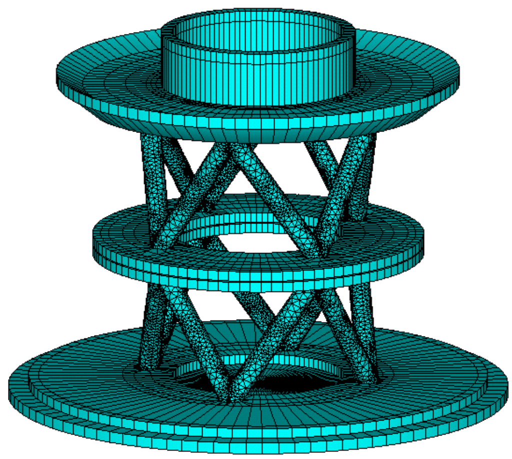
Analysis of the NSTX Upgrade Centerstack Support Pedestal

NSTXU-CALC-12-09-01

Rev 0 May 2011

Rev 1 July 2014



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PPPL Calculation Form

Calculation # <u>NSTXU-CALC-12-09-01</u> Revision # <u>01</u> ____ WP #, <u>1672</u>
(ENG-032)

Purpose of Calculation: (Define why the calculation is being performed.)

The purpose of this calculation is to qualify the stresses in Pedestal support for the centerstack assembly. Additionally, the effect of the torsional stiffness of the pedestal will be assessed.

References (List any source of design information including computer program titles and revision levels.)

Included in the body of the calculation

Assumptions (Identify all assumptions made as part of this calculation.)

At the time this calculation was prepared, the torsionally stiff Vee-Pipe pedestal w as coupled with a "bent spoke" lid that carried torques either through the cell floor or through the bellows. While analysis of this configuration did not show excessive bellows torsional shear, there was a concern that alignment of the center stack, slippage at the concrete anchors and the lower halo currents on the centerstack could stress the bellows. As a result a stiffer lower spoked lid was added . This final design is closer to the CDR and PDR global models that included a more compliant pedestal and a stiff diaphragm or plate lower lid. Consequently results of both pedestal concepts are included. Stresses in the Vee-Pipe pedestal are inferred from available models and it is assumed that net loads and torques are adequately enveloped by the global model analyses [2] with a compliant lower spoked lid, and a "stand-alone" model to which loads from the design point spreadsheet can be applied directly.
In July of 2011, analysis was added of the pedestal based on a flat lower spoked lid. This confirmed the assumptions discussed above.

Other Assumptions are included in the body of the report

Calculation (Calculation is either documented here or attached)

See the following report

Conclusion (Specify whether or not the purpose of the calculation was accomplished.)

Stress levels in the support satisfy the NSTX CSU criteria. Torsional stiffness of the pedestal has minimal effect on the torsional shear stress in the TF inner leg. Torsional moments are computed and bolt shear stresses have been updated in a new section 10.0 in Rev 1 of the calculation. Pedestal "Vee" stresses have been found to be compressive where peak stresses develop and fatigue is not expected to be a concern.

Cognizant Engineer's printed name, signature, and date

Mark Smith Mark Smith Digitally signed by Mark Smith DN: cn=Mark Smith, o=PPPL, ou, email=msmith@pppl.gov, c=US Date: 2014.07.24 10:53:34 -04'00'

I have reviewed this calculation and, to my professional satisfaction, it is properly performed and correct.

Checker's printed name, signature, and date

Han Zhang Han Zhang Digitally signed by Han Zhang DN: cn=Han Zhang, o, ou=PPPL, email=hzhang@pppl.gov, c=US Date: 2014.07.24 09:32:43 -04'00'

2.1 Table of Contents
NSTX Centerstack Pedestal

2.2 Revision Status Table

Rev 0	Initial Issue
Rev1	Radius in Figure 9.2.2-4 corrected to .35 meter from .7 m – loads re-calculated
Rev. 1	Added Section 10, Bolting Calculations, Anchor Loads
Rev 1	Added Section 5.4 that includes final drawing details
Rev 1	Added Ref 16 in Reference List and in Attachments
Rev1	Added Section 5.1, Criteria , which was missing.

Rev 1	Added final drawings in section 5.4
Rev 1	Added Ref 17 which includes the halo loads at the pedestal top plate
Rev 1	Added Attachment 4 Unisorb Data (Sent by Mark Smith)
Rev1	Added Figure 9.1-2 showing the Deadweight reactions

3.0 Executive Summary:

The pedestal is a structure that provides gravity support for the centerstack and resists Coil Lorentz loads during operation.. Because it is connected to ground, the lower lid assembly, and the TF flags, and the skirt which supports the centerstack casing, it also is a contributor to the torsional stiffnesses that determine the distribution of the global torques in the machine. The pedestal must allow access to the service connections at the lower end of the centerstack. Provision must be made to allow passage of coolant lines, power leads and diagnostics. In order to service these lines, the pedestal may have to be able to be disassembled in pieces that do not capture the service connections. The current design for the FDR is shown in figure 3.0-1. The number of bolts at the mid flange is 6 pairs - but this was described as needing resolution in an email from Mark Smith[10]. The analysis model uses four bolts in a pattern around the vertices of the trusses for a total of 8 pairs. Shimming of the mid flanges is assumed to also align with the vertices of the trusses. Use of high strength bolts at the flange connections (Mid height and at the base) allows these connections to be capable of resisting the worst case power supply loads. The limit to the upward loading is the concrete anchors. Ninety four 3/4 inch anchors are required to resist the worst case power supply loads. It is not likely that this number will be used. Only five 3/4 inch anchors are needed to react the normal operating net load on the centerstack. Many more than 5 are suggested. This number will set the limit that must be maintained by the DCPS.

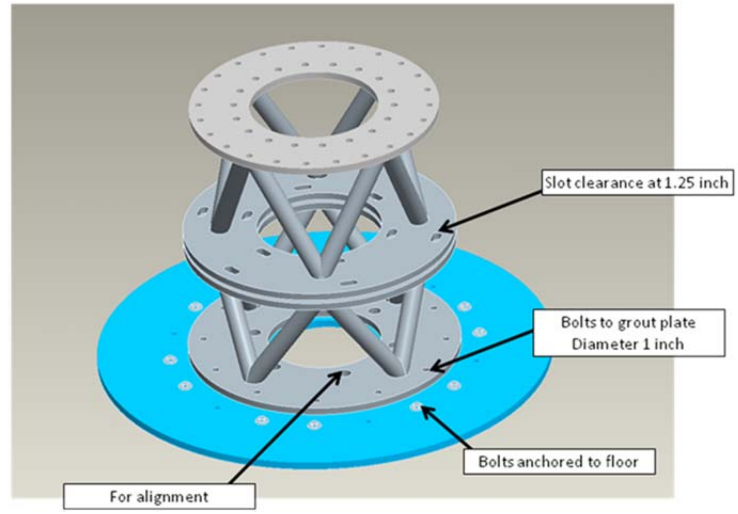
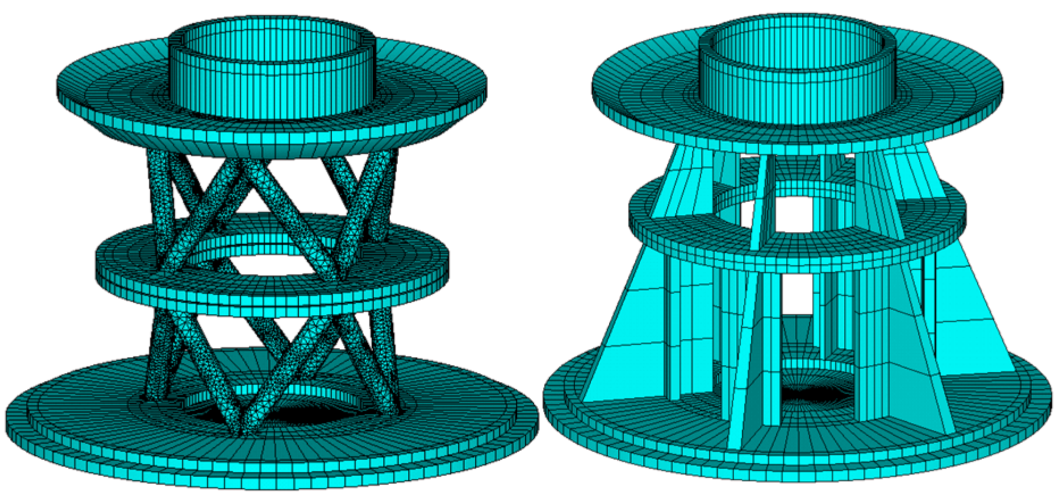
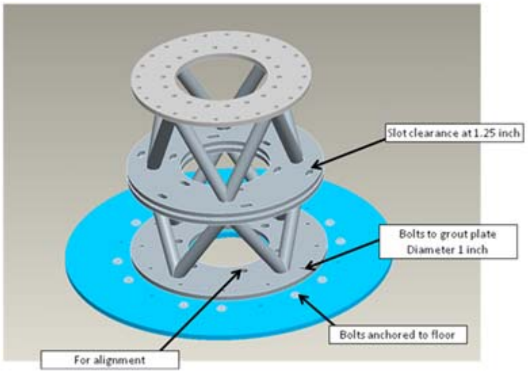


Figure 3.0-1 "Vee" Pipe Pedestal Evaluated for the FDR

There have been a couple of design concepts proposed for the pedestal. During the CDR, the pedestal was a bolted plate assembly. A number of analyses were performed based on this configuration, and the gusseted plate design was acceptable. Designers were concerned that a torsionally stiffer structure was needed, although the analyses (which also had a stiff lower lid structure) did not show this.





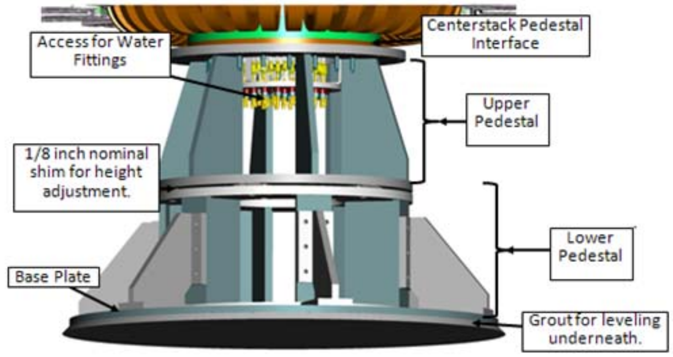


Figure 3.0-2 Two Concepts Proposed for the Pedestal; "Vee" Pipe (Left) and Gusseted Plate (Right)

Aside from qualifying the present Vee-tube structure , the global model used for the inner leg torsional shear calculation has been run with both the plate and vee-tube structure.

Section 9 is from the global model that includes the TF OOP loading, and the TF torque. It is the same model and analysis used for the TF torsional shear. Global machine torque is included. Global torque effects are discussed in a couple of places in the calc - Torque reactions at the base and bolt circle and in the discussion of the different stresses in the Vee legs indicating that a torque was being reacted by the pedestal.

If you look at the figure 9.2-4 you don't see a red contour. - the highest you see is a light green - around 135 level - stresses beyond this are very localized at the intersections and are an indication of the FEA capturing the stress concentration factor. The stress components in the pipes are shown in section 9.2.3, figures 9.2.3-1, and 2. The important observation is that the primary load in the pipes is compressive which, regardless of the stress concentration, will not allow cracks to propagate.

Also included in 9.2.3, plots from the latest global model run are included for a number of the equilibria - some go above the 135 MPa level, but not significantly. The 135 MPa quote is a reasonable reading of the contours, with the higher values more indicative of the very local peaks at the intersections. Section 9.2.3 shows results for the flat lower spoked lid. The pedestal stresses are varying a bit based on the lower lid design, but basic conclusions are not altered.

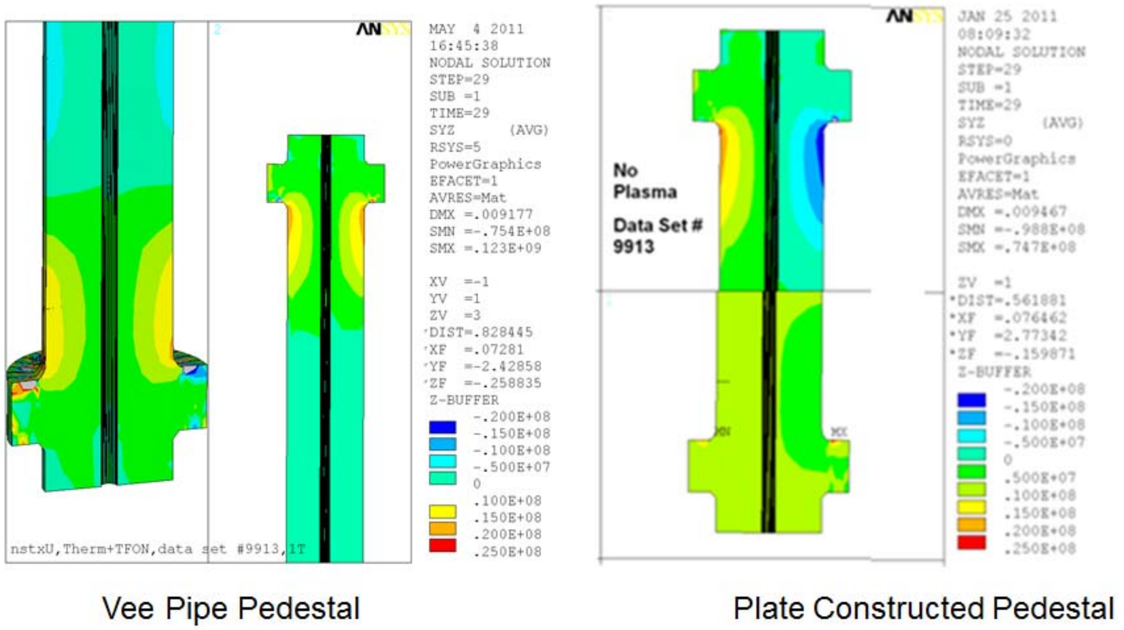


Figure 3.0-3 Inner Leg Torsional Shear For Two Pedestal Concepts

After reviewing results for the different pedestal designs, and a few scenarios, there is no difference in the max TF inner leg torsional shear of 25 MPa, but there is a difference in the shear in the lower end of the TF inner leg. This implies that there is a difference in torques transmitted via the TF flags and crown to the pedestal and lower spoked lid. For both these components, the torques have been based on an upper bound for the upper connections which have been found to be larger. So it is likely that the re-distribution of torque that is caused by the "Vee" Pipe pedestal will not be a problem, but rigorously, these should be re-investigated for the chosen pedestal design. In Bob Wooley's calculation of the inner leg torsional shear stress, he uses elements from Mark Smith's global model to construct a global torsional stiffness model that is consistent with the Vee-Pipe design - but possibly not the "flat" or not bent spoke compliance. The torsional shear values would be bracketed by the modeling available.

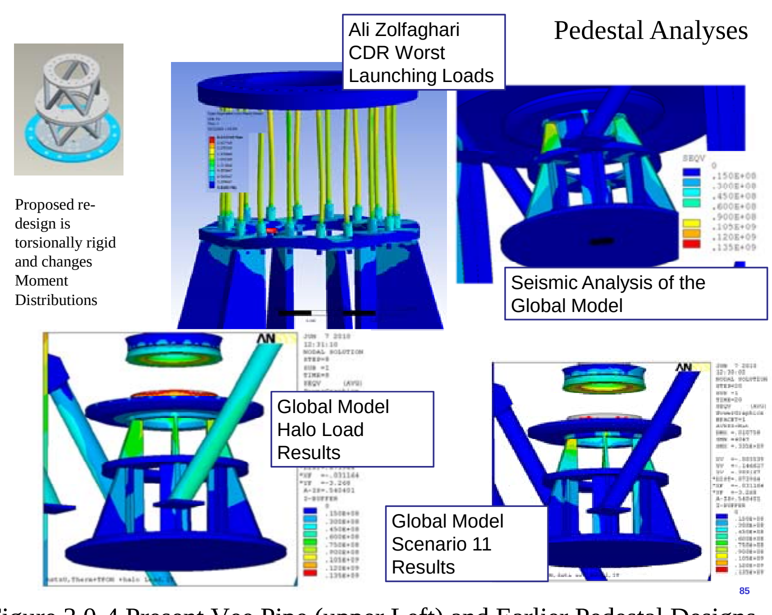


Figure 3.0-4 Present Vee Pipe (upper Left) and Earlier Pedestal Designs

Figure 3.0-4 shows the work performed on the pedestal up to the PDR . The gusseted plate design has upper "vanes" that are torsionally weak and appear weak with respect to side loads from seismic and halo loads, but their stresses are well within allowables. Halo Loads were considered only for the gusseted plate This was done in the global model [2] based on an early estimate of halo loads that was later confirmed by [11]. The "Vee" Pedestal design improves on the vane cross section in the gusseted plate design. . Stresses in the "Vee" pipe truss pedestal design are slightly lower than for the gusseted plate design. Both are less than 20 MPa for normal vertical loads and less than 200 MPa for the faulted vertical loads. This provides a large margin.. The global model results for the Vee Pipe design show 135 MPa typically for scenarios with significant torques. The bending allowable is 241 MPa for 316 weld material, and fatigue limit is 300 MPa (See figure 7.0-3) Assuming full penetration welds producing no stress multiplier on the stress that is reported by the FEA analyses, the welds and structural elements have a large margin against normal loads and a normal design margin for faulted loads. . Connection to the TF flags is discussed, in Ali Zolfaghari's calculation [9]

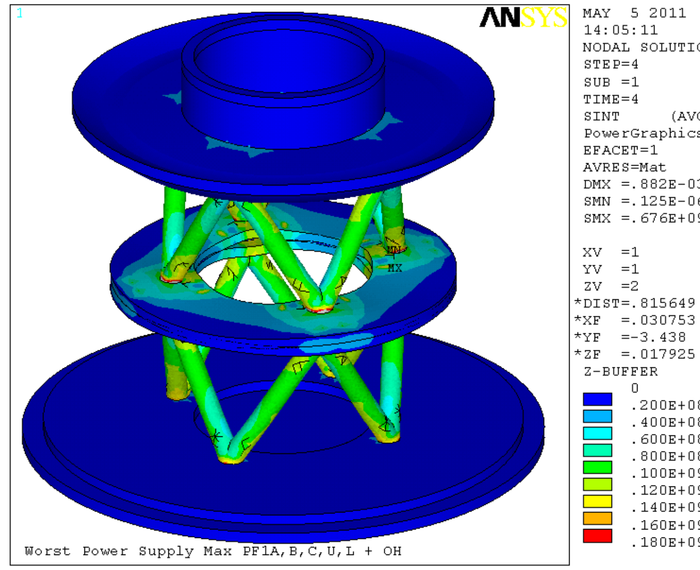


Figure 3.0-5 Representative Pedestal Stress for the Worst Case Power Supply Loads

The seismic analysis [6] was checked for the "Vee" pipe design - most of the modeling was with the plate design- and the seismic stress levels in the pedestal are acceptable. In section 9.3 of this calculation and in the global model analysis [2], a static 0.5g lateral loading was done with the Vee Pipe pedestal design and the seismic stresses are about 40 MPa - below the 135 MPa in the pipe trusses for the scenario loads.

4.0 Digital Coil Protection System Input

Conceptual design of the upgrade to NSTX explored designs sized to accept the worst loads that power supplies could produce. Excessive structures resulted that would have been difficult to install and were much more costly than needed to meet the scenarios required for the upgrade mission, specified in the General Requirements Document (GRD). Instead the project decided to rely on a digital coil protection system (DCPS). For the pedestal the critical loads are the vertical loads from the OH and PF1 a and b Upper and Lower coils interacting with the rest of the PF system. For the "Vee" Pipe design torsional loads are added to the vertical loads. For the downward loads from the PF coils, both pedestal designs are adequate even for the "worst case power supply" loads.

The limit to the upward loading is the concrete anchors or Hilties. Ninety four 3/4 inch Hilties are required to resist the worst case power supply loads. It is not likely that this number will be used. Only 5 3/4 inch anchors are needed to react the normal operating net load on the centerstack. Many more than 5 are suggested. The actual number will set the limit for the DCPS.

5.0 Design Input

5.1 Criteria

Criteria are as outlined by the NSTX Structural Design Criteria, [1]. The bolting section follows:

I-4.1.4.3 Stress Limits for Bolting Material
For preload:
• Bolt preload stress shall not exceed the lesser of 0.75 Sy at room temperature or 0.75 Sy at operating temperature.

For operating loads:
• Average tensile stress due to primary loads shall not exceed 1.0 Sm.
• Maximum direct tension plus bending stress due to primary loads shall not exceed 1.5 Sm.
For preload combined with operation:
At any point in time, combined operating loads and preload shall be evaluated for compatibility with joint design but in any case the maximum direct tension plus preload stress shall not exceed 0.9 Sy.

