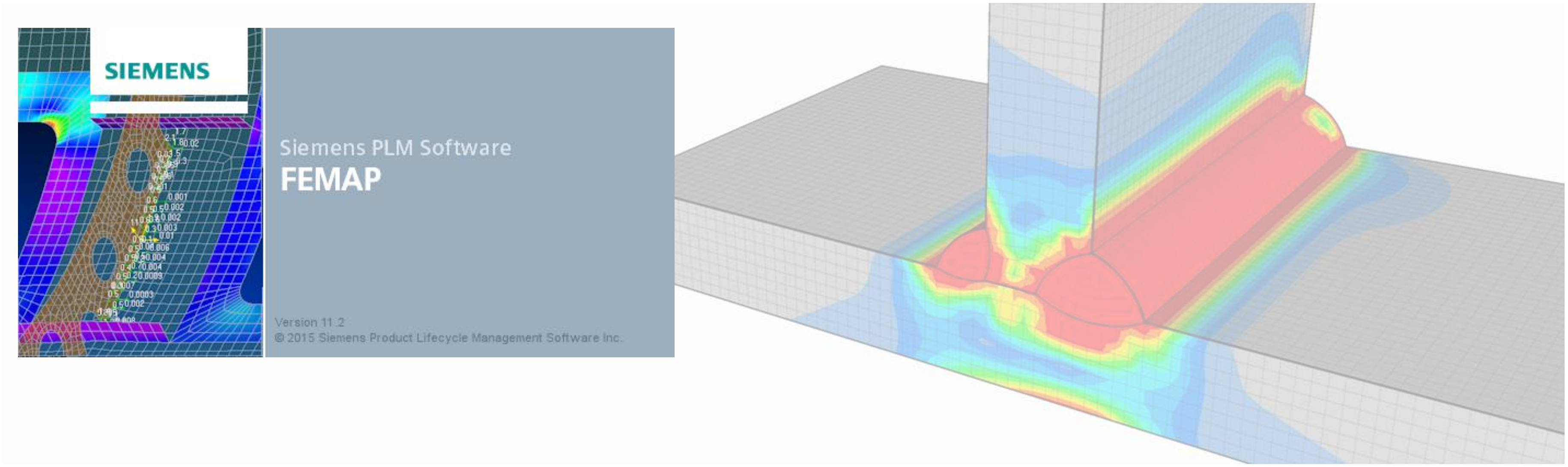

Weld Modeling and Analysis

A Seminar for FEMAP and NX Nastran Users

Adrian Jensen, PE | Senior Staff Mechanical Engineer
Kyle Hamilton, EIT | Staff Mechanical Engineer

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1. WARPING AND RESIDUAL STRESSES IN FILLET WELDED T-JOINT

How detailed do you really want to get? Figure 1 shows an example of a transient nonlinear stress analysis with temperature dependent mechanical properties. How accurate are the stresses? Well, that depends on how accurate the material properties are. Most likely, you're not going to find this information for your material-of-interest with a simple web search. Making things even more difficult is the fact that you will need to perform a coupled transient heat transfer analysis to get the right temperature profile. For 99% of the industrial analysts, this just isn't feasible. The intent of this seminar is to outline straight-forward procedures that provide conservative results in a reasonable amount of time.

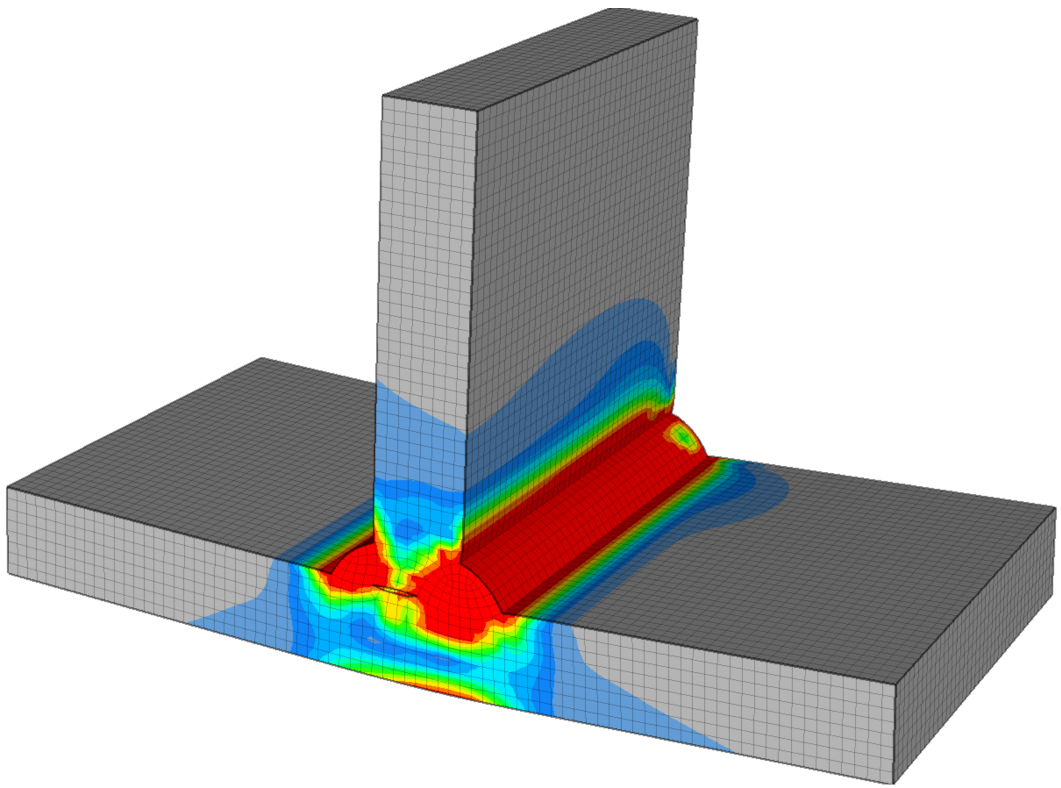


Figure 1: Fillet welded T-joint with weld induced warping and residual stresses

2. MODELING

Where to begin? Regardless of how complex your analysis will be, it all starts with the right model setup. Let's start with a 3D idealization of a structure. There are a few important things to keep in mind while preparing your geometry.

2.1 Partial penetration welds (i.e., internal free face)

Does your structure have fillet welds or full penetration welds? Not only will the type of weld affect the local stresses, it can change the stiffness of the joint and the load path through your structure. Figure 2 shows an example of the differences you might find.

