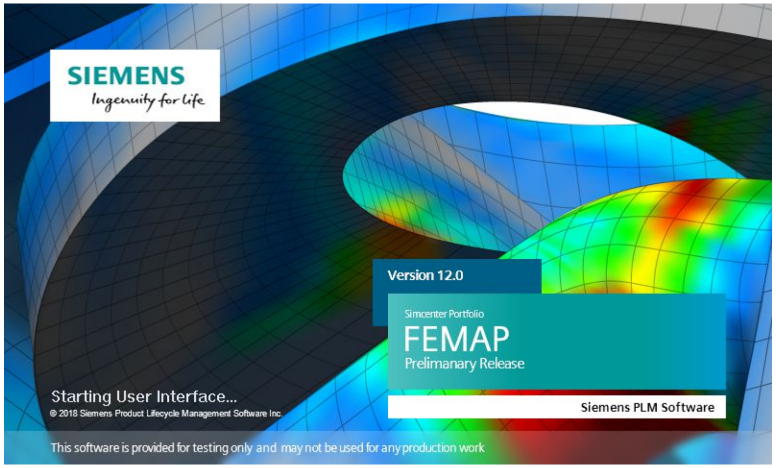

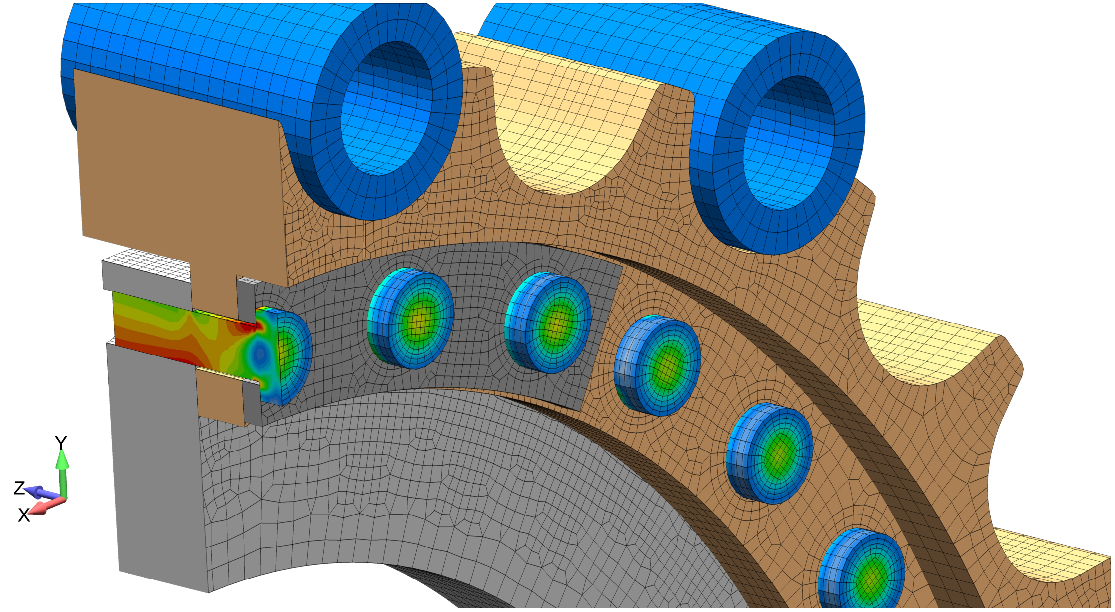

User Guide: FEA of Bolted Joints

A Seminar for Simulation Engineers

George Laird, PhD, PE – Principal Mechanical Engineer

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TABLE OF CONTENTS

1. FEA OF BOLTED JOINTS: WHAT THIS SEMINAR IS ABOUT

The literature has plenty of information on the modeling of bolts. It is the usual story about compromise versus accuracy or do I throw a beam/RBE at it (super-simple and efficient) or go full 3D with contact (super-slow and numerically expensive). We know that if someone wants to throw stones at your model, a reviewer can easily critique your model for using beam/RBE combinations to model a bolted joints whereas a 3D approach is more or less bomb-proof. But this is engineering and not pie-in-the-sky and we have to get our models built with reasonable accuracy and out-the-door.

1.1 WHAT WE WILL COVER

- Why all bolt modeling has it challenges
- Basic problems with linear analysis of bolted connections
    - Is the loading tensile or bending?
    - What type of bolted joint?
- Linear and Nonlinear Analysis – the basic beam/RBE combination
- Why going 3D is not always the answer and can be a trap
- Fatigue analysis of bolted joints (don't get your hopes up, but we'll make a few comments)

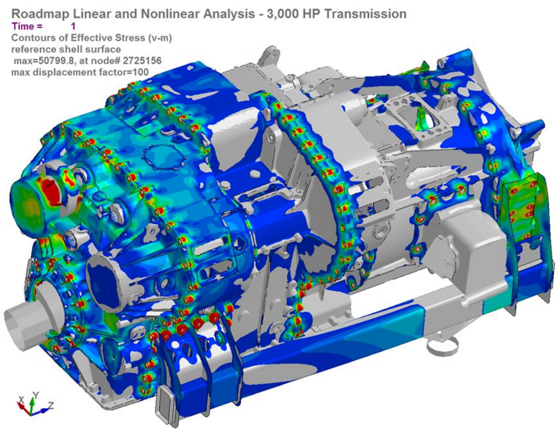
1.2 WHAT WE DON'T COVER

- Nothing to do with spot welding since it is another class of joints
- No discussion on what is the best connection for all bolted connections since there is no best idealized connection
- No proprietary software/solution that promises bolt connection nirvana

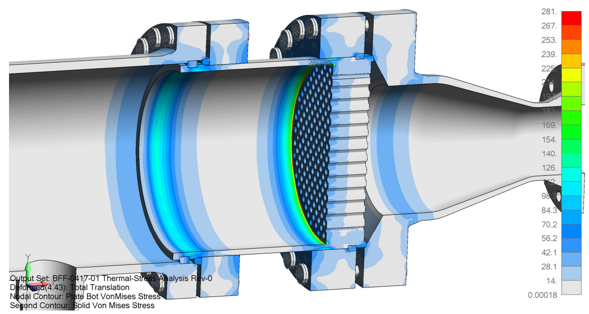
2. WHAT WE KNOW

A selection of some consulting projects at Predictive Engineering that have had bolt modeling challenges:

