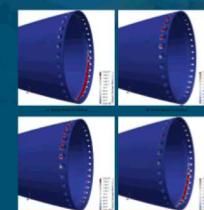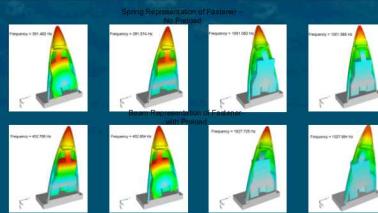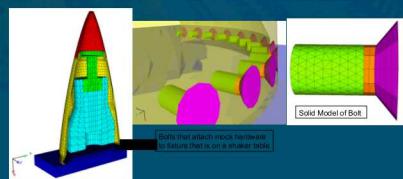SAND2019-0424C

Fastener Modeling Effects on Fatigue in Random Vibration


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1/21/2019

Abstract

- Explore the effects of different (low fidelity) modeling techniques of fasteners on fatigue predictions
  - Spring Model, Beam Model, and Solid Model (no threads)
  - Preload in FEA vs postprocessing the Preload

- Design engineers are asking analysis for margin of safety in concern to fatigue and ultimate/single passage failure
  - We can test a lifetime with *time compression*, but only up to a certain level. Testing is costly.
  - Normal environments (vibration and shock), small deformations, high-cycle

- Fatigue Analysis of mock hardware
  - The uncertainty created by the fastener modeling approaches is minuscule compared to the uncertainty from inputs
  - Small changes in modal analysis: is within calibration efforts
 - Goes without saying, but need test data for fastener parameters
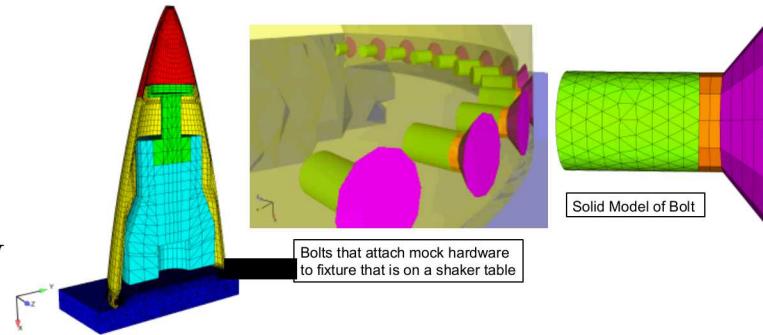
Solid Model of Bolt

Bolts that attach mock hardware to fixture that is on a shaker table

Outline

- Introduction

- Hardware

- Fastener Modeling Approaches – FE Analysis

- Fatigue Analysis Methods

- Credibility

- Results

Introduction

- Applications: Full System Models that have hundreds of fasteners that are joining subassemblies, components, etc. together with various sizing fasteners
  - Large models: 20 million DOF
  - How do we have confidence in the modeling techniques

- Fatigue has been cited as one of the major causes of in-service failure and it has been estimated that fatigue contributes to approximately 90% of all mechanical service failures.

- Normal environments with long duration random vibration events (captive carry environments for stores)

- Using mock hardware to generalize observations on actual hardware

- Literature Review
  - Papers on Spring/Beam Modeling (Low Fidelity)
    - Doubrava, "Effect of mechanical properties of fasteners on stress state and fatigue behavior of aircraft structures as determined by damage tolerances analyses", Procedia Engineering (2015)
    - Chandregowda et. al. "Evaluation of Fastener Stiffness Modeling Methods for Aircraft Structural Joints"
  - Papers on High Fidelity modeling fastener and threads
    - Sub-model approaches with Abaqus
    - Difficult including into a full system model
  - Papers on solid modeling of fasteners but not threads (Low Fidelity)
    - Mersch, Smith, and Johnson, "A Case Study for the low fidelity modeling of threaded fasteners subject to tensile loadings at low and high strain rates"
  - Analytical approaches to fasteners
    - 26th Edition Machinery's Handbook
    - Shigley and Mischke, Mechanical Engineering Design, 5th Edition
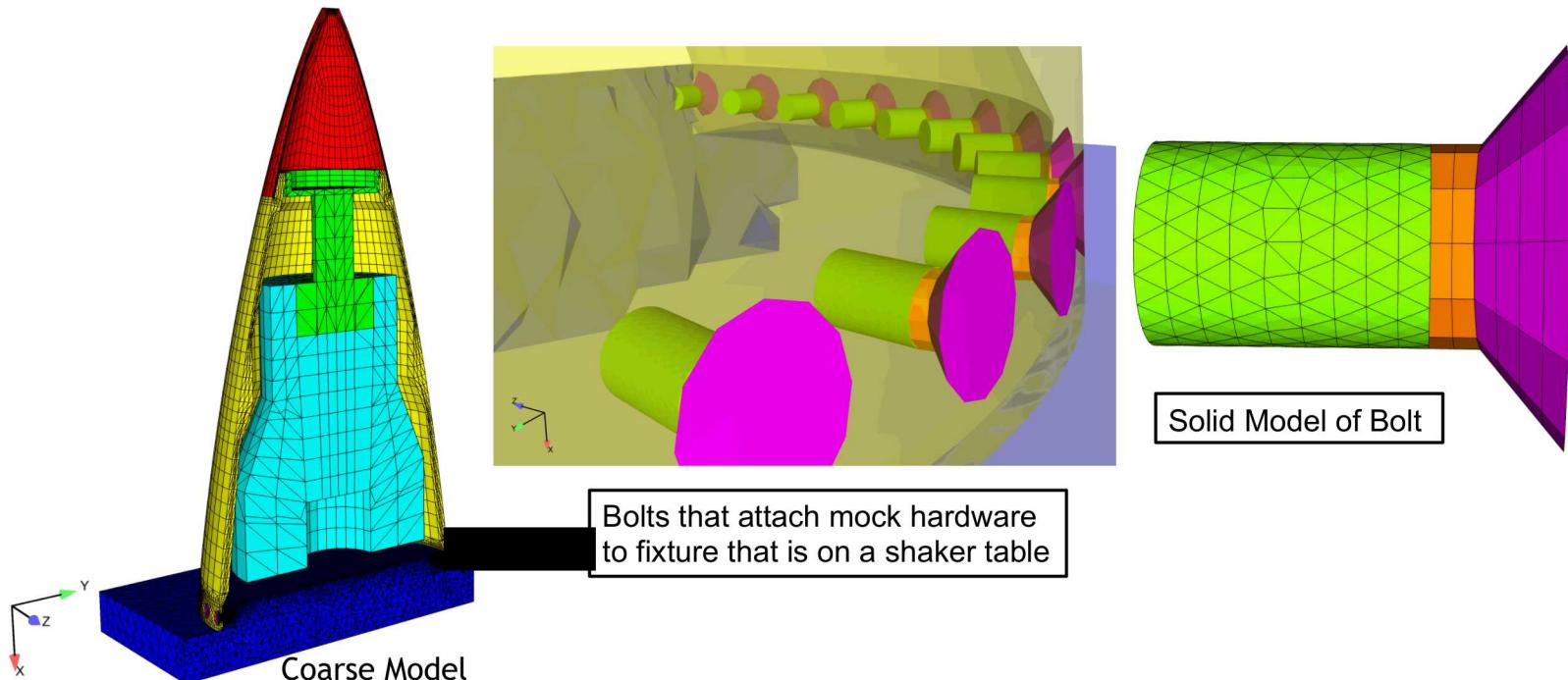 - Vibration Loosening of Fasteners
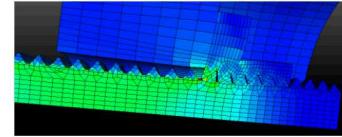
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Hardware

- Mock Hardware that would be attached to the shaker table with a fixture
  - Bolts attach case to fixture
  - 5/16-24, UNJF-3A with a torque Specification $135 \pm 13.5\ in-lbs$
  - Insert is Keenset Stainless Steel ½-13 external thread

- The case supports a mass that is hung like a pendulum
  - Produces high stresses away from attachment location at top of case near connection to pendulum mass
 - Produces high stress near attachment points

Solid Model of Bolt

Bolts that attach mock hardware to fixture that is on a shaker table

Coarse Model

Outline

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- **Fastener Modeling Approaches – FE Analysis**

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- Credibility

- Results

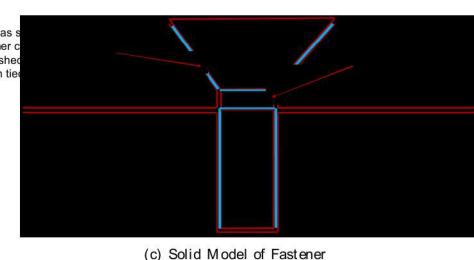# Fastener Modeling Approaches

- There are several methods for modeling fastener joints.