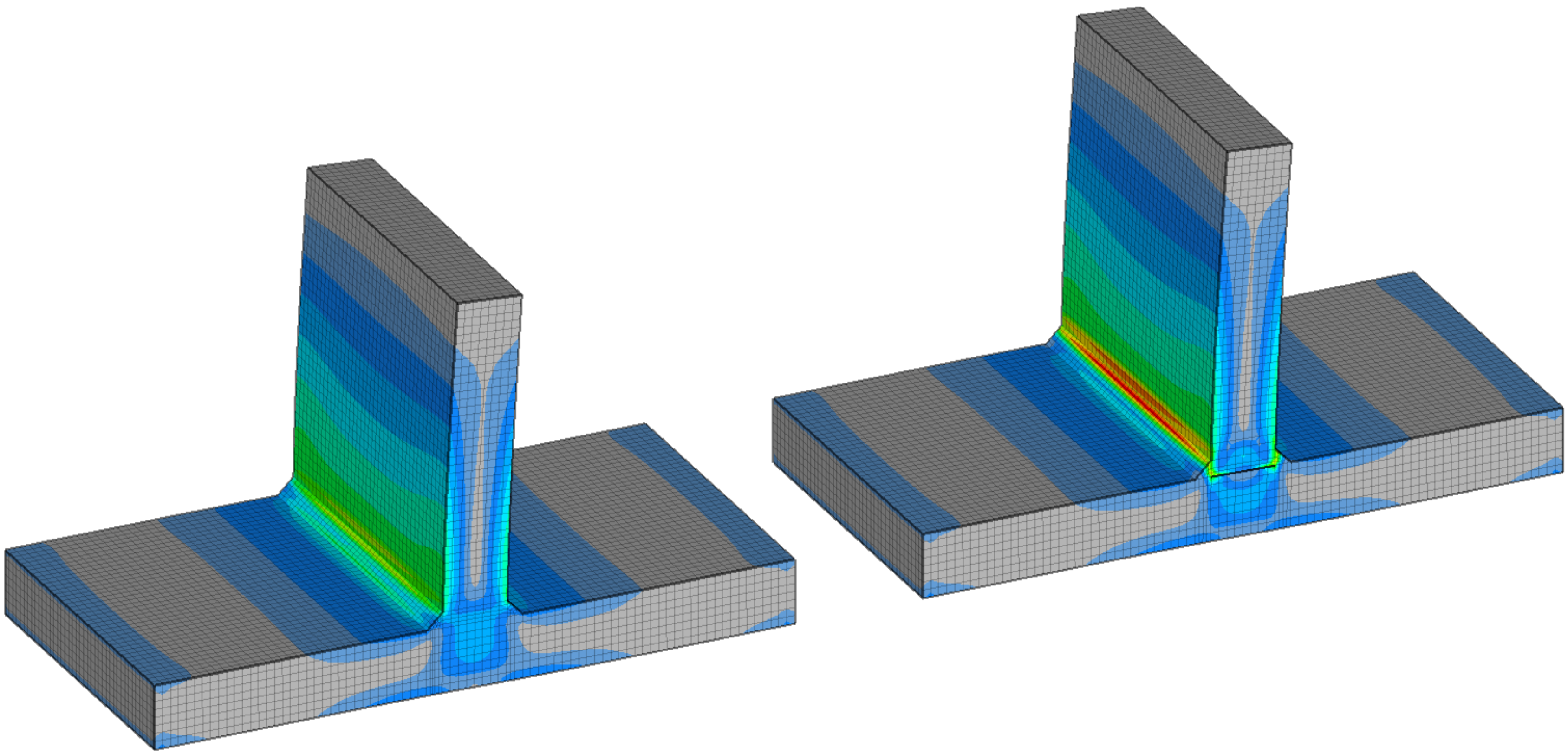


Figure 2: Full penetration weld vs fillet weld

The full penetration welded structure was created with standard hex meshing procedure. The fillet welded structure starts the same, but has an additional step to create the un-welded section between the fillets. Using Mesh > Mesh Control > Approach on Surface, one can remove the adjacent surface matching. The resulting free face can be visualized by viewing the mesh with transparency turned on.

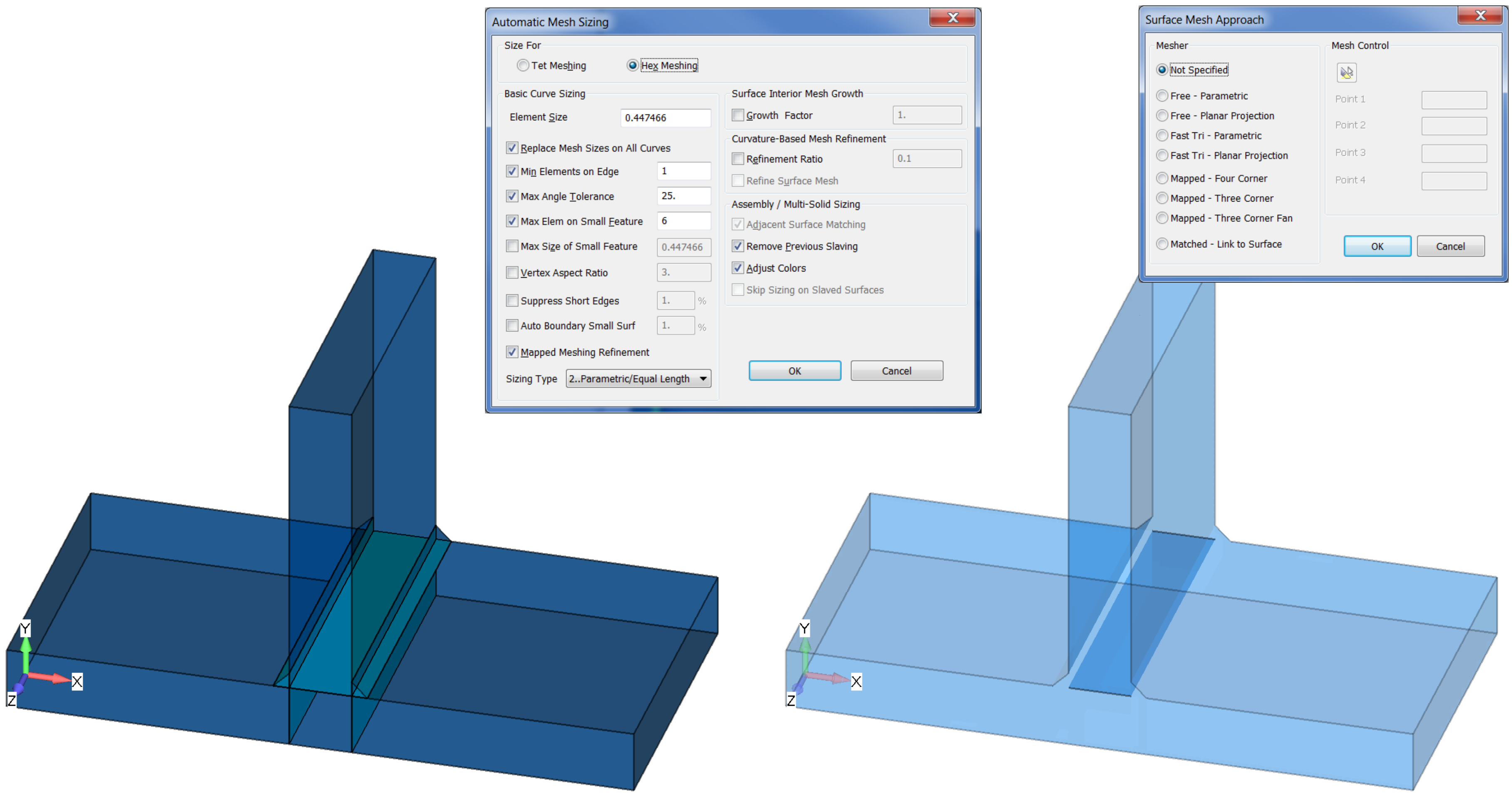


Figure 3: Fillet weld model setup

2.2 SHARP CORNER VS. CHAMFER VS. FILLET

While a good fabrication drawing will provide all of the information needed to fully define the welds, different fabricators will produce different welds. You can roughly capture the shape of the weld in the FEA model but the geometric details will never be a perfect match and therefore should not be agonized over. Figure 4 shows a T-joint with the weld modeled as a sharp corner, a concave face, a convex face and a chamfer. While the stresses at the weld change depending on the weld shape (and mesh density) the far field stresses are the same.