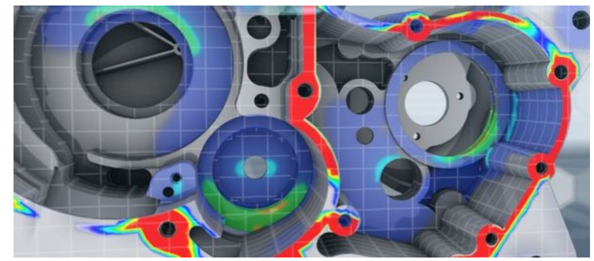
3,000 HP 8-Speed Transmission



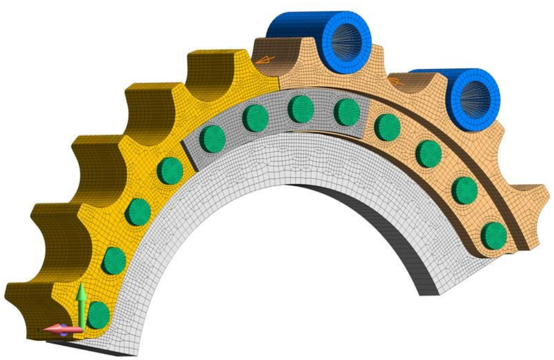
Bolted Flanges



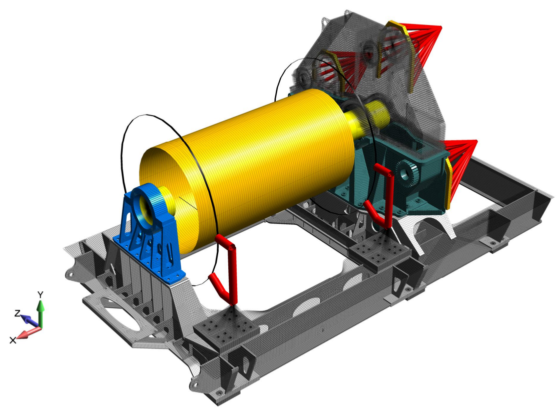
Norton Motorcycle Crankcase



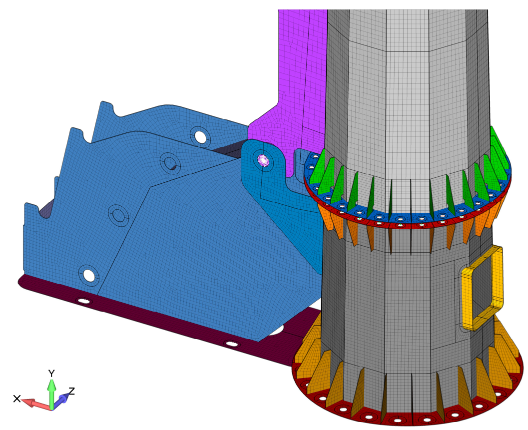
Open-Pit Mining Apron Conveyor



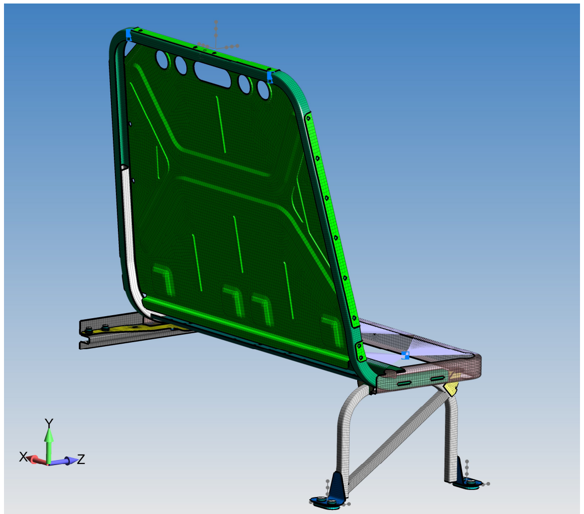
Off-Shore, High-Speed Winch



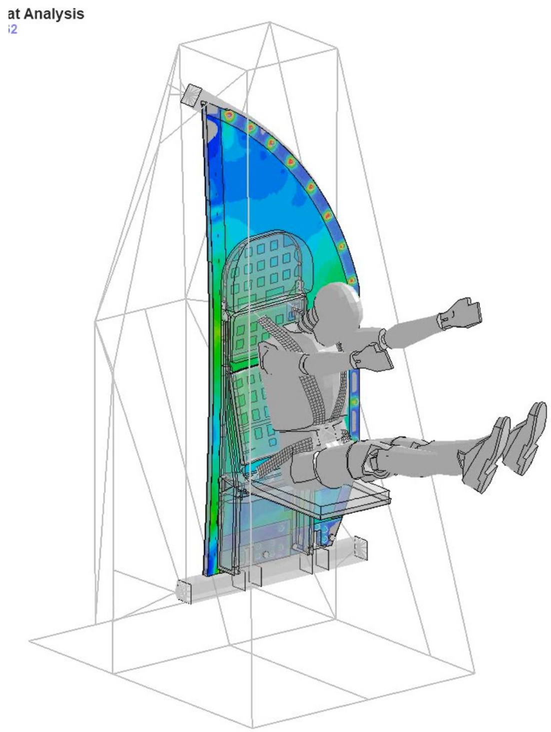
100 kW Wind Turbine Tower



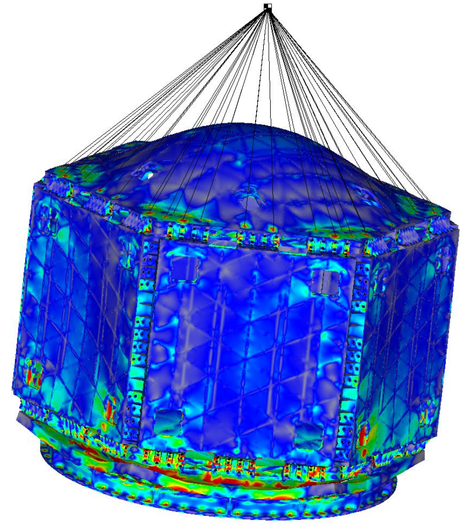
Bus Seat



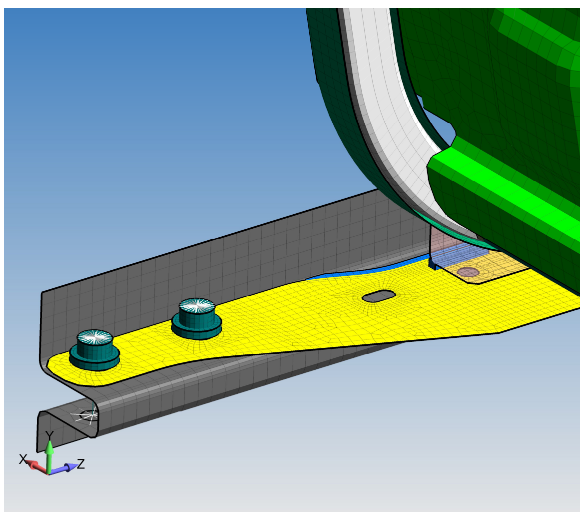
16-g Airplane Attendant Seat



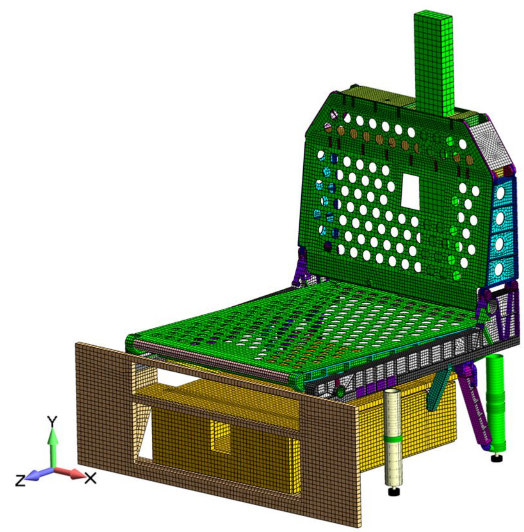
Research Satellite



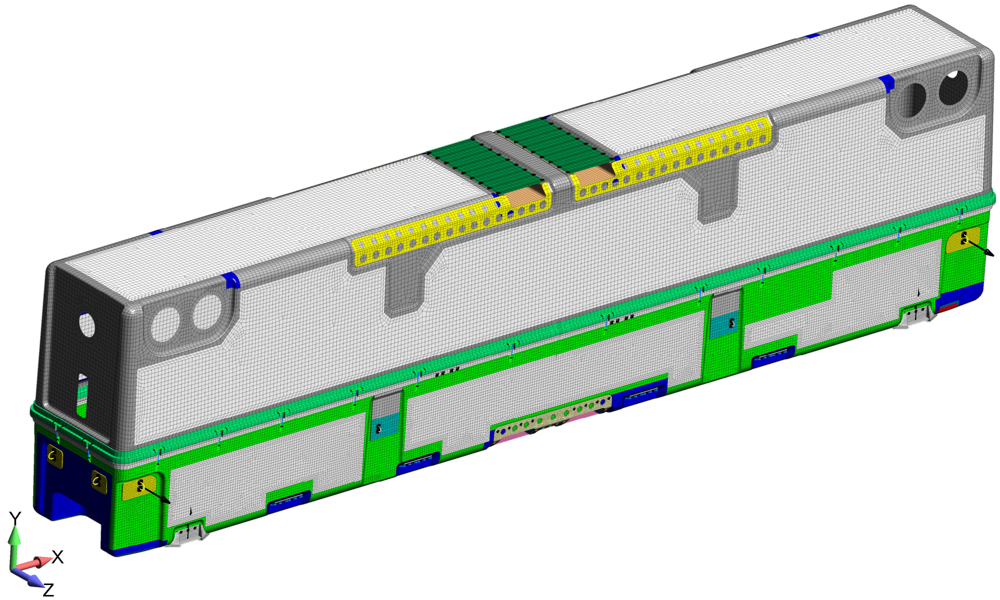
Close-Up View of Bolted Connection



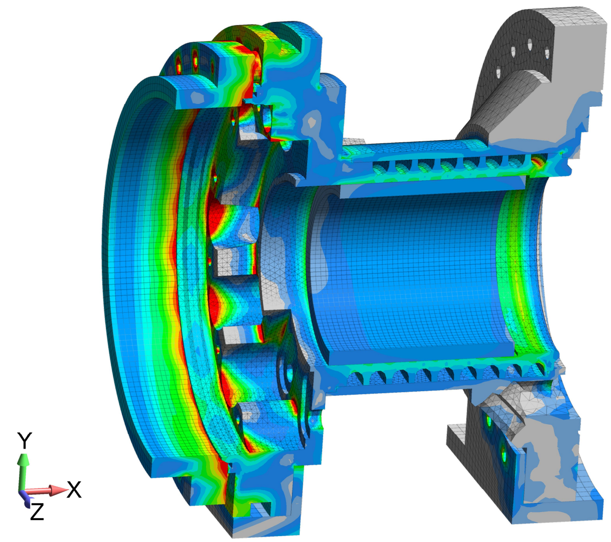
16 g Crash Analysis Airplane Divan Seat



Composite Aviation Container

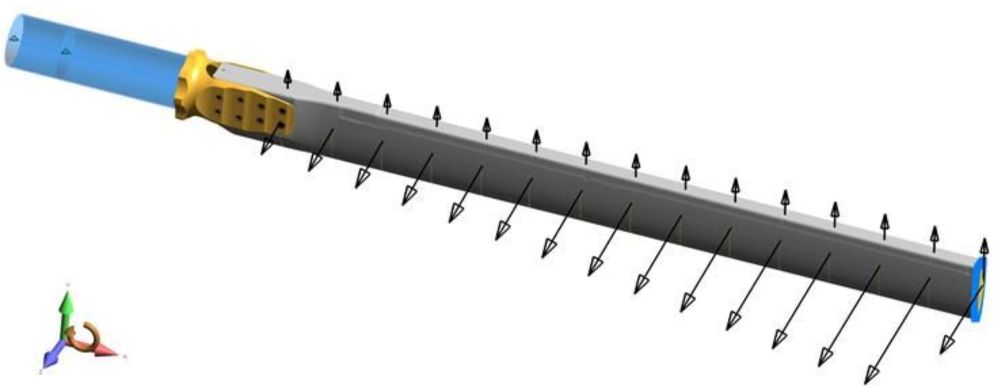


Magnetic Bearing 500 kW Generator

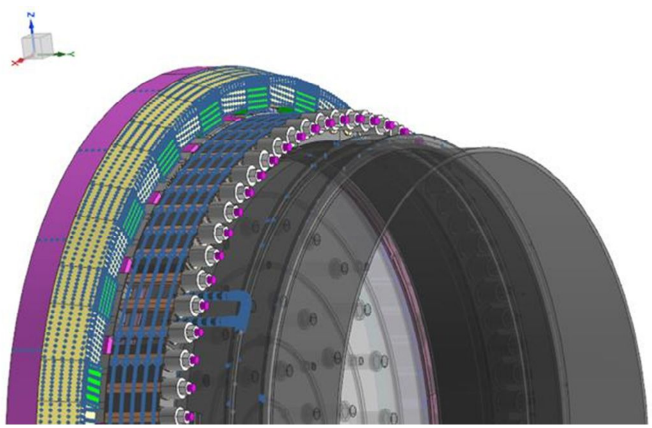


Output Set: Thermal-Stress in 500kW Turbine Generator
Nodal Contour: Nonlinear Solid Von Mises Stress

Helicopter Tail Rotor with Bolted Connection



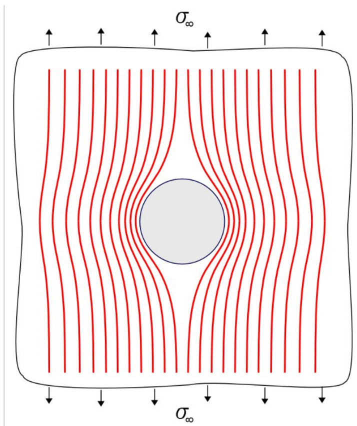
High-Temp Furnace Bolted Connection



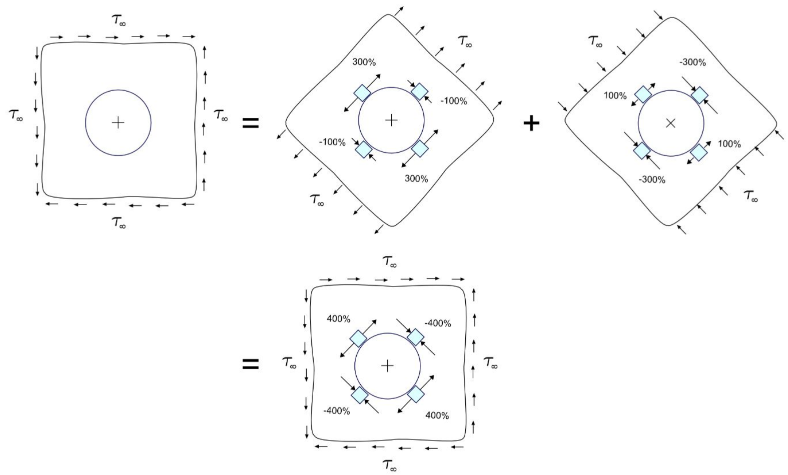
3. WHY ALL BOLT MODELING HAS IT CHALLENGES

