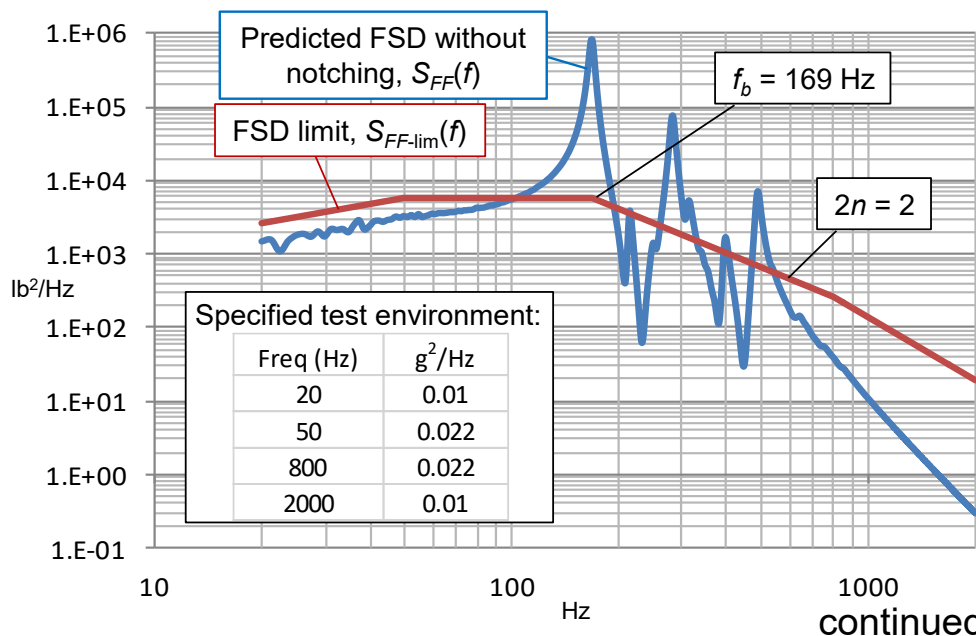
Example Problem 6-a, continued: Deriving an Appropriate Value for C^2

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.

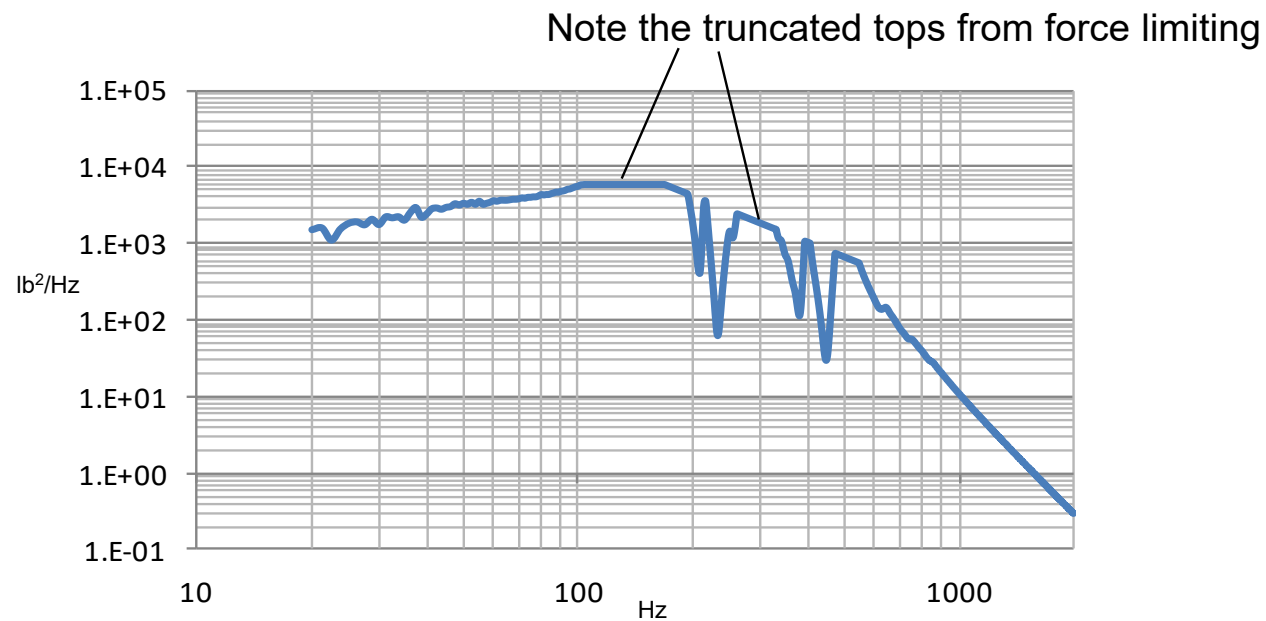
Results of iteration:
 $C^2 = 2$, yielding the force-limit curve shown at right.

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.

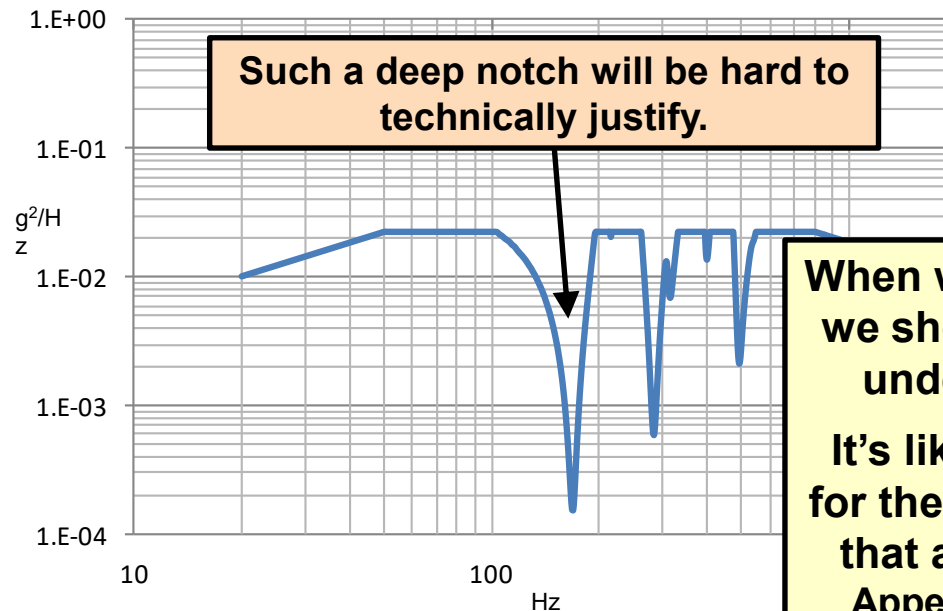


Example Problem 6-a, continued: Base FSD After Force Limiting



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:

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

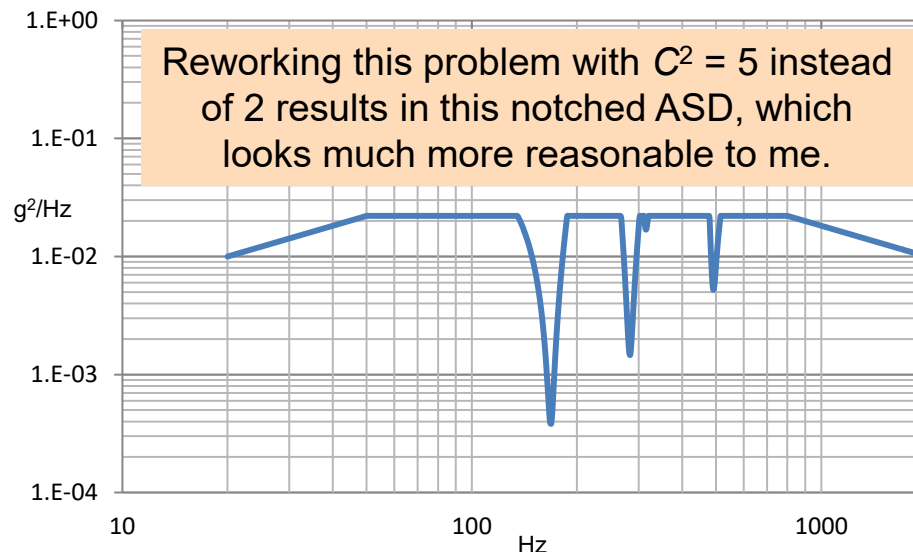
When we end up with such a deep notch,
we should ask ourselves if we might be
undertesting between 100 – 200 Hz.

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.

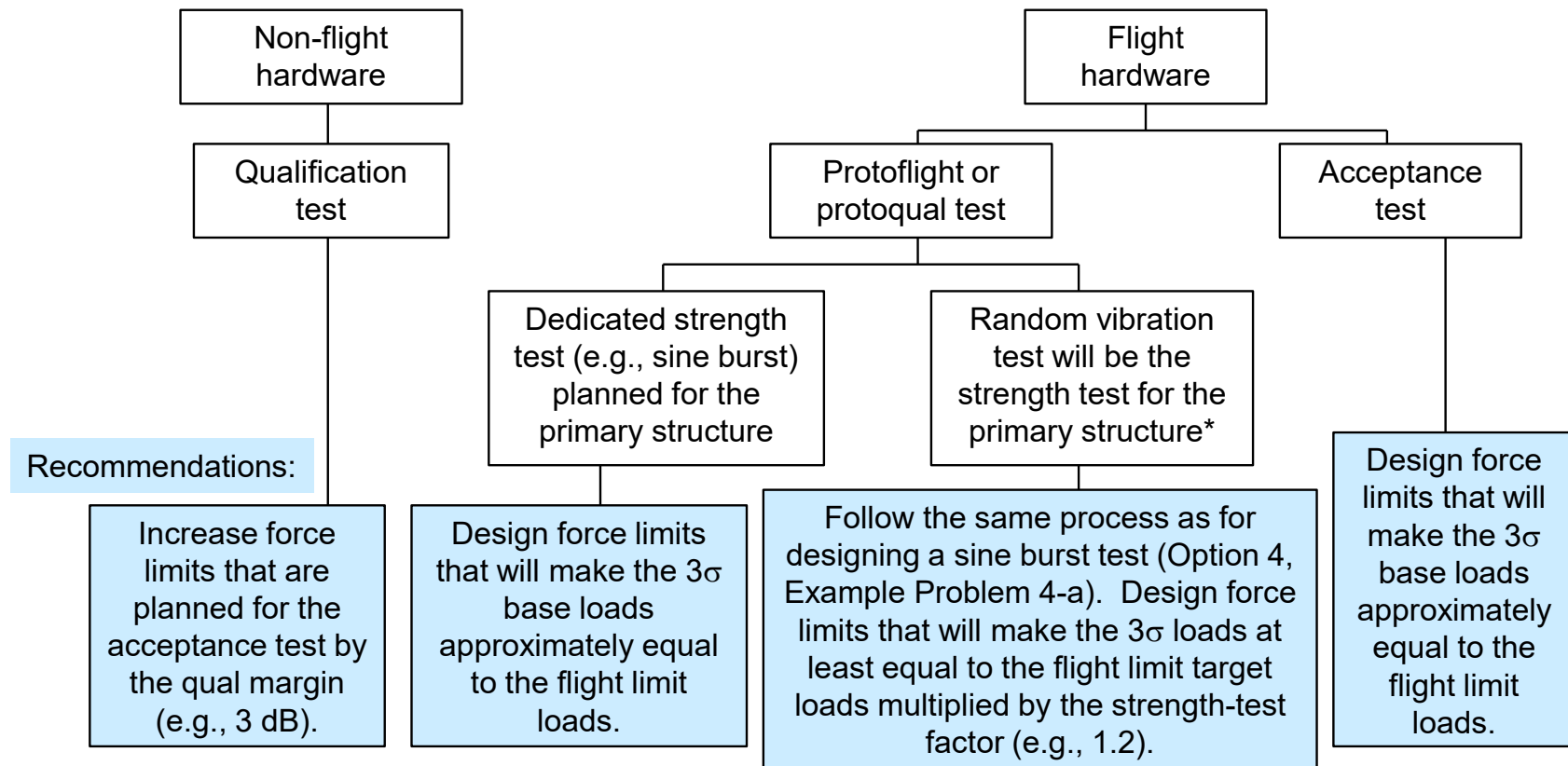
But the RMS base force is now 1390 lb, which is 36% higher than the target value of 1026 lb.

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

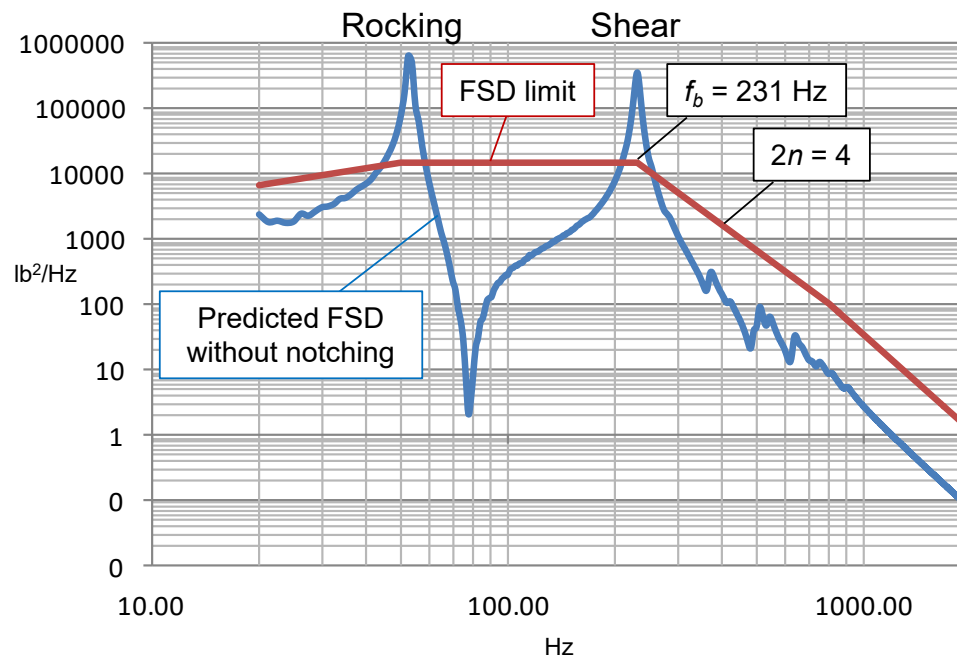


* As discussed in Sec. 4, the random vibration test should not be the strength test for the primary structure unless mass is low enough that the structure can be designed to have the needed strength and fatigue life for the planned test, accounting for uncertainty in response loads.

Example Problem 6-b: Force Limiting

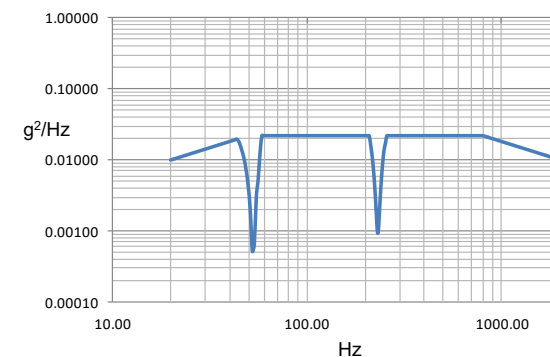
In this example, the 362-lb test article has two modes that have high effective mass in the lateral direction: a rocking mode and a shear mode (shearing of the separation mechanism).

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



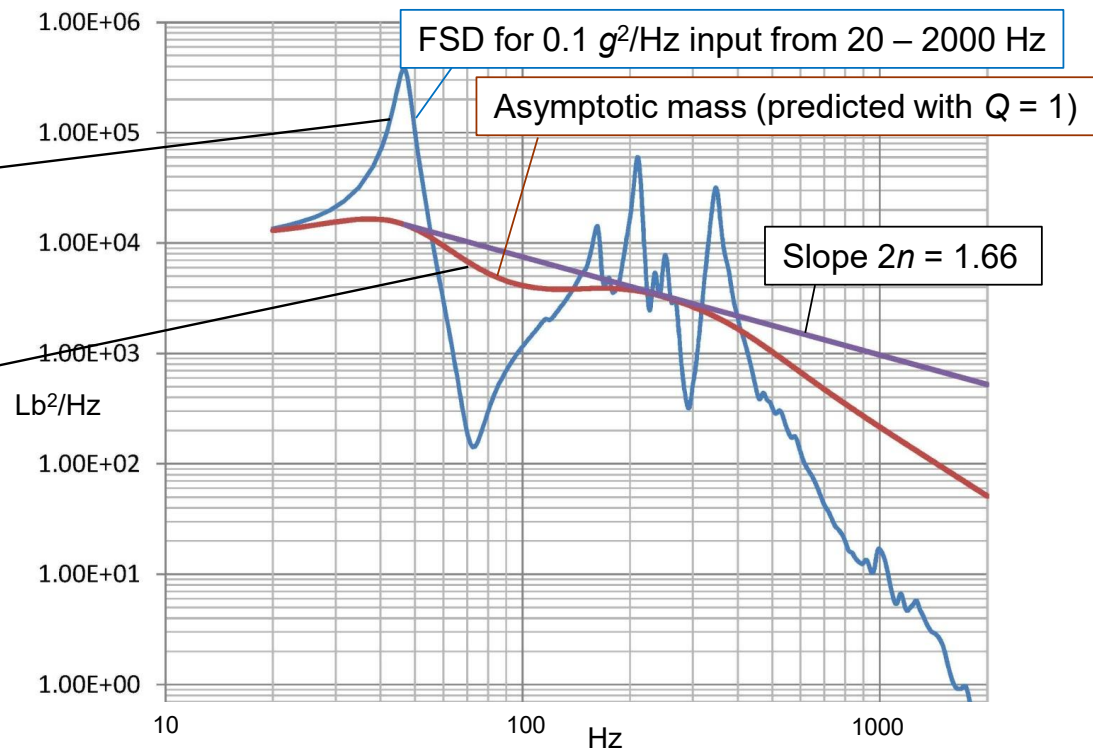
Example Problem 6-c: Calculating the Asymptotic Mass

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.

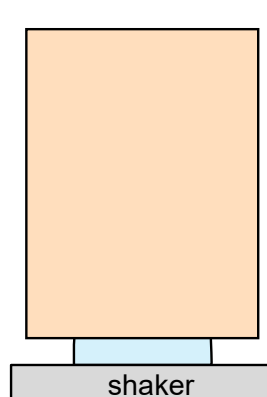
You can use either response limiting or manual notching to simulate force limiting, with the difference being that you have to trust your finite element model to predict the base force or moment.

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

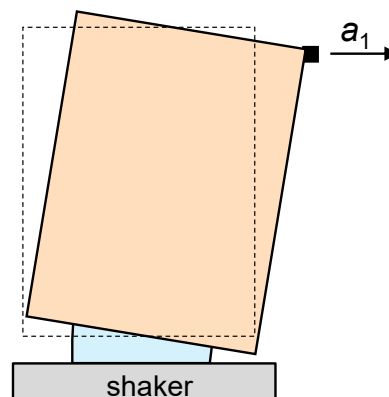
Example Problem 6-d: Response Limiting

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

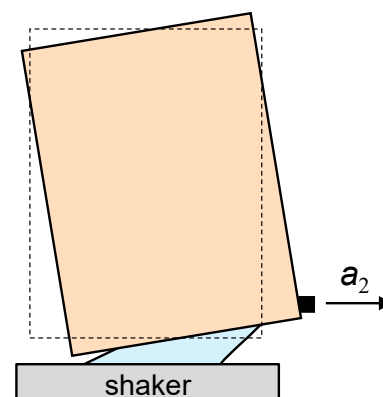
- Two modes with high effective mass for lateral testing: rocking and base shear. (Example Problem 6-b is for a small satellite with modes such as these.)
- Two accelerometer locations measuring accelerations a_1 and a_2 .
- Within the frequency range in which response is dominated by the first mode, 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

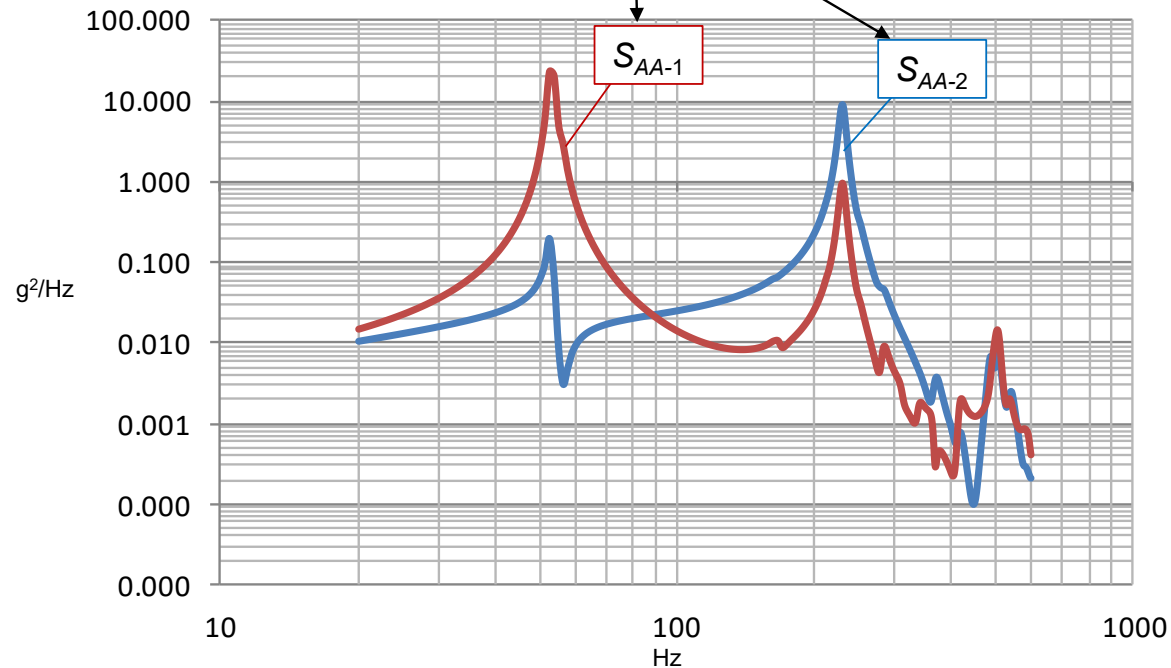
If we have high confidence in the FEM for these two modes, we can derive response limits for these two accelerometers to simulate force limiting.

