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# DESIGNING EFFECTIVE STATIC TESTS FOR SPACECRAFT STRUCTURES

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## Abstract

This paper describes and demonstrates a process for designing effective static tests for spacecraft structures. The emphasis is on devising load cases that effectively simulate the extreme dynamic loads experienced during launch and screen the structure for potential failures. The process entails devising preliminary load cases that deform the structure in its key excited mode shapes and then adjusting the cases to achieve target stresses and margins of safety. Although the discussion centers on bus (body) structures, the described techniques apply to smaller structural assemblies and launch vehicles as well.

## Introduction

Static testing is a common method of verifying that a spacecraft structure has enough strength to withstand launch and—in some cases—on-orbit loads. A *static test* is one in which loads are applied gradually to ensure no dynamic amplification. Loads can be applied with dead weights, hydraulic jacks (Fig. 1), or enforced displacements. We can also induce quasi-static inertia loads with a centrifuge.

A static test is consistent with how we normally do stress analysis, but launch loading is dynamic, dependent on how the spacecraft vibrates in response to time-varying forces. The challenge in static testing is thus to effectively and efficiently simulate the maximum expected dynamic inertia loading of the spacecraft. Our goal is to screen out any defects in design or workmanship with the simplest set(s) of statically applied loads. We want to ensure an effective test without unnecessary overtesting, balancing the risk of mission failure with that of test failure.

## When is a Static Test Required?

The answer to that question is easy: never. A static test should not be considered a firm requirement. We do static tests to verify a requirement (strength)—that is, we use them to gain confidence. In some cases, we can become confident enough simply through analysis. Factors of safety of 2.0 or higher for ultimate are used by many space programs to avoid a static test when they can accept the weight penalty. (A 1.25 ultimate factor of safety is common for an unmanned mission when a dedicated, nonflight structure will be tested to that level. Table 1 shows typical test options.)

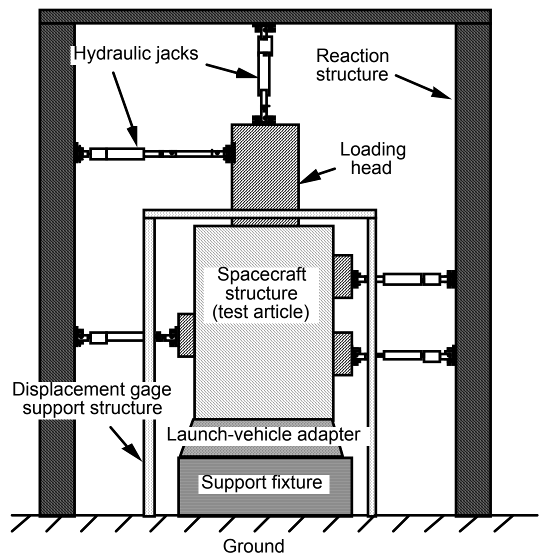



**Fig. 1. Typical Configuration for a Static Test with Hydraulic Jacks.**

The higher factor of safety when no test is planned helps cover human errors. This "no test" option is viable if the design is relatively simple, with predictable load paths and failure modes, and if strength is not very sensitive to workmanship or process variables. Other-wise, when the structure's capability is less predictable, a test is essential.

The right type of static test depends on the characteristic that is uncertain. Structures made of common, high-grade metals, such as many aluminum alloys, with members attached by high-quality fasteners, are candidates for options 1 and 3 from Table 1. Here, only the quality of the design is in doubt, so we test one structure and thus "qualify" the design. We then count on each build of that design to have a strength that is nearly the same as that for the tested structure, varying by no more than is covered by the test factor. The strength of advanced composite materials, brittle materials, and bonded joints is far more sensitive to process variables, so we normally proof test each flight article made of such materials (option 2).

**TABLE 1. Design Factors of Safety for Different Test Options.**[1] These factors apply when human safety is not at risk, such as for an unmanned mission.

| Option | Yield | Ultimate |
|---|---|---|
| 1. Ultimate test—test a dedicated unit to ultimate loads | 1.1 | 1.25 |
| 2. Proof test all flight units to 1.1 times limit | 1.1 | 1.25 |
| 3. Protoflight—test one flight unit of a fleet to 1.25 times limit | 1.25 | 1.4 |
| 4. No structural test | 1.6 | 2.0 |

Several tests other than static tests also verify strength; the best test depends on the type of structure. Acoustic tests are most effective for many solar panels, antenna dishes, and other lightweight, large surface-area structures that respond to sound pressure. Random vibration tests are best for electronic boxes, brackets, sensor stalks, and other relatively small structures whose fundamental frequencies are above 50 Hz or so. And sine vibration testing can verify the strength of structures not adequately loaded by acoustics or random vibration. A static test is called for when no other planned analysis or test will provide enough confidence in the structure's strength, or when static-test methods will provide such confidence most cost effectively.

