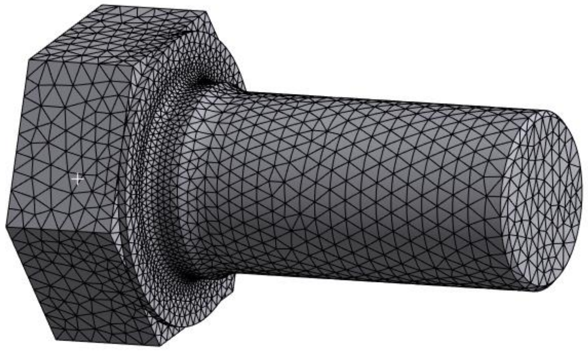
Twin Cities ANSYS® User Meeting

Simulating Bolts

ANSYS Toolset



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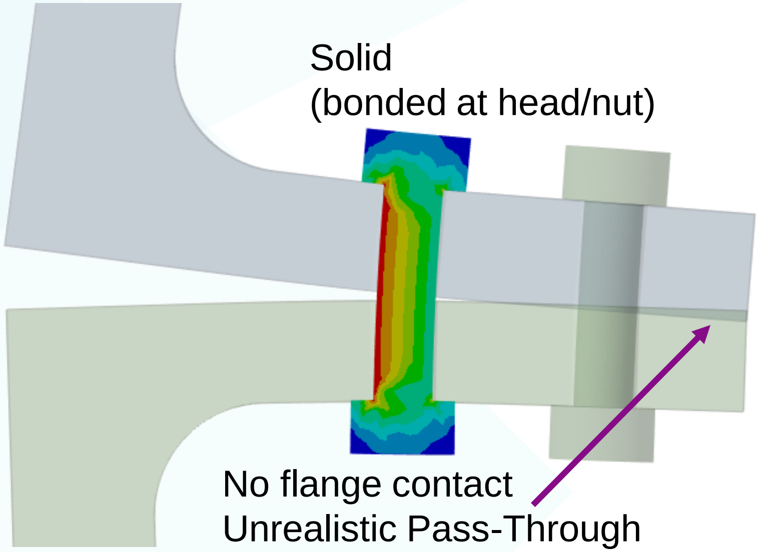
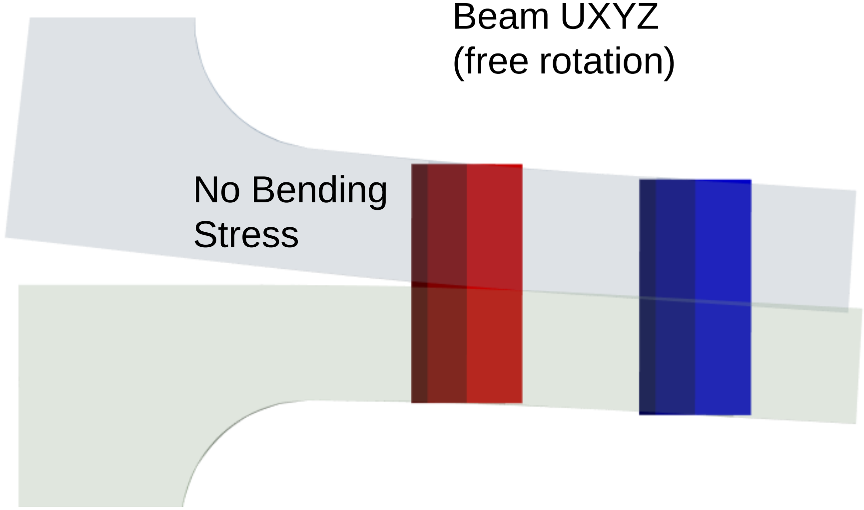
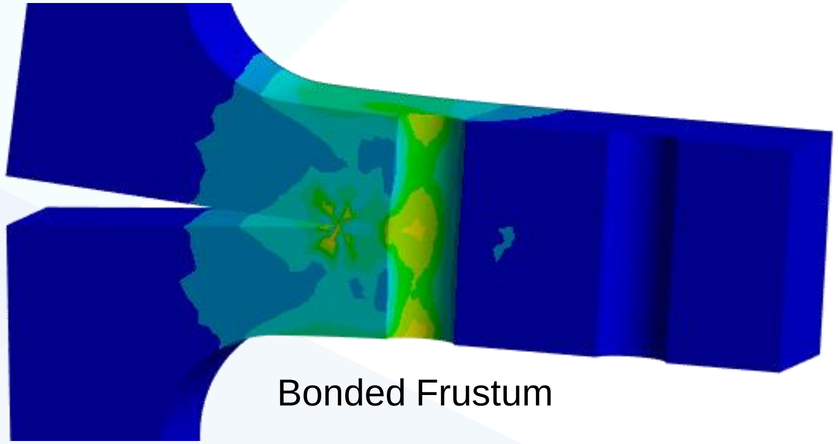
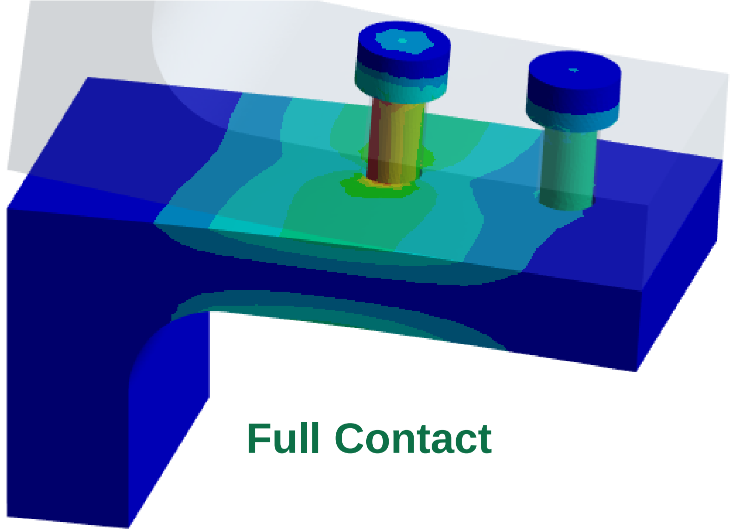
Bolted Interface Challenge

Three Approximations… and One with Full Contact









About Epsilon

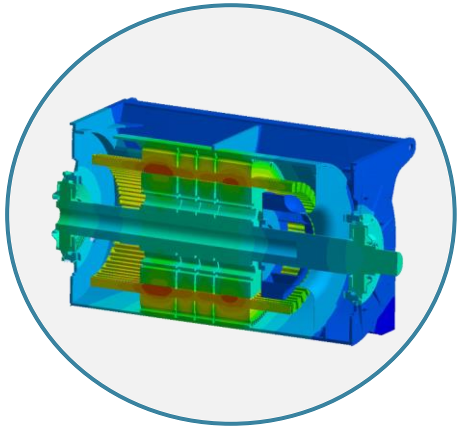
Company Mission

Our Core Values

Epsilon FEA was formed in 2008 in Minneapolis, Minnesota to provide a new class of Engineering Service utilizing the Finite Element method and related CAE tools.

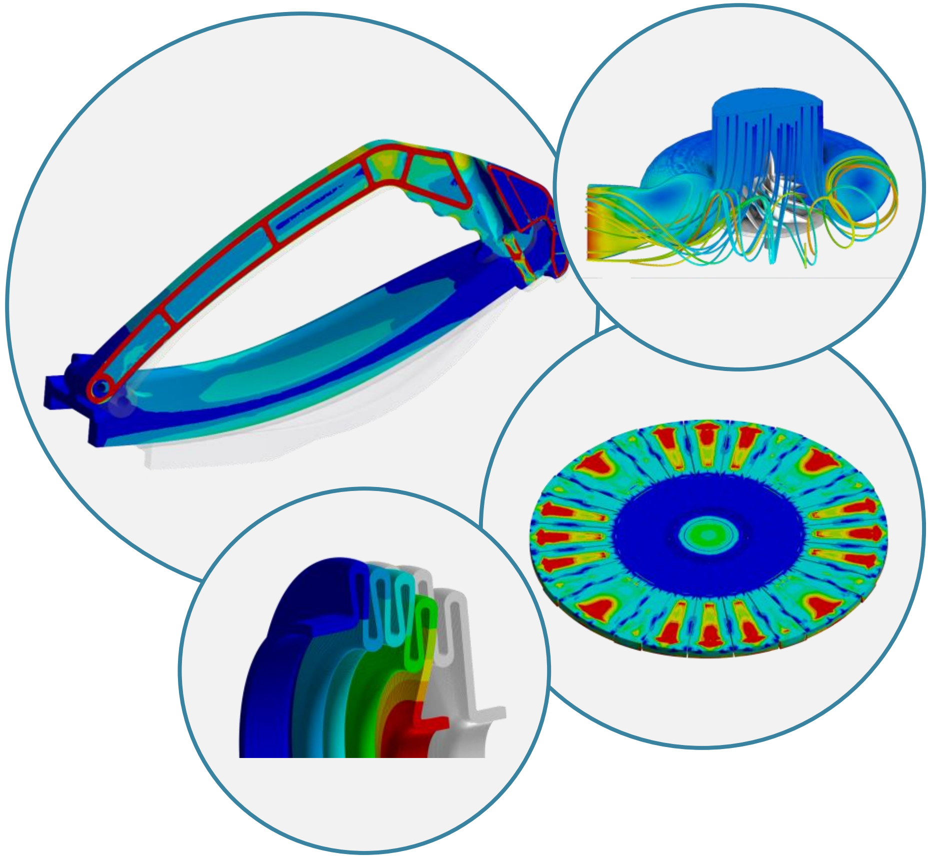
Recognizing this niche CAE tool requires large investment for companies competing in a technically challenging environment, Epsilon focuses on three cumulative characteristics of our services:

- ✓ **Exhibit Excellence with the Simulation Tools**
- ✓ **Infuse Technology into Customer Design System**
- ✓ **Communicate Thoroughly and Clearly**



What We Do

Our Tools



We utilize ANSYS tools both of Workbench as well as classic APDL (ANSYS Parametric Design Language) Fluent, CFX, along with a multitude of supporting engineering and business software.

Using up-to-date licensing and compute solvers we leave your costly internal resources intact, while leveraging our familiarity and expertise with our own in-house toolset that has been customized and augmented for over a decade and a half.

Pairing these long term investments with analysts (consultants) that perform FEA/CFD as their career focus results in a low total project cost thereby making Epsilon FEA a strategic partner for your next engineering challenge.

Why Epsilon?

Unique Value Proposition to Best Serve Our Customers

Superior Engineering Analysis

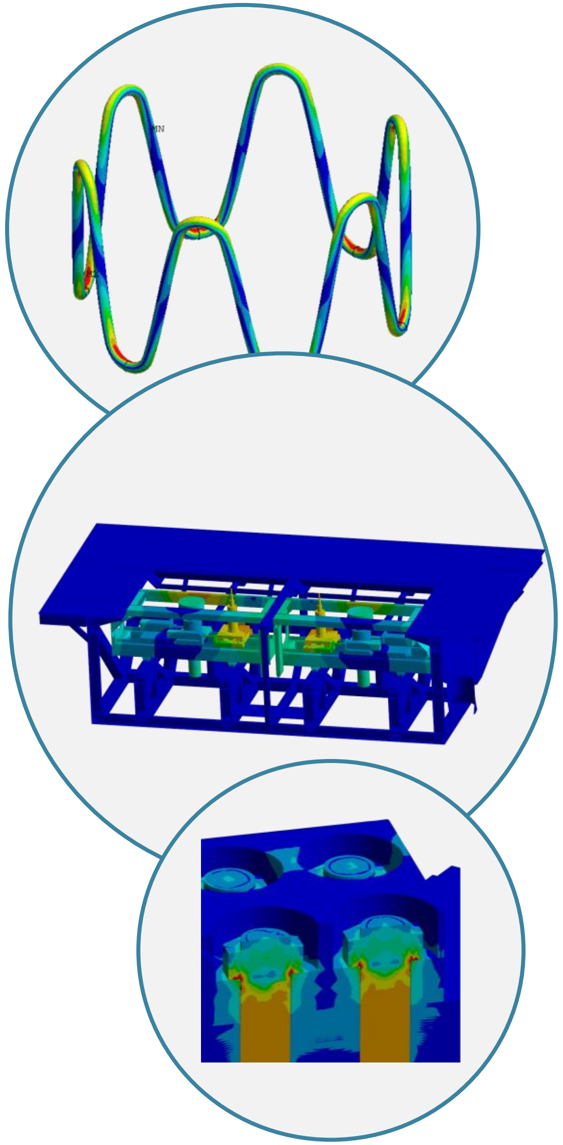
- 3 full-time simulation experts + network of additional experts as needed

Low Overhead

- Support functions outsourced
  - Accounting, IT, Finance, Technical writing, etc.
- Big Business Interface
  - Detailed invoicing/SOWs, updated toolset, insured, quality assurance, etc.

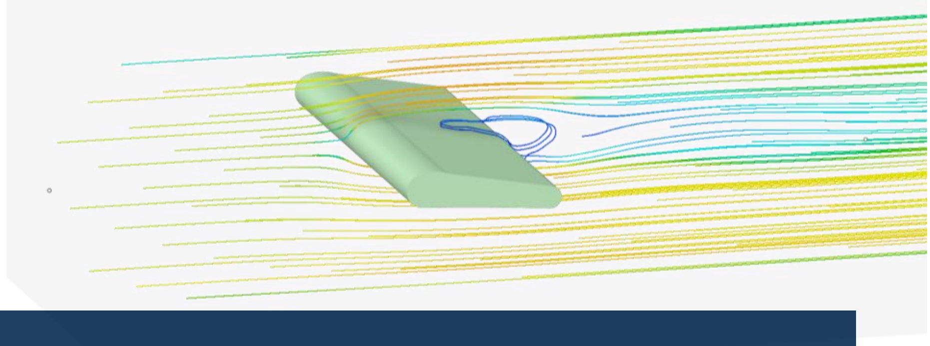
Small Business Service with Single Points of Contact

- Rod’s Cell Phone: 612-819-5288



Who Do We Serve?

Our Customers

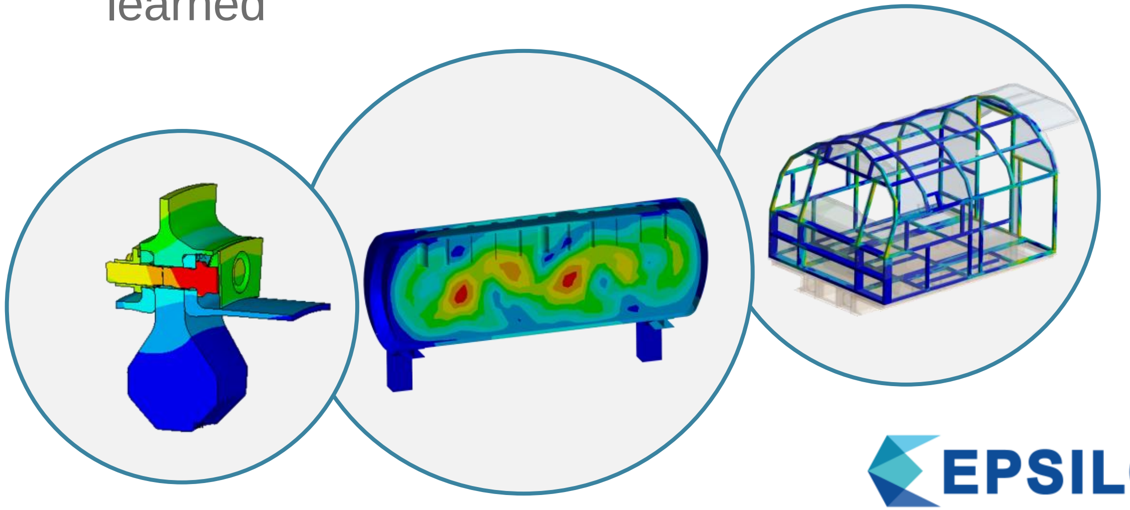


Load-Leveling

- Analyst is a team member, not a black box
  - Interface with same Epsilon analyst to leverage past experience
- Open and frequent communication
- Any new FEA methods/lessons learned are **well communicated**
- Schedule/budget fidelity with frequent status updates
  - Achieved by using the right person, tools, and technical approach

External Expertise

- We infuse up-to-date FEA methods/tools
  - Leverage other industries' FEA innovations
- We are not a software reseller
  - Unbiased tool selection, infrastructure advice
- We share our knowledge, files, and lessons learned



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Epsilon's Customers

Proudly served dozens of companies across numerous industries

BAE SYSTEMS	cannon EQUIPMENT	MTS	ROSEMOUNT	smiths medical	HORTON
American Superconductor	BARR	Donaldson. FILTRATION SOLUTIONS	Collins Aerospace	Panasonic	FLUIDYNE
LORAM	Park Industries	3M	SEAGATE	ZIMMER BIOMET	AMERIPACK
ergotron	ASE	GE Power	Honeywell	SAATI	E-M WEG Group
Calspan FluiDyne	Maxol Studios	United Technologies	VES-Artex a TURNTIDE company	First Solar.	SAFETY POLE
Tolomatic EXCELLENCE IN MOTION	RME	PIPELINE DESIGN & ENGINEERING	PRIDE SOLUTIONS	CREATIVE MACHINES	ORTHOSCAN
PHOENIX Solutions Co	QUALITY TOOL	HYDRA-FLEX FLUID INNOVATION	DANOTEK motion technologies	PADT	tes
CRAFT Tech	EASI	ScooterBug	NARVO ENGINEERING	Advatech Pacific	MODERN MOLDING, INC. CUSTOM INJECTION MOLDING

Epsilon’s Unique Capabilities

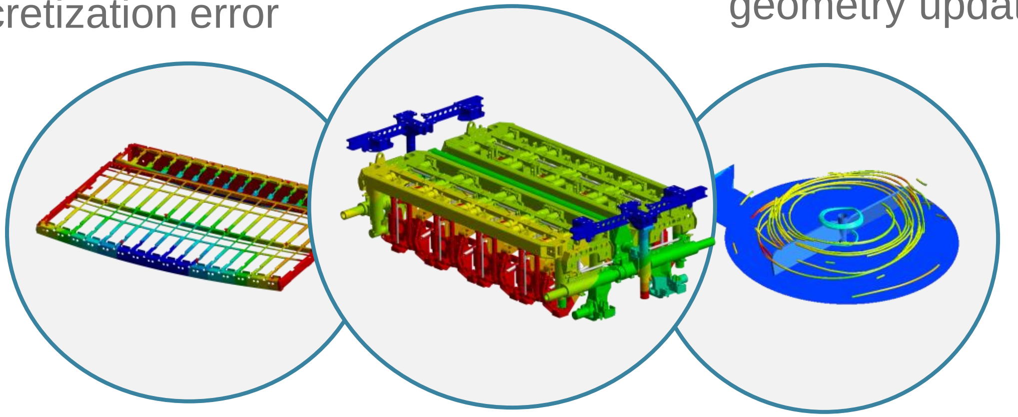
Expertise and Technology to Ensure Accuracy and Affordability

Accurate Simulation

- In-depth knowledge of tools
  - ANSYS® Suite of Multi-Physics software
- Experience with industry successes/failures
  - Aerospace, Rotating Machinery, Electronics, Manufacturing, Packaging, etc.
- Validation with calibration runs and hand-calcs
  - Experience assessing discretization error

Affordable Simulation

- Low hourly rates and/or fixed-price estimates
- Specialized, experienced engineers
- Detailed statements of work, scope, and budget tracking
- Automation (APDL, CAD-associativity)
  - Accommodates shifting inputs, materials, minor geometry updates, etc.