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)

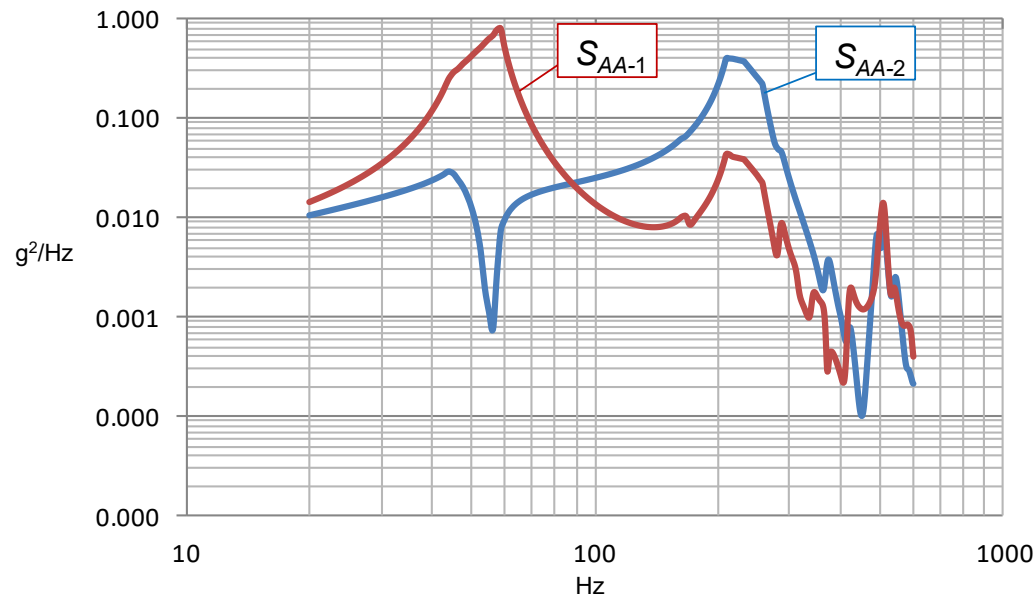
Corresponding to accelerations a_1 and a_2 , respectively, from previous page



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.



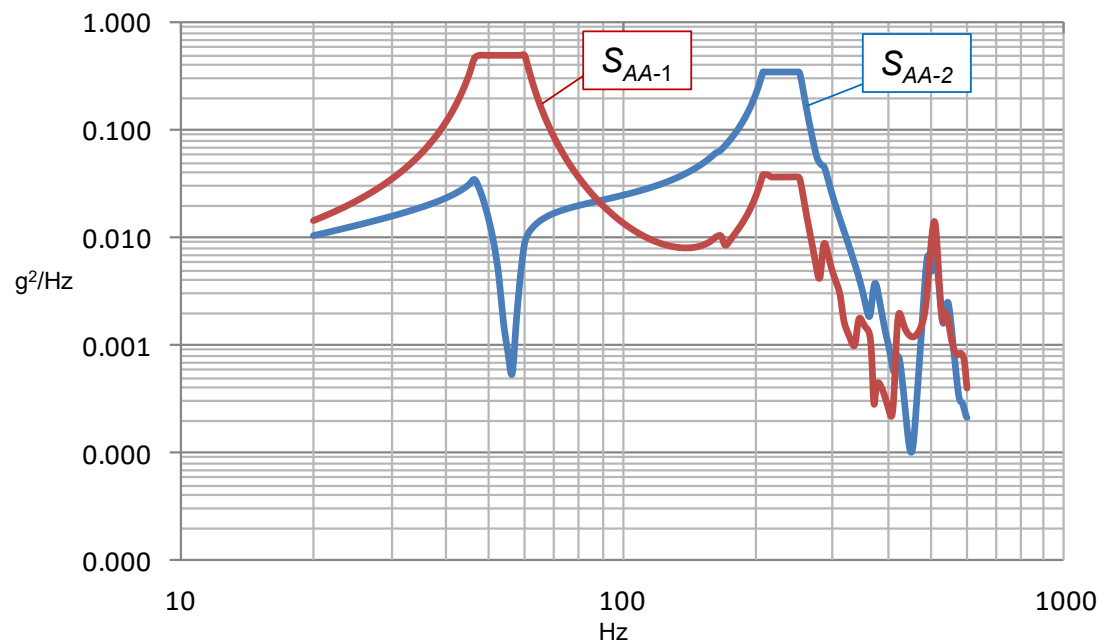
If base force and acceleration— a_1 for the first mode and a_2 for the second—were truly proportional within the target frequency bands, the peaks for S_{AA-1} and S_{AA-2} would be truncated with the same shapes as the FSD peaks, or nearly flat, as they would be for true force limiting (Example Problem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

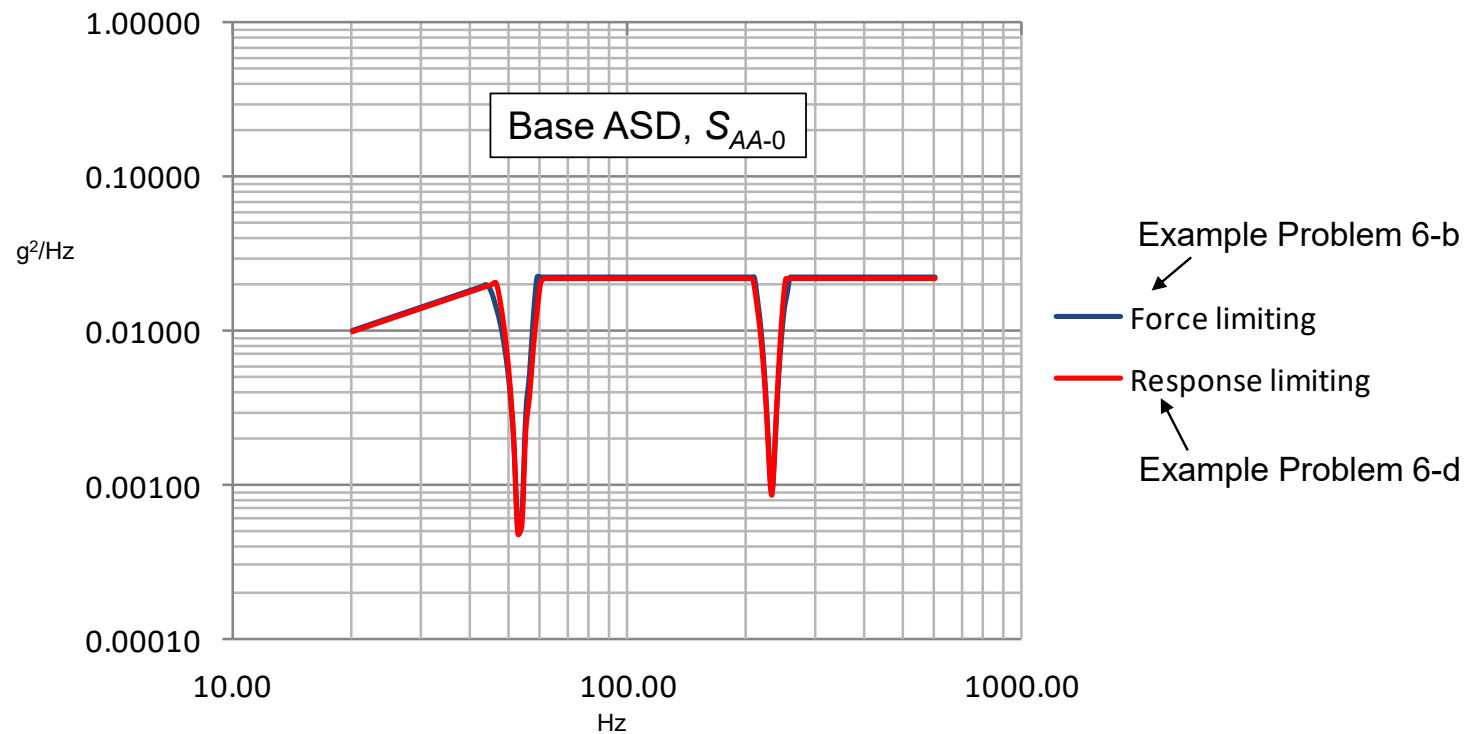
Flattening the peaks with response limits of $0.50 \text{ g}^2/\text{Hz}$ for S_{AA-1} between 20 – 100 Hz and $0.35 \text{ g}^2/\text{Hz}$ for S_{AA-2} between 150 – 300 Hz:



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

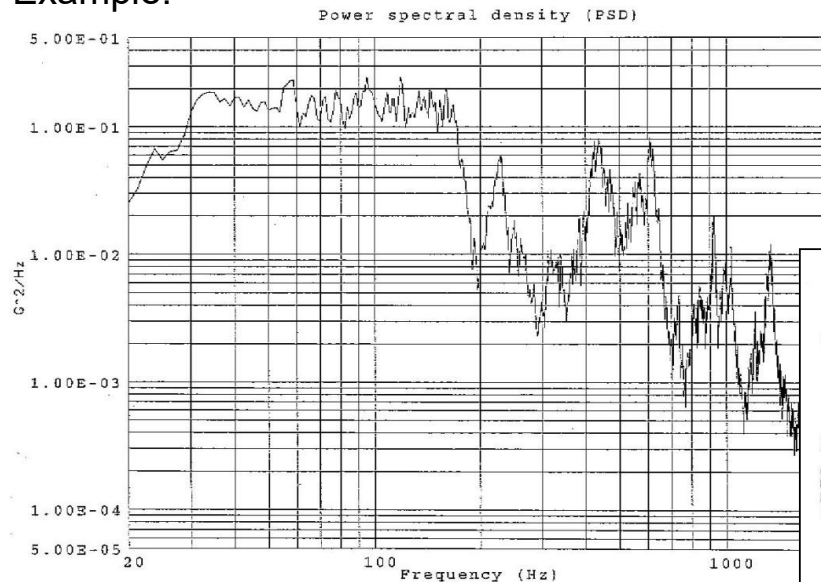


Conclusion: If our model is accurate, the notches are nearly the same as we'd get with true force limiting.

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

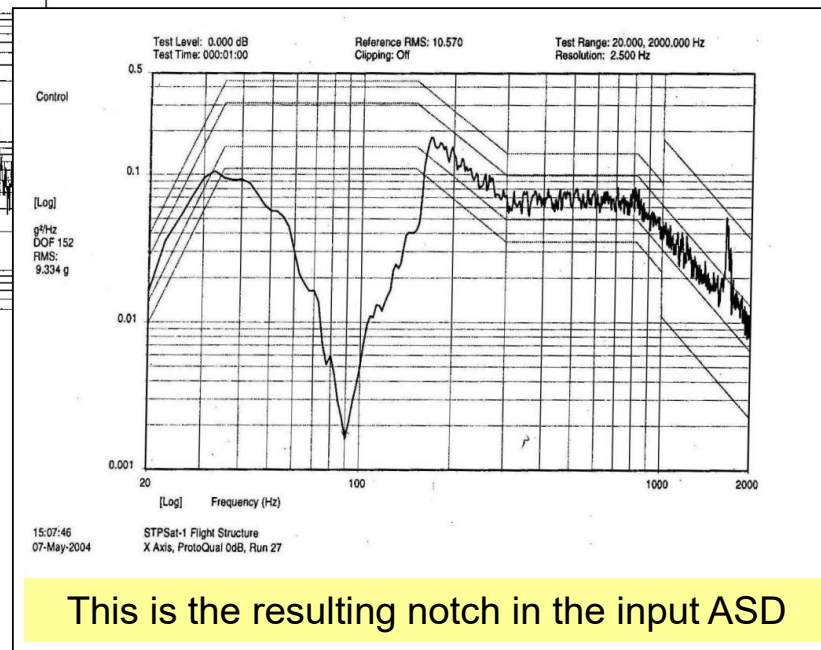
Response Limiting Requires Confidence in How the Response Acceleration Relates to Base Force

Example:



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

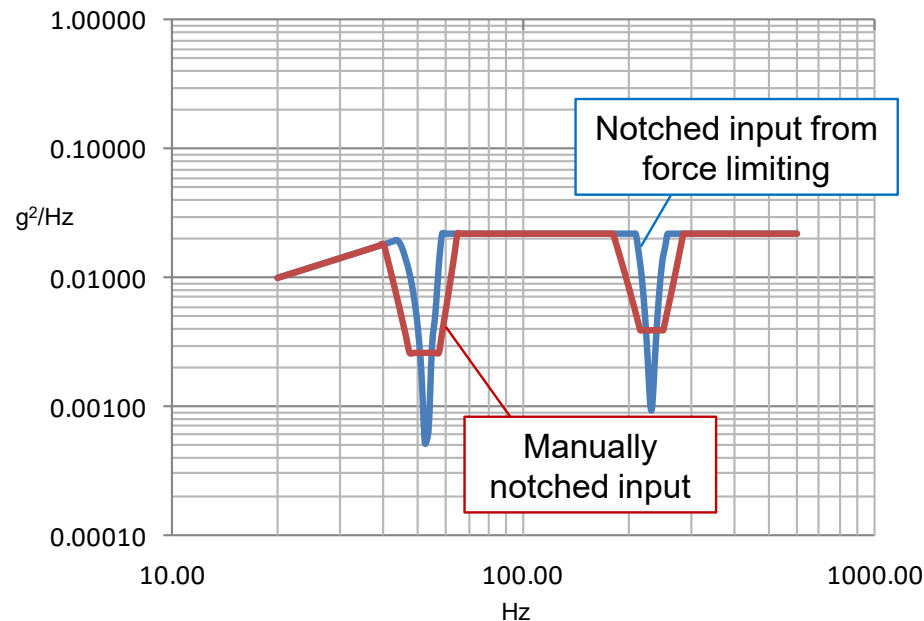
To avoid excessive loading for a 90-Hz mode, the shaker was controlled between 30 and 170 Hz by an accelerometer at the test article's CG. But acceleration at the CG was amplified by a local mode.



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.

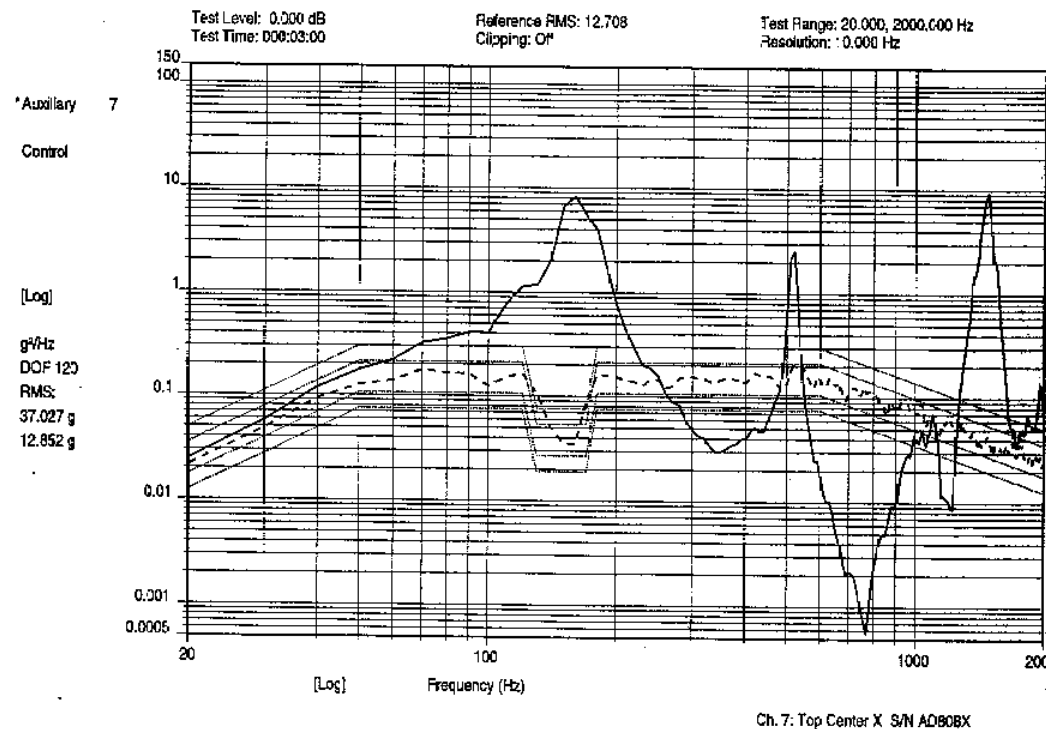


To generate these notches, I put the model-predicted data into a spreadsheet and assumed response scales linearly with input.

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”.

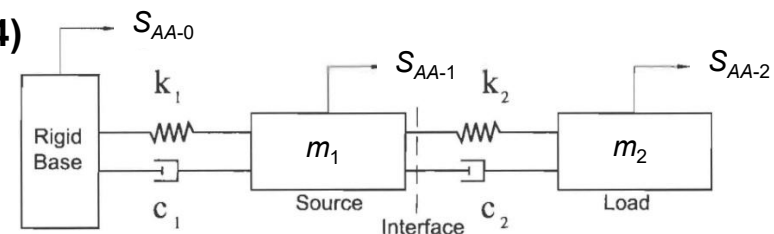
$S_{FF-\max}$ = maximum base FSD
 S_{AA-0} = base ASD (white noise)
 f = frequency
 f_{n2} = natural frequency of the load by itself
 m_2 = SDFS (load) mass
 Q = quality factor

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 $f = f_{\text{sys-1}}$



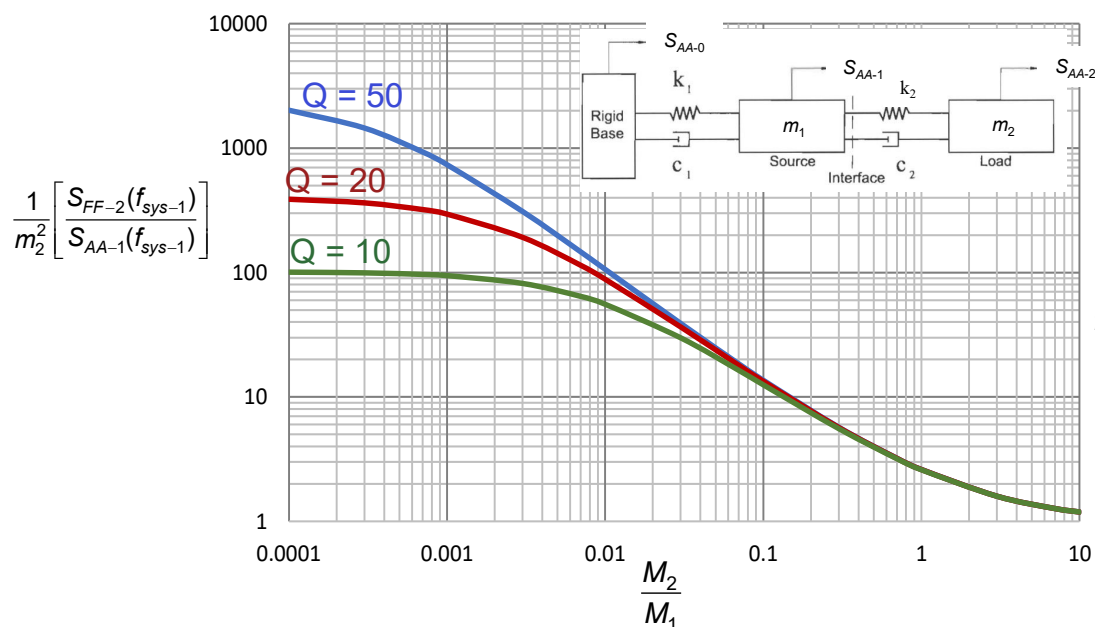
continued

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,
STD I Ref. 26)



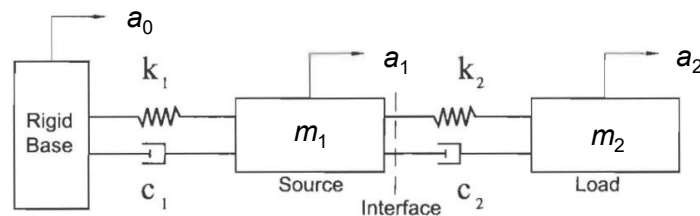
f_{n1} = natural frequency of the source as an SDFS
 f_{n2} = natural frequency of the load as an SDFS
 $S_{FF-2}(f_{sys-1})$ = maximum interface FSD (at f_{sys-1})
 $S_{AA-1}(f_{sys-1})$ = interface ASD at f_{sys-1}
 Q = quality factor
 $r = \frac{f_{sys-1}}{f_{n2}}$
 f_{sys-1} = first natural frequency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



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

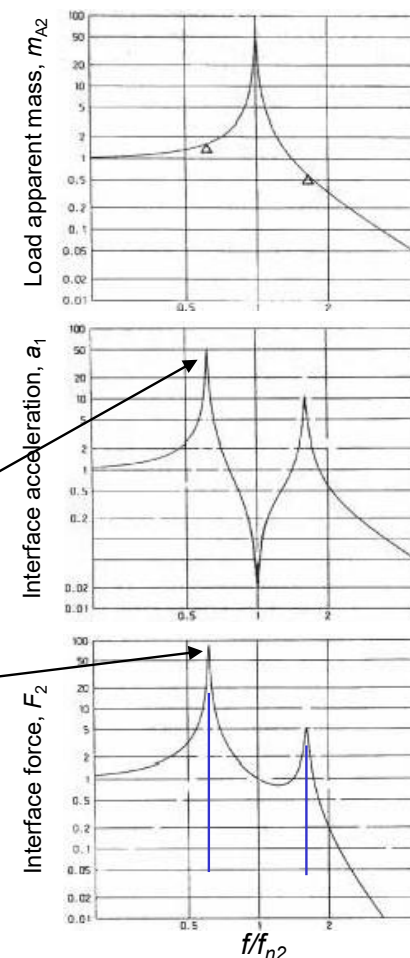
For this system driven with unit white noise
random vibration (1 g²/Hz base input),

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

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

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

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 ...