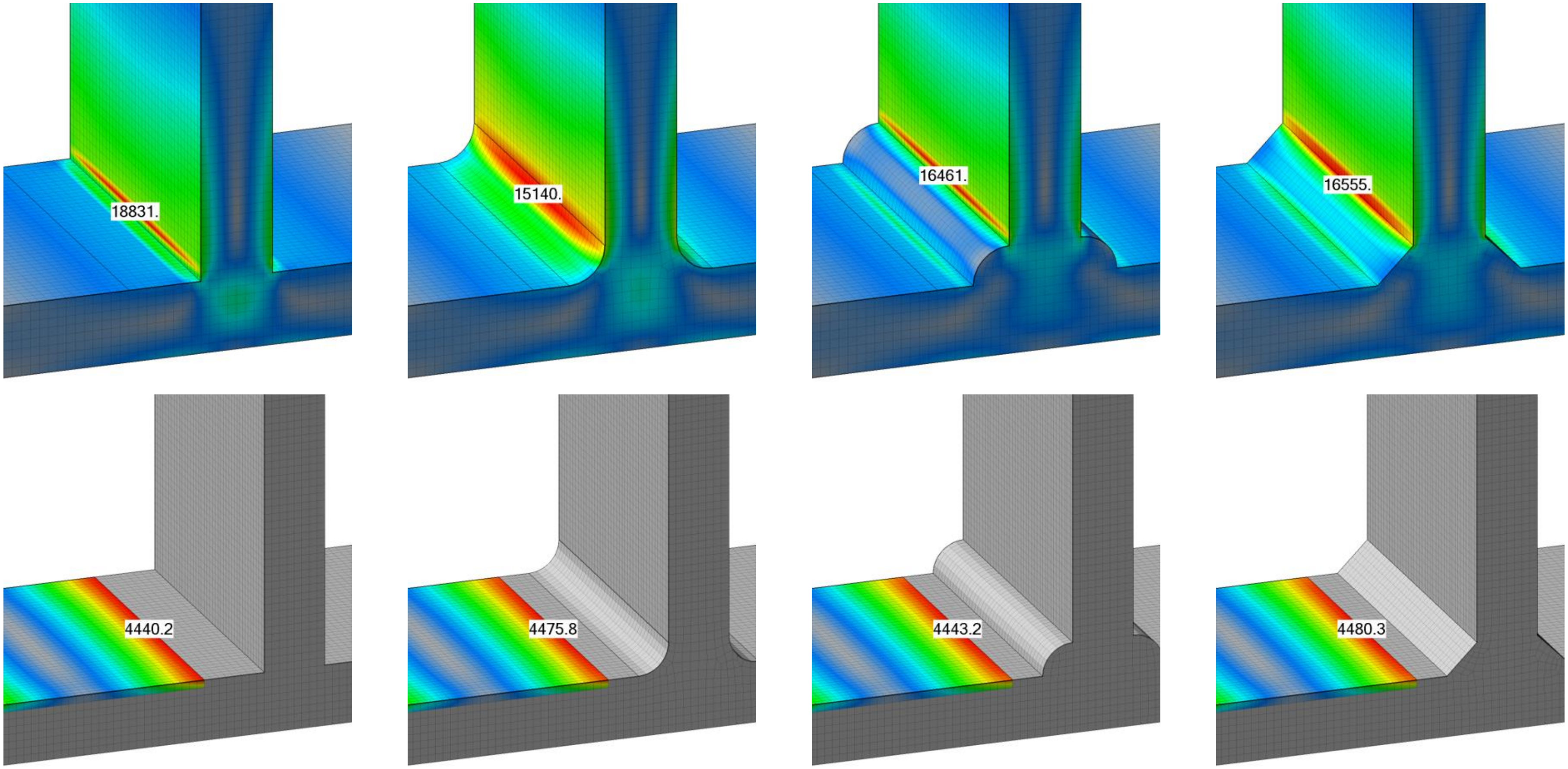


Figure 4: Various weld shape idealizations

2.3 Merged Nodes vs Glued connections

When joining 2D surfaces together, there are two major options to consider when setting up your model: put in the work extending the mid-surfaces and joining them together to create non-manifold geometry (resulting in merged nodes) or spend your time setting up glued connections. Both methods have pros and cons; the key is understanding when it makes a difference.

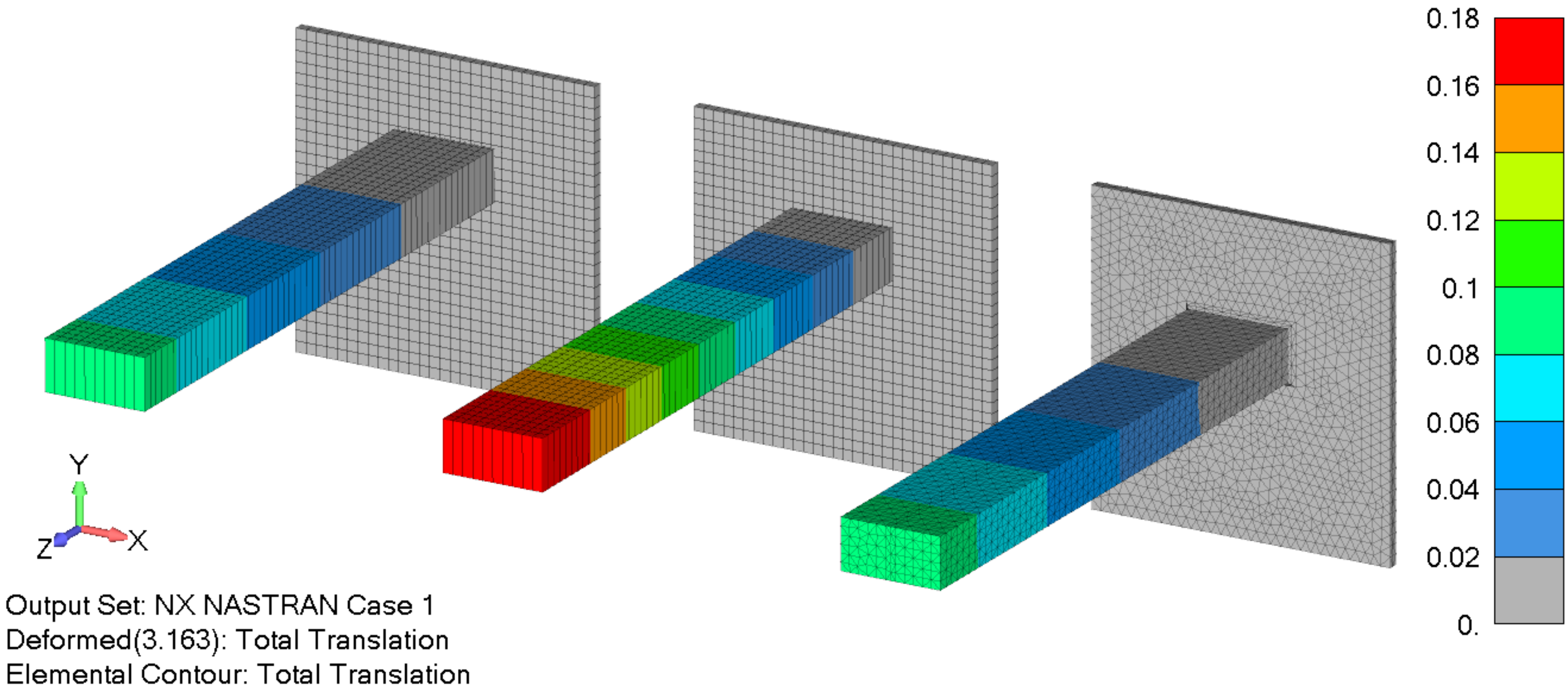


Figure 5: Cantilevered beam modeled with glued connection, merged nodes and solid elements

2.4 WELD ZONES (I.E., HAZ AND ASME SEP)

Whether you want to incorporate different material models or just organize your model, subdividing your surfaces into different zones will make life easier down the road. As demonstrated in Section 2.2, simple changes in the weld shape can have dramatic effects on the stresses. For this reason, most weld post-processing methods require recovering stresses away from the singularities found at the intersection of plates. For example, ASME BPVC Section VIII, Division 2 provides stress recovery points (SEPs), the first of which is located at the toe of the weld. To facilitate easier post-processing, incorporate these different regions into the mid-surface geometry.

Original CAD Geometry | **Mid-Surface Geometry** | **Finite Element Mesh**

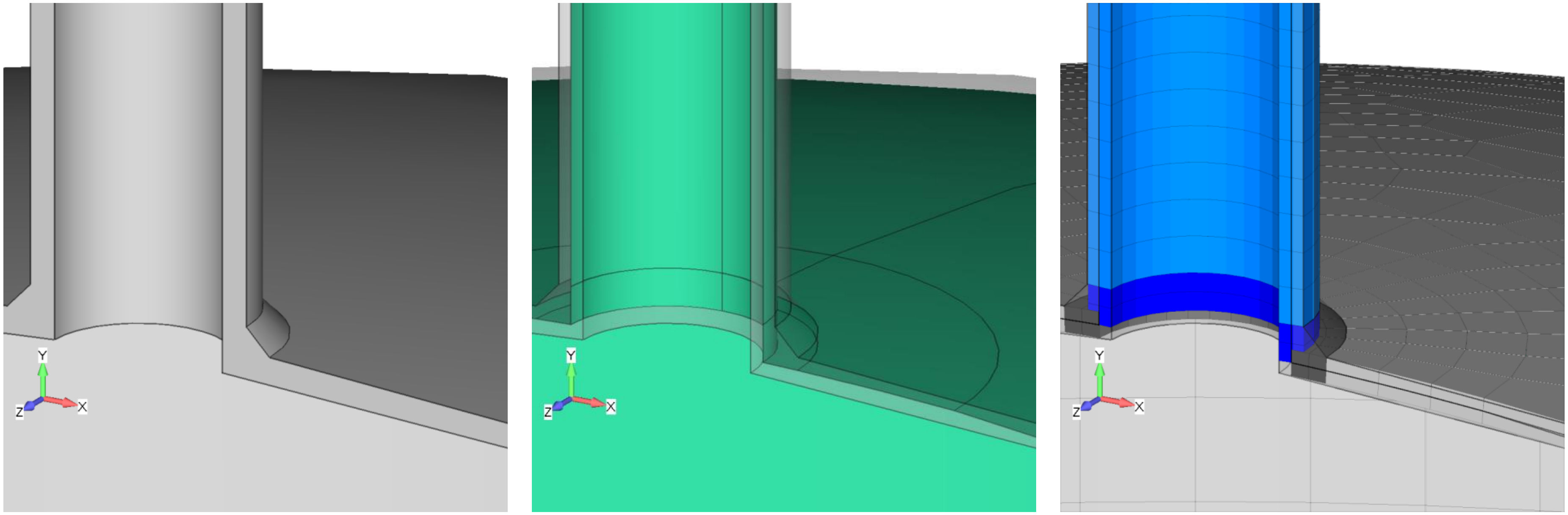


Figure 6: Pressure vessel nozzle modeled with various zones incorporated into the geometry and mesh

3. MESHING

Ok, we've decided how we want to idealize it. How do we mesh it? The most important thing to keep in mind is that local weld stresses cannot be determined using FEA. We have already explored the sensitivity of detailed weld geometry (and accepted that all idealization are wrong) but we must also consider material and numerical discontinuities.