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Agenda, Simulating Bolts

1. Selecting Modelling Approach
   a) Consider Analysis Goals
   b) Beams / Joint methods
   c) Important Concepts
   d) Interface Methods
   e) Decision Matrix
2. Four Case Studies
3. Bolt Strength / Grades
4. Bolt Fatigue Evaluation (Brief)
5. Flange Fatigue Evaluation (Brief)
6. ANSYS Thread Methodology (via Contact)

Consider Analysis Goals

What will be the choices made when you have the results?

- Always weighing risk vs cost of analysis
  - What are the past configuration and failure rates / modes

Analysis Questions	Yes/No
Check for joint sliding / separation?	?
Including Prying Load / Bending?	?
Determine Torque/Pretension?	?
Evaluate bolt strength / fatigue?	?
Evaluate base metal strength / fatigue?	?

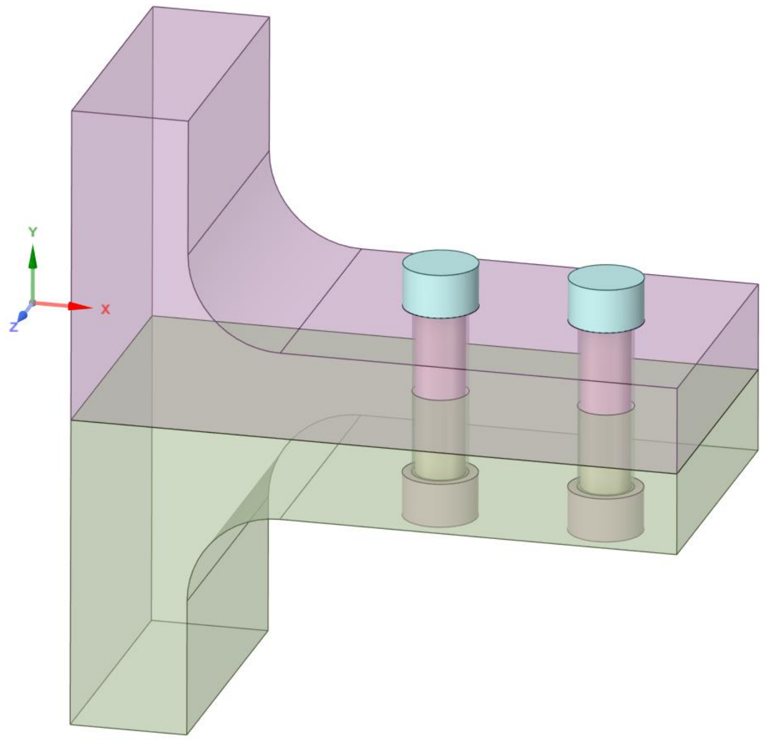
Common Methods / Important Concepts

Four Important Concepts

1. Joint/Beam
2. Pretension
3. Load Sharing
4. Bolts in Bending
5. Bolts Slipping

Three Common Interface Methods

1. Pass-Through Interface (no contact)
2. Bonded Frustum (circular imprint bonded)
3. Full contact (frictional or frictionless)



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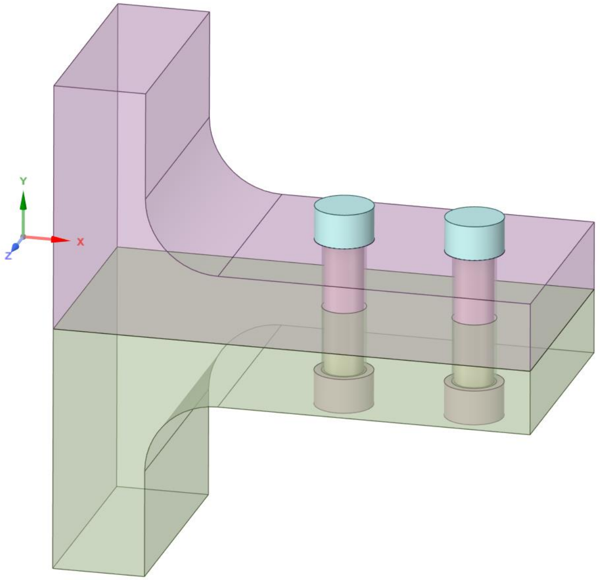
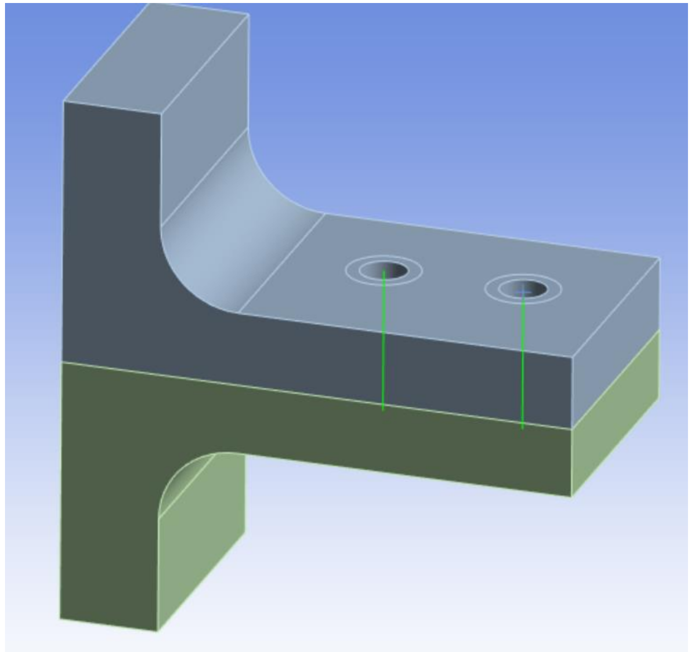
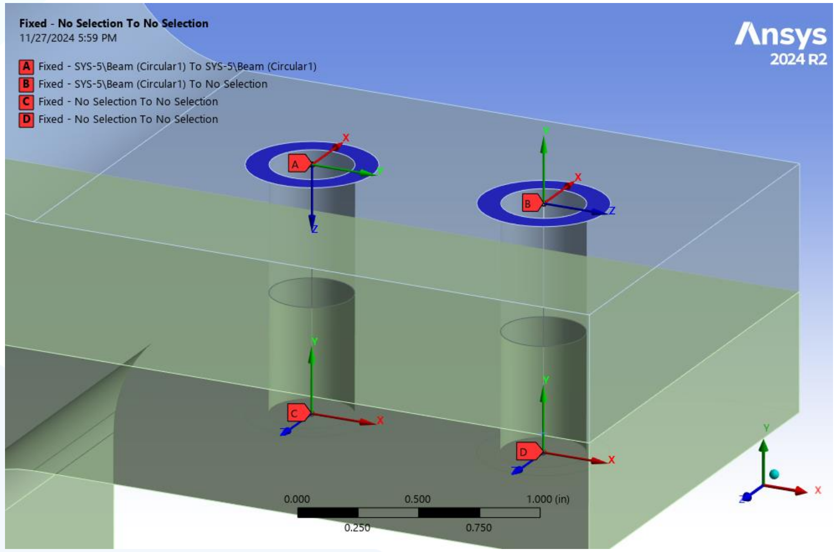
Joint / Beam Methods

Simplest Approach

1. Beam model used in many industries for decades
2. With automation can be fast / inexpensive
3. End join degrees of freedom
   - UXYZ only (ignores bending resistance)
     - Often conservative for tension
   - UXYZ and ROTXYZ
     - Often anti-conservative for tension







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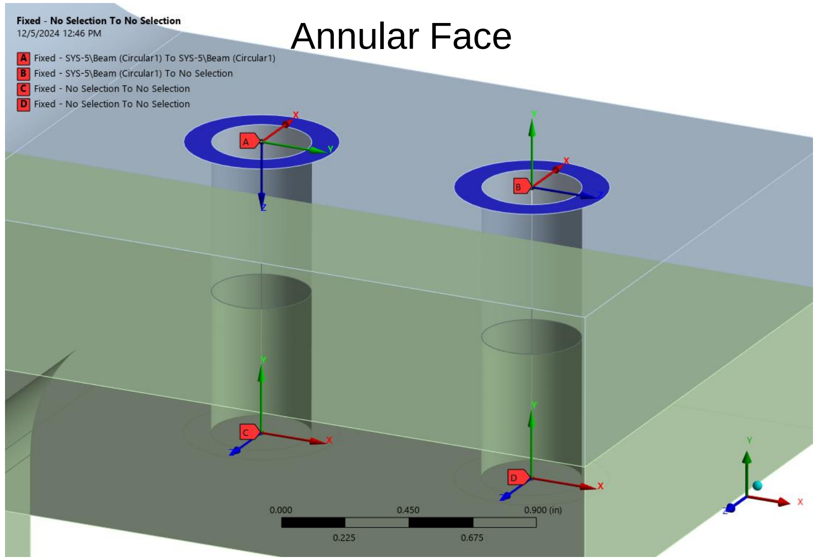
Joint / Beam Methods

End Connections

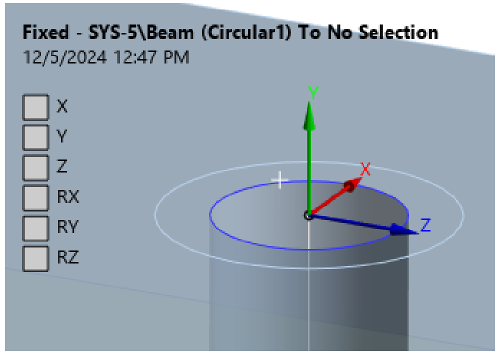
Methods apply to line bodies or solid body bolts

- Choices at the bolt head / nut / threads
  a) Connect to annular ring
    - CE's, Joint, Beams/"Wagon Wheel"
    - Rigid or deformable?
      – (deformable is usually conservative for bolt stress)
  b) Connect to edge
    - High singularity stresses
      – Unless connecting to shell elements
    - Same methods as annual ring available
  c) Connect to threaded hole
    - Best practice for engagement distance:
      – Three (3) threads (steel to steel)
      – Five (5) threads (steel to aluminum)

Annular Face



Edge Connection



Size of Ring? Various Methods

- 1.5XBolt Hole Diameter
- Bolt Head Imprint
- Washer Imprint
- Frustum cone projection from bolt head

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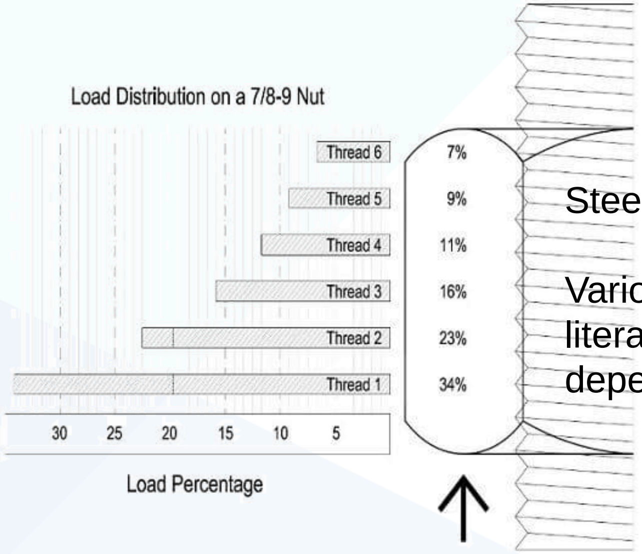
Joint / Beam Methods

End Connections

Solid images throughout slides do not show the washer, but consider including its geometry

- Choices at threaded hole
  a) First few threads bear most of load
  b) Best practice for engagement distance:
     - Three (3) threads (steel to steel)
     - Five (5) threads (steel to aluminum)

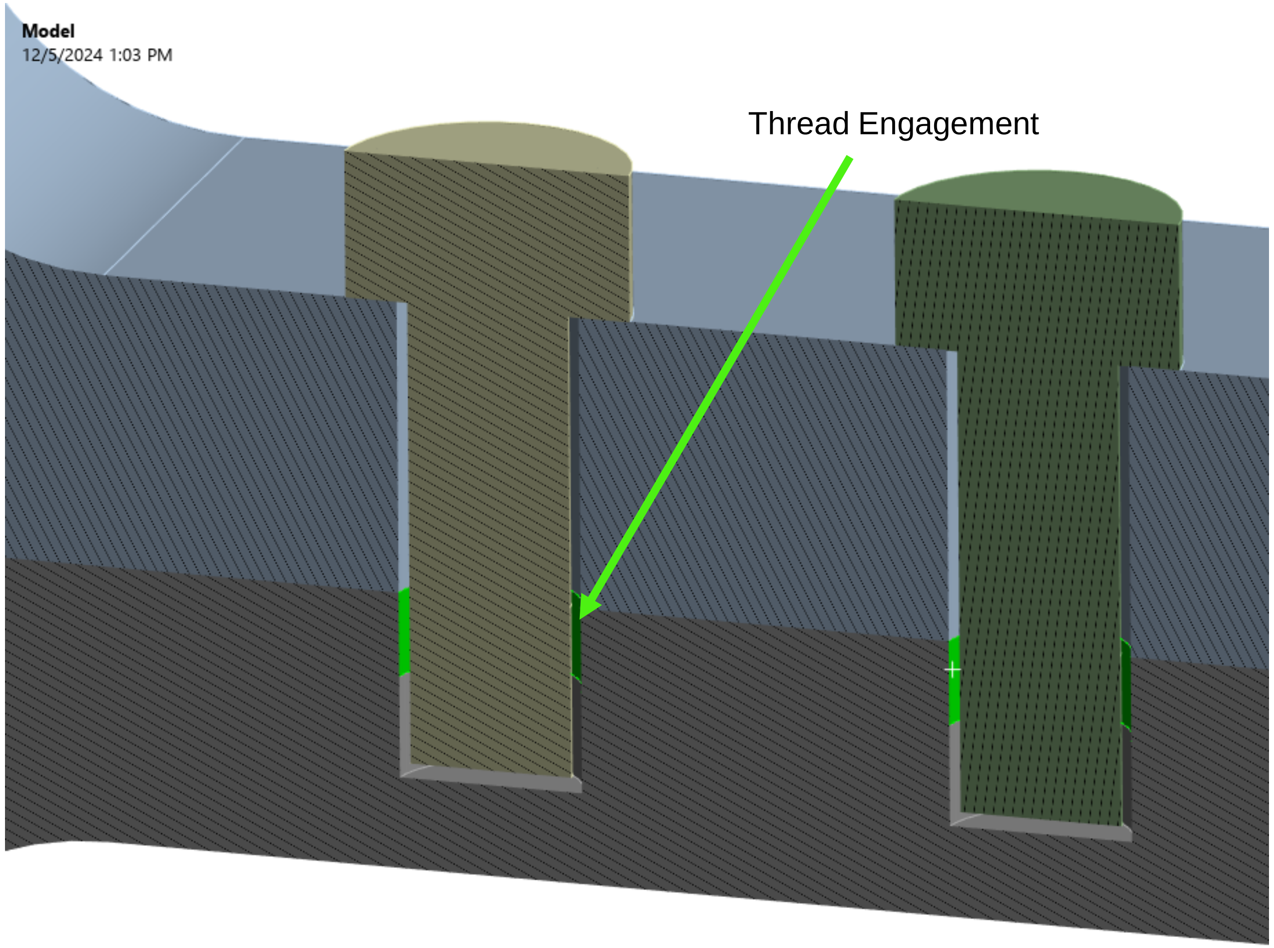


load distribution source: corrosionpedia.com

figure: Thread

Steel/Steel

Various distributions in literature, and configuration-dependent



Accuracy of stresses at the first thread is minimal given approximation (potential geometric singularity)