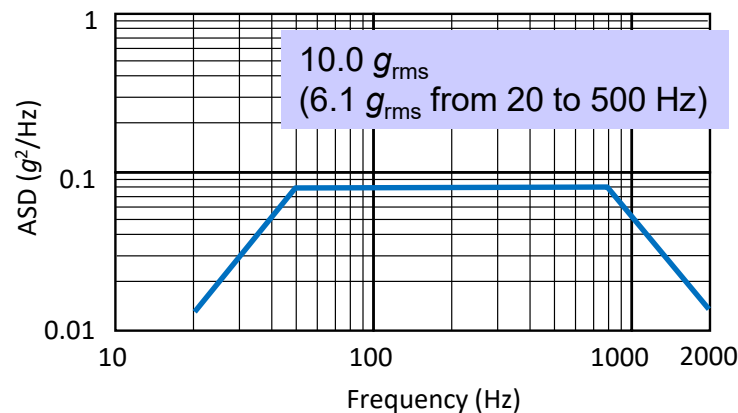
- what the FSD limit and notched base ASD would look like,
- how much the RMS base force would be reduced by force limiting,
- and how the RMS base force compares to the bus-payload interface force from the SC analysis.

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

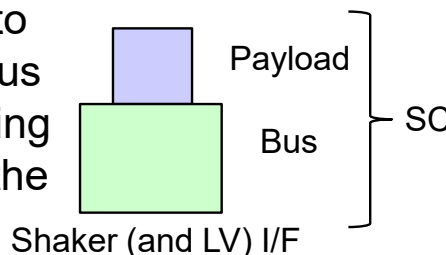
I ended the study by investigating several options for addressing these issues.

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

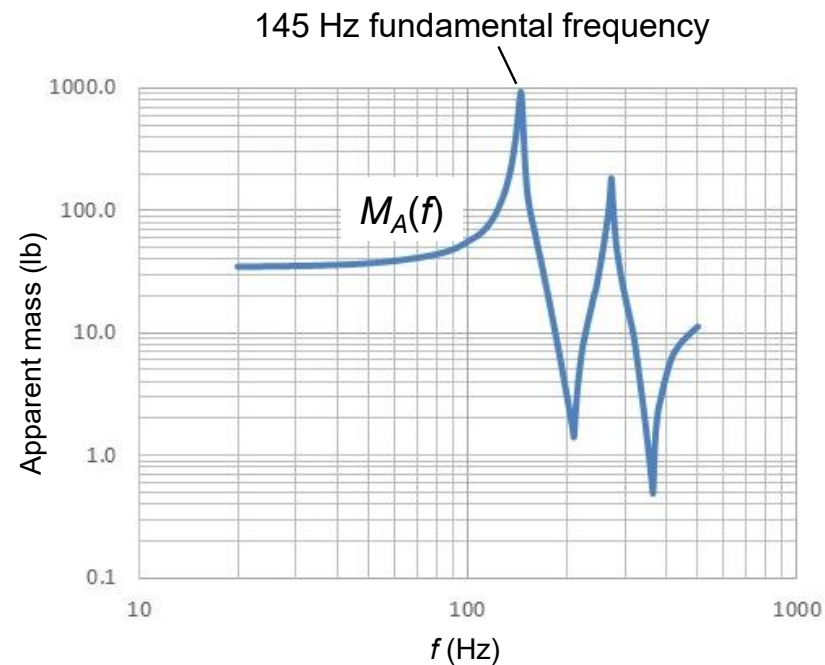
To determine the payload's apparent mass, $M_A(f)$, we base drive the payload FEM with any arbitrary ASD, $S_{AA-0}(f)$, and calculate the base FSD, $S_{FF}(f)$, assuming $Q = 20$ for this example.

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

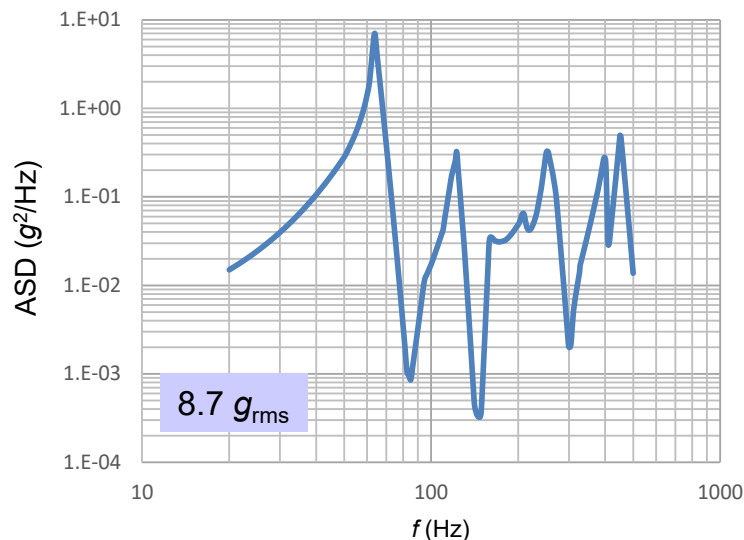
Base FSD



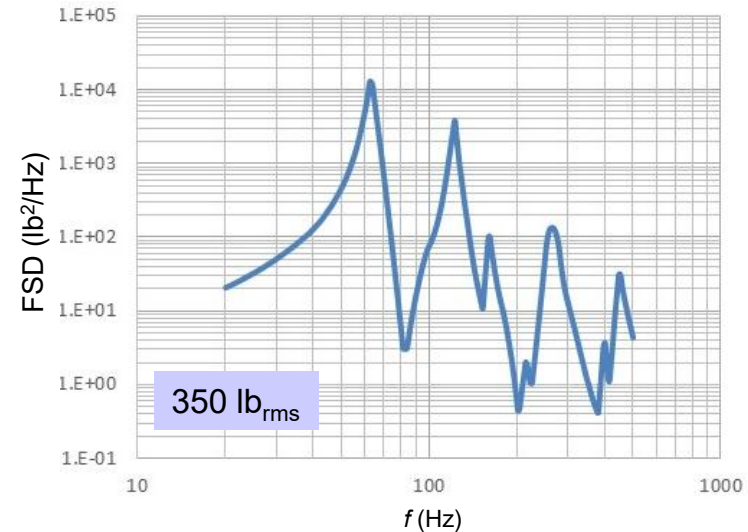
Step 2: Calculate Interface ASD and FSD for the SC Test

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)

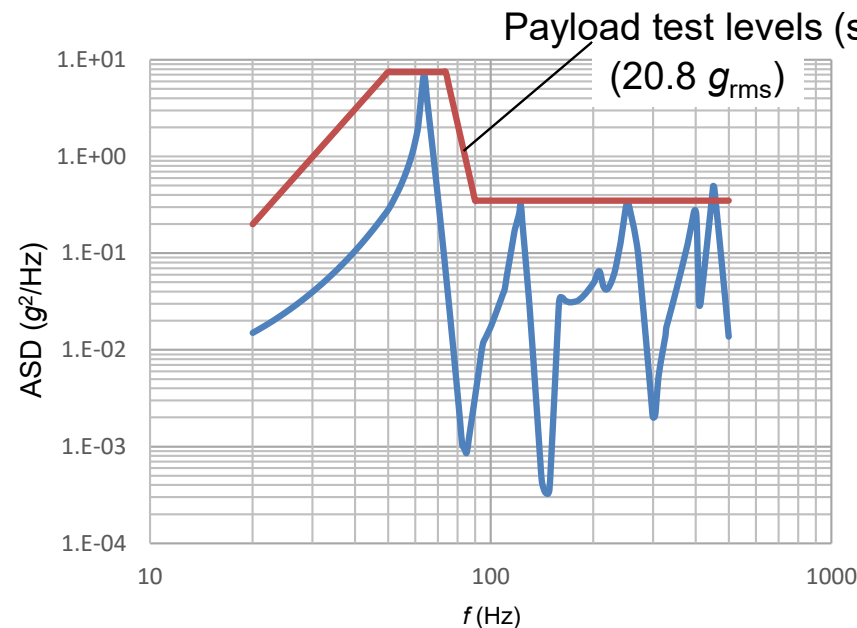


To sum FSDs at multiple interface nodes in the FEM, $FSD_{total} = \left[\sum \sqrt{FSD_{individual}} \right]^2$

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

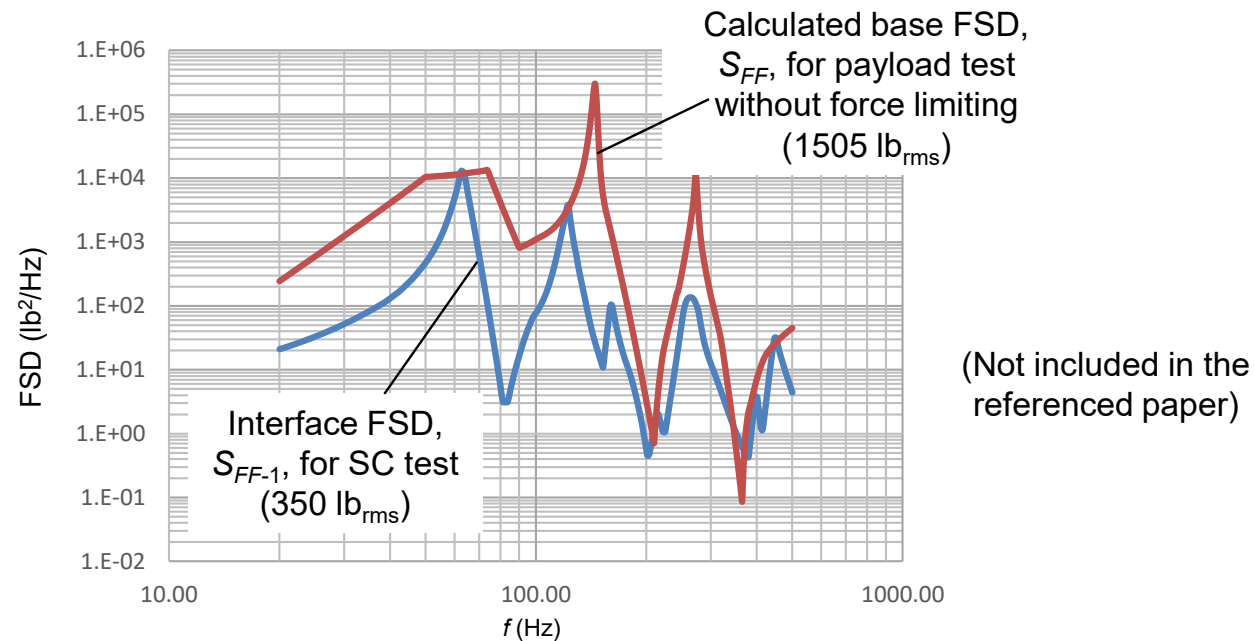
Based on the calculated interface ASD for the SC test, we can derive a test environment for the payload (from the referenced paper):



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

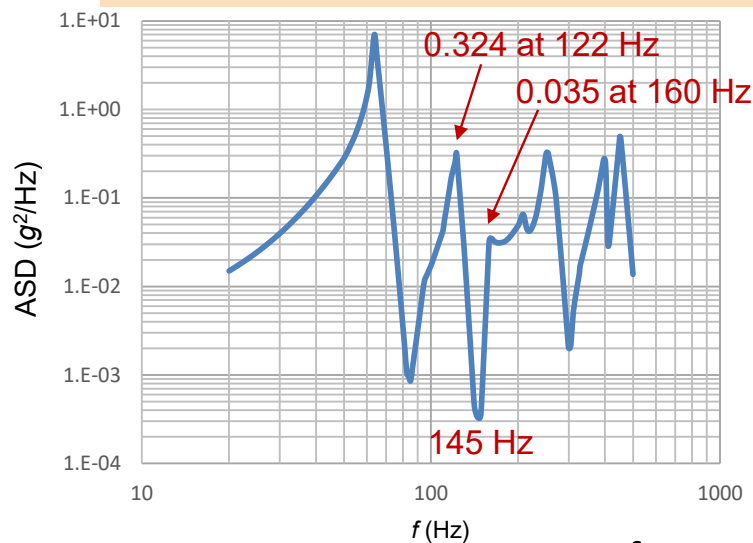
$f_{\text{max}-f}$ is the frequency of the higher of the two S_{FF} peaks on each side of the antiresonance

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

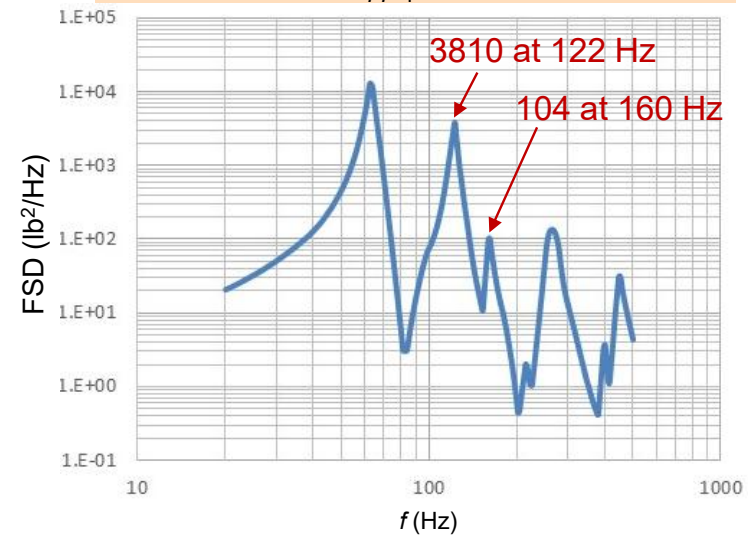
Let's do this for our example ...

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

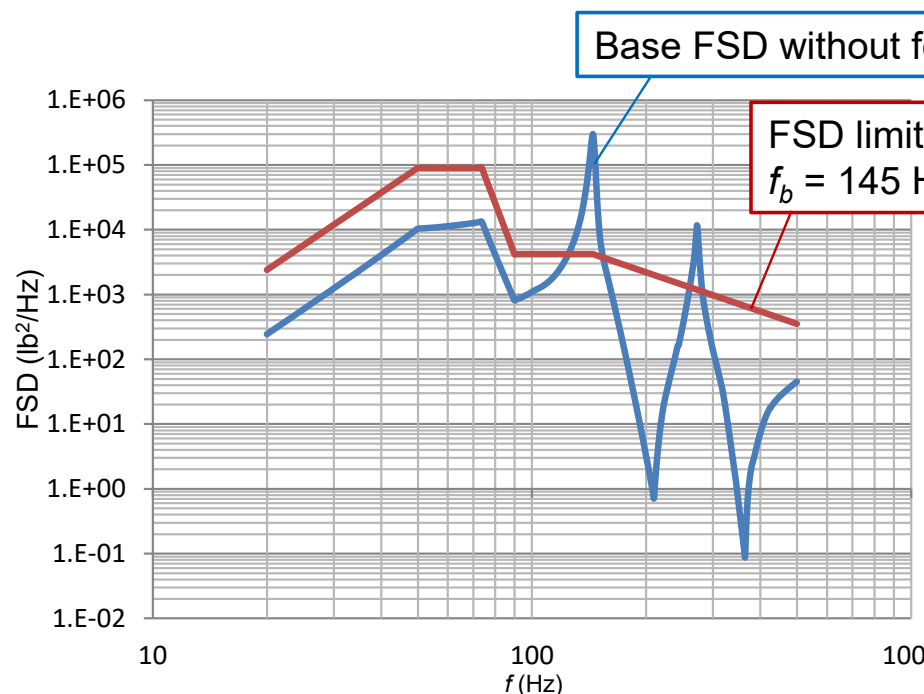


$$f_{\max-f} = f_{\max-a} = 122 \text{ Hz}$$

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

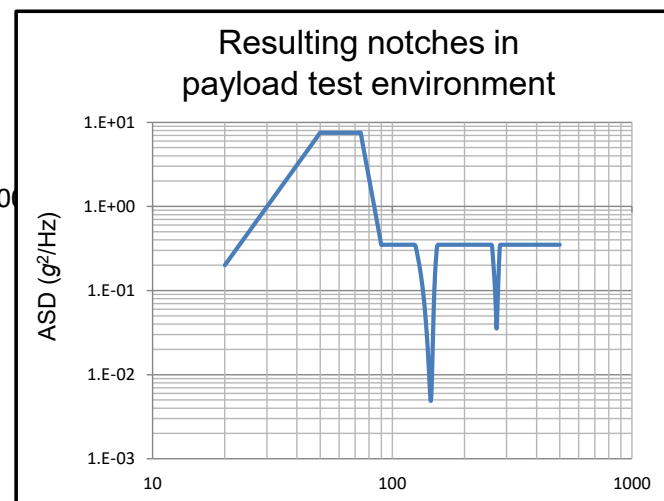
This is about as far as the referenced paper carried the example.
We'll continue it here...

Step 6: Define the FSD Limit



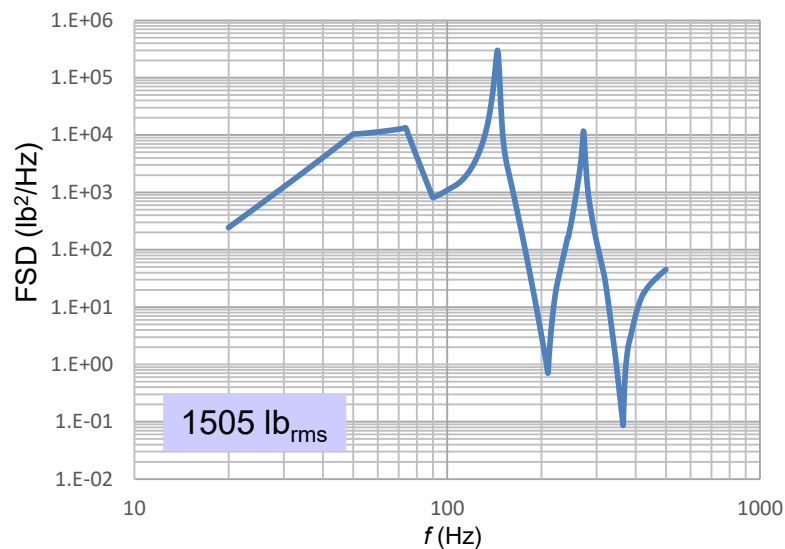
Note: The value of n was based on the approximate smooth-line average slope of the apparent mass when considered up to 2000 Hz, as shown in the referenced paper.

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.