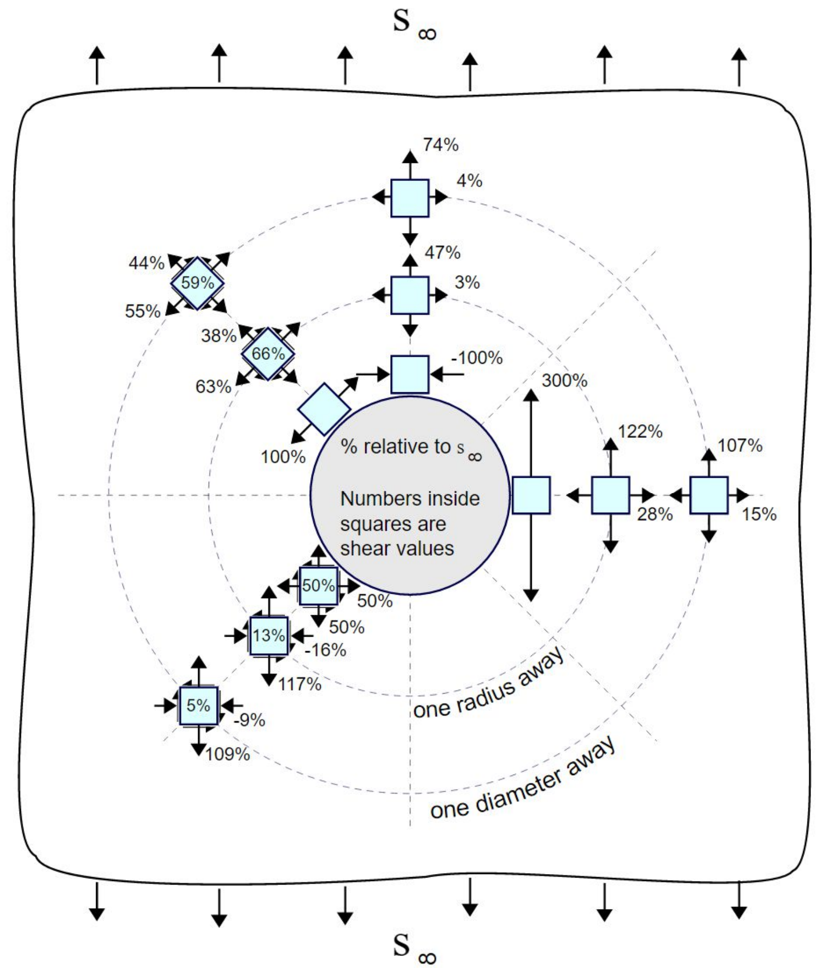
3.1 Let's Start With The Obvious – There is a Hole

Stress Flows $\dfrac{\partial^4 \phi}{\partial x^4} + 2\dfrac{\partial^4 \phi}{\partial x^2 \partial y^2} + \dfrac{\partial^4 \phi}{\partial y^4} = 0$



With Pure Shear – Stress Concentration 4x



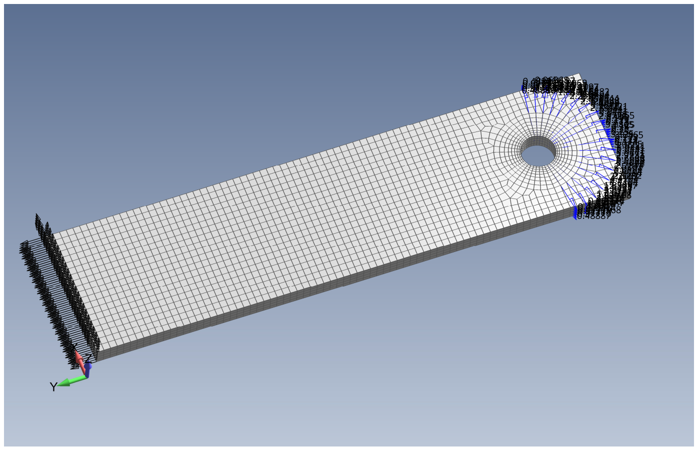


Images courtesy of www.FractureMechanics.org

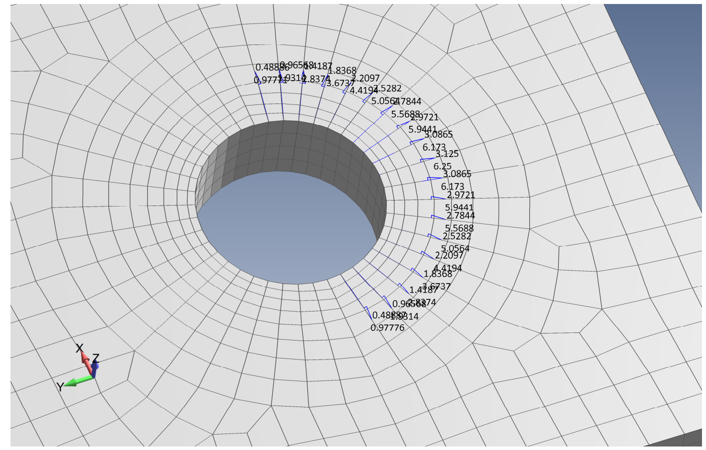
3.1.1 But What Does This Have To Do With Our Bolt Modeling?

Let's assume that our bolted joint is loaded in pure tension. If we narrow our scope to just one bolt in a wide plate, then one can visual the load transfer to the plate as a sinusoidal function where the peak bearing pressure is aligned with the direction of the tensile load. Let's show some stress images:

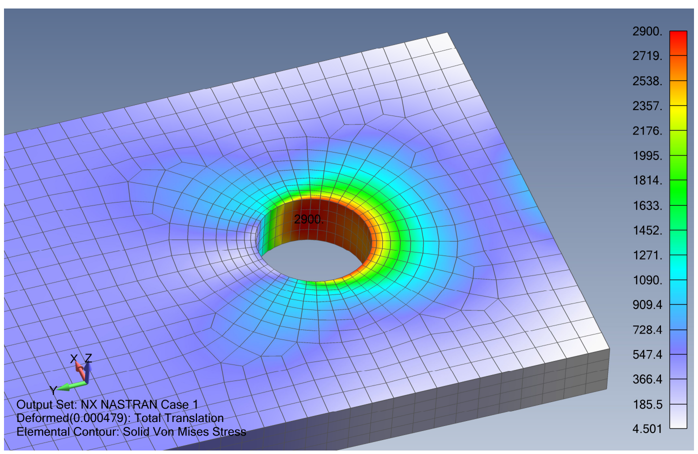
Bearing Load (250 lbf) to Mimic Bolt



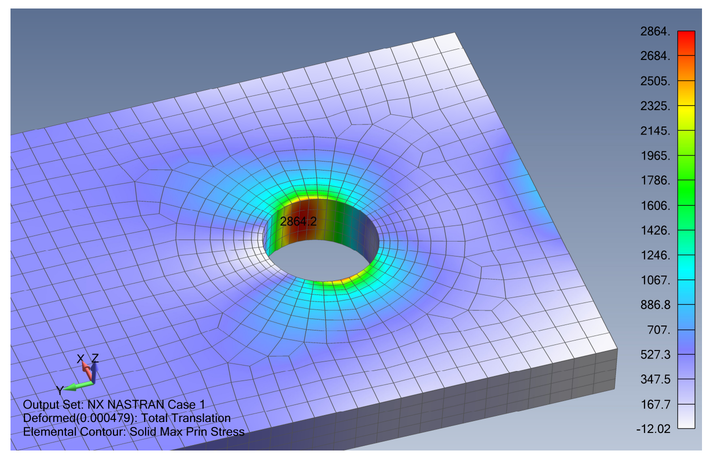
Max Pressure Aligned with Tensile Load



Von Mises Stress – 2,900 psi

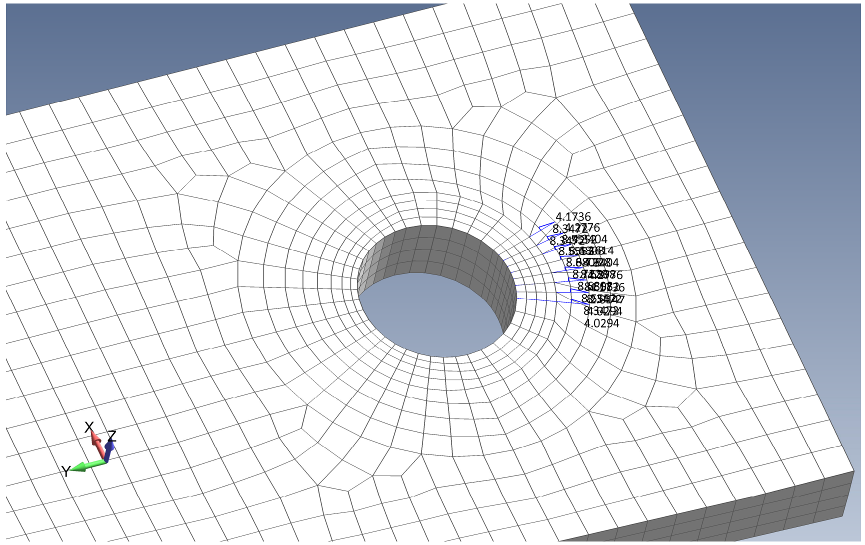


Maximum Principal Stress (σ_1) – 2,900 psi

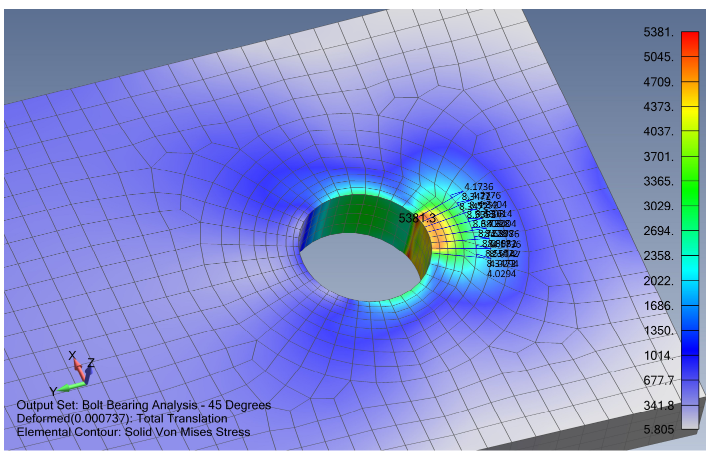


3.1.1.1 WHAT IF WE HAVE A NON-PERFECT BOLT FIT?

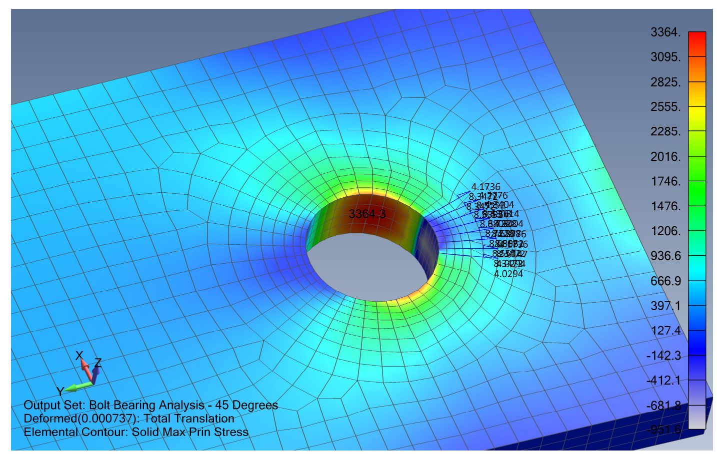
Bearing Load (250 lbf) to Mimic Bolt - Load Applied Over 45 Degree Region



