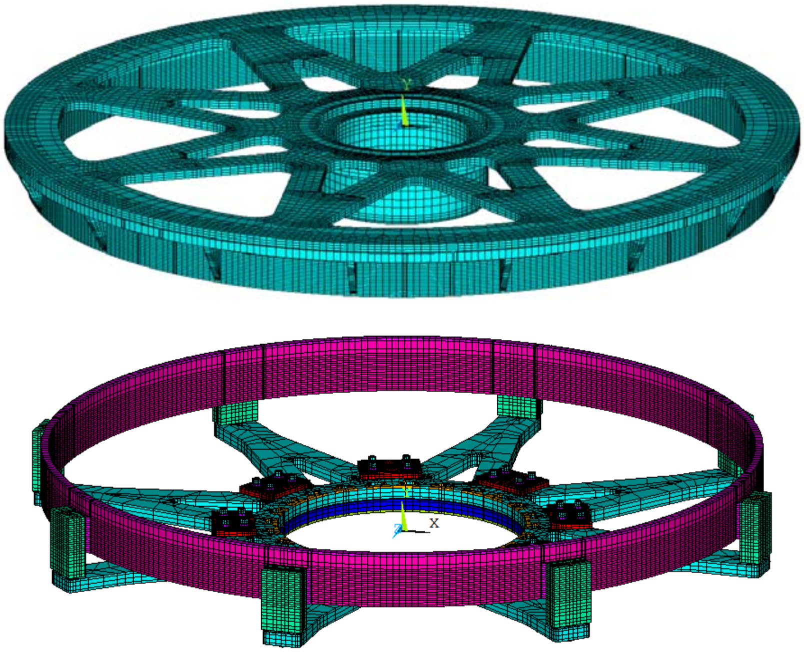
NSTX

Spoked Lid/Flex/Cover Analysis

NSTX-CALC-12-08-02

Rev 1 December 2011
Rev 2 May 2013



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PPPL Calculation Form

Calculation # **NSTX-CALC-12-08-02** Revision # **02** ____ WP #, **1677** (ENG-032)

Purpose of Calculation:

The purpose of this calculation is to qualify the upper and lower lid assemblies. These assemblies bridge between the upper or lower rims of the umbrella structures to the inner TF flags. The upper lid must allow thermal growth of the TF inner leg as it heats up during a pulse. The flexing of the upper lid produces bending stresses. The global machine torques are carried across the upper and lower lids, and also produce bending stresses in the spokes in an orthogonal plane. The primary purpose of this calculation is to qualify the stresses in the lids and also to address the interfaces at the ID connection to the TF flags and the OD interface with the umbrella structure rims.

References

-See the reference list in the body of the calculation

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

Halo loads do not load the upper lid because it is not directly connected to the upper end of the centerstack casing. Halo loads load the lower spoked lid. This is addressed in section 8 of this calculation. Lower halo current loads from the centerstack casing are assumed to be transmitted directly to the skirt and pedestal. Some of the halo current loads from the passive plates and divertor will be carried through the spoked lid. The lower spoked lid with the frictional bolted hub is analyzed with these loads applied.

Calculation **(Calculation is either documented here or attached)**

Attached in the body of the calculation

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

The upper lid lies in a plane and resists the machine torques with only the small offset that results from the thermal growth of the TF. Stresses in the upper lid due to bake-out, normal operational heat up and extension of the TF, are also acceptable. The final design includes a rigid bolted connection at the rim of the umbrella structure. Bolts used at the inner bolt circle are loaded similarly to the bolts that hold the ring to the G-10 crown, and need similar sizing, materials and frictional interface augmentation as the G-10 crown bolts. The outer bolt circle requires the higher strength bolts to react the prying action of the flexing lid caused by the expansion of the centerstack.

The lower lid must be mechanically assembled from segments to allow assembly and servicing of components at the bottom of the machine. Frictional preloaded bolted joints are used. These are qualified in section 6.1 for the normal torque, and Section 8.1 for the torque plus lateral halo load. The lower spoked lid has been designed to be stiff enough to provide torsional registration of the lower centerstack/pedestal and the outer vessel structures, and protect the bellows from unacceptable motions. The 5/8-inch splice plate bolts should be preloaded to 100 ksi, and the one inch inner bolt circle bolts should be preloaded to 100 ksi. Inconel 718 bolts will be needed. The alternating splice plate welds were adjusted in size to find an acceptable weld. 3/8-inch bevel groove welds on three sides were specified, but the missing side turned out to be the most important one. A ½ inch weld was needed where the splice plate overhangs the spoke plate. The other three sides could be decreased from 3/8-inch to 1/4-inch. A minimum friction factor of .35 is needed (0.45 is better). The criteria document requires 0.15 headroom on the coefficient. The Carbinite coating specified can achieve .6 and should be used. The 316, one inch thick pedestal plate can't take the thread shear, and 718 nuts will be required.

Cognizant Engineer's printed name, signature, and date

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

Checker's printed name, signature, and date

Irving Zatz Irving Zatz

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1.2 Table of Contents

1.3 Revision Status Table

Rev1	Added new section 6.1 with more detailed analysis of the flat lower spoked lid- shifted old section 6.1to 6.2 and old 6.2 to 6.3
Rev 2	Added another new section 6.1 with more detailed analysis of the flat lower spoked lid bolted plate hub. The FDR analysis from Rev 1 was shifted to 6.2 and old 6.2 to 6.3 etc
Rev 2	Added figure 1.2-3 to the executive summary along with some discussion of the bolted plate hub design
Rev 2	Added Section 8.1 that analyzes the bolted hub design for lateral halo loads.
Rev 2	Added Appendix D
Rev 2	Updated Figure 3.3-1 consistent with the Pedestal Fatigue Allowable

1.4 Executive Summary:

The purpose of this calculation is to qualify the upper and lower lid assemblies. These assemblies bridge between the upper or lower rims of the umbrella structures to the inner TF flags. The upper lid must allow thermal growth of the TF inner leg as it heats up during a pulse. The flexing of the upper lid produces bending stresses in the spokes.

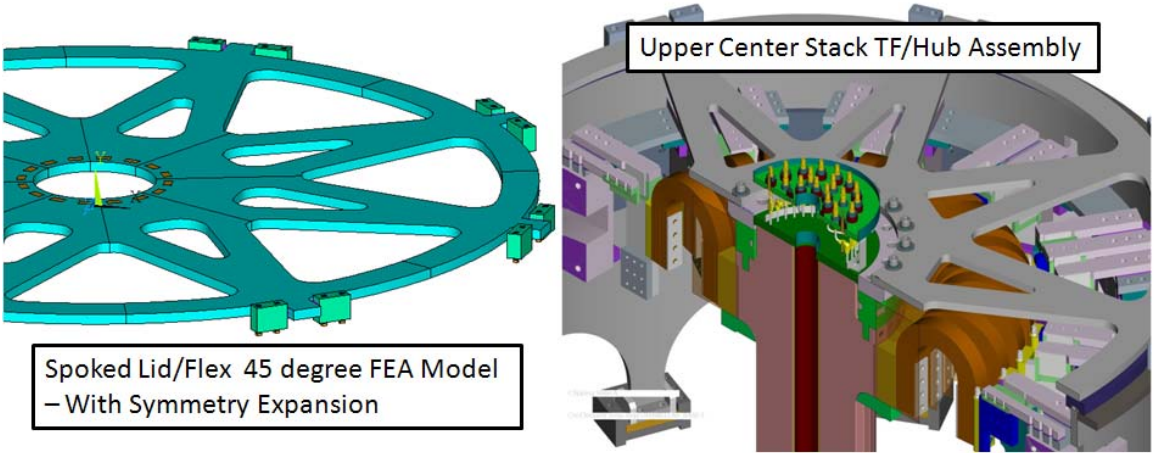


Figure 1.2-1 Earlier Upper Spoked Lid Model (left) and Machine Section (right)

One of the upper spoked lid models is shown in Figure 1.2-1. This had a pinned connection to cleats on the outer umbrella structure rim. The design shown at the May 2011 Peer Review is shown in Figure 1.2-1. This design employs a bolted fixed outer connection to a gusseted flange on the umbrella structure upper rim. This stiffer attachment increases the moments at the ID and OD and this has been investigated by A. Zolfaghari, ref [6]. This stiffer lid also experiences larger stresses. The pinned design had a spoke stress of 17 ksi and the design with a fixed outer ring, has a peak stress of 36 ksi. There is some possibility that the spokes could be thinned to reduce the stresses on the inner and outer bolt circle, if needed.

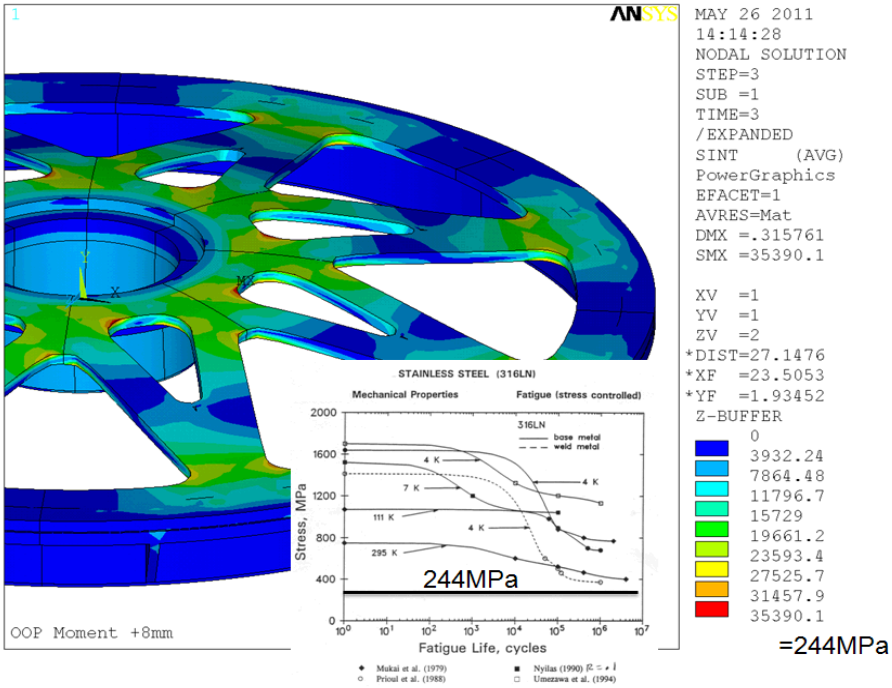


Figure 1.2-2 Upper Spoked Lid Stress with TF OOP Moment and Centerstack Expansion Displacement Imposed. Stress due to Flexure Allows Essentially Infinite Life

The advantage of the FDR design is that the OOP torques are transferred without reliance on any mechanism. The original NSTX had a cogged plate that slid vertically. Fit-up at the cogs was thought to have contributed to flag motion, that contributed to the flag failure.

The global machine torques are carried across the upper and lower lids, and also produce bending stresses in the spokes in an orthogonal plane. The torque load path is redundant. The OOP torque load paths are similarly redundant on the bottom of the machine.

The upper lid lies in a plane and resists the machine torques with the small offset that results from the displacements of the TF. Stresses in the upper lid due to bake-out, normal operational heat up and extension of the TF, are also acceptable. The stresses in the lower lid are also acceptable. The compliance of the bent spokes caused torques and lateral loads to be taken by other structures. There was a concern that having such a compliant member connecting the centerstack to the umbrella structure could introduce relative displacements and loads at the bellows. A flat concept was developed and it is the present FDR design approach.

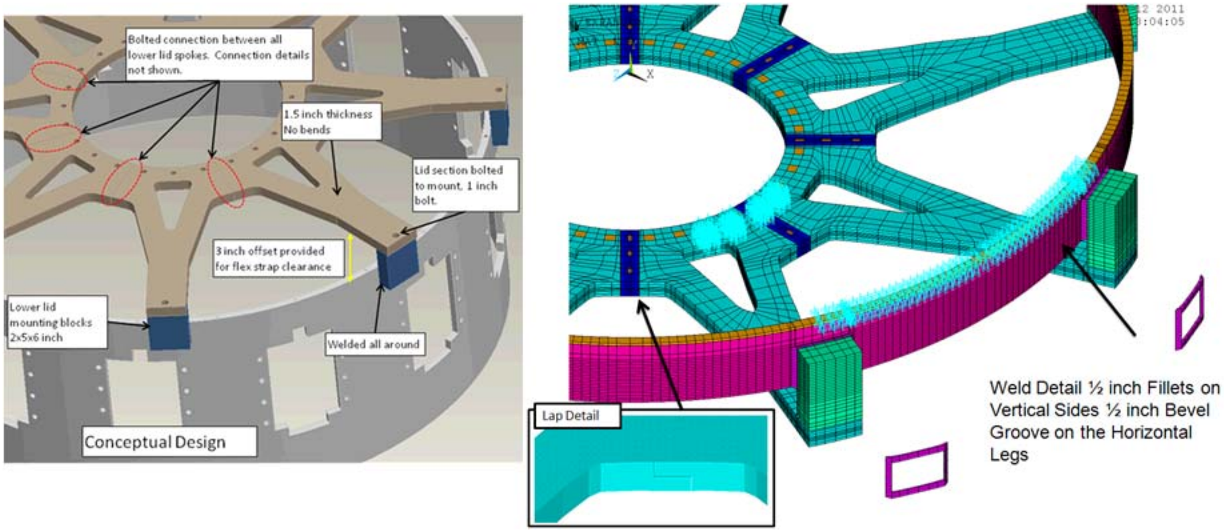


Figure 1.2-2 Flat Lower Spoked Lid Design Inverted for Clarity (Left) Analysis Model (Right)

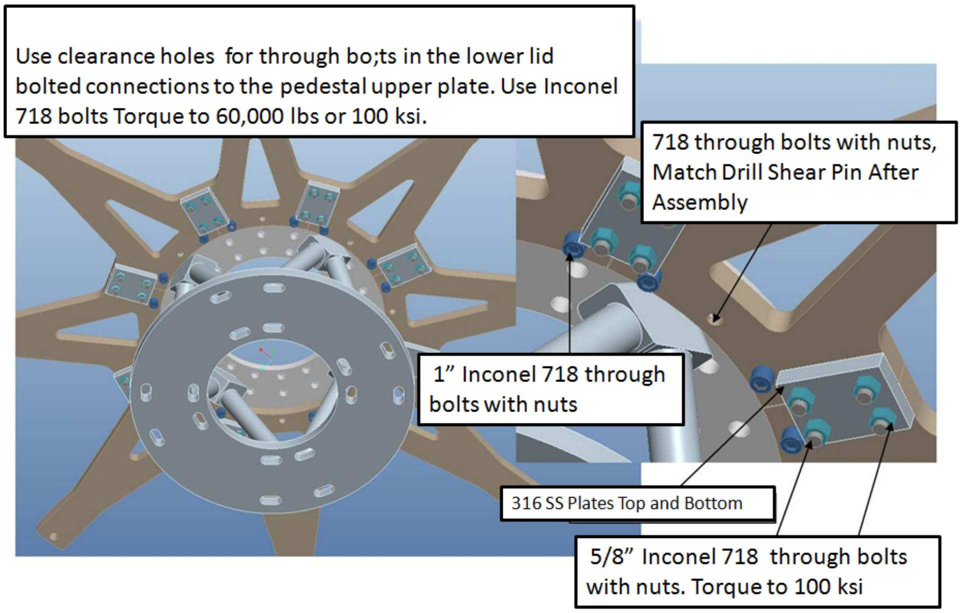


Figure 1.2-3 Lower Spoked Lid Design View from Below, Showing Joiner Plate or Splice Plate Bolting (May 2013)

Different design philosophies are used for the upper and lower lids. The upper lid is the primary torque transmission mechanism between the umbrella structure and the top of the TF flags. A secondary path of the machine OOP torque goes from the outer vessel, through the dome or dished head, through the ceramic break, then across the bellows [7] and into the centerstack casing.

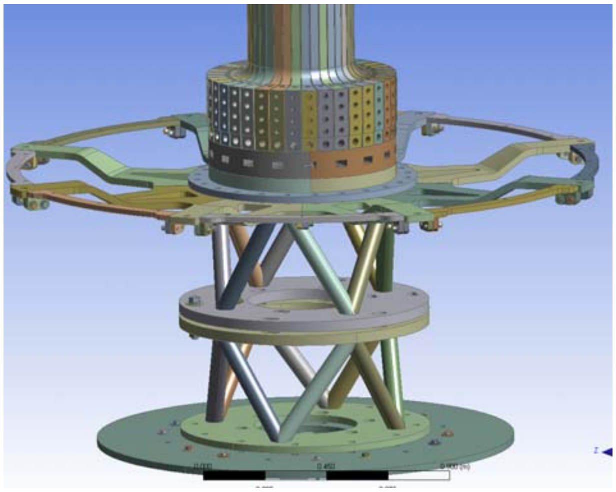


Figure 1.2-4 Lower Spoked Lid with Bent Spokes to clear the TF Straps. The stiffer flat concept was chosen.

The lower lid shares a part of three load paths that carry torque. The first is the spoked lid. The second is the bellows connection between vessel the centerstack casing. These are the load paths used for the upper structures, but the lower torque is also carried by a third load path through the pedestal to the floor and up through the braced vessel support columns to the vessel.

Designs have evolved from the CDR to the FDR. The lower torque load paths used the load path through the pedestal to the floor. This allowed a significant increase in access from below the machine. But compliance of the load path to the floor – particularly involving the vessel support legs made torsional loads in the bellows more significant. The final spoked lid design is torsionally stiff, protecting the bellows vacuum boundary.

The interfaces at the ID connection to the TF flags is addressed by reference [6]. The OD interface with the umbrella structure rims is addressed in this calculation.

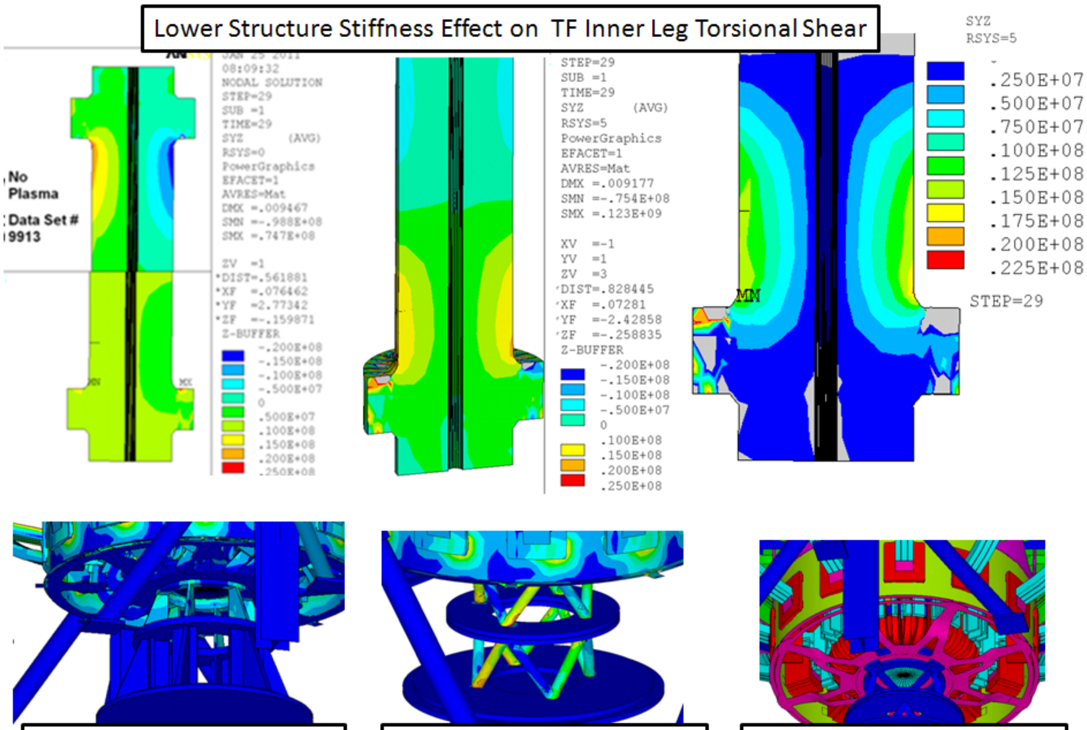
This calculation follows the torque being carried through the upper umbrella structure rim, across the lid assembly and to the upper TF flags [6]. The torque carried in this load path is quantified in the global model described in [1]. The inner TF flags and collar also carry this torque, and interface with the spoked lid. The lid also must allow the vertical growth of the TF inner legs.

The torsional moment for design of the lid/flex/diaphragm bolting and the TF steps or keys from [1] is 0.3 MN-m for the lower lid (Figure 3.2-1) and 0.25 MN-m for the upper lid from [1]. This is the torque being transmitted from the centerstack TF to the outer rim of the umbrella structure. This was translated into a load per TF flag of about 7000 lbs for the upper flag and 9000 lbs for the lower flag.