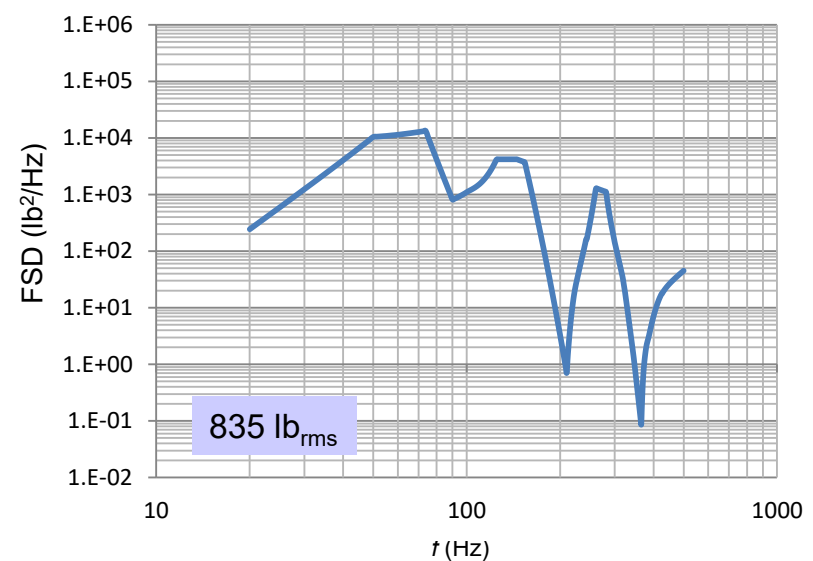


Step 7: Assess the Effects of Force Limiting

Payload-base FSD without force limiting



Payload-base FSD with force limiting



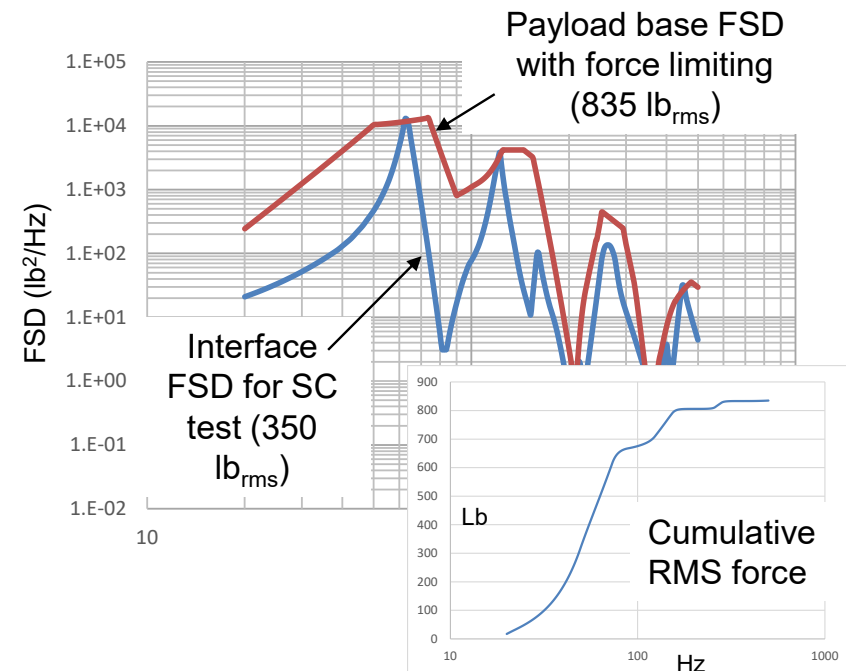
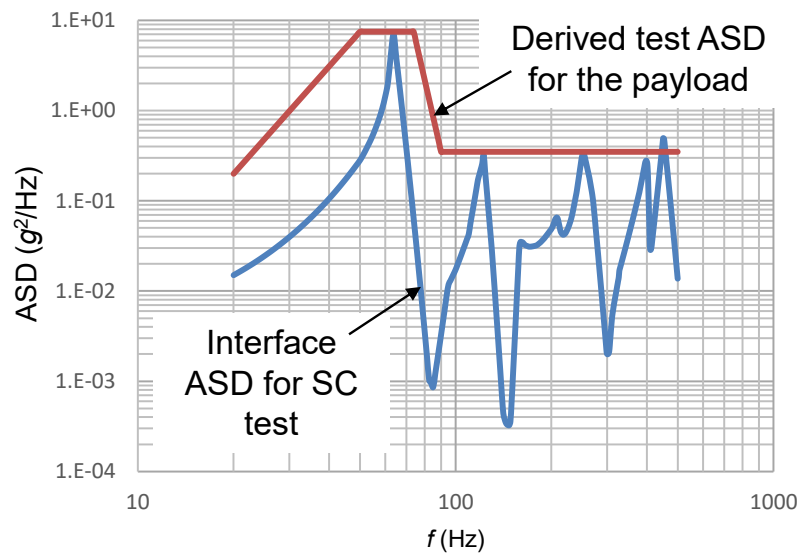
The RMS base force has dropped by about 45%.

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 $3 \times 835 = 2505 \text{ lb}$.

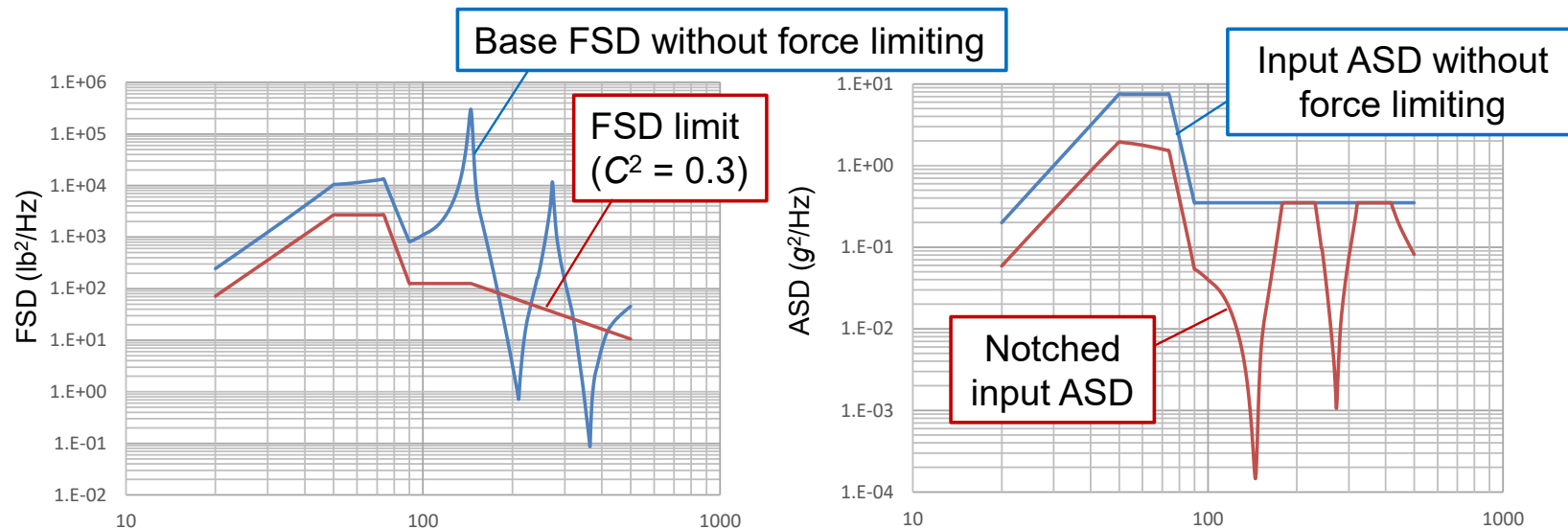
This means we would have to design the payload's primary structure to a limit quasi-static equivalent load of at least $2505/34.6 = 72 \text{ g!}$

And **121 g** if we choose to design to a 5σ load!

Option 2

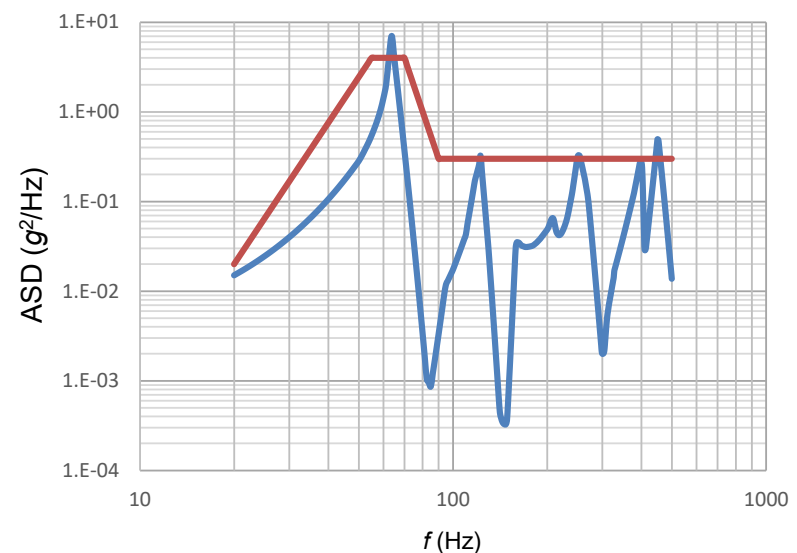
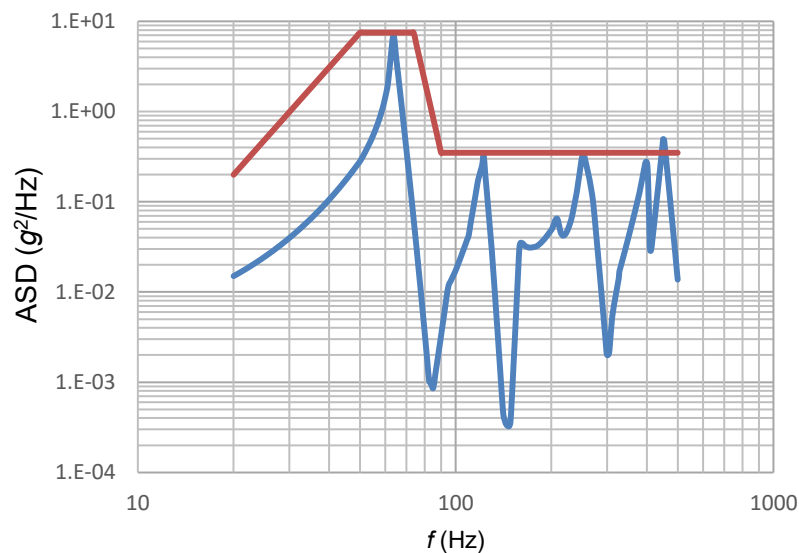
What if we set C^2 equal to the value that makes the RMS base force for the payload test equal the RMS interface force for the SC test?

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

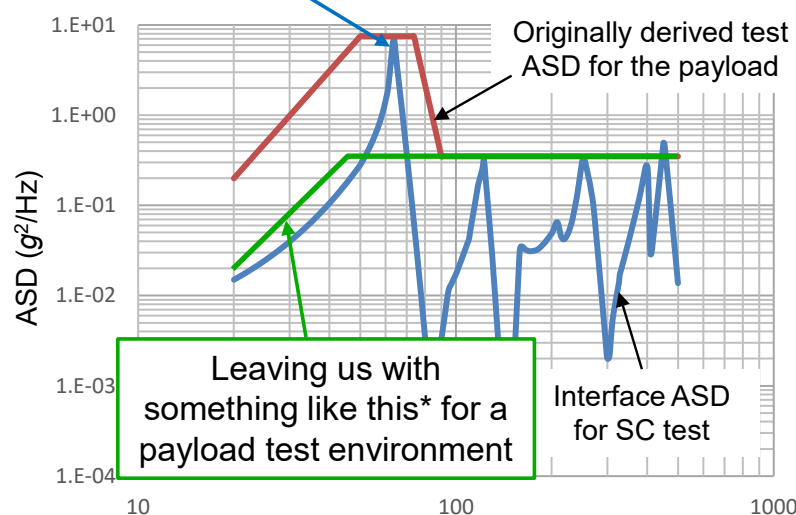


With the less-conservative enveloping shown above right, the base force for the force-limited ($C^2 = 10$) payload test drops from 835 lb_{rms} to 638 lb_{rms} (vs. 350 lb_{rms} interface force for the SC test).

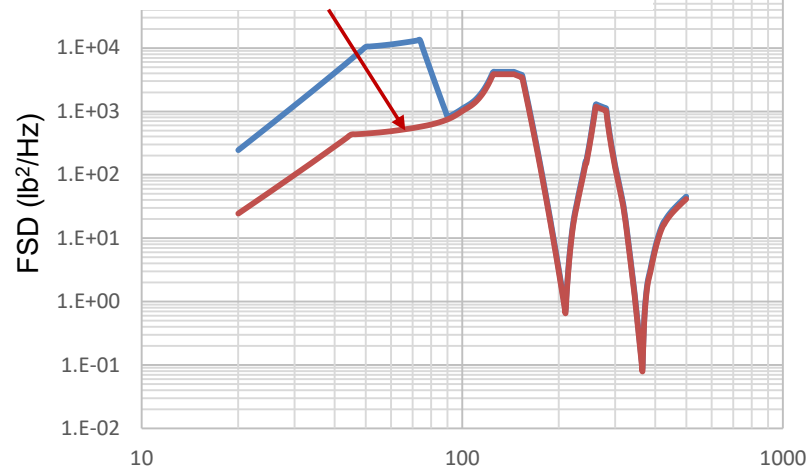
Helps, but doesn't solve the problem

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.



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})



*I do not have the data needed to derive force limits for the SC test in this example.

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:

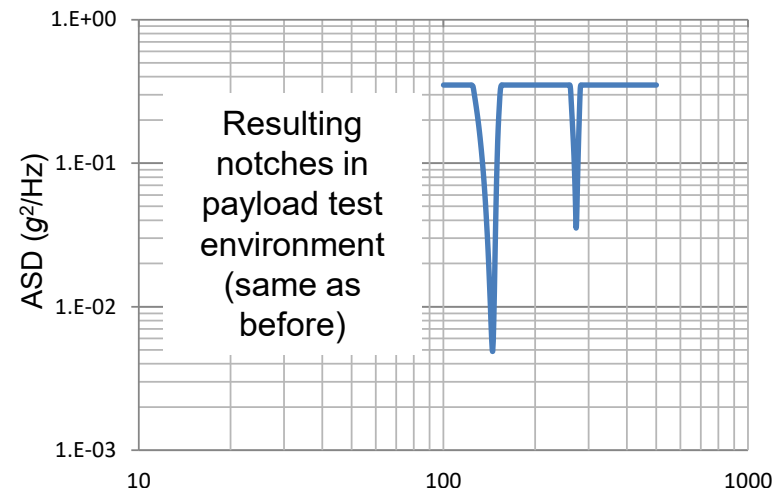
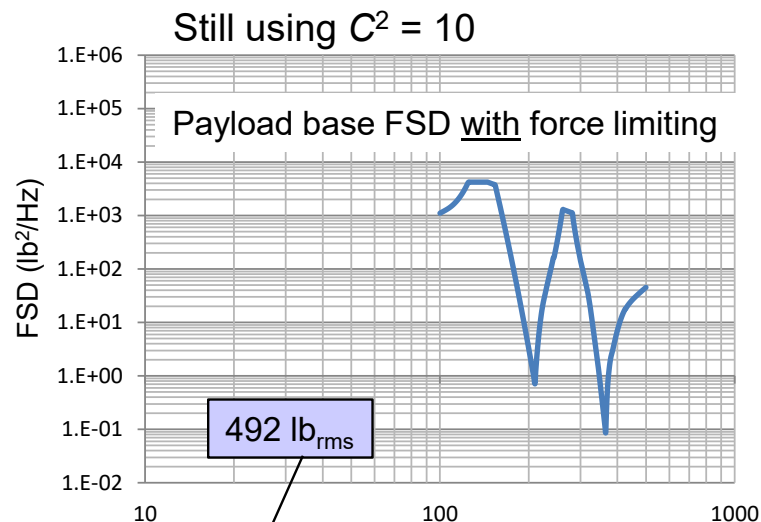
1. Low frequency, from 20 – 100 Hz without force limiting (base force is 448 lb_{rms})
2. High frequency, from 100 – 500 Hz with force limiting (base force is 492 lb_{rms})

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. Overview of Other Types of Structural Tests

- Modal Survey Testing and Correlation of the Finite Element Model
- Centrifuge Testing
- Acoustic Testing
- Shock Testing

This section includes contributions from Pete Abbott and Rick Webb

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Modal Survey Testing

A **modal survey** is a test performed to characterize a structure's modes of vibration (frequencies, shapes, and damping), most often for use in correlating a finite element model (FEM).

- Specific objectives relate to how the FEM will be used:
 - Dynamic loads analysis
 - Jitter or control-structure interaction analysis
 - Designing a vibration test (see the case history in the next section of this course)
- For each type of analysis performed with the model, certain modes of vibration will be important. Others will not, such as ...
 - Modes that will not be significantly excited in the event of interest
 - Modes that have little effective mass for the boundary conditions of interest (depends on objectives, though)
 - Modes that are above the frequency range of importance

Alternatives to Modal Survey Testing

- Can you show through sensitivity studies that uncertainty in the math model is not important?
 - Perform the analysis of interest with model variations; design the product for the envelope of results.
- Can you design the vehicle or component to have a fundamental frequency that is well above (perhaps 1.5 times) the frequency range of concern (e.g., truncation frequency for coupled loads analysis)?
 - May be feasible for small assemblies
- Will you be able to acquire adequate data for model correlation with a different planned test?
 - Random vibration or sine sweep
 - Static loads testing

Inclusion of a modal survey test in a government or industry test standard is not sufficient reason to spend time and money on such a test.

Some Basics Going In:
How Do We Get Mode Shapes from Accelerometer Data?

- Recall from Sec. 5 that, for harmonic motion of a single-degree-of-freedom system (SDFS), acceleration is proportional to displacement.
- The same is true at any given accelerometer location for a particular mode of vibration for your test article.
 - Or at least close enough to being true if damping is less than about 10% of critical
- Thus, we can derive mode shapes in the form of a modes matrix from measured accelerations in multiple degrees of freedom (DOFs).
 - Each accelerometer location and axis is a DOF.
 - The modes matrix consists of relative displacements for the DOFs; each row in the matrix is a DOF, and each column is a mode shape.

Some Basics Going In: Determining Damping by Test

By exciting an SDFS and then allowing it to vibrate freely, we can determine the damping ratio by

$$\zeta = \frac{\ln(x_1) - \ln(x_2)}{2\pi n} \quad (\text{Eq. 7-1})$$

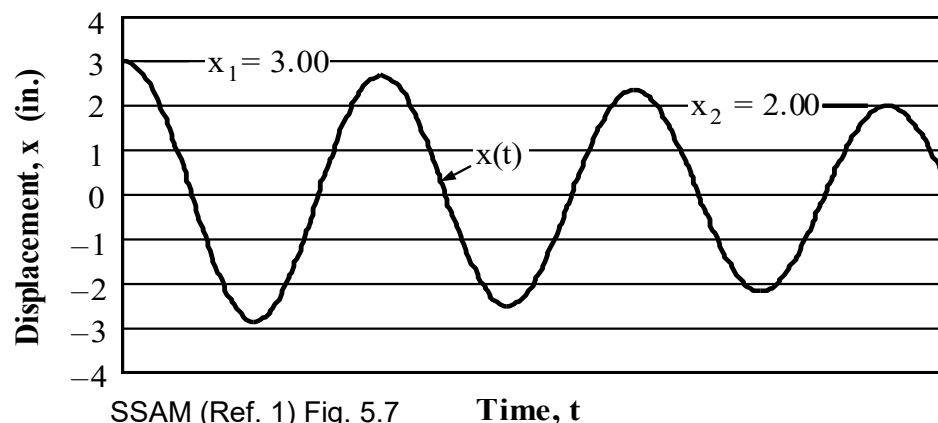
x_1 = peak displacement for one vibration cycle

x_2 = peak displacement for a later cycle

n = number of cycles separating x_1 and x_2

Example: We displace the mass of an SDFS 3.0 inches and then release it. At the third vibration cycle, we measure the peak displacement as 2.0 inches. What is the damping ratio?

$$\zeta = \frac{\ln(3.0) - \ln(2.0)}{2\pi(3)} = 0.022, \text{ or } 2.2\% \text{ of critical}$$



We can use acceleration in this equation as well because acceleration is proportional to displacement.

As long as damping is below about 10% of critical, this equation applies when measuring acceleration at any point (other than a modal node) in a structure vibrating in only one of its modes.

Nonlinearity: Linear Math Models vs. Reality

- Modal surveys are typically done at extremely low levels of excitation to minimize fatigue damage and risk of hardware failure.
- Unfortunately, many modes of vibration change with response amplitude.
 - Natural frequency (often goes down with amplitude)
 - Damping (usually goes up with amplitude)
 - Mode shape can change as well
- If the correlated model will be used for high-level excitation, plan to check linearity during test by increasing the input levels.
 - Then select the data set (at a particular input level) that is most appropriate for the model.
- Significant nonlinearity may warrant ...
 - nonlinear dynamic analysis
 - or inclusion of uncertainty factors when using the resulting “correlated” linear model.