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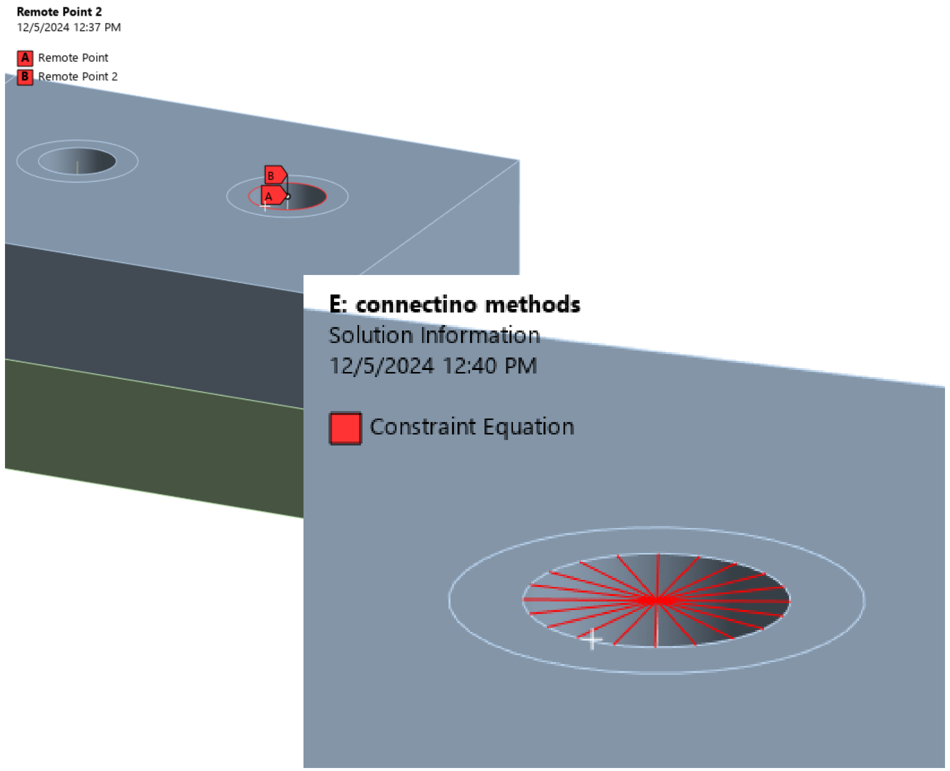
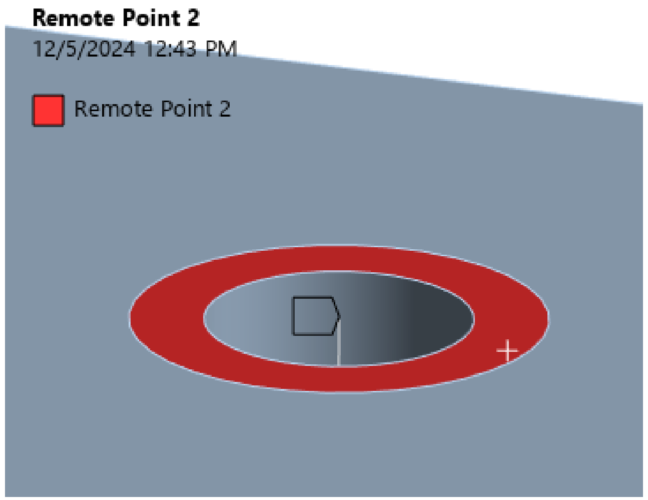
Joint / Beam Methods

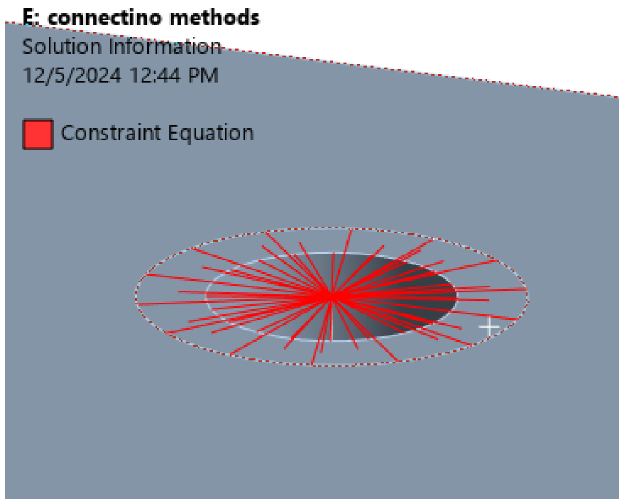
Remote Points



1. Avoids Joints/Contact
   a) Requires “remote point” within WB Mechanical
2. Other Methods used historically
   a) Can use constraint equations (CE’s)
      - Common practice historically
      - Can use “deformable” option to implement RBE3’s
   b) Can also use beams
      - Some legacy approaches still used
      - Will need guidance on beam stiffness

Coefficient	Units	Remote Point	DOF Selection
-1	1/in	Remote Point	X Displacement
1	1/in	Remote Point 2	X Displacement
-1	1/in	Remote Point	Y Displacement
1	1/in	Remote Point 2	Y Displacement
-1	1/in	Remote Point	Z Displacement
0	1/in	Remote Point 2	Z Displacement



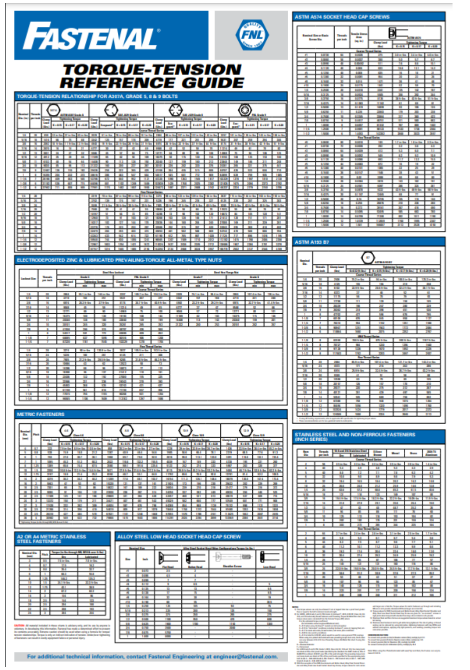


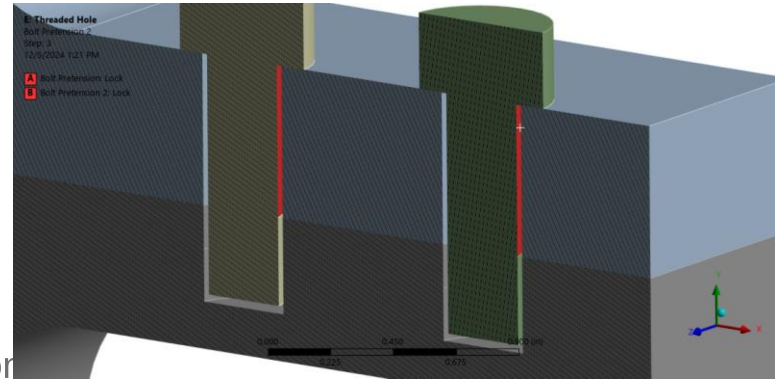
Bolted Pretension

Initial Applied Load

1. Can convert a torque to a tension force
   a) Torque =Force*K*D (more complex relations exist)
      - Note lubricated creates higher tension at same torque
   b) Consider thermal growth (different CTE's)
   c) Consider variation in installation torque
      - Depends on installation device
      - Sandia advice, shown Right
      - NASA has advice in NASA-STD-5020B for various conditions
        – Up to 35% for non-lubricated with a torque wrench!
2. In ANSYS Mechanical "Lock" the installation torque in second load step
3. Look for errors in applied pretension direction!
   - WB Mechanical sometimes has trouble finding the axis of preload for a geometry selection. Check coordinate system orientation for each bolt. Given rotations of CS's… you may still have to solve to verify it's working as expected.

K Factors	
Bolt Condition	**K**
Non-Plated, black finish (dry)	0.20 - 0.30
Zinc-Plated	0.17- 0.22
Lubricated	0.12 - 0.16
Cadmium-plated	0.11-0.15





SANDIA REPORT
SAND2008-0371
Unlimited Release
Printed January 2008

Guideline for Bolted Joint Design and Analysis: Version 1.0

Kevin H. Brown, Charles Morrow, Samuel Durbin, and Allen Baca

Table 3: Accuracy of Bolt Preload Based on Application Method

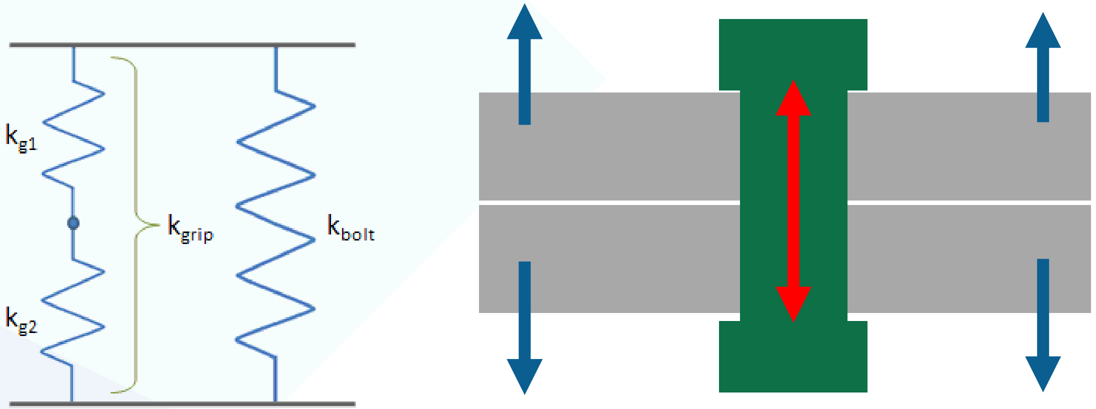
Method	Accuracy
Torque Wrench on Unlubricated Bolts [11]	±35%
Torque Wrench on Cad-Plated Bolts [11]	±30%
Torque Wrench on Lubricated Bolts [11]	±25%
Preload Indicating Washer [11]	±10%
Strain Gages[12]	±1%
Computer Controlled Wrench (Below Yield) [12]	±15%
Computer Controlled Wrench (Yield Sensing) [12]	±8%
Bolt Elongation [11]	±5%
Ultrasonic Sensing [11]	±5%

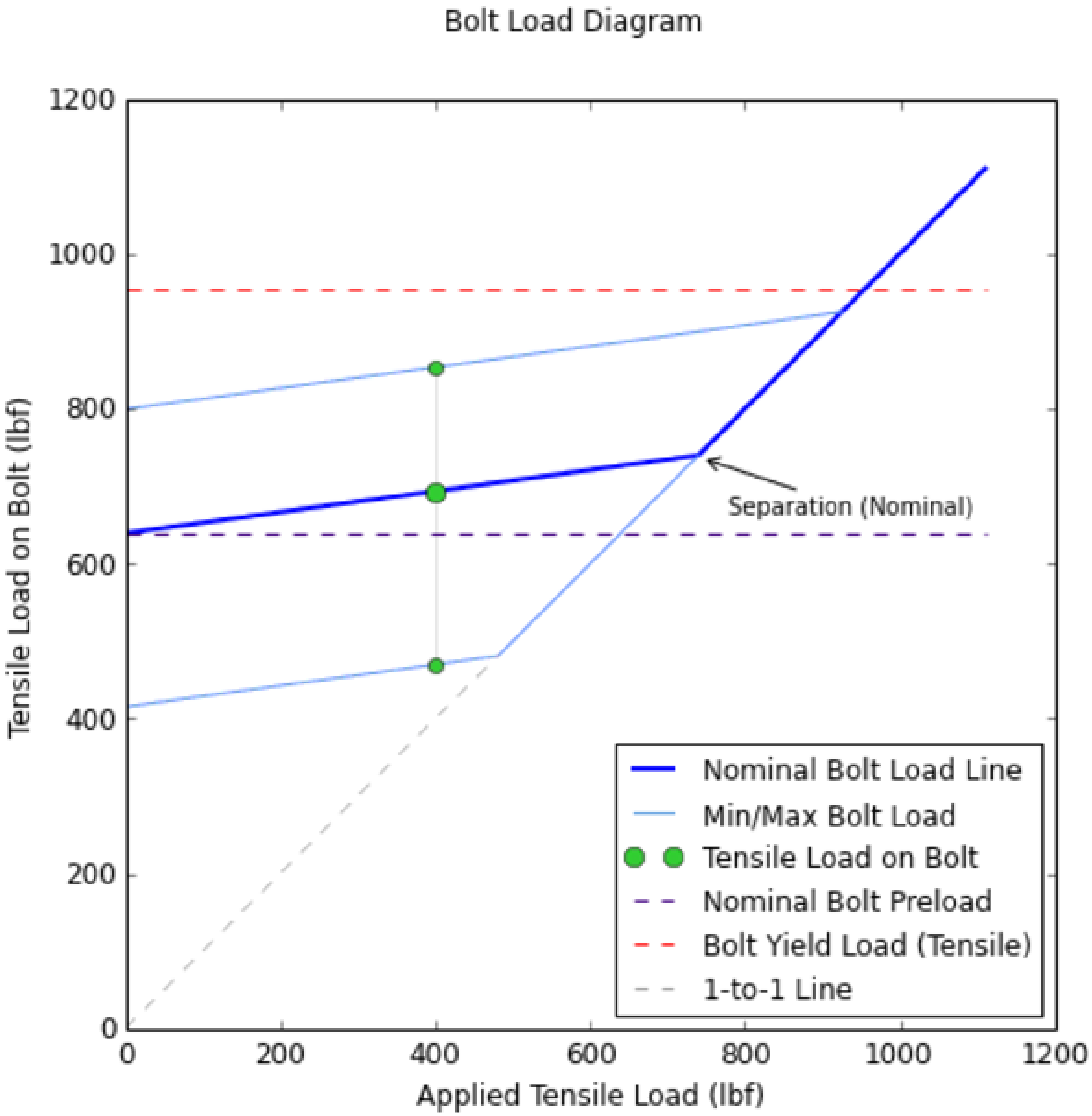
	Steps	Define By	Preload [lbf]	Preadjustment [in]	Increment [in]
1	1.	Load	20000	N/A	N/A
2	2.	Lock	N/A	N/A	N/A
3	3.	Lock	N/A	N/A	N/A
*					

Bolted Interface Load Sharing

Parallel Load Paths

Become familiar with loading charts similar to these



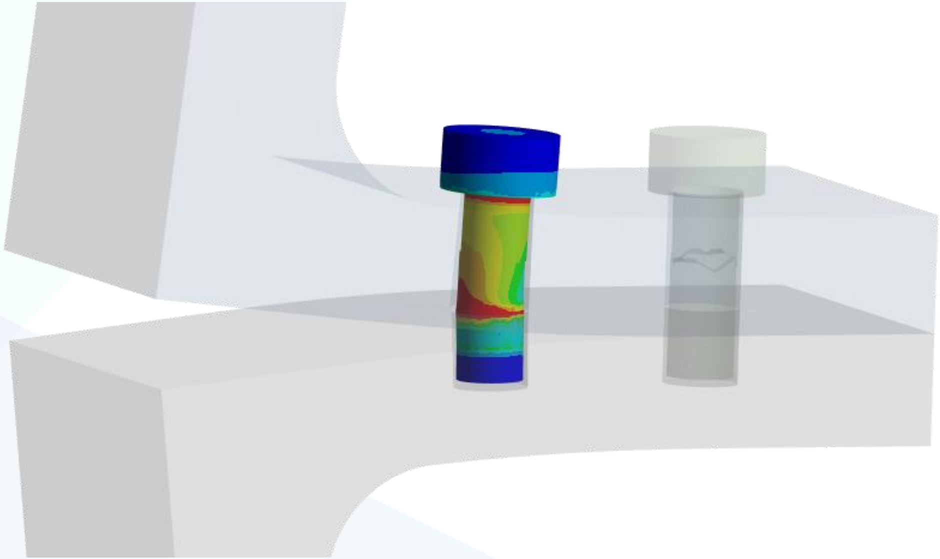


https://www.mechanical.com/reference/bolted-joint-analysis

Bolts in Bending

Moments

- Can be predicted depending on model setup
  a) Take peak stress as "bolt stress" in fatigue
  b) Usually, the stress under the bolt head is lower than the shank/threads
     - If this fillet region has high stress, include a KT if not in the geometry! (e.g. 2.1 to 2.3 – Shigley)
  c) Ideally bending occurs in shank, away from stress concentrations
     - Best to avoid bending at threads! (because KT is highest here)