Von Mises Stress – Hot Spot – Not Useful



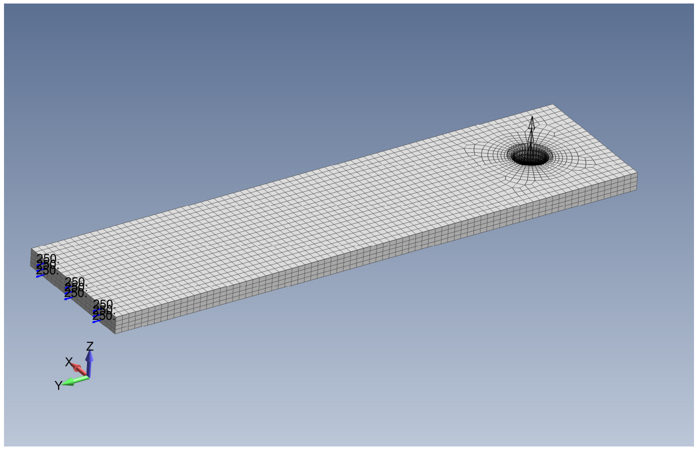
Maximum Principal Stress (σ_1) – 3,400 psi



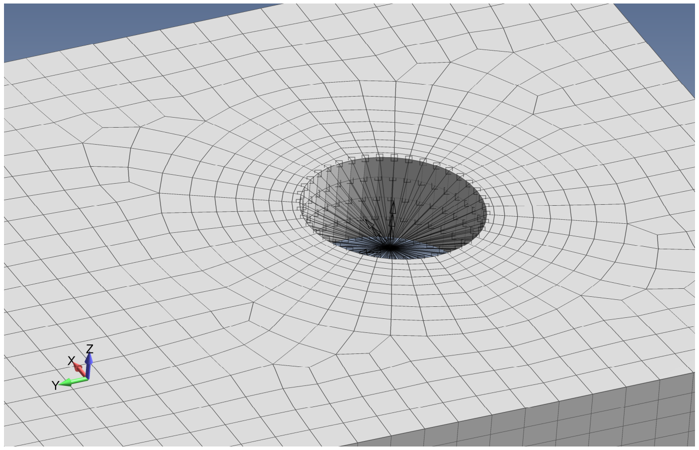
3.1.2 How Adequate is the RBE Idealization?

Let's take the same model and replace the bearing load with a RBE2. The RBE2 is constrained (6 DOF) at its independent node at the center of the hole while at the far end, a load of 250 lbf is applied. The results speak for themselves in comparison to the more realistic idealization using a bearing force.

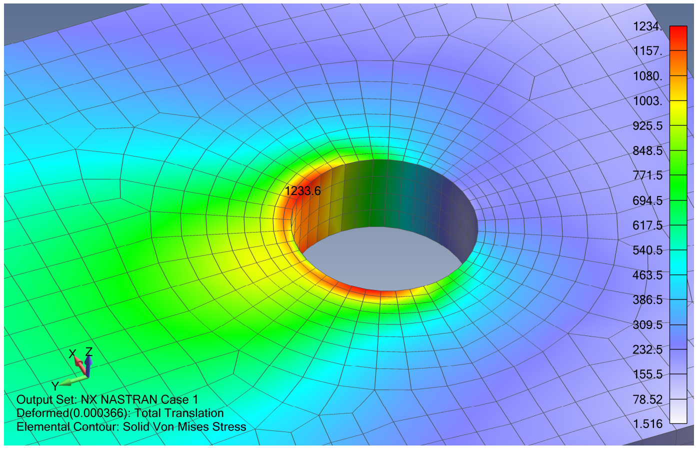
Same Load of 250 lbf



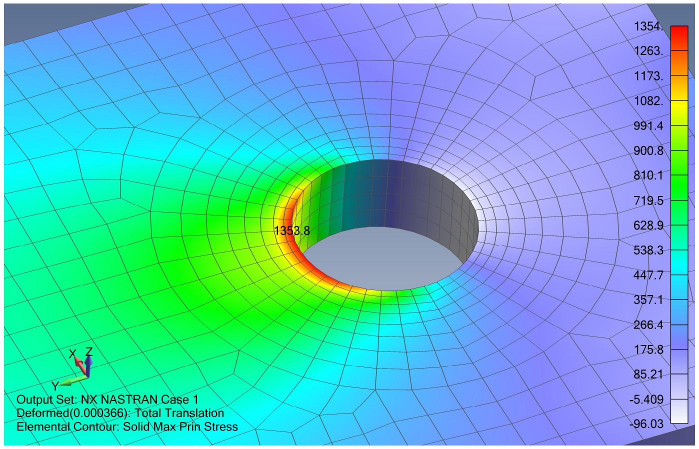
RBE2 with Independent Node Fixed



Von Mises Stress – 1,200 psi



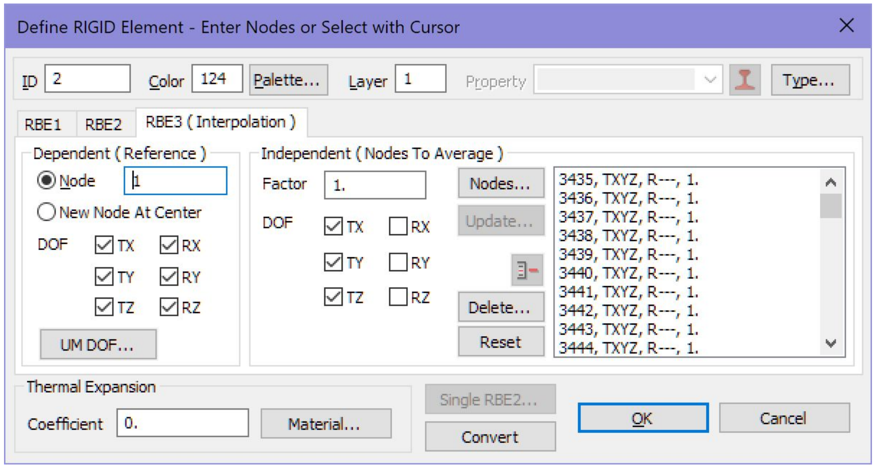
Maximum Principal Stress (σ_1) – 1400 psi



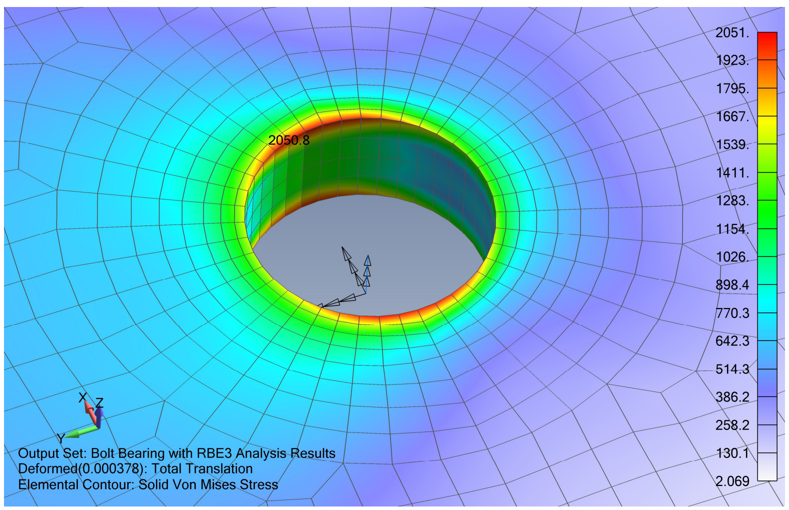
3.1.2.1 But Wait, We Could Use a RBE3!

Since this question would naturally come up, had to show that the results are not any better just more confusion.

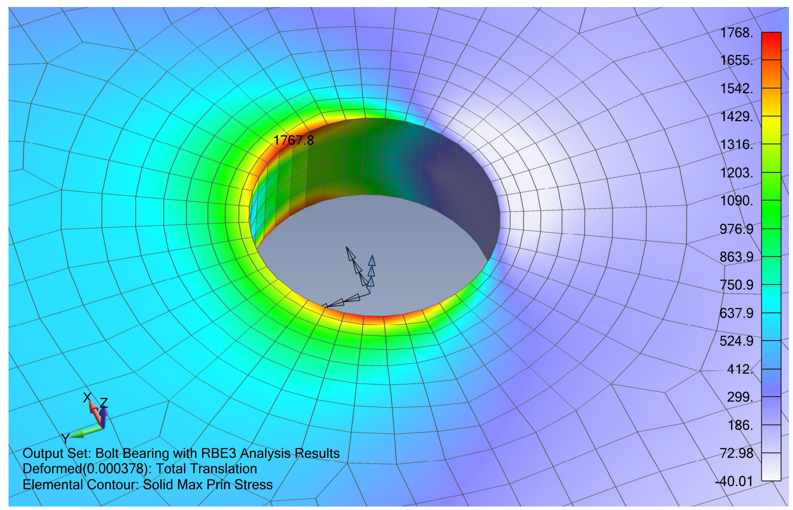
RBE3 – Force Interpolation Element



Von Mises Stress – 2,500 psi



Principal Stress (σ_1) – 1,700 psi

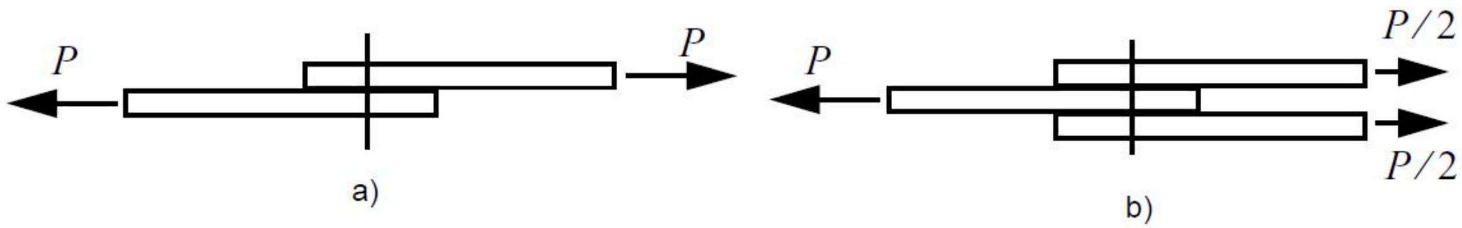


3.2 In Summary: About Plates with Holes with Bolts

If you put a hole in a plate and pull on it, you have a stress concentration of 3x. A little side note is that one often knocks downs composite data by 3x to set margins. This makes sense given that most composite structures have holes drilled into them and likewise why it is common to use inserts to spread out the connection bearing load.

- Holes are stress concentrations
- If you pull on a plate with a hole, you have a 3x effect
- The bending stress field is likewise bumped up by 3x
- Don't get your hopes up that any idealization technique will get you even close to the true stress state around a bolted hole (we'll get more into this later), that is, keep in mind that the RBE idealization may provide stresses that are lower than reality.