3.1 THE MESH CONVERGENCE TRAP

We already know that chasing weld stresses is a great way to spin your wheels. Let's take a closer look with a mesh convergence study.

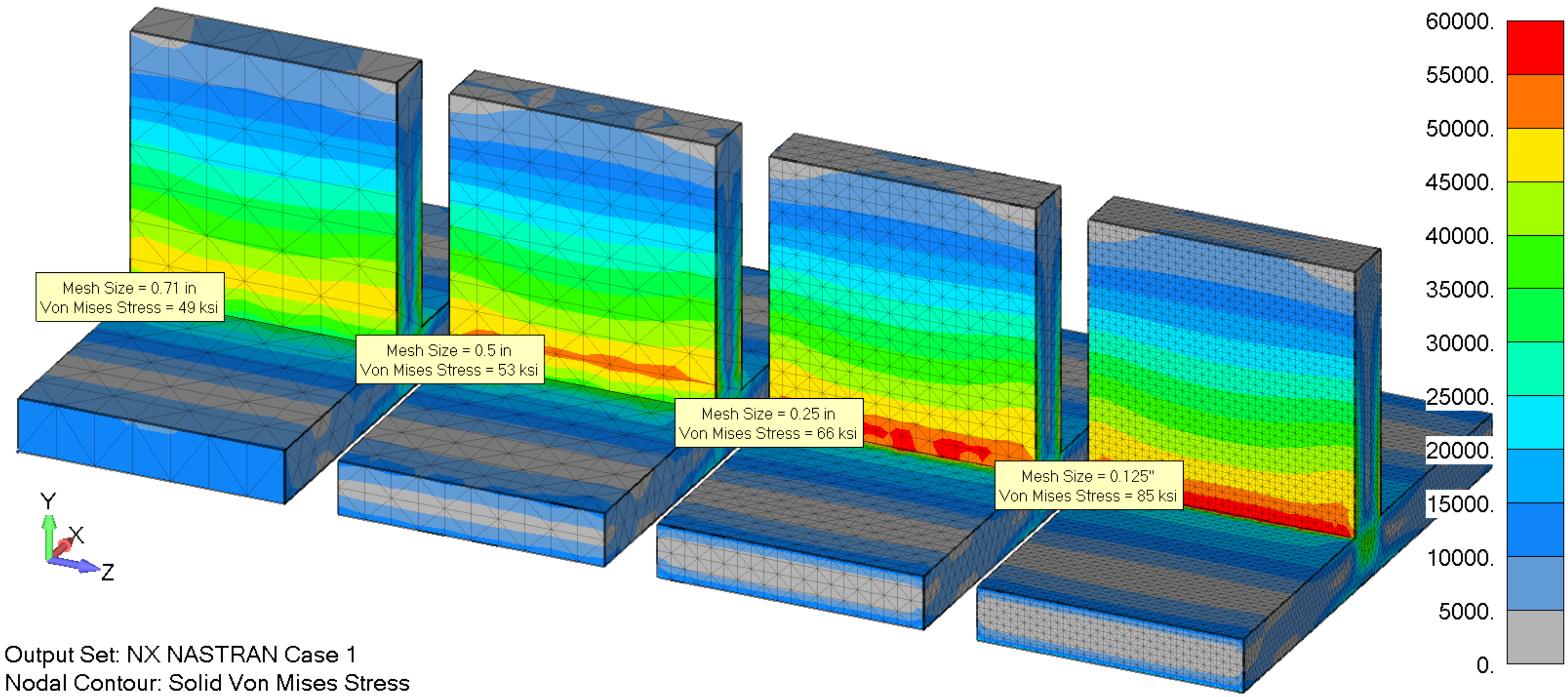


Figure 7: Mesh (non) convergence study

3.2 CONNECTING DISSIMILAR MESH TYPES

Nobody likes a sluggish FEA model. Now that we know there's no point in chasing stresses at the welds, let's be good analysts and take advantage of different element types. This will allow us to get the fastest possible model while maintaining accuracy. Here's a few examples of transitions between different element types.

Plate (2D) to Solid (3D)	Beam (1D) to Solid (3D)	Beam (1D) to Plate (2D)
Plate elements have six degrees-of-freedom (6DOF) while solid elements only have three. This introduces a problem as a T-joint now behaves like a hinge. Resolve this by embedding plate elements withing the solid, creating a small flange and merging nodes or by simply gluing parts together.	The same DOF discrepancy is a problem when connecting beams to solids, but worse. Since beams are 1D, a single node connection would behave like a spherical joint. Embedding elements works, but using RBEs to distribute the load over the footprint of the beam cross section is preferred.	While beams and plates both have 6DOF, one of the plate DOFs is a little less reliable than the rest. Resisting torque about the plate normal vector (also known as K6ROT or "drilling") should be avoided. Use an RBE to distribute the load over the footprint of the beam cross section.

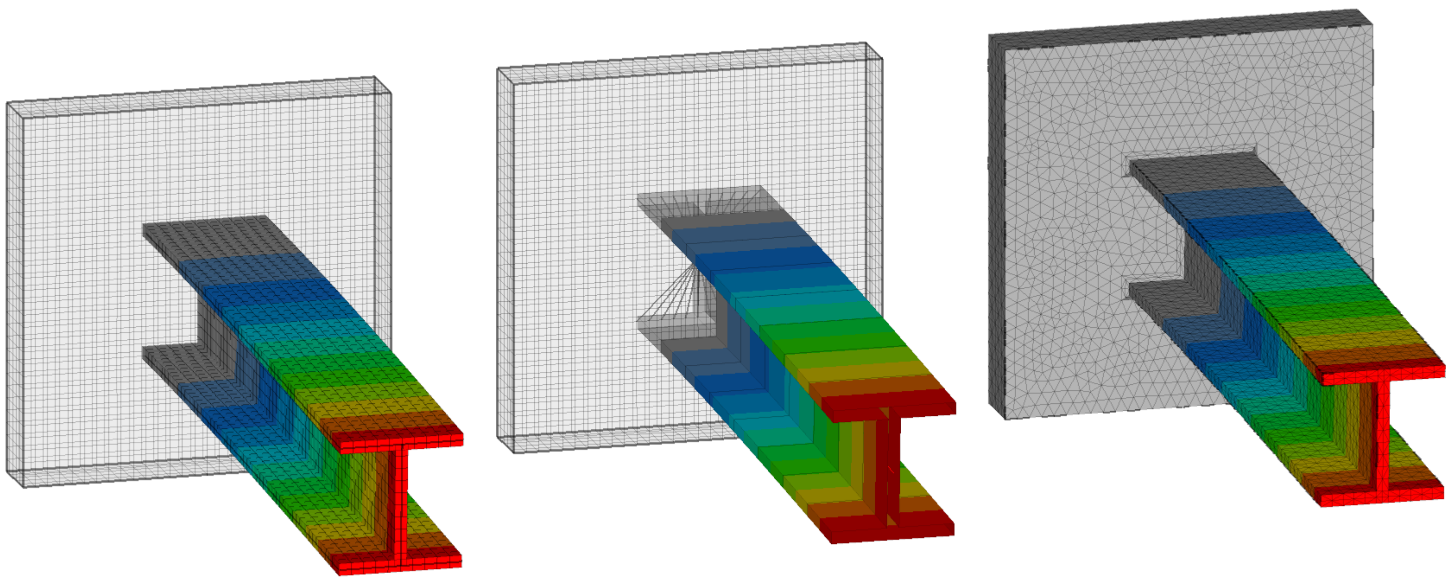


Figure 8: 2D-to-3D, 1D-to-3D and full-3D idealizations contoured with translation

3.3 Fillet welding (i.e., 3DOF)

When performing hand calculations on single-sided fillet welds, you might be left scratching your head when trying to figure out how to resolve those bending moments. In reality, a fillet weld cannot carry much bending load along the axis of the weld and structures should be designed to avoid this type of loading. Modeling this flexibility in an FEA model can be accomplished with 3DOF RBE2s.