Loads resulting from centerstack halo currents produce a lateral load and a moment at the lower connections to the pedestal. The bent spoked lid transmitted less of this load to the umbrella structure because of the compliance of the bent spokes. The upper Halo current load inventory goes through the upper bellows to the vessel and not the spoked lid. This effect is addressed in bellows [7] and centerstack casing calculations. This is included in the lid analysis via the 9000 lb load (the OOP torque load is around 7000 lbs). The lower centerstack Halo current load inventory goes through the skirt to the lower TF flag teeth/pins and splits between the pedestal and to the lower lid to the outer vessel leg supports. The lower TF G-10 collar must take the torques, the centerstack halo loads and the launching loads (OH + PF1a/b-U&L). Halo loads on the vessel and passive plates may go through the lower spoked lid being shared by the vessel support legs and the pedestal. The spoked lid bolts were checked for the full loading from the GRD specification of 700,000 amps across the face of the passive plates. This full load inventory is not expected to be applied to the lid; some will appear in the vessel support legs and be reduced by the inertia of the tokamak. Section 8.1 qualifies the frictional bolted hub design used for the lower lid, for halo loads.

The global model described in reference [1] was updated with the lower pedestal and spoked lid designs. This provides a means to qualify the stresses in the spoked lid, but the main purpose of including the lid in the global model is to address the need for torsional stiffness or compliance of the plate to ensure that the inner leg torsional shear stress is acceptable with the FDR configurations. The concern comes from the relative compliance of the bent spokes in the lower lid. Figure 4.0-3 shows the global model of the tokamak including the upper and lower spoked lids.

The hub/collar section is as Mark Smith and Jim Chrzanowski had designed - with only preloaded bolts and friction torsionally connecting the spoke/lid to the collar. Since the moment caused by the 8mm thermal expansion of the centerstack appears to impose minimal stresses on the collar, the outer lugs that connect to the umbrella structure flange can be pinned connections. The vertical growth can be absorbed by flexure of the spoked lid and a little flexure of the collar. Ali Zolfaghari preloaded the 18 bolts to 50,000 lbs each [6].



Rotationally Compliant Pedestal/Stiff Lower Lid

Rotationally Stiff Pedestal/Stiff Lower Lid

Rotationally Stiff Pedestal/Compliant Lower Lid

Figure 1.2-5 Effect of Pedestal and Lid Stiffness of TF torsional shear

The FDR chosen design for the lower lid is a flat, relatively stiff, spoked "wheel". This was chosen over the bent spoke design which was too compliant to protect the bellows from relative motions and lateral loads from halo loading and from global machine TF OOP loading. Figure 1.2-6 shows the load path between inner and outer vessel structures that would result from a weak lower lid.

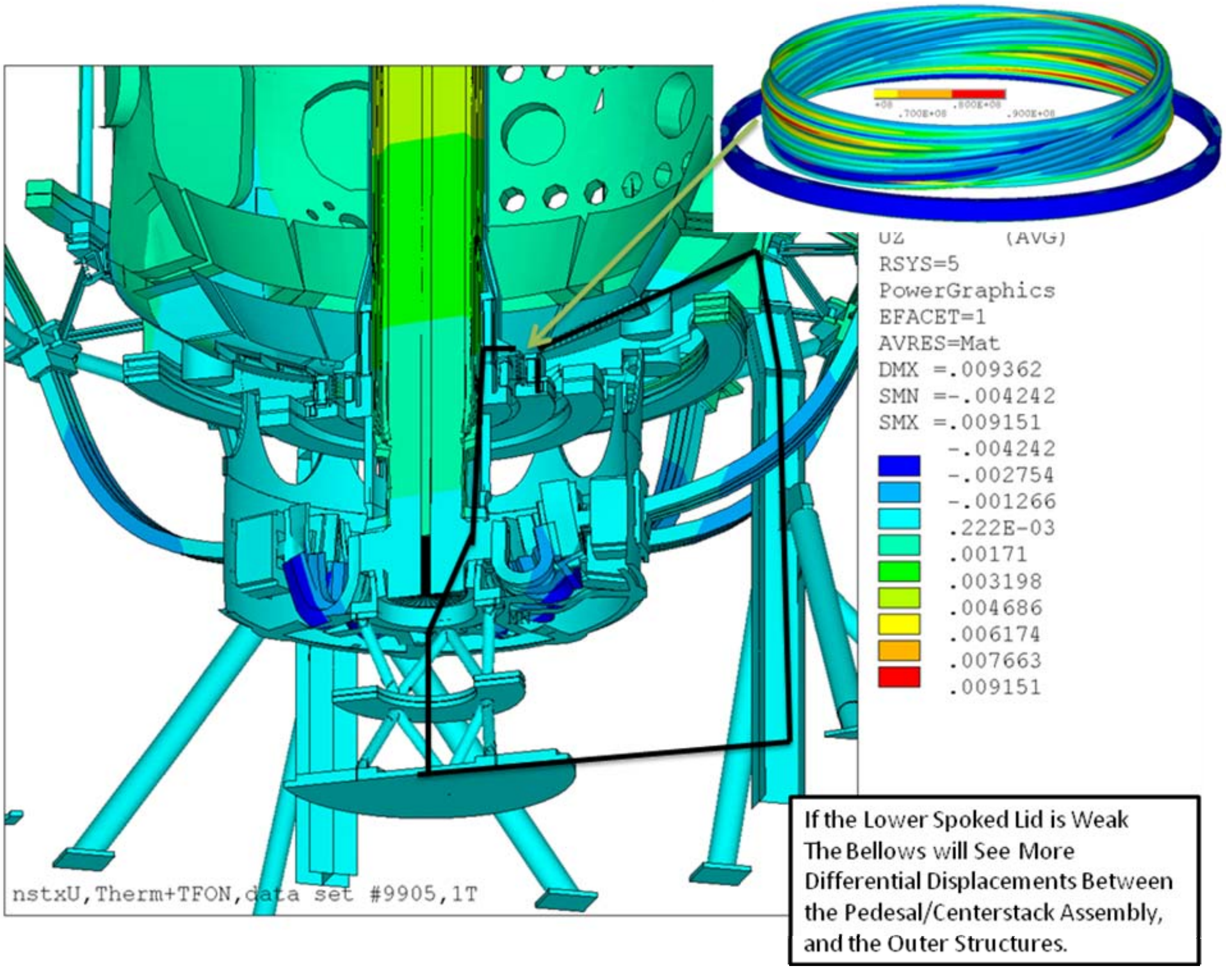


Figure 1.2-6 Structural Effect of a Laterally and Torsionally Compliant Lower Lid on the Bellows

Bent spoke lower lid analysis results were shown at the May 11, 2011, Wednesday project meeting based on models with the Vee pipe pedestal and the bent spoked lid. These two taken together behave differently than the CDR and PDR designs. The load path - exclusive of the spoked lid - for torsional and lateral loads - is shown with a dark line in the figure above. With the stiff pedestal and the softer lid, the bellows connection between the centerstack and the vessel will see more displacements. Jon Menard picked up on this and expressed a concern that this is a vacuum boundary and a problem here might affect the reliability of the machine. The net vacuum side load is included in the global model simulation. In the global model, the torsional stiffnesses are reasonably represented. None of the bellows stresses were troublesome. If the torsional shear is higher, these would have required a revision to Pete Rogoff's bellows calculation as well as Len Myatt's treatment of the ceramic break. The uncertain effect of the halo loading from the passive plates would require a more careful treatment if the lateral load path to the pedestal if the lower lid was compliant.

2.0 DCPS Algorithm

The load used in the analysis was based on the maximum torsional shear load being transferred through the crown to the lid, for all the 96 scenarios. This number is actually 7400 lbs (Ref 1, section 8.19). This was rounded up to 9000 lbs for design to allow for the 10% headroom for PF currents and to allow some headroom for halo current loads. The torsional moment at the TF collar teeth/pins will scale with the calculated torsional shear stress in the TF coil at the turn radius. For the 96 scenarios, this is 24 MPa [4]. Spoked lid stresses should be scaled based on the TF torsional shear stress calculated for the DCPS.

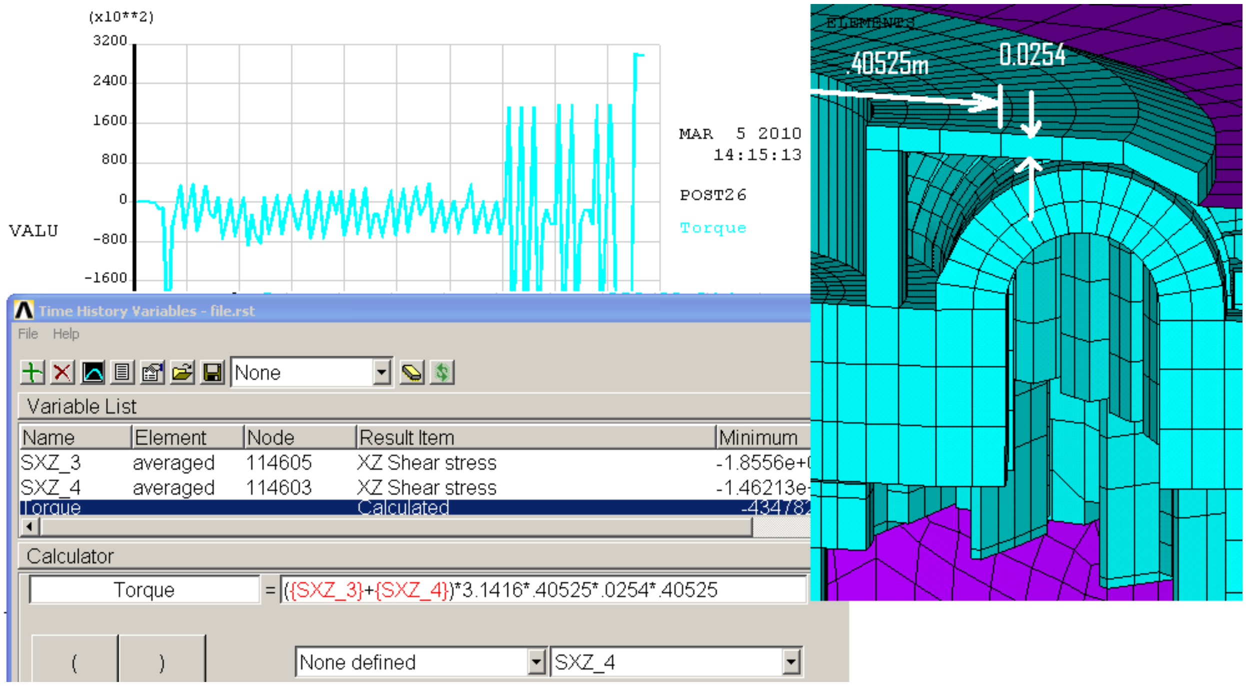
3.0 Design Input

3.1 References

[1] NSTX-CALC-13-001-00 Rev 1 Global Model – Model Description, Mesh Generation, Results, Peter H. Titus December 2010
[2] NSTX Structural Design Criteria Document, I. Zatz
[3] NSTX Design Point June 2010 http://www.pppl.gov/~neumeyer/NSTX_CSU/Design_Point.html
[4] NSTX-CALC-13-04-00 Rev 0 DCPS Inner leg torsional shear Stress, P.H.Titus, R.Woolley
[5] http://www.esk.com/en/products-brands/products/frictional-connection-elements/friction-enhancing-metal-shims.html
[6] Structural Calculation of the TF Flag Key, NSTXU-CALC-132-08-00 , A. Zolfaghari
[7] Center Stack Casing Bellows, NSTXU-CALC-133-10-0 by Peter Rogoff.
[8] NSTX Upgrade DISRUPTION ANALYSIS OF PASSIVE PLATES, VACUUM VESSEL AND COMPONENTS mNSTXU-CALC-12-01-01Rev 1 April , 2011 Peter Titus
[9] 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
[10] May 14 email from M. Smith with recommended design value for the Carbinite high friction coating

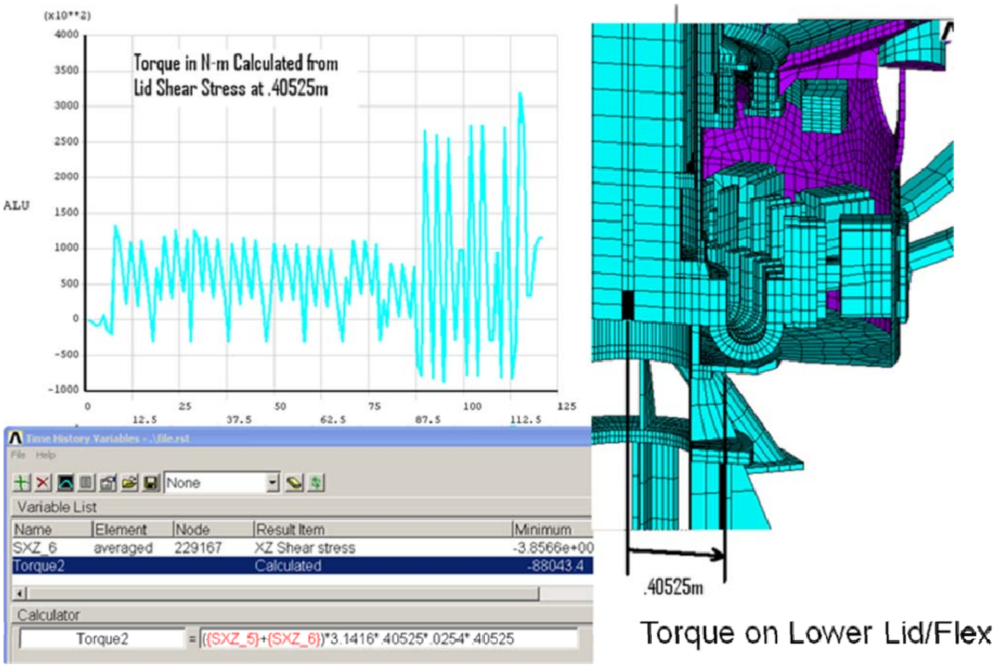
3.2 Torsional shear loading on the bolt circles and the TF steps, pockets or keys

The torsional load from the lid/flex/diaphragm is transmitted to a toothed collar that engages the torsional load from the TF inner leg. In the present design, the TF flags are staggered to engage a G-10 ring that is then bolted to the flex/lid. The keyed connection of the G-10 ring appears to have a larger capacity to carry torque than the bolt circle. Maybe shear keys or pins should be added here as well. To calculate the torsional moment being transmitted across the lid/flex, the torsional shear stress in the solid element portion of the model is post-processed using the ANSYS time history post-processor, Post26. All thermal cases and the 96 scenarios shear stress results are then used to compute the moment within Post26. The moment is then plotted.



Name	Element	Node	Result Item	Minimum
SXZ_3	averaged	114605	XZ Shear stress	-1.8556e+
SXZ_4	averaged	114603	XZ Shear stress	-1.46213e
Torque			Calculated	-43478

Torque = ({SXZ_3}+{SXZ_4})*3.1416*.40525*.0254*.40525



Variable List

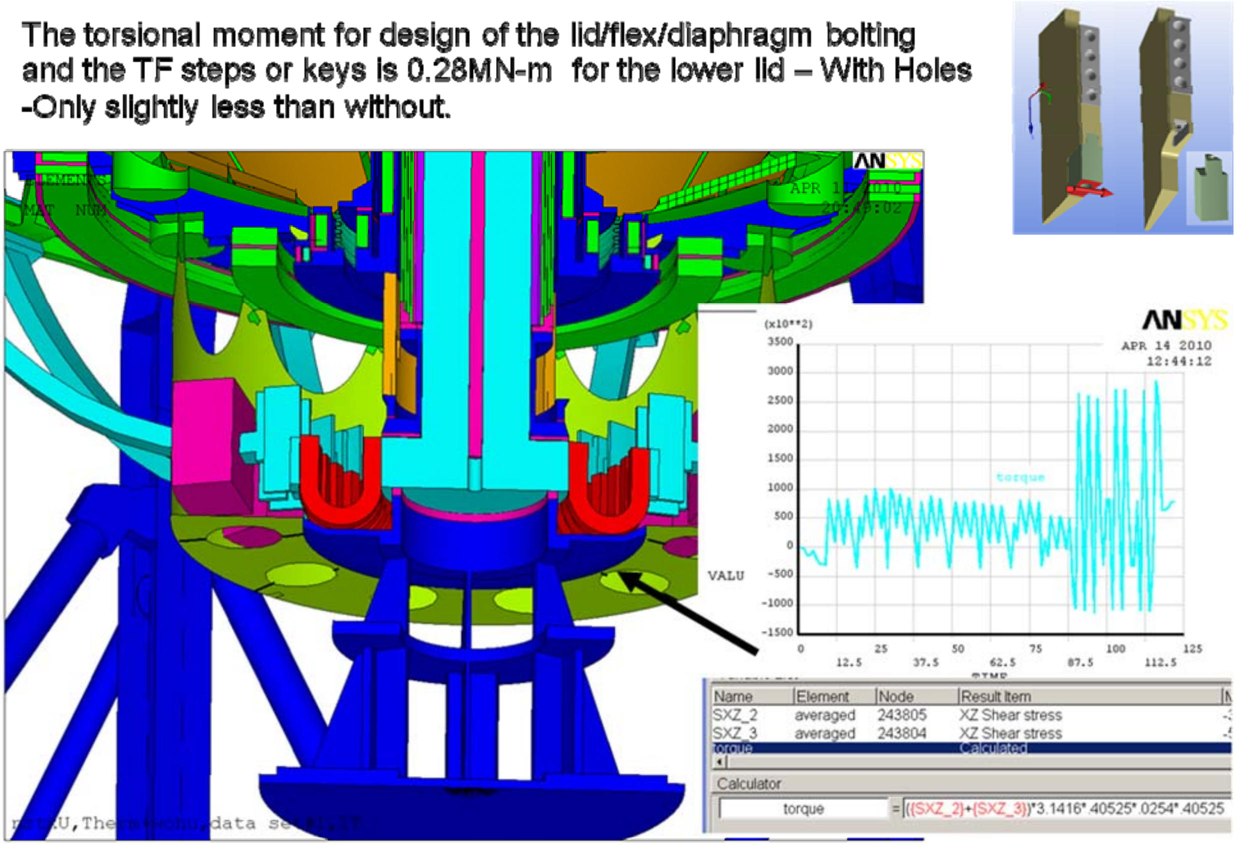
Name	Element	Node	Result Item	Minimum
SXZ_6	averaged	229167	XZ Shear stress	-3.8566e+00
Torque2			Calculated	-88043.4

Calculator

Torque2 = (({SXZ_5}+{SXZ_6})*3.1416*.40525*.0254*.40525

Figure 3.2-1 Torque at the Lower Lid, from ref [1]

The torsional moment for design of the lid/flex/diaphragm bolting and the TF steps or keys is 0.28MN-m for the lower lid – With Holes -Only slightly less than without.



Name	Element	Node	Result Item
SXZ_2	averaged	243805	XZ Shear stress
SXZ_3	averaged	243804	XZ Shear stress
torque			Calculated

Calculator

torque = (({SXZ_2}+{SXZ_3})*3.1416*.40525*.0254*.40525

1

Figure 3.2-2 Torque at the Lower Lid with Circular Access Openings

3.3 Materials and Allowables

Table 3.3-1 Tensile 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 (33.6 ksi)	640 (93 ksi)
304 SST 50% CW	1089 (158 ksi)	1241 (180 ksi)

Table 3.3-2 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	276
316 Weld	161	241
304 Stainless Steel (Bar,annealed)	156MPa(22.6ksi)	234 MPa (33.9ksi)

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 up to 3/4 inch ASTM A193 B8M Class 2 bolt would be the lesser of 125/3 or 2/3*100 =41.66 ksi

Using the 295K curve,

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

Or

20 on life(~400 Mpa)

The Allowable is =300
MPa

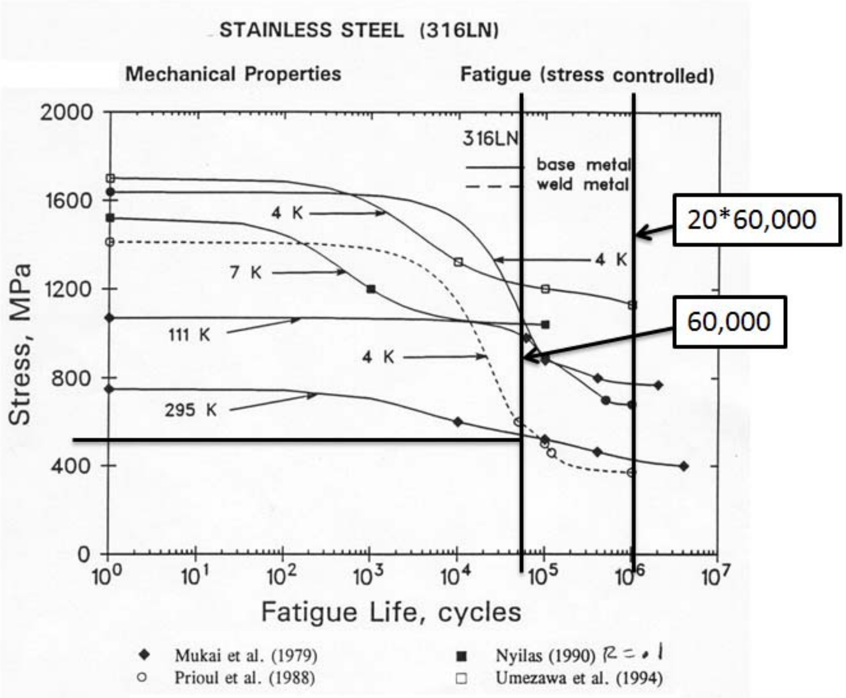
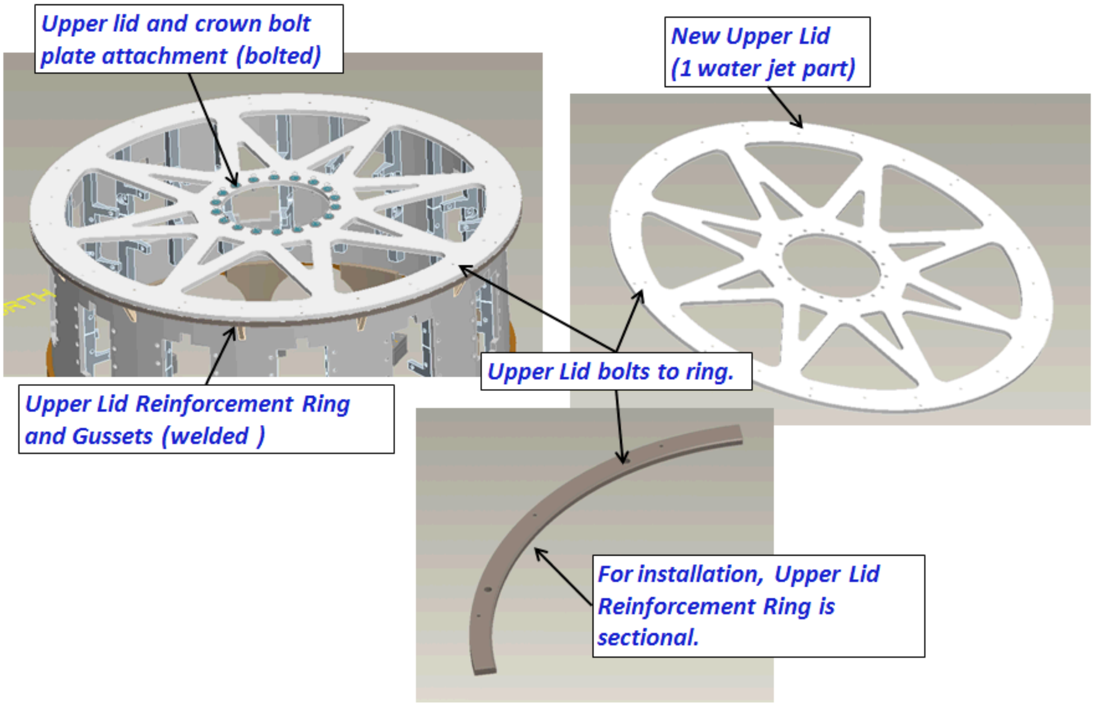


Figure 3.3-1 Fatigue S-N Curve for 316 Stainless Steel

The allowable R=0 stress based on figure 3.3-1 would be for 30000*20 = 600,000 cycles = ~400 MPa or 600/2 = 300 MPa. Use 300 MPa or 43 ksi for the fatigue allowable.