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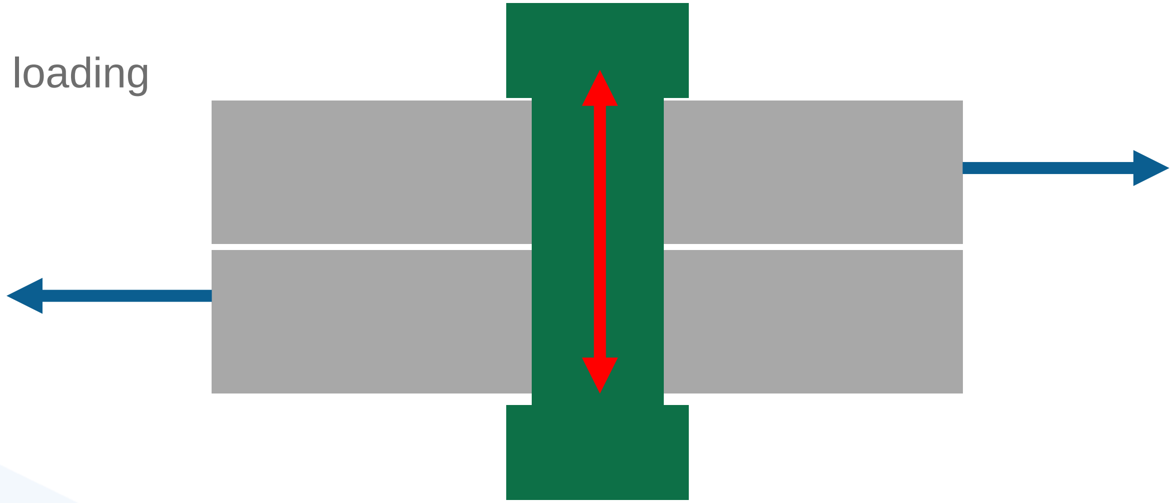
Bolts Slippage / Shear

Often done with Hand Calc's

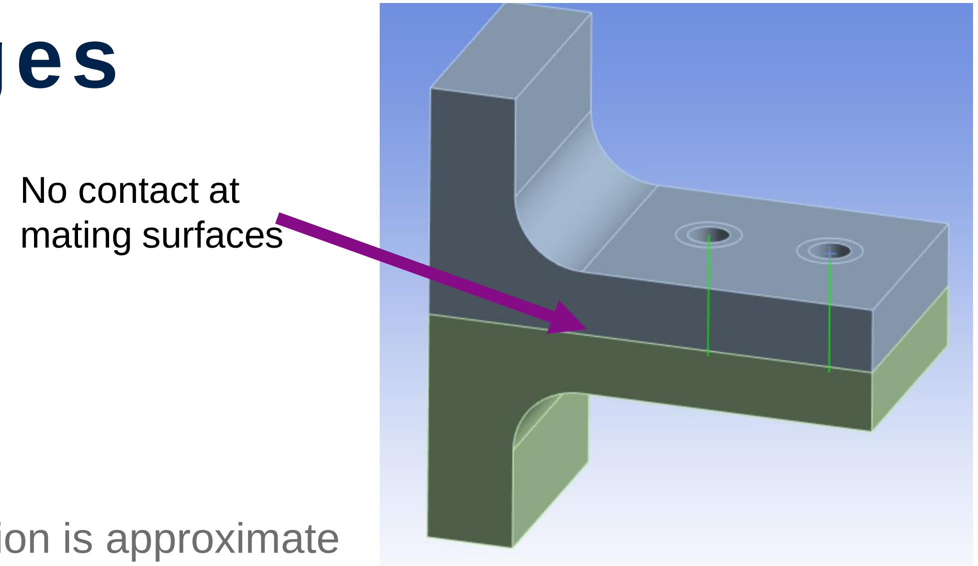
1. See common sources on shear strength of bolts
   a) Don't forget to evaluate shear+tensile when doing hand-calculations as both are present together!
2. Slippage calculation will be driven by coefficient of friction (COF)
   a) Do not be anti-conservative! Consider including presence of contaminates / oil
   b) If slippage is acceptable, and bolts are loaded in shear…
      - Consider tolerancing vs. yielding… will just one bolt bear all the shear load, or will yielding distribute it to others?
      - Consider wear / erosion
      - Consider fatigue at edge loading

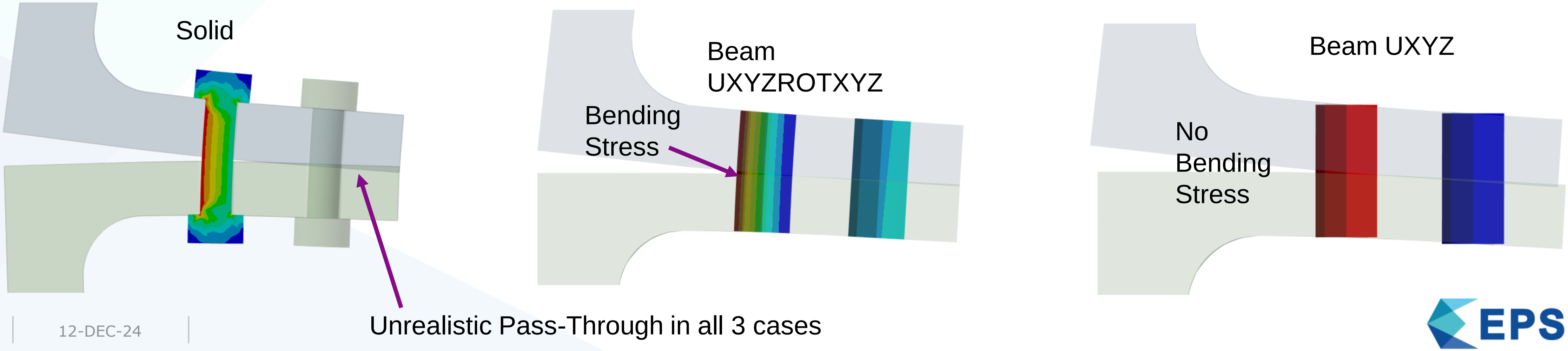


Interface: No Contact at Flanges

Bolt-Only connections

- No Contact at Flanges
  a) Relatively Rare in FEA
  b) Close to bolt load hand-calc's
    - If ROTXYZ ignored, can be very close to hand-calculations
  c) Accuracy is limited because moments are approximate, and so tension is approximate
    - **Attractive in cases with double rows of bolts (or circular pattern) where bolt bending is very low**
  d) No pretension (use linear superposition to include preload)
  e) Some design systems use this method for evaluating bolts (as conservative) and then re-analyze with another method for base metal.





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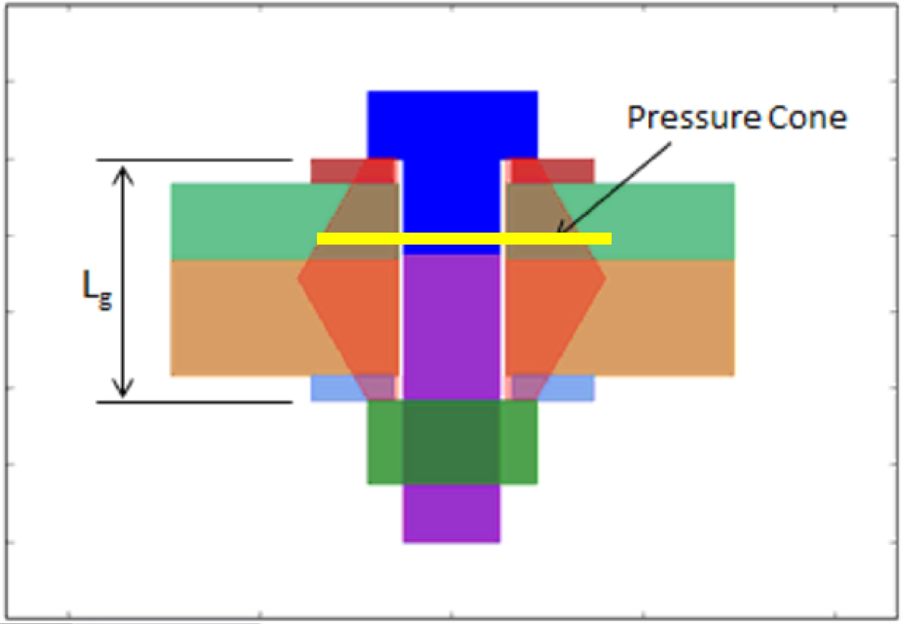
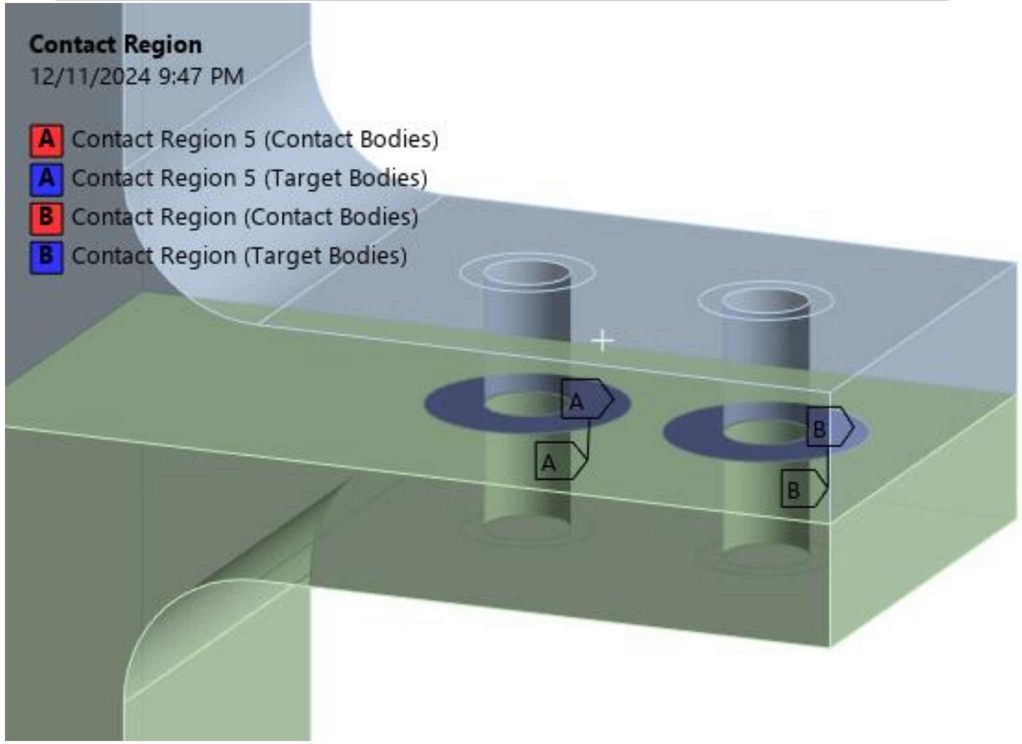
Interface: The Frustum Cone

Load Path Through Joint

- Load Path of Compressed Flanges
  a) Majority of load is localized under the bolt head
  b) Shigley provides simplification of a frustum cone
    - Drawn from bolt head (includes washer thickness)
  c) Annular ring where majority of load passes
    - Yellow Line
    - A simplified approach is to bond the annulus while letting the rest of the flanges separate or inter-penetrate
  d) Angle of the cone is commonly chosen at 30°
    - Consider modifying this angle for cases with different bolt/flange material

Bolt With Nut

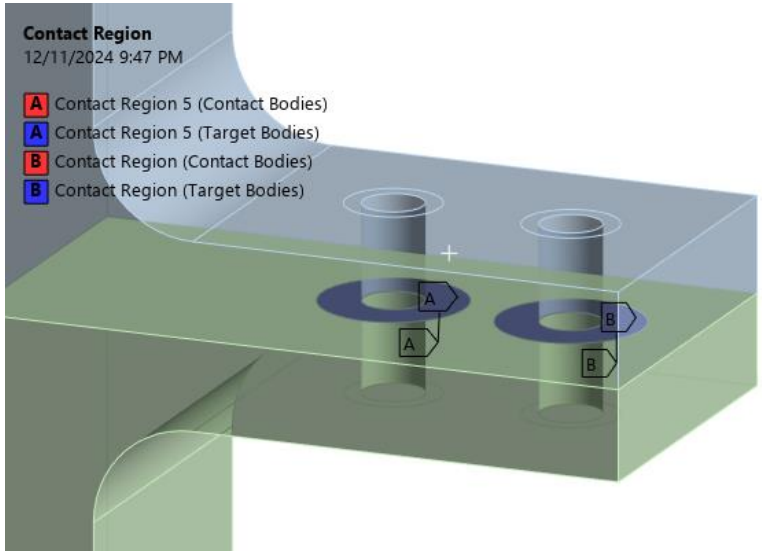


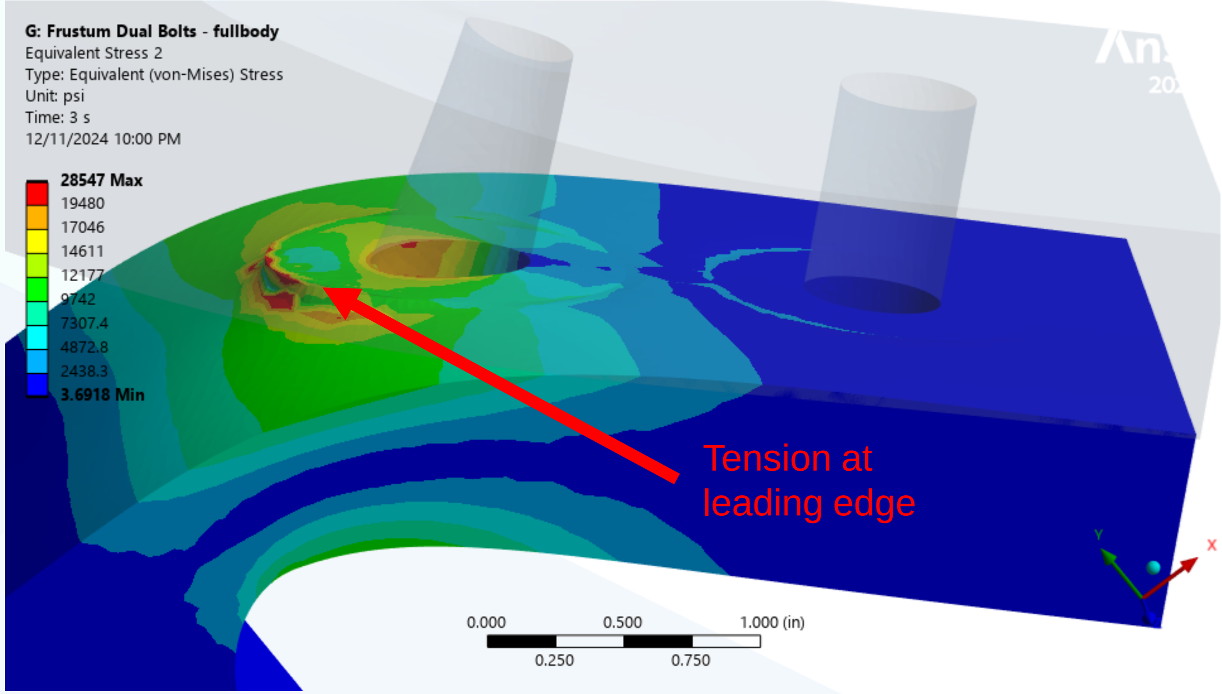


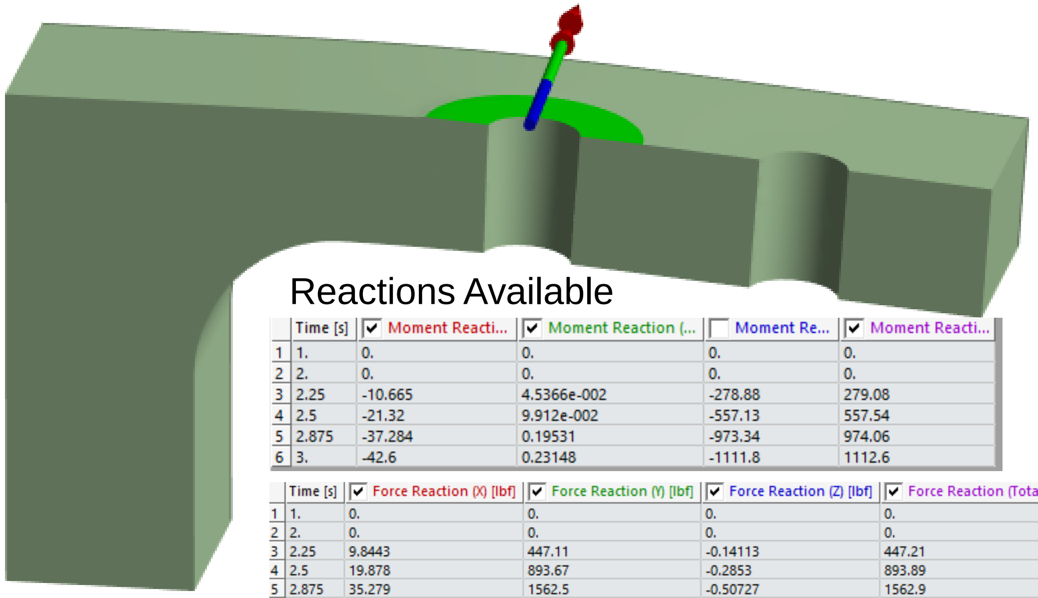
Interface: Bonded Frustums

Easiest to implement

- Bonded Contact over Annular Ring
  a) Allows separation / pass-through at rest of flange
  b) Ring size based on frustum cone
  c) No sense in adding a bolt (beam or solid) or pretension
  d) Still possible to get tensile/moment loads (not with MPC)
    - But accuracy is limited because moments are approximate, and so tension is approximate