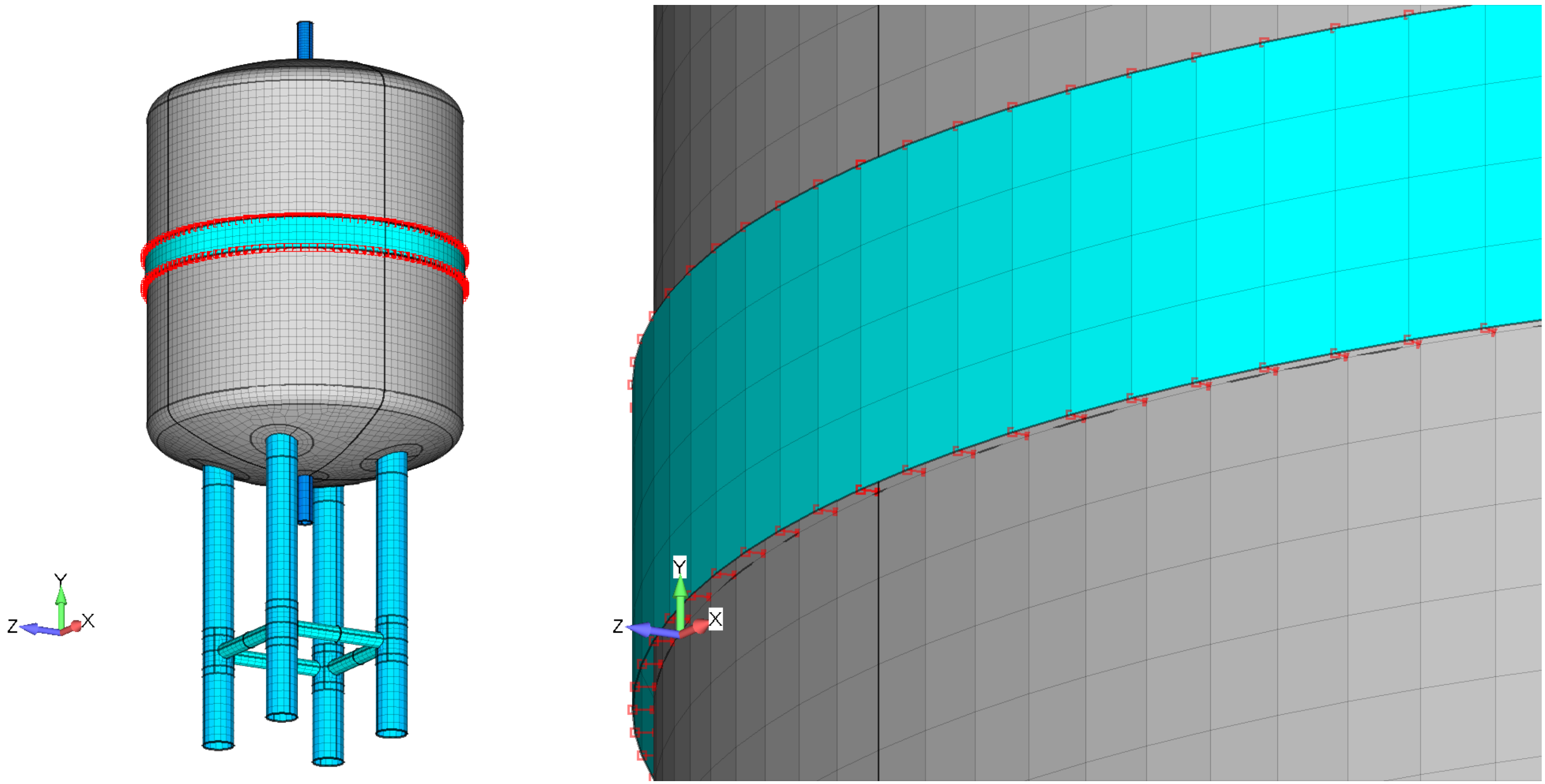
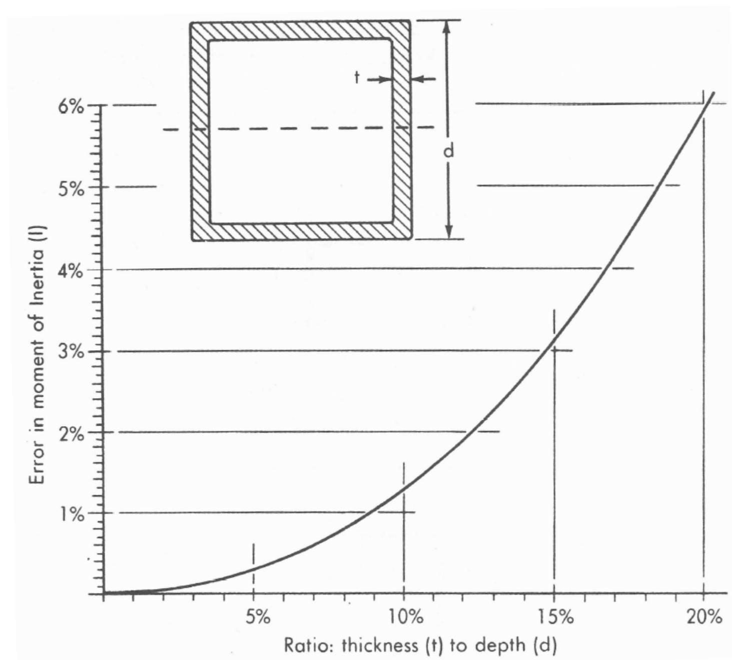


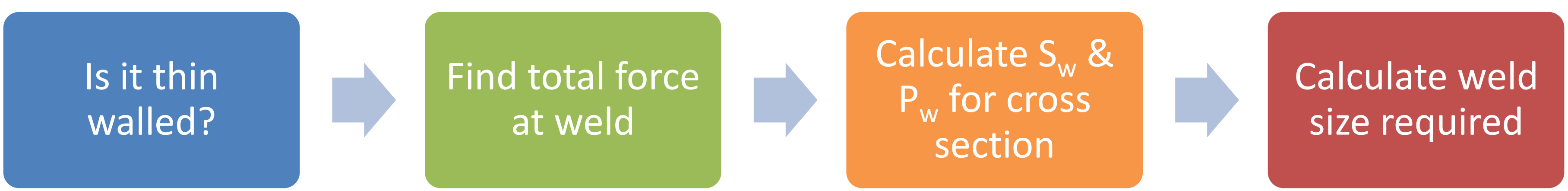
Figure 9: Pressure vessel with fillet welded belly band (repad)

4. POST PROCESSING

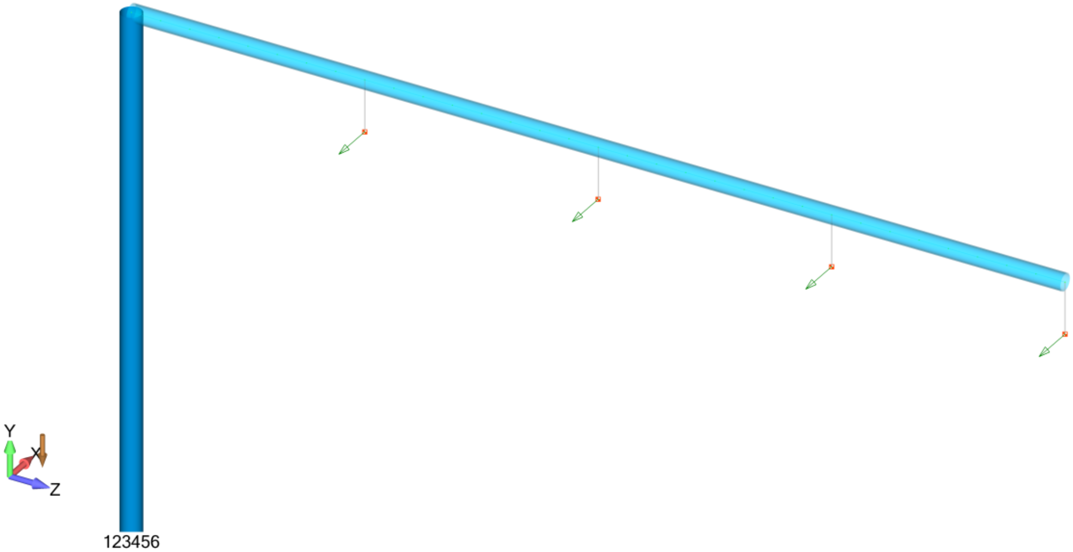
4.1 Weld Calculations – Forces on Beam Elements

In thin walled structures it is possible to simplify the analysis of the welds significantly by following Blodgett's method of evaluating the welded area as a line. The chart below shows an example for a thin walled square tube, where a 10% thickness to depth ratio results in approximately 1% error in the moment of inertia of the cross section.