3.4 Drawings, Screenshots of the Designs


Upper lid and crown bolt plate attachment (bolted)

New Upper Lid (1 water jet part)

Upper Lid bolts to ring.

Upper Lid Reinforcement Ring and Gussets (welded)

For installation, Upper Lid Reinforcement Ring is sectional.

Figure 3.4-1 Upper Spoked Lid Details from Mark Smith's May 2011 Peer Review Presentation

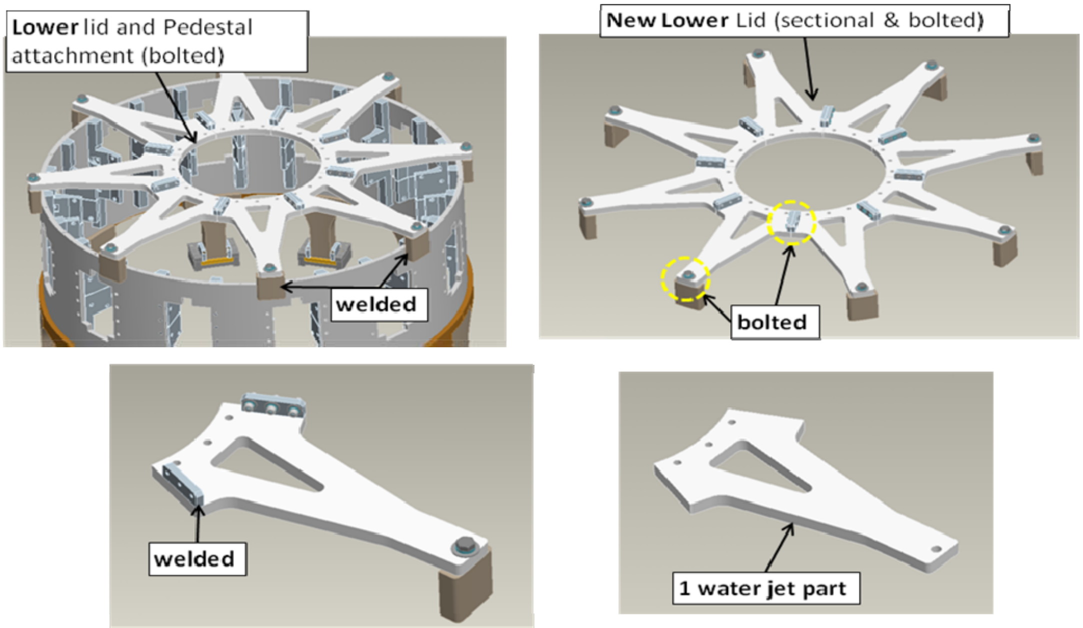


Figure 3.4-2 Lower Spoked Lid Details from Mark Smith's May 2011 Peer Review Presentation

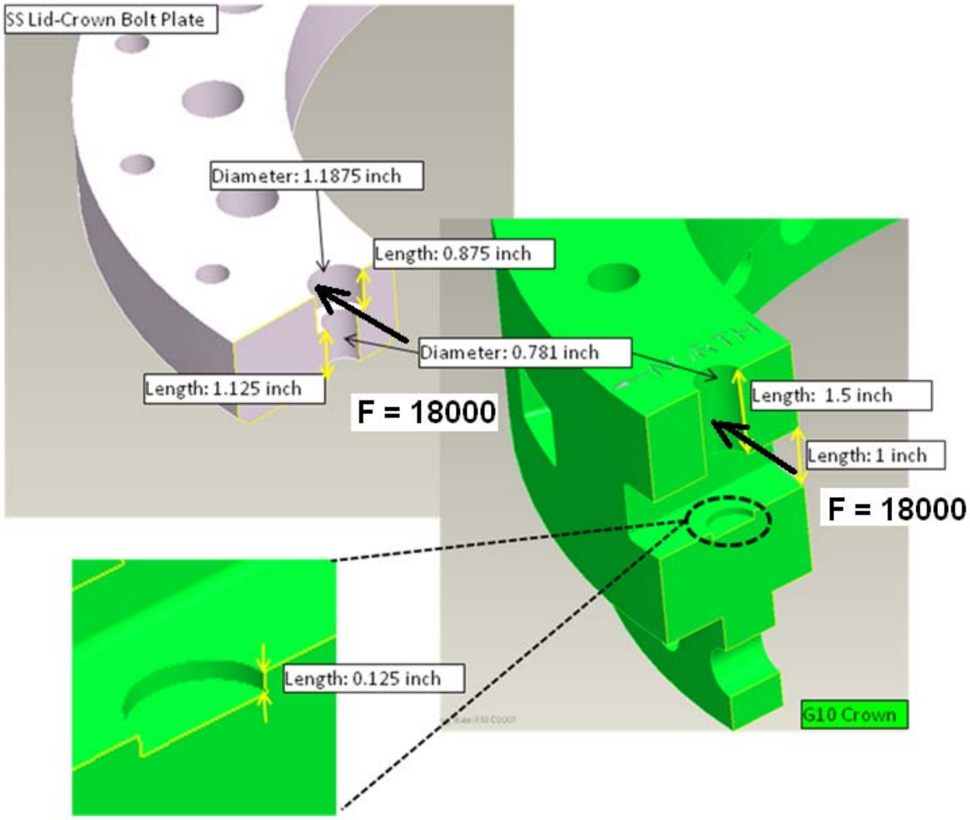


Figure 3.4-3 Early Details of the Spoked Lid to TF Flag Collar

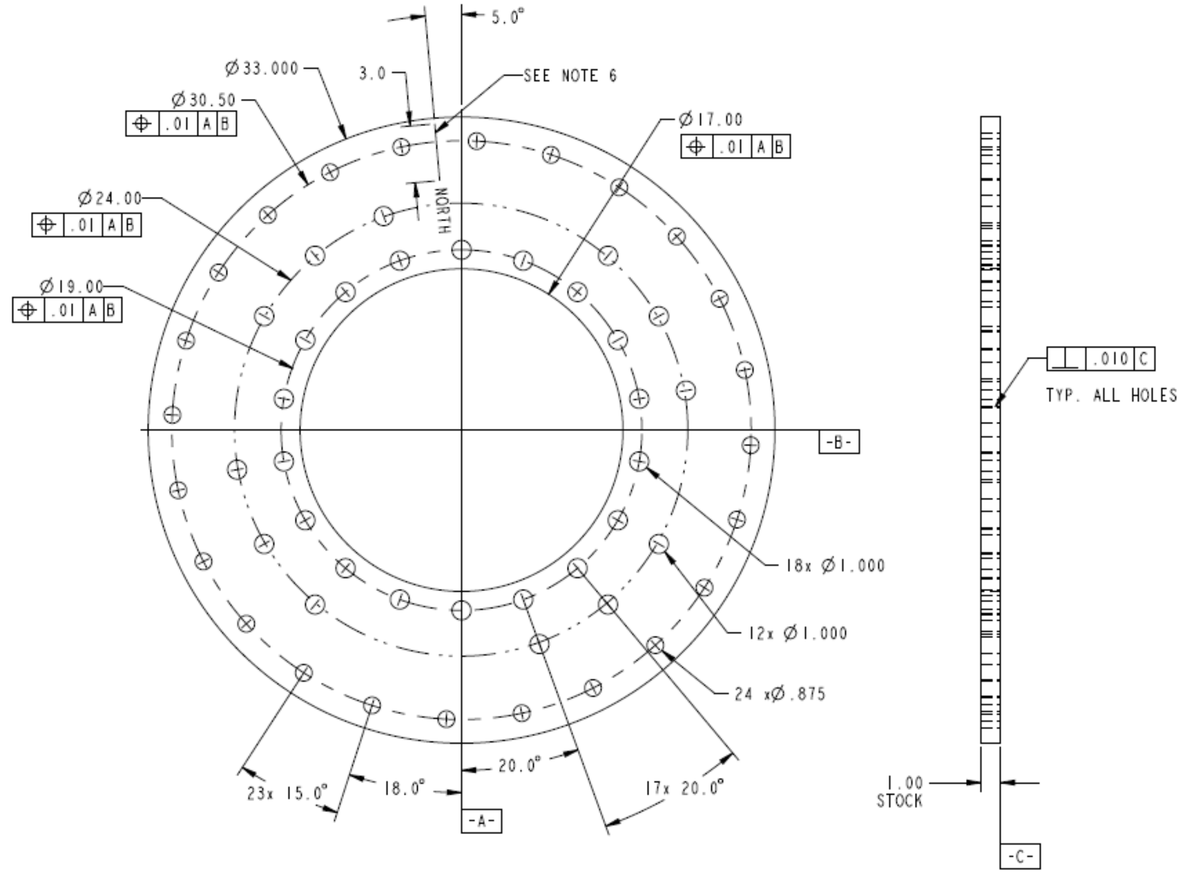


Figure 3.4-4 Pedestal Top Plate (Lower Spoked Lid Bolts to This)

4.0 Analysis Models

Two types of models are used to qualify the spoked lid assemblies. Separate models of the "wheel" are used with torsional loads and centerstack expansion displacements applied. The torques are derived from the global analysis [1]. Initial loads were derived from the global analysis model that had a full thin flex plate. This is shown in Appendix A. As of May 2011, the spoke geometries are modeled consistent with the FDR design of the spoked lids. Shown in this section are plots and descriptions of both the local or separate models and the global model which includes the latest (as of May 2011) geometry.

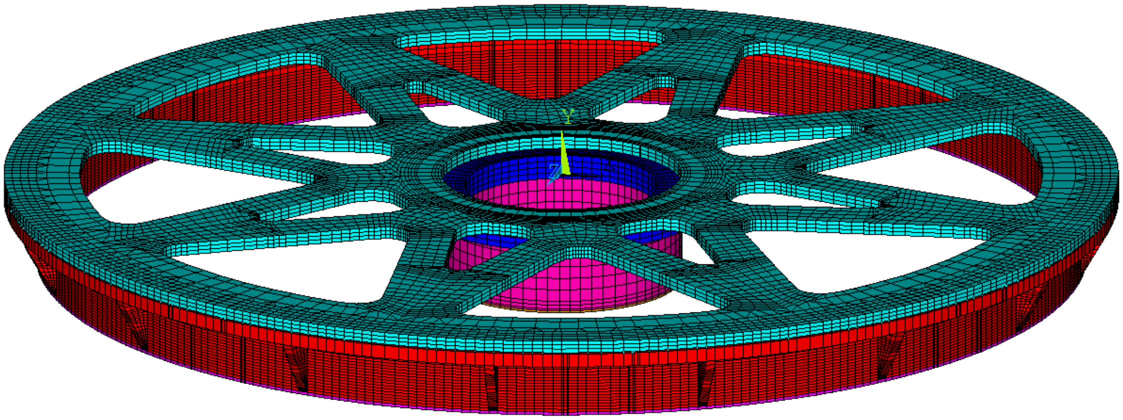


Figure 4.0-1 Symmetry Expansion of the Bolted Outer Flange Design.

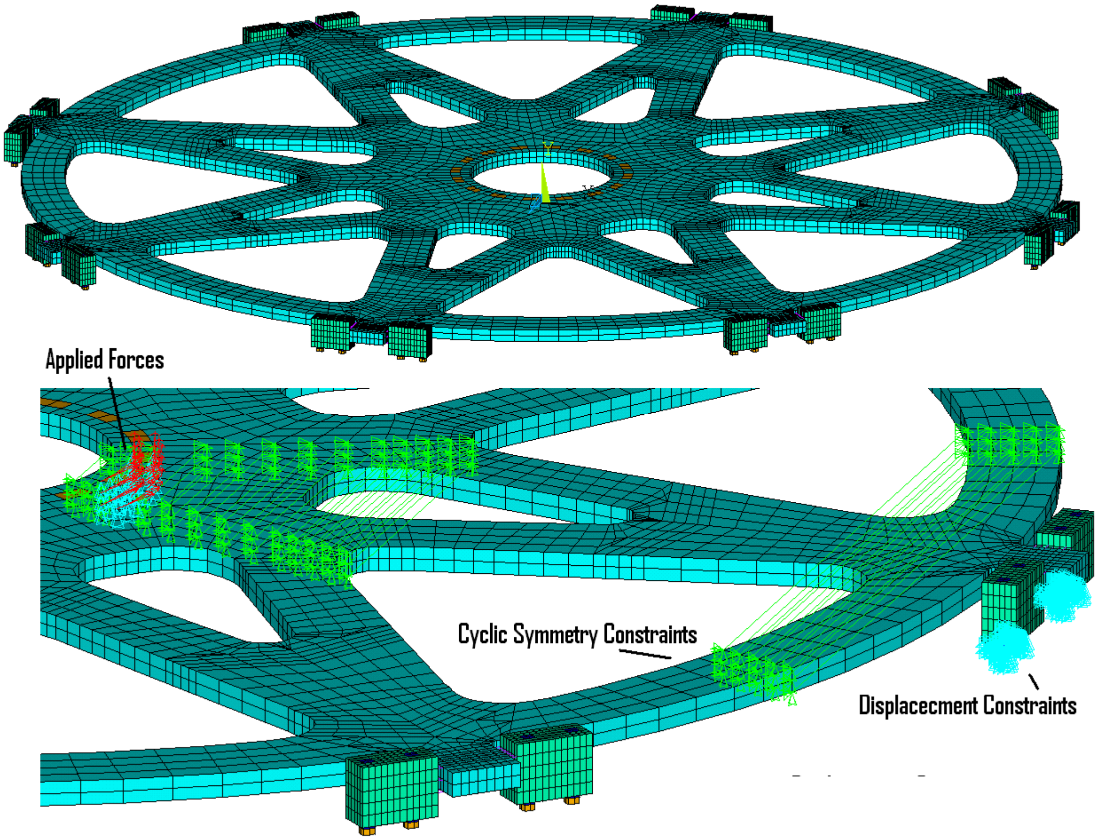


Figure 4.0-2 Local Model of the Upper Spoked Lid - Outer Lug Restraint Design. This is also a symmetry expansion with the symmetry coupling shown.

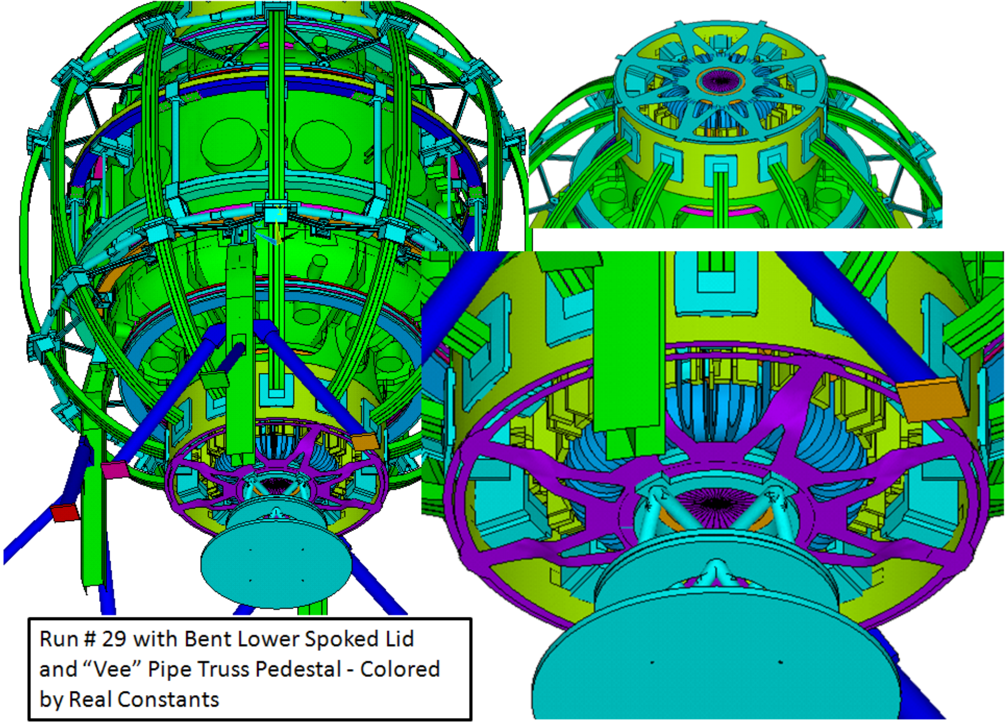

Run # 29 with Bent Lower Spoked Lid and “Vee” Pipe Truss Pedestal - Colored by Real Constants

Figure 4.0-3 Global Model of NSTX showing the Upper and Lower Spoked Lids. In this model, the Lower "Bent" lid is shown.

5.0 Upper Spoked Lid

The spoked lid or flex plate must allow the relative motions of the central column which is fixed vertically at the lower end by connections to the pedestal and to the lower TF flag extensions. The upper connections between the outer rim of the umbrella structure and the TF flags must allow the full vertical expansion of the central column. This is 9 mm at the elevation of the connection. The lid/flex plate is intended to bend and absorb the vertical motions elastically. Bending stresses develop at the ID and OD of the plate which produce prying moments at the bolt circles. Earlier designs had mechanisms or hinges to allow rotation at the ID and OD. These have been replaced with bolted flanged connections.

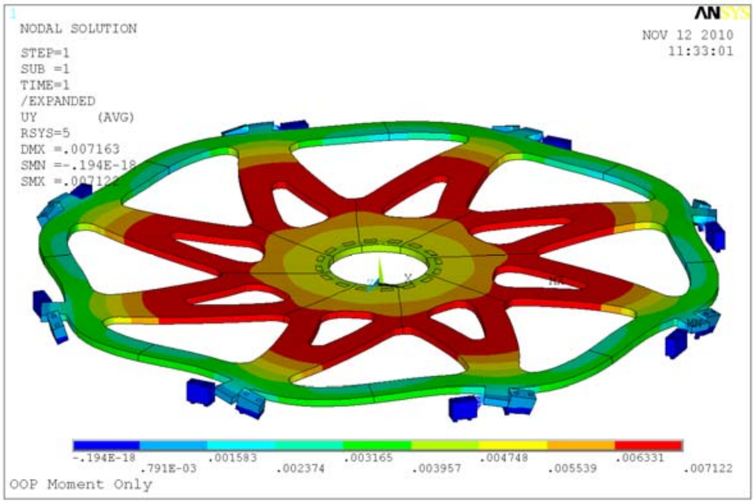


Figure 5.0-1 Earlier Pinned Outer Lug Model

5.1.0 Spoked Lid with Bolted Fixed and Gusseted Outer Flange

This is close to the CDR configuration with bolt circles at the ID and OD of the lid. Vertical growth of the centerstack is accommodated by elastic flexure of the spokes.