**Mostly
because of
fastened joints**

A High-level Process for Designing a Modal Survey Test

1. Identify and clearly define the objectives and how the data will be used.
2. With pretest analysis, identify the modes of vibration that are important to the problem (**target modes**). Use sensitivity analysis to scrub the list.
3. Match the test configuration and boundary conditions to those of the event (e.g., launch) for which the correlated math model will be used.
4. Find the most practical method of exciting the target modes in a manner that will meet the objectives.
5. Identify the fewest degrees of freedom (hence accelerometer channels) needed to adequately characterize the target mode shapes.

Schedule the test early enough in the program to enable model correlation in time for the model's intended use.

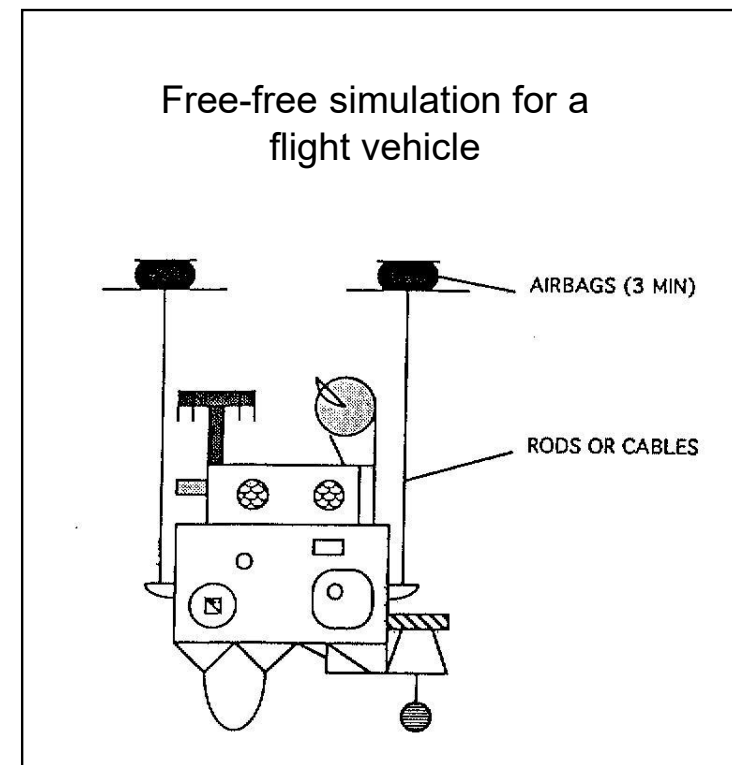
Test Configuration for a Modal Survey

The test configuration should closely match the configuration in which the model will be used.

- Mass and mass distribution
 - Actual flight (or flight-like) components for any items whose modes of vibration, damping, or stiffness contributions are important
 - Mass simulators for other missing items and electrical cables
- Boundary conditions
 - Match the test interface degrees of freedom with those in flight configuration.
 - Excitation delivered through mounting base with a shaker if target modes can be excited this way
 - Otherwise use small suspended shakers with stingers strategically located to excite the modes of interest. A **stinger** is a thin, flexible rod that transmits dynamic force in the stiff axial direction and flexes laterally.
 - If the correlated FEM will be used for on-orbit jitter analysis, simulate free-free boundary conditions. (See next page.)

Providing “Free-Free” Boundary Conditions

- Not possible here on Earth
- But you can usually test free-free modes adequately by suspending the test article from flexible (low-stiffness) supports.
 - Rods or cables with flexible airbags
 - Or bungee cords
- Design the supports such that the suspension (“bounce”) frequency is lower than 10% of the predicted fundamental frequency for the “free-free” test article.
- Deployed appendages, if they can’t support their own weight, are often omitted to avoid complex suspension systems for each. In such a case,
 - test them separately,
 - correlate models separately,
 - then combine models.



Fixed Boundary Conditions for a Modal Survey

- The mass and stiffness of the support structure should be an order of magnitude higher than that of the test article.
 - During the test, measure the response of the support structure, even when it appears “infinitely” stiff, and back the fixture effects out of the test-article measurements.
- If the modal survey will be at low levels of excitation, other sources of vibration may contaminate the data:
 - Cranes, elevators, heating, air conditioning, local traffic
 - Turn off as many operating devices as you can if they present a problem.
- We may need to either increase the input levels or provide support structure that is isolated from facility vibration with pneumatic or visco-elastic isolators.
 - or mount the test article to a massive base
- Note that electrical noise from improperly grounded instruments can be even more of a problem than facility mechanical noise.

Methods of Modal Survey Testing

- Sine dwell testing
 - Position multiple, suspended shakers (stingers) at the predicted **anti-nodes** (points of maximum displacement) for a particular mode.
 - Adjust frequency and phase of the stingers to excite the expected mode.
 - Remove the excitation and allow the vibration to decay.
 - If more than one mode is excited, the decay will not be a clean, damped sinusoid.
 - In such a case, eliminate the “beating” through adjustment of the stingers.
 - You may need one or more stingers to suppress motion at certain locations.
- More common nowadays: multi-point random—**Transfer Function Testing (TFT)**
 - Random force (random vibration) is introduced at one or more locations.
 - Response accelerations are Fourier transformed to **frequency response functions (FRFs)**, which are ratios of response acceleration to input acceleration as a function of frequency.
 - Use the FRFs to define mode shapes, natural frequencies, and damping using “fitting” algorithms such as poly-reference or eigensystem realization.
- Hybrid: **Multi-shaker Stepped Sine Testing**—combination of the above
- Tap testing—Tap the structure in a predetermined location to excite a target mode.

See MIL-HDBK-340A (Ref. 31) for more information.

Make Sure the Data Channels Can Adequately Capture the Target Modes

Mathematically reduce the finite element model to the test DOFs defined by the accelerometer set.

Pretest orthogonality check:

$$[\Phi_{\text{FEM}}]^T [M] [\Phi_{\text{FEM}}] = [I + \varepsilon]$$

(Eq. 7-2)

where

$[\Phi_{\text{FEM}}]$ = modes matrix from FEM, partitioned to test-channel DOFs

$[M]$ = mass matrix from FEM, reduced to test-channel DOFs

$[I + \varepsilon]$ = identity matrix plus some error, ε , in diagonal and off-diagonal terms

Reasonable goal: off-diagonal terms < 0.05 when diagonal terms are normalized to 1.0

A highly orthogonal (near identity) matrix assures ...

- the reduced mass matrix has adequate fidelity for the target modes
- and the selected instrumentation set can adequately define the target mode shapes.

Make Sure the Test Adequately Captures the Modes

Test data orthogonality check:

$$[\Phi_{\text{test}}]^T [M] [\Phi_{\text{test}}] = [I + \varepsilon]$$

(Eq. 7-3)

where

$[\Phi_{\text{test}}]$ = modes matrix from test data, partitioned to test-channel DOFs

$[M]$ = mass matrix from FEM, reduced to test-channel DOFs

$[I + \varepsilon]$ = identity matrix plus some error, ε , in diagonal and off-diagonal terms

Typical criterion or goal: off-diagonal terms < 0.10 when diagonal terms are normalized to 1.0

Things that affect the success of the test-data orthogonality check:

- Accuracy of the mass matrix
- Quality of test data

It's important to go into the test with an accurate mass matrix, which is derived from the finite element model.

Then you'll know that poor orthogonality between modes is the result of poor test data and can investigate the reasons.

Typical Criteria for FEM Correlation

Natural frequencies:

- Per NASA-STD-5002A (Ref. 32): $\pm 5\%$ for all significant modes.
 - For correlating with a stiffness test, stiffness agreeing within 10% is consistent with 5% on natural frequency.
- Per MIL-HDBK-340A (Ref. 31): $\pm 3\%$

Mode shape—Cross-orthogonality check: $[\Phi_{\text{test}}]^T [M] [\Phi_{\text{FEM}}] = [I + \varepsilon]$ (Eq. 7-4)

- Per NASA-STD-5002A: diagonal terms (DT) > 0.90 , off-diagonal terms (ODT) < 0.10
- Per MIL-HDBK-340A: $0.95 \leq \text{DT} \leq 1.00$, $\text{ODT} \leq 0.10$
- Poor cross-ortho checks can result from small errors in modes that are closely spaced in frequency. Failure to meet the above criteria in such cases is usually not important.

Such criteria are usually treated as goals, subject to practical limitations. The effects of significant model inaccuracies should be assessed with sensitivity analysis.

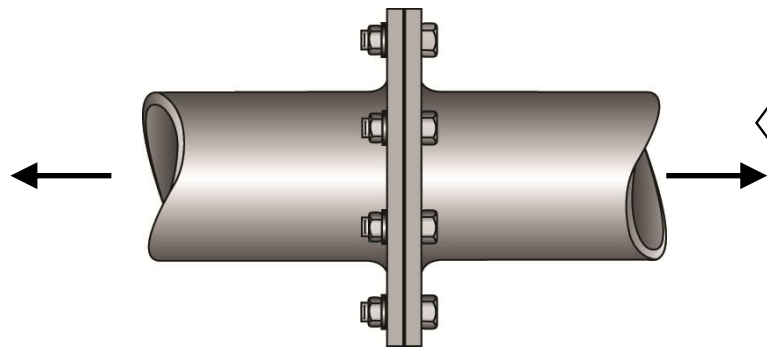
Make sure you've scrubbed your list of target modes down to the truly important ones!

Correlating the Finite Element Model with Test Data

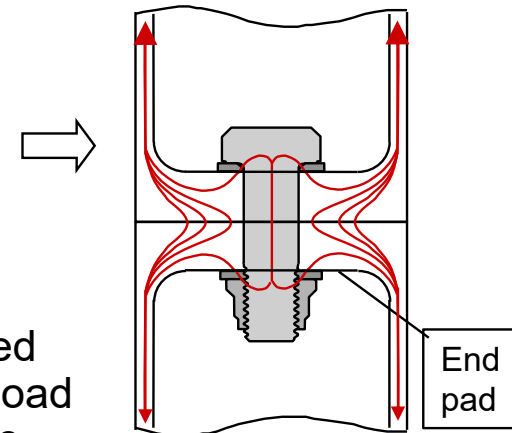
- It's important to investigate and understand where and why the FEM is wrong instead of making unrealistic changes.
 - There are many “dials” to turn when correlating a model.
 - If you're turning the right ones, non-target modes should improve in correlation along with the target modes.
 - So use all the collected modal data to help in the correlation effort.
 - Once the structure is out of test configuration and in flight configuration, any unrealistic model changes may make the model worse rather than better.
- Make sure the error is in the model of the test article rather than the test fixtures.
 - Fixture models may require test verification.
- The most likely regions of the model to be in error are those with the most strain energy for the target mode.
 - Solvers such as Nastran provide strain-energy distribution.

Most Modeling Inaccuracies are in Fastened Joints

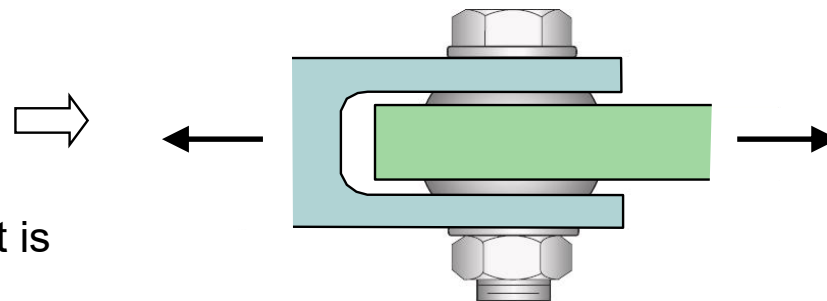
Does the model represent the load paths of a preloaded joint, in which the bolt's load path is **in parallel with** the stiffer load path of unloading the clamp load between fittings? Or is the bolt modeled to be **in series with** bending of the end pad?



Have you modeled only the tension load path, whereas the joint is far stiffer in compression?
(Improve the design!)



Lug-and-clevis shear joints under load often have significant strain energy (hence loss of stiffness) that is not captured in the model.



Failure to Satisfy the Correlation Criteria

- Remember, the “criteria” should be treated as goals!
 - Depending on the number of target modes, full compliance with the correlation criteria can be impractical.
- Use sensitivity analysis to determine an appropriate model uncertainty factor (MUF) for modes having less than desired correlation.
 - Change the model and see how much the critical analysis result changes.
 - MUF to be applied for subsequent use of the “correlated” model.
- Sometimes a MUF should be used even if the correlation criteria are satisfied.
 - Depending on how sensitive the critical answers are to the correlation error.

Keep your eye on the overall goal: a successful mission!

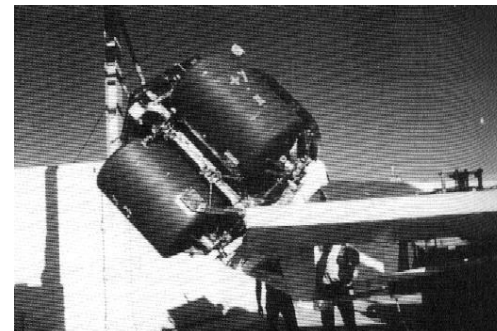
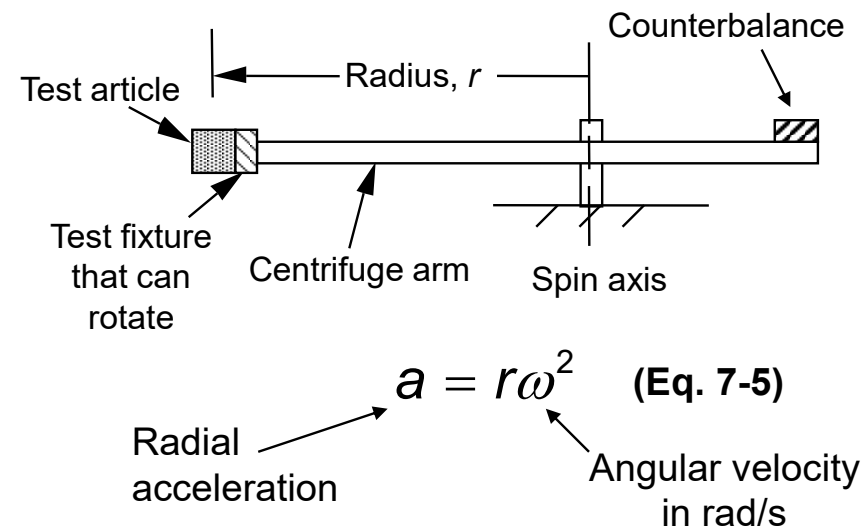
Centrifuge Testing

When to consider:

- Compact structures loaded in flight mostly by uniform (quasi-static) acceleration
- Test article too massive for sine burst testing (or concern that sine burst testing may not adequately verify strength because of potential buckling or ductile failure modes)
- Nearby, accessible test facility

Special considerations:

- One-g bias
- Sag of arm (can change load axis)
- Effect of non-constant radius
- Air drag



Propulsion module on Sandia Labs centrifuge, Fig. 11.8 from Ref. 1 (SSAM)

Centrifuges at Sandia National Laboratories



29-foot indoor centrifuge at Sandia National Lab in Albuquerque. This centrifuge is housed in an underground room that is 80 feet in diameter and 12 feet high, allowing a relatively clean and temperature-controlled environment. Payload capacity: 16,000 lb.

35-foot outdoor centrifuge at Sandia National Lab in Albuquerque. Payload capacity: 10,000 lb.



Centrifuge at NASA Goddard Space Flight Center



Acoustic Testing for Launch Vehicle Payloads

- Purpose:
 - Same as for random vibration tests
 - Done instead of random vibration when most of the vibration during launch is expected to be the result of the structure's response to acoustic pressure.
- Done separately on items with a lot of surface area relative to mass, such as solar panels and antenna dishes
 - We want to root out deficiencies at the lowest possible level of assembly.
 - Failure of a solar panel or other shell-like structure during vehicle-level acoustic testing can have a major impact to the program.
- Done at the vehicle level for spacecraft with large solar panels or other items sensitive to acoustics. Objectives:
 - Verify ability to withstand acoustics as a system. Prior to this test, we've predicted system response to derive subsystem environments; now we confirm it.
 - Verify integrity of mechanical and electrical interfaces.
 - Determine or verify random vibration environments for components to ensure component-level random vibration tests were adequate.

We Describe Acoustics with Sound Pressure Level

Sound pressure level (SPL) is a measure of relative pressure, equal to the root-mean-square (RMS) pressure within a frequency band, expressed in decibels.

$$P(f) = \sqrt{\frac{1}{T} \int_0^T p(f, t)^2 dt} \quad (\text{Eq. 7-6})$$

P = RMS pressure
 f = frequency
 T = duration
 p = pressure
 t = time

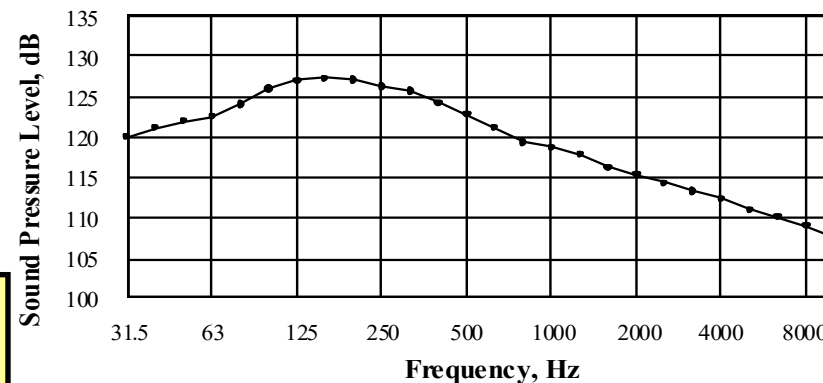
- In decibels (dB), the SPL is

$$\text{SPL}(f) = 20 \text{Log} \frac{P(f)}{P_{ref}} \quad (\text{Eq. 7-7})$$

P_{ref} = reference pressure = 2.0×10^{-5} Pa
 $(2.9 \times 10^{-9}$ psi)

- SPL is typically calculated at the center frequencies of one-third octave bands. An octave is a factor of two, so each region in which the frequency doubles has three plotted points.
- If SPL is calculated for larger frequency bands, such as full octaves, the values are higher for the same environment.

Example plot of sound pressure level:



A 6 dB increase means the pressure doubles, and so does response magnitude for linear systems.

Acoustic Test Approach

Historically, acoustic testing of spacecraft has been done in reverberant acoustic chambers.

- A **reverberant** chamber has solid walls that reflect sound waves, creating higher and higher harmonics to fill the frequency spectrum.
- In contrast, an **anechoic** chamber absorbs sounds.

Nowadays, many spacecraft are tested with a technique called **direct-field acoustic testing (DFAT)**.

- The spacecraft is surrounded by loudspeakers.
- No need for an expensive chamber; portable test equipment.
- For guidance, see NASA-HDBK-7010 (Ref. 33).



Image by Maryland Sound International (courtesy Orbital Sciences Corp., now part of Northrop Grumman)

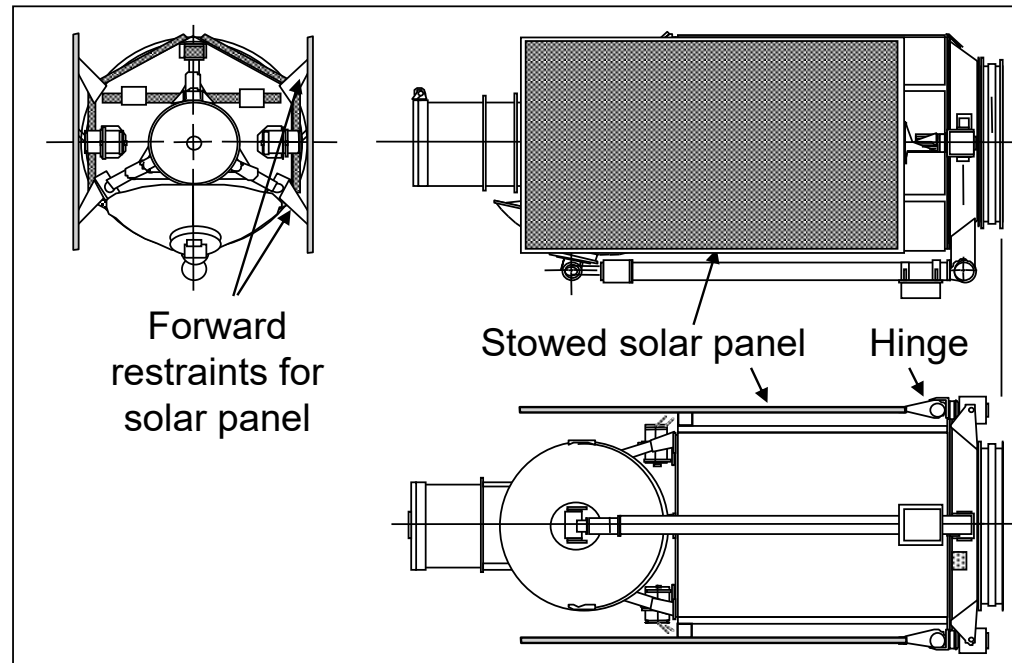
Acoustic Testing of Solar Panels: Example

Test plan:

1. Test the solar panels separately for acoustics.
2. Install the solar panels and integrate the vehicle.
3. Test the vehicle for acoustics.