Idealization	Von Mises Stress, psi	Maximum Principal Stress, psi
Bearing Load Idealization - 180°	2,900	2,900
Bearing Load Idealization - 45°	not useful	3,400
RBE2	1,200	1,400
RBE3	2,500	1,700
Reality	?	?



3.2.1 But the Reality Is: Airframe Stress Analysis and Sizing by M.C. Niu

Design for Tension and Shear Tear Out

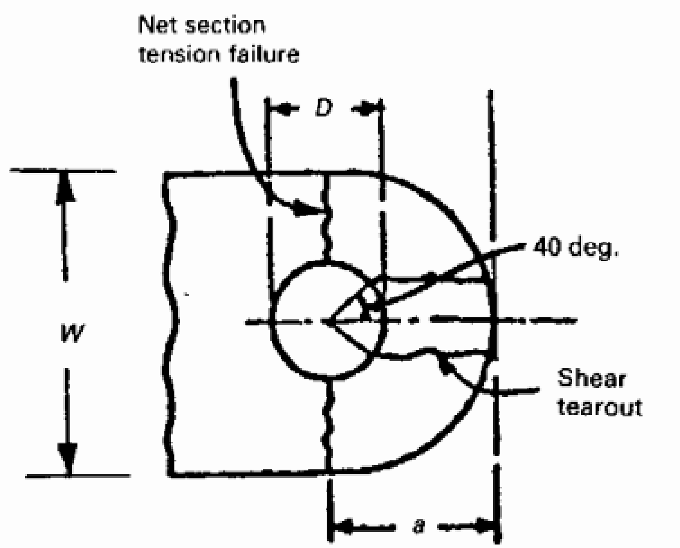


Fig. 7.5.2 Lug net section tension and shear tearout failure under axial load.

Material Bearing Loads

Basic Criteria of Fastener Strength Allowable

The allowable loads are based on the lowest values of the following criteria:

(1) Bearing load (protruding head only) $= F_{br} dt$

where

F_{br} — Allowable ultimate bearing allowable stress of sheet material is based on either MIL-HDBK-5D "B" value per Ref. 7.6 or other sources

d — Nominal shank diameter

t — Nominal sheet thickness

(2) Shear-off load $= F_{su}\left(\frac{\pi d^2}{4}\right)$

where

F_{su} — Ultimate shear allowable stress of fastener material is based on MIL-HDBK-5D per Ref. 7.6 or other sources

d — Nominal shank diameter

(See Fig. 7.1.3)

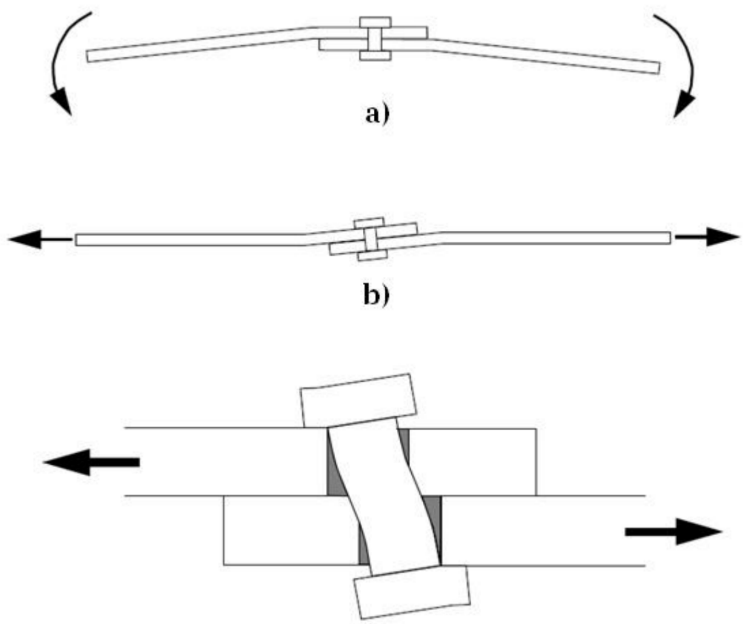
(3) Countersunk fastener and sheet combinations — the allowable ultimate and yield loads are established from actual test data.

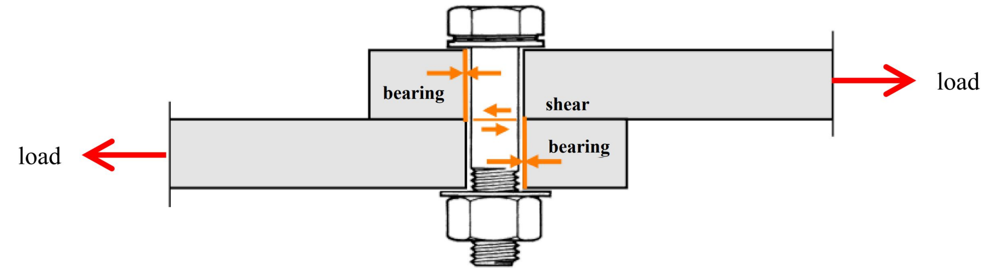
(4) Yield strength to satisfy permanent set requirements at limit load (limit load $= \frac{\text{ultimate load}}{1.5}$).

4. THE BOLTED LAP JOINT

Our test case is the lap joint where we will pull it and bend it. This is our foundation and but is sufficiently complex to get lost in the weeds.

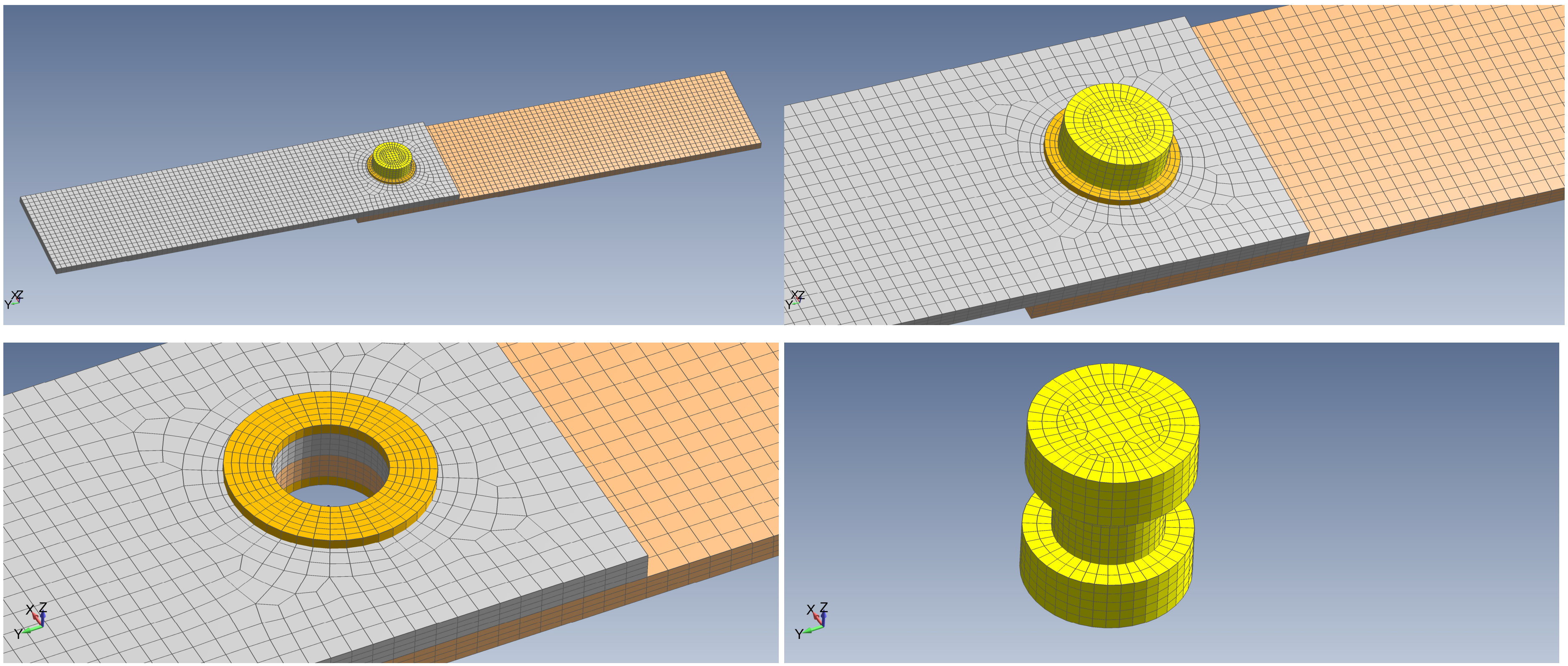
Thinking About Load Application to a Bolted Joint





4.1 LET'S START WITH REALITY: THE 3D BOLT MODEL

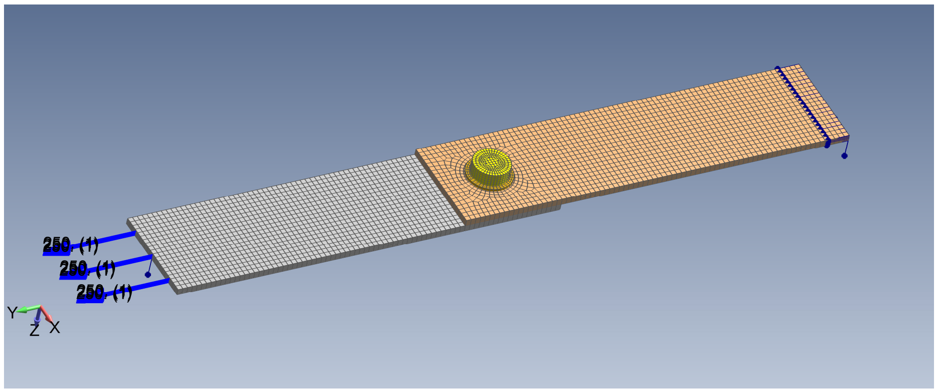
Stress flows (i.e., Airy Stress Function) and since structures are 3D, we'll build up a full-on 3D model with contact. The model follows the dimensions for the simple plate discussed in the prior section at 8" L, 2" W and ¼" Thk. The hole diameter is 0.5" with a bolt diameter of 0.48". The plates are overlapped by 2.0". The bolt head is 0.75" in diameter. For good measure, a 0.05" thick washer with a diameter of 0.9" was added. Contact was enforced across all parts.



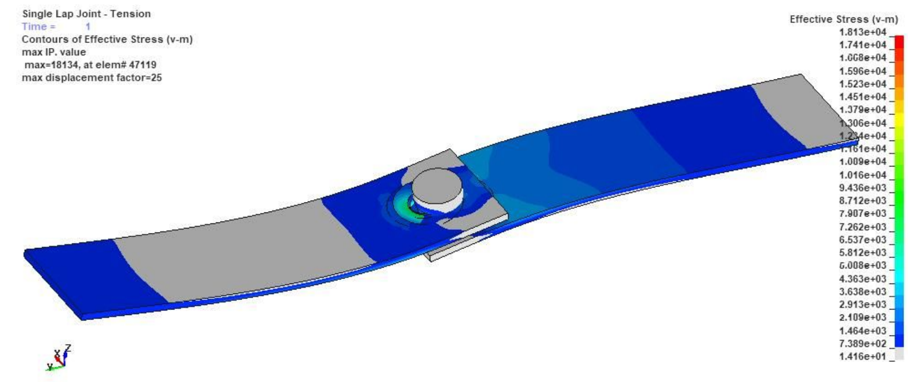
4.1.1 3D Stress Results: Tension (250 lbf)

The bolt is given just a small amount of preload (10 psi) to snug up the plates, washers and bolt heads. The 250 lbf load is then applied.