- There are three predominant methods for full system models (>10 million DOFs).
  - A forth, but I have not used option: One could do a full 3D model of threads and all, and then do a Global-Local Model (sub-model) approach.
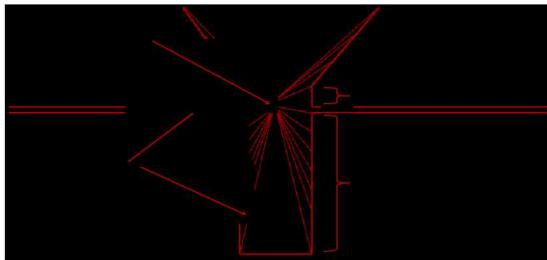
(a) Spring Model of Fastener
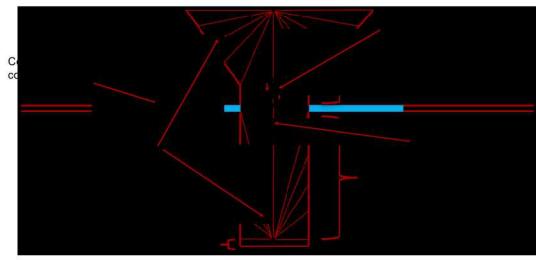
(b) Beam Model of Fastener

(c) Solid Model of Fastener

- To do the fatigue analysis near the fastener if preload is not included in the FEA, we use analytical approaches from basic machine design books (see Shigley & Mischke or Norton) in post processing.
  - This requires knowing the applied axial and transverse loads on the bolt from the environments.
  - In the post processing, one could account for friction given the tensile force and modify the shear force.

- Applying the preload in the FEA requires iteration.
  - This iteration will help with post processing the force or stress out of the solid model.

Fastener Axial Stress

- Preload $\quad F_i = \dfrac{T}{K_t d_{bolt}},$
  - T is the Torque, $K_t = 0.22$ with a coefficient of friction at 0.15
  - Torque: $135 +/- 13.5$ in-lbs

- Separation Load

$$P_o = \frac{F_i}{1-C}, \text{ where } C = \frac{k_b}{k_b + k_m}, \qquad k_b = \frac{A_t E_b}{L},$$

$$k_m = \frac{\pi E_m d_b}{ln\left[5\left(\frac{2L+0.5d_b}{2L+2.5d_b}\right)\right]}$$

- Define a tension load in the bolt

$$\text{if } F_{axial}(t) < P_o, \text{ then}$$
$$F_t(t) = F_i + C F_{axial}(t),$$
$$\text{else}$$
$$F_t(t) = F_{axial}(t),$$

$$L = l_1 + l_2 + (l_n + \frac{L_i}{2}),$$

$$E_m = \frac{L}{\frac{l_1}{E_1} + \frac{l_2}{E_2} + \frac{l_n - L_i/2}{E_n}},$$

- With tension load define axial stress

$$\sigma_t(t) = \frac{F_t(t)}{A_t}.$$

Note: This is a function of time!

Parent Material Thread Shear/ Insert Pull-out

- If the parent material is significantly weaker than the insert material, then this failure can happen
  - Aluminum Case with steel inserts
  - Common in aerospace structures

- Area for the shear stress of the case thread

$$A_{par} = \pi n_n L_e D_s \left(\frac{1}{2n_n} + 0.57735(D_s - E_n) \right),$$

- Shear Stress of parent material at the threads at the insert:

$$\tau_{par}(t) = \frac{F_t(t)}{A_{par}}.$$

Note: This is a function of time! – We can look at Fatigue around the fastener

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Fatigue Big picture

- Fatigue is caused by a fluctuation of the applied stress with a sufficiently large number of cycles

- We are doing High Cycle, because the environments generate a large number of cycles ($> 10^5$) and stress was analyzed to be in the elastic regime

- Given a stress state, cycle counting is performed with well established Rainflow techniques [10]
  - Algorithm provides a peak stress and an amplitude
  - We can get a stress ratio for each cycle

$$R = \frac{\sigma_{min}}{\sigma_{max}},$$

- S/N curve Data from MMPDS-11 (formerly MIL-HDBK-5) to get N_i
  - Use Goodman Modification if S/N data does not accommodate high stress ratio due to preload
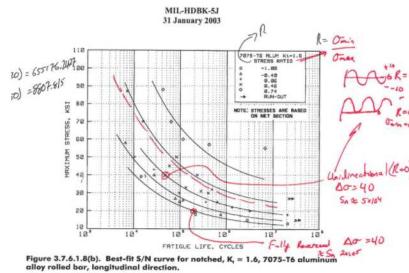
- Palmgren-Miner Damage Accumulation : $D \geq 0.5$ indicative of fatigue failure

$$D = \sum_{i=1}^{k} \frac{n_i}{N_i},$$

[10] S. D. Downing and D. F. Socie. Simple rainflow counting algorithms. *Int. J. Fatigue*, January 1982.

What Stress to Use for Fatigue Analyses

- Most test data is from uniaxial experiments (typical hourglass fatigue test)

- In a multiaxial random vibration simulation, the stress within a finite element will not be uniaxial
  - Stress tensor $(\sigma_{11}, \sigma_{22}, \sigma_{33}, \sigma_{12}, \sigma_{23}, \sigma_{13})$
  - Functions of time

- A method is needed to reduce the multiaxial stress state to an equivalent uniaxial stress for comparisons to uniaxial experiments

- Multiaxial Fatigue Life Methods
  - **Stress-based approaches**
  - Strain-based and energy-based models
  - Fracture Mechanics Models
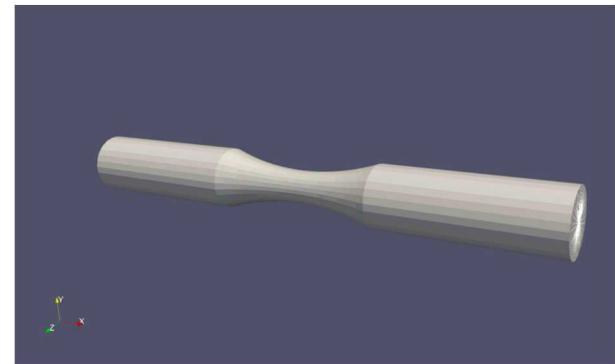
Multiaxial Fatigue

- "'For fatigue calculations, we usually are interested in stresses acting parallel to and perpendicular to the free surface of a component where fatigue cracks nucleate and start to grow" [9].
  - For a solid body, the free surface out-of-plane stresses must be zero, unless a pressure load is applied
  - Map σ_{p1} to σ_{p1} and σ_{p3} to σ_{p2}

- Proportional vs Nonproportional Fatigue
  - *Proportional loading, which is defined as any state of time varying stress where the orientation of the principal stress axes remained fixed with respect to the axes of the component". Thus, nonproportional loading is defined as any state of time varying stress where the orientation of the principal stress axes changes with respect to the axes of the component. [9]*
  - Biaxiality Ratio = σ_{p3}/σ_{p1}
    - Fairly constant throughout time, then multiaxial proportional fatigue methods are appropriate
    - Popular Proportional Methods: Signed Absolute Maximum Principal Stress & Signed von Mises
    - Recall: Derive an equivalent stress term that can be related to uniaxial testing

[9] Socie D.F. and Marquis G.B., *MultiaxialFatigue*, Society of Automotive Engineers, Inc., 2000.