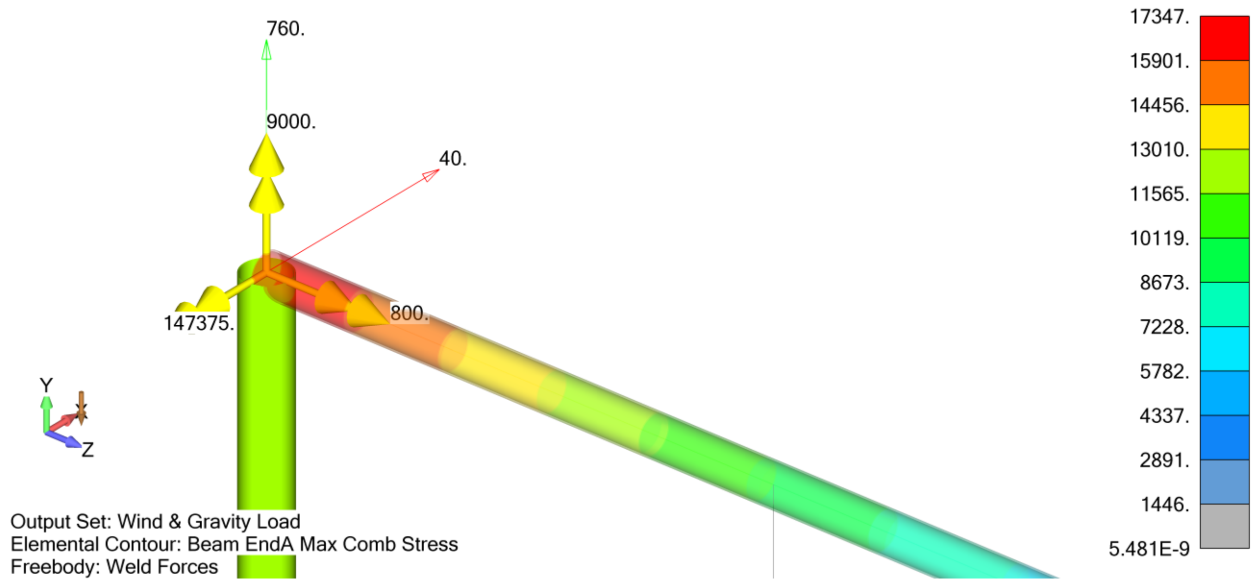




Let’s start by taking a look at a fairly common structure. The street light model below uses an 8.625 x 0.28” tube for the vertical support and a 6.625 x 0.28” tube for the horizontal support. The street lights are represented by mass elements placed along the horizontal beam. Forces are applied to the stoplight mass elements to represent wind loading.



The analysis results show a maximum stress at the base of the cantilever beam near the weld, right where it is expected. By adding a free body diagram to the structure at that node we can see the shear force and moment at the weld. The stress in the beam is 17,300 psi, but how does that translate to the stresses in the weld? At this point, we will grab our force and moment output from the free body diagram and input it to the hand calculations from Blodgett's.



$$F_1 = 40\ lb$$

$$F_2 = 760\ lb$$

$$F_3 = 0\ lb$$

$$M_1 = 147{,}375\ in \cdot lb$$

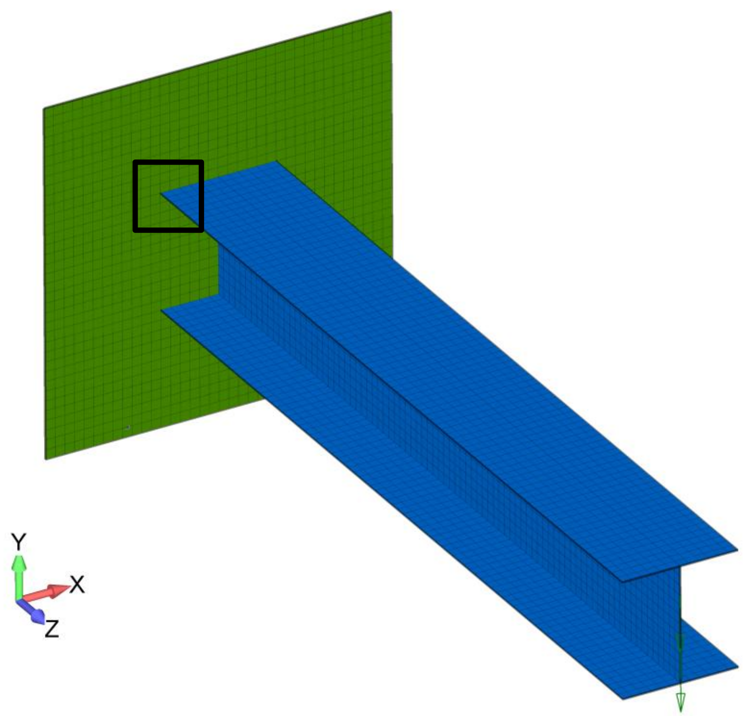
$$M_2 = 9{,}000\ in \cdot lb$$

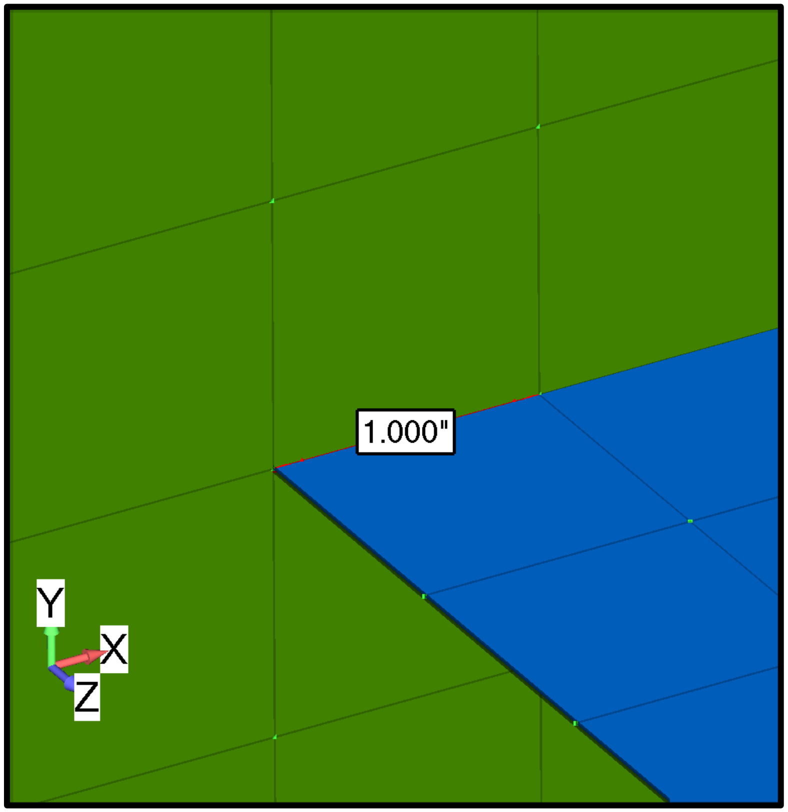
$$M_3 = 800\ in \cdot lb$$

Blodgett's hand calculations show that the minimum weld size is greater than the thickness of the structure, so some redesign is necessary. This is a case where the beam stress isn't concerning but the stress in the weld may be excessive.