5.2 Design Point Spreadsheet Loads

(PF1AU+PF1BU+PF1BL+PF1AL+OH)	Fz(lbf)	Fz(N)
Min w/o Plasma	-39635	-176312
Min w/Plasma	-53445	-237745
Min Post-Disrupt	-41843	-186134
Min	-53445	-237745
Worst Case Min	-375500	-1670374
Max w/o Plasma	20397	90733.99
Max w/Plasma	10748	47811.39
Max Post-Disrupt	19630	87322.06
Max	20397	90733.99
Worst Case Max	375501	1670378

Note that the deadweight of the centerstack is larger than 20,000 lbs, more like 48000 lbs – See Figure 9.1-2

5.3 References

[1] NSTX Structural Design Criteria Document, NSTX_DesCrit_IZ_080103.doc I. Zatz
[2] NSTX-CALC-13-001-00 Rev 1 Global Model – Model Description, Mesh Generation, Results, Peter H. Titus March 2011
[3] NSTXU-CALC-133-03-00 Centerstack Casing and Lower Skirt Stress Summary, P. Titus
[4] NSTX Design Point Sept 8 2009 http://www.pppl.gov/~neumeyer/NSTX_CSU/Design_Point.html
[5] OOP PF/TF Torques on TF , R. Woolley, NSTXU CALC 132-03-00, Feb 10 2012
[6] NSTX Upgrade Seismic Analysis NSTXU-CALC-10-02-00 Rev 0 February 9 2011 Prepared By: Peter Titus,
[7] "General Electric Design and Manufacture of a Test Coil for the LCP", 8th Symposium on Engineering Problems of Fusion Research, Vol III, Nov 1979
[8] "Handbook on Materials for Superconducting Machinery" MCIC- HB-04 Metals and Ceramics Information Center, Battelle Columbus Laboratories 505 King Avenue Columbus Ohio 43201
[9] NSTX Upgrade TF Flag Key Structural Analysis, Calculation number NSTXU 132-08-00 prepared by Ali Zolfaghari

[10] Email from Mark Smith:

Pete,

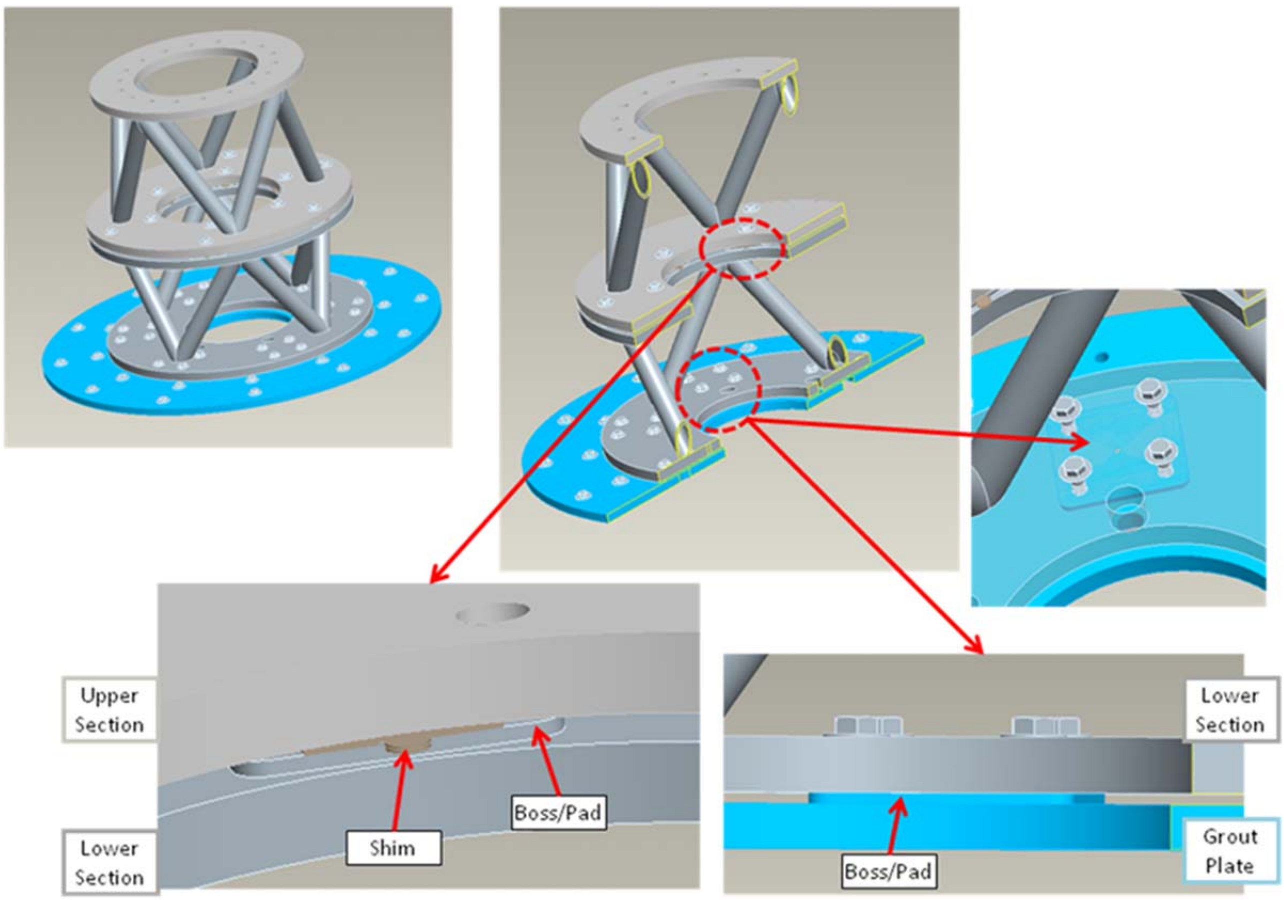
Below is a more detailed image of the pedestal design.

There are two sections: upper and lower.

There are 6 (bosses or pads) between the upper and lower sections as well as the lower section and grout plate.Also, shims will be placed between the upper and lower sections.Thus, there are gaps between these components.Also note, the structural tubing was aligned as you mentioned. However, the tubes are not aligned with all of the bosses. This needs to be resolved.

So, it will take some time to develop the CAD model for the FEA with some of these details. Hopefully, this will be completed by next week.

Note:
The bolt sizing, spacing, total number required and preload have not been determined. These details were scheduled for the final design analysis.



[11] Email from Art Brooks Thu 3/11/2010 8:21 AM, providing Upper and Lower design loads for the centerstack casing halo loads, copy of the email is included in the appendices
[12] WBS 1.1.2 Lid/Spoke Assembly, Upper & Lower NSTX-CALC-12-08-00 Rev 0 May 2011 Prepared
[13] NSTX Upgrade Centerstack Casing and Lower Skirt Stress Summary NSTXU-CALC-133-03-00 Rev 0 August 2011 Prepared By: Peter Titus
[14] Halo Current Analysis of Center Stack NSTXU-CALC-133-05-00 Prepared By: Art Brooks, Reviewed by: Peter Titus, Cognizant Engineer: Jim Chrzanowski, WBS 1.1.3 Magnet Systems,
[15] Bellows Qualification Calc # NSTXU CALC 133-10-00, Peter Rogoff, Checked by I. Zatz
[16] May 14 email from M. Smith with recommended design value for the Carbinite high friction coating
[17] NSTXU-Calc- 133-05-01 Halo Current Analysis of Center Stack, A. Brooks 12-19-213
[18] WBS 1.1.2 Lid/Spoke Assembly, Upper & Lower NSTX-CALC-12-08-02 Rev 2 May 2013 Prepared by: Peter Titus, Reviewed By: Irving Zatz, Cognizant Engineer: Mark Smith,

5.4 Drawings

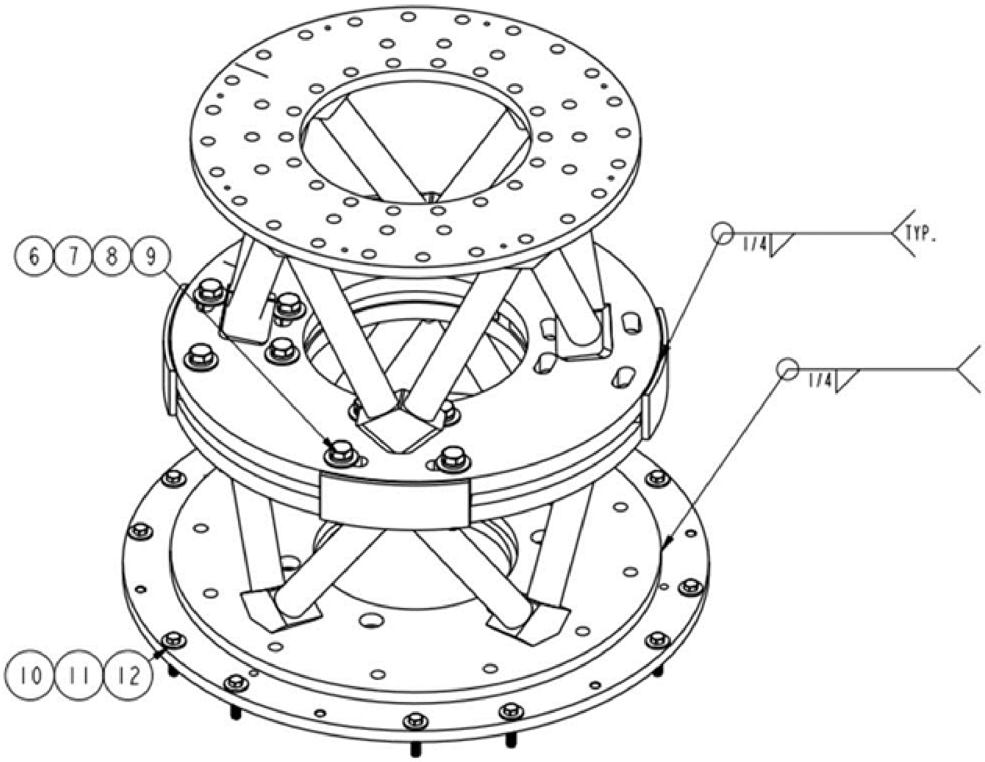


Figure 5.4-1

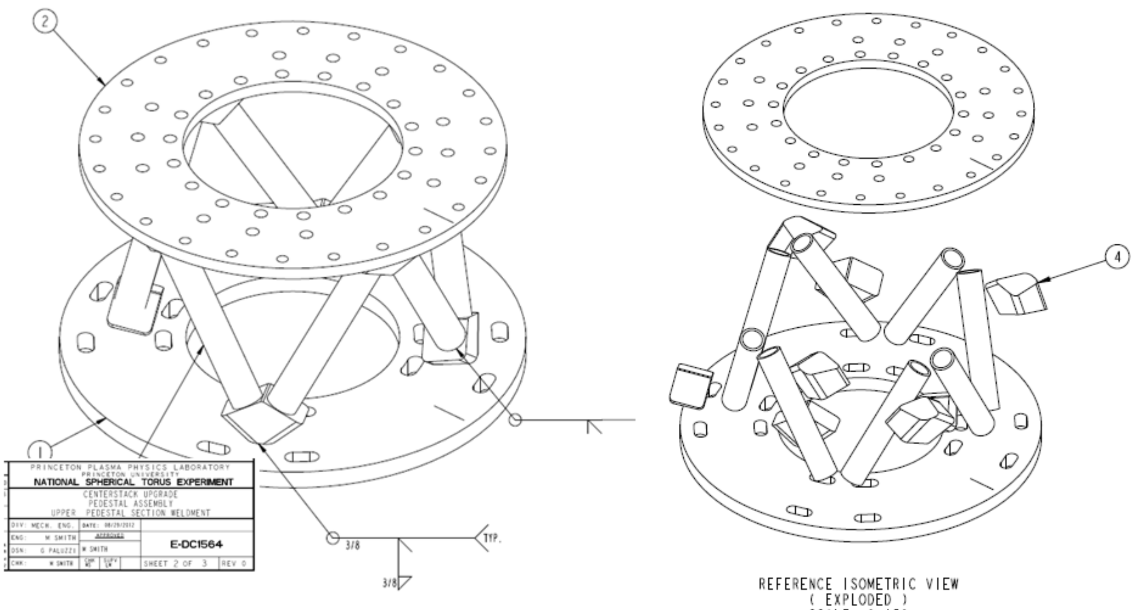


Figure 5.4-2

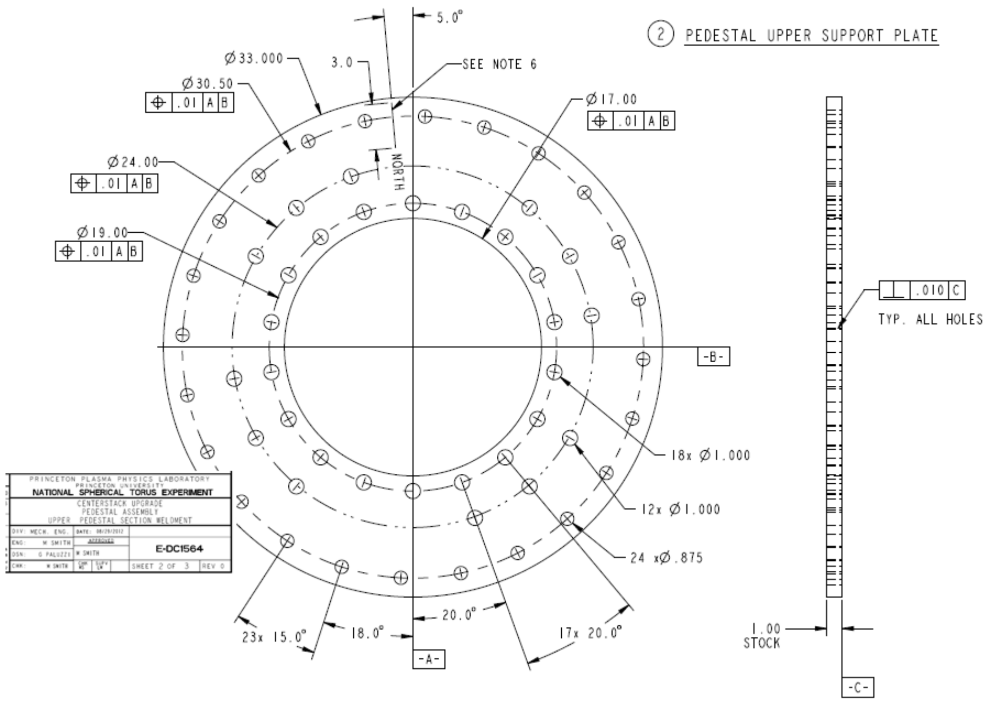


Figure 5.4-3

Seal

1

Figure 5.4-4

6.0 Analysis Model

The Pedestal is analyzed with two modeling approaches. It is included in the global model [2] and separate models of the pedestals are employed. Two designs have been evaluated. One which was chosen for the CDR and PDR analysis, uses gusseted plates. The second, introduced at the PDR and chosen for the FDR employs a trussed pipe design which is intended to be torsionally stiff. The pipe design basically has four stress areas at the pipes' intersection with the flanges. The gusseted plate design has six sets of gusseted plates which act as columns and flex plates (for torsion). The "Vee" Pipe design has two versions - one which is linear and is included in the global model and another version that models with a gapped interface, the shims planned to be placed between the mid height flanges to align the pedestal with the floor and centerstack elevations.

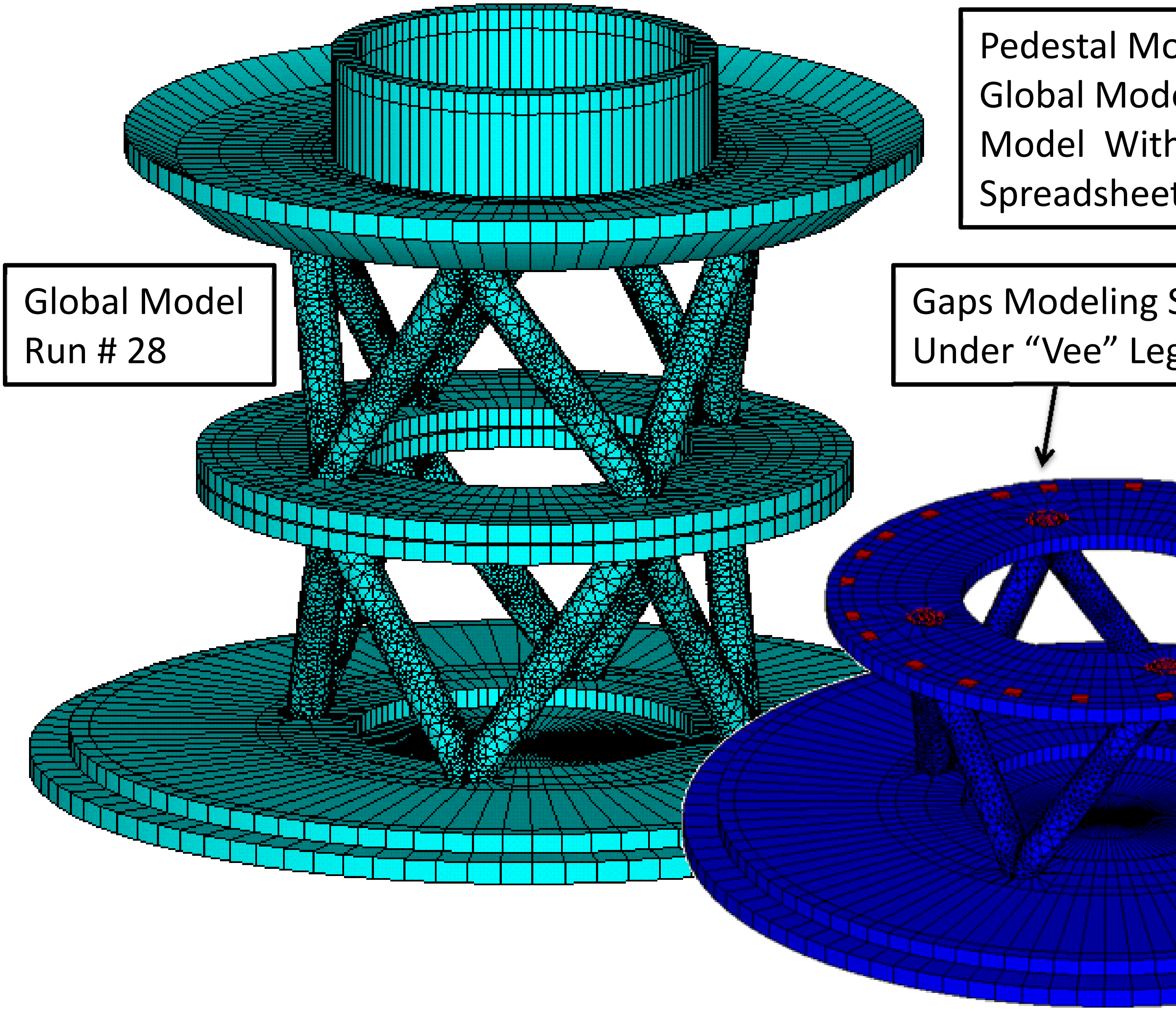


Figure 6.0-1 "Vee" Pipe model The pads modeling the bolts were repositioned, and an inner and outer bolt circle is used.

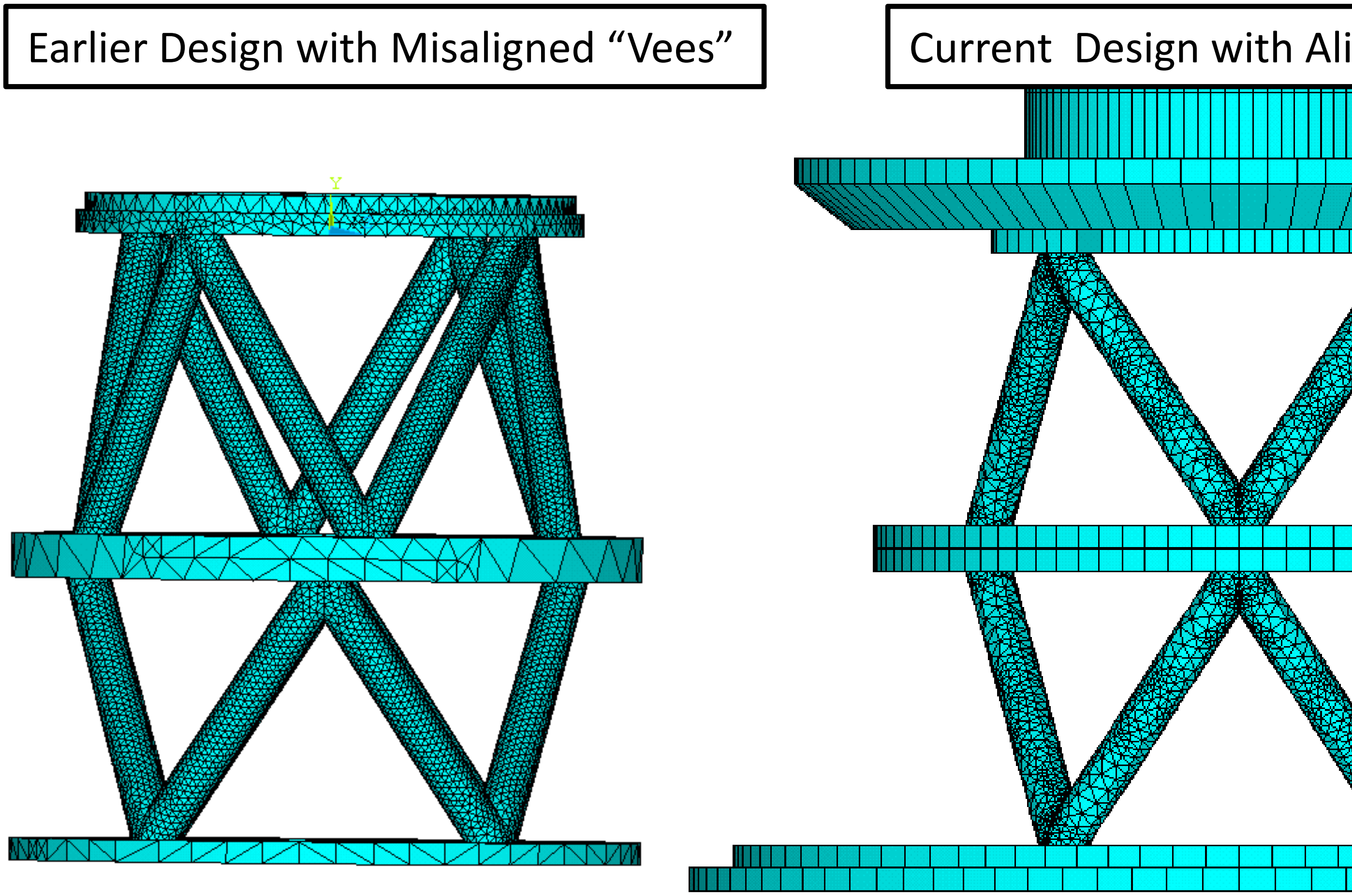


Figure 6.0-2 "Vee" Pipe models -Misaligned and Aligned

The first evolution of the Vee Pipe concept had "Vee" vertices misaligned at the mid plane where ideally, the compressive load should have been transferred directly without any offset and should not have required any plate bending to transfer the load. This was corrected in later versions of the design.