Multiaxial Fatigue: Biaxiality Ratio

- Biaxiality Ratio = σ_{p3}/σ_{p1}
  - In typical FEA for solid elements the stress tensor is obtained at the center of the element
  - Consequently the biaxiality ratio needs to be **fairly** constant
 - Just look at a simulation of a hourglass tensile test specimen
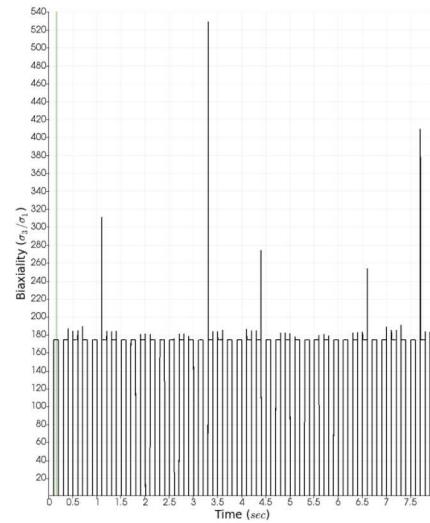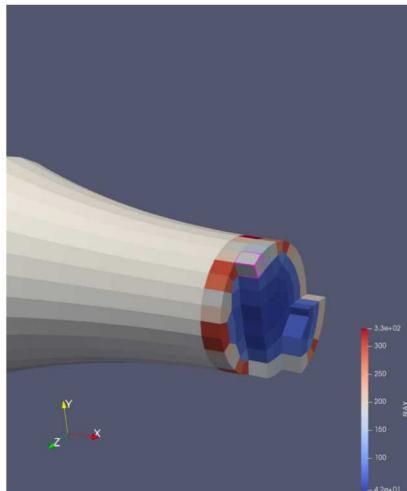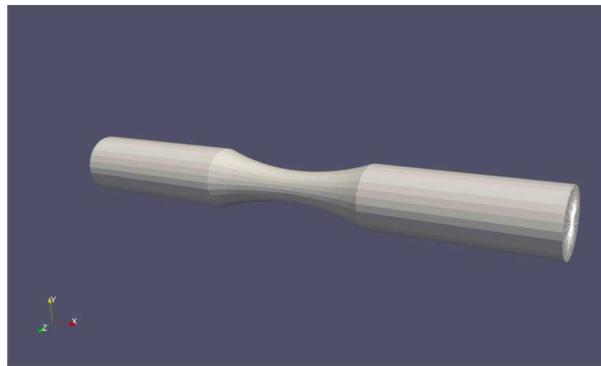
Mulitaxial Fatigue: Proportional Methods

- Yield Criterions: Signed von Mises & Signed Absolute Maximum Principal
  - Signed: Terms would always be positive and misleading in rainflow calculation

$$\sigma_{vm} = \frac{1}{3}\sqrt{(\sigma_{11} - \sigma_{22})^2 + (\sigma_{22} - \sigma_{33})^2 + (\sigma_{11} - \sigma_{33})^2 + 6(\sigma_{12}^2 + \sigma_{23}^2 + \sigma_{13}^2)}.$$

  - Absolute maximum principal stress sign is the sign used: Example

Time Step	Principal Stress 1	Principal Stress 3	Multiply σ_{vm} by
1	-40	-60	-1
2	10	-30	-1
3	40	-30	1
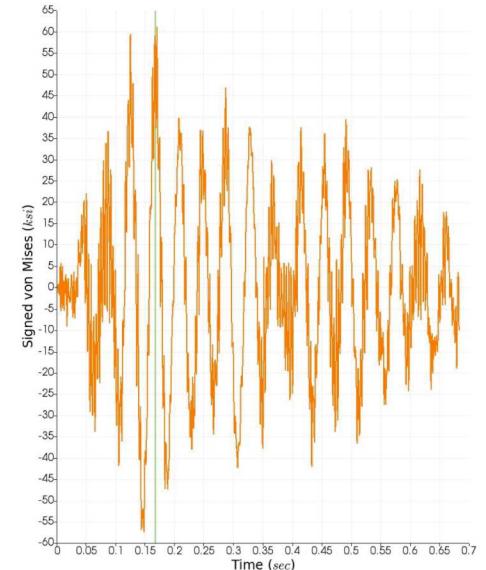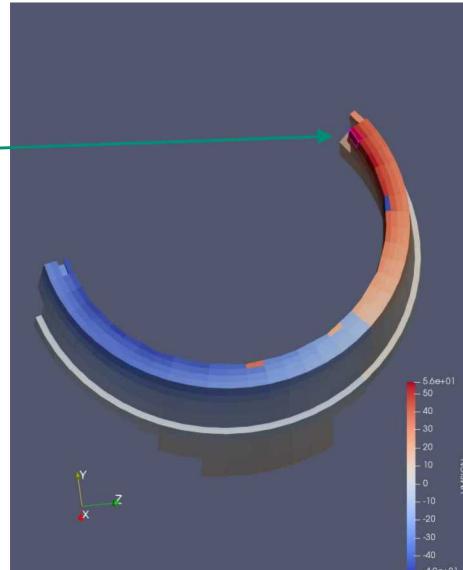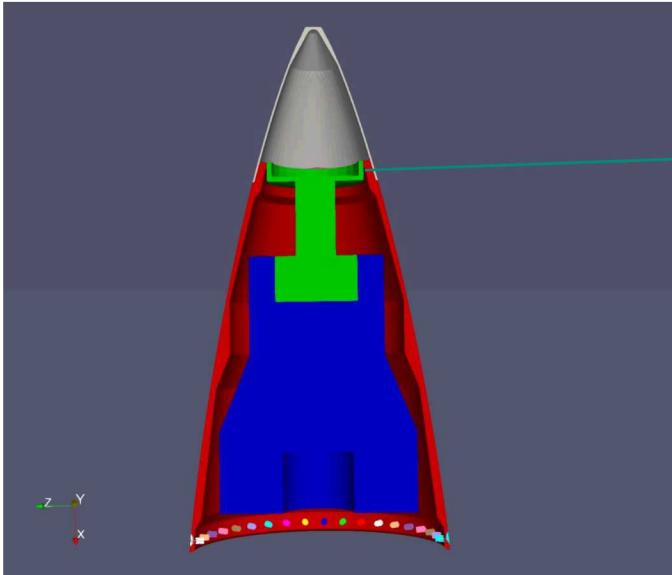
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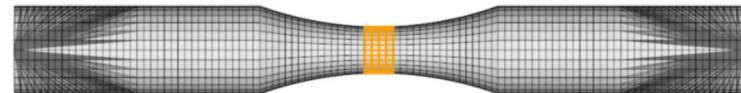- Fatigue Analysis Methods

- **Credibility**

- Results

Fatigue code Credibility

[10]