When devising a test program, we should focus on the best way to demonstrate the structure will be able to withstand mission environments. For example, we may initially perceive the need to test the spacecraft body structure for acoustics, random vibration, sine vibration, and static loads. But, if we focus on the true objectives—confidence in the structure's strength and fatigue life—we might be able to do one or two well-designed tests instead of four. For strength, we would start by predicting the limit loads, potential failure modes, and margins of safety for the mission life-cycle events. We would then design a test that analytically will be equally or slightly more severe.

## Devising Effective Load Cases

Once we have decided to do a static test, our goal is to load the structure in a manner that will screen out any failures that may occur during the mission. Because the launch loading environment is so complex, with many different types of input loads occurring simultaneously, devising effective load cases is not straightforward.

Most space programs define criteria for statistically combining transient, quasi-static, and acoustic loads for design. We also typically account for uncertainty in the finite-element model used for loads analysis by multiplying the computed loads by a model uncertainty factor (MUF) to arrive at design limit loads. These loads become the basis for the static test, subject to the appropriate test factor.

Let's look at an example of a poor static test. Figure 2 shows a panel used to mount an electronics box. The objective is to qualify the panel for three directions of inertia loads acting on the box, as shown in the figure. These load factors were derived by adding the extreme values computed in a coupled loads analysis, which normally covers the effects of transient and steady-state loading, and an acoustic-response analysis.

The simplest approach to static testing is to define three load cases, with each case applying the maximum and then minimum load in a single direction to the loading head that simulates the box. But it is hard to say whether this an effective test:

- For nearly any potential failure mode in the panel, or at the mounting interface, the stress is a function of all three directions of box loading. During launch, the box will experience inertia loads—at some levels—in all three directions simultaneously. Loading one direction at a time will result in an undertest in this regard.
- Adding the extreme responses calculated for transients and acoustics is overly conservative. Acoustic response is random, so

the "extreme" response calculated is typically statistically appropriate only for assessing the acoustic environment separately (e.g., for an acoustic test). There is little chance such a high load would occur precisely when the transient load peaks.[1] Also, the acoustic environment for most launch vehicles does not peak when transients are highest.

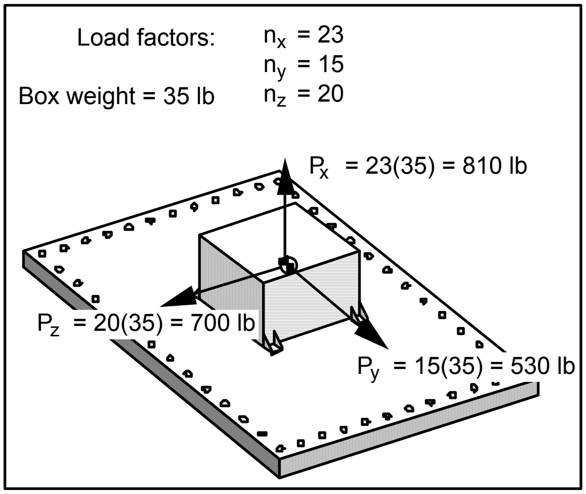



**Fig. 2. Example Structure that will be Statically Tested.** The objective is to verify the panel's strength for the design limit loads shown.

- Much of the peak box acceleration calculated for acoustics, which typically includes vibration up to 2000 Hz, may not load the mounting panel nearly as severely as uniform acceleration. Higher order box modes of vibration, in which parts of the box are accelerating in different directions at the same time, may be contributing to the total response. In this regard, static testing is overly conservative.
- During launch, the panel may be stressed by other loads than those introduced by the box. Distortions of the boundary structure may cause significant stresses. Or the panel itself may respond significantly to acoustic pressure. Thus, again, the planned static test may not be adequate.