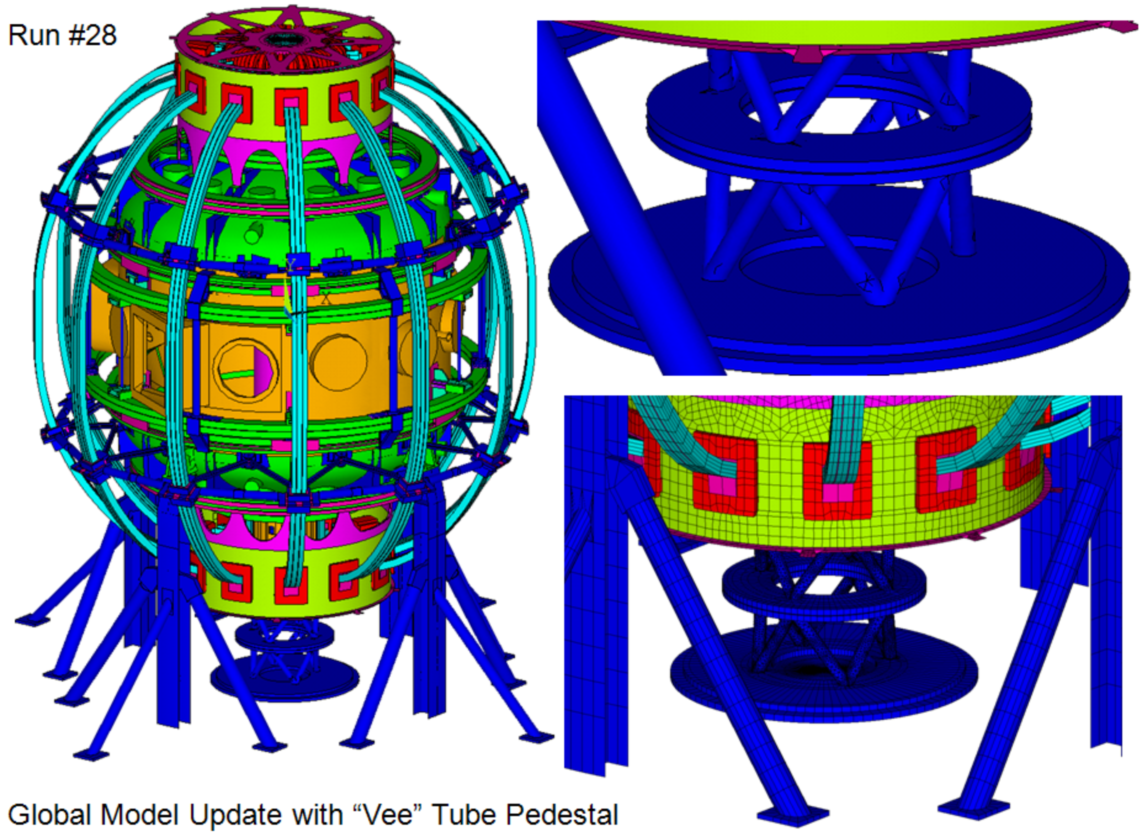


Figure 6.0-3 Ref 2 Global Model Udate with "Vee" Tube Pedestal

7.0 Materials and Allowables

Table 7.0-1Tensile Properties for Stainless Steels

Material	Yield, 292 deg K (MPa)	Ultimate, 292 deg K (MPa)
316 LN SST	275.8[7]	613[7]
316 LN SST Weld	324[7]	482[7] 553[7]
316 SST Sheet Annealed	275[8]	596[8]
316 SST Plate Annealed		579
304 Stainless Steel (Bar,annealed)	234 33.6ksi	640 93ksi
304 SST 50% CW	1089	1241 180ksi

Table 7.0-2 Coil Structure Room Temperature (292 K) Maximum Allowable Stresses, Sm = lesser of 1/3 ultimate or 2/3 yield, and bending allowable=1.5*Sm

Material	Sm	1.5Sm
316 Stainless Steel	184 (26.7ksi)	276 (40 ksi)
316 Weld	161	241
304 Stainless Steel (Bar,annealed)	156MPa(22.6ksi)	234 MPa (33.9ksi)

Using the 295K curve,

For the lesser of
2*stress (~300 Mpa)

Or

20 on life(~400 Mpa)

The Allowable is =300
MPa

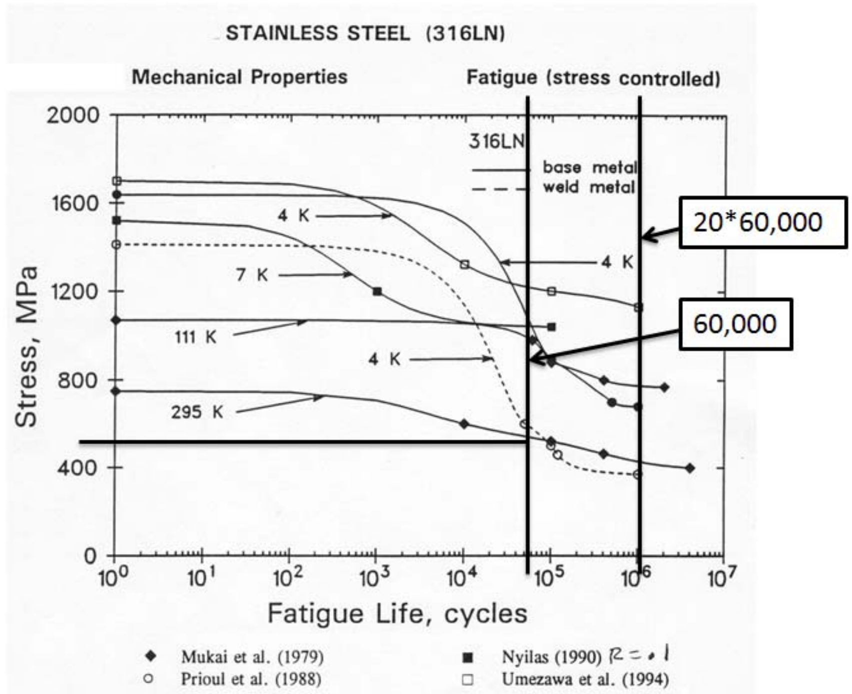


Figure 7.0-1 Fatigue S-N Curve for 316 Stainless Steel.

The limits are based on the early GRD requirement of 30,000 full power pulses which are double swung. The latest GRD requires 20,000 full power pulses and evaulations of the second OH swing show little fatigue damage because it is limited to 13.5 kA vs the first pulse current of 24 kA

ASTM A193 Bolt Specs from PortlandBolt.com

B8M	Class 1 Stainless steel, AISI 316, carbide solution treated.
B8	Class 2 Stainless steel, AISI 304, carbide solution treated, strain hardened
B8M	Class 2 Stainless steel, AISI 316, carbide solution treated, strain hardened

Mechanical Properties

Grade	Size	Tensile ksi, min	Yield, ksi, min	Elong, %, min	RA % min
B8 Class 1	All	75	30	30	50
B8M Class 1	All	75	30	30	50
B8 Class 2	Up to 3/4	125	100	12	35
	7/8 - 1	115	80	15	35
	1-1/8 - 1-1/4	105	65	20	35
	1-3/8 - 1-1/2	100	50	28	45
B8M Class 2	Up to 3/4	110	95	15	45
	7/8 - 1	100	80	20	45
	1-1/8 - 1-1/4	95	65	25	45
	1-3/8 - 1-1/2	90	50	30	45

The allowable for a one inch ASTM A193 B8M Class 2 would be the lesser of 115/3 or 2/3*80 =38.3 ksi
The allowable for up to 3/4 inch ASTM A193 B8M Class 2 bolt would be the lesser of 110/3 or 2/3*95 =36.66 ksi
The allowable preload stress, from the NSTX Criteria, for the ¾ inch B8M Class 2 bolts, is .75*Yield or 71 ksi

8.0 Stand-Alone-Model Results

The Pedestal is analyzed with two modeling approaches, the global model [2] and a separate sub model or stand-alone model. In the "stand-alone" model, the pedestal model is separated from other structures and loaded via displacement constraints. An initial guess is imposed and then the displacement is scaled based on the resulting reaction forces to obtain the vertical loading specified by the design point spreadsheet.

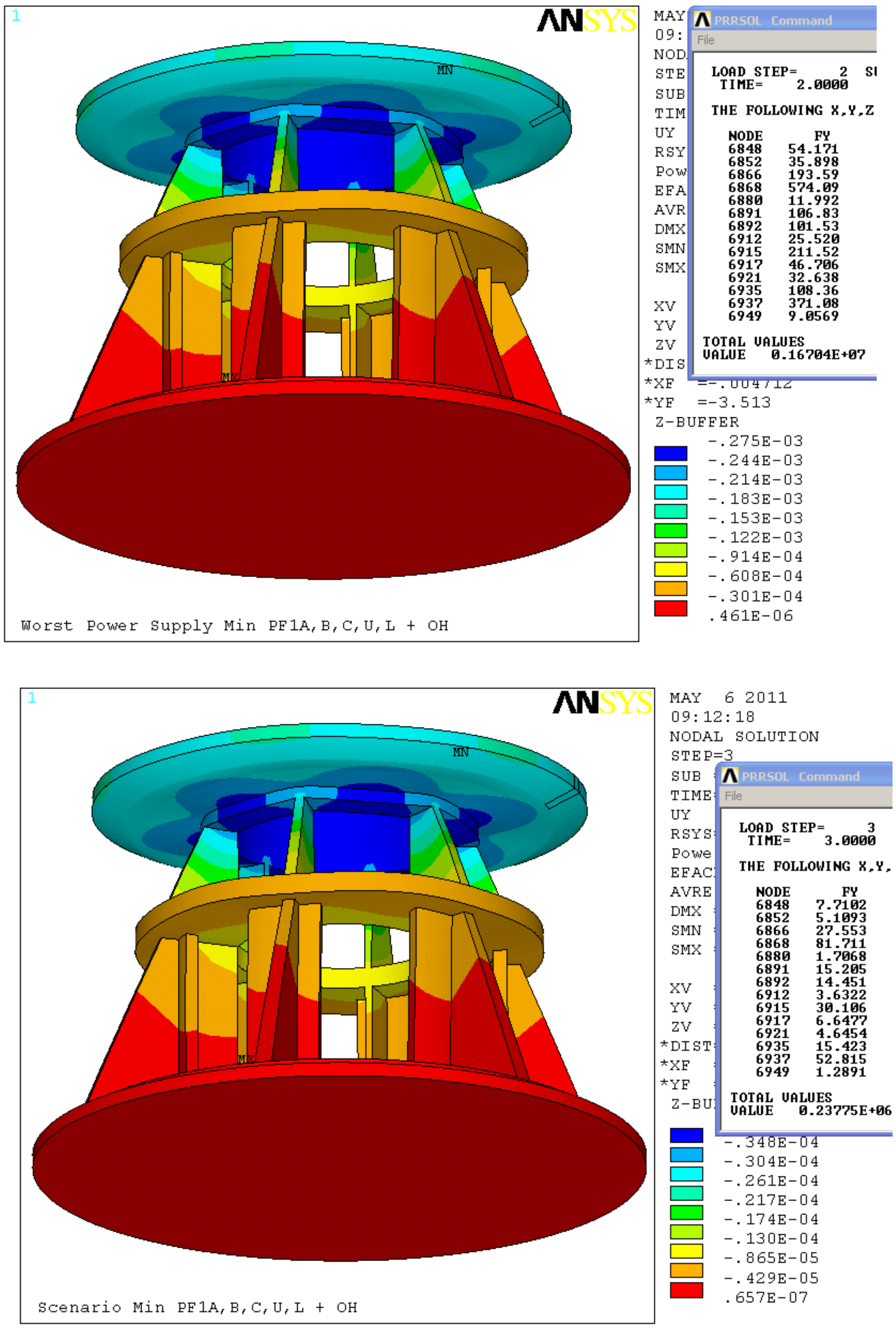
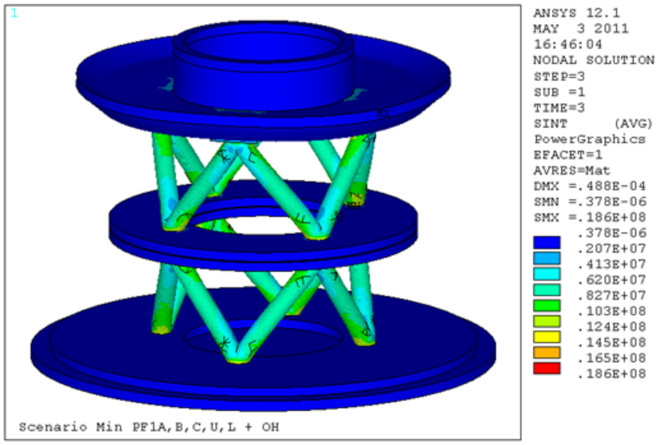


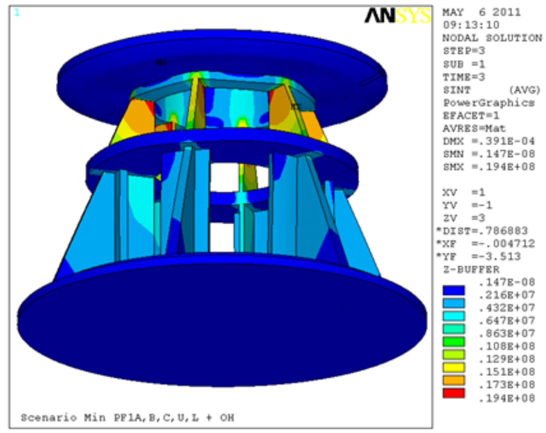
Figure 8.0-1 Displacement Constraints on the Gusseted plate model along with the script that applies displacement constraints scaled to produce the required applied load

8.1 Normal Operating Downward Loads

Scenario Min (Largest Normal Downward Load)

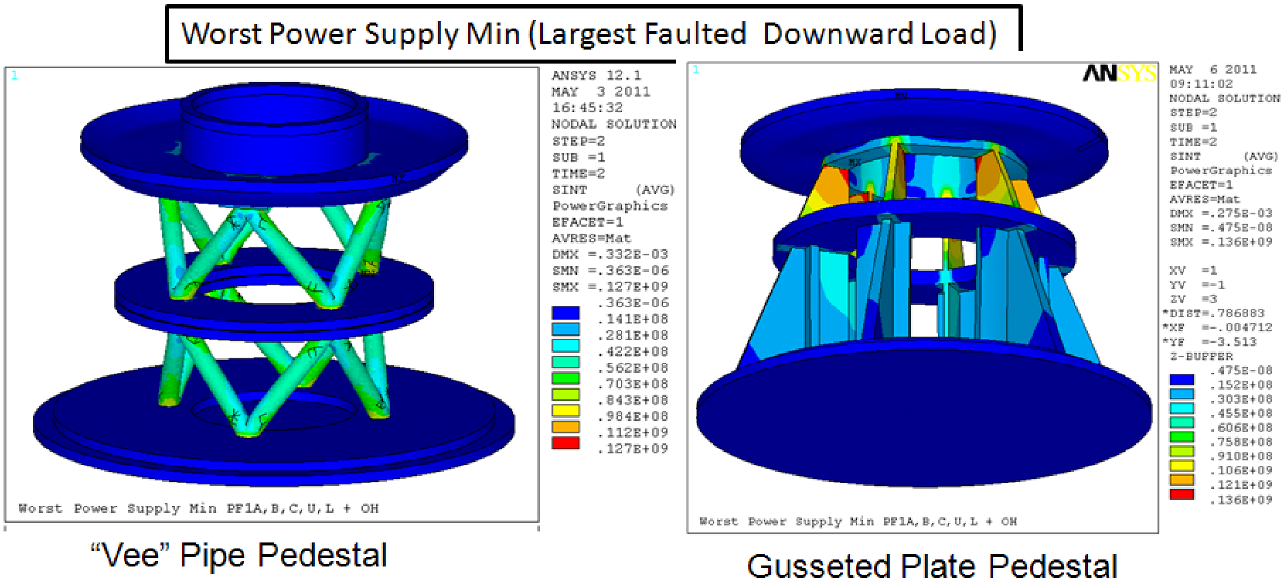


"Vee" Pipe Pedestal



Gusseted Plate Pedestal

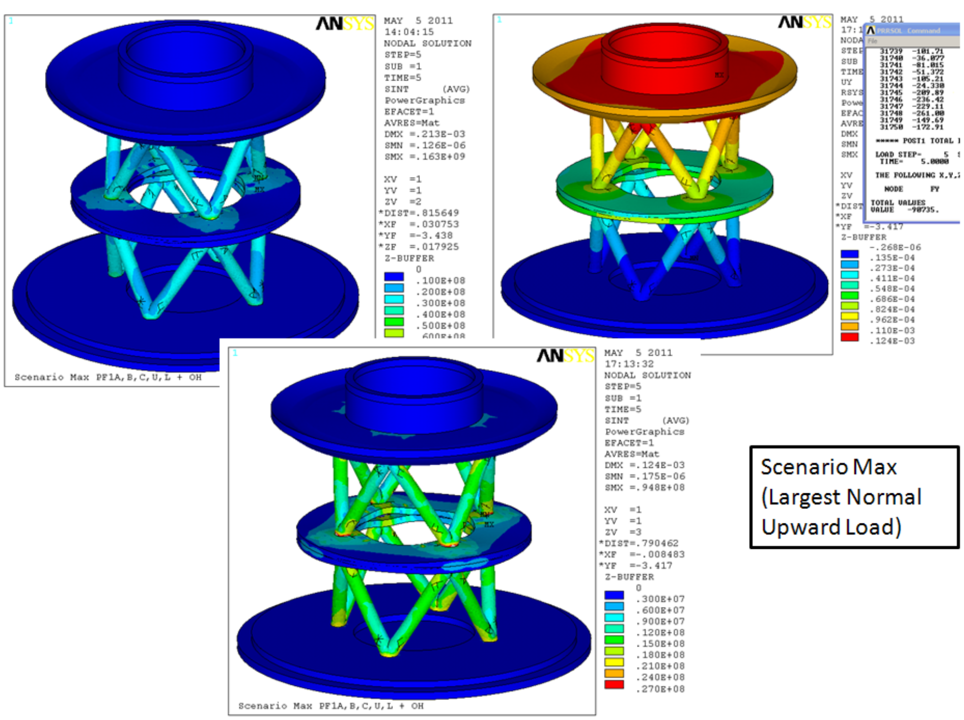
8.2 Faulted Downward Loads



Downward and Normal and Faulted Stresses are acceptable for both pedestal concepts. Stresses are almost the same for both concepts.

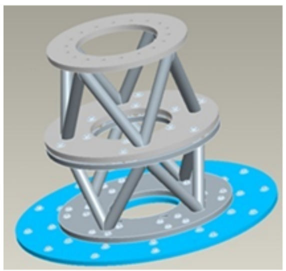
8.3 Normal Operating Upward Loads

8.3.1 Pipe and Plate Stresses for Normal Operating Upward Loads



8.3.2 Bolting and Embedment Anchors for the Normal Operating Upward Loads

(PF1AU+PF1BU+PF1BL+PF 1AL+OH)	Fz(lbf)	Fz(N)
Min w/o Plasma	-39635	-176312
Min w/Plasma	-53445	-237745
Min Post-Disrupt	-41843	-186134
Min	-53445	-237745
Worst Case Min	-375500	-1670374
Max w/o Plasma	20397	90733.99
Max w/Plasma	10748	47811.39
Max Post-Disrupt	19630	87322.06
Max	20397	90733.99
Worst Case Max	375501	1670378



Hilti Drop-In

Hilti HDI Concrete Flush Anchor Tests

	2000 psi	Concrete	4000 psi	Concrete	6000 psi	Concrete
Anchor Size	Tension	Shear	Tension	Shear	Tension	Shear
HDI – ¼	1904	1738	2251	1781	3075	3050
HDI – 3/8	3174	3970	4942	4225	5650	5900
HDI – 1/2	**3997**	**5873**	**6751**	**6224**	**10200**	**9350**
HDI – 5/8	**5549**	**8883**	**9696**	**12205**	**10400**	**13600**
HDI – 3/4	8857	15195	16034	17609	16400	21200

Allowable Design Loads are ¼ these Values, i.e. a F.S. of 4 is recommended

375550/(16000/4) = 94 ¾ Hilties to take the Worst Power Supply Loads
20397/(16000/4) = 5 ¾" Hilties to take the Max Scenario Load

There are really only 13 Effective in the Outer Two Rows.

Figure 8.3.2-1 Pedestal Hilti Capacity. Note that Unisorb anchors are being used –See Attachment 4.

The design point spreadsheet loads do not include deadweight of the centerstack, which is ~48,000lbs (See Figure 9.1-2). So if only the PF coil Lorentz loads due to normal operation were the basis of loading the embedments, then the net load would never go tensile. Normal operation also includes a torque , and faulted loads, halo loads and seismic loads will produce shear and tension on the anchors. Initially Hilti anchors were specified, but during construction, Unisorb anchors were chosen. These have better capacities.

Anchor Design Tensile Loads
¾ Hilti Anchor 16000/4= 4000 lbs
1" CS-100 Unisorb 6834 lbs

The analysis model uses four bolts in a pattern around the vertices of the trusses. Also shimming of the mid flanges is assumed to also align with the vertices of the trusses. Bolt sizes are assumed to be 1 inch diameter ASTM A 193 B8 bolts with an 80 ksi yield. There are 16 bolts in the final design. One inch bolts have a .6051 in^2 stress area and thus the total upward capacity of the mid flange connection is 16*80000*.6051 = 774528 lbs. which is above the worst power supply load of 375500 lbs. So the flange bolts capable of resisting the faulted upward tensile load.

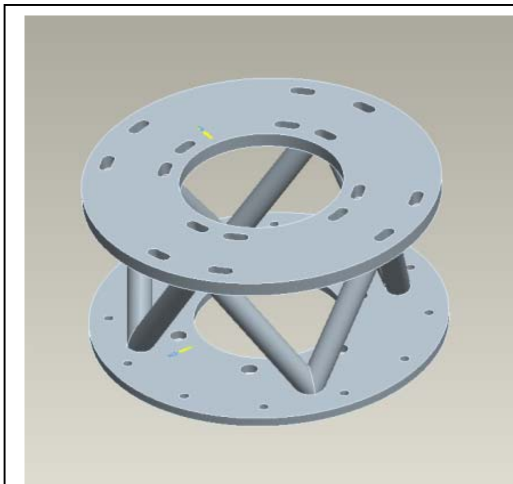
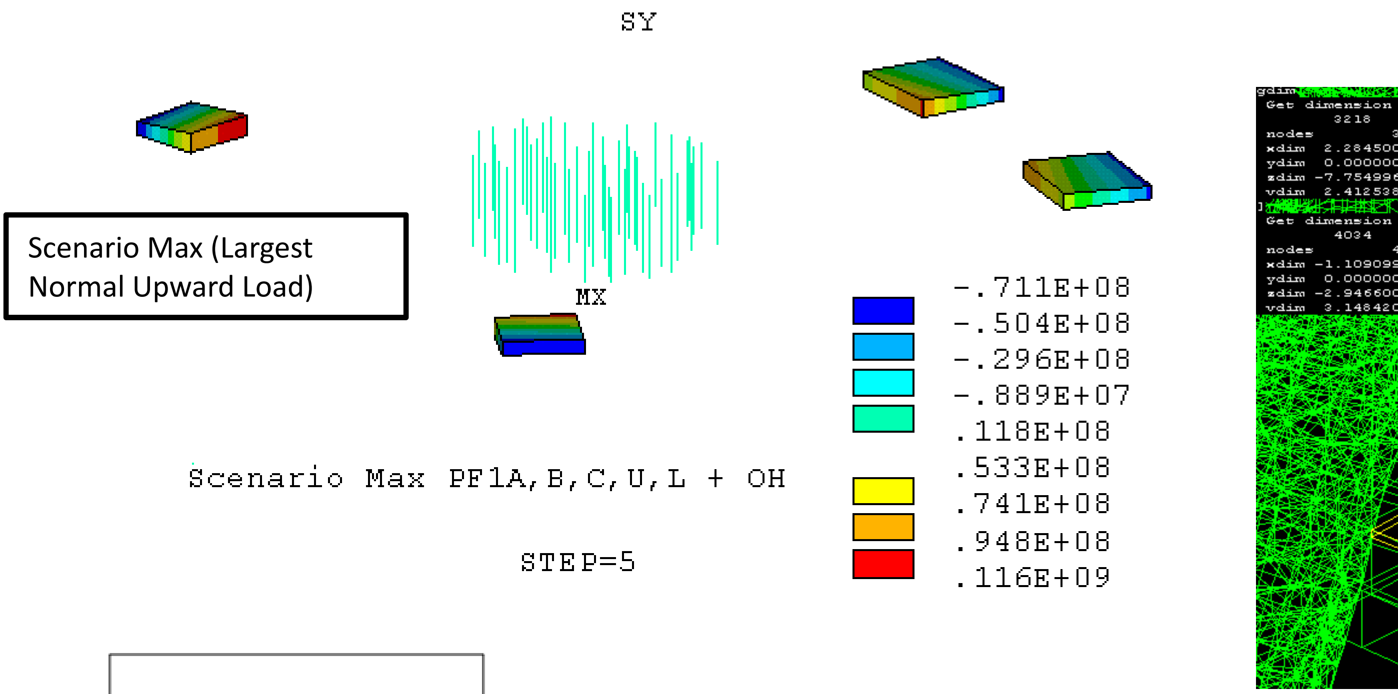


Figure 8.3.2-2 Pedestal final Design, showing 16 bolt holes t mid-height flange.