- Compare to published experimental results
  - Compared Rainflow Algorithm to other codes
  - Anes et al. hourglass specimen [10]
    - FEM, orange elements are analyzed in Fatigue Code
    - Time History analysis with 5Hz input for 428 MPa, 2.13×10^6 cycles, 4.26×10^5 sec
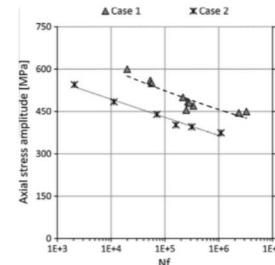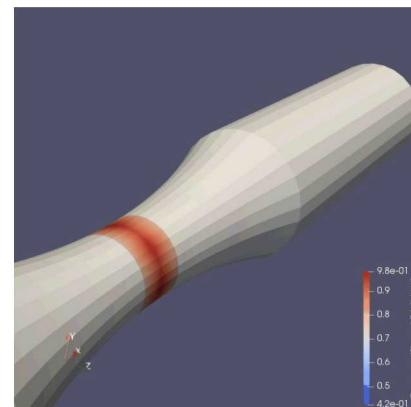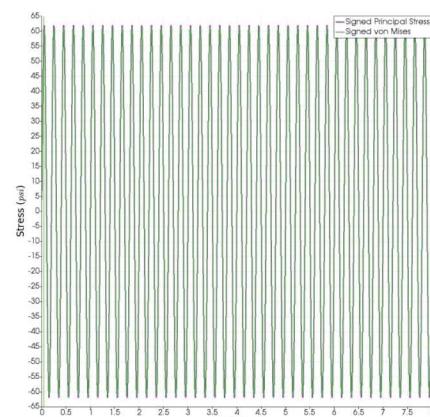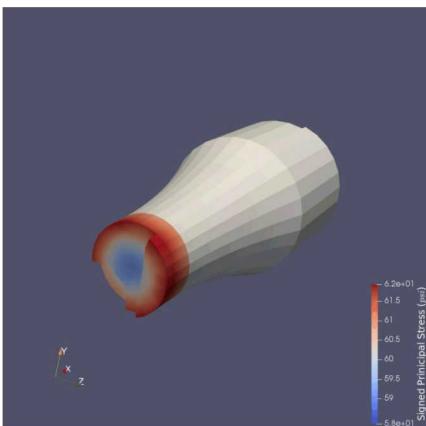
Fatigue Method	Max Stress (ksi)	Damage
Signed von Mises	61.3	0.82
Signed Principal 1	61.9	0.98

- Comparisons for Random Vibration

[10] Anes, V., Reis, L., Li, B., Fonte, M., and de Freitas, M., "New approach for analysis of complex multiaxial loading paths," *International Journal of Fatigue*, Vol. 62, 2014, pp. 21–33.

Outline

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- Fatigue Analysis Methods
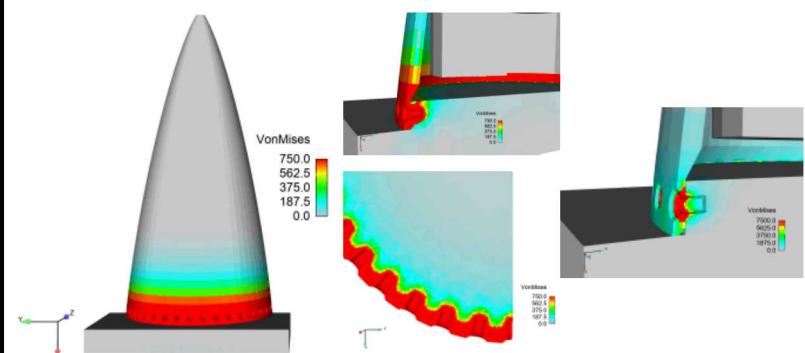- Credibility
- **Results**

Preload Stress State

- Solid 3D model of Bolt and Beam
  - Preload is an iterative process
  - Following resultant static von Mises stress states are small compared to yield

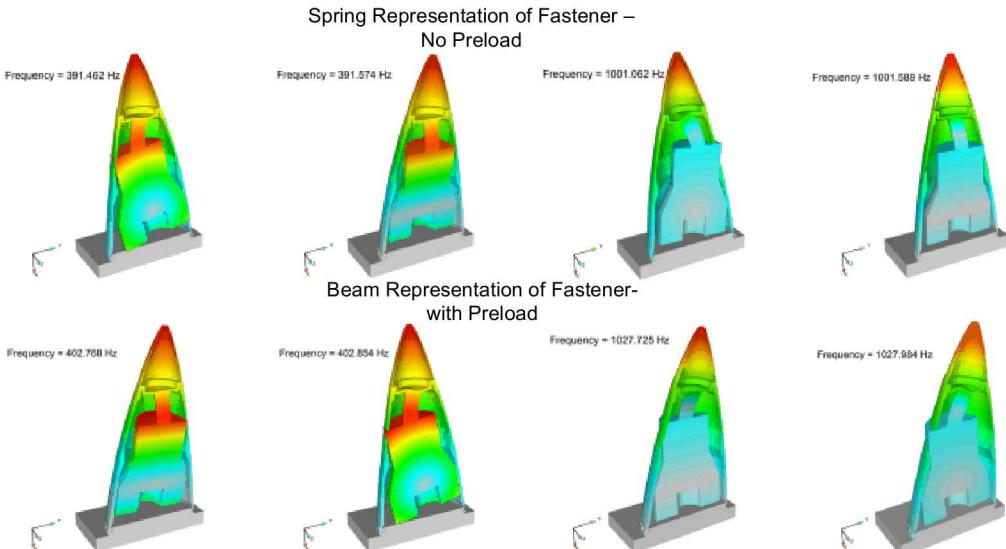
Beam Model Fasteners
von Mises (psi)
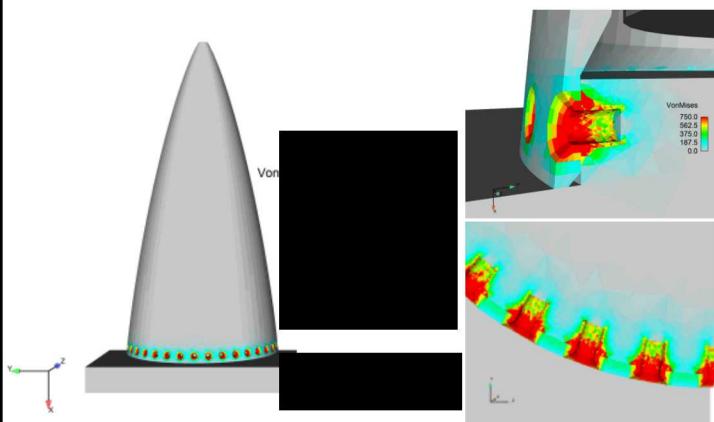
Solid Model Fasteners
Von Mises (psi)

Modal Analysis

- Modes below 2,000 Hz are well within typical calibration efforts of attempting to be within 5%

- Different fastener modeling methods can be adjusted to better match modal test data

Mode	Spring No Preload (Hz)	Beam No Preload (Hz)	Beam Preload (Hz)	Solid Preload (Hz)	Maximum Difference (%)
1	25.0	25.0	24.6	24.6	1.4
2	25.0	25.0	24.6	24.6	1.4
3	136.6	136.6	136.0	136.0	0.4
4	246.8	247.4	242.3	242.8	2.1
5	391.5	405.1	402.8	409.2	4.3
6	391.6	405.3	402.9	409.3	4.3
7	1001.1	1034.8	1027.7	1041.9	3.9
8	1001.6	1035.2	1028.0	1042.1	3.9
9	1102.6	1159.5	1154.7	1144.0	4.9
10	1102.7	1159.5	1154.8	1144.1	4.9
11	1620.6	1694.6	1690.0	1655.7	4.4
12	1620.6	1694.7	1690.0	1655.8	4.4
13	1933.5	1962.4	1952.0	1980.8	2.4
14	1933.6	1962.5	1952.2	1980.9	2.4
15	2093.6	2108.7	2096.5	2180.7	4.0
16	2100.9	2201.6	2184.1	2273.5	7.6
17	2284.5	2398.2	2387.0	2323.9	4.7
18	2284.5	2398.2	2387.0	2324.0	4.7
19	2370.3	2524.8	2515.1	2418.1	6.1
20	2370.3	2524.9	2515.2	2418.2	6.1
21	2431.5	2537.6	2519.9	2493.9	4.2
22	2431.9	2538.0	2520.4	2494.5	4.2
23	3037.6	3085.1	3073.9	3053.7	1.5
24	3037.6	3085.4	3074.0	3053.8	1.5

Spring Representation of Fastener – No Preload

Frequency = 391.462 Hz
Frequency = 391.574 Hz
Frequency = 1001.062 Hz
Frequency = 1001.588 Hz

Beam Representation of Fastener- with Preload

Frequency = 402.768 Hz
Frequency = 402.854 Hz
Frequency = 1027.725 Hz
Frequency = 1027.984 Hz