Clearly, we cannot design an effective static test based solely on box accelerations. Instead, we can address each of the above problems with an approach that is nearly as simple: First, we identify the potential failure modes of concern in the test article. Second, we directly calculate the parameters that relate to those failure modes. (Calculate stresses and bolt loads in the loads analysis in addition to accelerations.) Finally, we define static load cases that we predict will cause the critical stresses and bolt loads to be at or slightly higher than those for design, subject to the appropriate test factor.

In the above example, the failure modes of concern might be bolt tension/shear interaction, insert pull-out, and panel bending. In this case, we would calculate design limit bolt loads and panel bending stresses, multiply them by the test factor (say, 1.25), and then devise one or more load cases that are predicted to cause these target parameters. We do this by generating a load-transformation matrix (LTM) that relates the selected parameters to the applied loads. Figure 3 shows how to generate an LTM for computing bolt loads in terms of three forces applied to the box loading head, assuming the mounting interface is rigid. (In an actual problem such as this, to account for indeterminate load paths, we would use finite-element analysis to generate an LTM. For the purpose of this example, we will assume the LTM equations in Fig. 3 are accurate.)

To complete this example, let's assume our goal is to devise a static load case that will qualify the interface for peak bolt tension. By planning ahead, in the loads analysis used to establish design loads, we calculate tensile forces in the four interface bolts. We do this either by using another LTM or by constructing the model to enable direct force recovery from the finite-element analysis, such as with spring elements that represent the bolts. Assume we also calculate response accelerations at the four outboard corners of the box. The following limit loads result:

Peak accelerations at any corner:

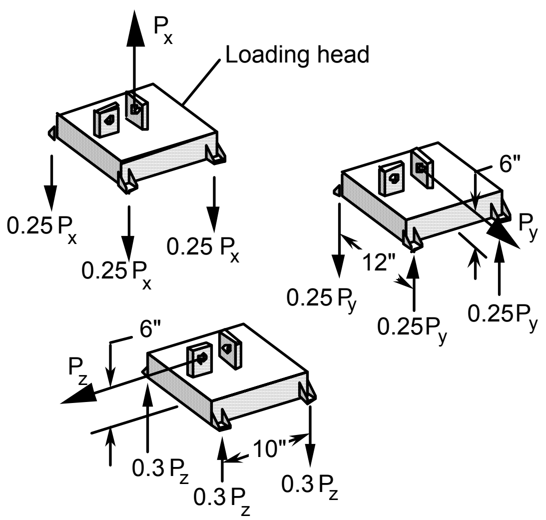
$$\ddot{x}_{\max} = 23\,\text{g}$$
$$\ddot{y}_{\max} = 15\,\text{g}$$
$$\ddot{z}_{\max} = 20\,\text{g}$$

Peak tension in any bolt:

$$p_{t\max} = 190 \text{ lb}$$

From the LTM in Fig. 3, we can generate a predicted 190 lb tension in any one fastener by any of three load cases, each with a single applied load: $P_x$ = 760 lb, $P_y$ = 760 lb, or $P_z$ = 633 lb. (We could also combine two or all three load components, but one makes for the simplest test.) Of course, this example is simple enough that we could have arrived at the same conclusion without generating an LTM. But as the structure gets more complex, and we need to combine applied loads to achieve the target load parameters, an LTM becomes much more useful.

**Bolt Tension Loads in Terms of Applied Loads**




**In matrix form,**

$$\begin{Bmatrix} p_{t1} \\ p_{t2} \\ p_{t3} \\ p_{t4} \end{Bmatrix} = \begin{bmatrix} 0.25 & 0.25 & 0.3 \\ 0.25 & 0.25 & -0.3 \\ 0.25 & -0.25 & 0.3 \\ 0.25 & -0.25 & -0.3 \end{bmatrix} \begin{Bmatrix} P_x \\ P_y \\ P_z \end{Bmatrix}$$

Tensile forces in four bolts

Load-transformation matrix

**Fig. 3. Simple Example of a Load-Transformation Matrix.** This matrix relates bolt loads to applied loads. We can also relate bolt loads (or stresses) to modal responses computed in loads analysis.