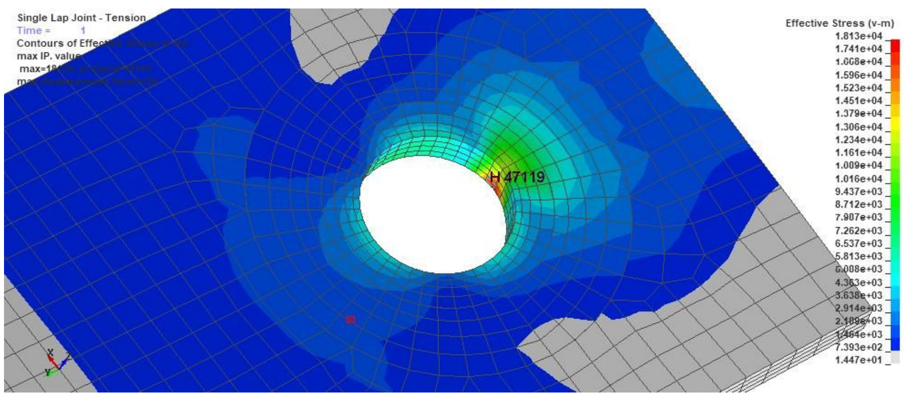
3D Model with 250 lbf Tension Load – Constrained Far End



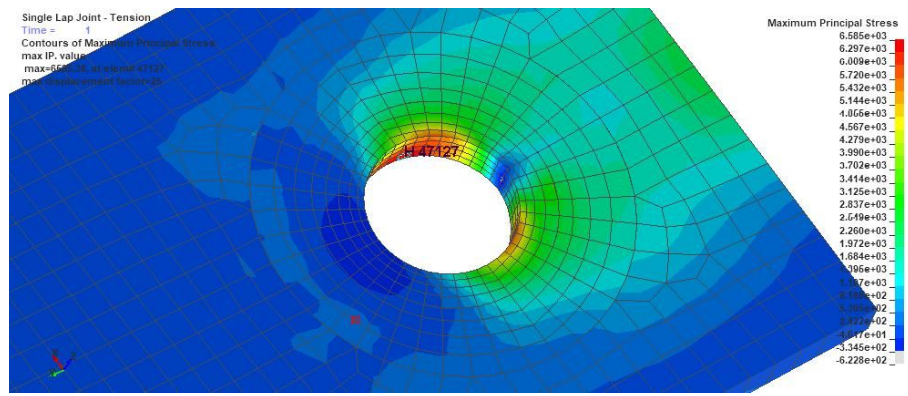
Stress Results – Deflection Scaled 25x



Von Mises Stress 18,000 psi



Max. Principal Stress 6,600 psi



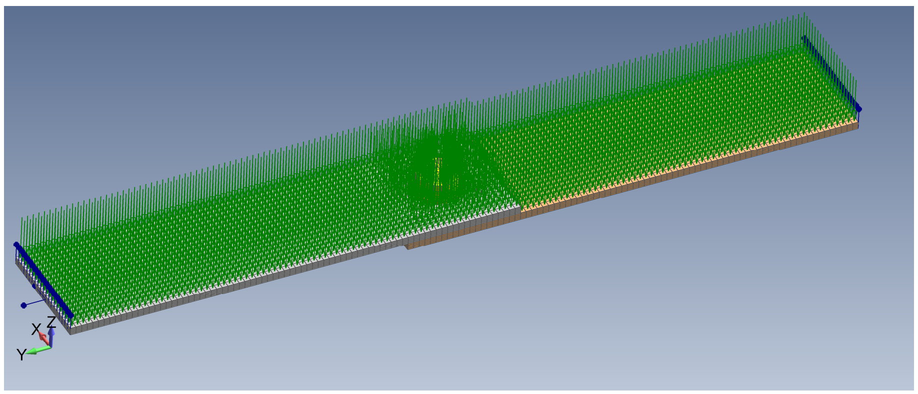
4.1.2 3D Stress Results: Bending

The bending load case has a uniform pressure of 0.33 psi applied along the top surface. The analytical solution for this simply-supported beam is a max. stress of 3,100 psi.

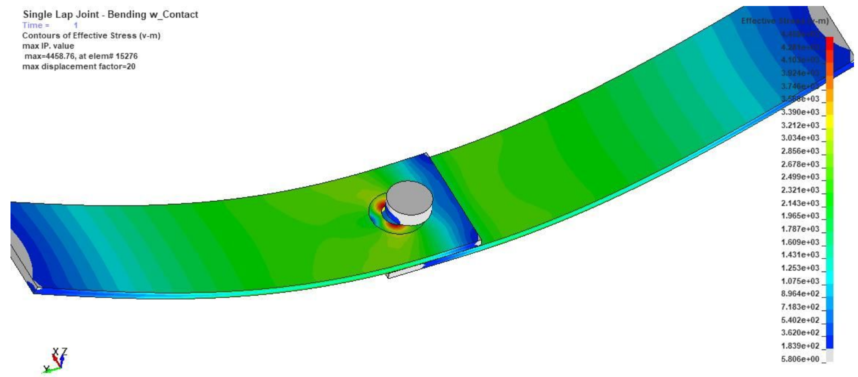
4.1.2.1 No Bolt Preload / No Friction

It is assumed that the bolt has been snugged but there is no real bolt preload and to assure that we have a conservative solution, there is no friction between the plates; thus we have contact but nothing more.

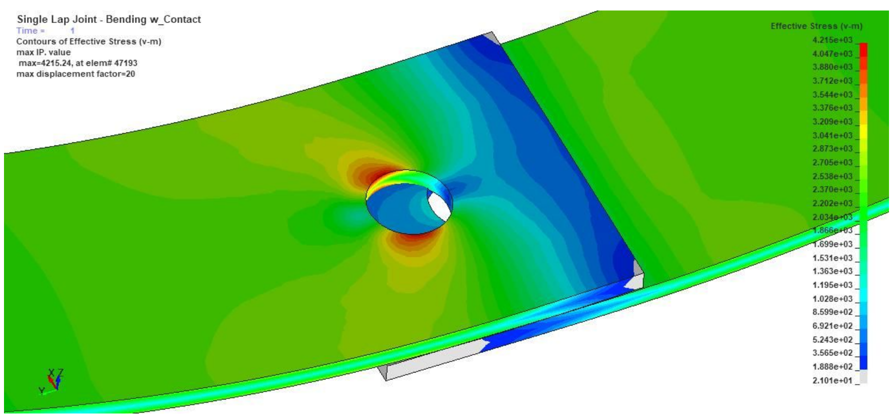
3D Model with pressure load



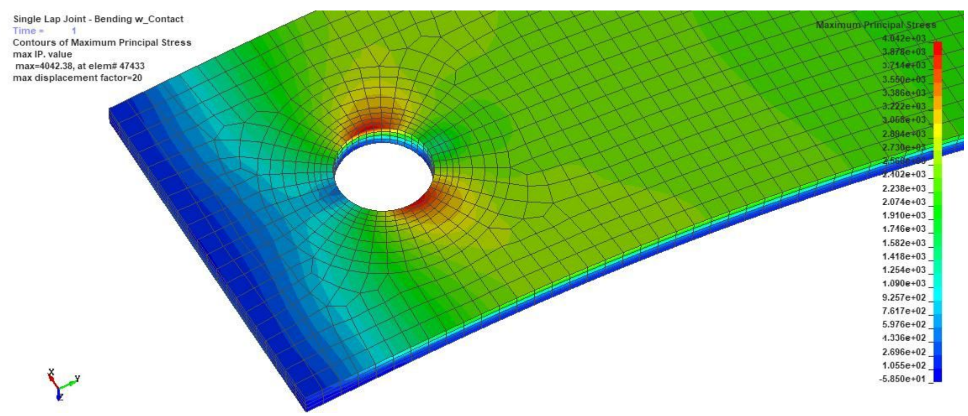
Stress Results – Deflection Scaled 25x



Von Mises Stress 4,200 psi



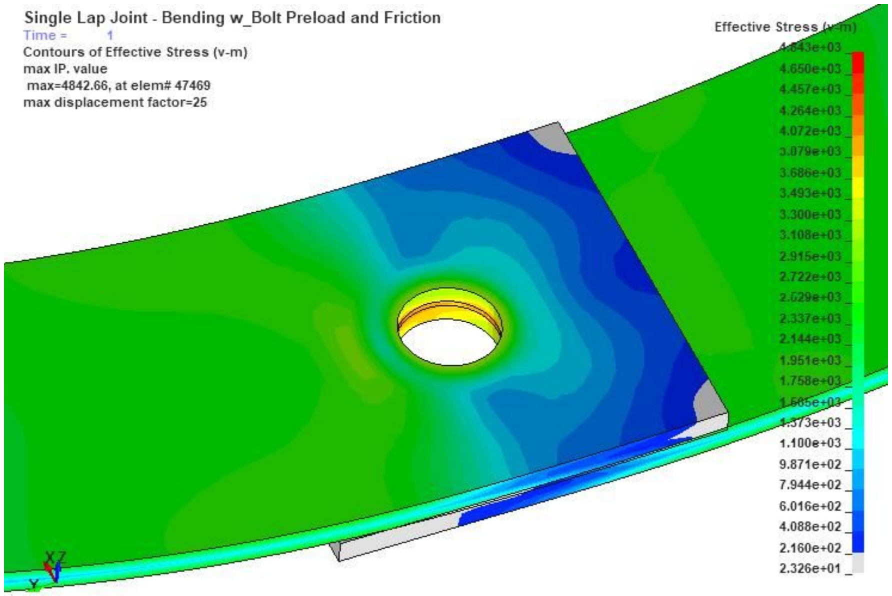
Max. Principal Stress 4,000 psi



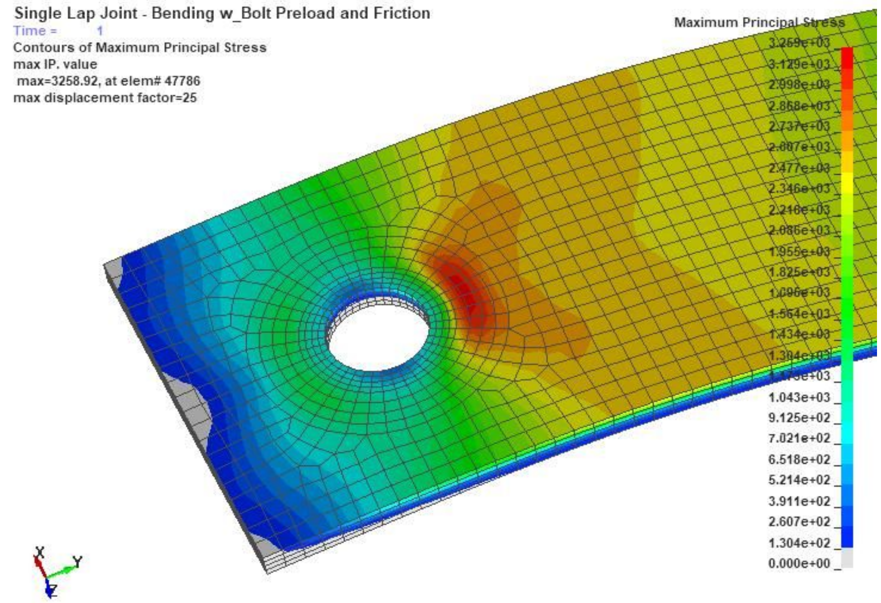
4.1.2.2 Bolt Preload with Friction

Here we crank down the bolt preload to lock the plates together.