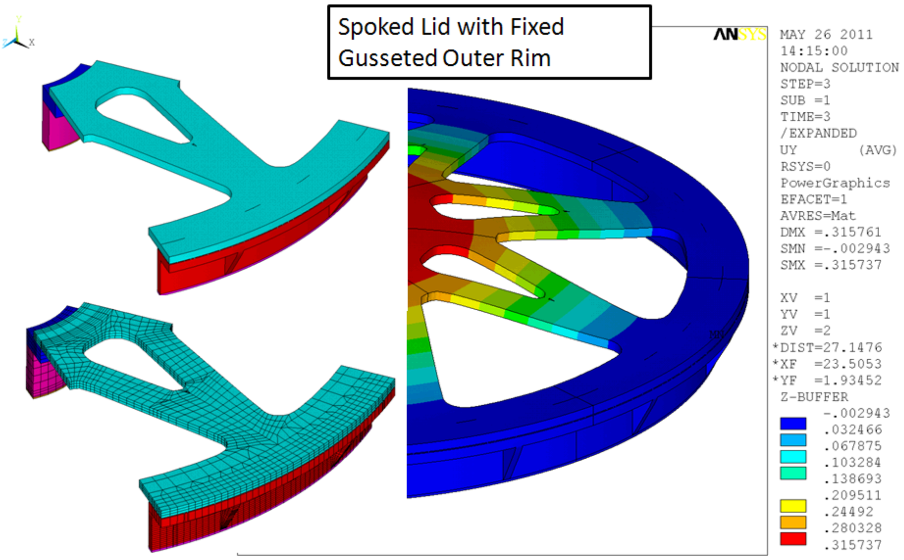


Figure 5.1.0-1 Spoked Lid with Bolted Outer Flange

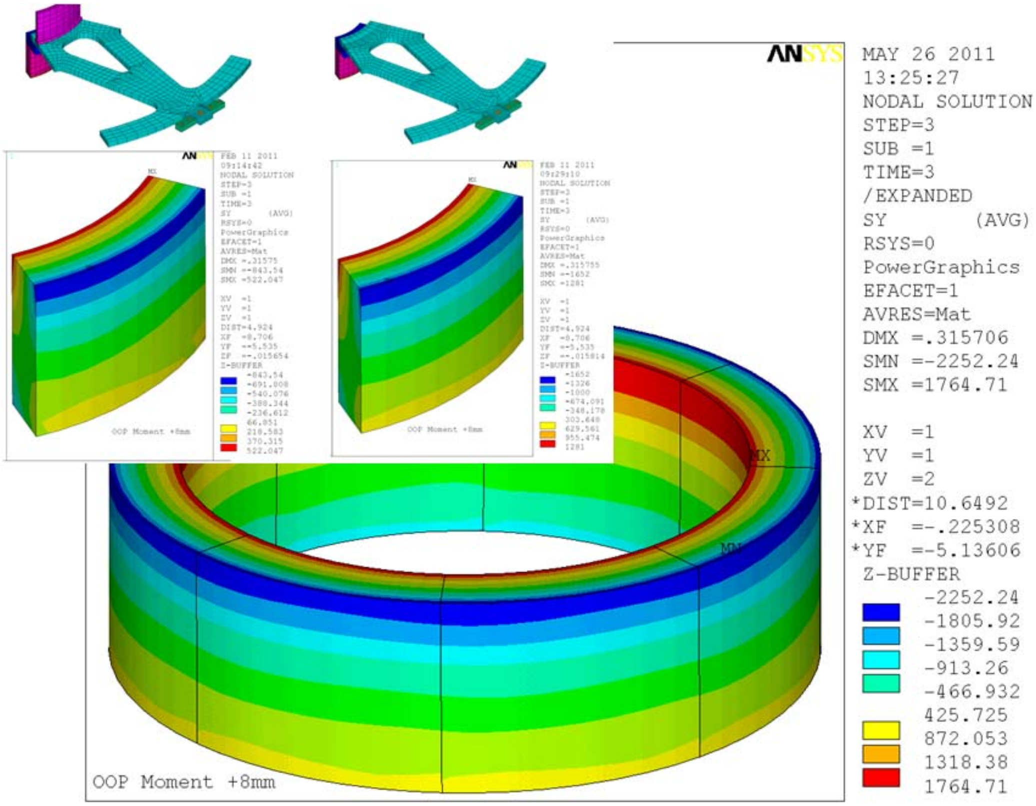


Figure 5.1.0-2 Spoked Lid with Bolted Outer Flange - G-10 Collar Vertical Stress

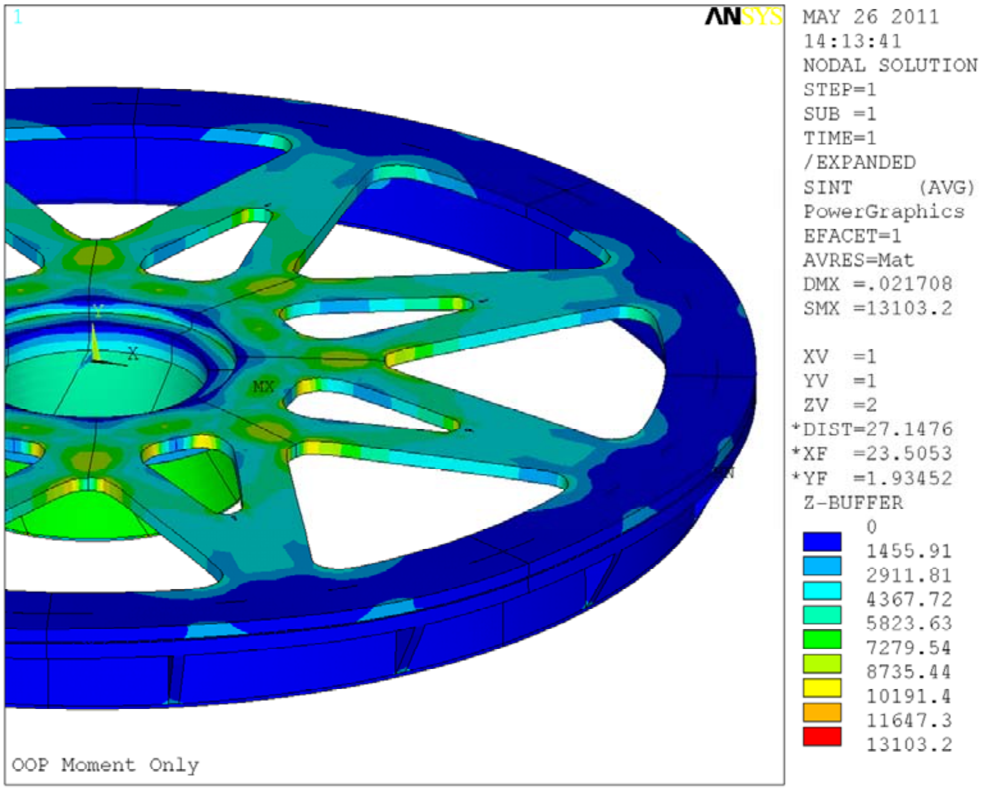


Figure 5.1.0-3 Spoked Lid with Bolted Outer Flange, Stress with OOP Torque, Only

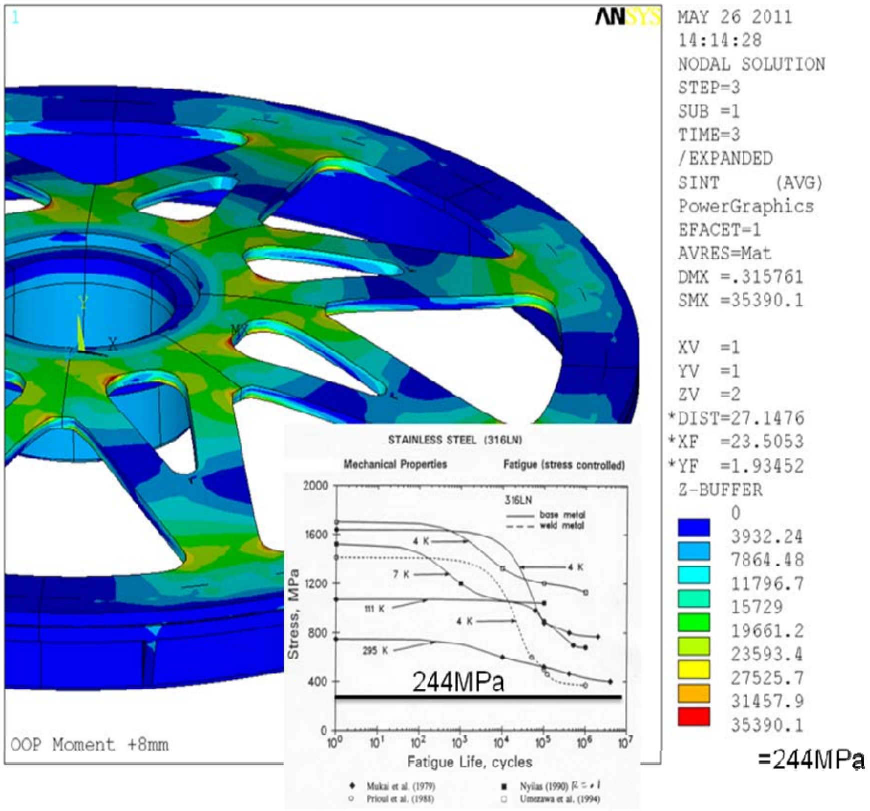
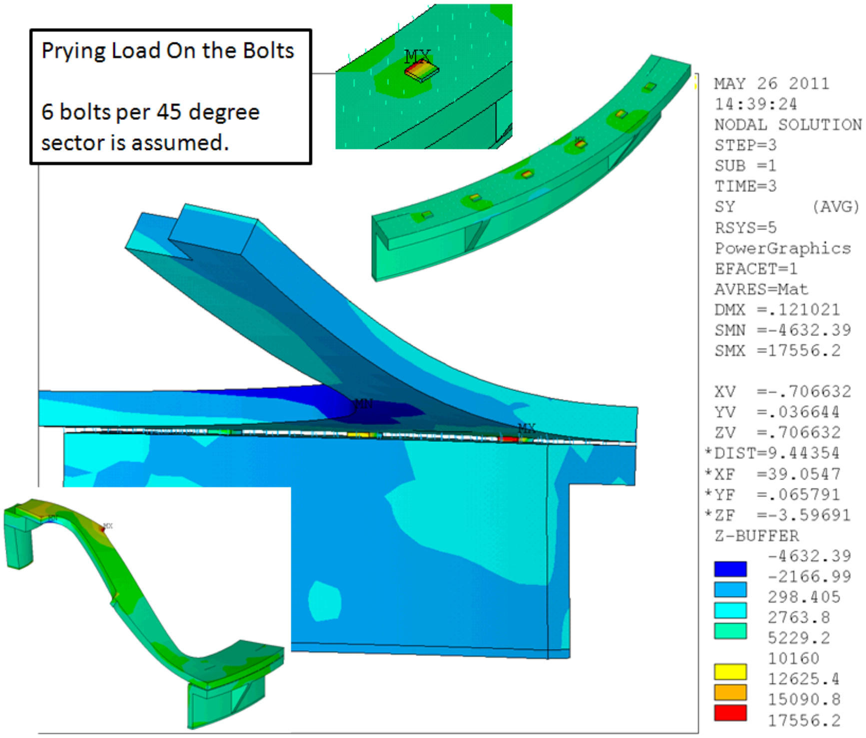


Figure 5.1.0-4 Spoked Lid with Bolted Outer Flange, Stress with OOP Torque, and Centerstack Expansion

5.1.1 Outer Spoked Lid Bolt Circle

Based on Figure 3.4.1, the outer bolt circle has 12 pairs of bolts with another 12 between the pairs for a total of 36 bolts. The finite element model has 6*8 = 48 bolts.

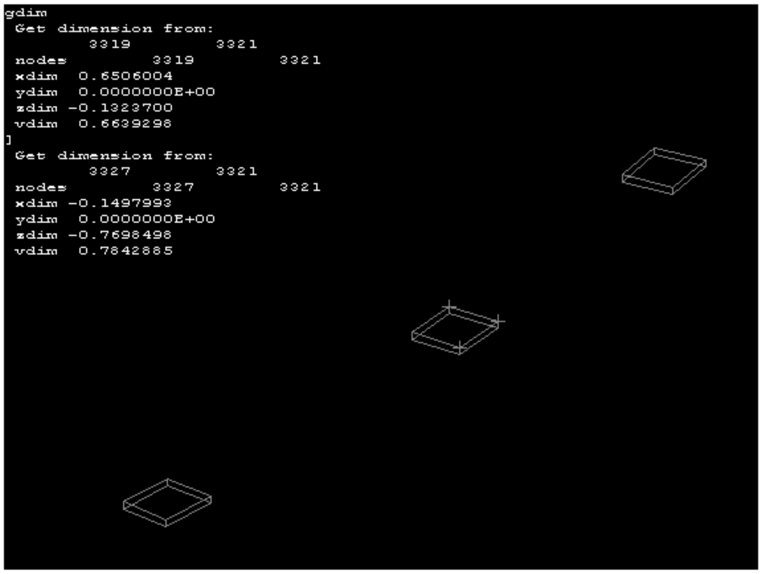

Prying Load On the Bolts

6 bolts per 45 degree sector is assumed.

Figure 5.1.1-1 Spoked Lid with Bolted Outer Flange - Bolt Prying Load

Prying Load On the Bolts

The pad is .66in X .78 in and is stressed to 10 ksi for a 5/8 bolt,
The stress would be 10000*.66*.78/.2256 = 22819 psi



```
gdim
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 nodes            3319            3321
 xdim   0.6506004
 ydim   0.0000000E+00
 zdim  -0.1323700
 vdim   0.6639298
]
 Get dimension from:
          3327            3321
 nodes            3327            3321
 xdim  -0.1497993
 ydim   0.0000000E+00
 zdim  -0.7698498
 vdim   0.7842885
```

Figure 5.1.1-2 Spoked Lid with Bolted Outer Flange

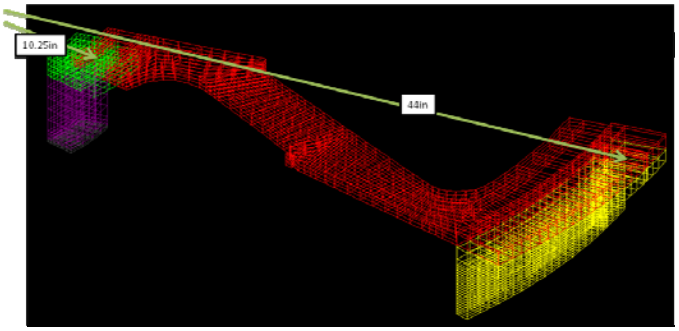
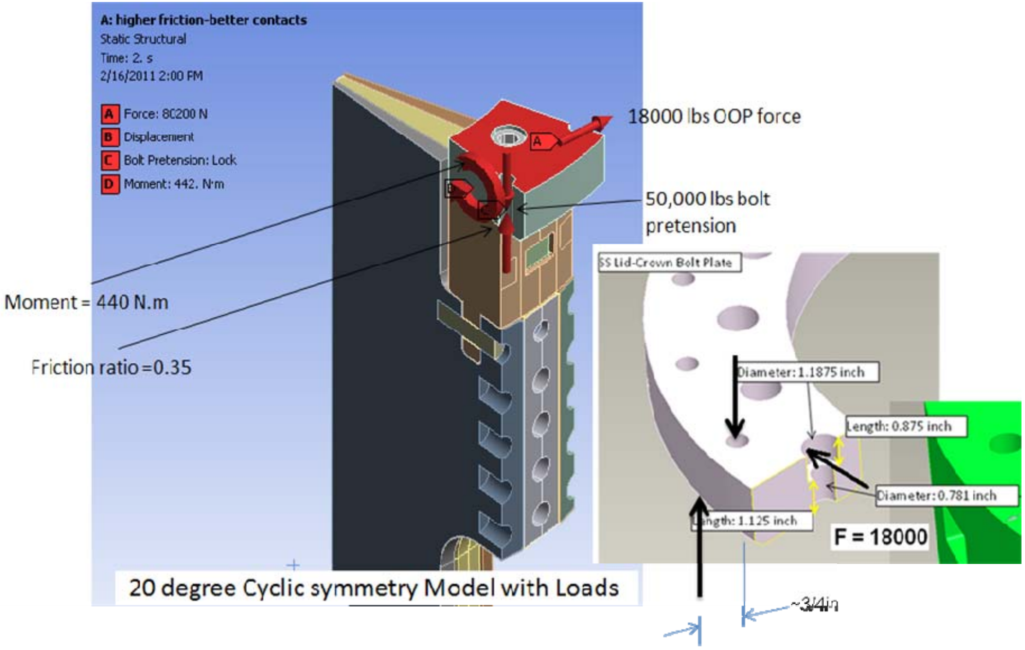


Figure 5.1.1-3 Spoked Lid with Bolted Outer Flange

The bolt stress needs to be scaled from the 48 modeled in the finite element model to the 36 bolts shown in Figure 3.4.1. This yields a bolt stress = 30425 psi. This is another application for high strength bolts. ASTM A-193 B8M Class 2 are recommended. The allowable for up to 3/4 inch ASTM A193 B8M Class 2 would be the lesser of 125/3 or 2/3*100 =41.66 ksi. The torque load at the outer bolt circle is 18000*18*10.25/36/44 = 2096 lbs, so the torque loading is not nearly as challenging for the outer bolt circle as for the inner bolt circles. The frictional capacity needed from the outer bolt circle bolts will be met by a normal preload (typically 70% yield).

5.1.2 Upper Inner Spoked Lid Bolt Circle

There are 18 bolts at the inner bolt circle that connect the spoked lid to the inner TF collar assembly. The 18 bolts are shown in Figure 3.4.1. The prying load below is from reference [6]. Bolt design for the crown bolting ring is challenging, and is addressed in reference [6]. Prying loading on the spoked lid bolts and the preload needed to support the global torque are very similar for the two bolt circles at the ring. These will require the same solutions, including the augmented friction coefficient and bolt sizing. In the figure below, a friction factor of .5 was assumed based on surface preparations planned for the crown to ring interface. The friction factor was de-rated in accordance with the structural design criteria.



Additional Prying Force=440*0.2248*39.37/0.75 = 5192 Lbs
Needed Preload to Support the Global Torque Load =18000/(.50-.15) = 51428 lbs
Total Bolt Load = 51428+5192 = 56620 lbs

Figure 5.1.2-1 Spoked Lid Inner Bolt Circle Bolting

5.2 Frictional Effects of Sliding Umbrella Restraint Blocks

A concept that employed restraint lugs at the umbrella outer rim was investigated. This concept was intended to allow vertical motion via slippage at the lugs. Bolted cleats that appear in the aluminum block CAD models were added at the lugs. The cleat bolts are 1/2 inch bolts spaced two inches apart. The 9000 lb design load per turn (times 36 teeth/8 cyclic symmetry sectors) was added at the inner hub. The lid spoke design is within allowable stresses. The cleat bolts are OK at 45 ksi if high strength bolts are used.

If the lugs at the outer perimeter are frictionally restrained, one potential issue is the stick-slip at the cleat-lug detail. With a friction coefficient of 0.1, the OOP torque load is sufficient to stick the lug/cleat. As the TF heats and expands, the lug/cleat remains stuck, bending the spokes and putting a moment on the inner TF collar. At 50% OOP load, the cleat/lug remains stuck. At 25% of the load, it slips and springs upward the 8mm (or more, dynamically) that the TF has expanded. This study led to the conclusion that the outer lugs need to be pinned or bolted.