(PF1AU+PF1BU+PF1BL+PF1AL+OH)	Fz(lbf)	Fz(N)
Min w/o Plasma	-39635	-176312
Min w/Plasma	-53445	-237745
Min Post-Disrupt	-41843	-186134
Min	-53445	-237745
Worst Case Min	-375500	-1670374
Max w/o Plasma	20397	90733.99
Max w/Plasma	10748	47811.39
Max Post-Disrupt	19630	87322.06
Max	20397	90733.99
Worst Case Max	375501	1670378

Estimating the Bolt Load from the
.022845*.03148*(116e6-71e6)/2
3637 lbs per bolt
For 16 bolts the net load is 58200

Figure 8.3.2-3 Bolt Loads including the non-linear prying/bending action on the Bolts - -

8.4 Faulted Upward Loads

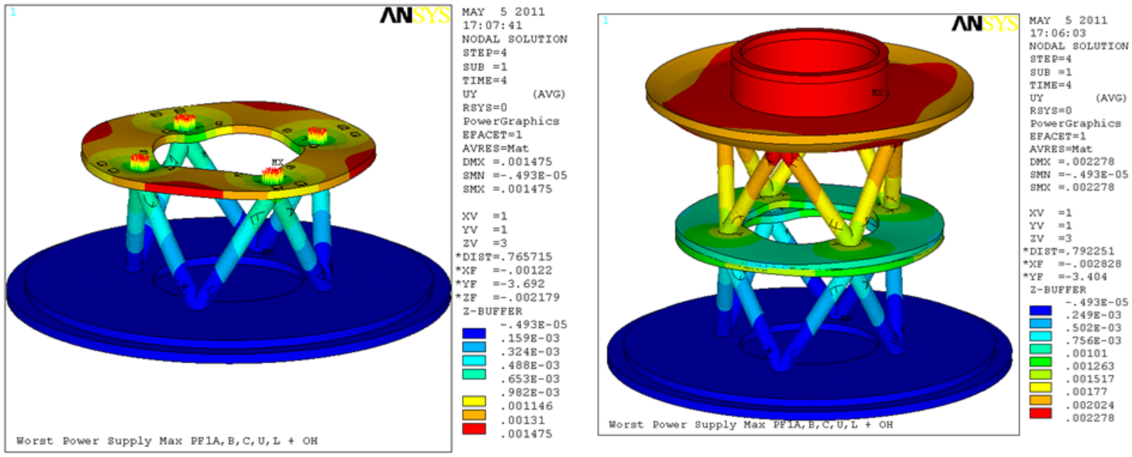


Figure 8.4-1 Vertical Displacements With Max Power Supply Loads Applied.

In figure 8.4-1, the displacement profile shows the lift-off at the gap elements that model the shims under the Vee vertices.

Again, the flanges are capable of resisting the faulted upward tensile load.

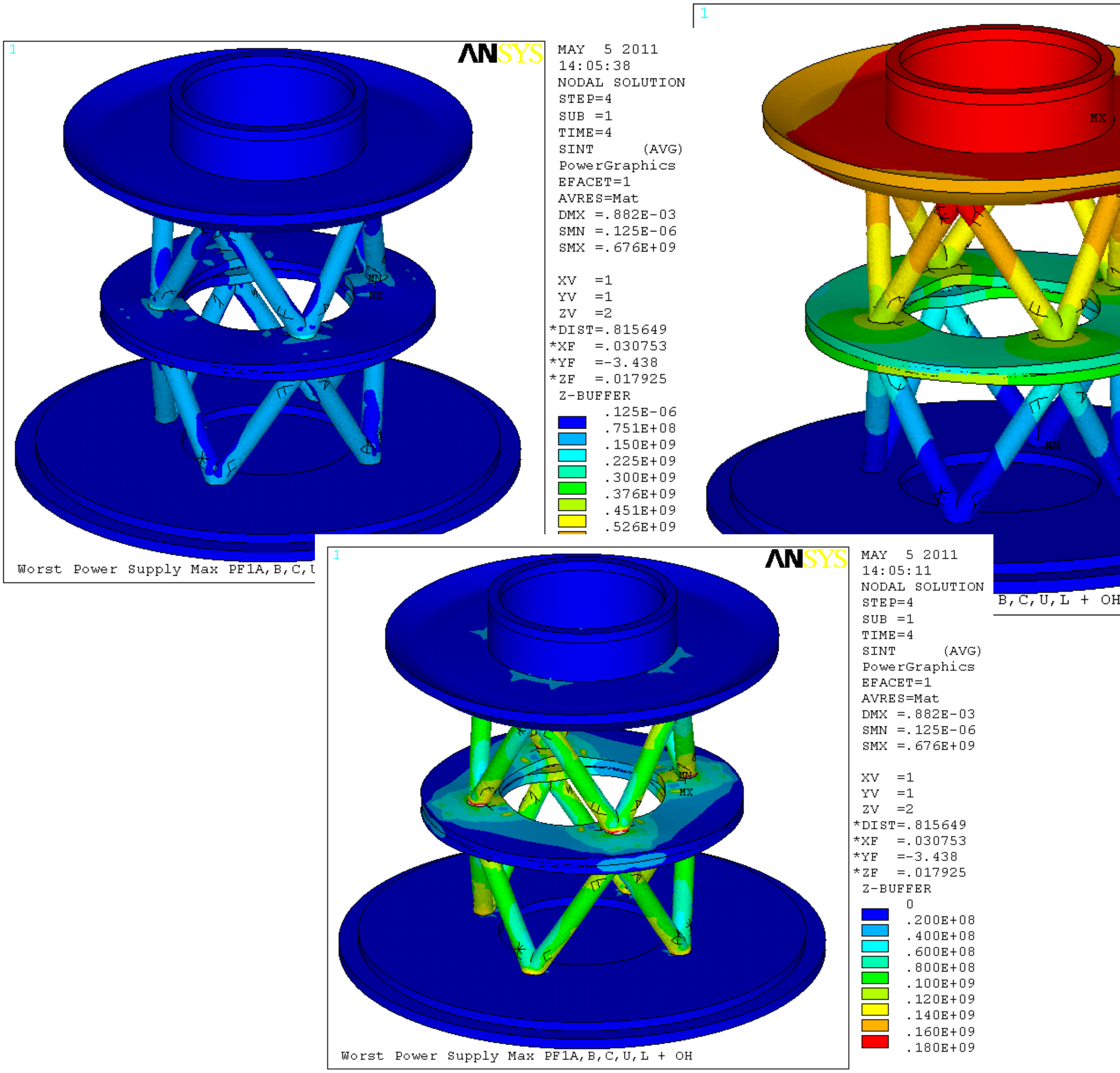


Figure 8.4-2

Upward Normal and Faulted Stresses are acceptable for both pedestal concepts. Stresses are similar for both concepts. Note, in figure 8.4-2, the peak stress is 676 MPa – well above yield. In the lower figure the stress contours were chosen the show regions close to yield which are at the intersection of the Vee and flange and shows the peak is very localized. Since this is for the max power supply loads, these stresses would be avoided by the DCPS, but the design is robust and would survive a faulted application of the maximum load the power supplies could develop.

9.0 Global Model Results

Ref [2] describes the global model of the tokamak that was updated with the Vee tube pedestal in run#28. This analysis provides results for a number of load cases not readily available from the design point spreadsheet[4] . The design point spreadsheet provides only axisymmetric loads from the PF coil currents.

9.1 Deadweight

Figure 9.1-1 shows the deadweight stresses . The average stresses in the pipe sections is 20 to 30 MPa. These would be P/A or membrane stresses and are below the Sm allowable for 316 Stainless Steel which is 26.7ksi or 184 MPa (Table 7.0-2). There are some local high stresses at the Vee intersections, which would be compressive and self limiting.

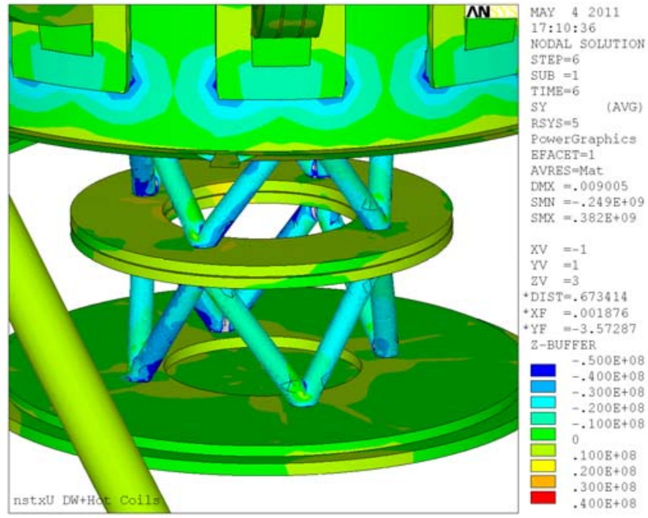


Figure 9.1-1 Deadweight Stresses in the Pedestal

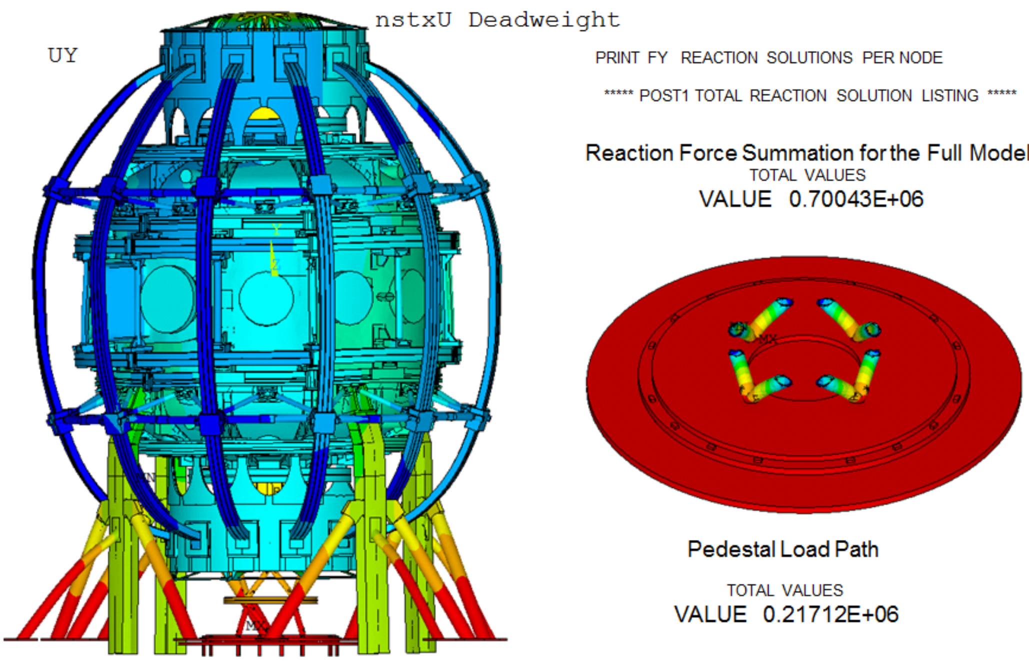


Figure 9.1-2 Deadweight Vertical Reaction Force for the Full Model and Pedestal Portion

9.2 Normal Operating Loads

9.2.1 Gusseted Plate Pedestal

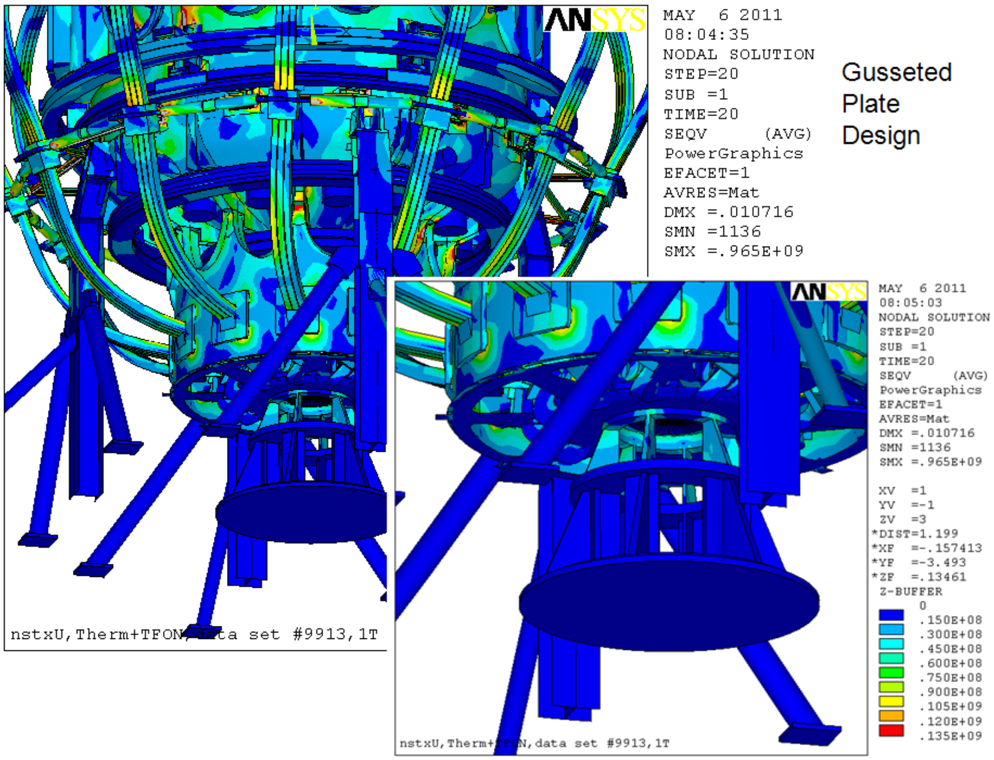


Figure 9.2.1-1 Scenario 13 Gusseted Plate Pedestal Stresses

9.2.2 Vee Pipe Pedestal Design with Bent Spoke Lower Lid

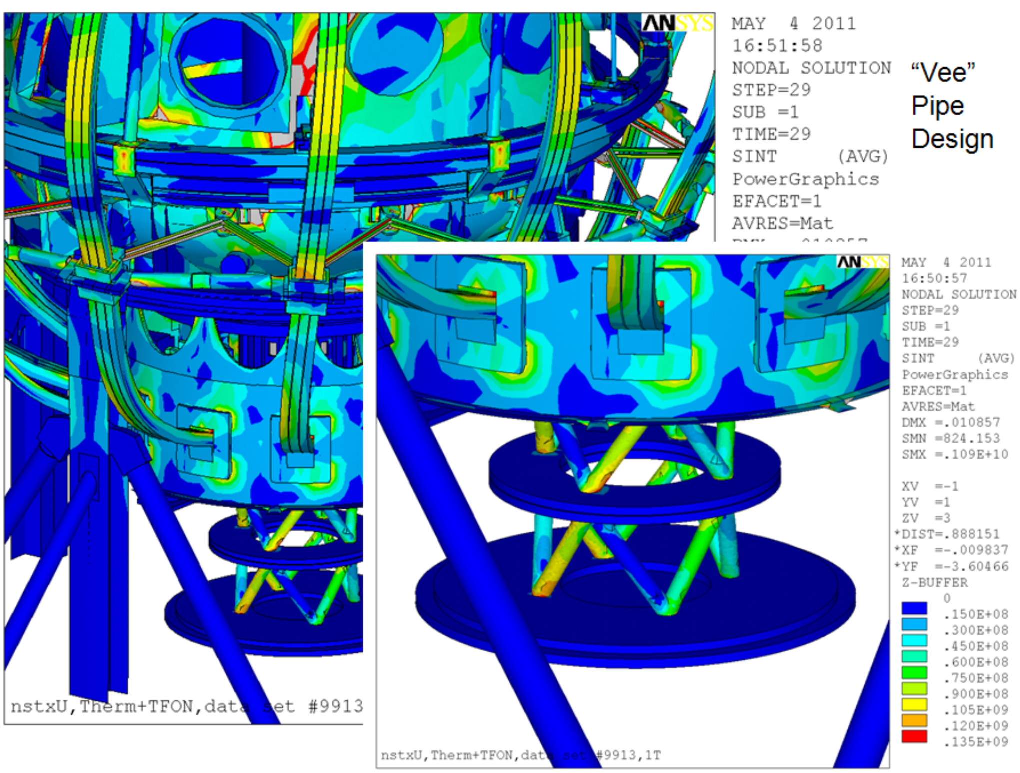


Figure 9.2.2-1 Scenario 13 "Vee" pipe Pedestal Stresses

Note that the Stresses in the "Vee" truss are not equal - this is an indication that some portion of the machine global torque is being transmitted into the truss.

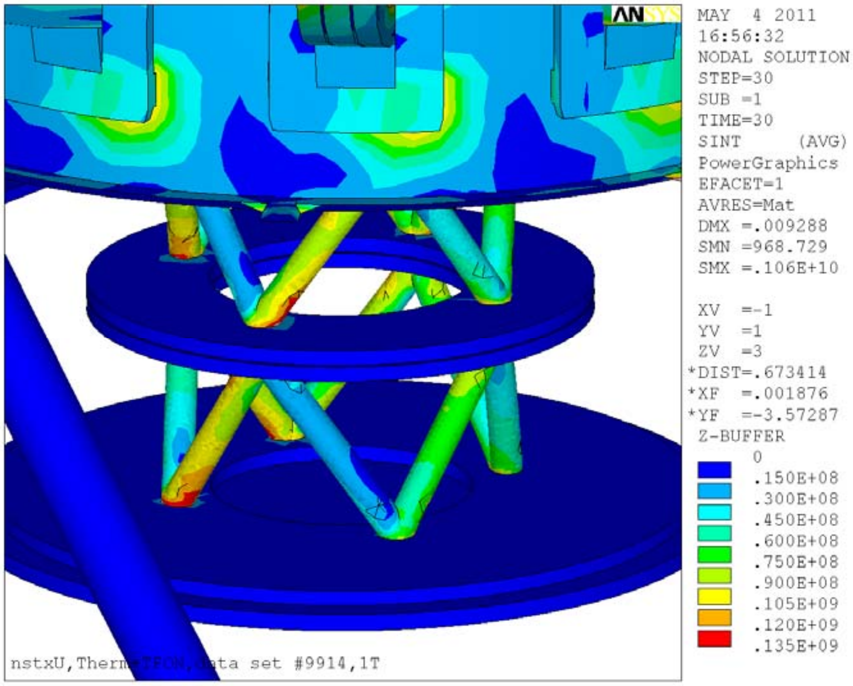


Figure 9.2.2-2 Scenario 14 "Vee" Pipe Pedestal Stresses

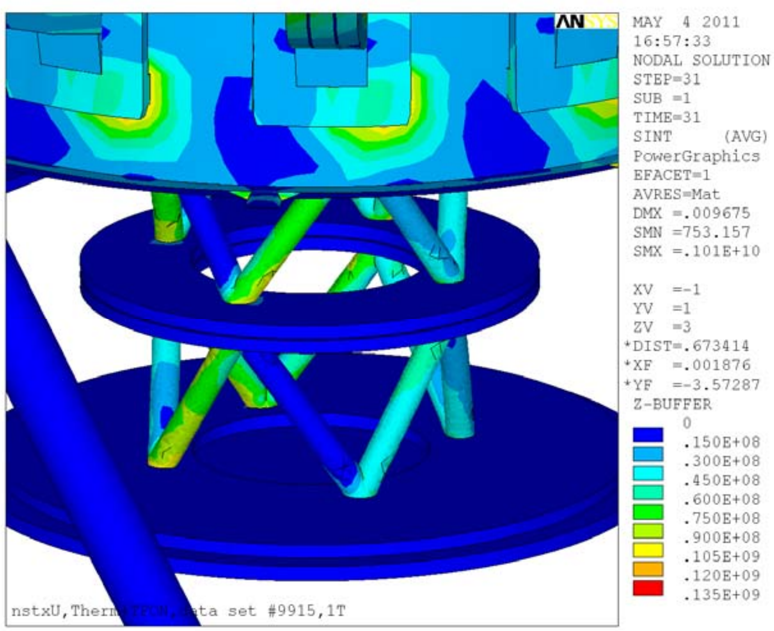


Figure 9.2.2-3 Scenario 15 "Vee" Pipe Pedestal Stresses

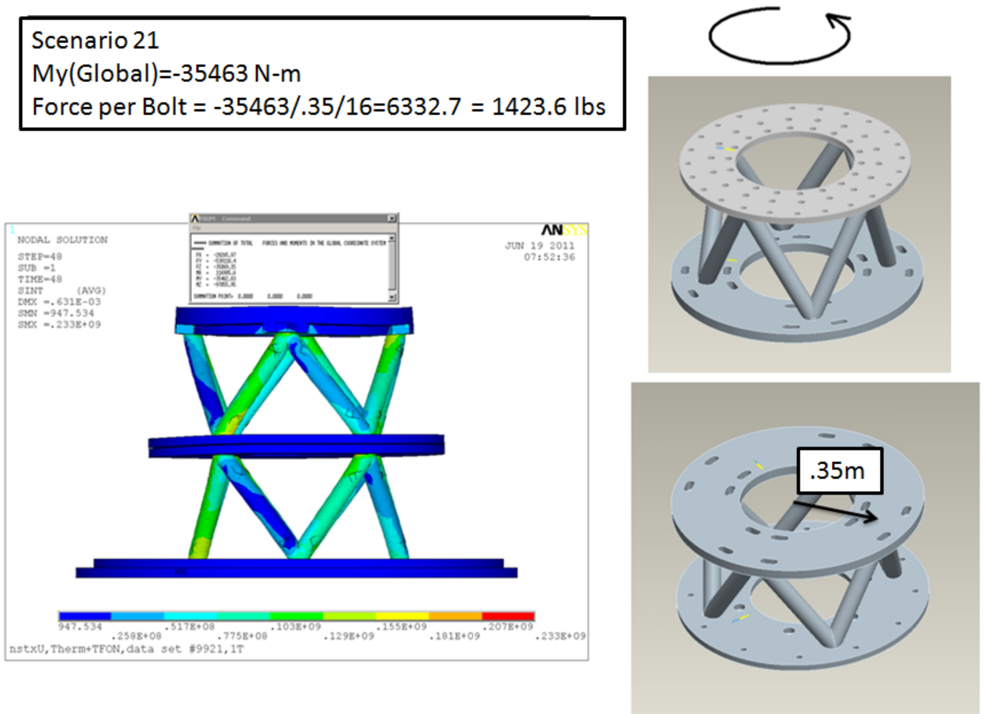


Figure 9.2.2-4 Pedestal Moment Diagram

The torques that are carried through the pedestal have been determined only for a few scenarios. Scenario 21 is larger than one of the usual larger torque scenarios, #32. The maximum moment found so far is 35463 N-m or 313860 in-lbs. More moment summations are included in [2] To envelope other scenarios, double the torque, and use 1 inch high strength bolts. The one inch bolts were recommended in section 8.3.2 to resist the worst case power supply vertical tensile or launching loads. These bolts also provide frictional resistance to the torque. with a stress area of .6051 in^2 The allowable for ASTM A193 B8M Class 2 would be the lesser of 115/3 or 2/3*80 =38.3 ksi. Each would be preloaded to 23175 lbs and each would have a frictional capacity of (.3-.15)* 23175 = 3476 Lbs - larger than twice the scenario 21 load. The other scenarios need to be addressed but it is expected that this margin is more than enough to envelope them all.