Input

- Obviously a random vibration specification can drastically change the response if a mode is at the bottom or top of an upslope
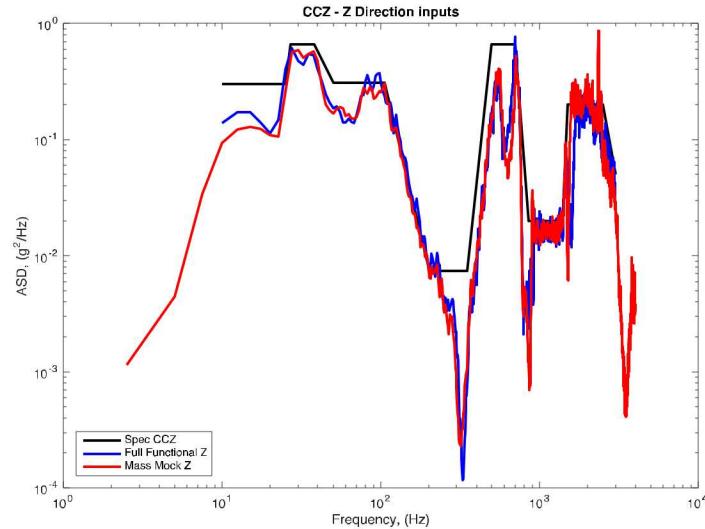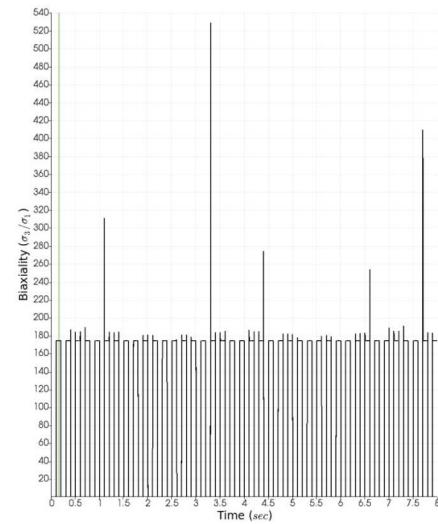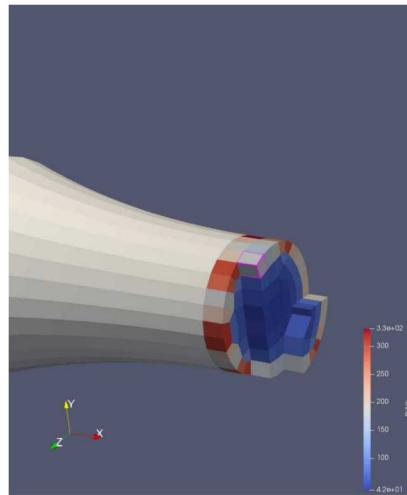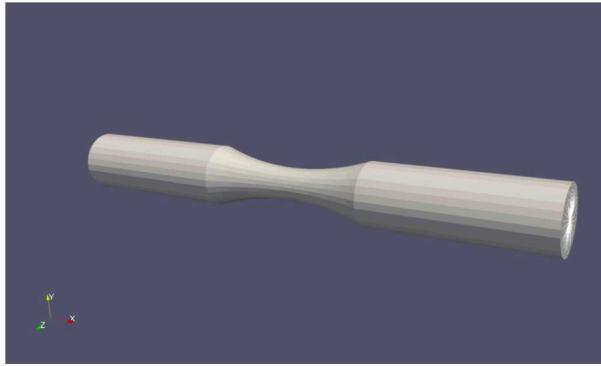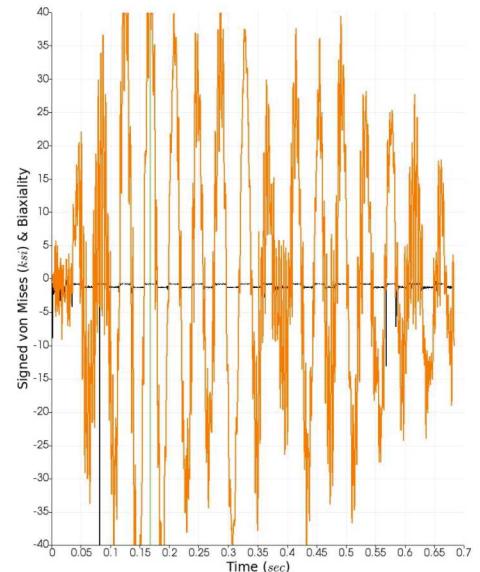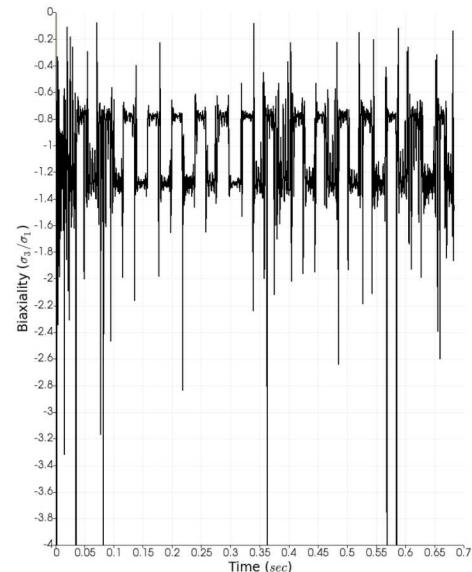

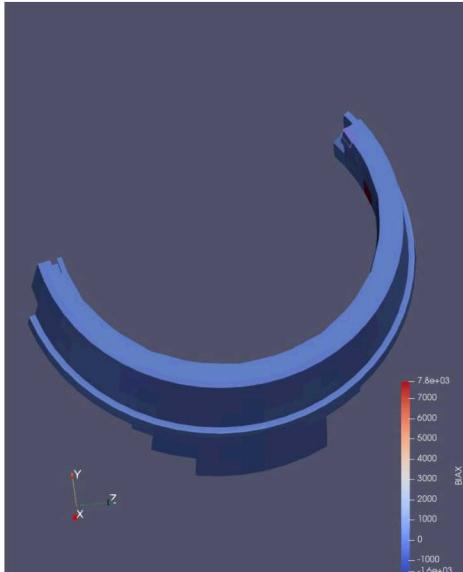- For this study, a flat spectrum is used for the random vibration analysis of mock case
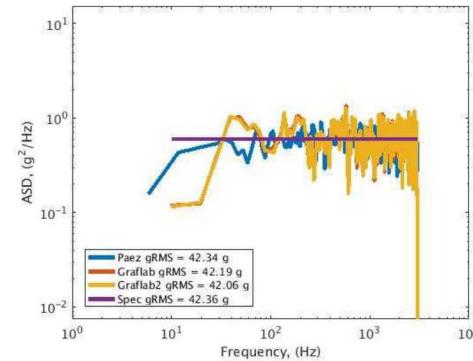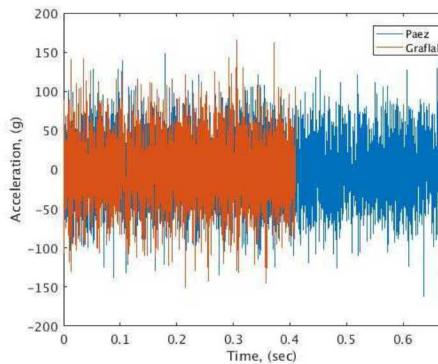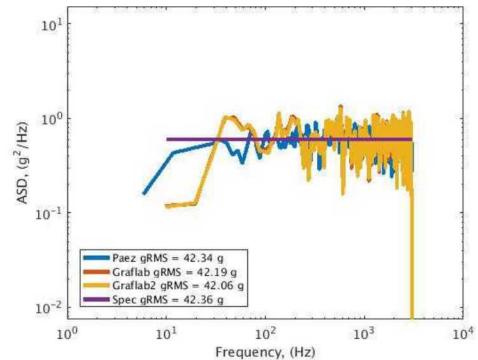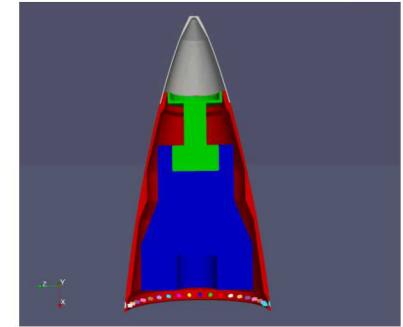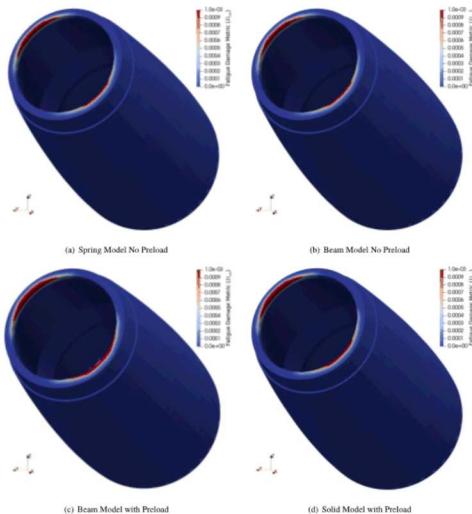
Biaxiality