Reactions Available

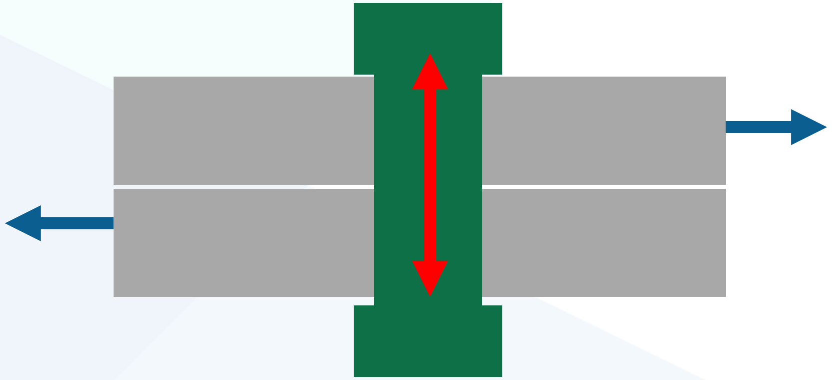
	Time [s]	Moment Reacti...	Moment Reaction (...	Moment Re...	Moment Reacti...
1	1.	0.	0.	0.	0.
2	2.	0.	0.	0.	0.
3	2.25	-10.665	4.5366e-002	-278.88	279.08
4	2.5	-21.32	9.912e-002	-557.13	557.54
5	2.875	-37.284	0.19531	-973.34	974.06
6	3.	-42.6	0.23148	-1111.8	1112.6

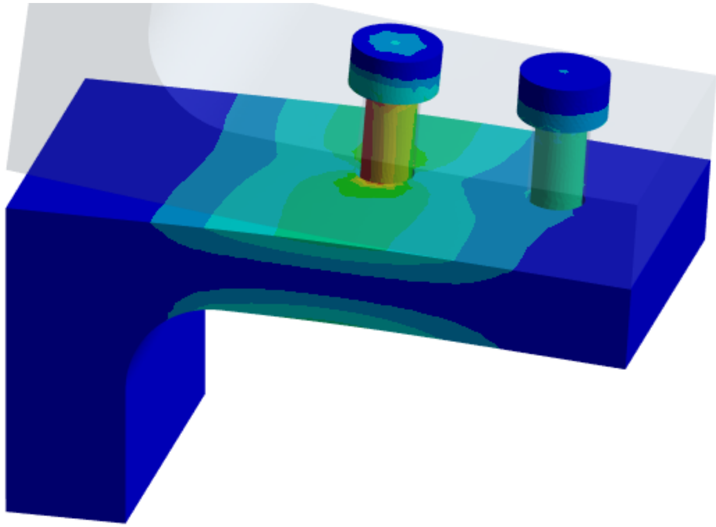
	Time [s]	Force Reaction (X) [lbf]	Force Reaction (Y) [lbf]	Force Reaction (Z) [lbf]	Force Reaction (Total) [lbf]
1	1.	0.	0.	0.	0.
2	2.	0.	0.	0.	0.
3	2.25	9.8443	447.11	-0.14113	447.21
4	2.5	19.878	893.67	-0.2853	893.89
5	2.875	35.279	1562.5	-0.50727	1562.9
6	3.	40.503	1785.2	-0.58272	1785.6

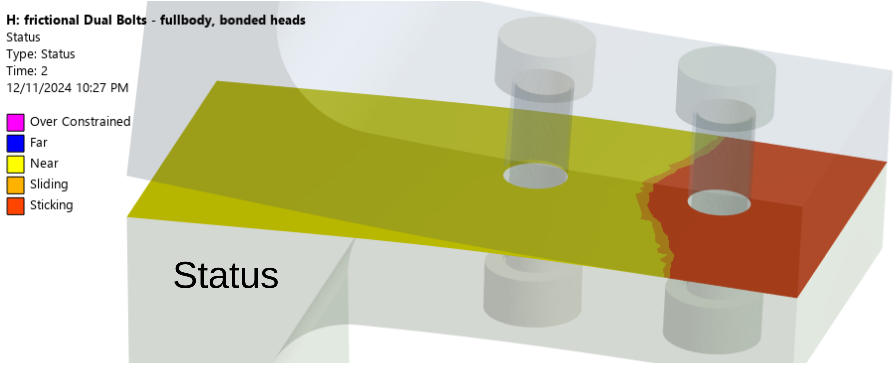
Interface: Full Contact

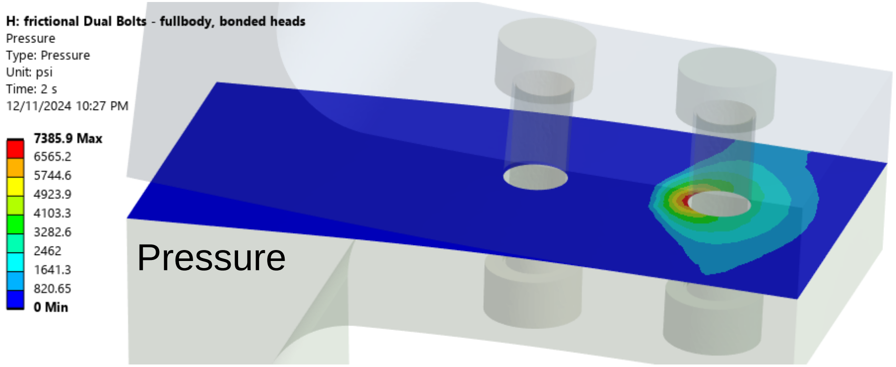
Most Accurate

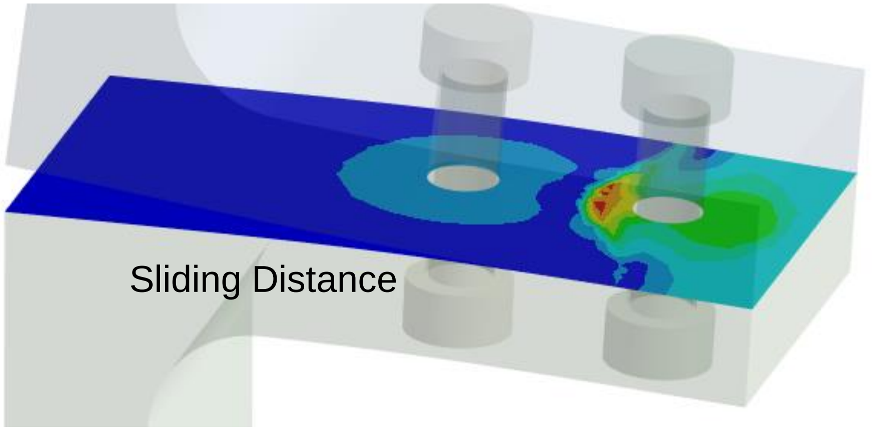
- Can use full frictional contact
  a) Long solution times
  b) Coefficient of Friction (COF) necessary
    - Usually difficult to know reliably
    - Often requires checking high/low (COF)
  c) Consider bonding the nut for stability
    - In some cases the bolt head and nut won't both slip simultaneously!











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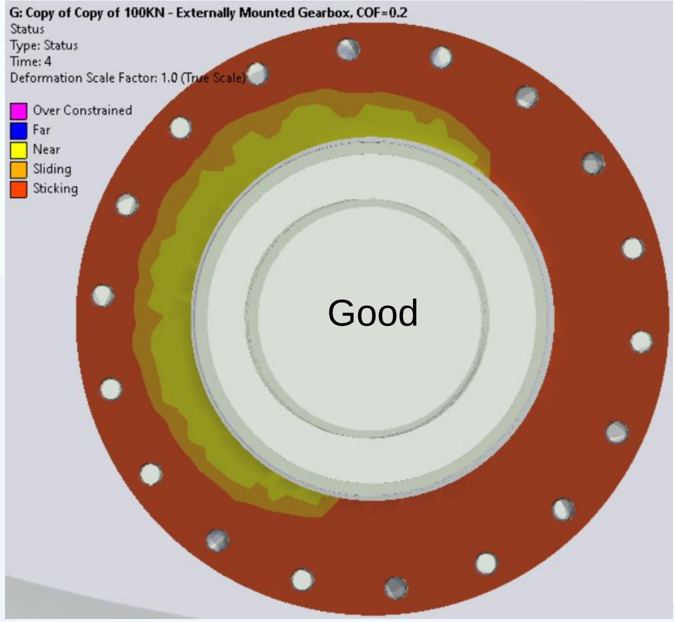
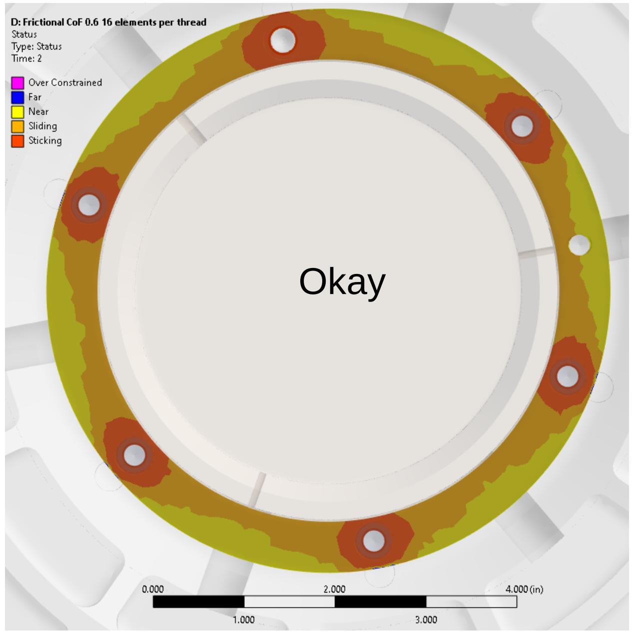
Interface: Full Contact

Local Slippage on Highly Loaded Joints

1. Sliding Contact means wear, and loss of preload
   - Wear accelerates loss of preload as sticking region shrinks!
2. Look for sticking contact in full annular ring around each bolt
   - Still slightly anti-conservative if wearing occurs on sliding surface

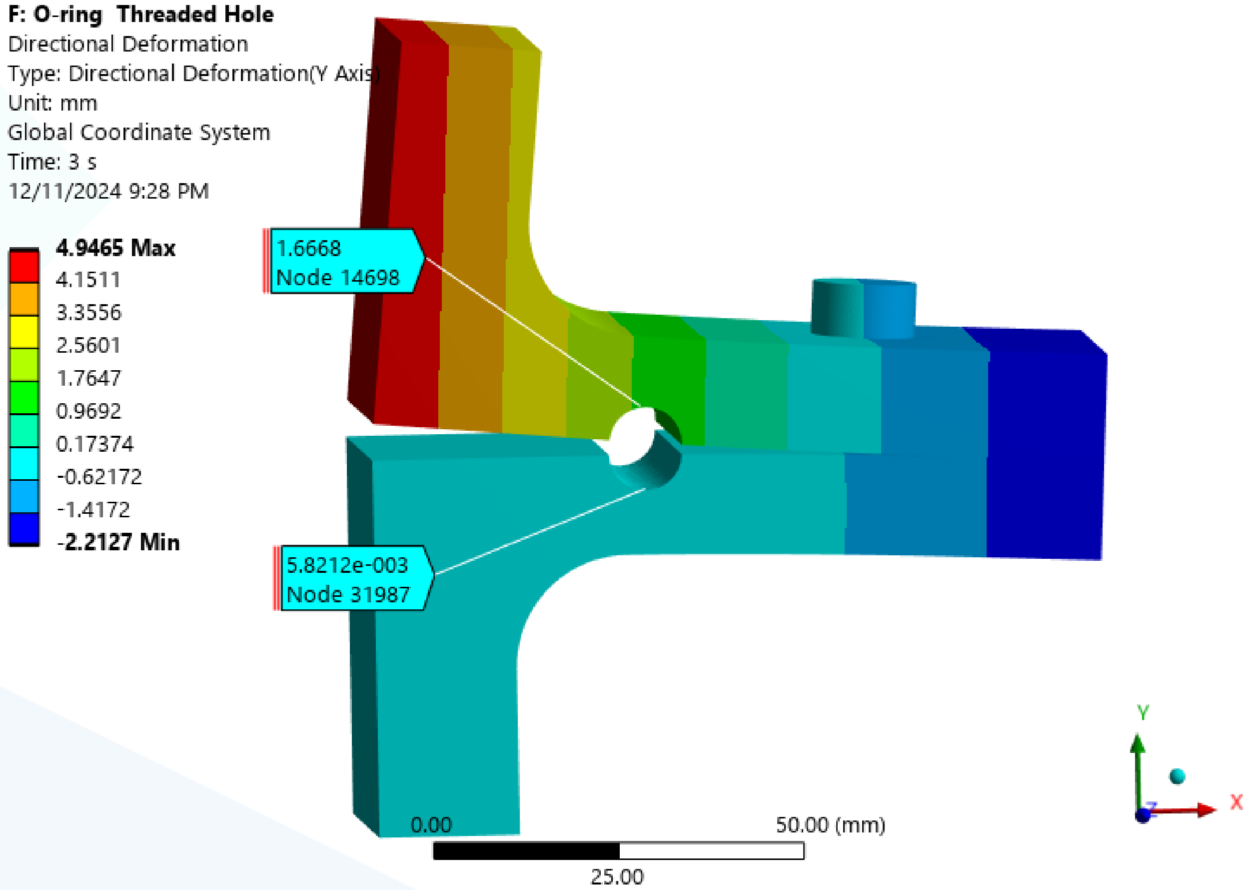




Interface: Full Contact

O-ring / Gasket Flange Sealing

- If verifying O-ring sealing… can use full contact for highest accuracy, but other methods could be conservative if maximum gap is below the compression distance



Selecting Approach at Flange Interface

Decision Matrix if using FEA Methods**

Analysis Questions	Beam/Joint	Bonded Frustum	Full Contact with Pretension
Check for joint sliding/separation?	Hand Calc's	Hand Calc's	Yes
Including Prying Load / Bending?	No	No	Yes
Determine Torque/Pretension?	Hand Calc's, but ignores bolt bending*	Hand Calc's, but ignores bolt bending*	Yes
Evaluate bolt strength / fatigue?	Hand Calc's, but ignores bolt bending*	Hand Calc's, but ignores bolt bending*	Yes
Evaluate base metal strength/ fatigue?	NA	No	Yes

*Or if ROTXYZ is included, can include too much/little bolt bending, and thus tension has error

**Many Empirical Methods exist for all of these (a lot of things built without doing FEA)!

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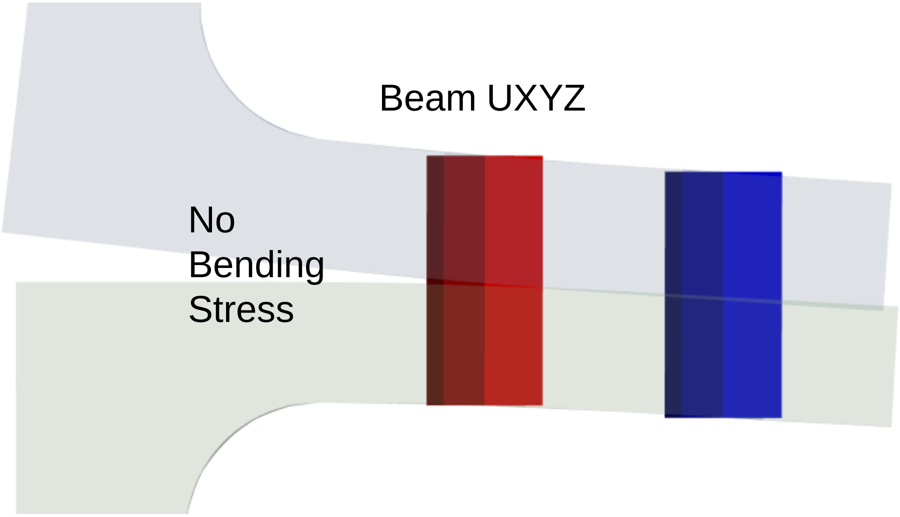
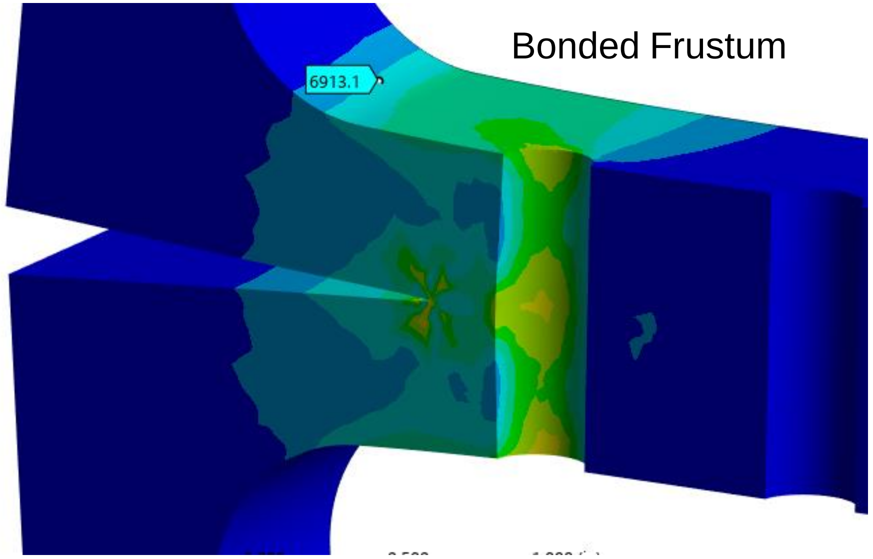
Case Study 1 – Flange in Low Bending