9.2.3 Vee Pipe Pedestal Design with Flat Spoke Lower Lid

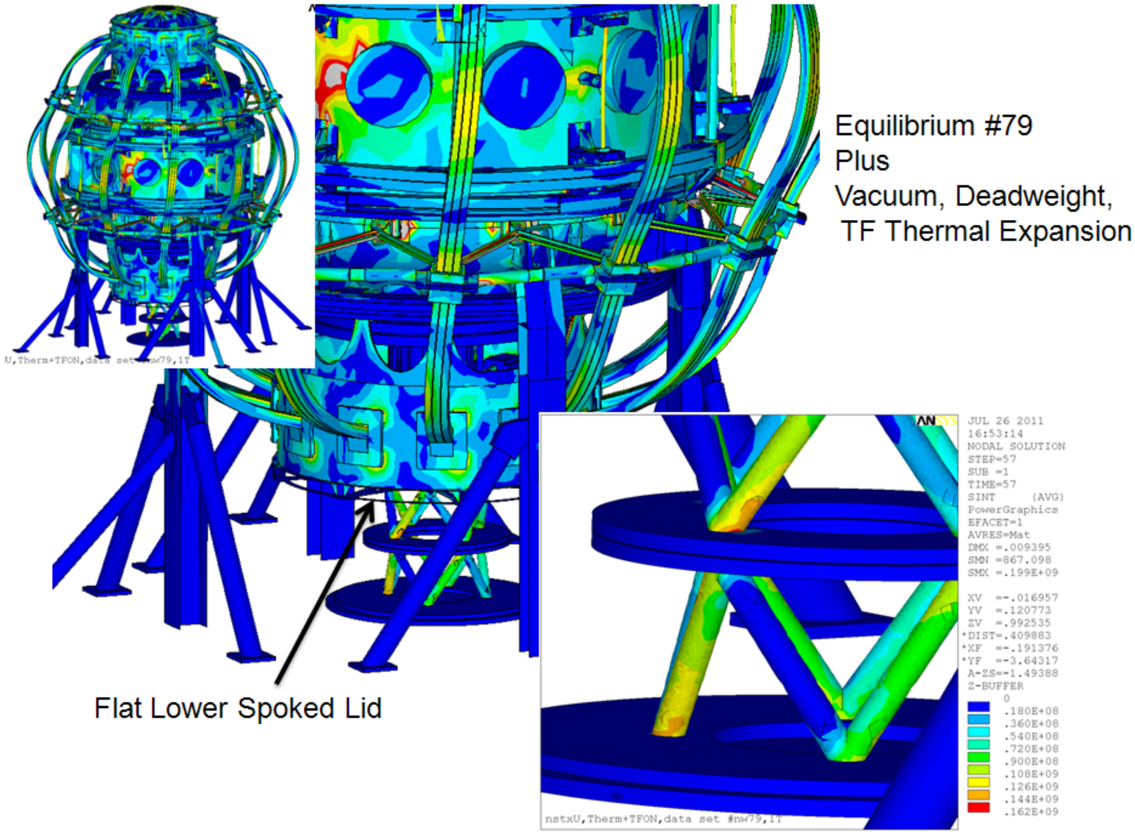


Figure 9.2.3-1 Model with Flat Lower Spoked Lid

Figure 9.2.3-2 TimHis 6 ANSYS Vertical Stress Postprocess for Most of the Scenarios

Note that the vertical stress component at the Vee intersection is compressive. except for the initial seismic and bake-out and other thermal load cases.

The global run load cases are:

1 deadweight (not turned off)
2 seismic (turned off in subsequent load steps)
3 deadweight plus vacuum (not turned off)
4 hot centerstack casing, 500C (operating Condition)
5 bake-out vessel at 150, PP at 350 (turned off in subsequent load steps

6 dead weight + 100C TF inner leg, 50C outer legs 50 C PF coils (This remains on for all the EQ with and without plasma loads)
7 TF on only. -No PF loads
8 TF on + Centerstack Halo (This is turned off for the remaining EQ load steps) and was not used in the pedestal plot - I reinstated the load case in run 35
9 through 104 96 EQ -no plasma
104 through 200 EQ with plasma

The tensile spike is the bake-out load case which has a hot vessel, and cold TF. As the vessel expands it flexes the spoked lid opposite to its normal 8mm positive displacement and puts the TF inner leg assembly in tension and pulls upward on the pedestal. The tensile stress on the Vee due to bake-out is 25 MPa. The seismic stress is -62 MPa and would be tensile in the opposite side of the pedestal. The bolting was qualified for the seismic and other upward loads.

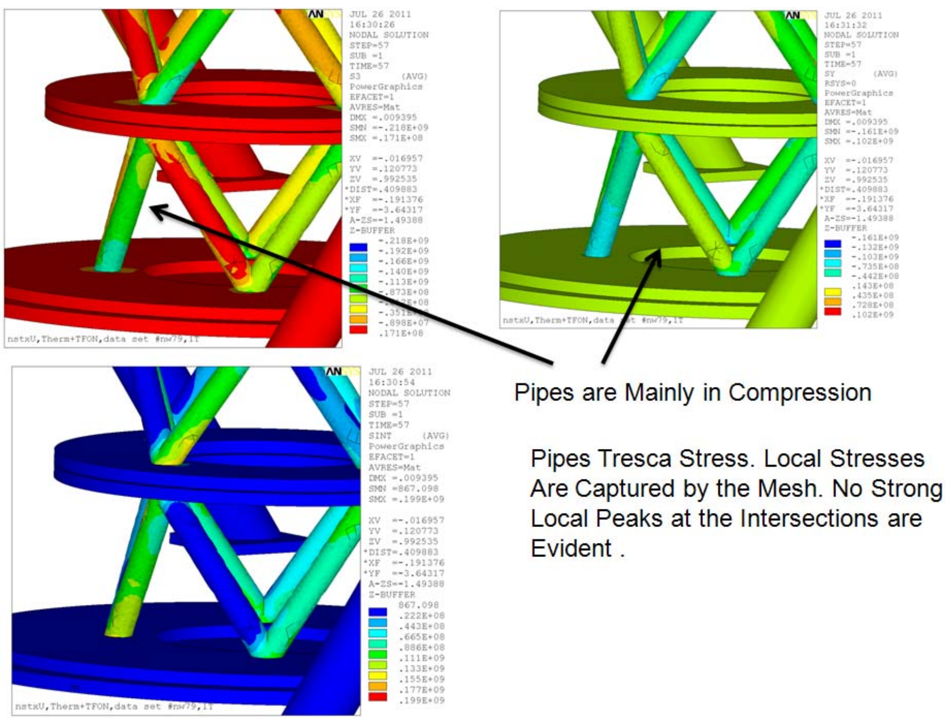


Figure 9.2.3-3 ANSYS Contour plots of Sig3, Vertical Stress, and Tresca Stress for EQ79

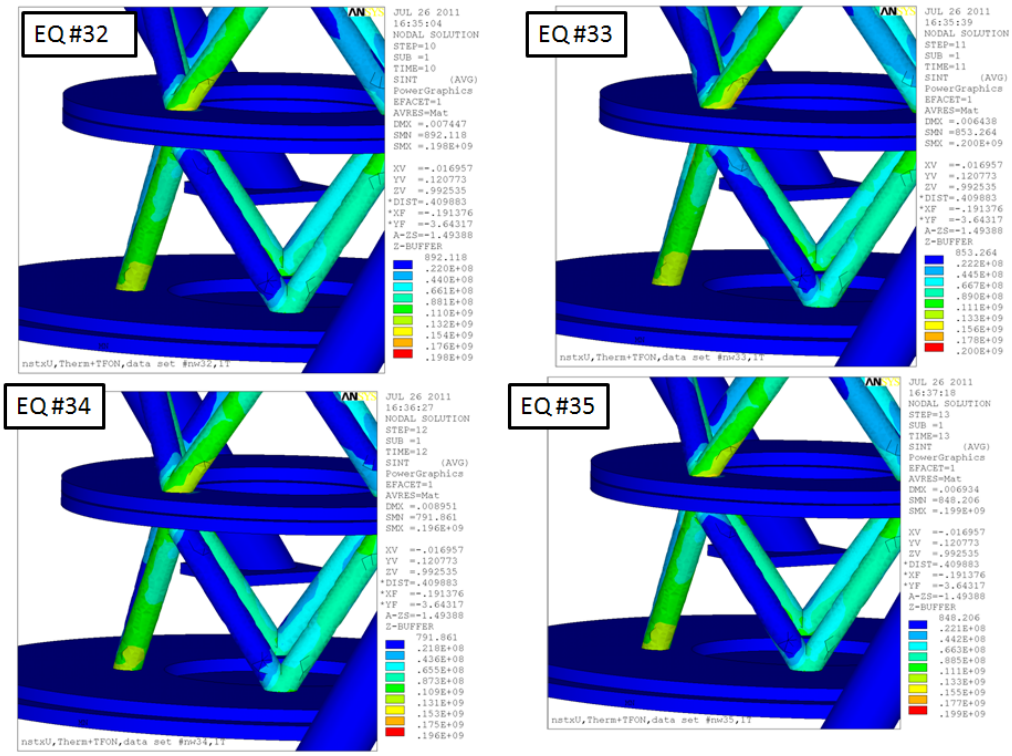


Figure 9.2.3-4 ANSYS Contour plots of Tresca Stress for Various Equilibria

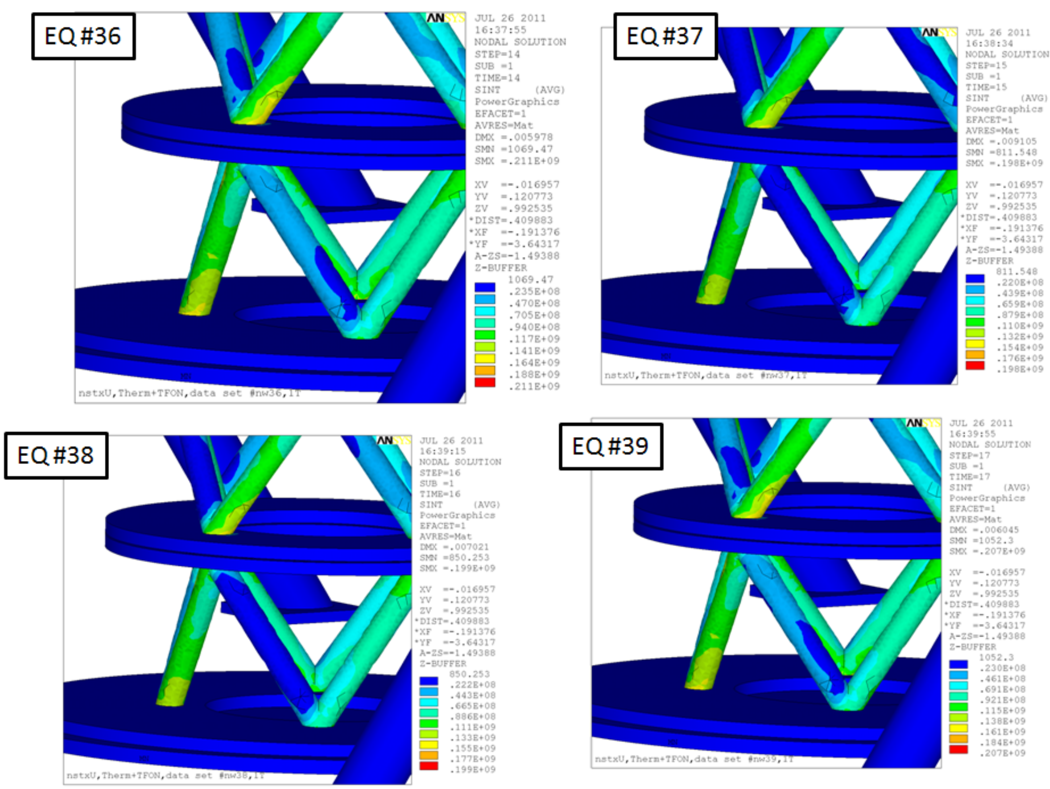


Figure 9.2.3-5 ANSYS Contour plots of Tresca Stress for Various Equilibria

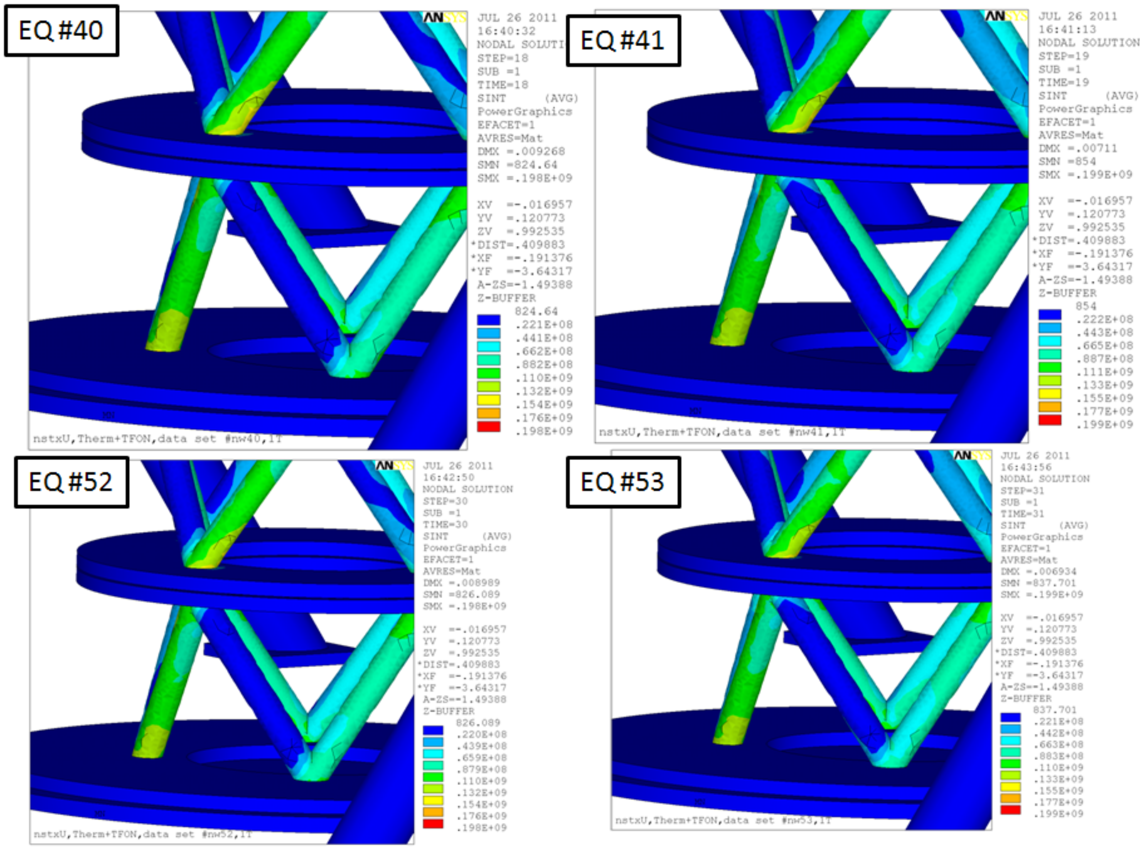


Figure 9.2.3-6 ANSYS Contour plots of Tresca Stress for Various Equilibria

9.3 Seismic Loads

Seismic analysis of NSTX may be found in Reference [6], based on the global model analysis described in reference [2]. Both of these calculations - as of May 2011- were based on the earlier gusseted plate pedestal concept. The global model was re-run with the .5 g lateral load applied, which is representative of the seismic response based on the more elaborate response spectra modal analysis also included in reference [6] The seismic stress in the V truss is only 40 MPa vs. 135 for a typical operating scenario.

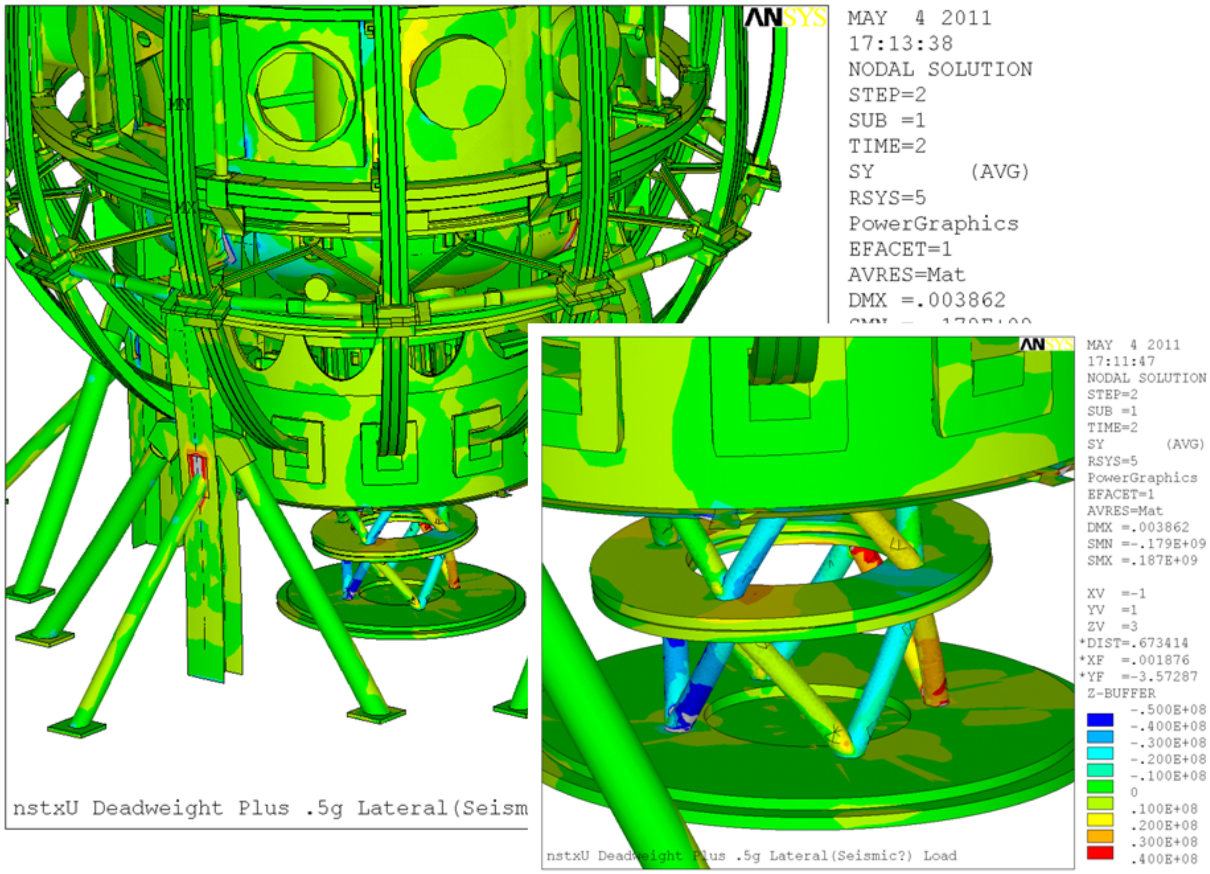


Figure 9.3-1 Seismic Stresses in the "Vee" Pipe Pedestal Design.

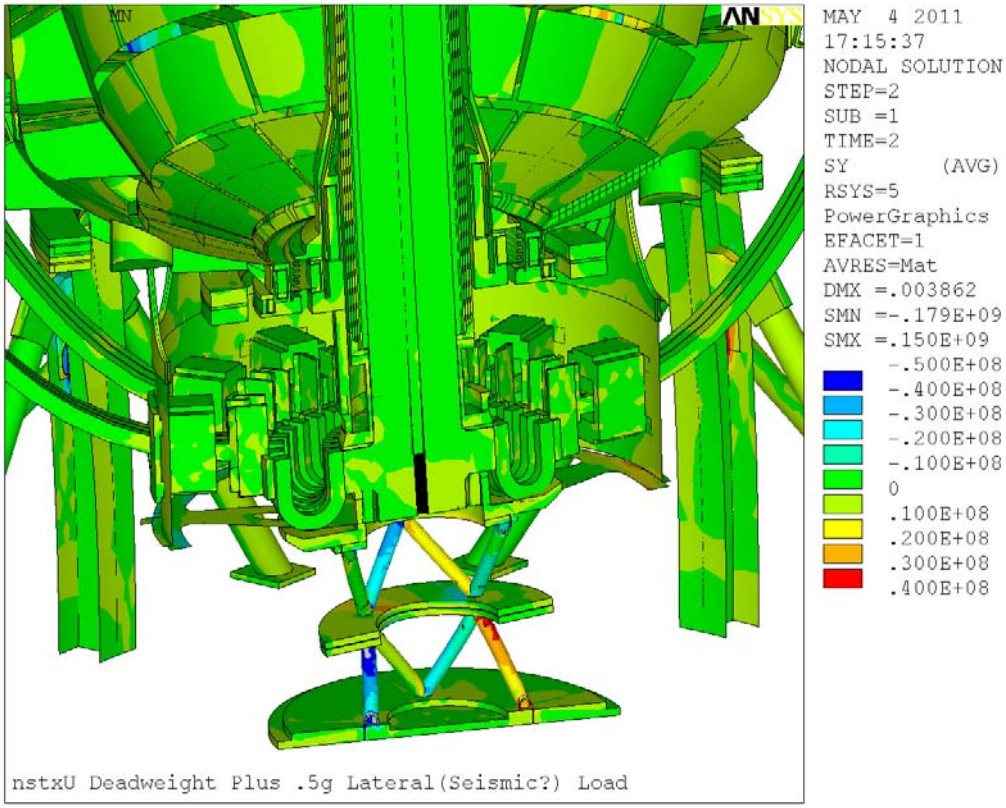


Figure 9.3-2 Section Through the machine with .5g Lateral Accelerations Applied.

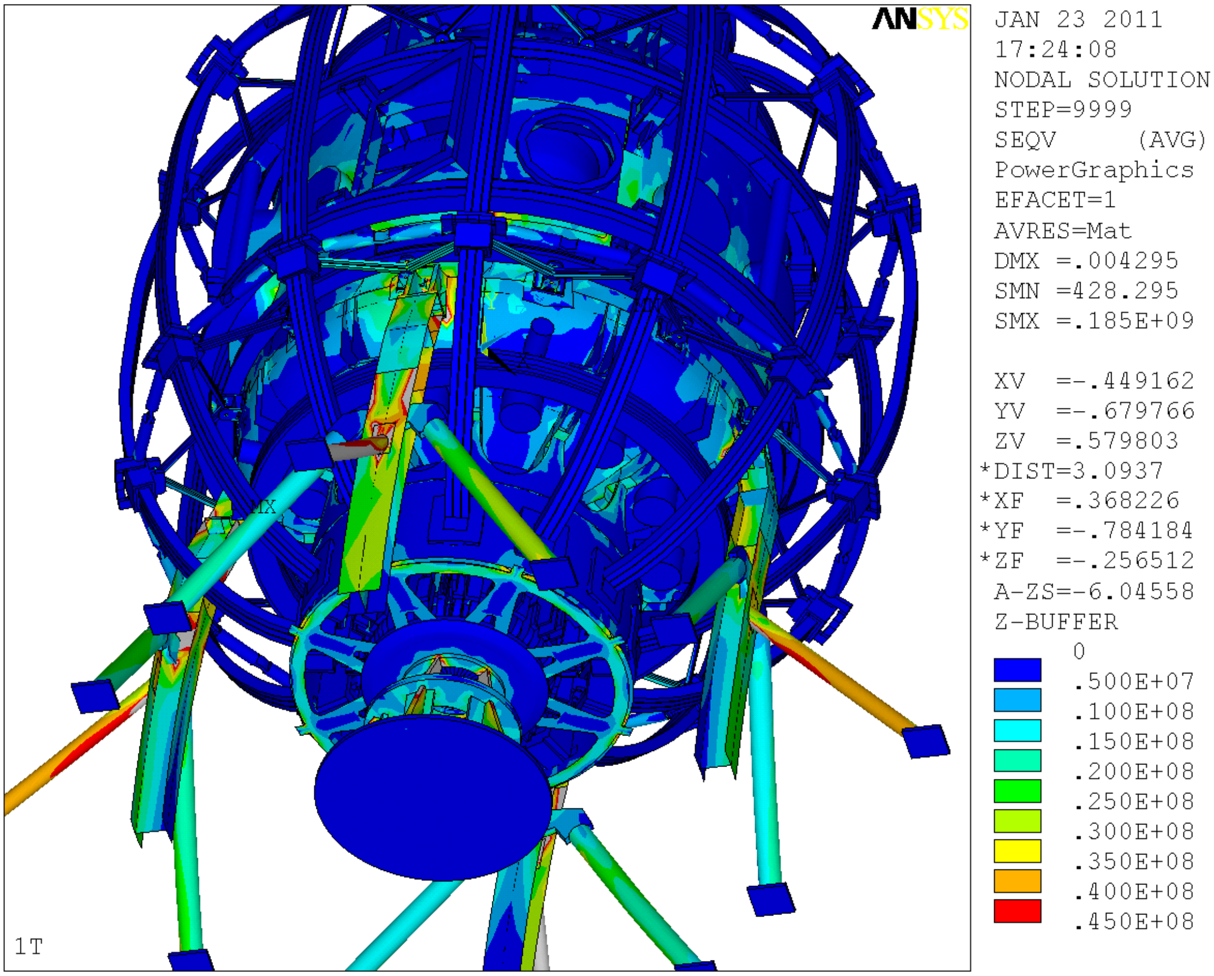


Figure 9.3-3 Plate Gusset Pedestal Seismic Stresses From Reference [6]

The seismic stresses in the pedestal are modest for both pedestal concepts.