Note that in this example, the key to an effective test was to identify the failure mode of concern. If we had used an overly general interpretation of the test's objective, we may have concluded that we needed to load the box loading head to a force equal to the box's weight multiplied by the peak design acceleration. And, to be conservative*, we would apply the peak loads in all three directions simultaneously, as shown in Fig. 2: $P_x$ = 810 lb, $P_y$ = 530 lb, and $P_z$ = 700 lb. The resulting predicted peak bolt tension would be

$$p_{t\max} = 0.25(810) + 0.25(530) + 0.3(700) = 550 \text{ lb}$$

or almost three times as high as we need!
Even the full design X load, 23 g, or 810 lb, would cause too much bolt tension. Although this example is contrived, such an apparent conflict is common. What it tells us is that, although one of the box corners sees 23 g, the entire mass of the box does not.

Note also that it is not important whether the actual member loads and stresses developed during the test envelop the design loads and stresses. Rather, the predicted loads and stresses must do so, based on the same finite-element model—and, if applicable, the same transformation equations—used to calculate the design loads. This is an "apples to apples" comparison. Of course, we need to modify the model to match the test configuration (boundary conditions, loading heads, one-g effects), but such changes should be small.

The above example is simplistic, of course, but it demonstrates the basic approach to generating effective static test cases. Table 2 gives an overview of how to extend this approach for a full spacecraft bus structure. The following text describes and demonstrates the steps in this process.

### 1. Generating an Effective LTM for Loads Analysis

Figure 3 showed an example of an effective LTM: one that provides the information we really want. Once we have the peak tensile force in any interface bolt, we can assess without

* Structural engineers are taught almost exclusively to be conservative—always. Unfortunately, this mindset often leads to weight-critical space structures being designed to be several times as strong—and heavy—as they need to be. Even worse, in some cases, test loads are often far too severe, many times leading to unnecessary test failures. Failure of a flight structure during testing at a high level of assembly can be the death of a competitive, commercial program. We must become more attuned to the risk and consequence of test failure when trying to be "conservative."

excessive conservatism several potential failure modes, including bolt tension, end-pad bending, and insert pull-out.

**TABLE 2. How to Develop Load Cases for a Spacecraft's Static Test.** See text for discussion.

1. Generate an effective load-transformation matrix (LTM) for loads analysis
2. Calculate internal loads to serve as test targets
3. Identify key mode shapes and uniform inertia loads to simulate statically; dedicate a load case to each
4. For each load case, define applied loads that are consistent with the selected mode shape
5. Generate a test LTM that relates the target load parameters to unit applied loads at the selected loading-head locations
6. Assess the preliminary load cases from step 4 with the test LTM
7. Modify the load cases to achieve all targets while also staying below defined upper limits
8. Scrub the load cases to minimize their number and complexity

A full spacecraft body structure will have many potential failure modes. The first step in generating an effective LTM for loads analysis is to identify all potential failure modes of concern. Next, we use preliminary design loads, such as the inertia loads defined in the launch-vehicle user guide, to whittle down the list of failure modes to those having low margins of safety. We then write equations for those failure modes, relating the critical load or stress parameters to loads in the model's elements, and put them in the form of an LTM. In some cases, such as for column buckling, we would include a single member load as a row in the LTM. We then multiply this LTM by the matrix relating element loads to modal accelerations, interface displacements, and interface accelerations.[1] The product is an LTM relating critical stresses and bolt loads to the param-eters directly computed in loads analysis.

The main limitation of LTMs is that each equation must be linear, with the following form:

$$A = c_1 B_1 + c_2 B_2 + \ \dots \ + c_i B_i$$

where $A$ is the parameter (e.g., bolt load) of interest, $c_i$ are coefficients, and $B_i$ are member loads or other directly calculated response parameters. For any loads or stresses requiring nonlinear equations, we must find linear approximations. For example, the peak bending stress on a cylindrical cross section is the RSS of bending stresses from two orthogonal bending moments. To approximate this relationship linearly, we could write stress equations for locations every 45° around the cylinder wall, with a possible error of only about 8%.

As an example of how to develop an effective LTM, consider the spacecraft structure shown in Fig. 4. Say we want to devise a load case for this structure that will test the joint in detail *A* for tension. The two identified failure modes of concern (low margins of safety), lug failure and end-pad bending, are solely dependent on tension in member 1 and bolt *a*, respectively. Before we can design a load case to test this joint, we need to define the design loads for member 1 and bolt *a*. We do this with a coupled loads analysis, using an LTM to compute these load parameters. The LTM contains equations for the selected parameters in terms of member loads. We derive the equation for tension in bolt *a*, $p_{ta}$, as follows: (Note: to keep this example simple, we will ignore how potential misalignments affect bolt tension.)