1. Assume no flange separation
2. Assume no bolt bending stress
3. Investigating which Failure Point?:
   a) **Bolts:** Use no interface, floating bolt approach
      - Many steel/steel standard designs will fail in bolts before flange; so if the bolt is okay, the flange hole/threads are okay!
      - With double rows of bolts (or circular pattern often check UXYZ is okay
      - If single row of bolts, and very low moments, use UXYZ+ROTYXZ
      - Include pretension in hand-calculations
   b) **Flanges:** Bonded Frustum
      - Stresses at holes are unknown!
      - Reaction moments/tensile loads from contact elements may not be that accurate if bending is significant… but sometimes these reactions can be used to prove bolt survival if large margin





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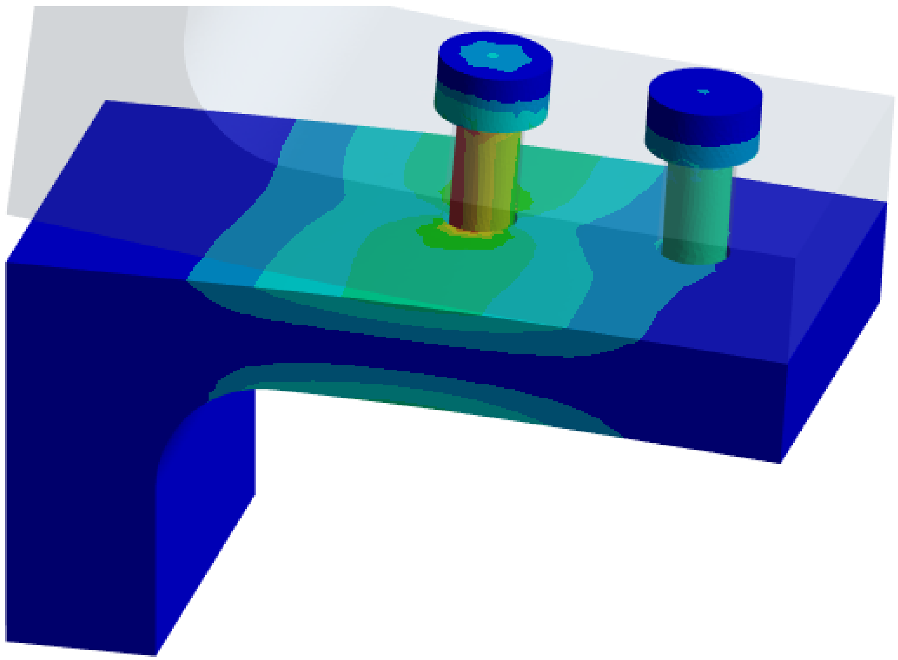
Case Study 2 – Flange in High Bending

1. Use full contact with friction
   a) May be possible to identify critical bolts to limit regions of full contact
      - Solve first with bonded contact, to verify model
2. Consider checking range of COF’s
   a) Sometimes its unknown whether high or low COF is conservative
3. Check bending stress in bolt
   a) Consider Kt’s at threads, under bolt head, etc.
   b) If doing fatigue run installation load, and other cases too
      - Don’t forget Mean Stress Correction / Goodman!



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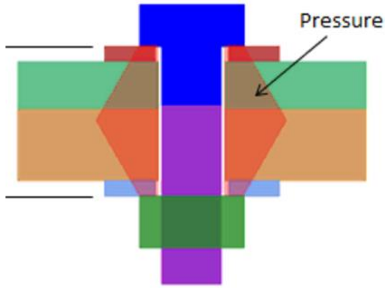
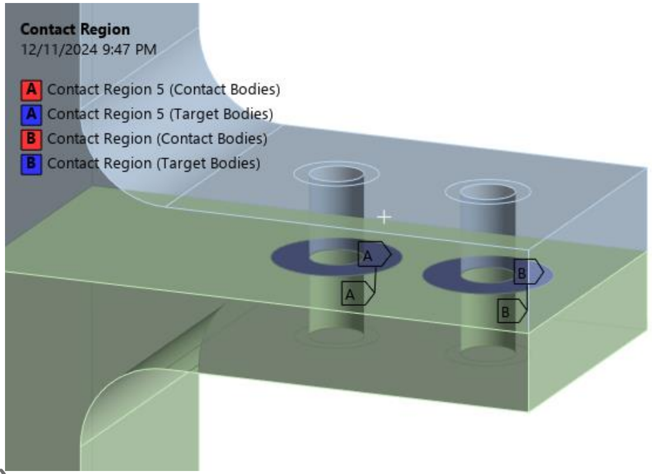
Case Study 3 – Bolted Joints in Vibration



1. Want to run linear in most cases.
   a) Transient dynamics is expensive!
2. Use bonded frustum
   a) Check reaction loads to be sure vibration stress range is very low compare to fatigue allowables
3. Choose appropriate frustum angle
   a) 30° is typical still
   b) Very low excitation could be closer to 45°… or 70°
   c) Very high excitation could cause flange separation!
   d) Sometimes a static equivalent acceleration case (at peak resonance levels) with full contact and pretension can determine the bonded contact annulus size
4. Check stress in bolt
   a) Consider Kt's from threads, under bolt head, etc.
   b) If doing fatigue don't forget Mean Stress Correction / Goodman!



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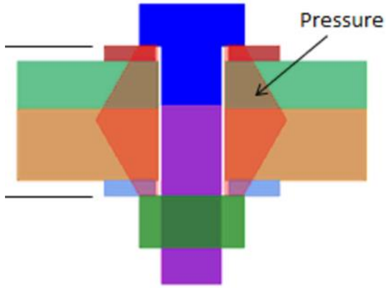
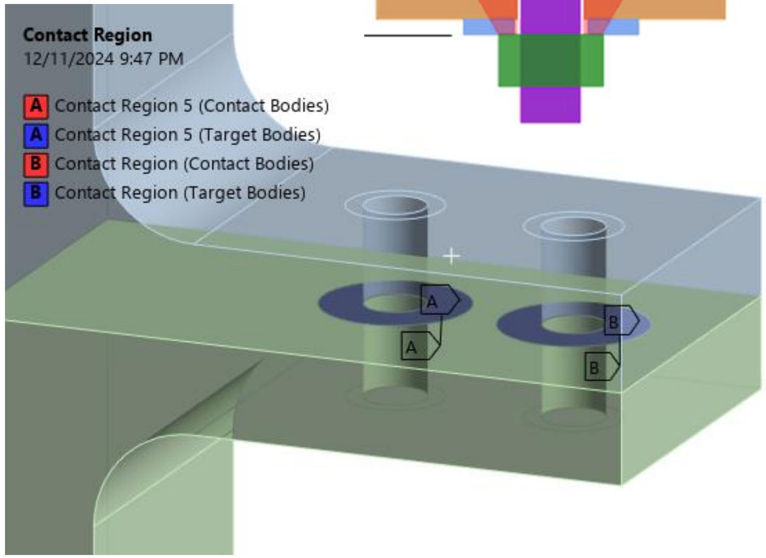
Case Study 4 – Bolts in Shock Loading



1. Want to run linear in most cases. (unless it is static equivalent shock)
   a) Transient dynamics is expensive!
2. If evaluating bolts, use floating interface
   a) Unless it is a single row of bolts (i.e. moments)
3. If evaluating base metal, use bonded frustum
   a) Most information on bolts is lost
4. Choose appropriate frustum angle
   a) Check for flange separation! (i.e. DDAM shocks are very high!)
   b) Sometimes a static equivalent acceleration case (with conservative shock amplification factor) and including full contact and pretension can determine the bonded contact annulus size if static equivalent loading is not acceptable
5. Check stress in bolt
   a) Consider rate dependent plasticity/strength if very low shock duration



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Bolt Strength / Grades

Usually Done with Hand-Calculations

- Once we have predicted stress we can evaluate the bolts
  a) Don’t forget to combine axial, shear, bending as appropriate
  b) If fatigue, include bending stress. If strength, verify peak stress is below ultimate
  c) Don’t forget bolt preload
  d) Fastenal is a great resource on bolts. But there are many, including Shigley

ASTM, SAE AND ISO GRAD MARKINGS AND MECHANICAL PROPERTIES FOR STEEL FASTENERS

Grade Marking	Specification	Material	Bolt and Screw Size in.	Proof Load. psi	Tensile Strength min.,psi
	SAE-Grade 1	Low Carbon Steel	1/4 thru 1-1/2	33,000	60,000
	ASTM-A 307	Low Carbon Steel	1/4 thru 1-1/2 Over 1-1/2 thru 4	33,000	60,000 55,000
	SAE-Grade 2	Low Carbon Steel	1/4 thru 3/4 Over 3/4 thru 1-1/2	55,000 33,000	74,000 60,000
	SAE-Grade 5	Medium Carbon Steel, Quenched and Tempered	1/4 thru 1 Over 1 thru 1-1/2	85,000 74,000	120,000 105,000
	ASTM-A 449	Medium Carbon Steel, Quenched and Tempered	1/4 thru 1 Over 1 thru 1-1/2 Over 1-1/2 thru 3	85,000 74,000 55,000	120,000 105,000 90,000
A325	ASTM-A 325	Medium Carbon Steel Quenched and Tempered	1/2, 5/8, 3/4 7/8, 1 1-1/8 thru 1-1/2	85,000 78,000 74,000	120,000 115,000 105,000
	SAE-Grade 8	Medium Carbon Alloy Steel, Quenched and Tempered	1/4 thru 1-1/2	120,000	150,000
	ASTM-A 354 Grade BD	Alloy Steel, Quenched and Tempered	1/4 thru 1-1/2	120,000	150,000
A490	ASTM-A 490	Alloy Steel, Quenched and Tempered	1/2 thru 2-1/2 Over 2-1/2 thru 4	120,000 105,000	150,000 140,000

ASTM Specifications
A 307- Low Carbon Steel Externally and Internally Threaded Standard Fasteners
A 325- High Strength Steel Bolts for Structural Steel Joints, Including Suitable Nuts and Plain Hardened Washers.
A 449 - Quenched and Tempered Steel Bolts and Studs
A 490 - High Strength Alloy Steel Bolts for Structural Steel Joints, Including Suitable Nuts and Plain Hardened Washers.

For reference only, do not use for design

Also, consider checking for **thread shearing** to ensure adequate thread strength. Typically, bolts are designed to fail in the minor diameter, and not strip threads off, and threaded holes have larger thread area (but if lower strength metal they also need checking.

There are hand-calculations to check this.

If bending is present, it is conservative to use the load / stress determined from the peak bending stress near the threaded region.

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Bolt Fatigue (Brief)

Using Standards

1. One can spend a lifetime learning to predict fatigue failure
2. Standards (ASME BPVC VIII, BS-7608, etc.) will provide allowable alternating stresses for different bolt materials
   a) Often assumes some standard assembly preload
   b) Don’t forget to use the peak stress (include bending stress)!
   c) Some standards provide stress vs. cycle count predictions
   d) Often fairly conservative because margin added for many variables
      - Temperature range, corrosion, variations in torque, tolerancing, safety factors

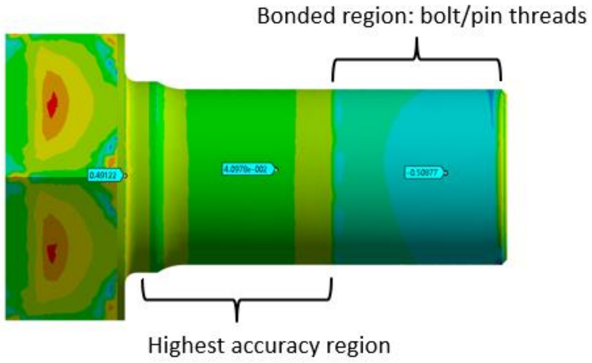
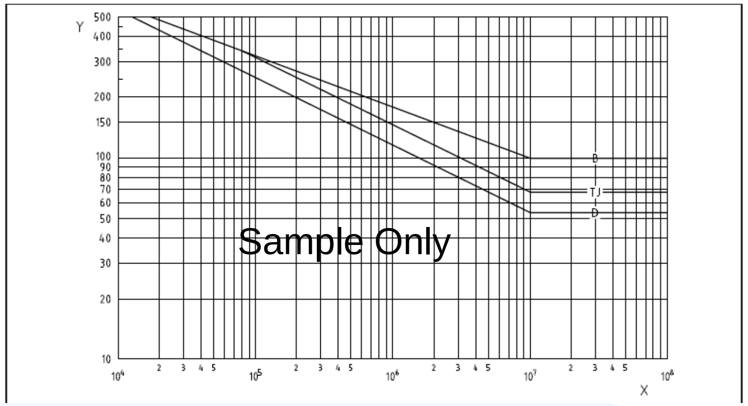
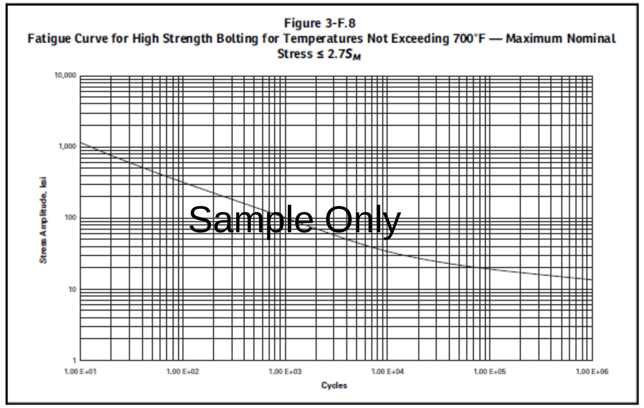


Figure 10 Standard basic design S_r-N curves

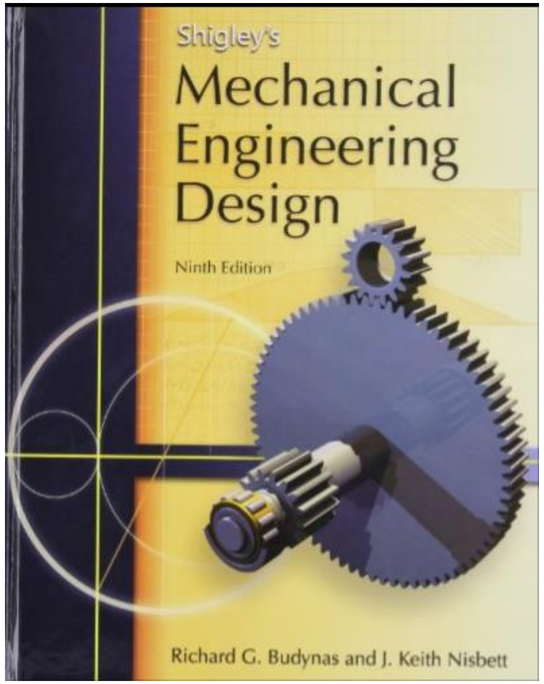




Bolt Fatigue (Brief)

Using Shigley

3. For evaluation to endurance limit, Shigley is an excellent resource
    a) Provides ranges for typical bolt grades and sizes
    b) Don’t forget goodman / mean stress correction
    c) Kt’s are provided
        - Different for rolled vs. cut threads (dependent on bolt grade)
        - Fillet Kt (under bolt head) is given as 2.1 to 2.3 (dependent on bolt grade)



Grade or Class	Size Range	Endurance Strength
SAE 5	$\frac{1}{4}$–1 in	18.6 kpsi
	$1\frac{1}{8}$–$1\frac{1}{2}$ in	16.3 kpsi
SAE 7	$\frac{1}{4}$–$1\frac{1}{2}$ in	20.6 kpsi
SAE 8	$\frac{1}{4}$–$1\frac{1}{2}$ in	23.2 kpsi
ISO 8.8	M16–M36	129 MPa
ISO 9.8	M1.6–M16	140 MPa
ISO 10.9	M5–M36	162 MPa
ISO 12.9	M1.6–M36	190 MPa

*Repeatedly applied, axial loading, fully corrected.