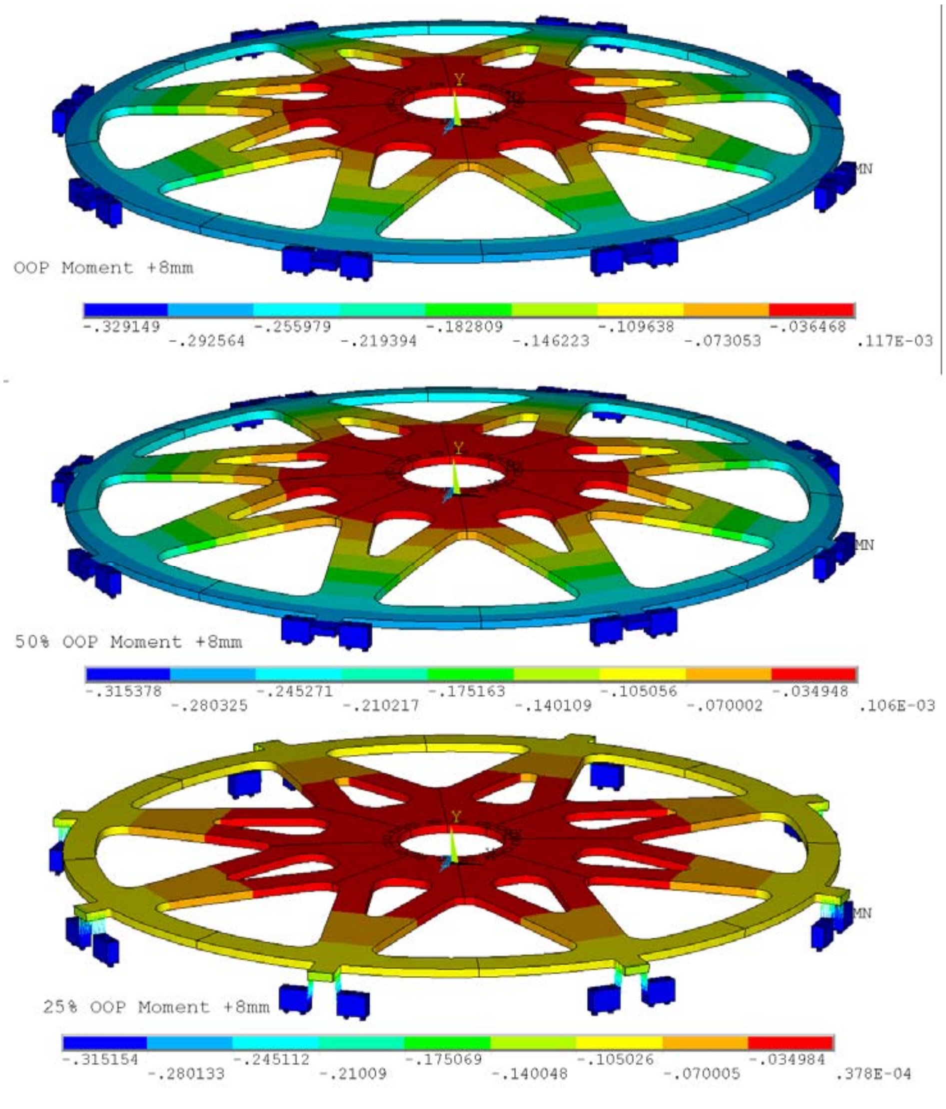


Figure 5.2-1 When the OOP force reaches 25% of nominal, the spoke lid springs upward

5.3 Upper Spoked Lid Pinned Umbrella Restraint Blocks

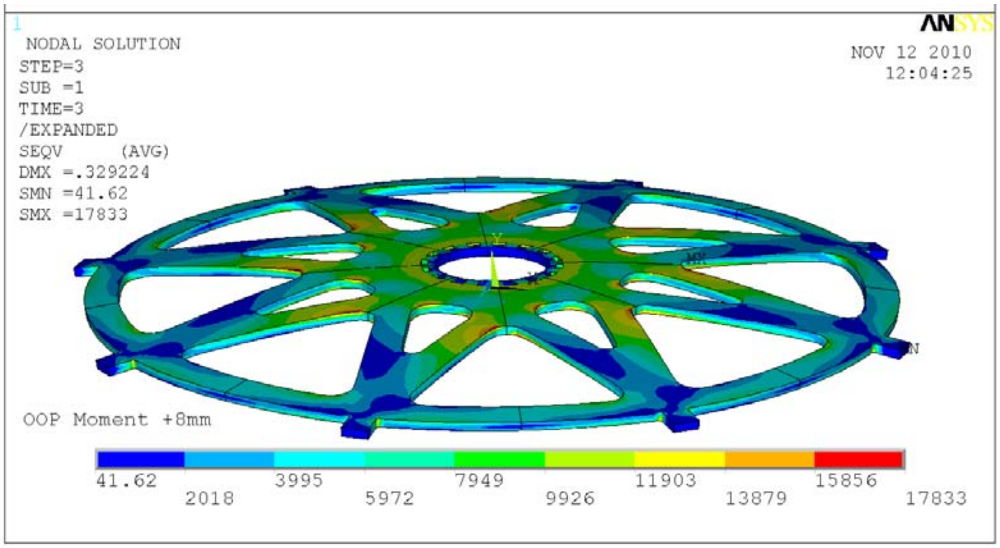


Figure 5.3-1 Pinned Lug Concept, Stress with OOP Moment, and Centerstack Growth

A concept that pinned the restraint lugs at the umbrella outer rim was investigated. This allowed vertical motion of the centerstack without imposing a prying moment on the outer connection hardware. It also minimizes the prying moment at the ID. The local pin hardware was not modeled. The "hinge" behavior was modeled by connecting a single node at each side of the lug - to the umbrella structure rim.

5.4 Upper Spoked Lid Bolted - Fixed Umbrella Restraint Blocks

In this concept, only the lugs were bolted to the umbrella rim. This concentrated all the moment and rotations at the lug and over stressed it. Spreading the moments around the full perimeter of the umbrella structure rim, discussed in section 5.1, eased the stresses where the lug detail had been, but imposed more flexure in the spokes, which translated to higher spoke stresses than in the pinned lug design.

5.5 Bending Moments at the ID TF Collar - Possible Mitigation

The inner radius of the upper spoked lid is attached to the G-10 Collar that is pinned to the TF flags. As the TF central column heats during a shot, it displaces 8 mm vertically, and the lid must flex or translate upward to absorb this displacement. Pinned or frictional restraint at the outer diameter connection to the umbrella structure produces a bending moment on the spokes that is reacted at the inner hub of the lid/collar assembly. The bending moment or rotation must be accommodated by the G-10 collar. In this study, the collar is included in the model. The vertical stress distribution is computed at the top of the collar. This stress distribution is then matched in Ali Zolfaghari's TF flag analysis, reference [6]. As a part of modeling the collar, an additional hub was considered that would have reduced the moment. The stresses in the collar have been found acceptable without the added hub [6].

The early estimate of the prying moment at the bolt circles was 6300 N-m per meter of perimeter. The prying moment can probably be reduced by reducing the assumed thickness of the 5/8 inch thick lid.
A flex plate or cover or “lid” is intended as the structure that extends from a connection to the TF central column flags to the outboard edge of the umbrella structure. These details are only concepts in the drawings currently, but a simple representation of the plate is included in the global model [1].

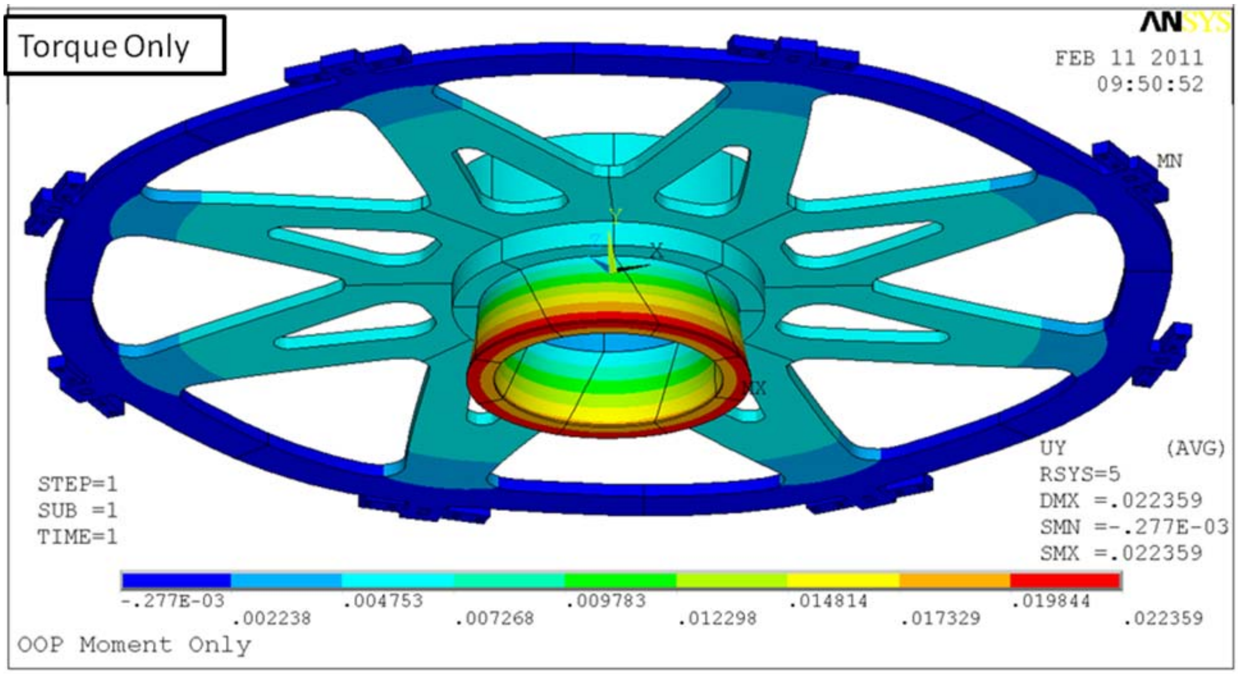


Figure 5.5-1 Hub Concept Torsional Displacements. The added hub is above the G-10 Collar that is Twisting

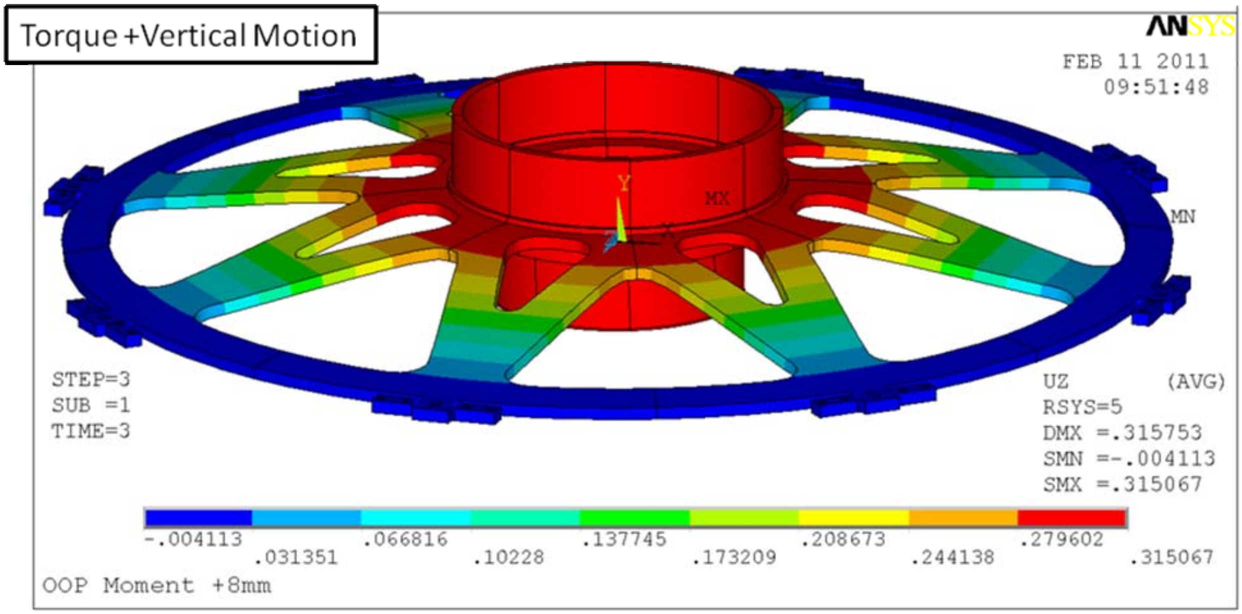


Figure 5.5-2 Hub Concept Vertical Displacements

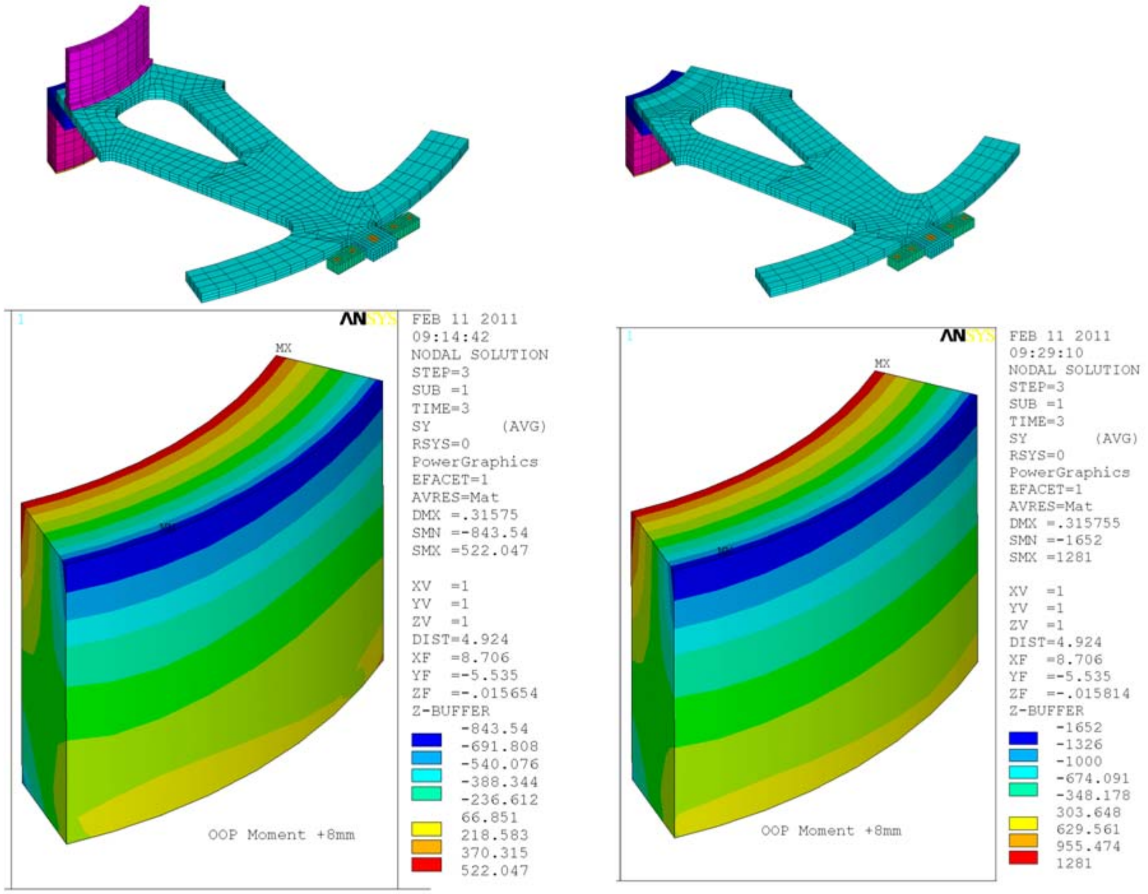
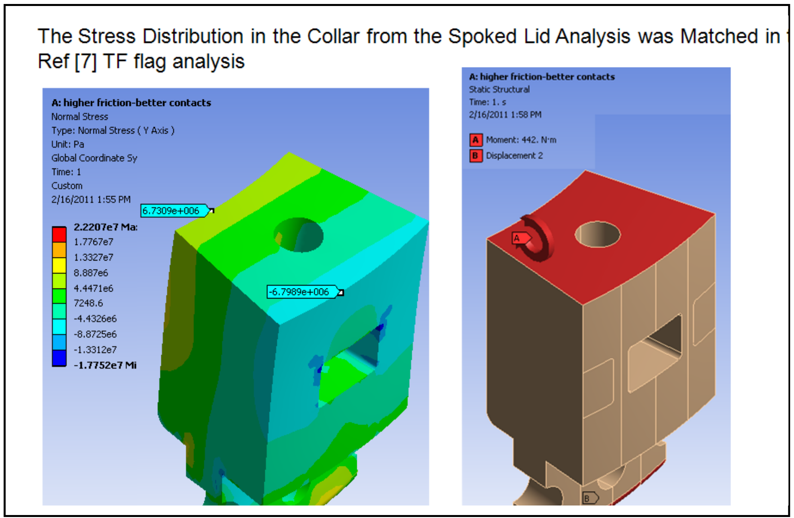


Figure 5.5-3 Vertical Stress distributions in the inner G-10 collar, with (left) and without the hub

In figure 5.5-3, the connection to the inner crown was modeled with and without a metal hub extension. The vertical displacement of the TF central column was imposed. The resulting vertical stress distribution was matched in the collar analysis [6]. A distributed moment was applied in the ANSYS WORKBENCH model which was then scaled to reproduce the vertical stress. The hub improved the prying stress in the G-10 collar, but Ali Zolfaghari qualified the collar stresses with the higher prying moment, and the metal hub has been omitted from the current design.

Loads and Moments on the TF Inner Leg Flag Collar

The vertical stress distribution in the collar was passed to Ali Zolfaghari. He first applied a unit moment and then scaled the moment to match the vertical stress distribution. This was updated at the end of May, 2011 consistent with the vertical stress distribution shown in Figure 5.1.0-4 for the bolted gusseted outer rim.


The Stress Distribution in the Collar from the Spoked Lid Analysis was Matched in
Ref [7] TF flag analysis

6.0 Lower Spoked Lid

The lower spoked lid serves a similar purpose as the upper spoked lid - bridging between the outer lower rim of the lower umbrella structure with the inner lower TF hub and pedestal. It is part of a complex multiply redundant load path for the OOP torque. The TF flex strap needed more territory and the spokes needed to be bent or offset to clear the straps. The first design employed the bend. The FDR design employs a flat, not bent, geometry. The bend introduced a torsional compliance issue that needed to be resolved. Analyzing and explaining this effect, and its impact on the total machine torque load paths, is important in understanding the design evolution.

6.1 Lower Spoked Lid Final Title III Bolted Plate Design

6.1.1 Hub Frictional Simulation

In the FDR design, the bolted connection between spoke segments had not been finally detailed. In April of 2013 the design shown in Figure 6.1-1 was chosen. The design is segmented and relies on bolted friction joints.

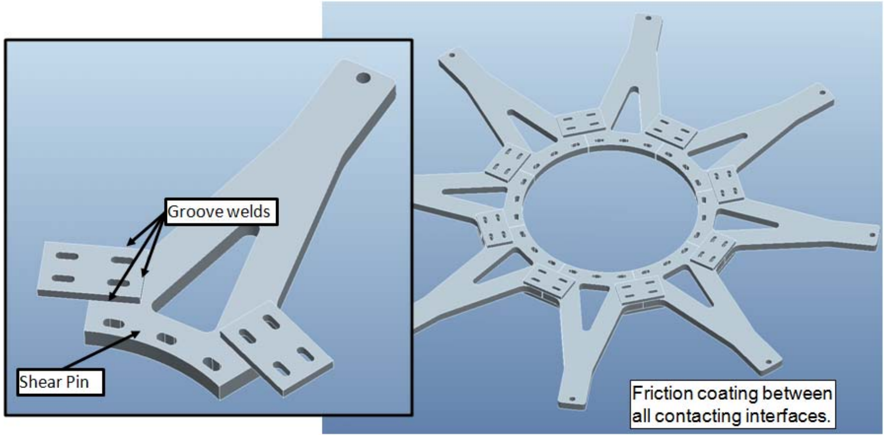


Figure 6.1.1-1 Bolted Plate Hub Design

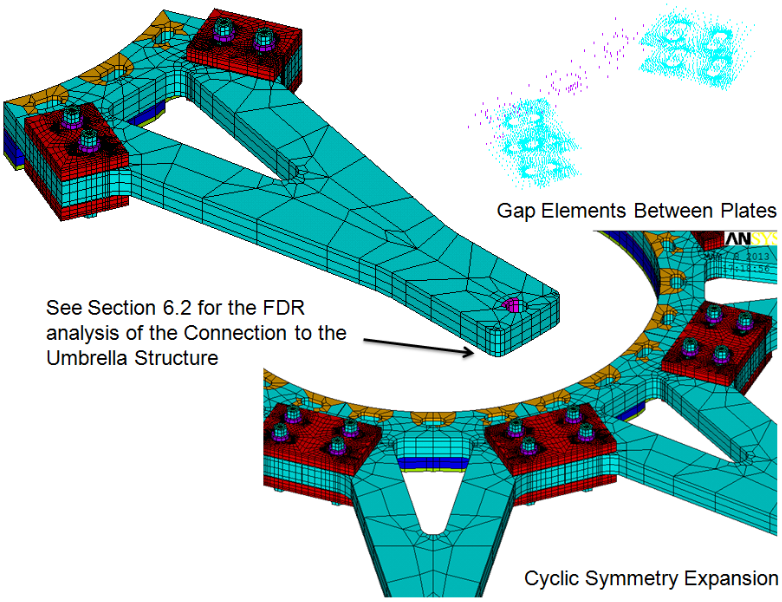
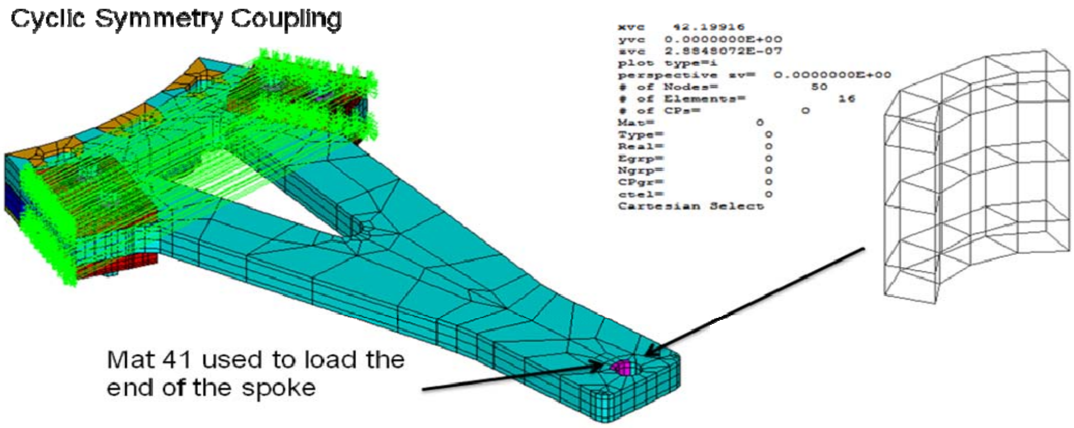