Axial load acting on the bolt pattern:

$$P_{z'} = F_{x1} + 0.707 F_{x2}$$

where $F_{x1}$ and $F_{x2}$ are axial loads in members 1 and 2, respectively. Moment acting on the bolt pattern:

$$M_{x'} = -0.60(0.707)F_{x2} \ = -0.424 F_{x2}$$

Tensile force in bolt *a*:

$$p_{ta} = \frac{P_{z'}}{4} - \frac{M_{x'}}{2(1.60)}$$

$$= \frac{F_{x1} + 0.707 F_{x2}}{4} - \frac{-0.424 F_{x2}}{2(1.60)}$$

$$= 0.250 F_{x1} + 0.309 F_{x2}$$

To help us understand the nature of loading for this spacecraft, in the coupled loads analysis we will also compute total shear, $V_y$, and moment, $M_x$, at the interface between the spacecraft and the booster adapter (station 0). We will consider this problem two-dimensional to keep it simple:

$$V_y = 0.707 F_{x2}$$

$$M_x = -15(F_{x1} + 0.707 F_{x2}) + 15F_{x3}$$
$$= -15F_{x1} - 10.6F_{x2} + 15F_{x3}$$

where $F_{x3}$ is the axial load in member 3. Let's now write the above three equations, plus the one for axial load in member 1, in matrix form:

$$\begin{Bmatrix} F_{x1} \\ p_{ta} \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 1 & 0 & 0 \\ 0.25 & 0.309 & 0 \\ 0 & 0.707 & 0 \\ -15 & -10.6 & 15 \end{bmatrix} \begin{Bmatrix} F_{x1} \\ F_{x2} \\ F_{x3} \end{Bmatrix} \quad (1)$$

where the matrix is the LTM. We multiply this LTM by the one (generated with finite-element analysis) relating the three member loads to parameters computed in the loads analysis.

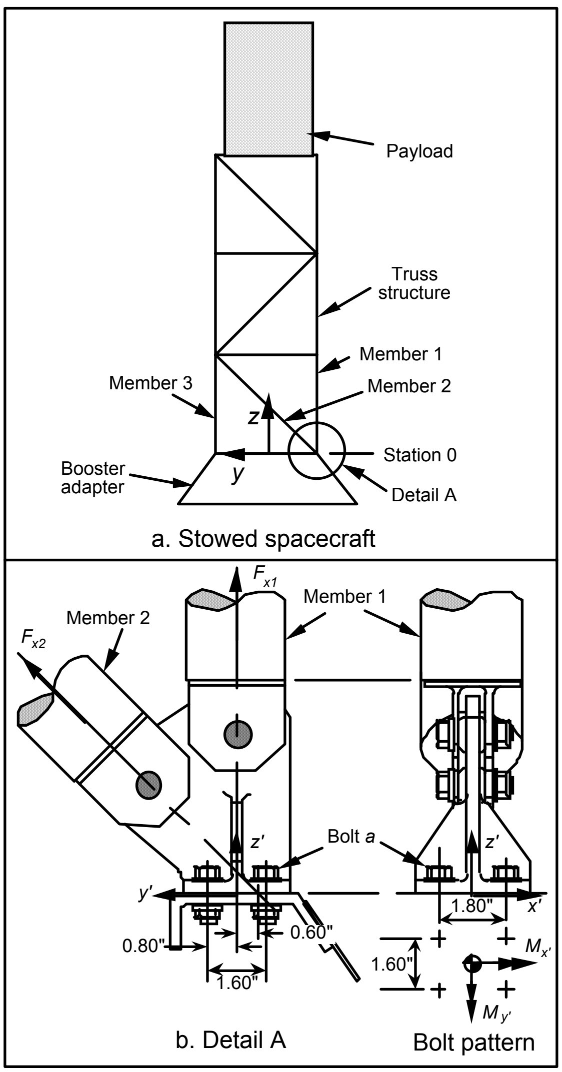



**Fig. 4. Hypothetical Spacecraft and Area of Concern.** The objective is to devise a static load case that will develop the design limit tensile forces in member 1 and in bolt *a*.

## 2. Calculating Target Loads

For a spacecraft body structure, this step entails coupled loads analysis for launch transients, using the LTM directly to get the needed information. These loads, with the appropriate test factor from Table 1, become test targets, or the values that must be achieved for the test to be fully effective.