- Recall Biaxiality
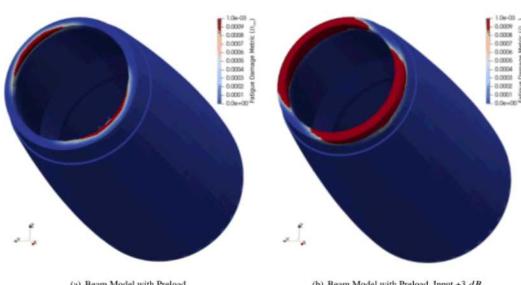

- What is the biaxiality on the case in the area of interest

Fatigue Maximum Damage in Case

- Maximum Damage was noted away from the fasteners, but at connection of the pendulum
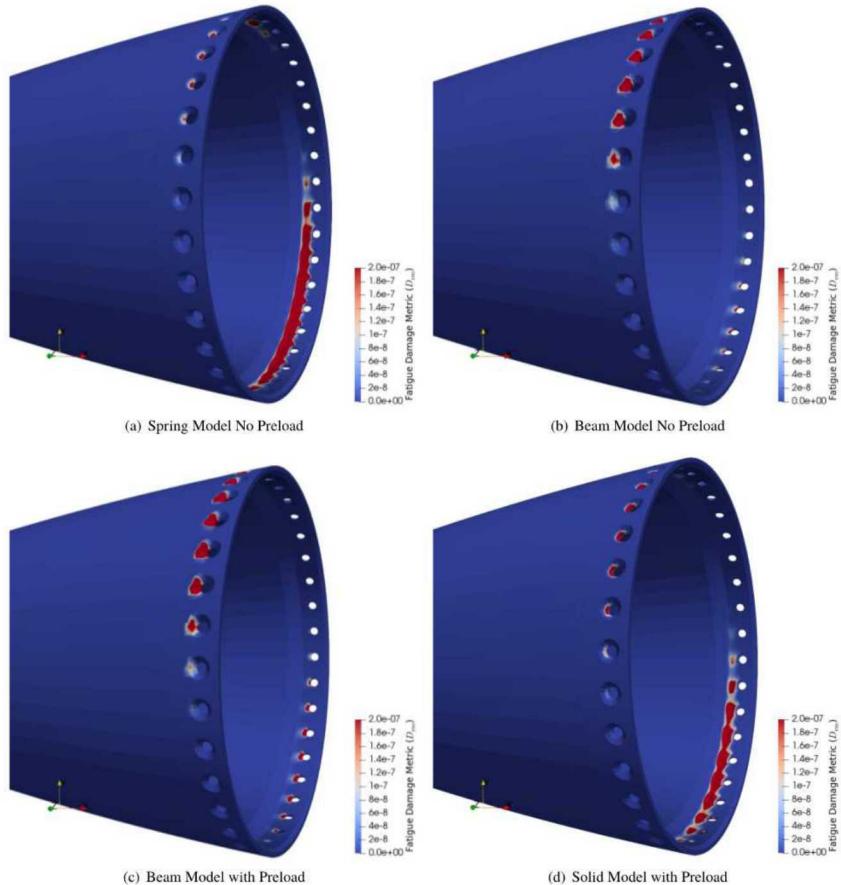

(a) Spring Model No Preload   (b) Beam Model No Preload

(c) Beam Model with Preload   (d) Solid Model with Preload

- Increase the input to be within test tolerance ($\pm 3\ dB$)

(a) Beam Model with Preload   (b) Beam Model with Preload, Input +3 dB

Model	Damage Metric	% Difference vs Beam PL
Beam PL	2.28E-03	0
Spring No PL	1.94E-03	-15
Beam No PL	1.76E-03	-23
Solid PL	2.26E-03	-1
Beam PL $\pm 3\ dB$	1.288	56391

Fatigue Damage Near Fasteners

- Look at the damage near the Fasteners

- Surprisingly the spring model looks a lot like the solid fastener model

(a) Spring Model No Preload

(b) Beam Model No Preload

(c) Beam Model with Preload

(d) Solid Model with Preload

Model	Damage Metric	% Difference vs Beam PL
Beam PL	4.98E-06	0
Spring No PL	1.24E-05	149
Beam No PL	3.26E-06	-35
Solid PL	1.22E-05	145
Beam PL ±3 dB	2.49E-04	4900

- Does the additional complication add value to your FEA?
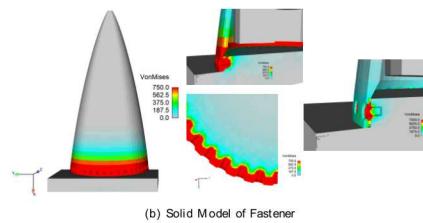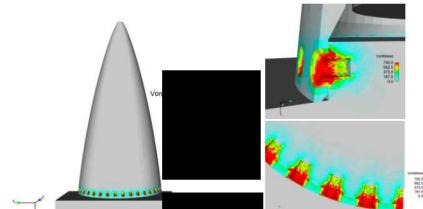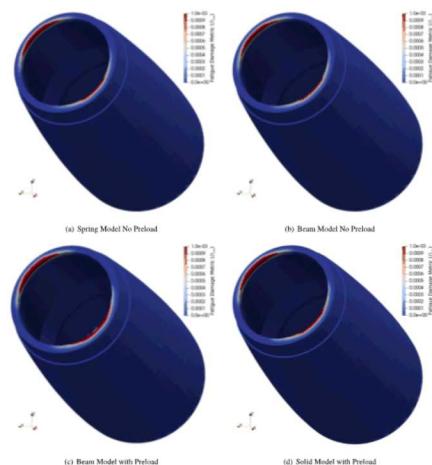

(a) Beam Model of Fastener

(b) Solid Model of Fastener
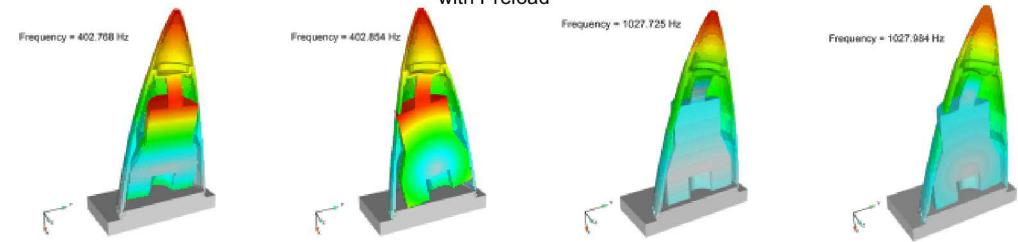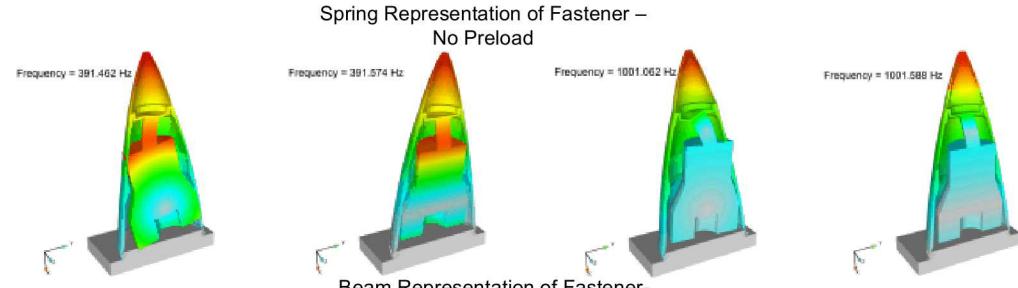
Figure 7. The static stress surrounding the fasteners after applying the preload to the fasteners modeled.