Shigley's Mechanical Engineering Design

For Reference Only

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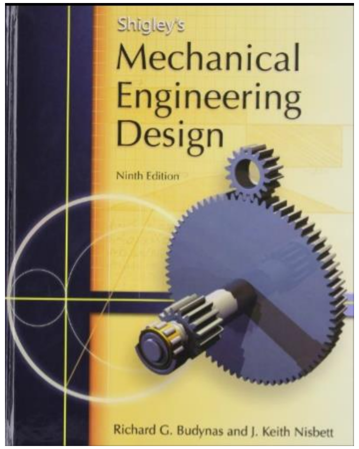
Bolt Fatigue (Brief)

More Detailed Evaluation

4. Some cases require full fatigue calculations
   a) IN718 Bolts?
   b) Very high, or cryogenic temperatures?
   c) High stress / yielding, low cycle fatigue (LCF)
5. Do stress-life or strain-life evaluation
   a) Use Kt's from Shigley
   b) Include mean stress correction
   c) Do not forget Marin Factors… all of them!
      - Surface finish, size factors, reliability (including number of critical bolts), temperature, axial/bending, corrosion…
      - These knockdowns usually reduce fatigue strength by >50% and fatigue life by order(s) of magnitude!

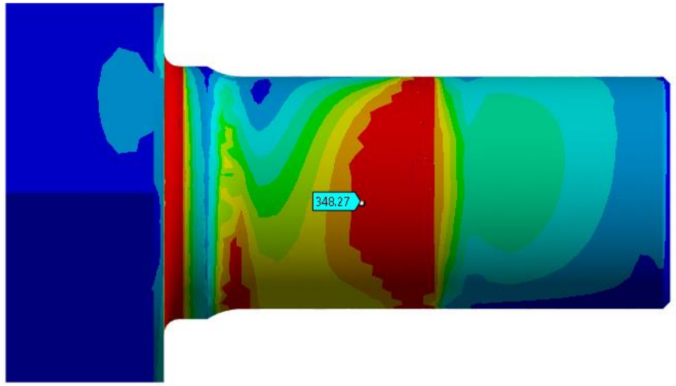
$$S_e = k_a k_b k_c k_d k_e k_f S'_e$$

k_a = surface condition modification factor
k_b = size modification factor
k_c = load modification factor
k_d = temperature modification factor
k_e = reliability factor[13]
k_f = miscellaneous-effects modification factor
S'_e = rotary-beam test specimen endurance limit
S_e = endurance limit at the critical location of a machine part in the geometry and condition of use



Shigley's Mechanical Engineering Design

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.21	1.38	−0.067
Machined or cold-drawn	2.00	3.04	−0.217
Hot-rolled	11.0	38.6	−0.650
As-forged	12.7	54.9	−0.758



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Flange Fatigue (Brief)

1. Near washer / bolt head, there is compression contact
   a) Fatigue failures in compression are rare (and hopefully less range than 2X yield stress)
      - Thus we don’t have to capture the washer/edge stress concentration typically
   b) But there is a tensile ring outside the compression ring
      - This circular region of surface tension is typically quite low… but not zero!
2. Check stresses in through-holes
   a) Account for the thread KT if not a through-hole
   b) Consider pretension/assembly case for goodman
   c) Fatigue is not typically an issue due to bolt installation compression, but if there is ovalizing of the hole due to other loads, this region must still be evaluated.

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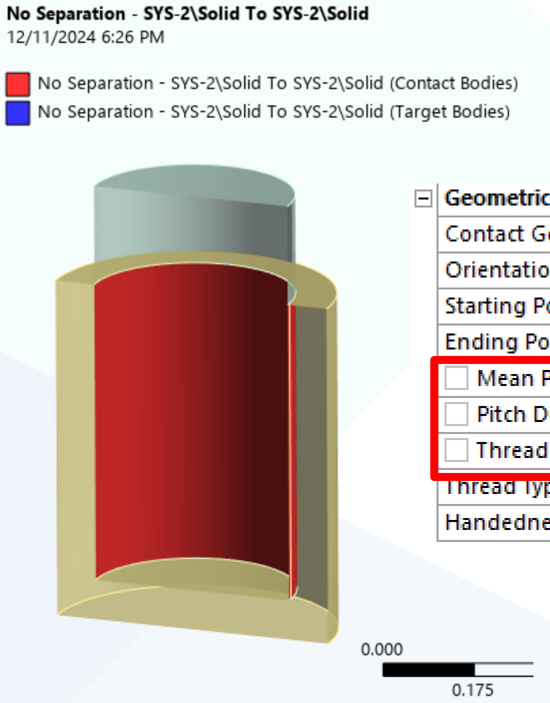
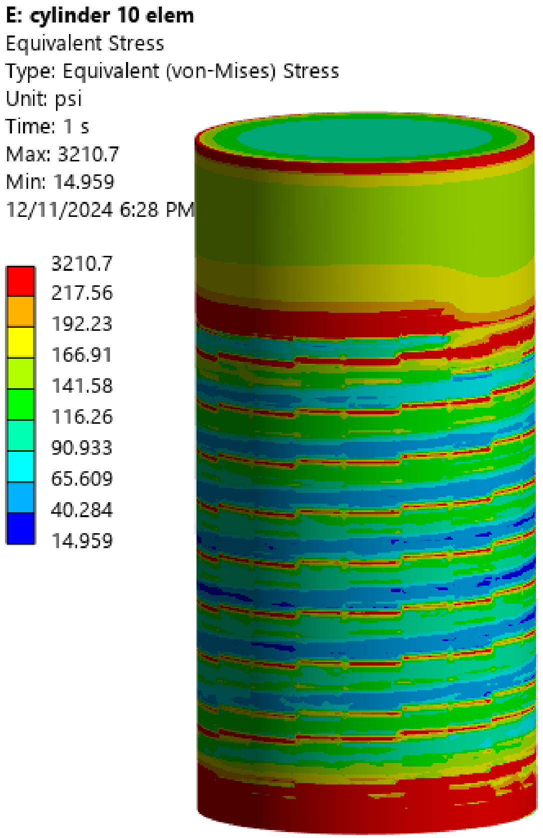
ANSYS Thread Methodology (via Contact)

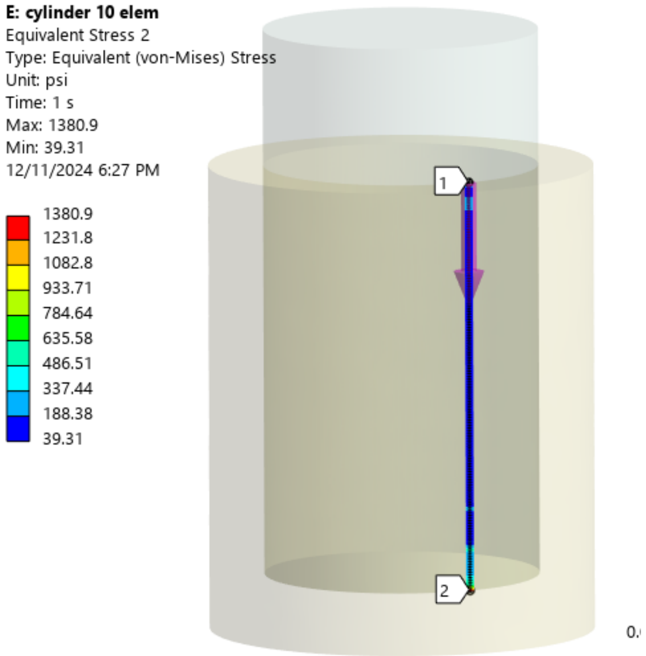
WB Mechanical Tool to simulate threads

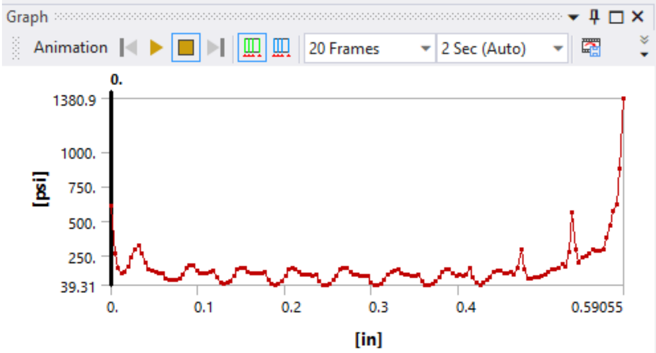
- Option on contact to predict thread stresses
  a) Assign to smooth cylindrical geometry (or any radially symmetric geometry)
  b) ANSYS will predict thread fillet stresses
  c) Can be used for tapered thread geometry



Geometric Modification	
Contact Geometry Correction	Bolt Thread
Orientation	Revolute Axis
Starting Point	Coordinate System
Ending Point	Coordinate System 2
Mean Pitch Diameter	0.35535 in
Pitch Distance	5.9055e-002 in
Thread Angle	60. °
Thread Type	Single-Thread
Handedness	Right-Handed







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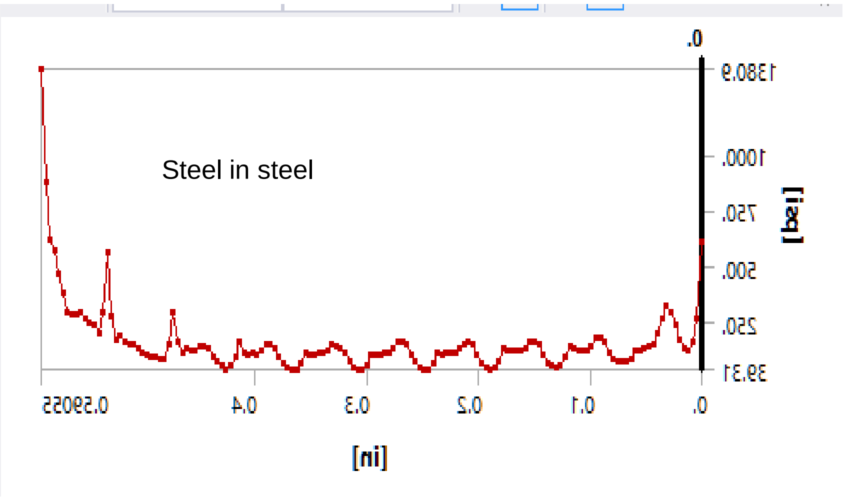
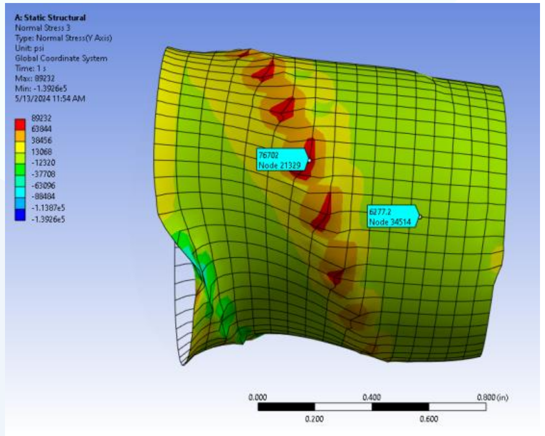
ANSYS Thread Methodology (via Contact)

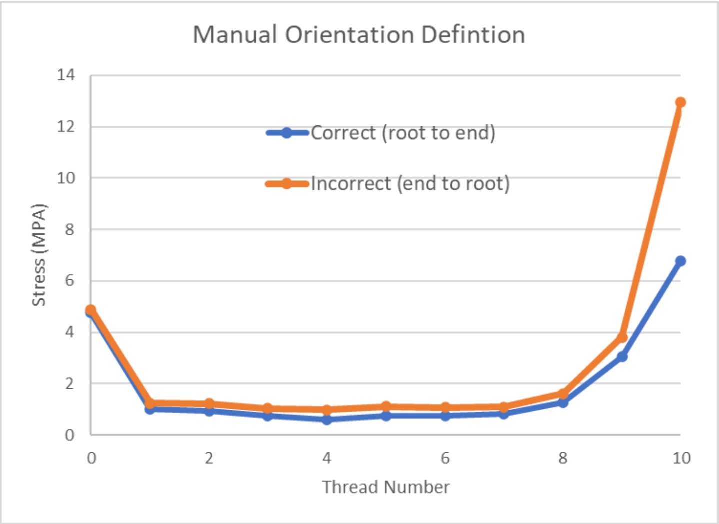
WB Mechanical Tool to simulate threads

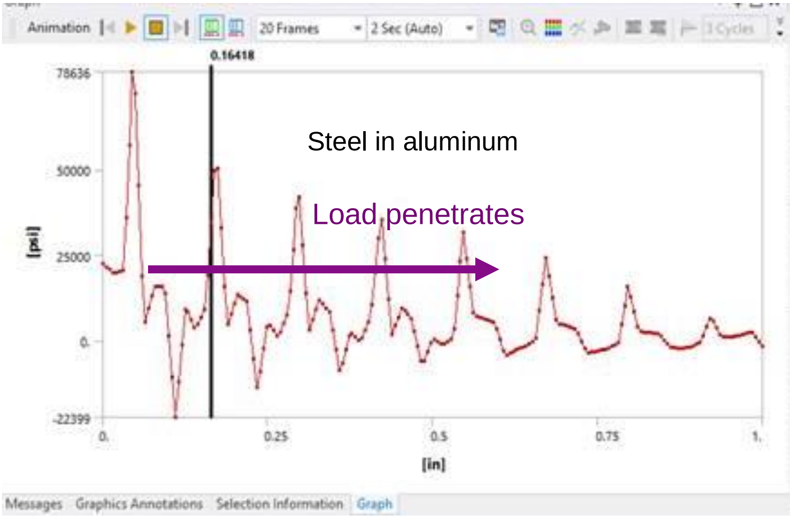
- Implementation
  a) Do **NOT** used bonded contact
    - Use “no separation”
    - As of 24R2, is not grayed out, so its possible to solve incorrectly
  b) Set hole diameter to thread major diameter
  c) If programmed controlled orientation fails, switch to manual
    - Define Revolute Axis starting point (root of thread) to ending point (end of thread)
    - Specifying in reverse order gives incorrect results





High pitch thread confirms spiral (not circular) stress pattern.





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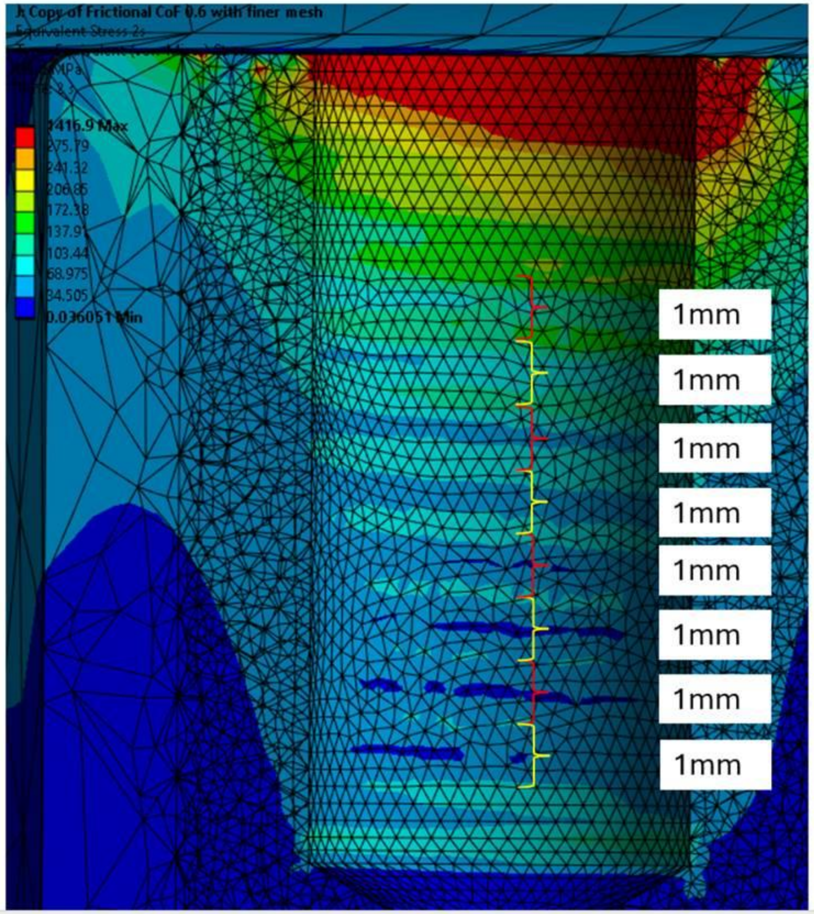
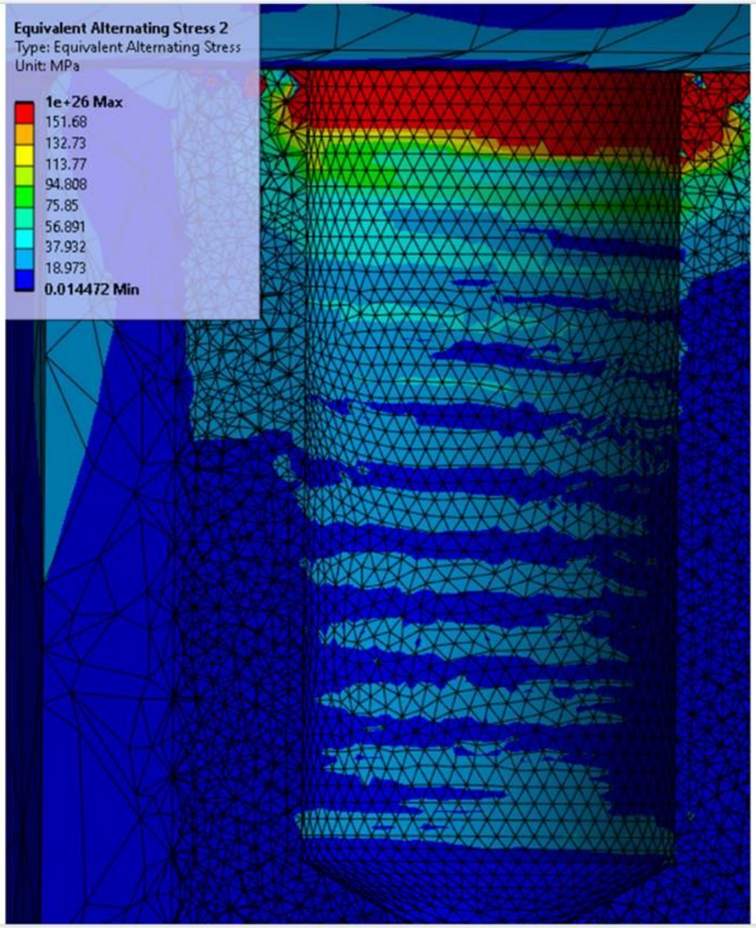
ANSYS Thread Methodology (via Contact)

Evaluating Results

- Consider built-in Goodman correction tool if checking thread fatigue



Peak Stress



Equivalent Alternating Stress

This particular case has high shear at the surface --- that won’t always be the loading.