Figure 6.1.1-2 Model with bolted plates at the hub



```
xvc   42.19916
yvc  0.0000000E+00
zvc  2.8848072E-07
plot type=i
perspective zv=  0.0000000E+00
# of Nodes=              50
# of Elements=              16
# of CPs=            0
Mat=           0
Type=           0
Real=           0
Egrp=           0
Ngrp=           0
CPgr=           0
ctel=           0
Cartesian Select
```

Figure 6.1.1-3 Model with Bolted plates at the Hub, Cyclic Symmetry Constraints and Spoke Loading

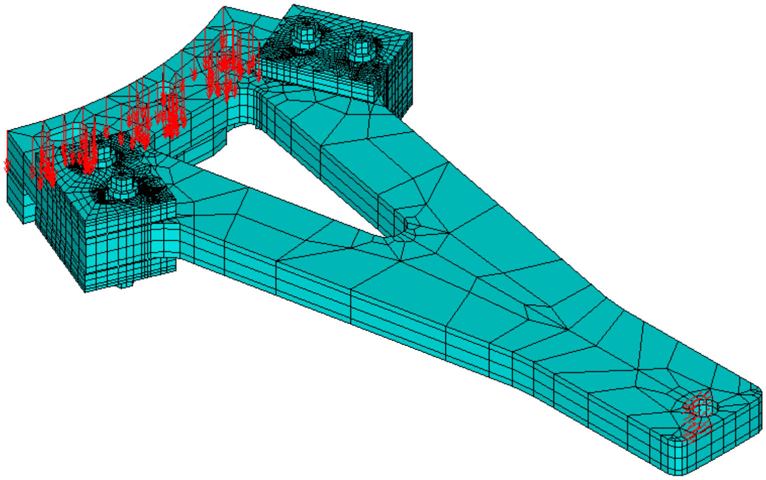
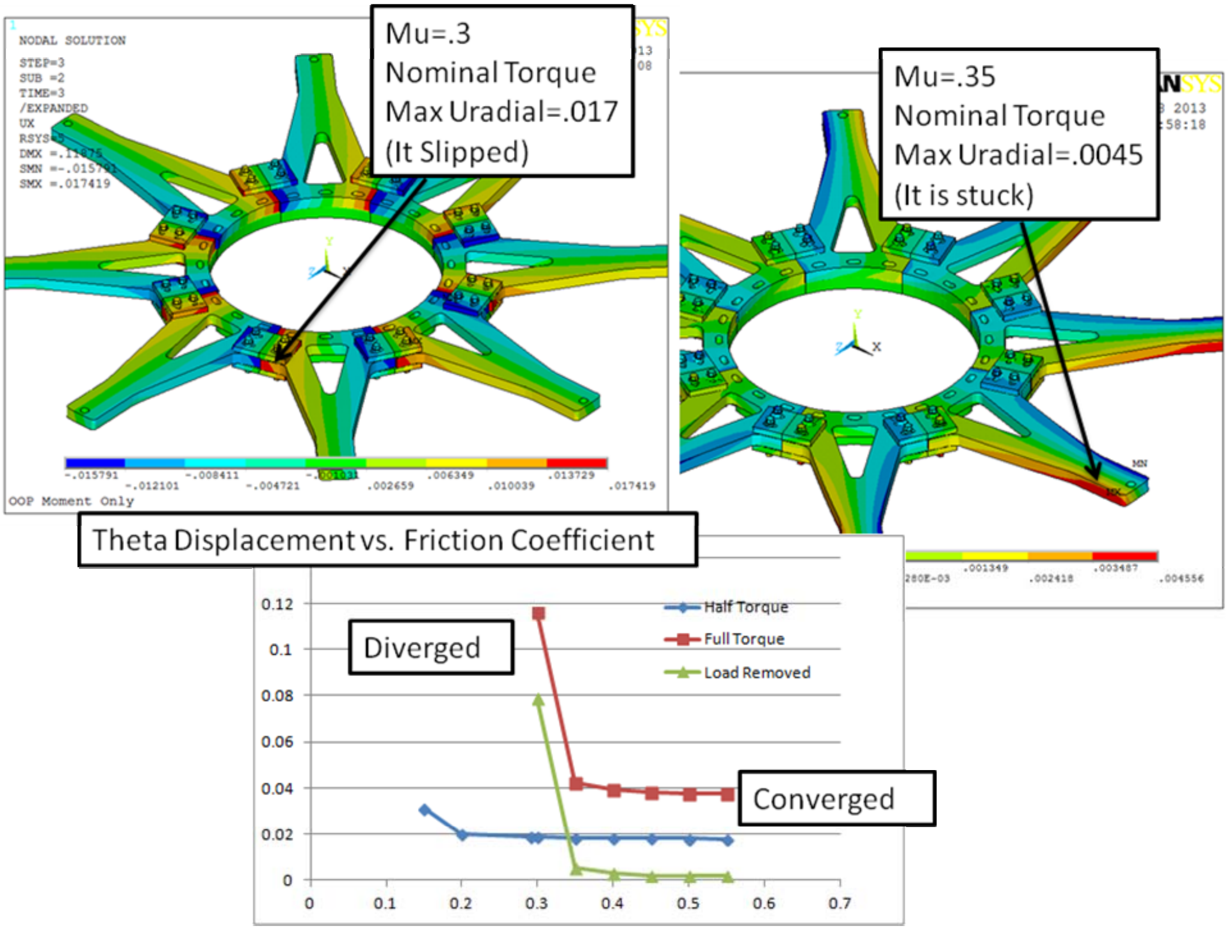


Figure 6.1.1-4 Inner Bolt Circle Preload is Applied with Forces, and the Load at the Outer Radius of the Spoke is applied at Nodes in the Bolt Hole

From the Global Model Analysis [1]
“Collar Tooth Load: = 0.3 MN-m/(10/39.37)*.2248/36 = 7375.3 lbs. This has been rounded up to 9000 lbs to provide some headroom for the halo current loads, and the 10% headroom on PF currents applied in the design point spreadsheet.”

```
FTorque=9000*36/8*10.918/14.65   !This force is applied at a spoke
f,all,fy,ftorque*14/42.62/50/2  ! 50 nodes in Mat 41
```

Later modeling of the plates have the edge welds on every other side modeled. Credit for the welds was required to address the halo loads which are considered in section 8.1. To begin with, only frictional restraint is modeled. The 5/8-inch bolts are tensioned around 60 ksi and the one inch bolts at the inner bolt circle are tensioned to 54 kips each. Friction is varied in this study. The concept works for the machine torque when the friction coefficient is above .35. At this level, it still has a bit of motion and frictional slip. With Halo loads applied (see section 8.0), the bolts had to be tensioned to 100 ksi, the splice plate welds had to be increased, and high friction Carbonite material with a friction coefficient of .6 was needed.



Mu=.3
Nominal Torque
Max Uradial=.017
(It Slipped)

Mu=.35
Nominal Torque
Max Uradial=.0045
(It is stuck)

Theta Displacement vs. Friction Coefficient

Diverged

Converged

Figure 6.1.1-5, Effect of Friction Coefficient on Dimensional Stability of the Hub Bolting

In the figure above, the contour plot on the left is of a case where everything slipped and it took a permanent deformed/slipped geometry when it was unloaded. There is a sharp transition between the diverged slipped results and ones in which the mechanically assembled hub doesn't slip. It is clear that a minimum friction factor of .35 is needed. The criteria document requires a margin of .15 on the friction factor, so a friction coating that can achieve a factor of .5 is needed. In section 8.1, a lateral load representing the halo disruption load was added. This led to increases in bolt preload and modeling of the plate welds.

With the addition of halo loads, the bolt preload values, friction coefficients and welds were all increased to provide adequate margin against slippage. In figures 6.1.1 – 6 and 7, the results with the higher preloads etc., are shown for only the torque loading.

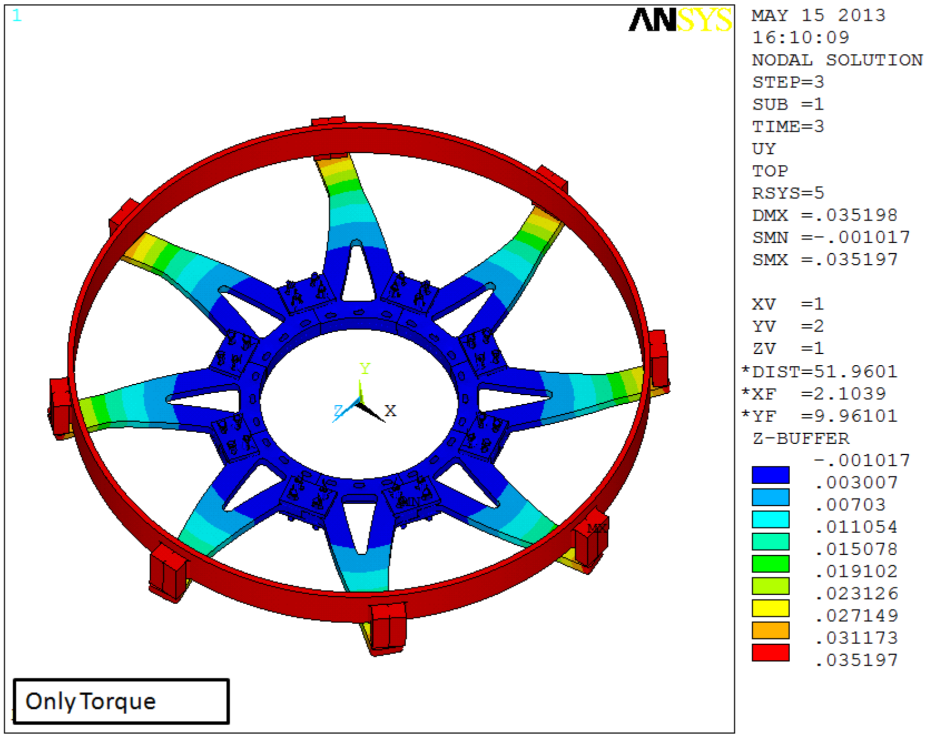


Figure 6.1.1-6, Theta Displacement of the Lower Spoked Lid Assembly For Machine Torque, Only

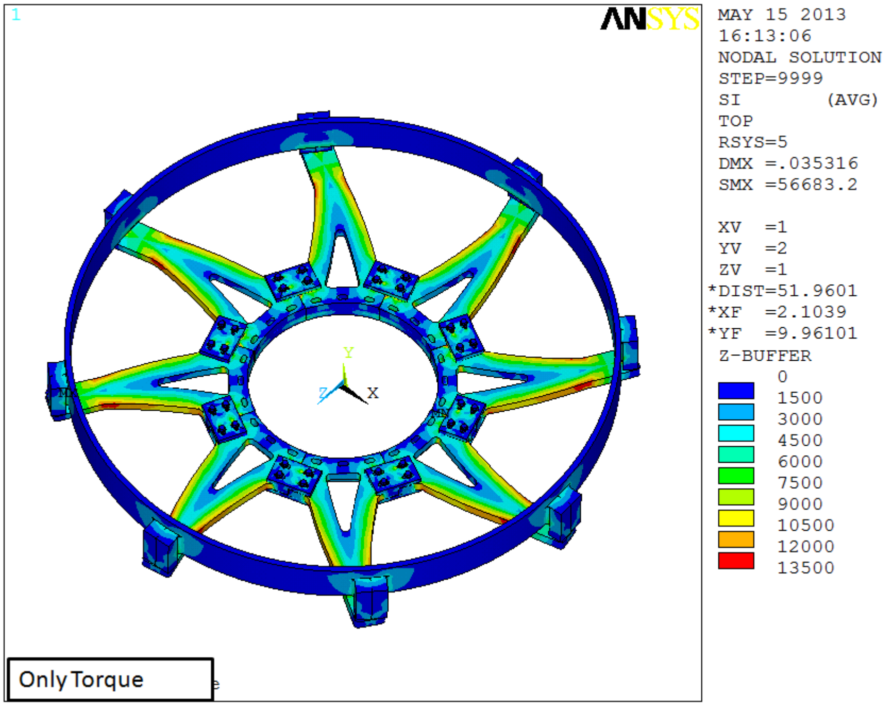


Figure 6.1.1-7 Tresca Stress in the spokes, Difference between Torque Load and Preload

The spokes are stressed to 13500 psi, which is lower than the 300 MPa/43 ksi fatigue allowable discussed in section 3.3. Local Stresses around some of the preloaded hardware are above this, but are not cyclically loaded.

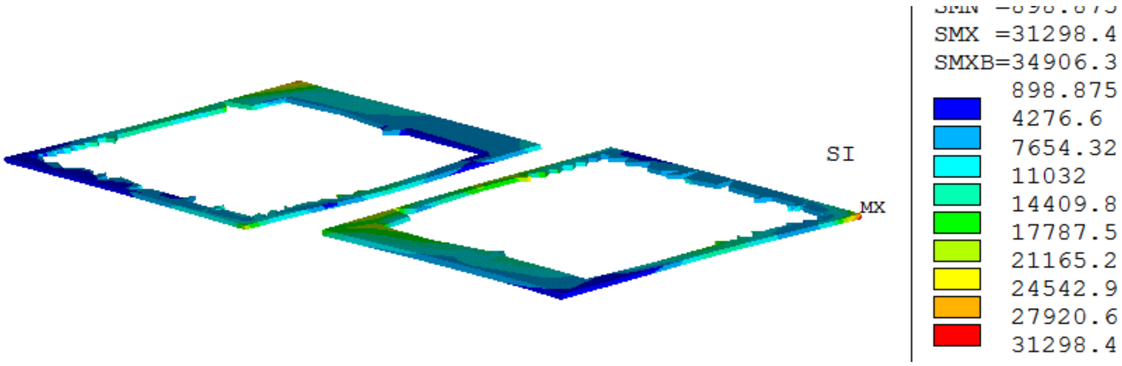


Figure 6.1.1-8 Tresca Stress in the Welds

Figure 6.1.1-8 is the stress in the plate welds. This result is for the model which does not include the halo loading discussed in section 8.1. This result is interesting because the friction factor study showed that, at least for the machine torque, the welds were not needed. This highlights the fact that the welds and the frictionally loaded bolts provide redundant load paths for part of the loads at the hub. The peak stresses at the corners of the weld might be of concern in terms of fatigue life, but the fact that the design works without the weld, would make a fatigue assessment of these welds unnecessary.

6.1.2 Lower Spoked Lid Inner Bolt Circle

The torsional moment for design of the lid/flex/diaphragm bolting and the TF steps or keys from [1] is 0.3 MN-m for the lower lid (Figure 3.2-1) and 0.25 MN-m for the upper lid from [1]. This is the torque being transmitted from the centerstack TF to the outer rim of the umbrella structure. This was translated into a load per upper TF flag of about 7000 lbs. which was increased to 9000 lbs to allow for the 10% headroom in the design point spreadsheet forces. The 0.3 MN-m torque is transmitted from the lower end of the TF column to the lower umbrella structure and vessel. Most goes through the lower spoked lid. Some goes through the pedestal to the floor and back up the support legs (.0354MN-m). A little goes through the lower TF straps. Assume the torque goes only through the lower spoked lid. The inner leg torque is carried through the upper plate of the pedestal to the inner bolt circle of the lower spoked lid.
The upper flange of the pedestal has two bolt circles that carry this torque. An inner bolt circle with 18 holes that connects to the TF collar, and an outer circle with 24 holes that connect to the spoked lid. If you take the torque load in friction, you need 1-inch 718 bolts tensioned to 100 ksi and a high friction coating. One inch bolts have a stress area of .6051 in^2 so the bolt load is 60,510 lbs.

The one inch bolts could be threaded into the pedestal top plate. The Sm allowable for 316 SS is 26.7 ksi and the shear allowable is .6* Sm or 16ksi. For bolts that are higher strength than the 316 SS, the shear area that the federal screw standards allows is .75 * the shear cylinder area. The shear cylinder is pi*.9188 * 1 = 2.886 in^2 and the shear stress is 60510/2.886/.75 = 27955 psi, which is larger than the 16 ksi allowable where .9188 is the pitch diameter of the 1 inch bolt. For 1 inch bolts, the preload tensile stress is 100 ksi.

The 316 pedestal plate can’t take the thread shear, and 718 nuts will be required.

According to the design criteria [2], this can be 75% of yield, so a bolt with 100000/.75 = 133 ksi yield or higher is needed.

6.1.3 Lower Bolted Plate Bolting

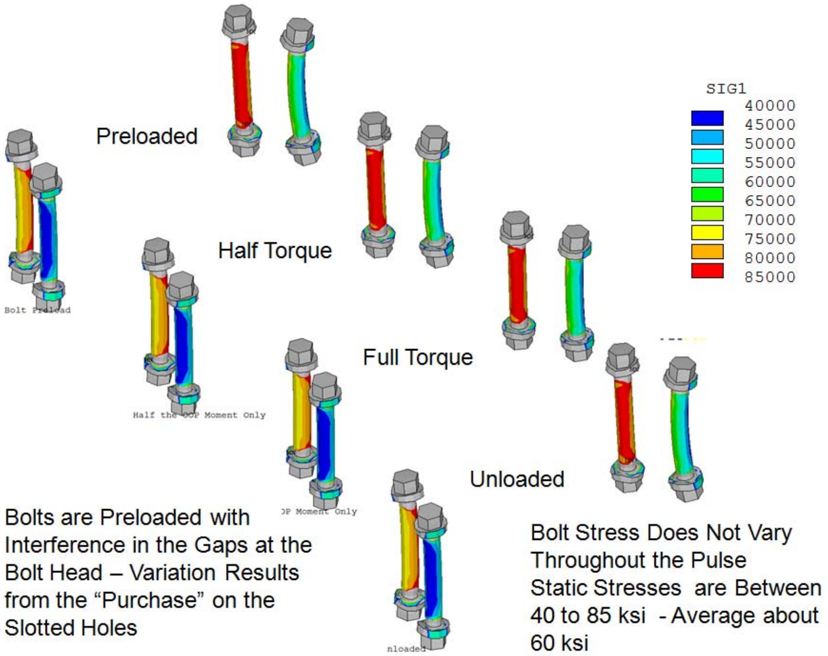


Figure 6.1.3-1 Bolt Stresses in the Bolted Plate Hub Simulation for Mu=.35

The hub joint needs to be qualified for cyclic loading. Fitted joints were considered. This approach would have loaded the bolts cyclically in shear. For a friction connection, the bolts are not cycled. Bolts which can be preloaded to 60 ksi are required.

6.2 FDR Flat Lower Spoked Lid

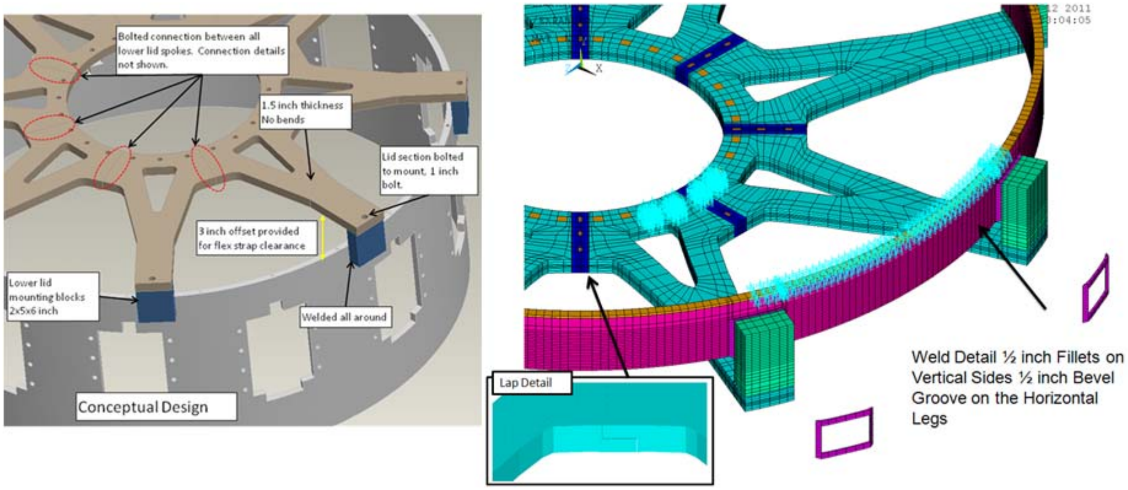
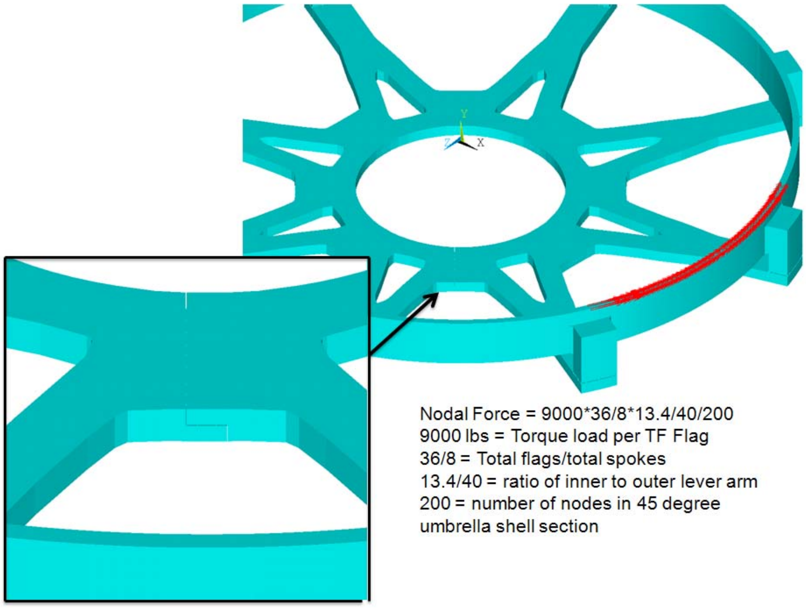


Figure 6.2-1 Flat Lower Spoked Lid Model

In Figure 6.1-1, the Pro-E model at left has been turned upside down to show the details of the connection of the lid to the umbrella shell. Lugs are used to obtain the required offset to clear the TF straps. At right, the FEA model is shown. Welds are modeled rather than analyzing a merged connection. Weld stresses can be extracted directly from this model.