Von Mises Stress 4,800 psi



Max. Principal Stress 3,300 psi

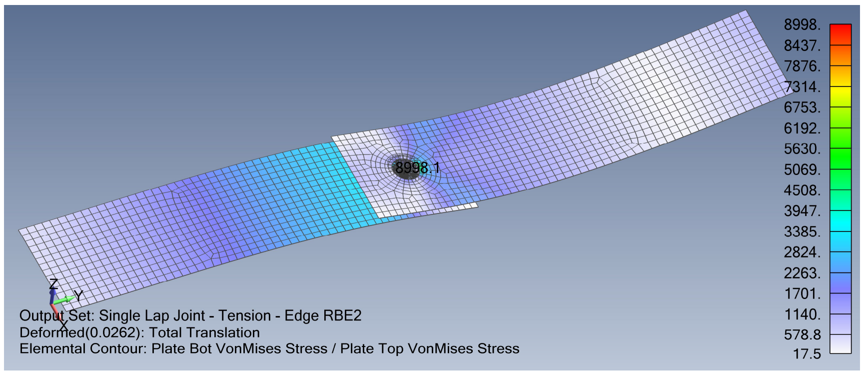
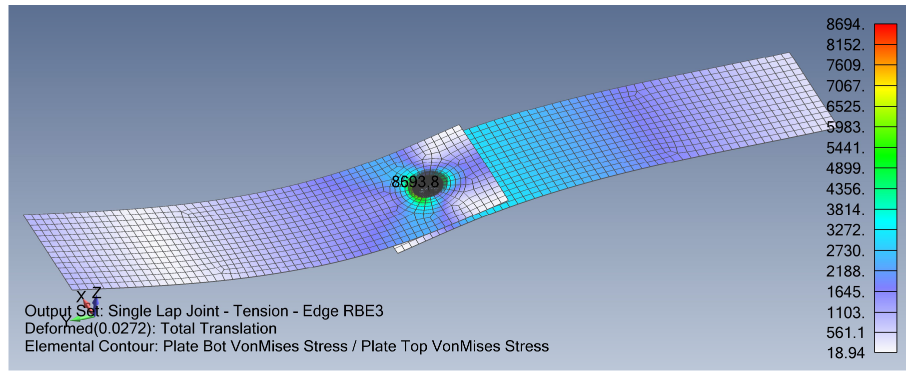
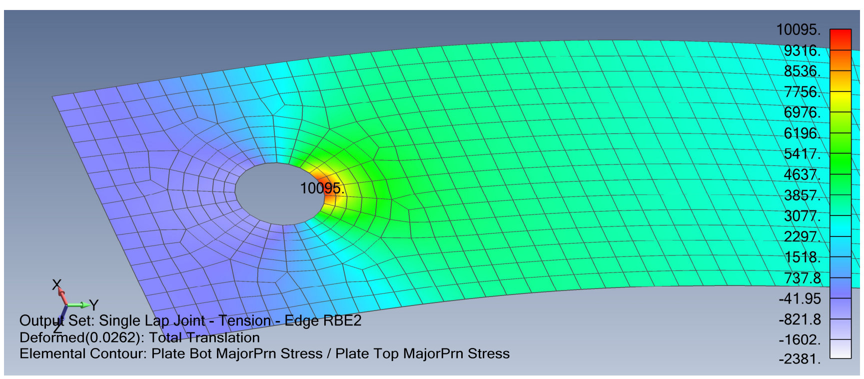
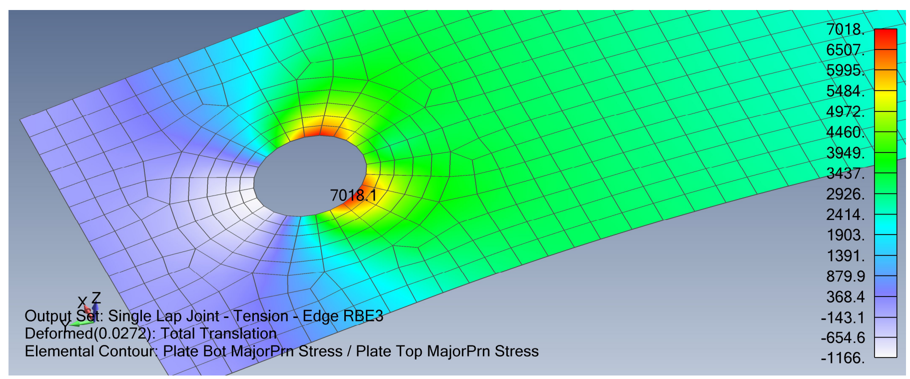


4.2 Lap Joint Bolting with Beam and RBE Elements

4.2.1 Tension Loading

We are modeling the bolt head and nut in two ways: (i) At the edge and (ii) As a pseudo-washer.

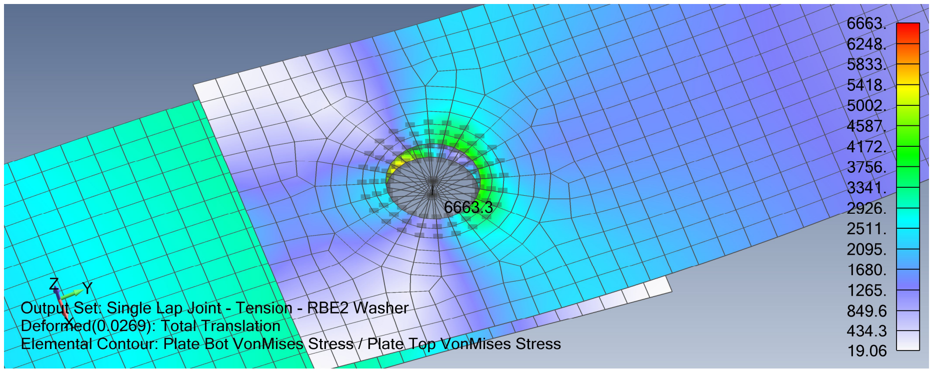
4.2.1.1 Edge Connection

RBE2	RBE3
Von Mises 9,000 psi	Von Mises 8,700 psi
	
Maximum Principal 10,000 psi	Maximum Principal 7,000 psi
	

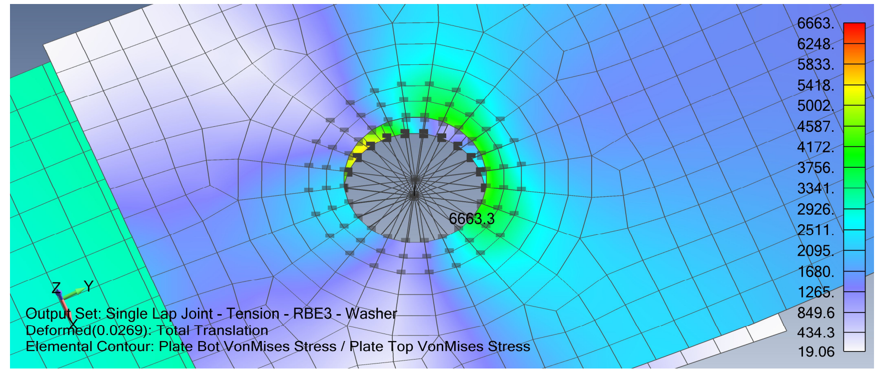
4.2.1.2 WASHER CONNECTION

RBE2	RBE3

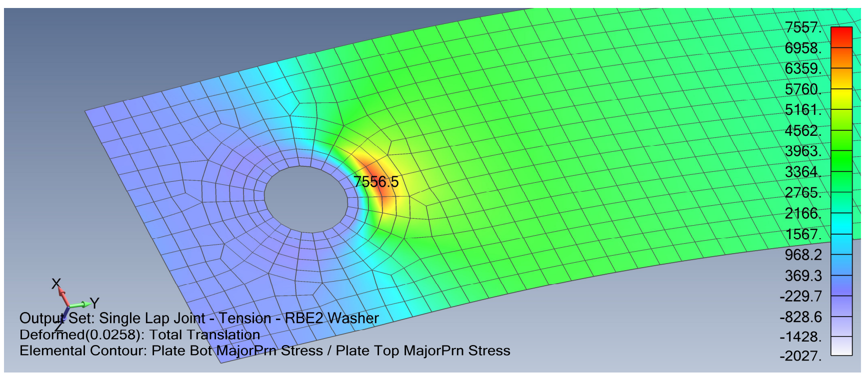
Von Mises 6,700 psi



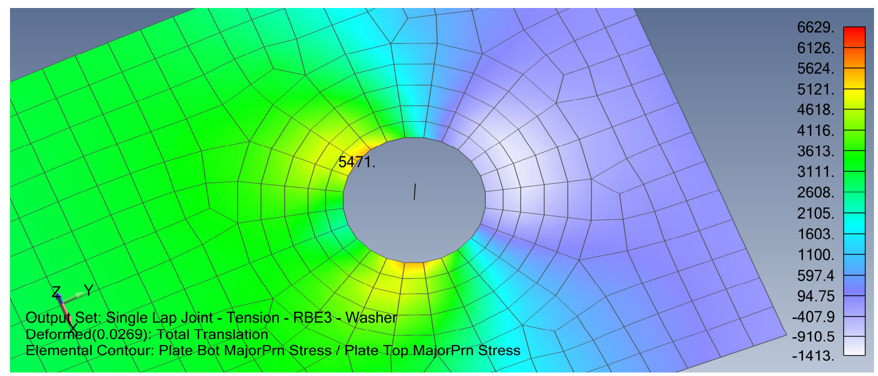
Von Mises 6,700 psi



Maximum Principal 7,600 psi



Maximum Principal 6,600 psi



4.2.2 SUMMARY: WHAT DO WE KNOW ABOUT BOLTED LAP JOINTS UNDER TENSION?

Idealization	Von Mises Stress, psi	Max. Principal Stress, psi
Plate with Hole – Tension Load		
Bearing Load Idealization - 180°	2,900	2,900
Bearing Load Idealization - 45°	not useful	3,400
RBE2	1,200	1,400
RBE3	2,500	1,700
Lap Joint Model - Tension		
3D Solid Model	18,000 (compressive)	6,600
2D Shell Model – Beam/RBE2 - Edge	9,000	10,000
2D Shell Model – Beam/RBE3 - Edge	8,700	7,000
2D Shell Model – Beam/RBE2 - Washer	6,700	7,600
2D Shell Model – Beam/RBE3 - Washer	6,700	6,600