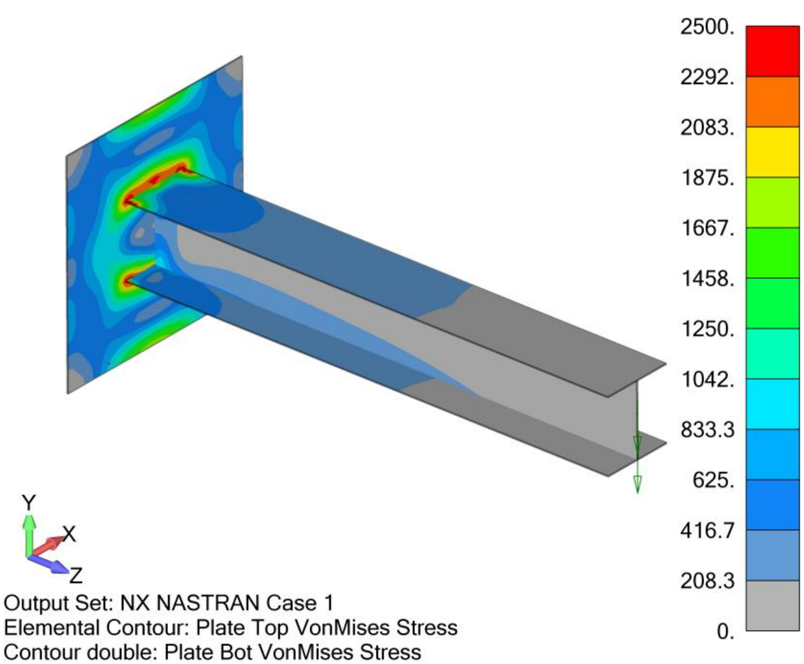
4.2 Weld Calculations – Forces on Nodes

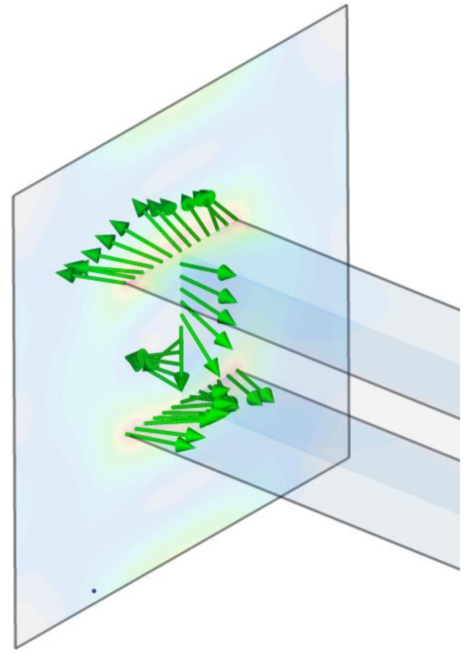
An alternative way to evaluate weld sizing on a more granular level is to directly pull out the weld force using free body diagrams. This takes a little model prep from the beginning. When you set up the model it is important to set mesh spacing to a consistent size to automate some steps later in the process.





We know that looking at the stresses does not provide reliable data for the welds, so we can pull nodal forces instead. With the force data one can build a spreadsheet with weld force per inch, and calculate the minimum weld size based on that.