Planned approach for testing the panels separately:

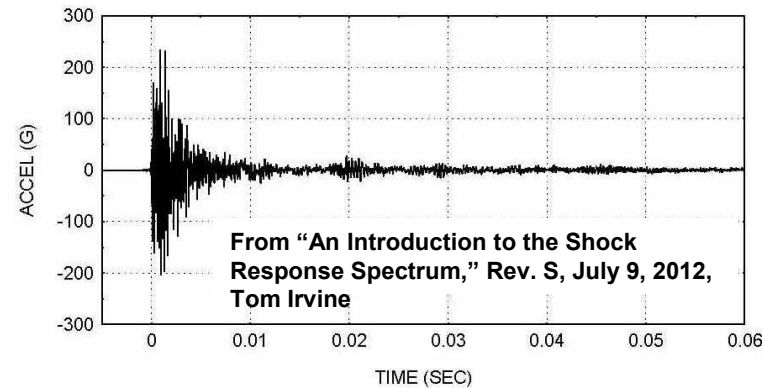
- Suspend panels from strings at two corners inside an acoustic chamber (or blast it with speakers).



The problem with this approach is that it does not adequately stress the panels. The panel should be mounted to a test fixture the same way it will be mounted to the SV bus structure.

Shock

Shock is high-intensity, short-duration vibration, with high frequency content.



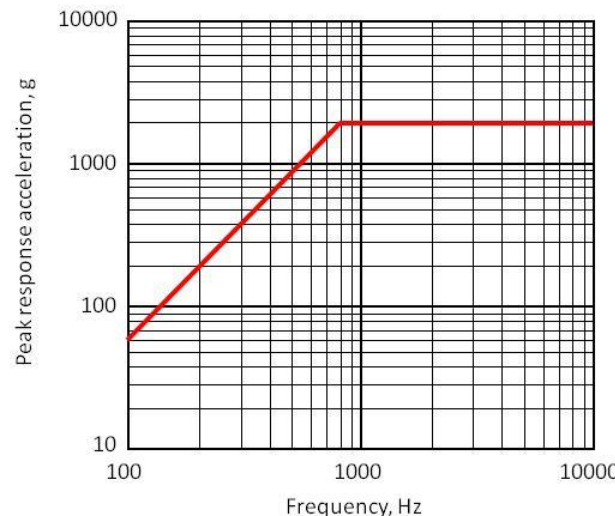
- Typically up to 10,000 Hz
- Caused by explosives (**pyrotechnic shock**, a.k.a. **pyro shock**), sudden release of strain energy, and mechanical impacts
 - Separation mechanisms are a common source of shock.
- Can damage electronics and cause electro-mechanical devices such as valves to fail
- Effects of shock are hard to predict; testing is important
- Relatively large structures are unharmed by most flight shock events, but such structures transmit shock to sensitive components.
- There's not enough energy introduced by most flight shock events to cause component housings to fail.
 - But certain test methods can cause such failures. (addressed in upcoming chart)

Shock Response Spectrum (SRS)

A shock environment is typically expressed as a **shock response spectrum** (a.k.a. **shock spectrum**), which shows the calculated response of a mass on a spring to a base-acceleration time history as a function of its natural frequency.

Example SRS →

Note: An SRS is based on some assumed level of damping (indirectly represented by Q , usually equal to 10).



An SRS is not intended for literal interpretation in deriving design loads. For example, because of its impedance (see Sec. 6), a 10-lb, 200-Hz component would NOT see an equivalent quasi-static load of 200 g in the flight event for which the SRS shown here applies.

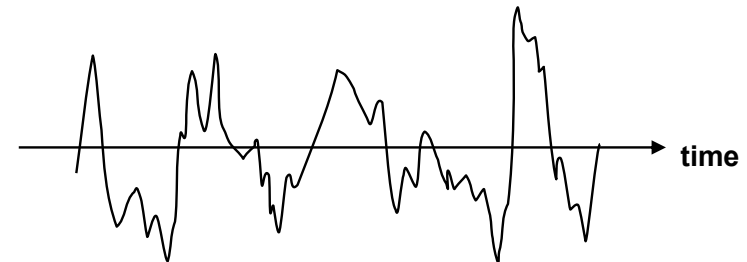
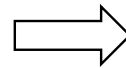
The most common failures from flight shock events are those in lightweight items made of brittle materials, such as crystal oscillators*.

*A **crystal oscillator** is a small device in which a vibrating crystal made of a piezoelectric material (typically quartz) produces an electrical signal of constant frequency; for use in clocks

For guidance on how to estimate an SRS for a component that is mounted some distance from the source, see SSAM (Ref. 1) Sec. 12.5.3 and Appendix A of GSFC-STD-7000 (GEVS, Ref. 23).

Understanding a Shock Response Spectrum (SRS)

Say we measure a time history of acceleration at a particular location (can be from shock or any other dynamic event).

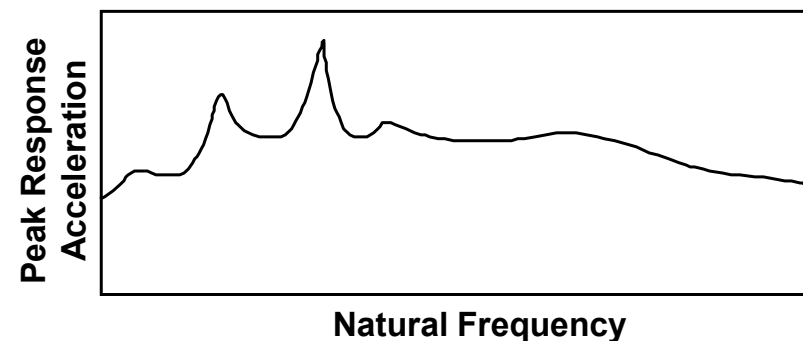


We then analytically subject a spring-mass system to this excitation, introduced at its base. Let's say the natural frequency of this oscillator is 5 Hz, and we assume a damping ratio of 5% of critical ($Q = 10$).

With these assumptions, we calculate a peak response acceleration of, say, 4.7 g.

We then repeat the analysis, but this time with the oscillator set to 6 Hz, and we compute a peak response acceleration of 5.2 g.

By repeating this analysis with incremental natural frequencies up to some upper limit, we can generate a shock response spectrum, which is a plot of peak response acceleration vs. frequency.



An SRS can be derived from any acceleration time history, not just that for shock.

Three Methods of Shock Testing

- Operation of a flight-like shock source, such as a separation mechanism
 - Most effective method but can be costly when explosives are involved
 - Typically done at the vehicle level as a combined separation and shock test; not usually practical for shock testing of spacecraft components
- Electrodynamic shaker
 - Can achieve relatively low shock levels only (limited by shaker's force rating)
 - Convenient: can test components for shock while set up for random vibration testing
 - Can significantly overstress hardware, same as for random vibration testing without notching (see Sec. 6 herein)
- Impact-type tester (often used for shock testing of equipment rather than vehicles)
 - Swing or drop a mass a controlled height onto the test fixture (**drop-hammer test**)
 - Tailor the test until the shock spectrum matches the specified one within reasonable tolerances (may not be able to get a good match at high frequencies)
 - Can significantly overstress hardware if the test fixture is massive relative to the mass of the flight structure that will support the component, especially if the test parameters are tuned to achieve the specified shock spectrum with a mass simulator present

8. Case History: Vibration Testing of a Large Spacecraft Telescope

- Overview
- Originally Planned Structural Test Program
- Problem Statement
- Revised Approach
- Telescope Testing Details
- Test Anomalies
- Lessons Learned and Conclusions

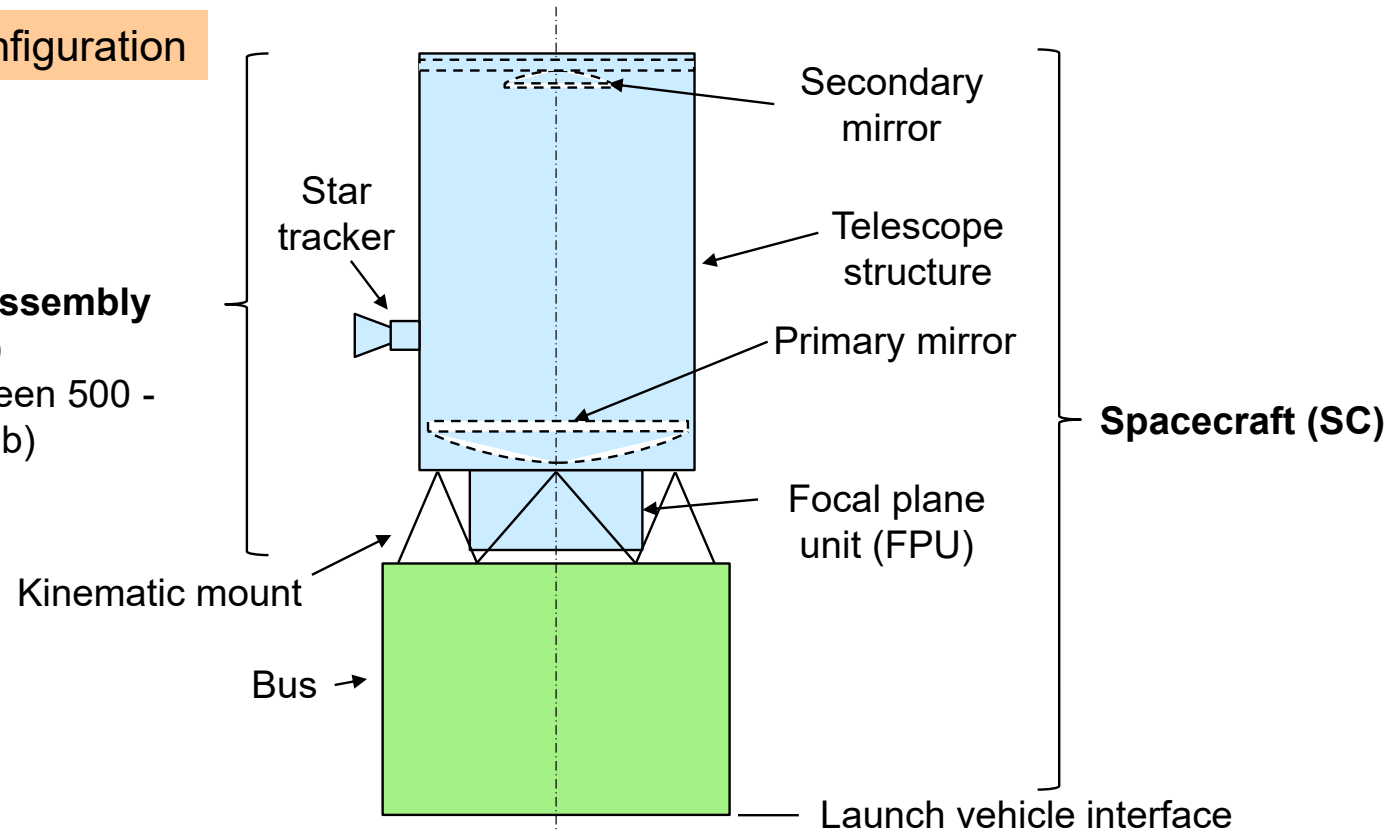
The information presented in this section is adapted from an actual case history, modified to protect company-proprietary data.

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Case History Overview

Simplified configuration

Telescope assembly (TA)
(weight between 500 - 1000 lb)



Objective: Establish a structural test program for the TA and for the integrated SC (= TA + bus).

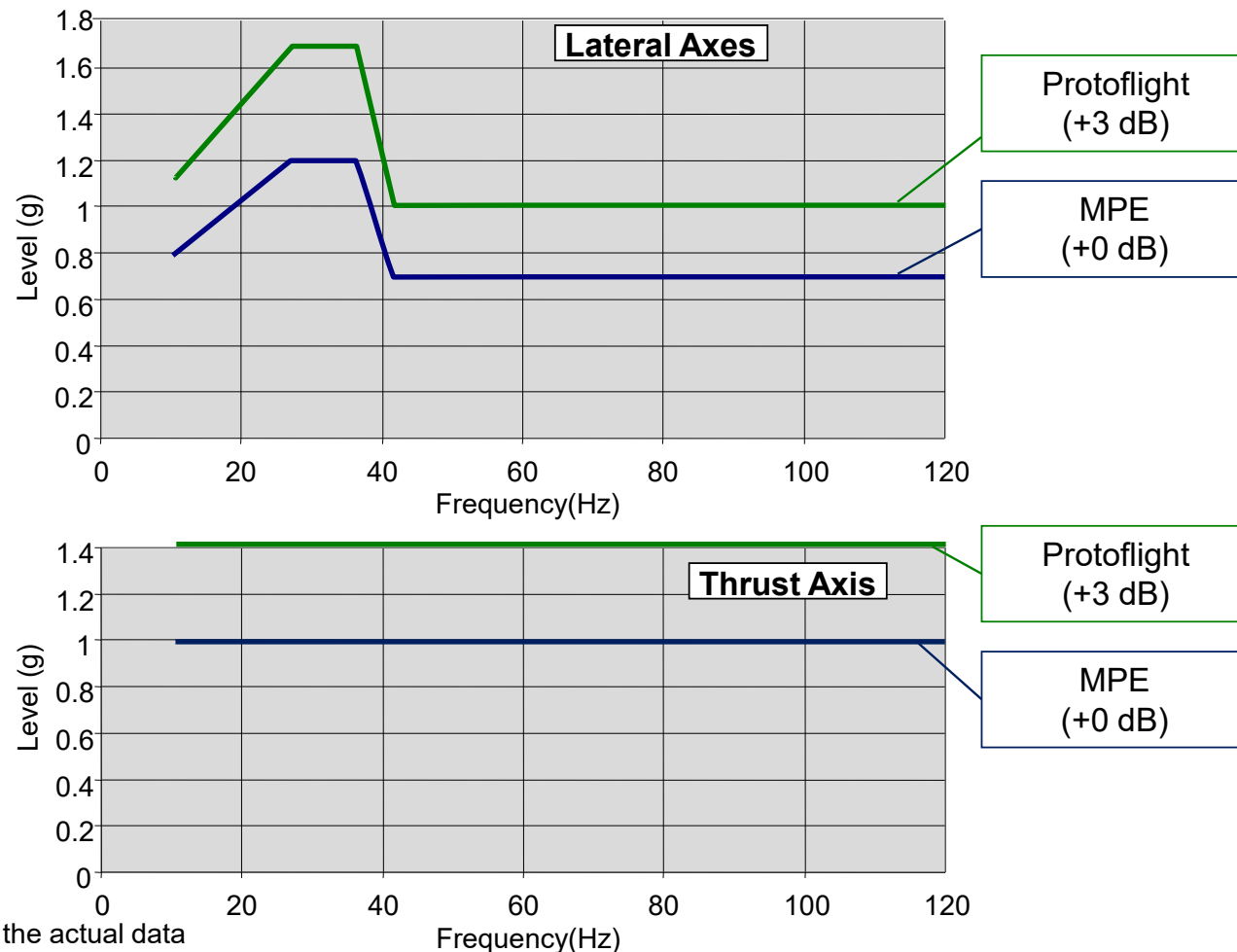
Special Challenges

- The optics (mirrors) were made of glass.
 - Brittle; could crack easily if overstressed
- The optics and other key components were expensive, with long lead times.
 - Especially the primary mirror
- The TA had extremely challenging requirements for optical surfaces and dimensional stability of the **metering structure** (structure supporting the optics and going between optics).
 - Virtually no permanent deformation tolerated
 - Often the case for optical instruments
- The telescope's performance could be tested at the TA level only, not after the TA is integrated with the bus to become the SC.
- In any testing done at the SC level, no accelerometers could be directly on the optics
 - Proximity only

Originally Planned Structural Test Program (per contract)

- Component-level random vibration testing to protoflight levels (MPE + 3 dB)
 - FPU and star tracker; bus equipment
- Modal survey of telescope structure with flight optics and mass simulators for FPU and star trackers
- Random vibration test of the telescope assembly (TA), which includes all optics, FPU, and star trackers, to acceptance levels
 - Random vibe test in two axes only (axial and one lateral)
 - Notching as required to protect the hardware, as determined by TA contractor (TAC)
 - No other structural testing planned for the TA
- Sine vibe test of the integrated SC, which includes the bus and the TA, to protoflight levels (MPE + 3 dB)
 - This test will be the strength test for the bus structure
 - Testing performed to the sine vibe environment specified by the launch-vehicle contractor (LVC)
 - Notching, if required (or desired) to protect the TA or the bus, will be determined during test, based on test data, but requires LVC approval
- Acoustic test of the integrated SC to protoflight levels (MPE + 3 dB)

Specified Sine Vibe Test Levels* for the SC

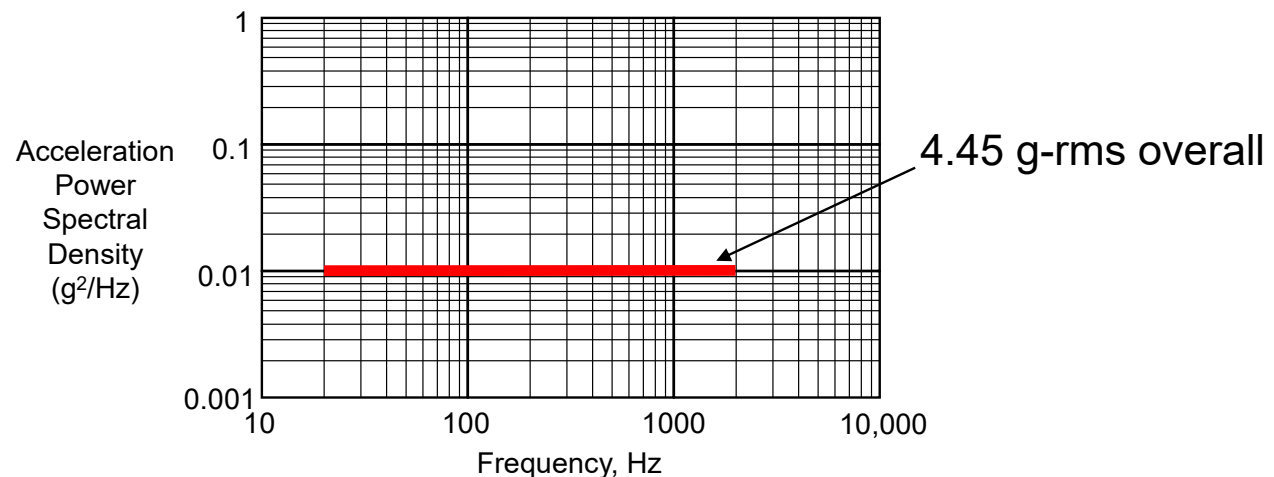


*Representative, but not the actual data

Original Random Vibration Test Environment for the TA

TA random vibration test: acceptance levels, $0.01 \text{ g}^2/\text{Hz}$ from 20 – 2000 Hz

- Thrust axis and one lateral axis only
- Source or basis unknown



What's wrong with the originally planned test program (p. 8-4)?

Problems With the Original Test Program

- TA tested to MPE (acceptance) random vibration, but integrated SC tested to protoflight (MPE + 3 dB) sine vibrate and acoustics.
 - Don't want to expose the TA in the SC test to levels that are higher than it sees during the TA test.
 - Can't do a performance test of the TA after the SC tests.
- Would not be able to accurately quantify or control response accelerations and stresses during random vibration testing of the TA.
 - Input levels typically not controlled accurately in a random vibration test; NASA-STD-7001A allows +/-3 dB tolerance on PSD at any frequency.
 - Response will be random; a high (e.g., 5-sigma) acceleration could crack the glass of the mirrors.
- Many regions within the TA structure may not be stressed adequately during random vibration testing, especially with only one lateral axis tested.
 - Structural verification would be too reliant on stress analysis.

continued

Problems With the Original Test Program, continued

- No criteria established for notching during the TA random vibe test.
 - Based on how high the TA contractor feels the hardware can be stressed without risk of damage.
 - Not tempered by the objective of adequately testing the hardware for the launch environment.
- No criteria established for notching during the SC sine-vibe test.
 - Would not know until the SC sine vibe test whether the LVC will approve the requested notches.
- The delicate TA would be unnecessarily risked during SC sine vibe testing in order to meet the objective of structurally testing the bus.

Class Exercise

We need to design a new test program such that the structure and optical assemblies are stressed at least as severely during TA testing as they will be stressed later—not only during the mission but also during the SC tests—while also protecting against the risk of costly failure from over-test.

Assignment:

As a team, brainstorm ideas for a redesigned test program.

Address levels of assembly (up to SC level), types of tests, and test levels relative to acceptance (max expected) levels.