Spring Representation of Fastener – No Preload

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Frequency = 391.574 Hz

Frequency = 1001.062 Hz

Frequency = 1001.588 Hz

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Frequency = 1027.725 Hz

Frequency = 1027.984 Hz

(a) Spring Model No Preload

(b) Beam Model No Preload

(c) Beam Model with Preload

(d) Solid Model with Preload

In Summary

- The area of the joint surrounding the fastener is put into compression
  - Metals are not going to fatigue in compression
  - Beware of loss of preload for this leads to fretting
  - In this example and seen in full system models the compression stress state is low

- The uncertainty created by the modeling approaches is minuscule compared to uncertainty from inputs.

- Small Small changes in modal analysis: within calibration efforts

- Used multiaxial proportional fatigue methods for the solid parts of interest

- Used multiaxial non-proportional fatigue methods for the fasteners

Back-ups

Fastener Axial Stress

- Preload $F_i = \dfrac{T}{K_t d_{bolt}}$,
  - T is the Torque, $K_t = 0.22$ with a coefficient of friction at 0.15
  - This is highly variable, uncertainty, Torque: 135 +/- 13.5 in-lbs

- Separation Load

$$P_o = \frac{F_i}{1-C}, \text{ where } C = \frac{k_b}{k_b + k_m}, \qquad k_b = \frac{A_t E_b}{L},$$

$$k_m = \frac{\pi E_m d_b}{ln\left[5\left(\frac{2L+0.5d_b}{2L+2.5d_b}\right)\right]}$$

- Define a tension load in the bolt

$$\text{if } F_{axial}(t) < P_o, \text{ then}$$
$$F_t(t) = F_i + C F_{axial}(t),$$
$$\text{else}$$
$$F_t(t) = F_{axial}(t),$$

$$L = l_1 + l_2 + (l_n + \frac{L_i}{2}),$$

$$E_m = \frac{L}{\frac{l_1}{E_1} + \frac{l_2}{E_2} + \frac{l_n - L_i/2}{E_n}},$$

- With tension load define axial stress

$$\sigma_t(t) = \frac{F_t(t)}{A_t}.$$

Note: This is a function of time!

Parent Material Thread Shear/ Insert Pull-out

- If the parent material is significantly weaker than the insert material, then this failure can happen
  - Aluminum Case with steel inserts
  - Common in aerospace structures

- Area for the shear stress of the case thread

$$A_{par} = \pi n_n L_e D_s \left(\frac{1}{2n_n} + 0.57735(D_s - E_n) \right),$$

- Shear Stress of parent material at the threads at the insert:
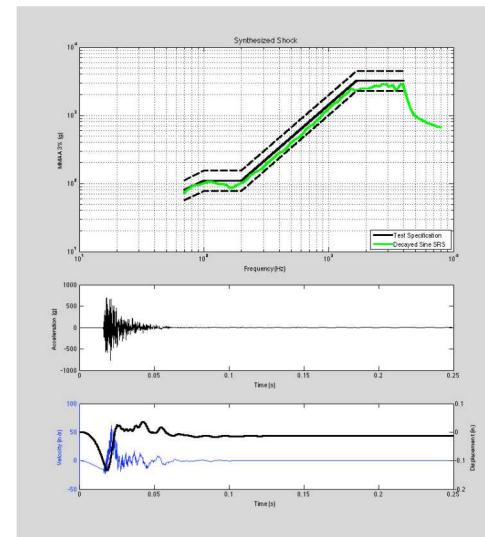
$$\tau_{par}(t) = \frac{F_t(t)}{A_{par}}.$$

Note: This is a function of time!

Typical Inputs for Structural Dynamics

- Random Vibration Specification is almost always in a ASD form
  - Inputted directly with SIERRA SD and simulated using modal random vibration
  - ASD can be converted to a time realization as if going on a shaker table
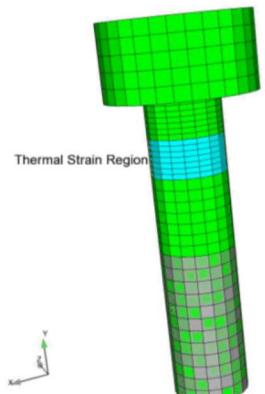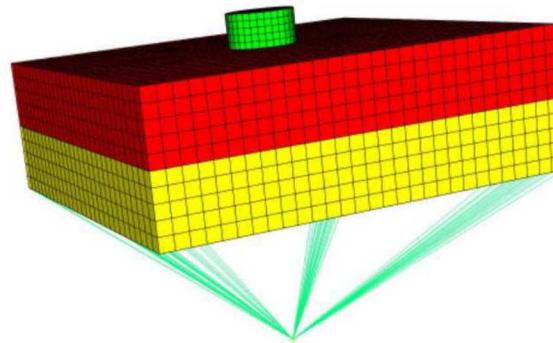   - Make sure you do this appropriately. It is easy to screw up
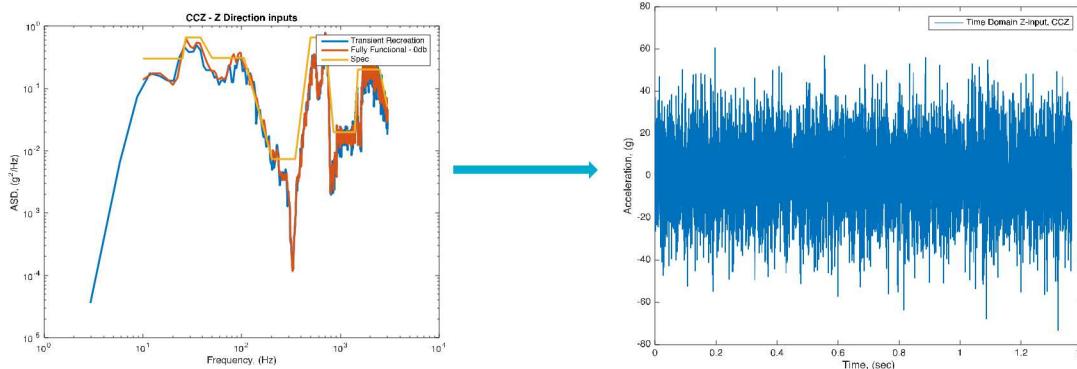
- Shocks Specifications are usually SRS or a typical signal such has a haversine.
  - A time realization for input into a transient analysis can be derived from the Shock specification.

- For the derivation and discussions on fasteners, let's assume inputs are time histories.

- At the end, I will provide a simple method for doing fasteners in the frequency domain.