9.4 Halo Loads

The stiffness of the pedestal and lower lid partially determine where the halo load goes. In the spoked lid calc it is claimed that the pedestal is stiff enough that it will see the halo from the centerstack casing and that the halo load from the passive plates is carried through the spoked lid[12] and reacted at the pedestal[24]. The halo loading (from Art) on the skirt and skirt to pedestal bolting is addressed in the centerstack casing calc: NSTXU-CALC-133-03-00[14], and its revision, [17] . The halo load is tracked more rigorously in the bellows calculation[15]. Judgmentally, if the (upper) bellows can take the centerstack load, the heavier skirt, pedestal, vessel, spoked lid structures will be able to take the load. A. Brooks[14] and P. Rogoff [15] trade reaction forces at the bellows. The halo loads are dynamic impulsive loads and are treated in a very conservative manner in this global model calculation as static loads - P. Rogoff and A. Brooks reduced the loads by including P. Rogoff's bellows stiffness in the dynamic analysis. Art further reduced the reaction at the pedestal by including the compliance of the G-10 ring that connects the casing to the TF lower flags [17]. In the global model calculation an earlier estimate by A. Brooks of 50,000 lbs is applied on the upper and lower region of the centerstack. This is also the basis for the loads that A. Zolfaghari uses to qualify the TF crown bolting calculation.

Halo Loads are included in one of the static load cases in the global model run - Figure 9.4-1 shows the earlier pedestal model and the stresses were low, 135 MPa, and are about the same for the pipe truss design.

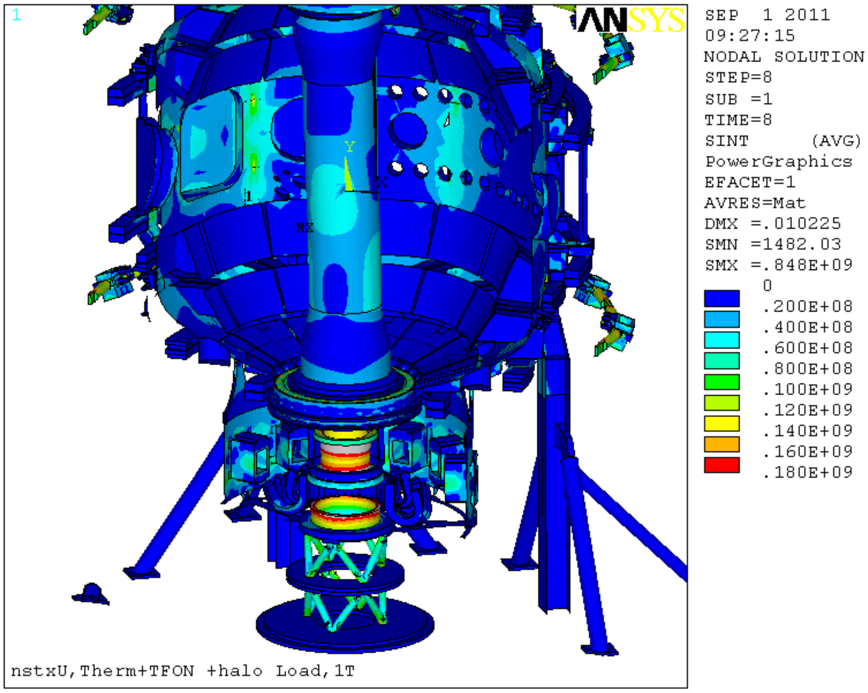


Figure 9.4-1 Global Model Results With 50,000lb assumed Halo Load

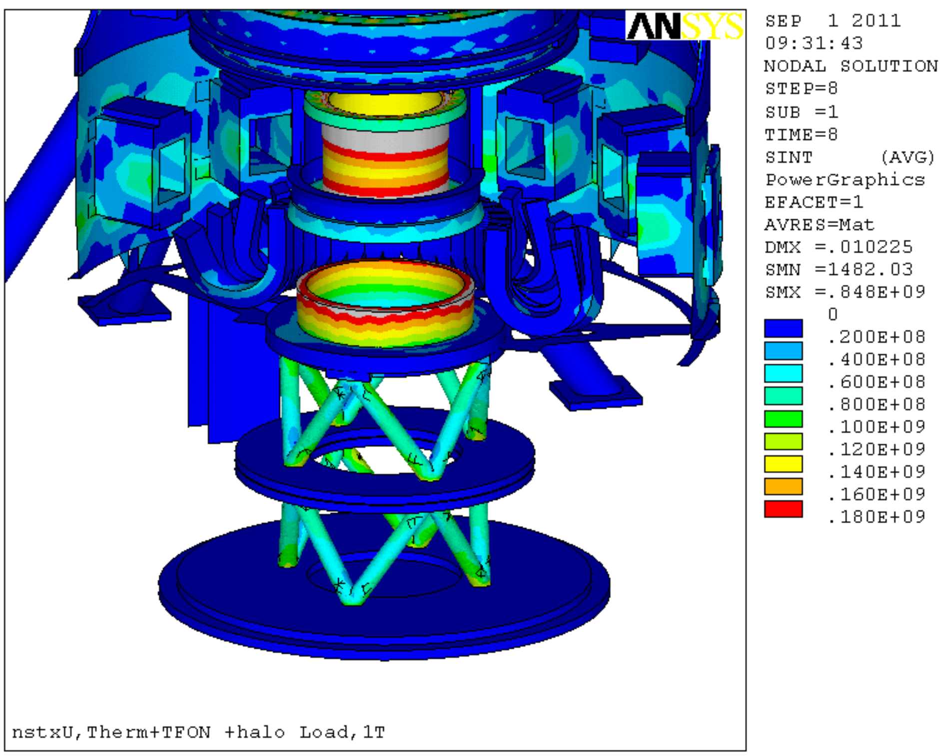


Figure 9.4-2 Pedestal Area Global Model Results With 50,000lb assumed Halo Load

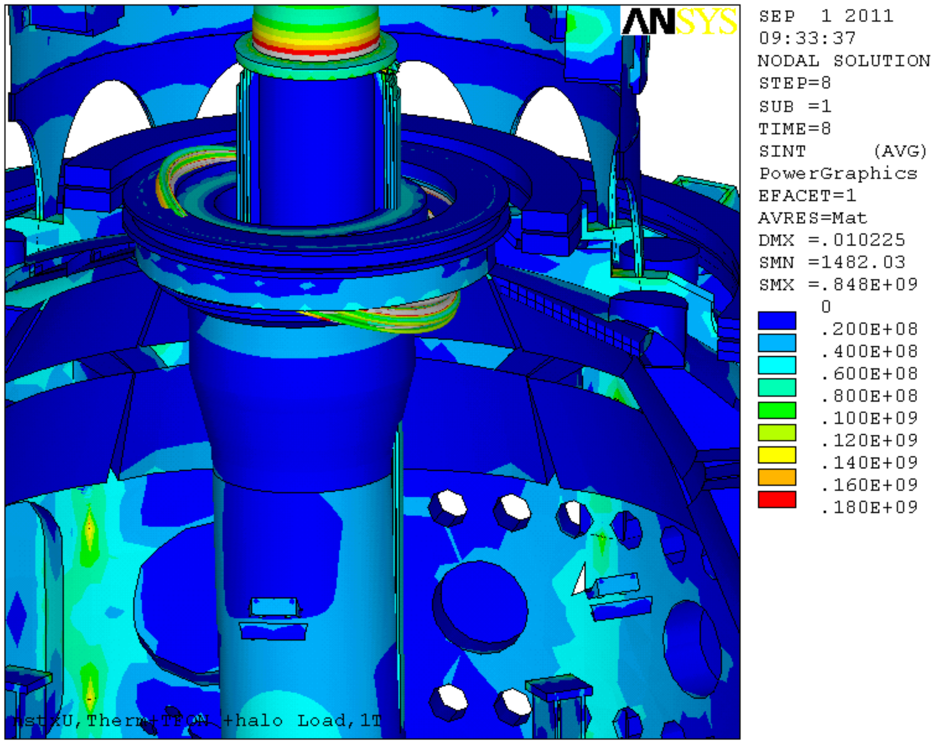


Figure 9.4-3 Global Model Upper Bellows Area Results With 50,000lb assumed Halo Load

Based on the stress levels above, Halo loads have the potential of severely loading the upper bellows. This structural interaction is addressed in the two calculations discussed above, [15] and [14]. With dynamic effects appropriately applied bellows loading is acceptable.

10.0 Bolting Calculations

This section adds halo loading to previously calculated bolt loads from section 9, and summarizes them.

The global model described in [2] was updated with better modeling of the pedestal to allow better quantification of the bolt loads at the intermediate flanges and the concrete anchors. Figure 10.0-1 shows the updated modeling and the torsional shear at "blocks" that model 16 concrete anchors. The number of concrete anchors is assumed at the time of writing this report. This can be scaled based on capacity of the actual anchors used. The updated global modeling is used for the bolts below the pedestal top plate and joining ring. Figure 10.1-1 provides some indication of the naming convention of the bolts and flanges.

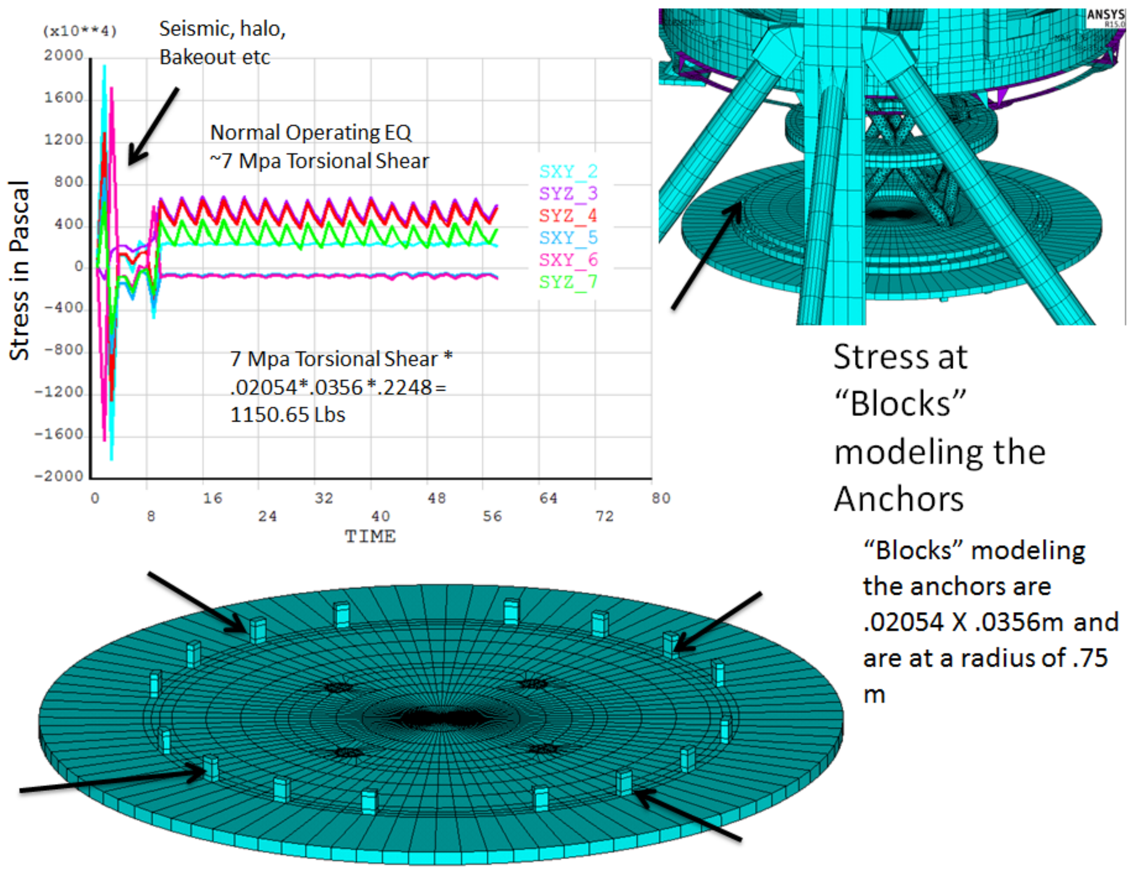


Figure 10.0-1 Revised Global Modeling to Allow Better Post-Processing of bolt and Anchor Loads

10.1 TF to Pedestal Joining Ring Bolts

Figure 10.1-1 is an attempt at tracing the load path from the TF flag to the pedestal and spoked lid segments. Bolting in this area is considered in four calculations. the TF flag key calculation [9], centerstack casing calculation[3], the spoked lid calculation[18], and the pedestal (this) calculation. Loading of this area comes from the global model analysis [2], and the halo load analysis [17]

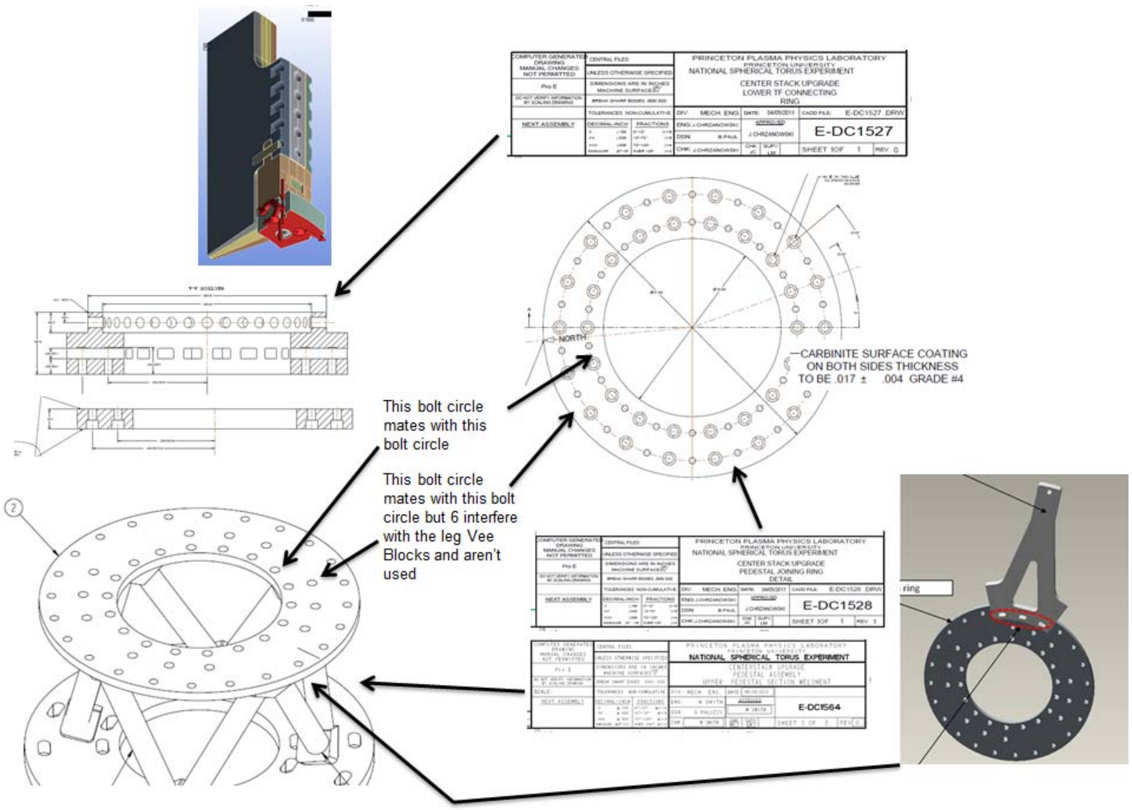


Figure 10.1-1 Upper Pedestal Flange Connections

From [17], and attachment 3, for the pedestal to centerstack connection, halo disruption loads will produce a lateral load of 160kN (36000 lbs) and a moment of 95 kN-m at the upper G-10 ring above the TF flag. The inner two bolt circles must take this load. There are 30 bolts in these two bolt patterns so the friction shear loading that is taken is 36000/30 = 1200 lbs per bolt. The moment will produce vertical loads at the bolts.

```
sect
 Element Group for which Section Properties are to be Calculated
1
  Section Properties for Group Number:          1
 AREA=    12.65657        IXX =     734.3295        IYY =     680.9030
 MAX DISTANCE TO EXTREME FIBER, CX=    12.18700          CY=    12.37500
 RINX=     734.3295          SNX=     60.25516         RAD OF GYR=    7.617063
 RINY =    680.9030        SNY=     55.02245        RAD OF GYR=    7.334739
 PRODUCT OF INERTIA ABOUT ORIGEN=RIXYT = -8.3617872E+07
 PRODUCT OF INERTIA ABOUT NEUTRAL AXIS=IXYN = -8.3617872E+07
 ROTATION ANGLE TO PRINCIPAL MOMENTS OF INERTIA =   -44.99999
 COORDINATES OF THE CENTROID, XBAR=  1.3699980E-06    YBAR=  1.0172250E-0
```

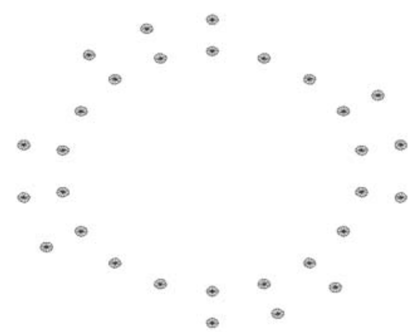


Figure 10.1-2 Section properties of the Bolt pattern that Connects the Pedestal Joining Ring to the Pedestal Top Plate. Bolt areas are modeled as .75 in diameter

The min section modulus of the bolt pattern is 55.022 in^3. The axial bolt stress due to the moment is 95000 N-m*.2248*39.37/55.022 = 15281 psi. The bolts will be preloaded to ~75 ksi so the halo moment will not cause lift-off. The moment will reduce the frictional load on one side but increase it on the other, so it shouldn't affect the lateral load carrying capacity.

The torque from the TF blades is transmitted through the G-10 collar to the pedestal joining ring. Some of this torque is transmitted to the pedestal and most is transmitted through the spoked lid to the outer vessel, but all the load goes through the pedestal top plate. The design load for the G-10 ring is 9000 lbs per TF blade[9]. This includes some allowance for halo loads so halo loads are double counted in this evaluation. The total perimeter load is 9000*36 = 324000 lbs which is reacted by 30 bolts or 10,800 lbs per bolt. Adding the halo contribution from A. Brooks [17], the load per bolt is 10800+1200=12000 lbs These are ¾ inch bolts which are threaded into the joining ring from the bottom. The surface is carbonite coated so a friction coefficient of .6-.15 = .45 is allowed. The bolt tension required is then 12000/.45=26700 lbs. The installation torque is .2*26700*.75/12= 333 ft-lbs. The stress area of a ¾ bolt is .334 sq in. so the preload tensile stress is 79940 psi. From the NSTX Criteria, the allowable bolt preload stress is 75% of yield or 71.25 ksi for the ASTM 196 B8M Class 2 bolts, and 75 ksi for 718 bolts. The overage will be OK because of the conservatism in the original 9000 lbs, the unlikely pairing of worst torque and worst disruption loading and NSTX criteria allows .9*yield with applied loads included.

The joining plate is 2 inches thick and the through hole is threaded with a ¾ inch thread. The pull out shear area with a 1.5 inch engagement is .5*1.5*.685*pi = 1.614 sq. in. The shear stress is 10800/1.6139 = 6691 psi shear or 13382 psi Tresca equivalent. The Sm allowable for 316 Stainless Steel is 26.7ksi (Table 7.0-2), above the shear equivalent stress.

10.2 Pedestal Intermediate Flange Bolting

Bolt loads were computed from shear stress contour plots of the “pad” elements that model the bolted connection. In the summary below the figure number that shows the loading calculation from the contour plot data is listed along with the bolt shear load.

	16 Pedestal Mid Height Flange Bolts Peak Shear load. per bolt	
Normal Operating (EQ 21)	1463. lbs	Figure 9.2.2-4
Normal op plus headroom	1609.3 Lbs	1.1* 1463
Halo	2526 lbs	(Fig 10.2-1)
Normal + Disruption(Halo)	4135 lbs	
Seismic	7058 lbs	(Figure 10.2-3)

The frictional capacity of the 1 inch flange bolts is 4500 lbs torqued to 30000 lbs with a friction factor of .3 -.15 = .15.

For Seismic, take credit for the capacity in shear or 80 ksi*.6051*.6 = 29045 lbs, which is well above the applied load. Seismic+normal+halo is considered an unlikely faulted load, but still 4135+7058<29045

	16 Pedestal Mid Height Flange Bolts Peak Tensile load. per bolt		
Deadweight	-.217e6 N*.2248/16 =-3048lbs	(See Figure 9.1-2)	-
Normal Vertical	1274 lbs		
Faulted Power supply	23468		

Dead weight offsets the normal DCPS protected vertical load. The 1 inch bolts can take the faulted power supply load. The bolt capacity for faulted loading is 80 ksi*.6051 = 48,400 lbs

The loads were based on the worst loaded bolt from an elastic linear analysis. During a seismic event, the bolts will slip, and redistribute the load more evenly to all 16 bolts. Looking at the contours in the figure 10.2-3, it looks like the average shear is well below half the peak, making the average below the 4500 lb frictional capacity,. Loading the bolts in shear produces a 29045 lb capacity per bolt, If just a few load in shear, there is plenty of capacity.