Also was steel bolt into aluminum, so threads deep in the hole were highly loaded.

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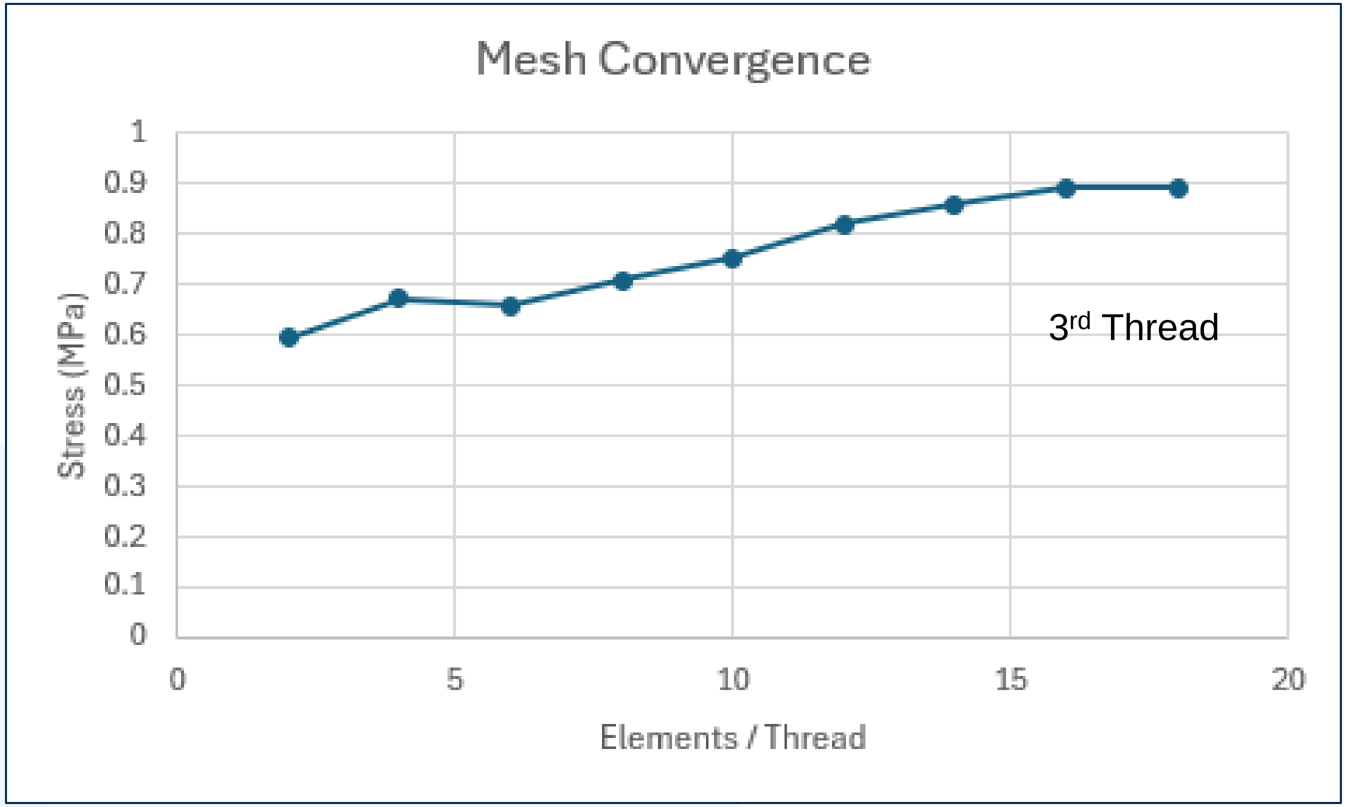
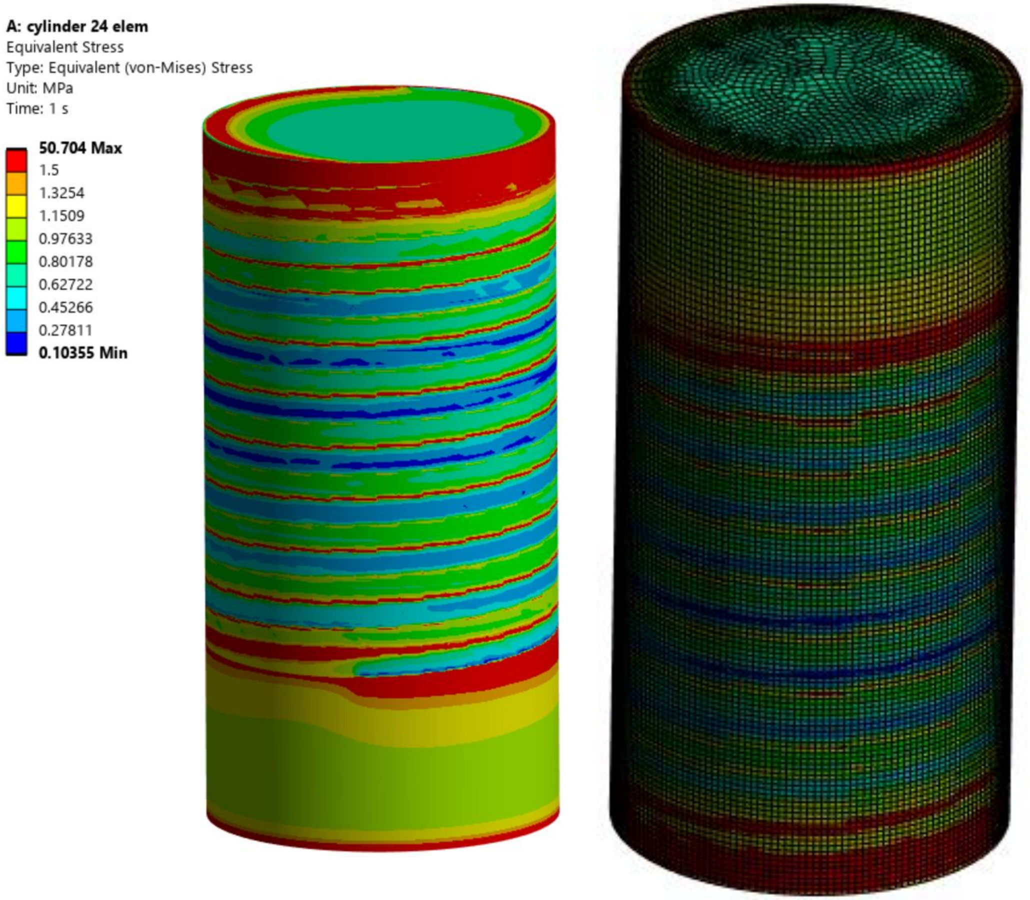
ANSYS Thread Methodology (via Contact)

Mesh dependent

1. Peak stresses will be mesh dependent.
   - Many elements per thread are required for accuracy
2. Performed mesh convergence study on test case



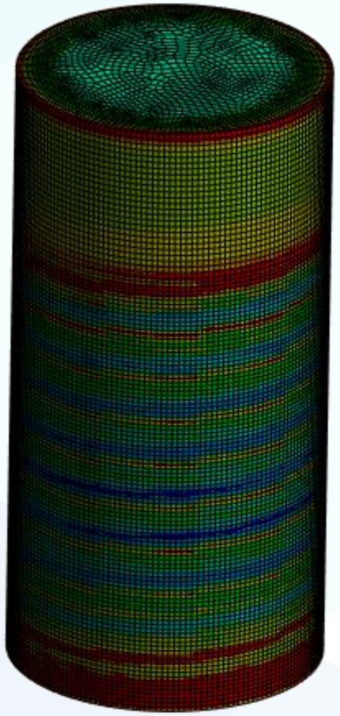


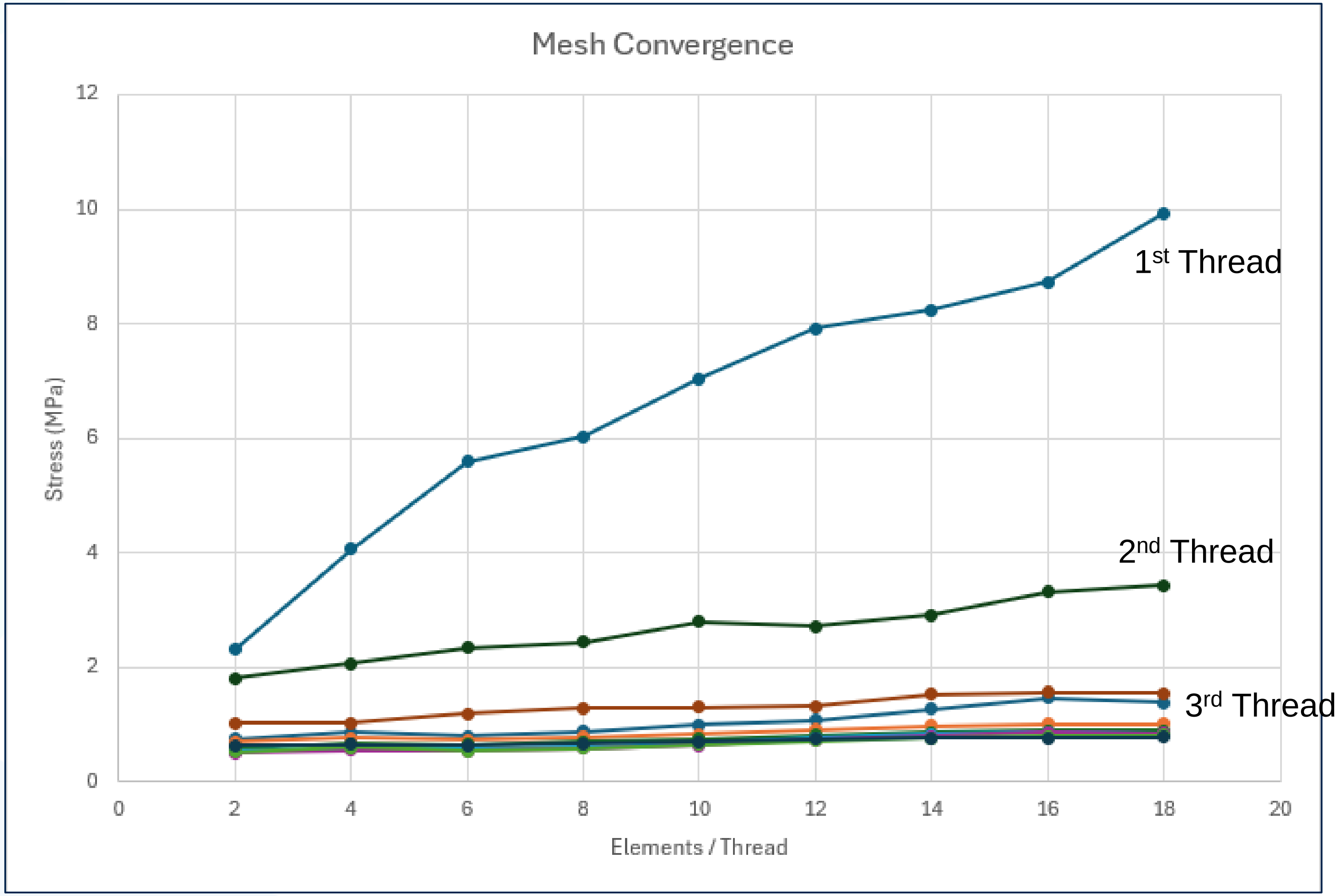
24 Elements / Thread

ANSYS Thread Methodology (via Contact)

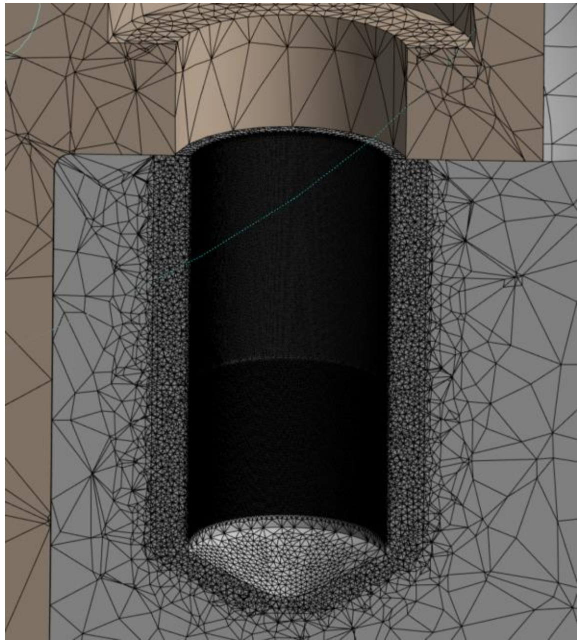
Full Convergence Study

24 Elements / Thread





Example from actual simulation



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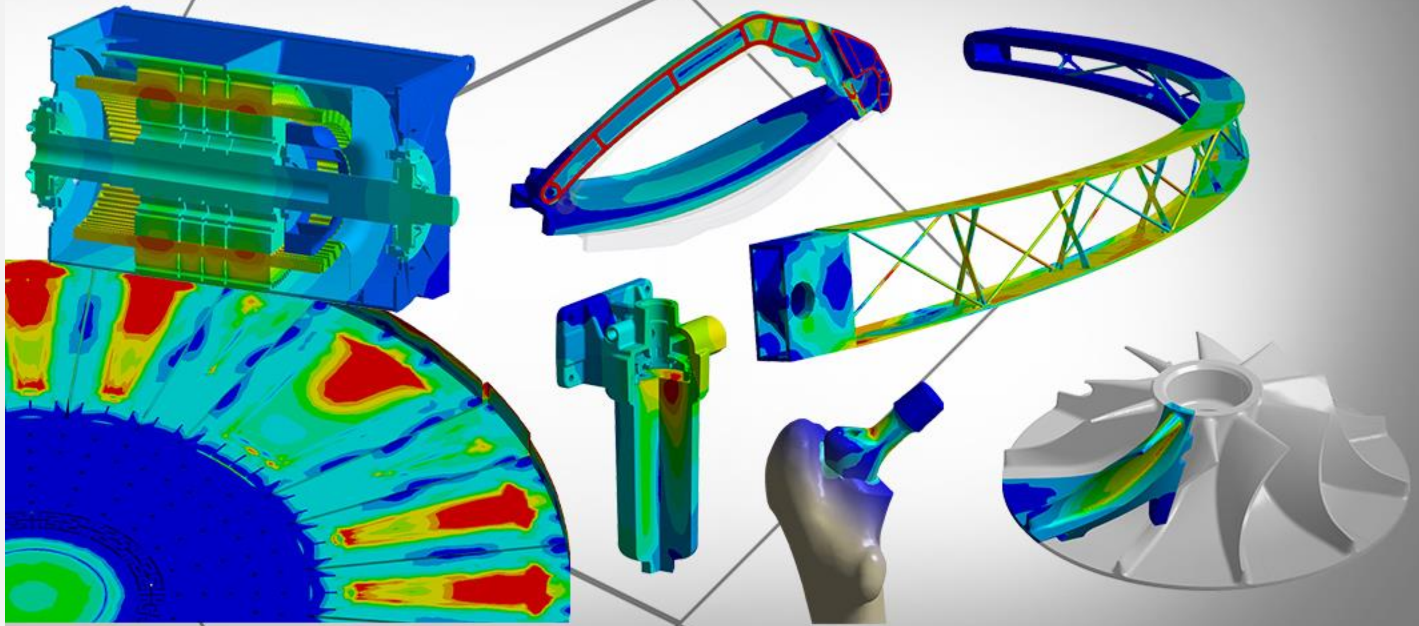
Input / Questions

*Image used *without* permission from ChatGPT.



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EXPERIENCE INNOVATION