Nodal Force = 9000*36/8*13.4/40/200
9000 lbs = Torque load per TF Flag
36/8 = Total flags/total spokes
13.4/40 = ratio of inner to outer lever arm
200 = number of nodes in 45 degree
umbrella shell section

Figure 6.2-2 Flat Lower Spoked Lid Model with Lap Joint at Inner Hub

Figure 6.2-2 shows the applied forces and the inner lap joint modeling. A flanged joint may be used in place of the lap joint.

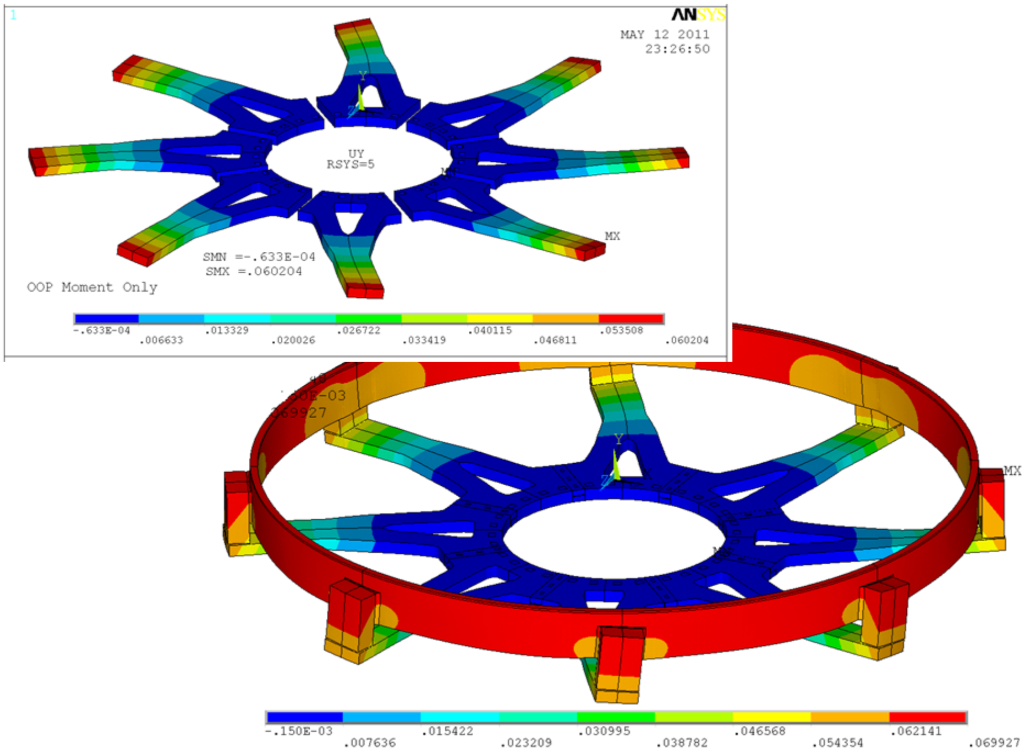
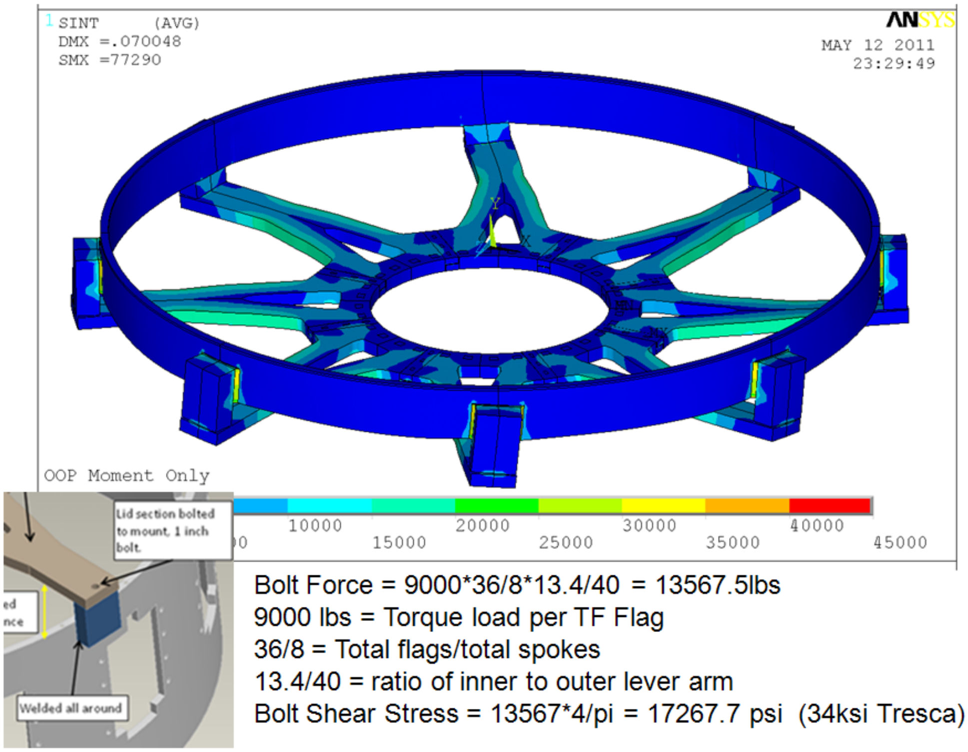


Figure 6.2-3 Flat Lower Spoked Lid Rotational (Theta) Displacement



Bolt Force = 9000*36/8*13.4/40 = 13567.5lbs
9000 lbs = Torque load per TF Flag
36/8 = Total flags/total spokes
13.4/40 = ratio of inner to outer lever arm
Bolt Shear Stress = 13567*4/pi = 17267.7 psi (34ksi Tresca)

Figure 6.2-4 Tresca Stress in the Lower Flat Spoked Lid and "Hand" calcs for the Bolt to Lug Connection for Moment Loading Only

Figures 6.2-4 and 5 show the results for the lap joint that was modeled as the connection between the spoke segments. A flanged connection may be used, but the lap joint bolting is representative of bolt loads and stresses for whatever connection is chosen.

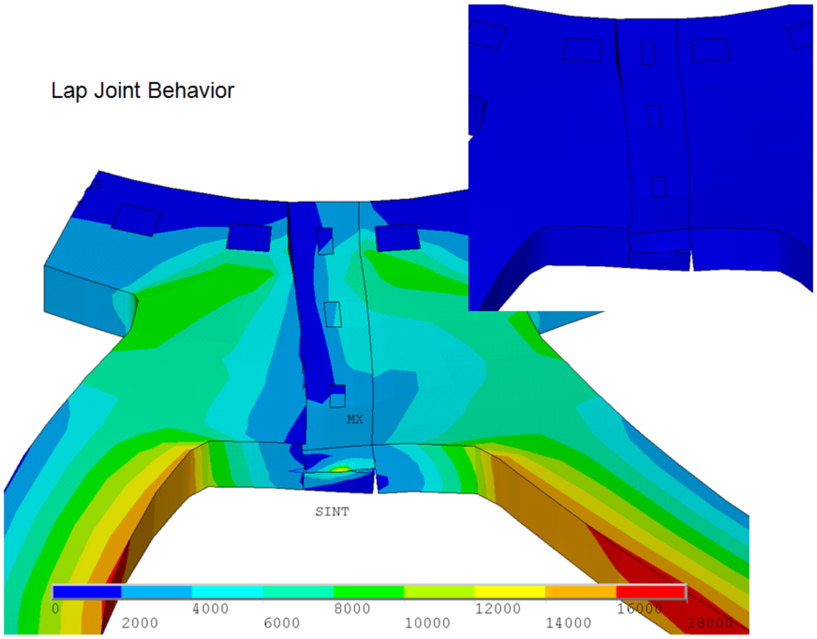
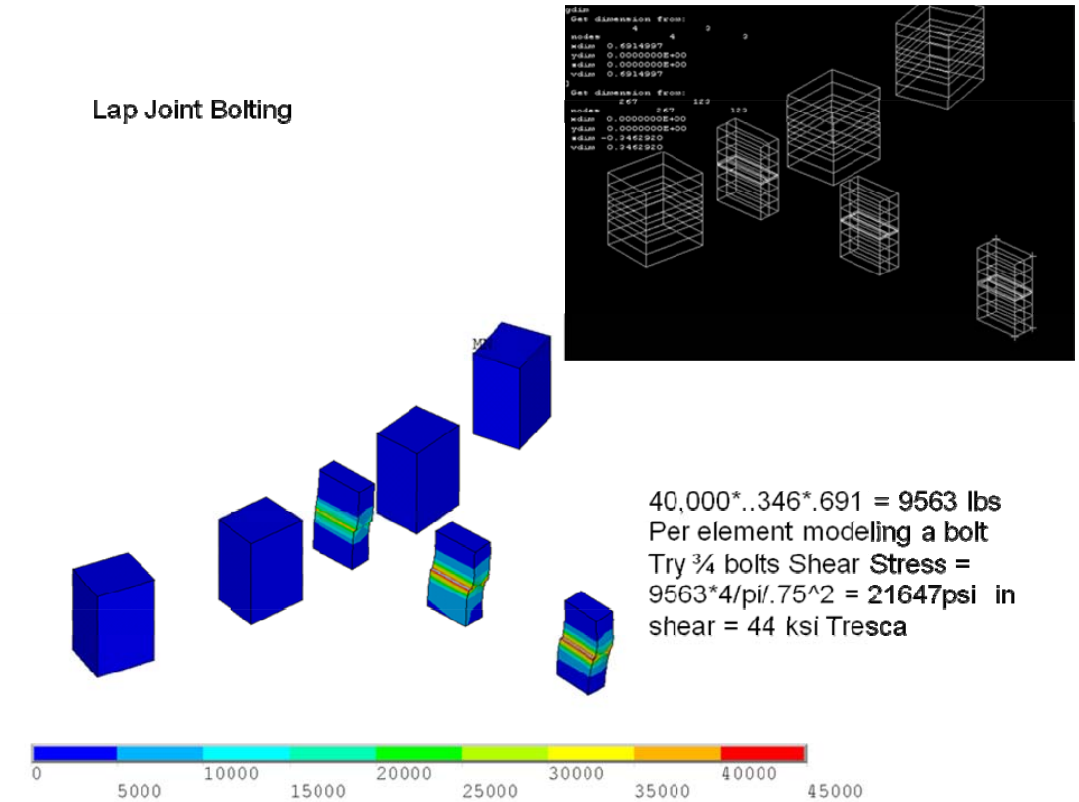


Figure 6.2-5 Lower Flat Spoked Lid Lap Joint Behavior

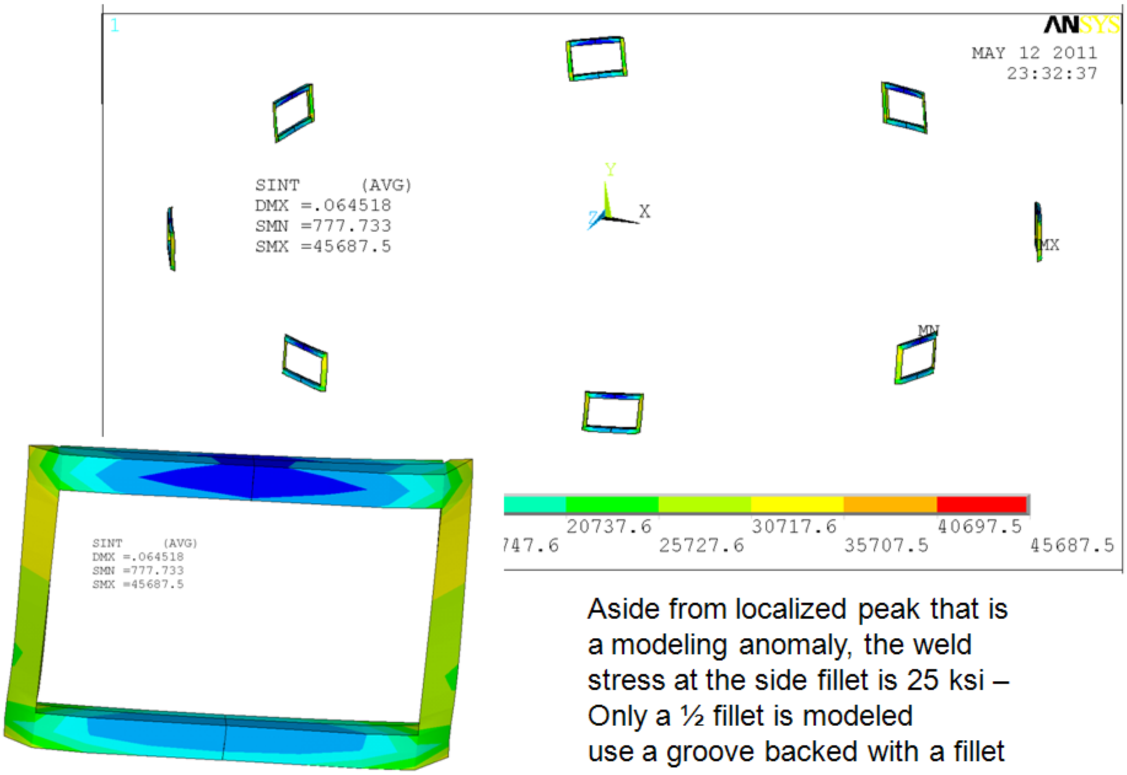


Lap Joint Bolting

```
gdim
 Get dimension from:
              4            3
 nodes              4            3
 xdim  0.6914997
 ydim  0.0000000E+00
 zdim  0.0000000E+00
 vdim  0.6914997
)
 Get dimension from:
           267          123
 nodes             267          123
 xdim  0.0000000E+00
 ydim  0.0000000E+00
 zdim -0.3462920
 vdim  0.3462920
```

40,000*..346*.691 = 9563 lbs
Per element modeling a bolt
Try ¾ bolts Shear Stress =
9563*4/pi/.75^2 = 21647psi in
shear = 44 ksi Tresca

Figure 6.2-6 Lower Flat Spoked Lid Lap Joint Bolting.



Aside from localized peak that is
a modeling anomaly, the weld
stress at the side fillet is 25 ksi –
Only a ½ fillet is modeled
use a groove backed with a fillet

Figure 6.2-7 Lower Flat Spoked Lid Lug to Umbrella Shell Weld

6.3 Lower Spoked Lid with Offset

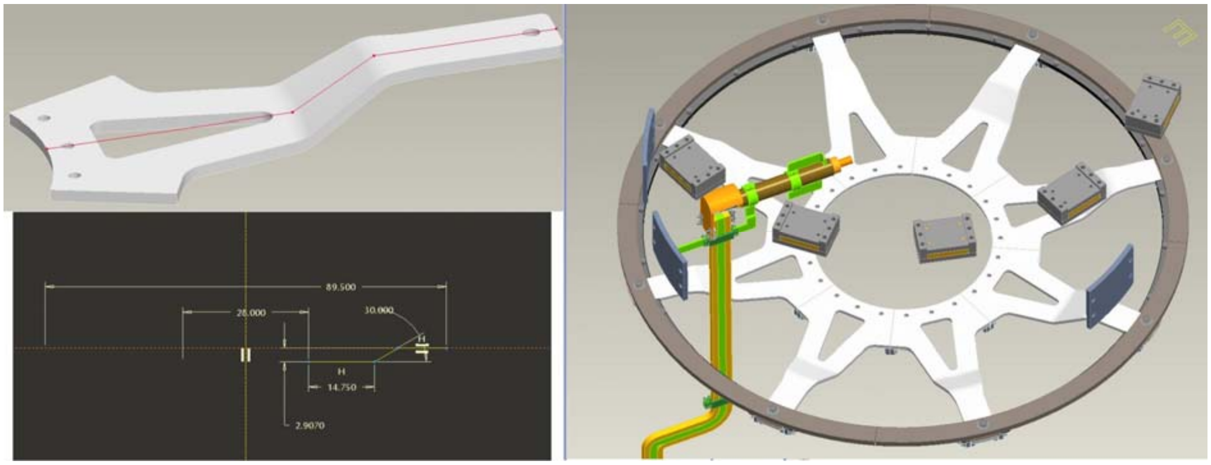


Figure 6.3-1 ProE plots of the lower spoked lid. Dimensions of the offset are indicated (left) and the view (right) showing the routing of the OH coax lead - one of many connections passing through the lower lid.

The bend in the spokes in the lower lid will produce a different torsional stiffness than in the flat upper lid. To investigate this, the upper lid model was "upset" and the torque loads were applied.

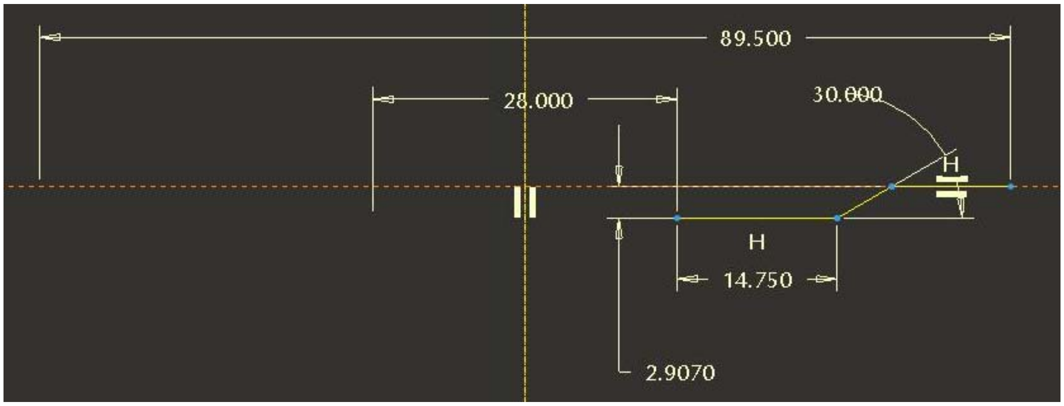
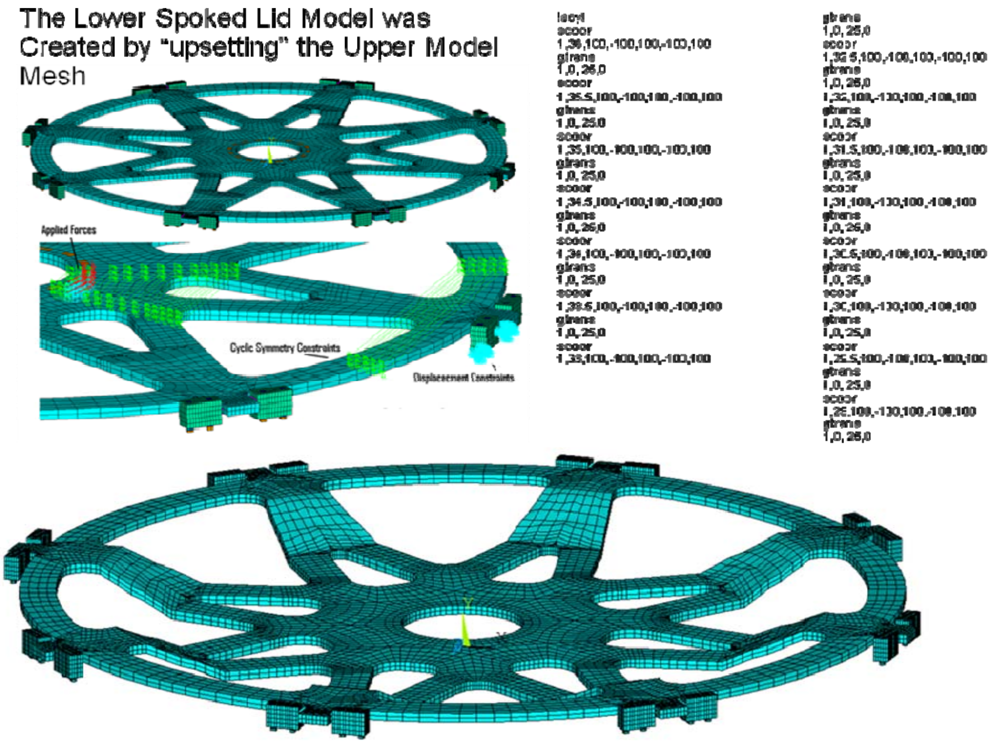


Figure 6.3-2 Dimensions of the offset of the spoke to accommodate the lower TF Straps.



```
lecyl
scoor
1,36,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,35.5,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,35,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,34.5,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,34,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,33.5,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,33,100,-100,100,-100,100
```

```
gtrans
1,0,25,0
scoor
1,32.5,100,-100,100,-100,100
gtrans
1,0,25,0
1,32,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,31.5,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,31,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,30.5,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,30,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,29.5,100,-100,100,-100,100
gtrans
1,0,25,0
scoor
1,29,100,-100,100,-100,100
gtrans
1,0,25,0
```

Figure 6.3-3 Symmetry Expansion of the Bent Spoke Model. and methodology of developing the lower spoke Model from the upper Spoked Lid Model.