The conservatism on the friction factor is not appropriate for a one time seismic loading - If a realistic friction factor were used, the capacity of the bolts is 9000 lbs per bolt, more than 7058 lbs per bolt. In an earthquake you want some bolt slippage to increase damping,

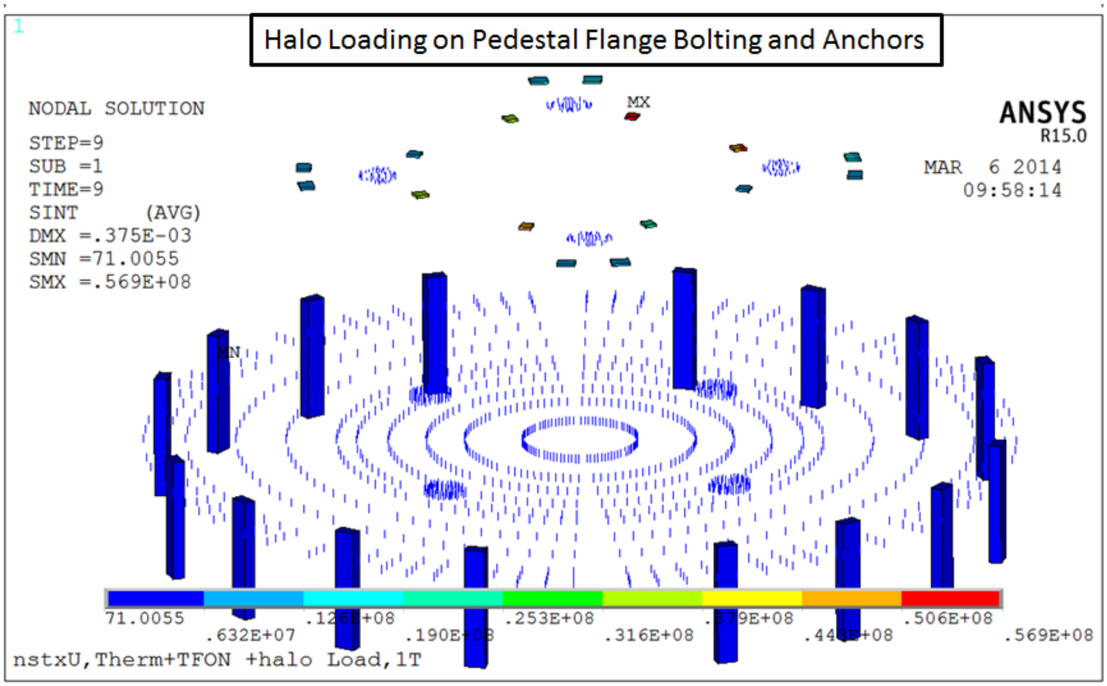


Figure 10.2-1

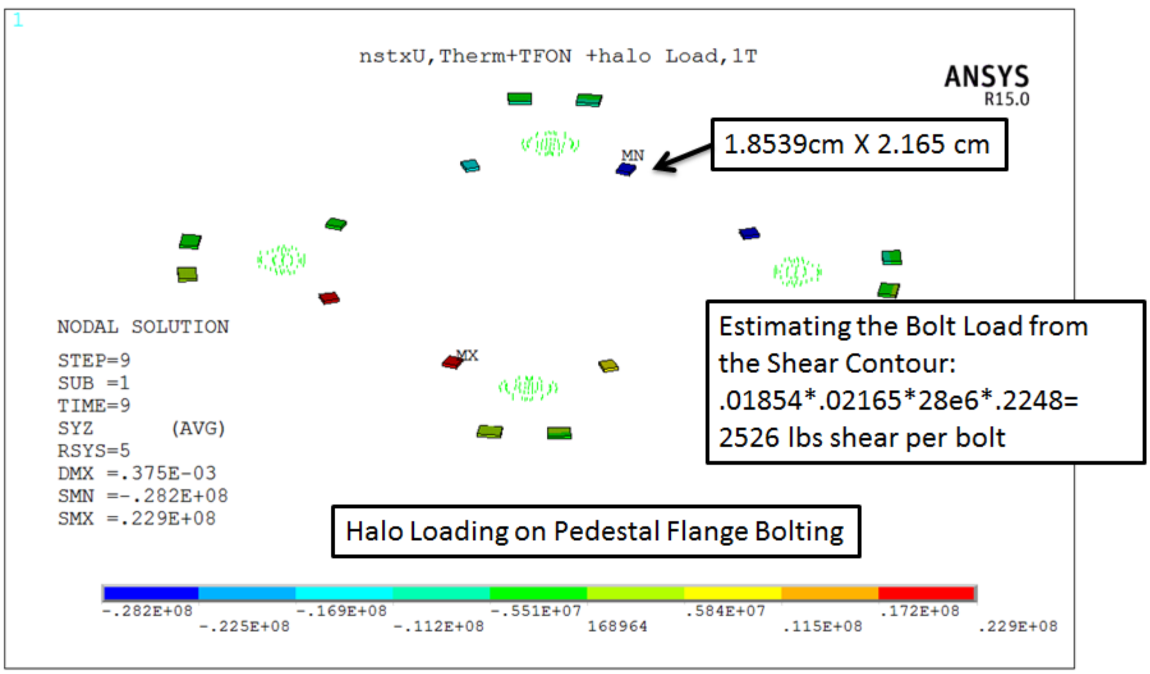


Figure 10.2-2

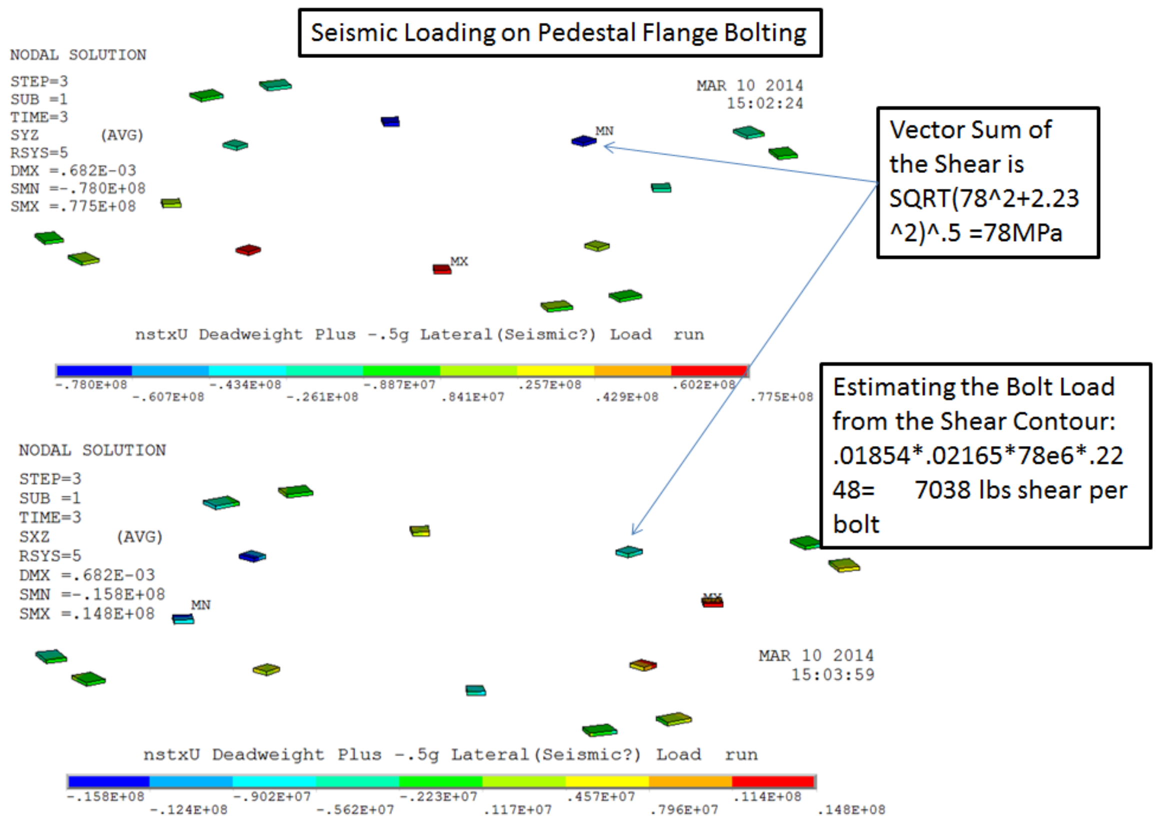


Figure 10.2-3

10.3 Anchor Loads

Anchor loads were considered in section 8.3.2. At the time of the initial issue of the calculation, the number, type and position of the anchors had not been finalized. As of April 2014, anchors have been chosen, but the number and position of the anchors will be worked out in the field to avoind rebar and other obstructions. In the following discussion, 16 uniformly distributed anchors are analyzed. Loads in the following tables are derived from stress contour plots of “pads” in the global model that represent the bolts.

	16 Anchor Bolts Peak shear Load per anchor	
Normal Operating	1150 lbs	Figure 10.0-1
Normal op plus headroom	1265 lbs	
Halo	1380 lbs	Figure 10.3-1
Normal + Disruption(Halo)	2645 lbs	
Seismic	3228 lbs	Figure 10.3-5

	16 Anchor Bolts Peak Tensile Load per anchor		
Deadweight	-.217e6 N*.2248/16 =-3048lbs	(See Figure 9.1-2)	-
Normal Vertical	20397/16=1274		
Faulted Power supply	375500/16 = 23468		

16 anchors are needed, with a shear capacity of 2645 lbs each for normal loading, 3228 lbs for seismic loading. Tensile loads are offset by deadweight for normal and seismic loads, Any added tensile capacity will give margin for the DCPS.

Anchor Design Loads
1” CS-100 Unisorb 6834 lbs Tension 3520 lbs Shear (From Attachment 4)

The loads were based on the worst loaded bolt from an elastic linear analysis. for seismic, assume the anchors slip and redistribute the load more evenly to all 16 anchors. Looking at the contours in figures 10.3-3 and 4, it looks like the average shear is well below half the peak, making the average below the 3520 lb frictional capacity. Seismic loading is a faulted load

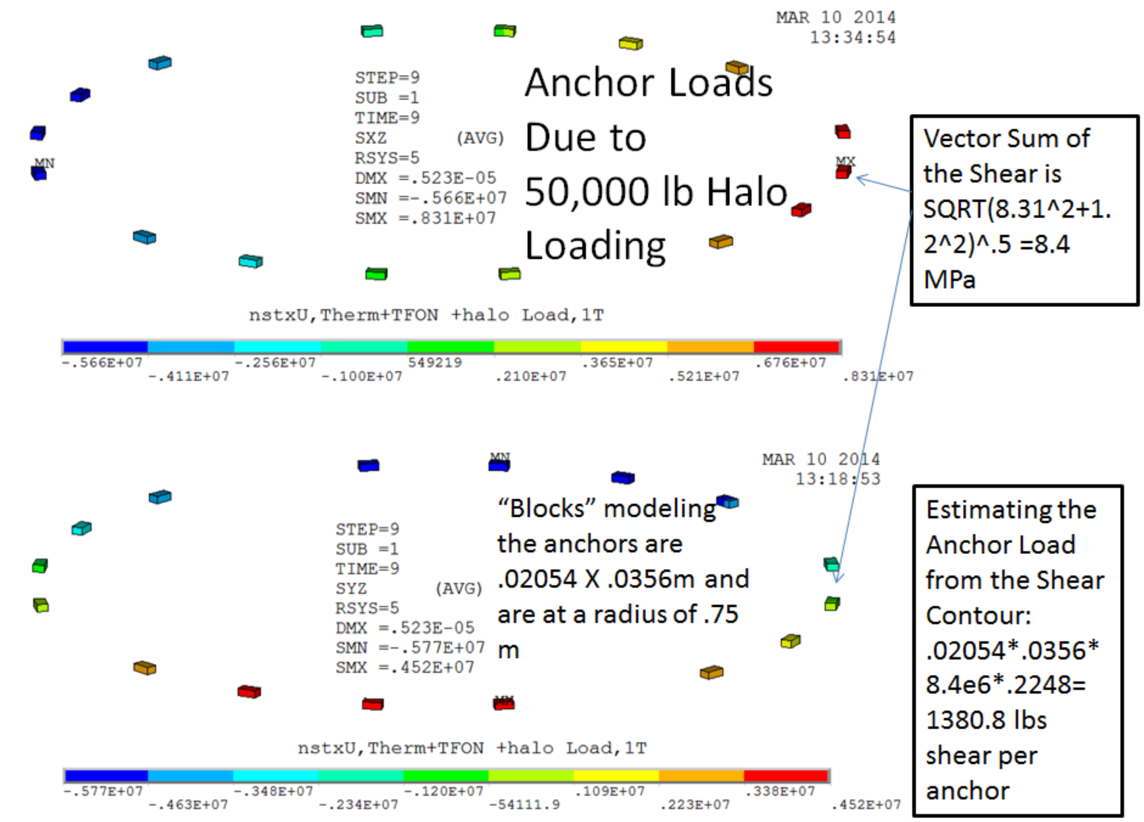


Figure 10.3-1

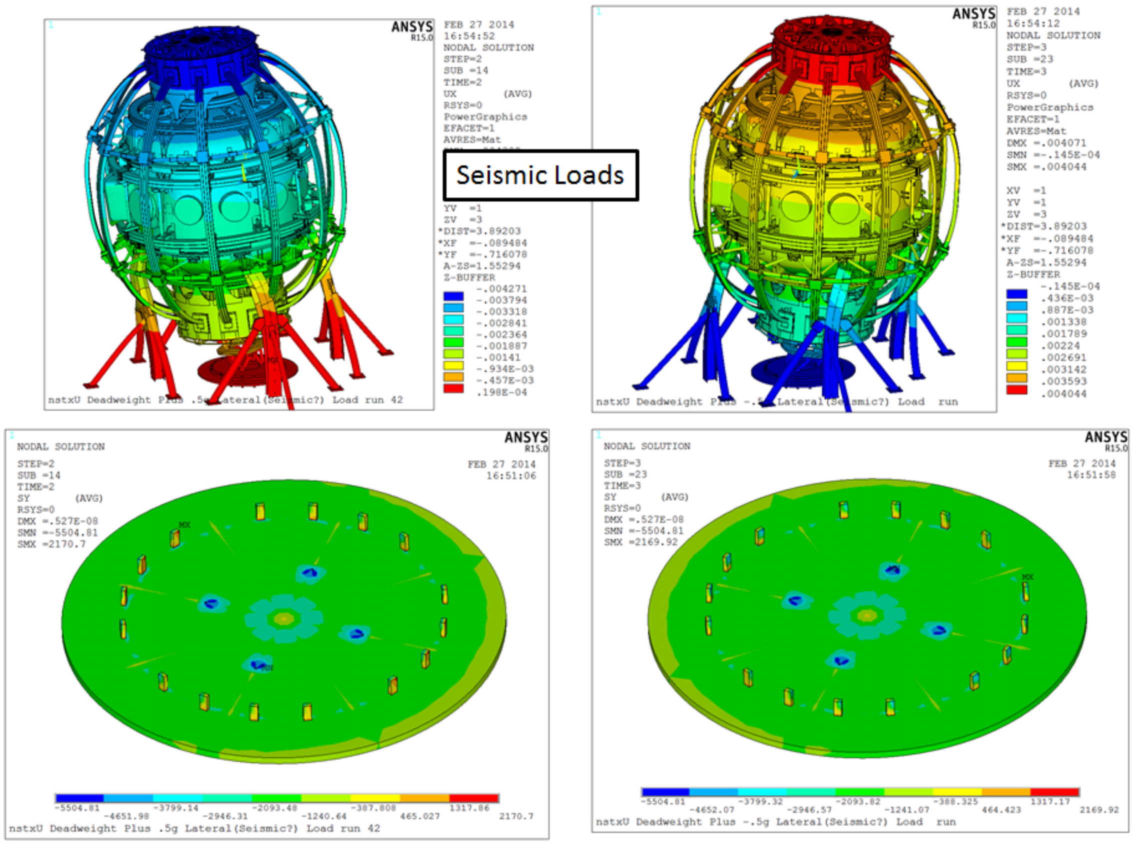


Figure 10.3-2

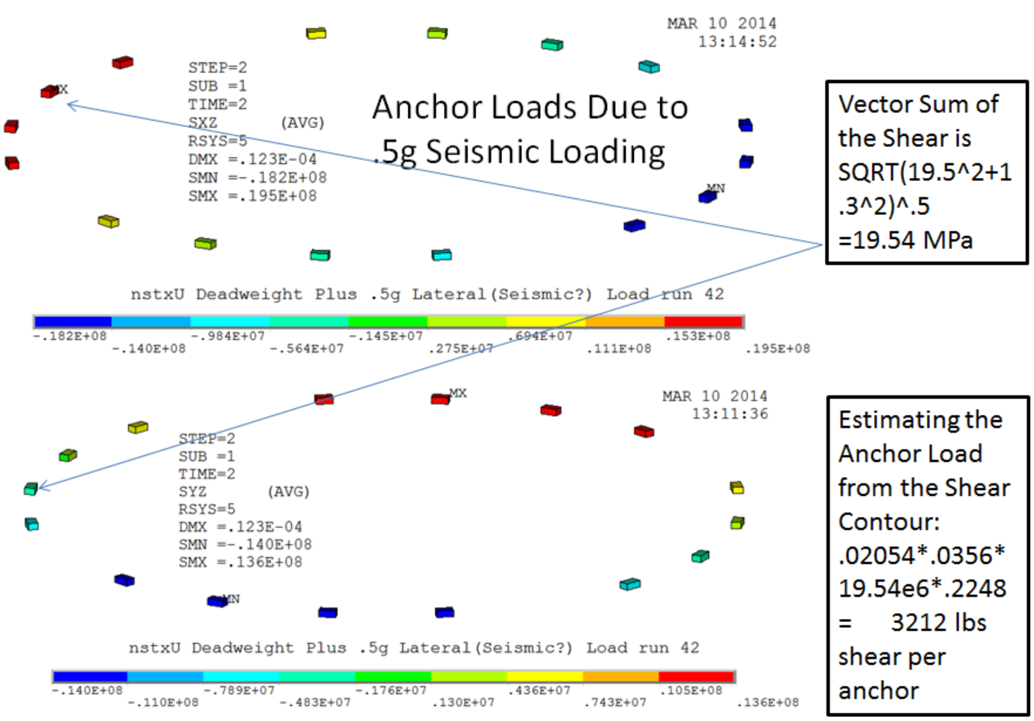


Figure 10.3-3

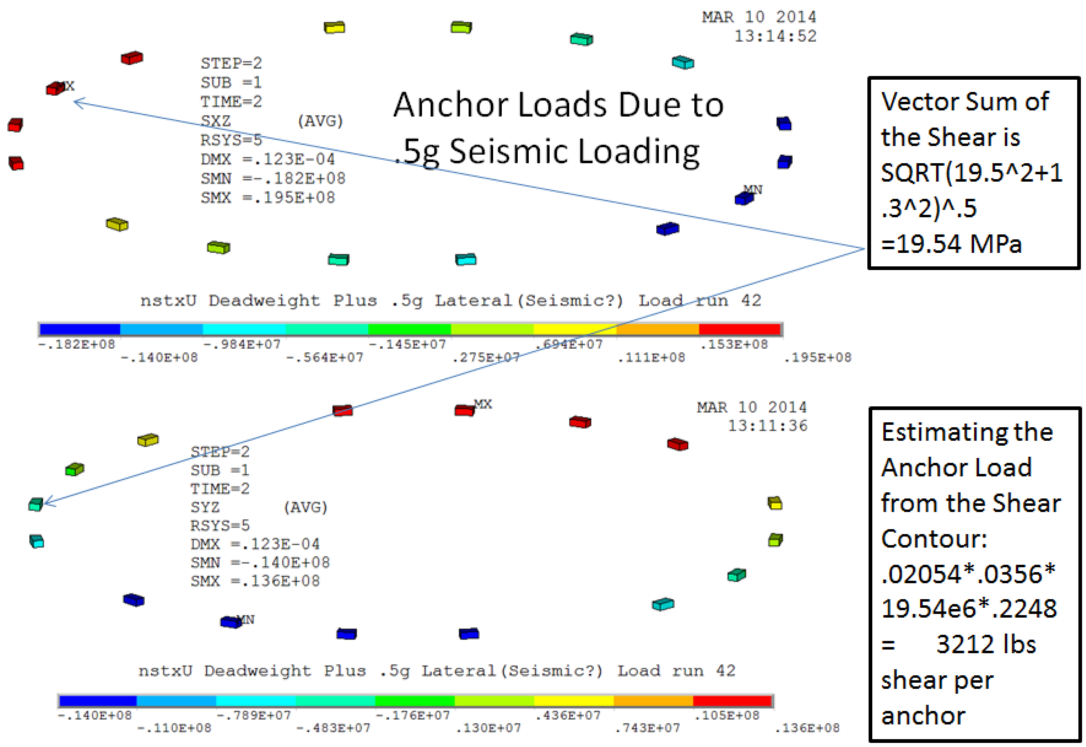


Figure 10.3-4

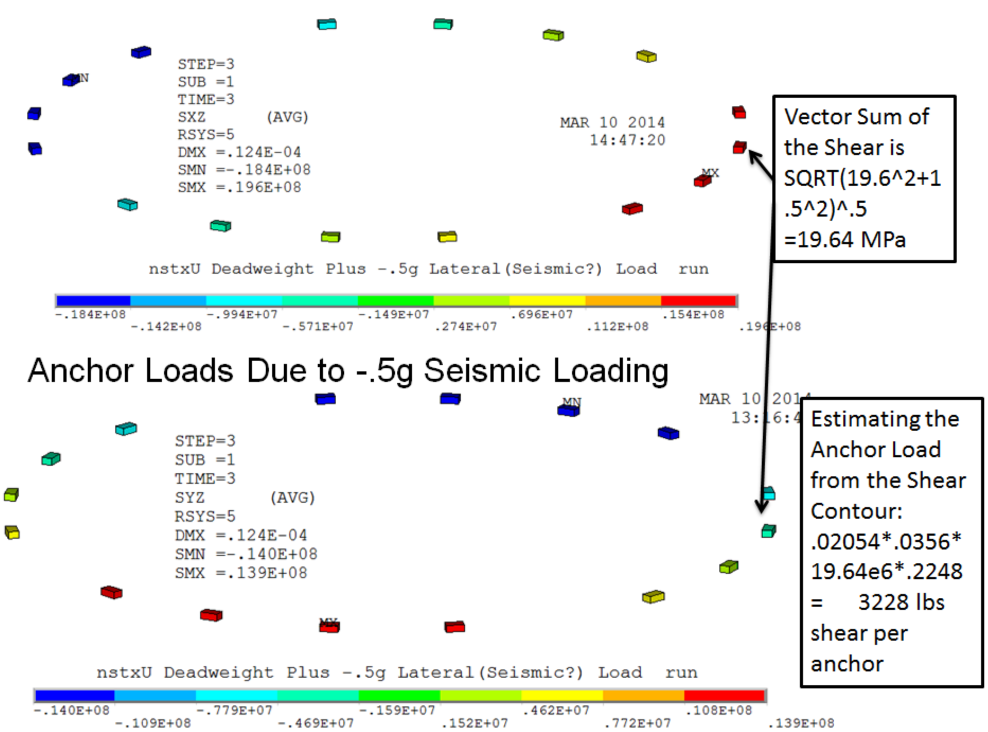


Figure 10.3-5

Attachment 1 Ref 11 text

Thu 3/11/2010 8:21 AM

Peter,

Summing up the applied halo forces for the resistive distribution scenario (for the strike at z=+/-0.6m) with PF and TF (1/R) fields I get:

Applied Load Sum on CS

Fx = -30695.6 N, Fy=Fz=0
Mx = 80400.7 N-m, My=Mz=0

I ran these thru a stress pass constraining all the points on the top and bottom flanges and looked at the reaction loads:

Reaction Loads on CS when Upper&Lower Flanges Fully Constrained

	Fx, N	Fy	Fz	Mx, N-m	My	Mz
Up	15347.	32464.	44662.	-40200.9	56846.7	-201.8
Low	15349.	-32463.	-44661.	-40199.6	-56848.9	201.8

The sum of the Up and Low values do add to negative the applied loads as expected. It just highlights the need to look at the reaction moments as well when considering support design loads.

Art

Attachment 2 Ref 16 Email Text

Ref [16] May 14 email from M. Smith with recommended friction coefficient

We have friction test results for carbinite coated SS against G-10.

The test report shows photos of the carbinite samples from the vendor.
But, the report doesn't match the sample with the carbinite coating thickness received.

By visual comparison of the report photos, I'd say sample 4 matches the coating we plan to use.
The friction coefficient associated with this sample is 0.689.

Being really safe matching the photos, the samples, and the coating thickness we'll specify, I'd use a **friction value of 0.6 mu for the analysis.**

However note, the friction test were for the lid-crown interface, therefore carbinite coated SS and G-10 were used. If we're concerned about the design margin, additional testing using all SS parts can be performed.
Please advise and thanks

Attachment 3 Email Text from Ref 17

Nov 4 2011 email from Art Brooks

Peter,

Adding the compliant G10 plate and structure sitting on the TF flags has reduced the moment (now measure at the G10, z=-2.7m) to a peak of 95 kN-m during the dynamic response. The net lateral force has dropped to 160 kN. The bellows/bumper reaction drop slightly to 200 kN and again is not in phase with the reaction load at the base (see attached plots).