- Constraints:
- All the flight-hardware designs are complete, and manufacturing is underway.
 - The optics and the telescope structure have low structural margins of safety and thus cannot tolerate significant over-test.
 - No time or money available to build qualification hardware.
 - See special challenges listed on page 8-3.

Intentionally left blank

Actual Revised Test Program for This Case History

- Work with the LVC to strip unnecessary conservatism from the SC-level sine vibe environment and understand the acceptable notching.
- Add a protoflight strength test (1.2 times limit) for the bus primary structure prior to bus and SC integration.
- Reduce SC sine vibe test to acceptance (MPE) levels instead of protoflight.
 - Now a test for workmanship given that all subassemblies will be tested earlier
 - SC acoustic test still planned for 3 dB above MPE
- Replace TA random vibration test with a sine vibe test to 120 Hz in all three axes and a sine burst test in the telescope's longitudinal axis.
 - Protoflight levels (1.25 x MPE)
 - Lateral sine burst not necessary; primary structure to be tested with sine vibe
- TA sine vibe environment designed (using apples-to-apples finite element models (FEMs)) to cause response accelerations of key components to be at least as high as the maximum of ...
 - 1.25 times predicted levels for acceptance SC sine vibe test
 - 1.0 times predicted levels for protoflight SC acoustic test
 - 1.25 times limit loads for launch: results of coupled loads analysis (CLA) combined with results of vibroacoustic analysis

The TA Modal Survey Plan Also Was Revised

Original plan:

- Objectives: Test correlate the FEM for use in ...
 - Coupled loads analysis
 - Planning and notching the SC sine vibe test
- No defined frequency ranges or criteria for FEM correlation
- No identified objective of correlating the FEM for use in jitter analysis
- TA vibration test to be done immediately following the modal survey
 - No scheduled time for model correlation prior to TA vibration test

Revised plan:

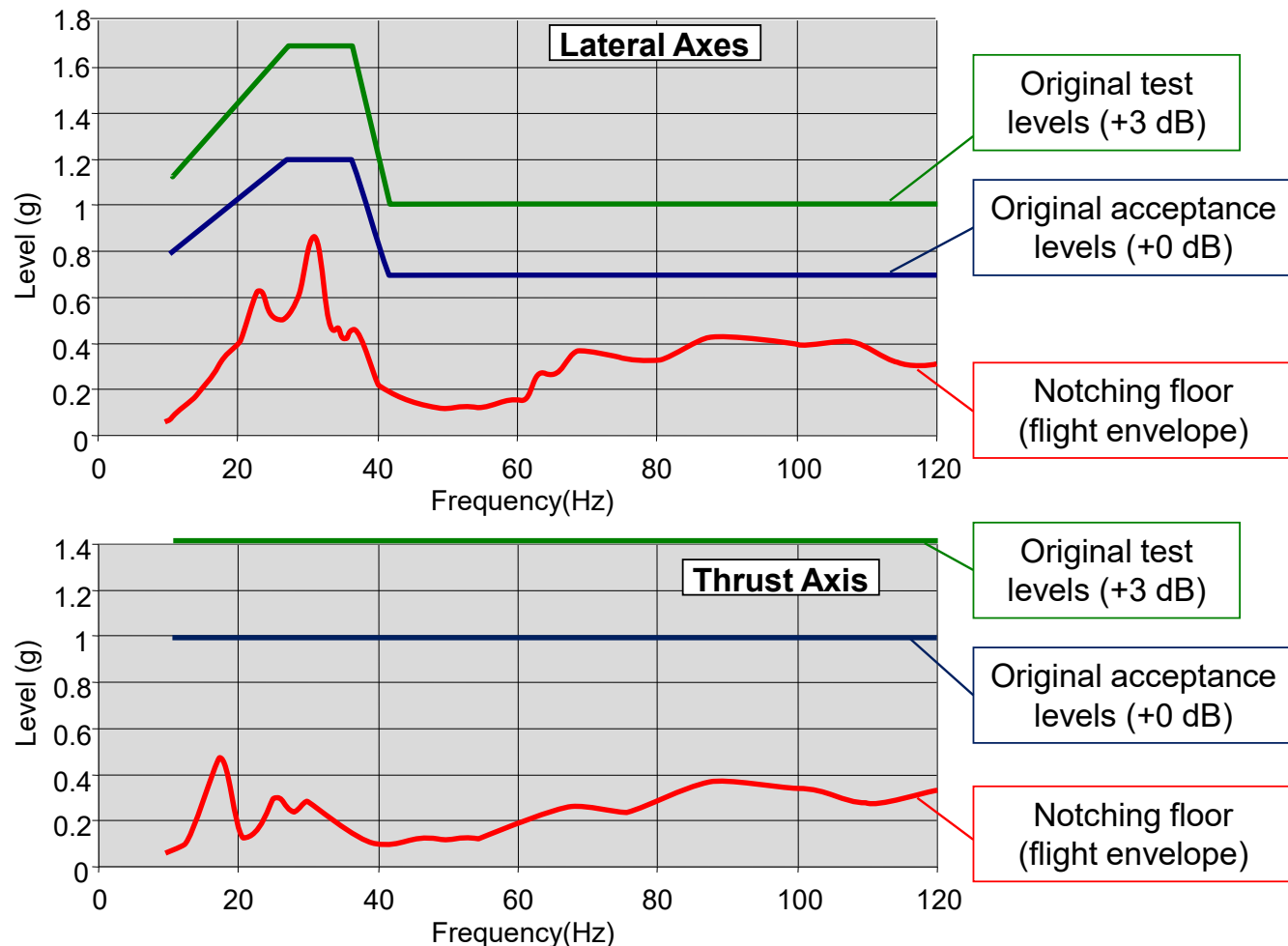
- Objectives broadened to include higher-frequency vibration modes, when practical, to enable partial correlation in support of on-orbit jitter analysis
- Reached agreement on model correlation goals
- Scheduled time to allow at least partial model correlation prior to the TA sine vibe test in case certain analyses need to be repeated

Stripping Conservatism from the SC Sine Environment

- We requested and held several meetings with LVC loads and dynamics engineers.
 - Explained the problems we faced
 - Obtained a better understanding of their flight data and concerns
- We got LVC's concurrence that, given all the other planned tests, SC sine vibe testing could be done to acceptance (MPE) rather than protoflight levels.
- We discovered that the sine vibe responses up to 60 Hz were enveloped by coupled loads analysis.
 - So we could notch as needed up to 60 Hz in the SC sine vibe test to ensure CLA results are not exceeded.
- We requested and received a less conservative sine vibe environment that could be used as a "floor" for notching above 60 Hz.
 - Envelope of flight data to date

Comparison of Original SC Sine Vibe Test Levels with Notching Floors*

Plan:
Test SC to acceptance levels, with notching as needed to keep the TA's critical components from exceeding their "capability" loads (see next page), but not penetrating the notching floors above 60 Hz.



*Not the actual data

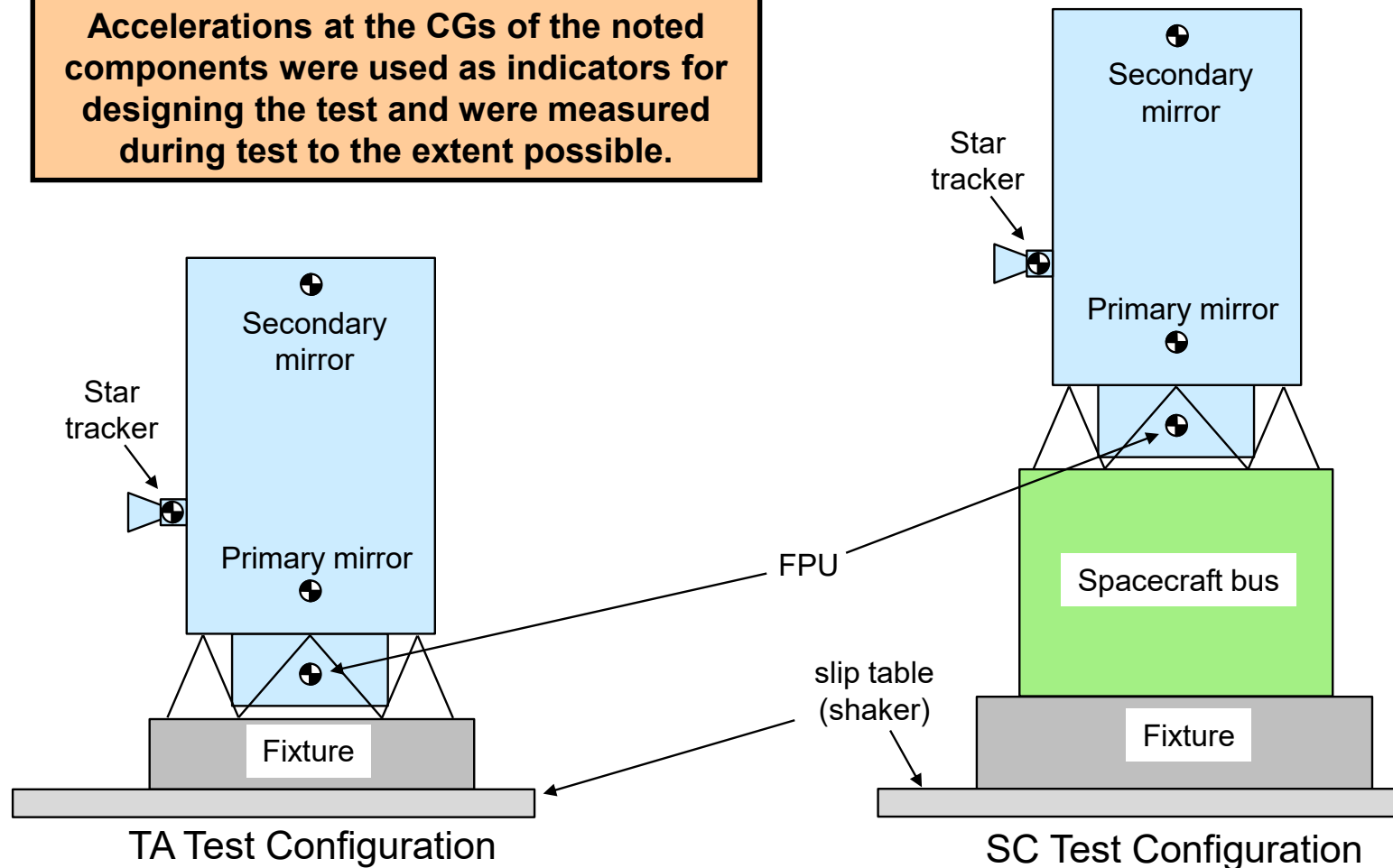
Process Used to Design the TA's Sine Vibe Test

Step	Description	Discussion
1	Identify TA component "capability" loads. Components include primary mirror, secondary mirror, star tracker, other items.	"Capability" is defined as the greater of the original design limit acceleration and the highest level that the TAC's analysis shows would not present unacceptable risk of damage.
2	Define "target" response loads for TA components (values lower than the "capability" loads from Step 1).	Targets equal the maximum predictions for 1.25 x acceptance SC sine vibe testing, 1.0 times 3-sigma loads predicted for protoflight SC acoustic testing, and 1.25 x flight limit (max expected).
3	Define a preliminary input sine vibe environment for the TA.	This environment is 1.25 times the time history of average acceleration at the TA mounting interface, as determined from SC sine vibe analysis.
4	Predict component responses to the preliminary sine vibe environment.	This analysis is to be done with a FEM representing the TA test configuration, including the fixture.
5	Modify the sine vibe environment as needed to analytically hit target accelerations without exceeding capabilities.	This step uses the same FEM as for step 4.
6	From results of the TA modal survey test, decide whether the TA FEM is accurate enough to be the basis for the sine vibe test environment.	If not, correlate the FEM with test results and repeat steps 2 through 6 before commencing with the TA sine vibe test.

Use the same FEMs and damping assumptions throughout (apples to apples).

Test Configurations and Critical TA Components

Accelerations at the CGs of the noted components were used as indicators for designing the test and were measured during test to the extent possible.



Problem: Knowledge of Accelerations at Component CGs

- During testing, acceleration could be measured at accelerometer locations, not at component CGs.
- Deriving accelerations at component CGs would require use of model-generated transformation factors.
 - Both for deriving the TA sine environment and during TA and SC testing
 - Dependent on model accuracy
- For TA testing, accelerometers could be directly on the components (e.g., secundar mirror), close to component CGs, so we weren't too concerned about accuracy of derived CG levels.
- But, for SC testing, accelerometers could not be mounted directly on the mirrors.
 - Proximity only
 - Less confidence in derived accelerations at component CGs

Model uncertainty was the main reason we decided to design the TA test to achieve 1.25 times the component loads predicted for SC sine vibe testing.

Another Problem Encountered During Test Planning

We discovered the vibroacoustic and sine vibe analyses used different damping assumptions.

- TA assumed to have lighter damping ($Q = 100$) than the bus ($Q = 20$)
- TA sine vibe analysis used $Q = 100$
- SC sine vibe analysis used appropriate coupled-system damping
- Coupled loads analysis was based on 2% damping ($Q = 25$) throughout, but all of the TA's component modes were above the CLA truncation frequency.
- Here's the problem: SC vibroacoustic analysis used $Q = 20$ throughout.

Recall:

We used these analyses to derive target loads and the TA sine vibe test levels.

We needed to use the same models with consistent damping assumptions.

We had to repeat vibroacoustic response analysis with coupled-system damping.

Issues Arising During TA Sine Vibe Test Performance

Damping was nonlinear (often the case) and higher than predicted (also common).

- Q of between 40 and 50 when at full levels rather than 100 (assumed)
- Overall good news: components saw less response acceleration during test; same should be true for SC tests and launch.
- But the effects of higher damping may not be the same for all the different types of analyses we had used to derive the TA test levels.

Nonlinearity in higher-order modes (>80 Hz)

- Including fundamental modes for some of the smaller components we were tracking when developing the TA test levels

These observations caused us to rethink the plan of depending so heavily on the FEM ...

How We Dealt with the Nonlinear Structural Behavior

The damping data and nonlinearities invalidated our approach!

- We had depended heavily on the assumption that the linear FEMs with assumed damping would be accurate, based on contractor assurance.
- The 1.25 margin between TA testing and SC testing was intended to account for model uncertainty, but it clearly wasn't enough.

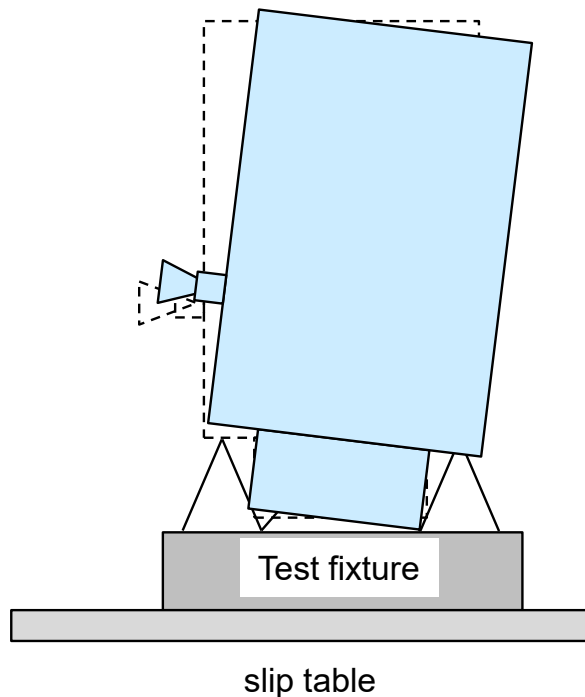
Fortunately, the higher damping discovered at full test levels provided additional inherent margin in the process.

- Lower response loads predicted for SC acoustic testing and sine vibe testing
- Could still notch down to the notching floor in the SC sine vibe test, if needed

In the TA test, we decided to identify target response loads for the key components and then tune the test environment until those loads were achieved, based on extrapolation from the -6 dB run. (There was no -3 dB run.)

- Could not do so with much accuracy
- But we all felt comfortable that those loads would not be exceeded during SC testing or the mission.

Test Anomaly During Full-level Lateral Sine Vibe Test: Nonlinear Rocking Frequency



Natural frequency
for rocking mode

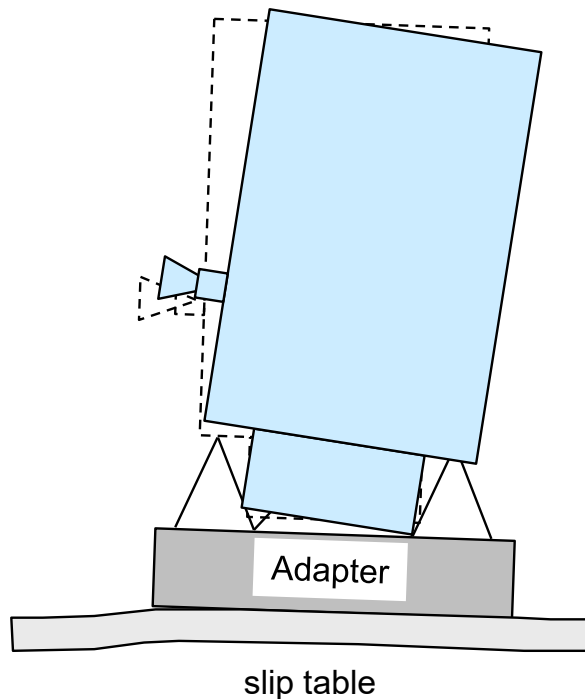
Low-level sweep	50 Hz
High-level sweep	39 Hz
Low-level sweep (repeated)	50 Hz

Investigation found the nonlinearity to be in the test fixture or in the slip table, not in the TA.

But the fixture and table were just chunks of metal.

Any ideas for what caused the nonlinearity in the fixtures?

Root Cause of Nonlinear Rocking Frequency



Our conclusion:

The slip table was too thin (2") for the test article's mass properties.

At low test levels, one-*g* loading kept the table supported over its length.

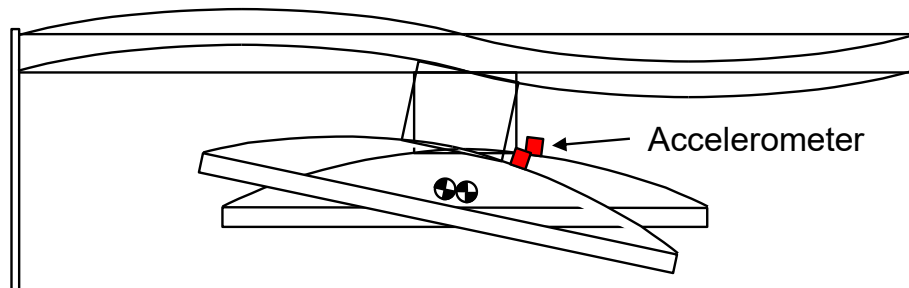
But at high levels, gravitational force was overcome, and either the table bent or there was other nonlinear deformation (bearings?) under the applied moment.

Unfortunately, we discovered this anomaly only after the contractor broke test configuration.

So we could not fully confirm our conclusion.

Human Error Invited by Schedule Pressure

- Load parameters being tracked for the sine vibe test were component CG accelerations.
- But, as stated earlier, accelerometers could not be put at the CGs.
- So the FEM was used to generate frequency-dependent transformation factors, γ , such that $A_{cg} = \gamma A_{accel}$
- For the lateral mode of the secondary mirror, the transformation factor was 4.5.
 - This means that, when aiming to achieve a CG acceleration of 18 g, the target acceleration measured in test was 4 g.
- As a consultant and an independent reviewer, I questioned this factor.
 - The responsible engineer checked and found a coordinate transformation error.
 - The correct transformation factor was 1.6.
 - The test level at this frequency was raised accordingly.



Schedule pressure and workload had prevented the responsible engineer from adequately documenting and checking his work.

Problems in the Sine Burst Test

As noted previously, sine burst testing was done only in the TA's longitudinal axis because lateral sine vibrate testing achieved the required overall base shear force and moment.

As is often the case, we had problems with the sine burst test!

- 18 Hz burst, with full level = 8.2 g
- -12 dB first (2.05 g), followed by -6 dB (4.1 g), followed by full level (0 dB) if lower-level runs look good
- All looked good for the -12 and -6 dB runs
- At 0 dB ...
 - 18 Hz input had additional superimposed higher frequencies (harmonics)
 - Several optical components in the TA saw amplified response (example: 13.4 g for the secondary mirror)

We were lucky the excessive loading didn't damage the hardware!

Lessons Learned

- Make sure the shaker and test fixtures are appropriate for the test.
 - The slip table was too thin in this case.
- Don't rely too heavily on FEA.
 - It's unreasonable to expect a FEM to be accurate to 100 Hz or higher when there are many modes of vibration below 100 Hz.
 - To some extent, nonlinearities are unavoidable and unpredictable.
- Allow adequate time in the test schedule and provide extra sets of eyes.
 - To allow adequate documentation and review
 - To ensure anomalies are sufficiently investigated before tear down
 - To avoid human errors
- Avoid sine burst testing for expensive, delicate items such as a large space telescope.
 - Environment is too hard to control, leading to high risk of over testing.
- Reduce the sweep rate in the TA sine vibe test from 4 oct/min to 2 oct/min.
 - As a result of the high sweep rate combined with low damping, there was little consistency in response peaks from one low-level run to the next, making it difficult to assess the health of the hardware.