Fastener Modeling Credibility

- Bolt Modeling: Solid, Beam, Spring

Thermal Strain Region

Spring Element Representing Joint Mechanics

Beam

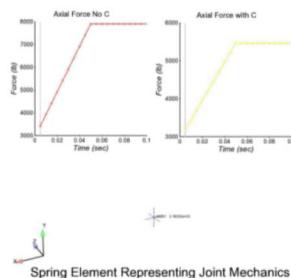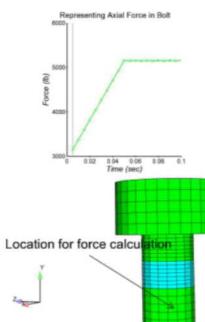Insert surface rigidly tied to bottom of beam
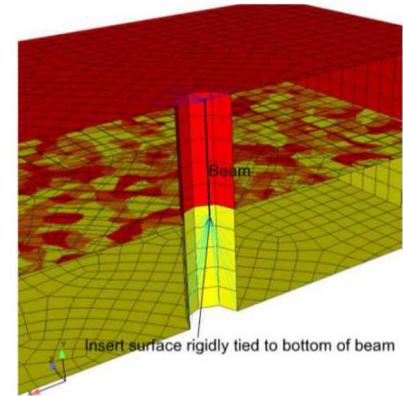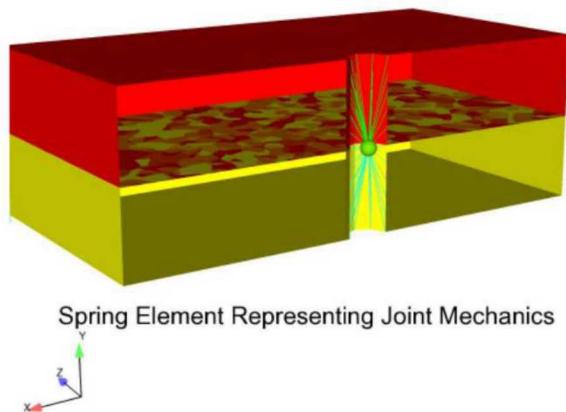
- Interface contact at mating surfaces can greatly change these results
  - Do you use the frustrum for the contact area?
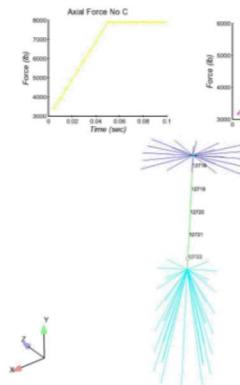 - Ultimately, you are going to want to tune things to match some test data, so will it matter?
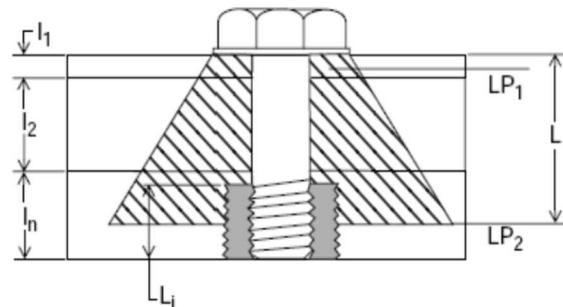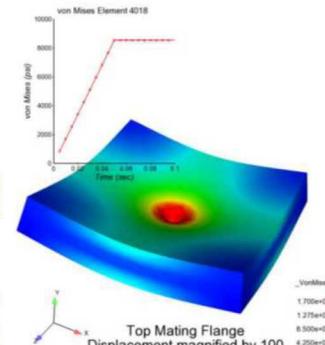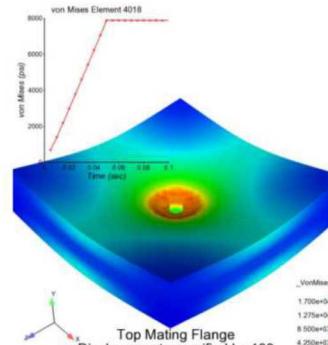

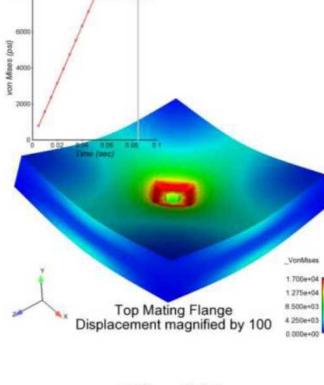| Mode	Beam Frequency (Hz)	Spring Frequency (Hz)	Solid Frequency (Hz)	Difference Spring vs Solid
1	2944.0	2942.1	2896.0	1.6%
2	6420.3	6414.4	6445.7	-0.5%
3	9163.2	9176.0	9097.9	0.9%

- Do we get the same load in the Bolt
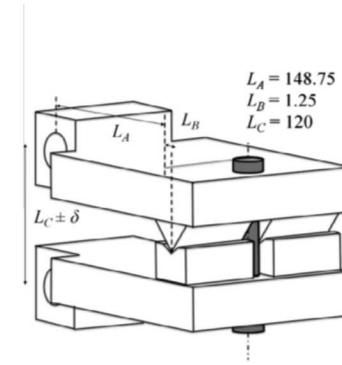
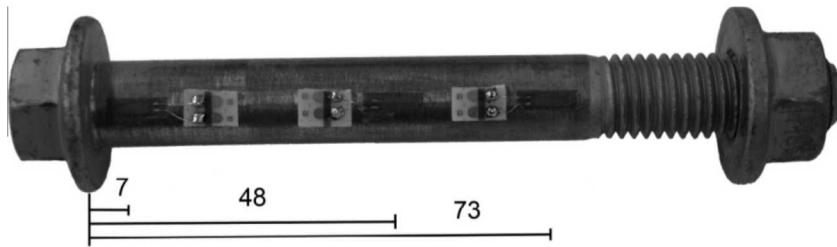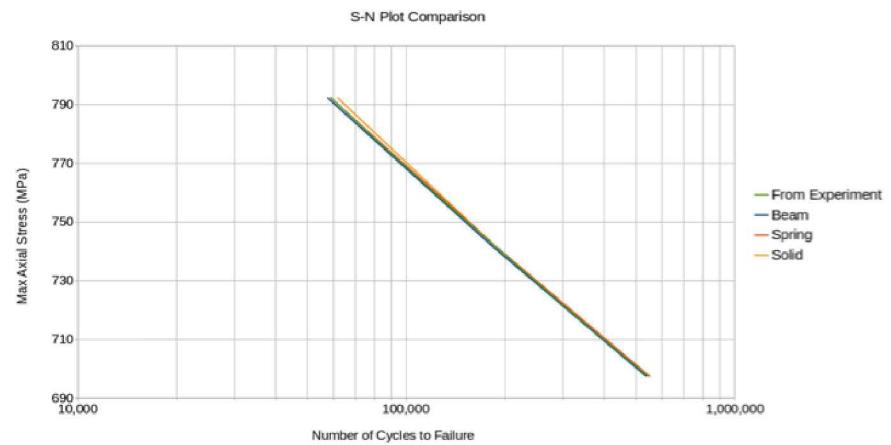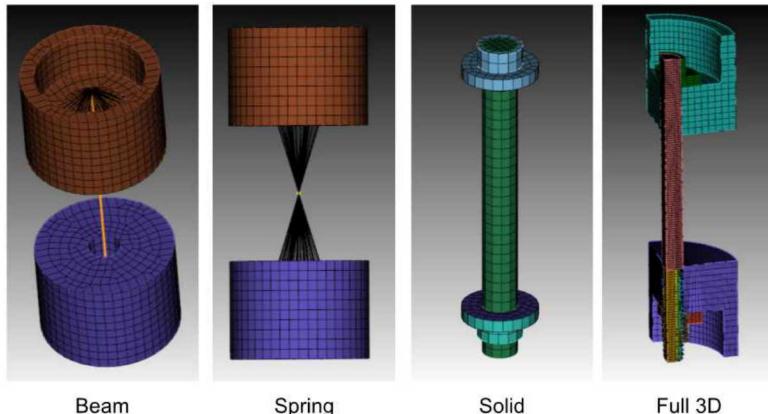- Do we get the same load in the Flange

Fix Figures

Fastener Fatigue Code Credibility

- Fastener Credibility: Wentzel and Huang's Experiment [15]

- Beam Model Results with SN Curve Generation

[15] Henrik Wentzel and Xiyue Huang. Experimental characterization of the bending fatigue strength of threaded fasteners. *International Journal of Fatigue*, 72:102–108, 2015.