Art

This is documented in "Halo Current Analysis of NSTX CS Calc-133-05-011_r1" [17]. In a 12-19-2013 email Art updated the results for a compliance calculated by Len Myatt in the inner PF calculation:

Peter,

For consistency I reran the Halo current structural analysis with Len's 420,000 lbs/in stiffness. The bellows displacements are back to being ~ .5 mm (with the larger stiffness at the support the displacements were less than .1 mm). The bellows reaction loads are also much less (40 kN vs 200 kN). The loads at the base of the pedestal and at the base of the CS are about the same. Art

So for the Pedestal, a lateral load of 160kN and a moment of 95 kN-m at the G-10 ring are appropriate loads

Attachment 4 Unisorb Data (Sent by Mark Smith)

Email from Mark Smith, Feb 25, 2014

"The pedestal anchors are Unisorb capsule anchors, not hilti brand.
The calculation sites hilti brand." Attached are the Unisorb load specs.
The capsule anchor is the CS100 or 1 inch capsule.
The plan was to use the internal threaded insert which then requires a 3/4 inch bolt/stud.

UNISORB®
CAPSULE ANCHOR SYSTEMS

CAPSULE ANCHOR WITH STUD ASSEMBLY

This system provides a superior method of heavy duty anchoring using a high strength adhesive to retain a threaded rod and other materials such as rebar in concrete or other masonry material. The system consists of a glass capsule containing the proper proportion of base resin, hardener and aggregate for the anchor, an appropriate length stud with washer and nut and a drive unit to allow the stud to be inserted into a standard hammer drill.

To install the anchor a clearance hole is prepared and a capsule is inserted. The stud is driven into the hole with a standard hammer drill using the drive unit.

This action breaks the glass capsule and mixes the premeasured components. At room temperature the anchor nuts may be torqued down within approximately 30 minutes.

Extensive testing and field trials have proven the UNISORB Capsule Anchor Systems are among the most dependable on the market. They are far superior to expansion type anchoring systems, and stronger than the concrete itself.

SPECIFICATIONS

Anchor Size	Capsule Number	Drill Dia.	Hole Depth	Anchor No. & Length	*Allowable Tensile Load	*Allowable Shear Load
in.		in.	in.	in.	lbs.	lbs.
mm		mm	mm	mm	kg.	kg.
3/8"	C-38	7/16	3-1/2	S-38 x 5-1/8	2115	1090
9.5		11	89	S-38 x 130	961	495
1/2"	C-12	9/16	4-1/4	S-12 x 6-1/2	3755	1940
12.7		14	108	S-12 x 165	1707	882
5/8"	C-58	11/16	5	S-58 x 7-5/8	5870	3025
15.9		17	127	S-58 x 194	2668	1375
3/4"	C-34	7/8	6-5/8	S-34 x 9-1/2	8455	4355
19.1		22	168	S-34 x 241	3843	1980
7/8"	C-78	1	7	S-78 x 10-1/4	11510	5930
22.2		25	178	S-78 x 260	5232	2695
1"	C-100	1-1/8	8-1/4	S-100 x 12	15035	7745
25.4		29	210	S-100 x 305	6834	3520
1 -1/4"	C-114	1-1/2	10-1/4	S-114 x 15	23485	12100
31.8		38	260	S-114 x 381	10675	5500

Metric dimensions are for reference only.

*ALLOWABLE LOAD DATA

-Tensile loads are based on a 4:1 safety factor applied to the tested bond strength of the adhesive to 4,000 psi (279.9 kg/cm²) concrete.

-Shear loads are based on an F 1554 Gr. 36 anchor and are based on the methods described in the AISC Manual of Steel Construction (Ninth Edition).

-Greater allowable loads are possible by using different anchor materials and by altering the embedment depth. Please contact the factory for information on custom anchoring applications.

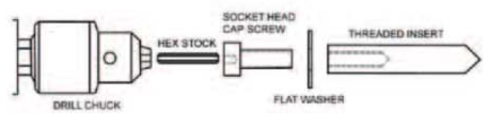
Contact factory for information for drive unit models available for all drill types.

To order specify capsule, stud and driver (if required).

CAPSULE ANCHOR WITH INTERNALLY THREADED INSERT

This system provides a superior method of heavy duty anchoring using a high strength adhesive to retain a threaded insert in concrete or other masonry material. The system consists of a glass capsule containing the proper proportion of base resin, hardener and aggregate for the anchor and an appropriately sized threaded insert. The customer provides an appropriately sized socket head cap screw and washer, along with a short length of hex stock to be used as a driver when inserted into the chuck of a standard hammer drill. The cap screw can then be used to secure the machine.

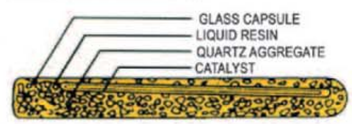
To install the anchor a clearance hole is prepared and a capsule inserted. The threaded insert is driven into the hole with a standard rotary hammer drill using the equipment shown in the illustration below. This action breaks the glass capsule and mixes the pre-measured components. At room temperature the anchor bolts may be torqued down in approximately 30 minutes.



SPECIFICATIONS

Insert Size	Capsule No.	Drill Dia.	Hole Depth	Thread Length	•Allowable Tensile Load	•Allowable Shear Load
in.		in.	in.	in.	lbs.	lbs.
mm		mm	mm	mm	kg.	kg.
3/8 x 4-1/4	C-12	11/16	4-1/4	7/8	3755	1090
10 x 108		17	108	22	1707	495
1/2 x 5	C-58	7/8	5	1-1/4	5870	1940
13 x 127		22	127	32	2668	882
5/8 x 6-5/8	C-78	1-1/8	6-5/8	1-5/8	11510	3025
16 x 168		29	168	41	5232	1375
3/4 x 8-1/4	C-100	1-1/4	8-1/4	2	15035	4355
19 x 210		32	210	51	6834	1980

Metric dimensions are for reference only.



MINIMUM CURE TIMES

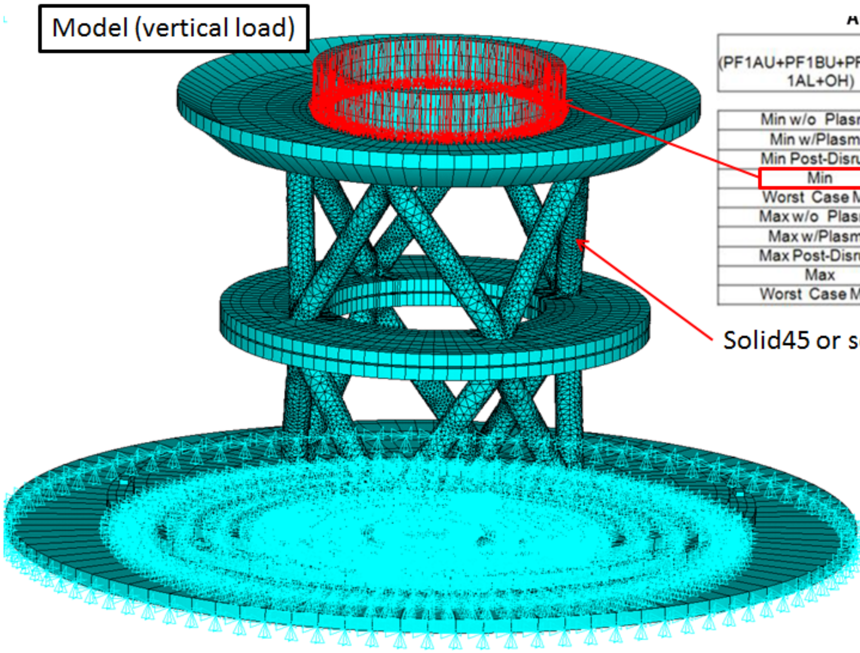
CONCRETE TEMPERATURE	CURE TIME
68° F (20° C) & Over	20 Minutes
50° F to 68° F (10° C to 20° C)	30 Minutes
32° F to 50° F (0° C to 10° C)	1 Hour
23° F to 32° F (-5° C to 0° C)	5 Hours
14° F to 23° F (-10° C to -5° C)	10 Hours

Cure time should be doubled for wet concrete.

To order specify capsule and threaded insert.

Attachment 5
Han Zhang Solid 45/Solid 185 Study

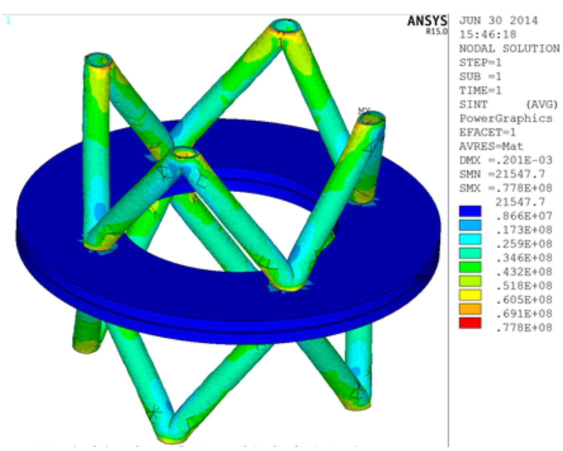
"In your model, tetrahedral shape of solid45 was used for pipe mesh. It seems this shape is only recommended for low stress area. I extract the pedestal from your model and change the elem to solid187 and the stress gets a little higher."



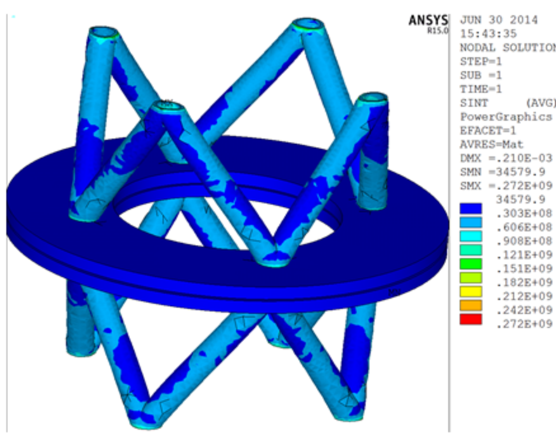
AN

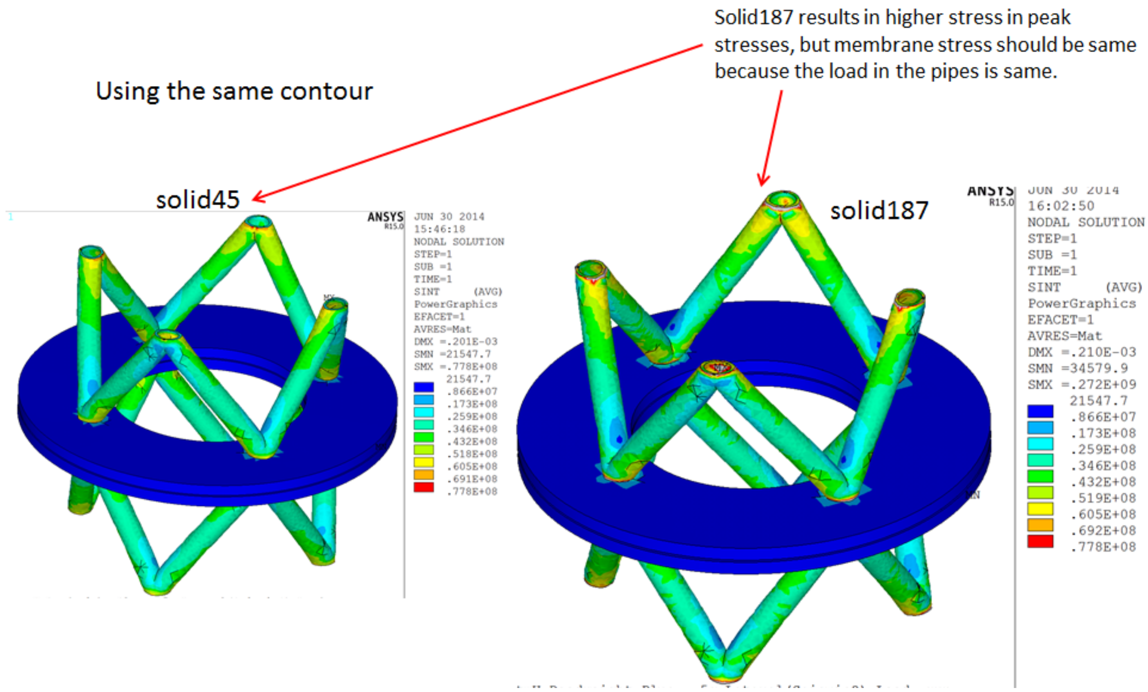
(PF1AU+PF1BU+PF1BL+PF1AL+OH)	Fz(lbf)	Fz(N)
Min w/o Plasma	-39635	-176312
Min w/Plasma	-53445	-237745
Min Post-Disrupt	-41843	-186134
Min	-53445	-237745
Worst Case Min	-375500	-1670374
Max w/o Plasma	20397	90733.99
Max w/Plasma	10748	47811.39
Max Post-Disrupt	19630	87322.06
Max	20397	90733.99
Worst Case Max	375501	1670378

solid45



solid187




Using the same contour
Solid187 results in higher stress in peak stresses, but membrane stress should be same because the load in the pipes is same.
solid45
solid187
ANSYS
R15.0
JUN 30 2014
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NODAL SOLUTION
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SUB =1
TIME=1
SINT (AVG)
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.201E-03
SMN =21547.7
SMX =.778E+08
21547.7
.866E+07
.173E+08
.259E+08
.346E+08
.432E+08
.518E+08
.605E+08
.691E+08
.778E+08
16:02:50
NODAL SOLUTION
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SUB =1
TIME=1
SINT (AVG)
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.210E-03
SMN =34579.9
SMX =.272E+09
21547.7
.866E+07
.173E+08
.259E+08
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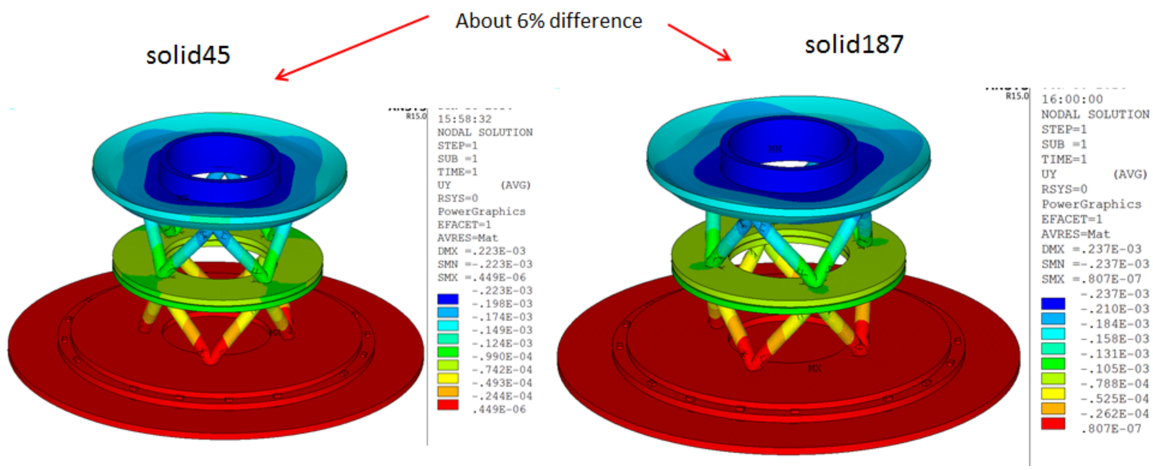
.692E+08
.778E+08


About 6% difference
solid45
solid187
R15.0
15:58:32
NODAL SOLUTION
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SUB =1
TIME=1
UY (AVG)
RSYS=0
PowerGraphics
EFACET=1
AVRES=Mat
DMX =.223E-03
SMN =-.223E-03
SMX =.449E-06
-.223E-03
-.198E-03
-.174E-03
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-.990E-04
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16:00:00
NODAL SOLUTION
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SUB =1
TIME=1
UY (AVG)
RSYS=0
PowerGraphics
EFACET=1
AVRES=Mat
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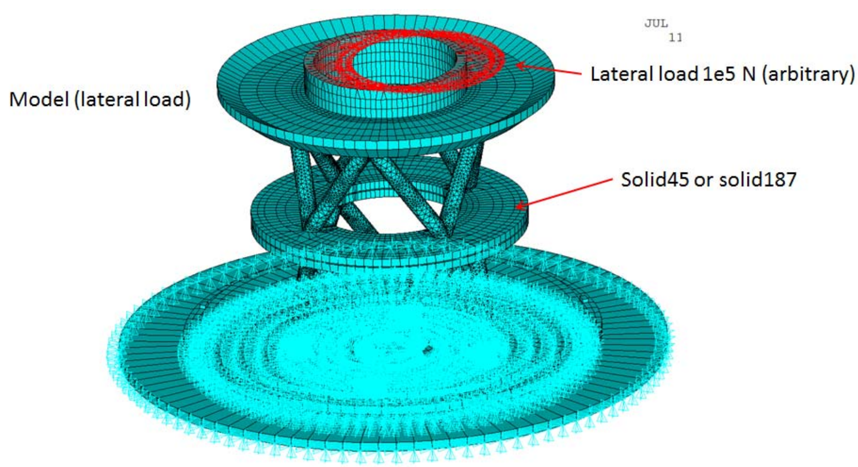
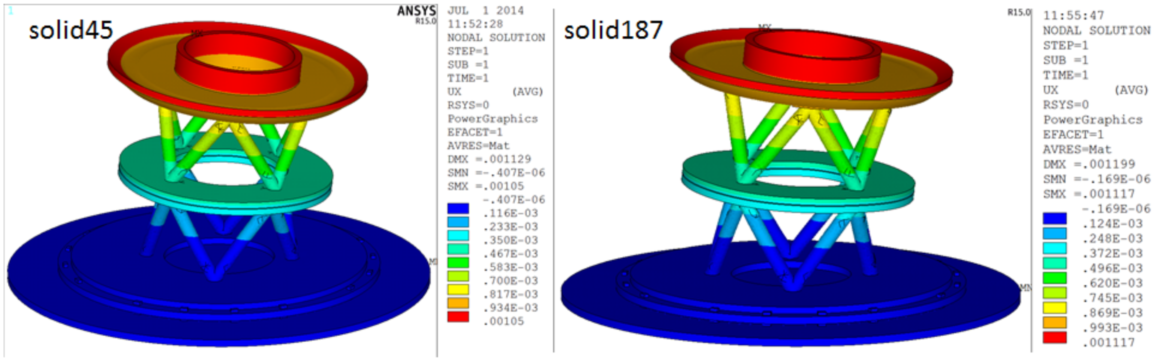
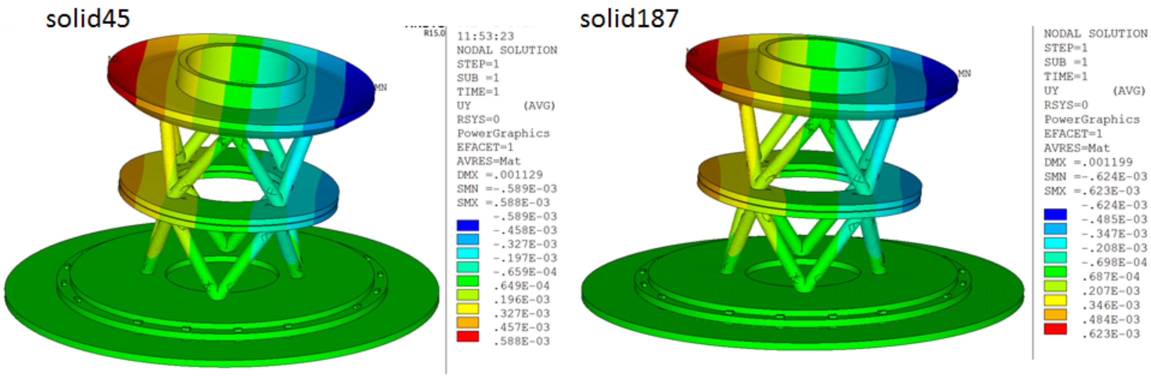
.807E-07

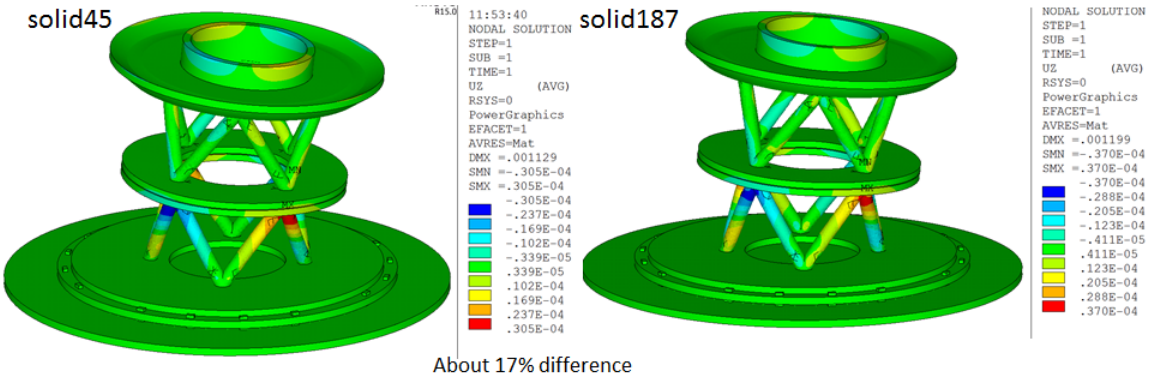

JUL
11
Model (lateral load)
Lateral load 1e5 N (arbitrary)
Solid45 or solid187



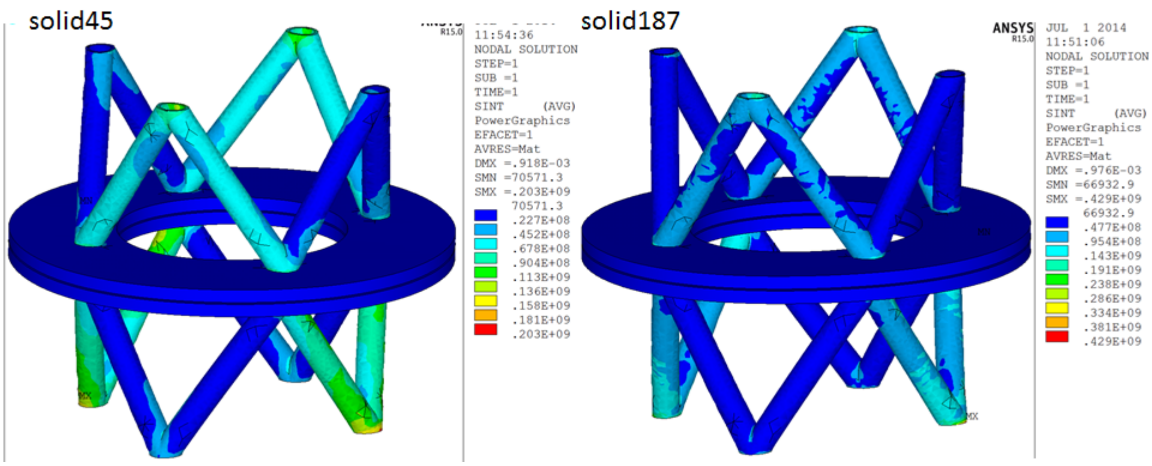
About 6% difference



About 6% difference



About 17% difference



Using same contour

solid45

solid187