The perfect static test is impossible for a complex structure; to reach all the targets, we must overshoot some of them. To balance the risk of an undertest (undetected deficiency) with that of an overtest (unnecessary test failure), we need to understand the structure's capability regarding each of the monitored failure modes. With defined capabilities, simple post-processing of the peak loads and stresses will provide margins of safety.

Returning to the example in Fig. 4, let's say the loads analysis, using the LTM generated in Step 1, above, results in the design ultimate loads in Table 3. We want to test member 1 and bolt *a* to these ultimate loads. The table also shows the hypothetical allowable ultimate loads and computed margins of safety.

**TABLE 3. Design Loads and Margins of Safety for the Example Problem.** These loads are peak values from the loads analysis and do not necessarily occur at the same point in time. Units: lb, in·lb.

| Parameter | Ultimate Load | Ultimate Allowable | Margin of Safety |
|---|---|---|---|
| $F_{x1}$ | 12,100 | 13,300 | +0.10 |
| $p_{ta}$ | 6230 | 6670 | +0.07 |
| $V_y$ | 8140 | – | – |
| $M_x$ | 585,000 | – | – |

## 3. Identifying Key Mode Shapes

An effective static test simulates the global loading conditions for the structure. For example, if, during launch, the spacecraft's fundamental torsion mode is highly excited, we want a test case that will twist the structure. If the second bending mode responds significantly to transient loads later in launch, we should have a test case that will deform the structure in approximately that mode shape. And we may need another load case that simulates the high quasi-static, uniform acceleration during, say, second-stage shutdown. This approach to devising test cases helps ensure unrecognized modes of failure will be detected.

An important point to remember is that we test structures to detect unrecognized potential deficiencies in the design or in workmanship. In the earlier example (Fig. 2), the lead test engineer could have decided simply to test a single fastener in tension to show it can carry the needed load. But this would be an ineffective test of the component's interface. For two reasons, it is much better to load the component in a manner predicted to cause the target bolt load. (1) We may not have recognized the critical failure mode associated with bolt tension, such as shear tear-out of the fitting. (2) The bolt pattern is statically indeterminate, so loads may not distribute as predicted—in the test and during launch. With more highly indeterminate structures, it is important to develop realistic global loading conditions so the loads distribute much like they will at launch.

Thus, before devising load cases, we should identify the key mode shapes for the structure and any critical quasi-static loads. We can do this by computing modal contributions for critical load parameters; i.e., percentages of peak load caused by different response modes. For transient analysis, response time histories will provide this information; for random vibration, we can estimate these contributions from the computed response power spectral density.

4. Defining Preliminary Load Cases

Identifying suitable locations for introducing concentrated loads is often challenging. Because we are trying to simulate inertia loads—whether uniform or associated with a given mode shape—we try to put loading heads (test fixtures) in locations of high mass. The goal is to minimize unrealistic local effects. Putting a loading head in place of an intentionally missing component, as we did for the example in Fig. 2, is a good approach. Mounting interfaces for ground-handling equipment, such as lift points, are also good candidates.

Figure 5 shows a preliminary load case that generally simulates the first bending mode of the spacecraft truss from Fig. 4. This appears to be a likely mode to simulate in order to test member 1 and bolt *a*. This example uses only four load points, for simplicity, and again considers only two dimensions. In a real load case such as this, we usually target several potential failure modes and may have to add axial loads and loads in the other lateral direction.

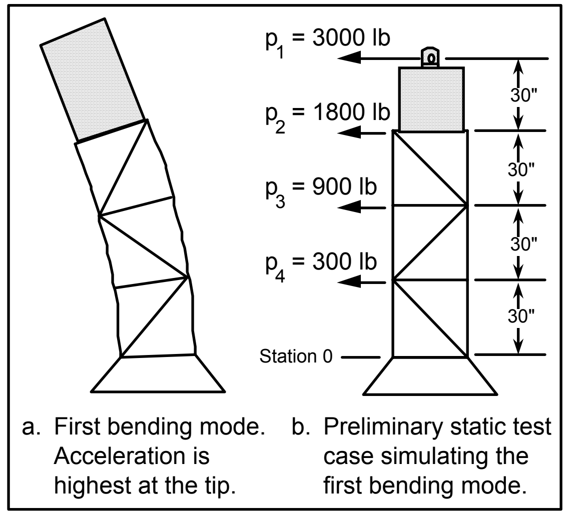



**Fig. 5. Simulating the Example Space-craft's First Bending Mode with a Preliminary Load Case.** Load magnitudes were chosen to be approximately proportional to the mode shape's displacements while also causing a bending moment at station 0 equal to the design ultimate load of 585,000 in·lb. For this example, mass is assumed to be uniformly distributed along the spacecraft's length.