4.2.3 Bending

4.2.3.1 Edge Connection

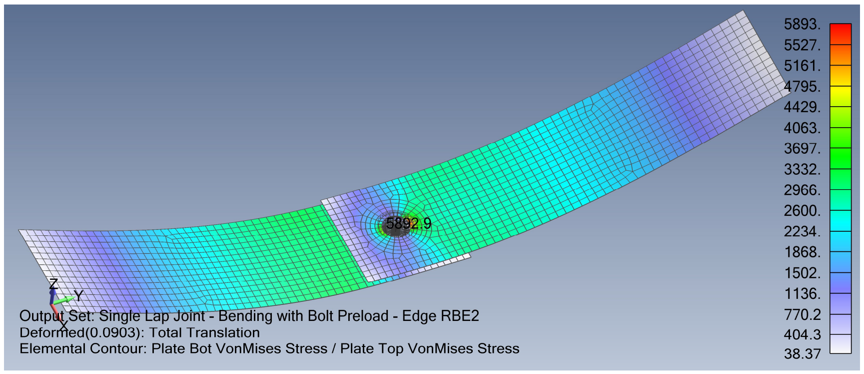
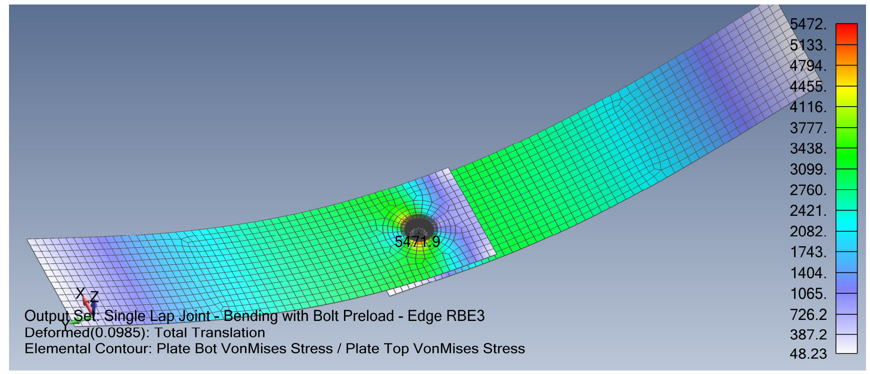
The same bending load as used in the 3D model is morphed over to the shell model.

4.2.3.1.1 Edge Connection / No Contact with No Bolt Preload

RBE2	RBE3
Von Mises 6,800 psi	Von Mises 6,400 psi
Maximum Principal 7,600 psi	Maximum Principal 6,100 psi

4.2.3.1.1 Edge Connection / Contact with Bolt Preload

BTW, with shell elements, the only out-of-plane stiffness we have is the bolt and some fuzzy contact spring stiffness. Thus, it is a far cry from reality where the stored elastic energy of the 3D plates (i.e., real structure) is part of the preload equation.

RBE2	RBE3
Von Mises 5,900 psi	Von Mises 5,500 psi
	
Maximum Principal 6,600 psi	Maximum Principal 5,500 psi

4.2.3.2 Washer Connection

4.2.3.2.1 Washer Connection / No Contact with No Bolt Preload

RBE2	RBE3
Von Mises 5,000 psi	Von Mises 4,900 psi
Maximum Principal 5,700 psi	Maximum Principal 4,800 psi

4.2.3.2.2 Washer Connection / Contact with Bolt Preload

RBE2	RBE3
Von Mises 4,700 psi	Von Mises 3,200 psi
Maximum Principal 5,300 psi	Maximum Principal 3,300 psi

4.2.4 Summary: What do we know about Bolted Lap Joints under Bending

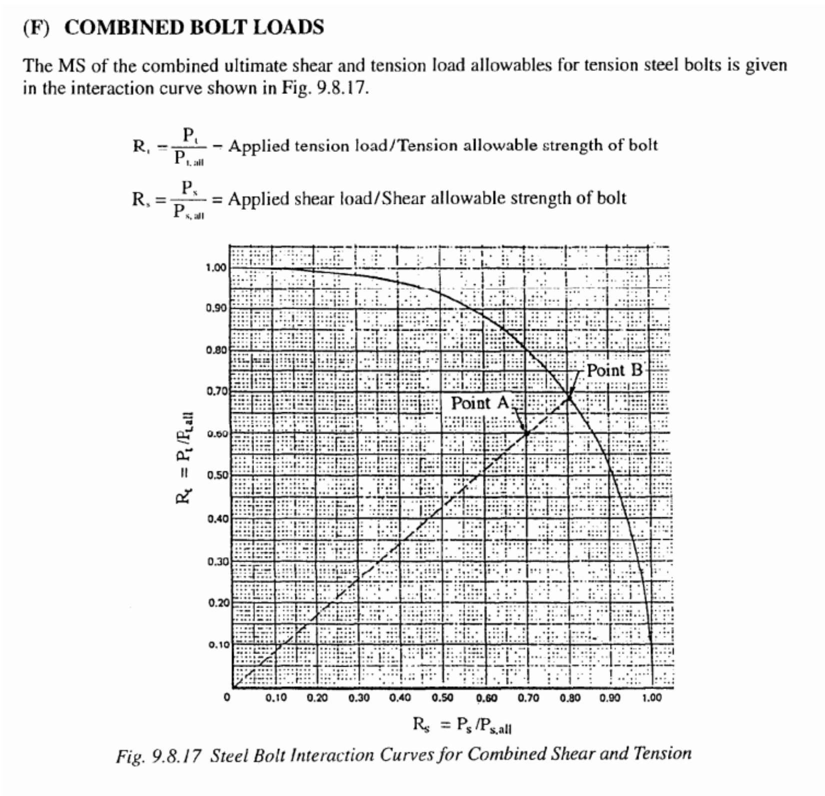
Idealization	Von Mises Stress, psi	Max. Principal Stress, psi
Lap Joint Model - Bending		
3D Solid Model with Bolt Snug (Contact)	4,200	4,000
3D Solid Model with Bolt Preload (Contact)	4,800	3,300
2D Shell Model – Beam/RBE2 - Edge	6,800	7,600
2D Shell Model – Beam/RBE3 - Edge	6,400	6,100
2D Shell Model – Beam/RBE2 – Edge w/Contact	5,900	6,600
2D Shell Model – Beam/RBE3 – Edge w/Contact	5,500	5,500
2D Shell Model – Beam/RBE2 - Washer	5,000	5,700
2D Shell Model – Beam/RBE3 - Washer	4,900	4,800
2D Shell Model – Beam/RBE2 – Washer w/Contact	4,700	5,300
2D Shell Model – Beam/RBE3 – Washer w/Contact	3,200	3,300

5. SOME COMMENTS

5.1 LET'S JUST CUT TO THE CHASE

What are some thoughts on how to handle a bolted connection?

Niu



(F) COMBINED BOLT LOADS

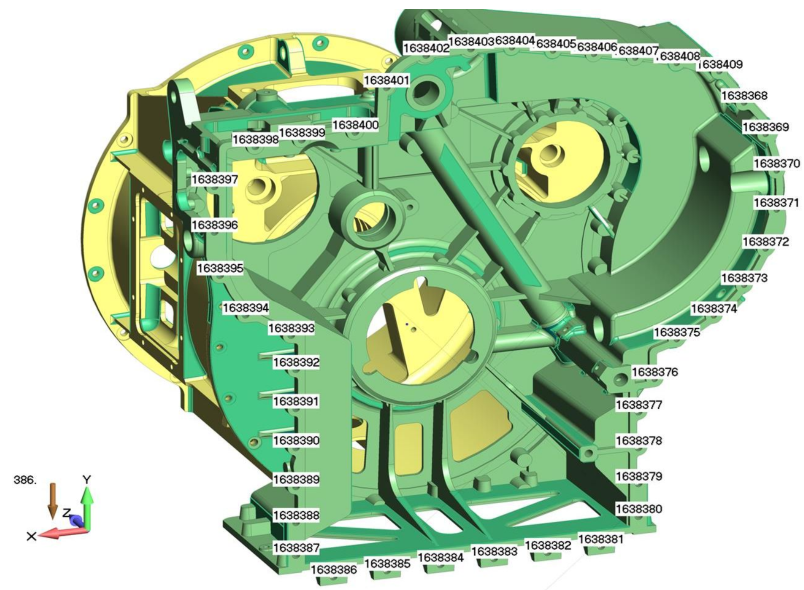
The MS of the combined ultimate shear and tension load allowables for tension steel bolts is given in the interaction curve shown in Fig. 9.8.17.

$R_t = \frac{P_t}{P_{t,all}}$ = Applied tension load/Tension allowable strength of bolt

$R_s = \frac{P_s}{P_{s,all}}$ = Applied shear load/Shear allowable strength of bolt

Fig. 9.8.17 Steel Bolt Interaction Curves for Combined Shear and Tension

Spreadsheet Analysis – Axial and Shear Forces



5.1.1 WHAT ABOUT BOLT BENDING STRESSES?

We don't. Although one can add a bending stress component by taking the shear force and assuming a contact gap between the bolt shank the plates (e.g., 0.01") and thus via your spreadsheet analysis, include a bending stress.

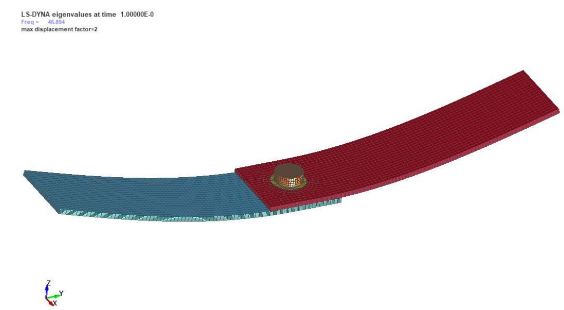
This information courtesy of EneavorAnalysis.com

5.2 What About Vibration: RBE2 or RBE3 or with Bolt Preload?

RBE2 (washer) 56 Hz

RBE3 (washer) 52 Hz

3D 47 Hz



5.2.1 What About Bolt Preload?

Yes – it can be added and yes it does change the results (see FEMAP User Guide Section 8.8.1.9). What is the right answer? Most likely closer to the 3D model.

Model	1st	2nd	3rd	4th	5th
RBE2	47[1] / 47[2] Hz	239 / 333	480 / 481	493 / 562	554 / 991
RBE3	43 / 44 Hz	234 / 318	432 / 444	480 / 540	530 / 937
3D	47 Hz	236	479	499	645
3D Transient	47 Hz	time step 0.001 sec, hence tight resolution to 100 Hz (10 points per cycle)			

[1]Regular linear analysis
[2]Pre-stiffened (bolt preload with contact)

5.3 Fatigue Analysis of Bolted Connections, i.e., the Bolt

We do our fatigue analysis on bolts via spreadsheet. It is not fancy but it gets the job done. What is the first step? Getting the information out of FEMAP and into the spreadsheet.

5.4 API of the Month