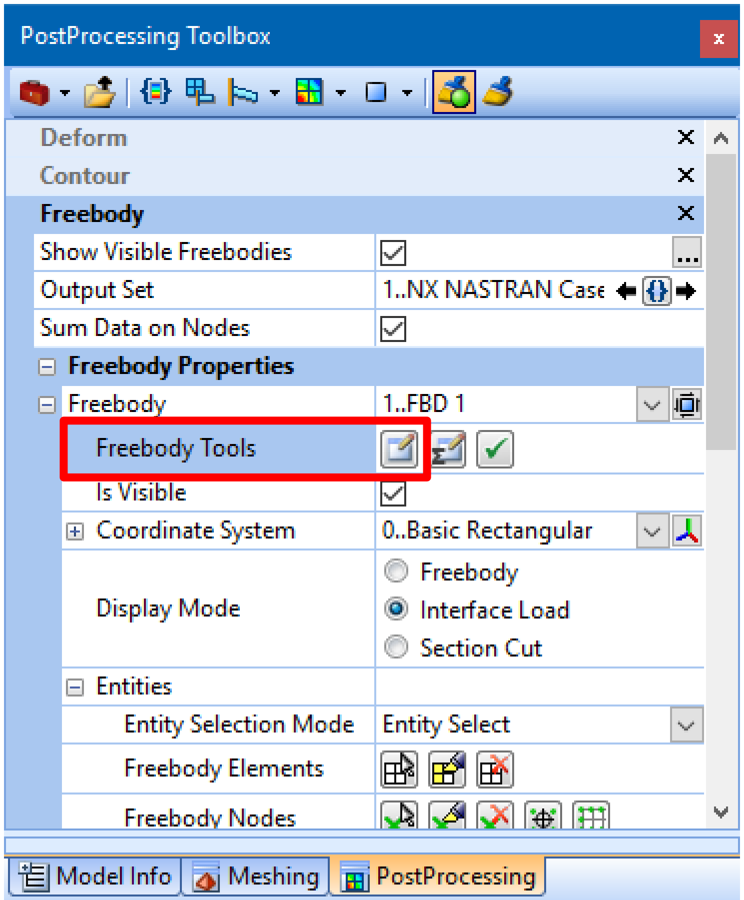


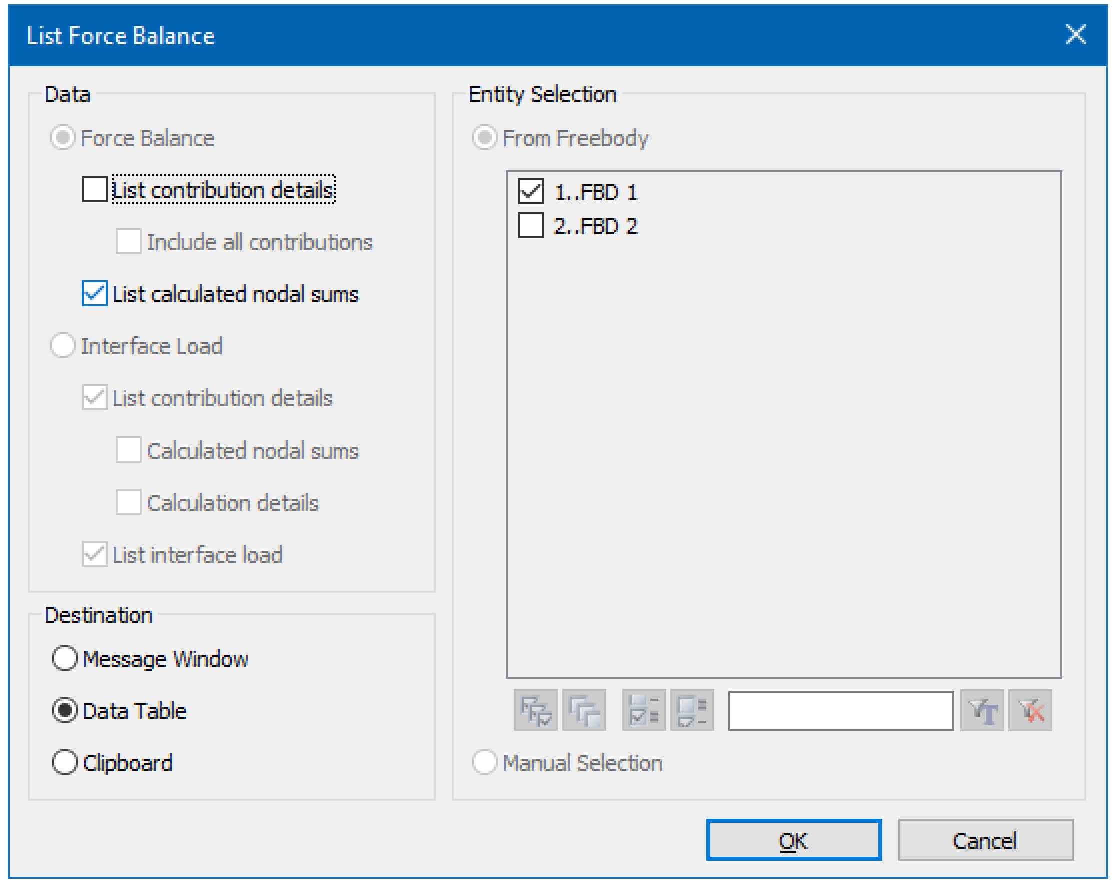


						Shear Allowable:	10000	psi	**Max Magnitude**	833.467		**Max Size**	0.472
Freebody ID	**Node ID**	**Output Set ID**	**CSys ID**	**Source**	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Magnitude of force**	**Weld Size**	
1	3327	1	0	**SU	0	232.1942	-392.69	-112.7	0	0	456	0.258	
1	3328	1	0	**SU	0	52.26164	308.891	-0.035	0	0	313	0.177	
1	3329	1	0	**SU	0	-18.6338	369.734	-0.007	0	0	370	0.209	
1	3330	1	0	**SU	0	-62.47629	283.085	0.0078	0	0	290	0.164	
1	3331	1	0	**SU	0	-84.03688	183.462	0.0146	0	0	202	0.114	
1	3332	1	0	**SU	0	-94.36764	88.6585	0.0175	0	0	129	0.073	
1	3333	1	0	**SU	0	-97.278	0	0.0182	0	0	97	0.055	
1	3334	1	0	**SU	0	-94.36764	-88.658	0.0175	0	0	129	0.073	

Nodal forces from the freebody diagram can be listed in the Data Table with a couple of clicks from the Freebody Post Processing Toolbox.

Select *List Force Balance*, then choose *List calculated nodal sums*, and the *Freebody entity* you are interested in.





4.2.1 SPREADSHEET: MINIMUM WELD SIZE

This is pretty great—you have forces and moments at each node listed which you can now use to calculate weld forces. Remember earlier when we set up the model we used constant node spacing, so it is trivial to convert the nodal forces into lbf/in. From the data table one can export the table to CSV and then process weld calculations in Excel or your favorite spreadsheet program.

$$\omega = minimum\ weld\ leg\ length\ [in]$$

$$\tau = shear\ stress\ allowable\ [psi]$$

$$f_w = weld\ force\ per\ linear\ inch\ [\frac{lbf}{in}]$$

$$\sigma = stress\ allowable$$

$$f_w = .707 * \omega * \tau$$

$$f_w = \sqrt{{f_{wX}}^2 + {f_{wY}}^2 + {f_{wZ}}^2}$$

$$\omega = \frac{f_w}{0.707 * \tau}$$

Data Table

Freebody ID / Output Set ID

Freebody ID: 1

Output Set ID: 1

Free...	No...	O...	CSys ID	Source	Fx	Fy	Fz	Mx	My	Mz
1	3327	1	0	**SUM*...	0.	232.1942	-392.6898	-112.7429	0.	0.
1	3328	1	0	**SUM*...	0.	52.26164	308.8912	-0.0349274	0.	0.
1	3329	1	0	**SUM*...	0.	-18.6338	369.7343	-0.00662994	0.	0.
1	3330	1	0	**SUM*...	0.	-62.47629	283.0854	0.00783157	0.	0.
1	3331	1	0	**SUM*...	0.	-84.03688	183.4623	0.0146303	0.	0.
1	3332	1	0	**SUM*...	0.	-94.36764	88.65848	0.0174611	0.	0.
1	3333	1	0	**SUM*...	0.	-97.278	0.	0.0182243	0.	0.
1	3334	1	0	**SUM*...	0.	-94.36764	-88.65848	0.0174611	0.	0.
1	3335	1	0	**SUM*...	0.	-84.03688	-183.4623	0.0146303	0.	0.
1	3336	1	0	**SUM*...	0.	-62.47629	-283.0854	0.00783157	0.	0.
1	3337	1	0	**SUM*...	0.	-18.6338	-369.7343	-0.00662994	0.	0.
1	3338	1	0	**SUM*...	0.	52.26164	-308.8912	-0.0349274	0.	0.
1	3339	1	0	**SUM*...	0.	232.1942	392.6898	-112.7429	0.	0.
1	3340	1	0	**SUM*...	49.6758	79.44401	235.6547	-92.63086	0.000118256	0.0345526
1	3341	1	0	**SUM*...	47.12846	31.87773	151.6801	-71.4653	0.00261688	0.0467247
1	3342	1	0	**SUM*...	62.49435	17.1811	115.6914	-58.63513	0.00947571	0.030397
1	3343	1	0	**SUM*...	106.8502	14.23273	137.0038	-50.97841	0.0213928	0.0306327
1	3344	1	0	**SUM*...	172.7437	14.21411	425.2362	-42.12747	0.0420532	0.0266298

					Shear Allowable:	2750	psi	Max Magnitude	133.621	Max Size	0.275 in
Node ID	**Output Set ID**	**CSys ID**	**Source**	**Fx**	**Fy**	**Fz**	**Mx**	**My**	**Mz**	**Magnitude of force**	**Weld Size**
1777	1	0	**SU	0.27	-92.56	-60.26	-0.64	7.68	0.00	110.445	0.227
1778	1	0	**SU	-2.78	-92.71	-66.54	-0.54	8.87	0.00	114.150	0.235
1779	1	0	**SU	-6.36	-93.01	-73.64	-0.42	10.33	0.00	118.804	0.244
1780	1	0	**SU	-10.26	-92.43	-82.84	-0.29	12.26	0.00	124.543	0.256
1781	1	0	**SU	-16.19	-90.06	-95.37	0.00	14.55	0.00	132.171	0.272
1782	1	0	**SU	-17.84	-80.41	-105.22	1.08	16.79	0.00	133.620	0.275
1783	1	0	**SU	-15.51	-73.54	-91.77	2.47	16.20	0.00	118.617	0.244
1784	1	0	**SU	-4.22	-65.79	-65.37	3.88	12.09	0.00	92.842	0.191
1785	1	0	**SU	9.16	-56.33	-52.36	3.36	4.85	-2.55	77.447	0.159
1786	1	0	**SU	4.47	2.90	-32.04	0.01	0.20	-0.45	32.480	0.067
1787	1	0	**SU	5.87	19.14	-30.99	0.00	-0.62	1.43	36.891	0.076
1788	1	0	**SU	5.08	21.04	-32.56	0.00	-0.93	2.14	39.103	0.080
1789	1	0	**SU	3.71	20.35	-33.33	0.00	-0.96	2.21	39.233	0.081
1790	1	0	**SU	2.64	18.97	-33.45	0.00	-0.89	2.04	38.540	0.079
1791	1	0	**SU	1.85	17.64	-33.11	0.00	-0.77	1.77	37.559	0.077
1792	1	0	**SU	1.29	16.45	-32.43	0.00	-0.65	1.49	36.387	0.075
1793	1	0	**SU	0.91	15.39	-31.54	0.00	-0.53	1.22	35.106	0.072
1794	1	0	**SU	0.65	14.42	-30.45	0.00	-0.43	0.98	33.696	0.069
1795	1	0	**SU	0.47	13.52	-29.19	0.00	-0.33	0.77	32.166	0.066
1796	1	0	**SU	0.33	12.62	-27.71	0.00	-0.25	0.58	30.456	0.063
1797	1	0	**SU	0.15	11.70	-26.00	0.00	-0.19	0.43	28.514	0.059
1798	1	0	**SU	-0.22	10.71	-23.93	0.00	-0.13	0.30	26.215	0.054
1799	1	0	**SU	-1.13	9.62	-21.41	0.00	-0.08	0.19	23.501	0.048

4.2.2 Freebody Diagrams: Maximum Nodal Force

Another way to utilize the free body diagrams and weld sizing is to work backwards through the process. If you calculate your maximum weld force based on the known maximum weld size, you can use free body diagrams to quickly highlight the areas of your structure where weld force exceeds the allowable force.

Fillet Weld: Maximum Weld Force for Weld Size

$$\omega = maximum\ weld\ leg\ length\ [in]$$

$$\tau = shear\ stress\ allowable\ [psi]$$

$$f_w = weld\ force\ per\ linear\ inch\ [\frac{lbf}{in}]$$

$$f_n = weld\ force\ per\ node\ [\frac{lbf}{node}]$$

$$f_w = 0.707 * \omega * \tau$$

$$f_w = 0.707 * 0.25" * 2{,}750\ psi = 486\frac{lbf}{in}$$

$$f_n = 486\frac{lbf}{in} * \frac{1\ inch}{4\ nodes} = 121\frac{lbf}{node}$$