Note that the load magnitudes in Fig. 5 were merely approximated. At this point in the process, precision is unnecessary. We simply want to establish initial load cases that are subject to later refinement.

5. Generating a Test LTM

The goal here is to develop a tool that will allow us to quickly assess and iterate the preliminary load cases. We start by modifying the finite-element model used for loads anal-ysis to make it match the test configuration. Because we will be judging the adequacy of the load cases by comparing test predictions with the computed design loads, the models used for these analyses must be nearly identical. The only differences should pertain to boundary conditions and loading heads.

To form the test LTM, we start with the initial LTM generated in step 1, which relates critical stresses and loads to element loads. We generate a second matrix that relates element loads to unit applied loads at the load points selected in step 4. (Do this by subjecting the test model to unit load cases and requesting element loads.) The product of these two matrices is an LTM that relates critical stresses and loads to unit applied static loads.

Returning to our example, which is a statically determinate planar truss, we can derive the following equations for the loads in members 1 through 3 in terms of the four applied loads:

$$\begin{Bmatrix} F_{x1} \\ F_{x2} \\ F_{x3} \end{Bmatrix} = \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0 \\ 1.41 & 1.41 & 1.41 & 1.41 \\ -4.00 & -3.00 & -2.00 & -1.00 \end{bmatrix} \begin{Bmatrix} p_1 \\ p_2 \\ p_3 \\ p_4 \end{Bmatrix}$$

By multiplying the LTM in this equation by the one in Eq. (1), we get equations for the selected critical parameters in terms of the applied loads:

$$\begin{Bmatrix} F_{x1} \\ p_{ta} \\ V_y \\ M_x \end{Bmatrix} = \begin{bmatrix} 3.00 & 2.00 & 1.00 & 0 \\ 1.19 & 0.936 & 0.686 & 0.436 \\ 1.00 & 1.00 & 1.00 & 1.00 \\ -120 & -90 & -60 & -30 \end{bmatrix} \begin{Bmatrix} p_1 \\ p_2 \\ p_3 \\ p_4 \end{Bmatrix} \tag{2}$$

where $F_{x1}$ is the load in member 1, $p_{ta}$ is the tension in bolt $a$, and $V_y$ and $M_x$ are the total shear and moment at station 0.

6. Assessing the Preliminary Load Cases

Here, we simply use the test LTM to calculate the selected load parameters for the preliminary load cases. For our example, using Eq. (2), the loads defined in Fig. 5 cause the computed loads in Table 4, as compared with the target levels.

**TABLE 4. Predicted Loads Caused by the Preliminary Load Case.** In this example, we are trying to achieve the target values for $F_{x1}$, $p_{ta}$, and $M_x$; total shear, $V_y$, is provided for information only. Units: lb, in·lb.

| Parameter | Predicted Load | Target Load | Ratio |
|---|---|---|---|
| $F_{x1}$ | 13,500 | 12,100 | 1.12 |
| $p_{ta}$ | 6000 | 6230 | 0.96 |
| $V_y$ | 6000 | 8140 | 0.74 |
| $M_x$ | -585,000 | -585,000 | 1.00 |

Note that the preliminary load case would overtest member 1 and undertest bolt $a$. We need to modify the applied loads both to ensure an adequate test and to minimize the risk of test failure. The test load for member 1 exceeds the allowable load from Table 3, resulting in a negative margin of safety.

7. Modifying the Load Cases

This step involves iterating the load cases until all the target parameters are achieved without any being uncomfortably surpassed, based on computed margins of safety. Automating the LTM equations with a spreadsheet or other means enables rapid iteration.

For our example, the problem boils down to how much shear at station 0, $V_y$, should co-exist with the peak bending moment, $M_x$. The preliminary load case does not have enough shear; increasing it without raising the bending moment requires moving the resultant applied shear vector aft. Table 5 shows one solution that achieves the target loads, based on iteration with Eq. (2).