Case History Summary and Conclusions

- The structural test program was far from perfect.
- But we feel we did a good job of identifying and addressing issues and balancing risks.
- One area for improvement: load parameters should be tracked and used to design, perform, and evaluate the tests.
 - Instead of using component accelerations,
 - develop equations for peak stress at critical locations in terms of all three orthogonal accelerations at accelerometer locations,
 - and then calculate stresses time consistently for use as evaluation parameters.

Overall results:

**The cost of the test program was reasonable,
and the space mission was successful.**

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):

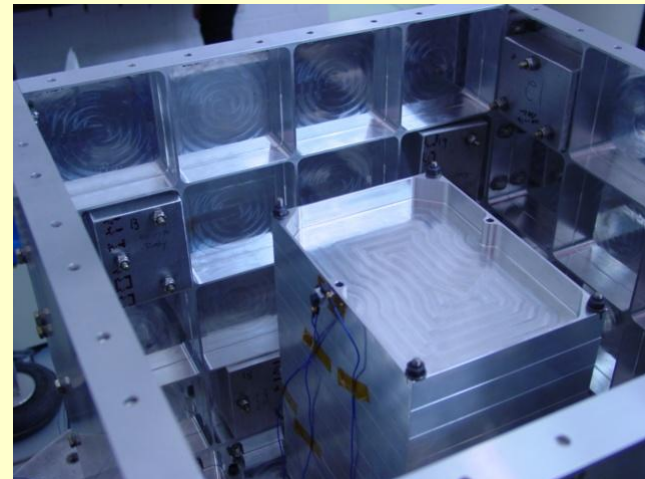
The subject test was performed April 14 and 15, 2003, at Kirtland Air Force Base. The test met all its objectives (see below) and the EM suffered no damage. The random vibration environment used in the test was considerably more severe than we expect to use for qualification testing next year. The maximum 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

- Design the overall structural and environmental test program with a systems engineering approach.
 - Rather than simply doing the tests that are specified in a government standard, user guide, or Interface Control Document
- For a given test, start with a clear definition of objectives and applicable criteria.
- 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.
- Try to find budget and time for a strength test of the spacecraft structure prior to vehicle integration.

continued

Key Points from This Course, continued

- Design the test to meet the objectives and enable assessment of the criteria:
 - Establish appropriate boundary conditions.
 - Make the applied load cases or loading conditions representative of the way in which the structure will be loaded during the mission.
 - Make sure analysis shows, with an apples-to-apples comparison, that the test loads are at least as severe as the maximum predicted flight loads multiplied by the test factor.
 - Design effective instrumentation to satisfy the objectives and criteria.
- Design the flight structure to pass the planned structural tests.

continued

Key Points from This Course, continued

- For vibration testing ...
 - Strengthen your understanding of the mechanics of vibration and the limitations of testing on a shaker.
 - Design fixtures and mass simulators that allow the test to meet its objectives.
 - Dry run the test before mounting the spacecraft to the shaker unless the shaker and control software have been used recently for similar tests.
 - 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.

continued

Key Points from This Course, continued

- For vibration testing (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 by force limiting, response limiting, or manual notching—without defensible technical rationale.
 - 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.
- Come to the test prepared to interpret data, and make sure the data are valid before dismantling the test setup.

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

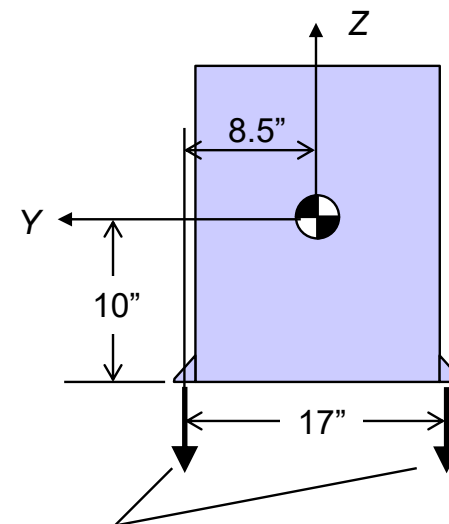
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Solution to Problem 2-1: Designing a Sine Burst Test

Use the process defined in Option 4 for Example 2-b:

- 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 2-1, 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,

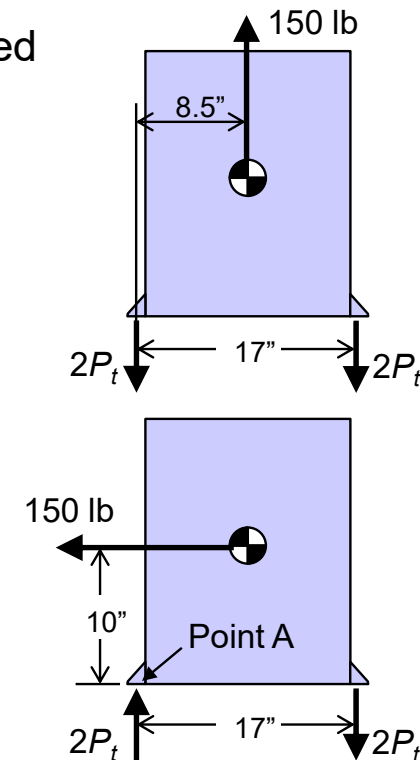
$$\sum M_A = 0 = 10(150) - 17(2P_t)$$

$$\text{So the peak bolt load is } P_t = \frac{10(150)}{2(17)} = 44.12 \text{ lb}$$

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 2-1, 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 2-1, 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 2-1, 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

$$a = \frac{938 + 37.50}{44.12} = 22.1 \text{ } g$$

In Sec. 5 we'll look at completing the process of designing a sine burst test by selecting the burst frequency and accounting for dynamic amplification of the test article.

Solution to Problem 5-1, Deriving an Equivalent Sine Burst Level

First, let's estimate a reduced input level by use of the transmissibility equation (Eq. 5-1):

$$T_R = \sqrt{\frac{1 + \left(\frac{2\zeta(25)}{100}\right)^2}{\left[1 - \left(\frac{25}{100}\right)^2\right]^2 + \left(\frac{2\zeta(25)}{100}\right)^2}} = 1.067$$

Sine burst frequency, f
Fundamental frequency, f_n

For damping < 10% of critical

Estimated equivalent reduced input acceleration to achieve the same base force and moment as 21.3 g uniform acceleration:

$$a = \frac{21.3}{1.067} = 20.0 g$$

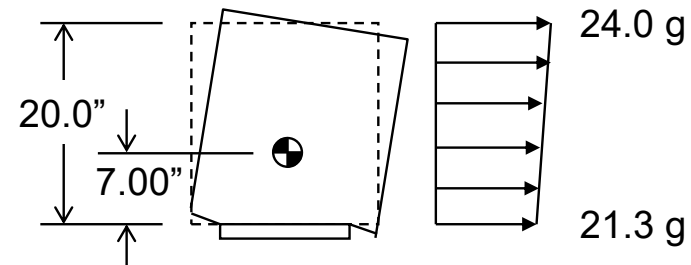
But a rocking mode such as this not the same as a mass on a spring, so let's derive an equivalent acceleration using the actual measured response ...

(continued)

Solution to Problem 5-1

For the acceleration profile shown at right and with the method used in Example 5-2, Option 4,

Average acceleration, as calculated at CG:



$$a_{cg} = 21.3 + \left(\frac{7.00}{20.0} \right) (24.0 - 21.3) = 22.2 \text{ g}$$

Angular acceleration:

$$R = \frac{(24.0 - 21.3)(386 \text{ in/s}^2)}{20.0 \text{ in}} = 52.1 \text{ rad/s}^2$$

$g = 386 \text{ in/s}^2$

(continued)

Solution to Problem 5-1 (continued)

Moment at base, $M = wa_{cg}x_{cg} + RJ$ ————— (from Example 5-2)

$$= 150(22.2)(7.00) + 52.1(30.0) = 24,900 \text{ in-lb}$$

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

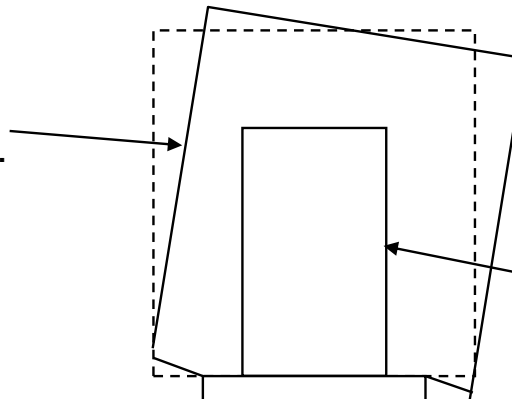
$$a = \left(\frac{22,400}{24,900} \right) (21.3) = 19.1 \text{ g}$$

(continued)

Solution to Problem 5-1, continued

The problem statement said to assume all of the mass rotates as a rigid body. However, this is usually not the case.

Here, the exterior structure is rotating ...



... but the interior structure is not.

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!

STDI Index

Last revised February 2026

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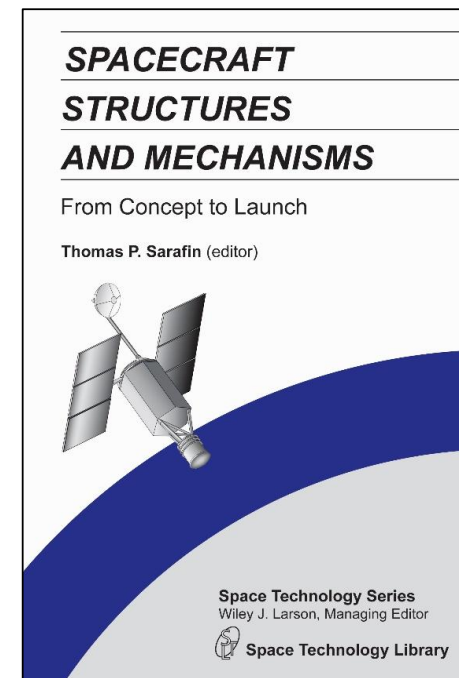
Contact: tom.sarafin@instarengineering.com

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

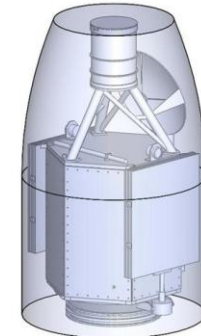

Space Mission Structures: From Concept to Launch

Understanding a Shock Response Spectrum (SRS)

Say we measure a time history of



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.

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5-14

“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

Instar 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.

Unwrapped exploded side view of panels and stringers

Start at stringer 3 (top): For equilibrium, ...

$$P_3 = 2V_3 ; V_3 = \frac{7.50}{2} = 3.75 \text{ kN}$$

Now move down:

$V_1 = V_2$

$V = 30 \text{ kN}$

$M = 4500 \text{ kN-cm}$

continued

Instar Structural Design and Analysis for Aerospace Engineers

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

Or mated to a structure stiff enough to distribute loads

Monocoque cylinder mated to a monocoque conical (frustum) shell

Include a bulkhead or a deep ring frame to react the radial load from cylinder

Load from bulkhead or ring frame

Load from frustum

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Instar 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:

Displacement plots

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

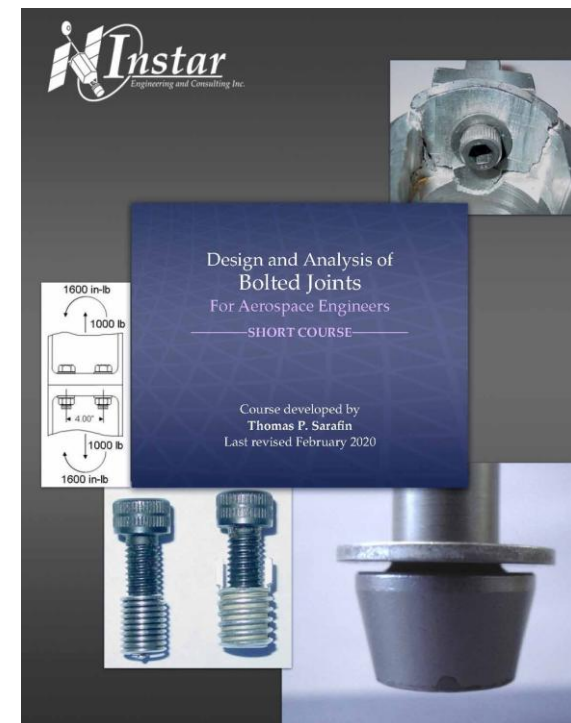
Don't rely solely on FEA!

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DABJ—Design and Analysis of Bolted Joints

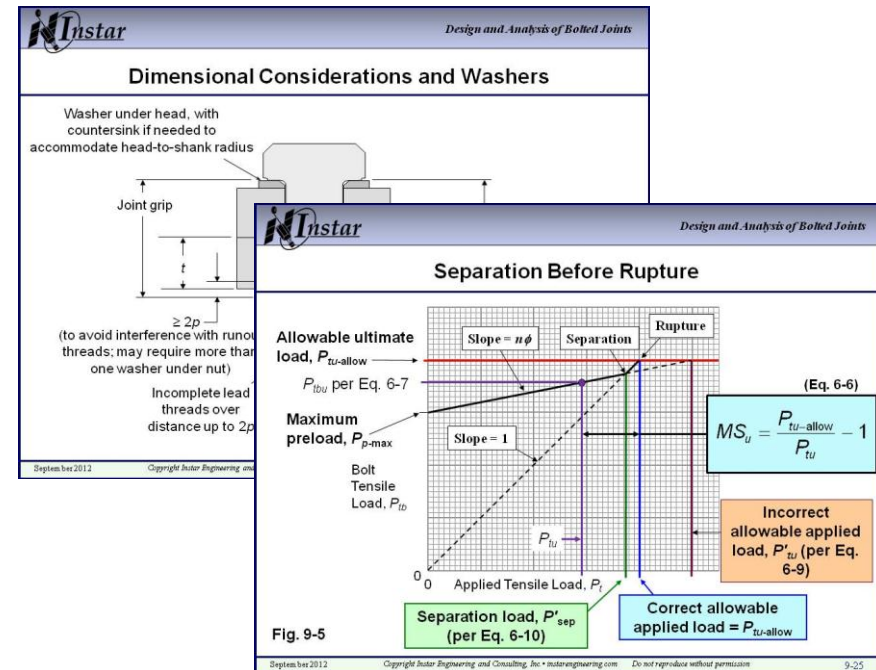
***Includes a close look at NASA-STD-5020 (including Revs. A and B),
“Requirements for Threaded Fastening Systems in Spaceflight Hardware”***

- 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."

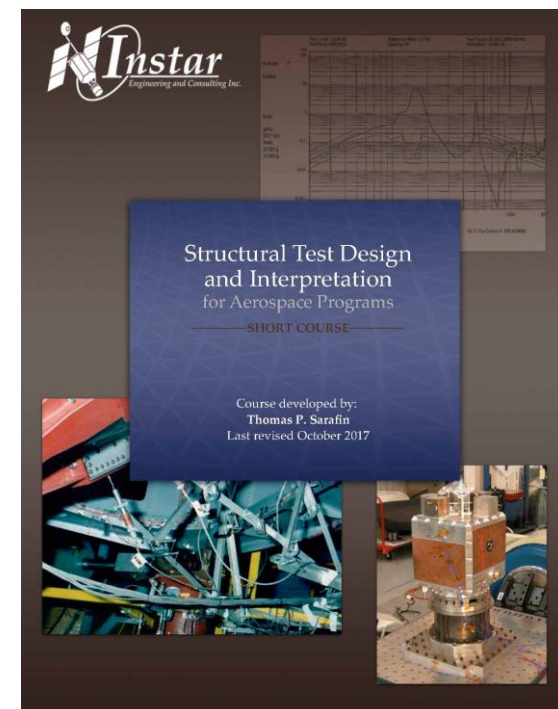
"Really good course. More people need to take this."

"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

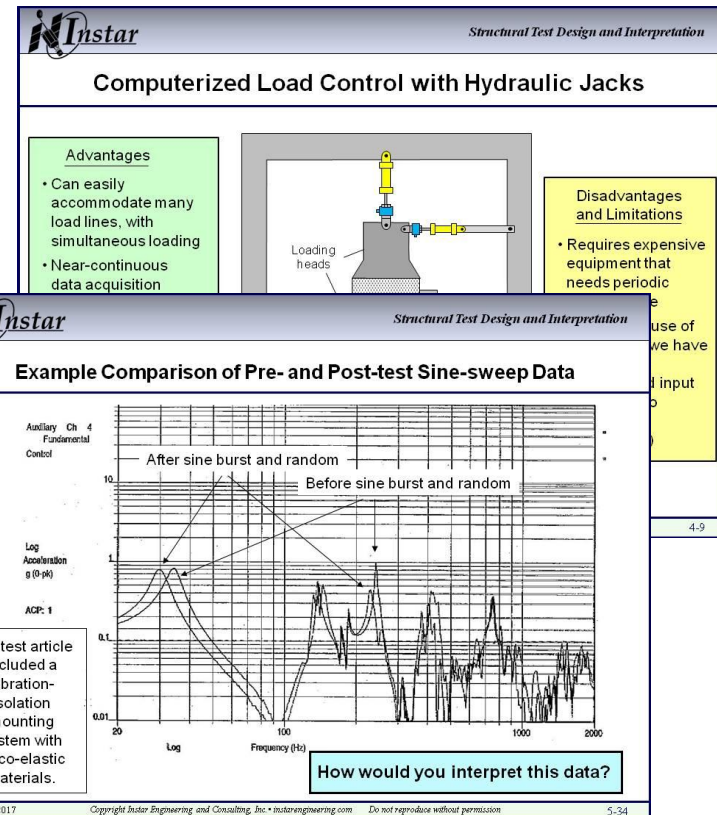
1. Overview of structural testing
2. Designing and documenting a test
3. Loads testing of small specimens
4. Static loads testing of large assemblies
5. Testing on an electrodynamic shaker
6. Notching and force limiting
7. Overview of other types of structural tests
8. Case history: vibration testing of a large spacecraft telescope

"Good job, Tom. These courses are a big help and give us a lot of great fundamental information."

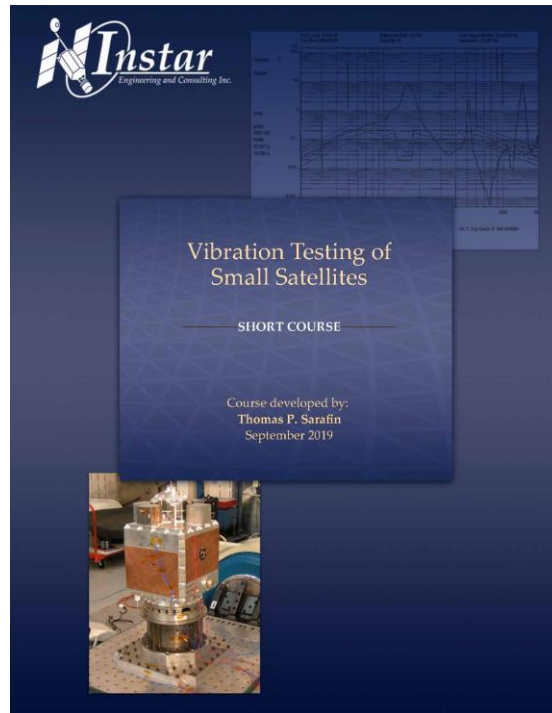
"Tom Sarafin's courses never disappoint. This class offers a well-balanced blend of fundamentals, examples, and lessons learned that any aerospace engineer involved in structural test design and interpretation would benefit from."

"This is a great course."

Can be combined with a one-day computer workshop on notching and force limiting (NFLW)



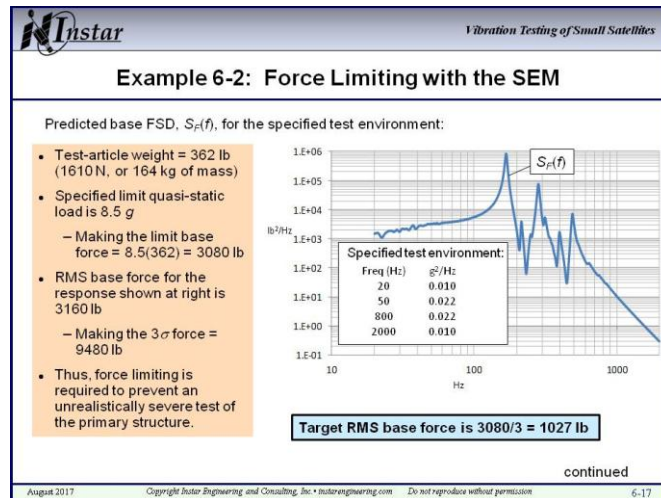
VTSS—Vibration Testing of Small Satellites and NFLW—Notching and Force Limiting Workshop



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."