**TABLE 5. Final Load Case for the Example Problem.** Units: lb, in·lb.

| $p_1$ | $p_2$ | $p_3$ | $p_4$ |
|---|---|---|---|
| 2290 | 1820 | 1650 | 1550 |

The above applied loads result in the following predicted loads, based on Eq. (2):

| Parameter | Predicted Load | Target Load | Ratio |
|---|---|---|---|
| $F_{x1}$ | 12,200 | 12,100 | 1.00 |
| $p_{ta}$ | 6240 | 6230 | 1.00 |
| $V_y$ | 7310 | 8140 | 0.90 |
| $M_x$ | -584,000 | -585,000 | 1.00 |

Modifying a preliminary load case in this manner is almost always necessary. In this case, the mode shape we tried to simulate was not quite right: the actual shape corresponds to a coupled mode of the spacecraft/booster system. Or perhaps a quasi-static lateral air load has been superimposed on the computed modal response during launch. In many cases, superimposing two mode shapes may give the best shape for a static test case to simulate. Launch loads consist of responses of multiple modes of vibration combined with quasi-static loads caused by thrust and wind gusts.

8. Scrubbing the Load Cases

The last step entails balancing the desire for ideal static representation of dynamic loads with any test constraints, such as cost and schedule.

The objective is to simplify the test to the greatest extent possible. We strive for the fewest test cases and the least number of applied loads for each case.

Using the steps 1 through 7, above, we may devise twenty load cases, whereas the test schedule may permit only five. If the test schedule and budget are consistent with early studies regarding the needed scope of testing, as opposed to being arbitrarily chosen, we should be able to design an effective test within those constraints.

When trying to reduce the number of load cases, look for structural symmetry and for similarity between load cases. For a qualification test, in which we are verifying a design, any symmetry within the structure will reduce the needed extent of testing. Only one member or joint of identical design needs testing. If, however, the test is a proof test, for the purpose of verifying workmanship, we need to test each member and each joint.

If two load cases are similar in terms of the mode shapes being simulated, we may be able to combine them into a single case that meets the objectives of the two. With this approach, we normally have to accept added risk of local overtesting, so we do it only when we believe the risk is low.

By assessing the full set of cases devised using steps 1 through 7, above, with the test LTM, we might find that one case tests only a single failure mode, whereas another case comes within, say, 7% of achieving the required load in that region. If the computed margins of safety indicate the risk is acceptable, we could increase loads for the latter load case by 7% and eliminate the first load case. Or, by examining the terms in the LTM, which are effectively influence coefficients, we might be able to identify adjustments to one or two applied loads in another load case that would achieve the target parameter without significantly affecting loads in the regions initially targeted by that case.

Another common constraint is the number of discrete loads we can apply in any one test case. Most spacecraft structures are tested with hydraulic jacks using an electronic control system. Such systems have limited numbers of channels they can control. If a desired load case requires more applied loads than the available channels, we could design a system of spreader bars (*whiffletree*), as shown in Fig. 6, to transform a single controlled load into two or more discrete loads applied to the test article. But such systems add complexity, and they usually are not stable under compression. Thus, we try to use them only when we can't make the test work with fewer applied loads.

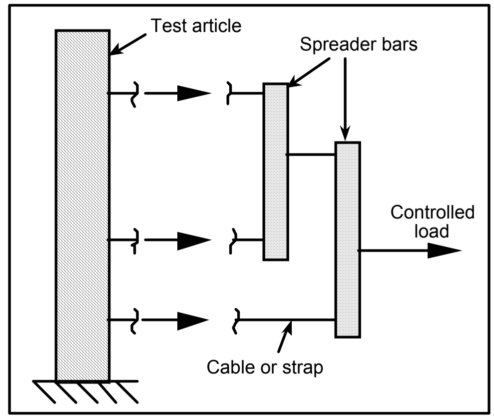



**Fig. 6. Example Whiffletree for Distribut-ing Loads.** Each spreader bar is statically determinate.

Finally, it is sometimes wise to accept an undertest in certain regions of the test article. To save money and schedule, or reduce risks elsewhere, space programs often accept risk associated with a less-than-perfect test. Such a decision is usually made at a level well above the engineer designing the test—and it should be—but the engineering staff provides key information to management in order to ensure a good decision. Without adequate testing, the area of concern must be verified by analysis, and when analysis is the sole method of verification, it deserves thorough documenta-tion and independent review.

## Reference

1. Sarafin, Thomas P., editor. 1995. *Spacecraft Structures and Mechanisms: From Concept to Launch*. Microcosm, Inc., and Kluwer Academic Publishers.