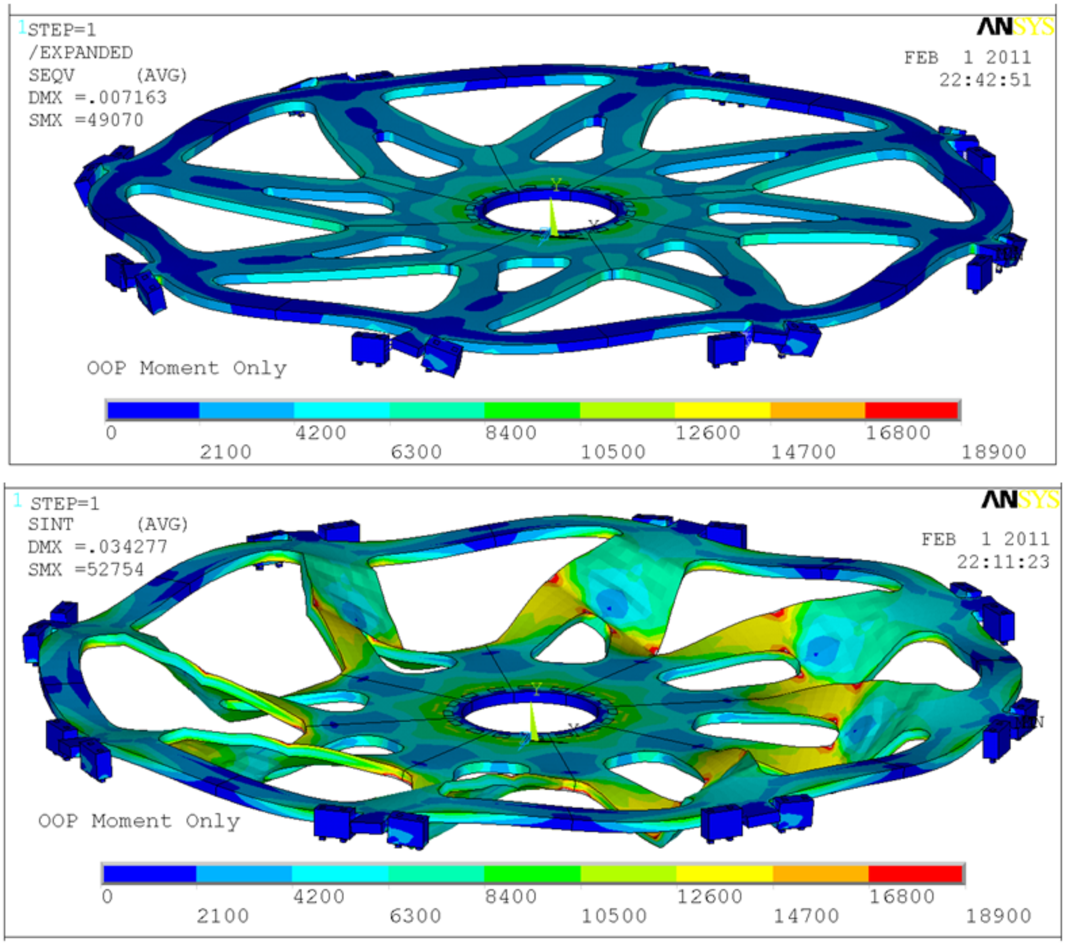


Figure 6.3-4 Results showing the Rotations of the legs Caused by the Bends

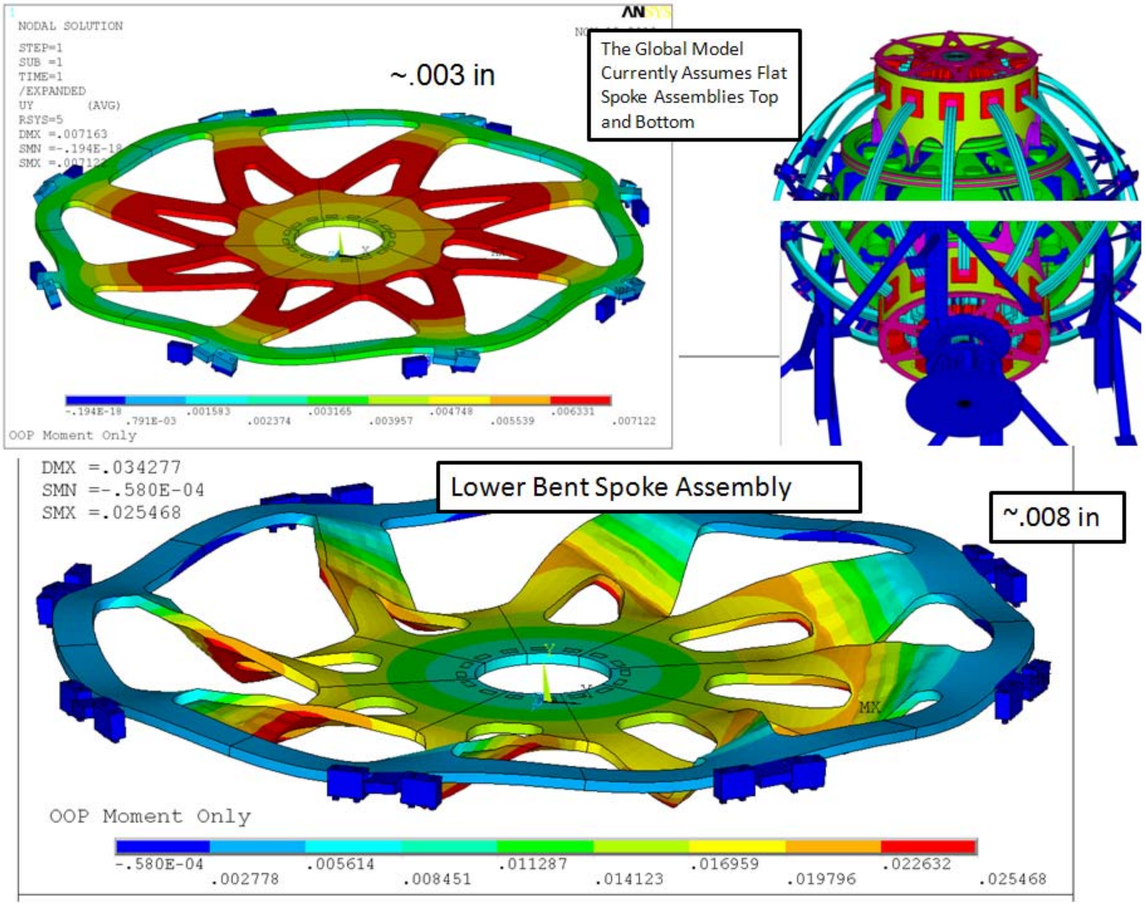


Figure 6.3-5 Results showing the Rotational Displacements for Flat and Bent Spokes.

The March 2011 IGES model of the lower spoked lid with the bend will be substantially less stiff than the upper. The spokes are narrower and the thickness is the same. The torsional resistance goes as b*t^3 so a modest increase in thickness should improve stiffness, but some of the "b" or spoke width has been lost in the process. Having the upper lid stiffer than the lower one will distribute more torsional shear stress to the upper end of the TF inner leg, which is already at the limit. It also shifts the torsional registration features of the lower structures to the bellows, which is a delicate part of the vacuum boundary, and could introduce operational reliability issues.

7.0 Global Model Results

The global model described in reference [1] was updated with the lower pedestal and spoked lid designs. This provides a means to qualify the stresses in the spoked lid. The pedestal included in the final design is a torsionally stiff design different than the pedestal used in the existing NSTX. In the global model response, the stiffness of the pedestal and the compliance of the lower spoked lid do not seem to be causing anything unacceptable. The inner leg torsional shear is altered, and this is discussed in section 7.2.

7.1 Stresses in the Spoked Lids

7.1.1 Normal Scenario Upper and Lower Spoked Lid Stress

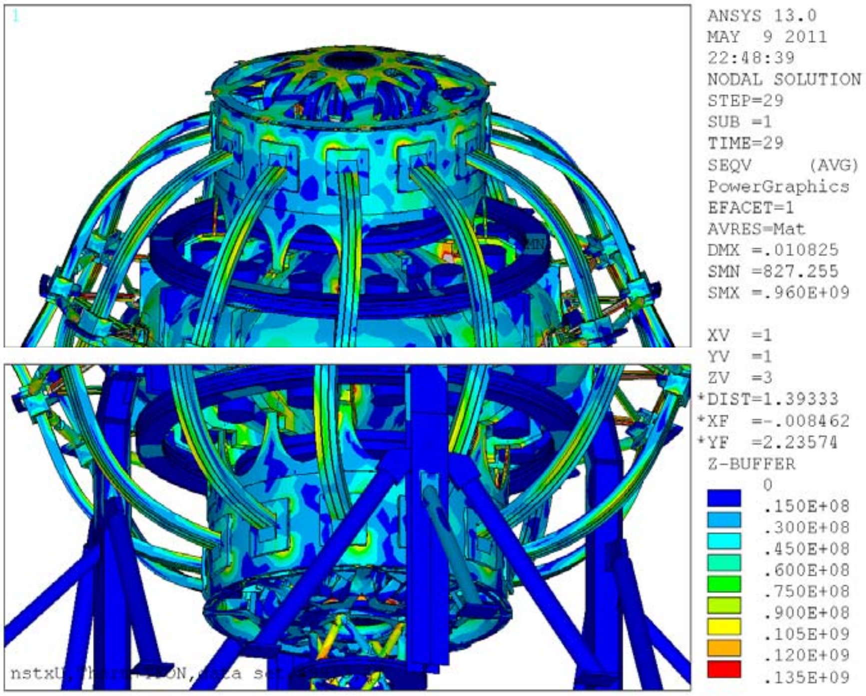


Figure 7.1.1-1 Global Model Results with Views of the Upper and Lower (Bent) Spoked Lids

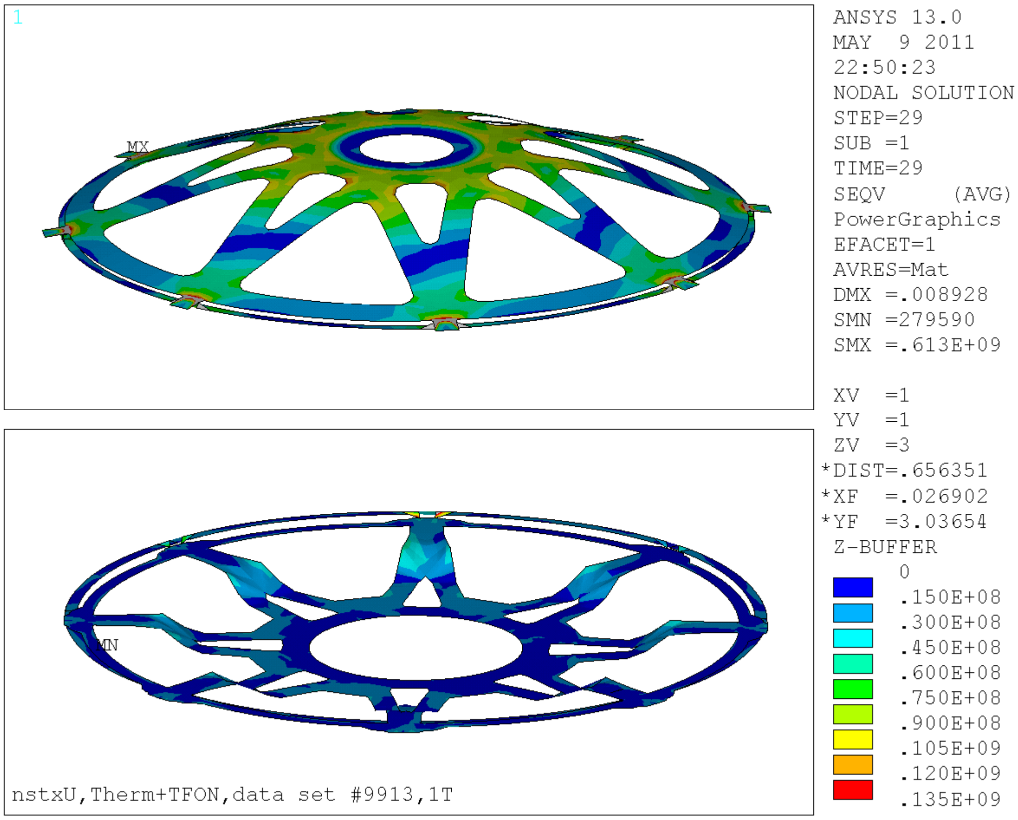


Figure 7.1.1-2 Global Model Results with the Upper and Lower Spoked Lids Selected

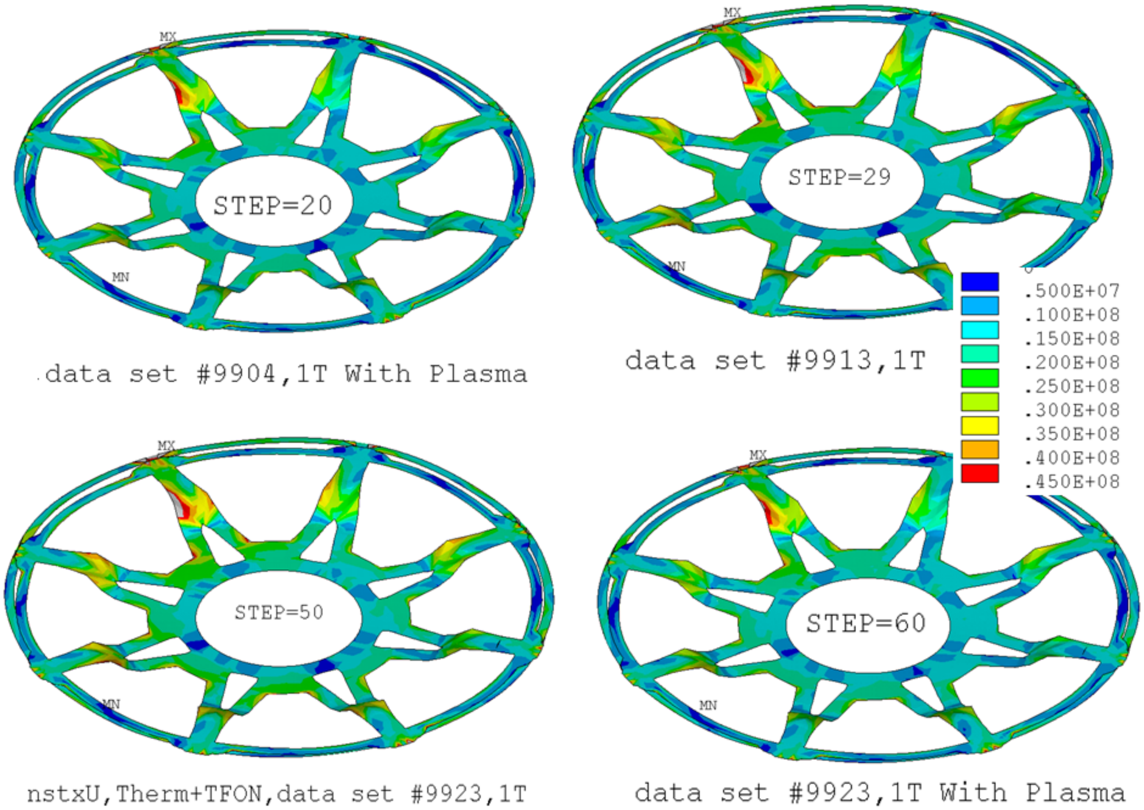


Figure 7.1.1-3 Global Model Results Lower Spoked for four different Equilibria

Note that there is very little difference between the results for various load files/equilibria.

7.1.2 Bake Out Spoked Lid Stress

Bake out is simulated in the global model. The bake out expansion of the vessel is comparable in magnitude to the normal operating expansion of the centerstack, only the displacements are in opposite directions. Stresses are low and there are many fewer cycles of bake out loading.

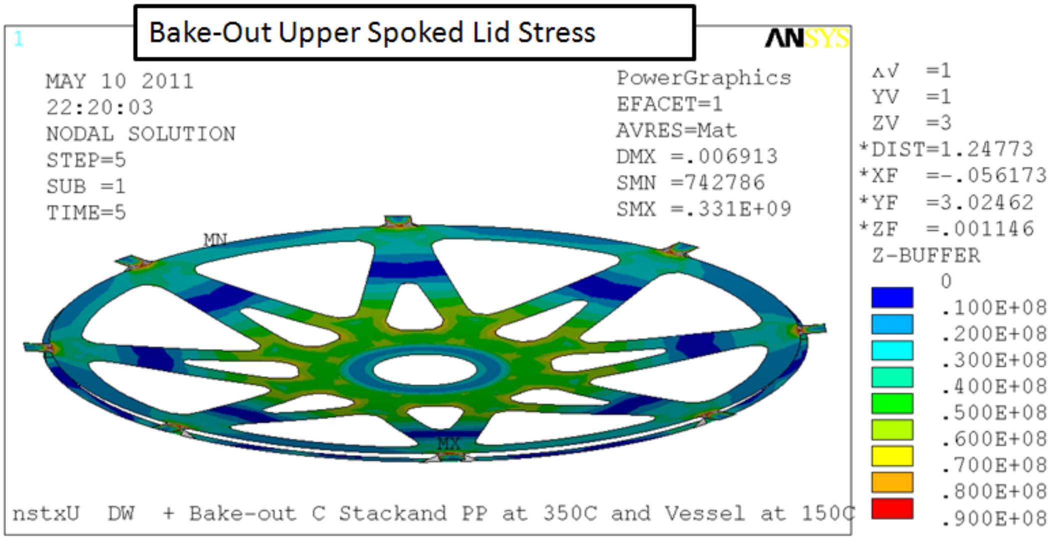


Figure 7.1.2-1 Global Model Bake Out Stresses in the Upper Spoked Lid

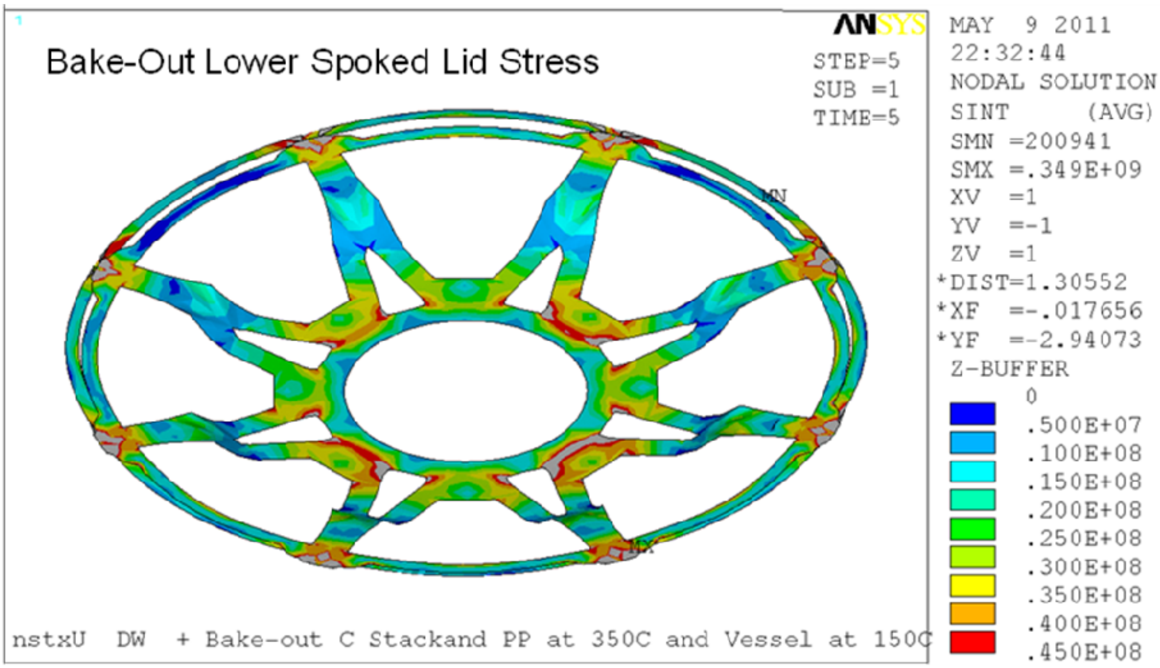


Figure 7.1.2-2 Global Model Results Lower Spoked Lids Bake-Out Stress

7.2 OOP Load Distribution with the FDR Spoked Lid Design

The main purpose of including the lid in the global model is to address the need for torsional stiffness or compliance of the plate to ensure that the inner leg torsional shear stress is acceptable with the FDR configurations. The concern comes from the relative compliance of the bent spokes in the lower lid. Figure 4.0-3 shows the global model of the tokamak including the upper and lower spoked lids.

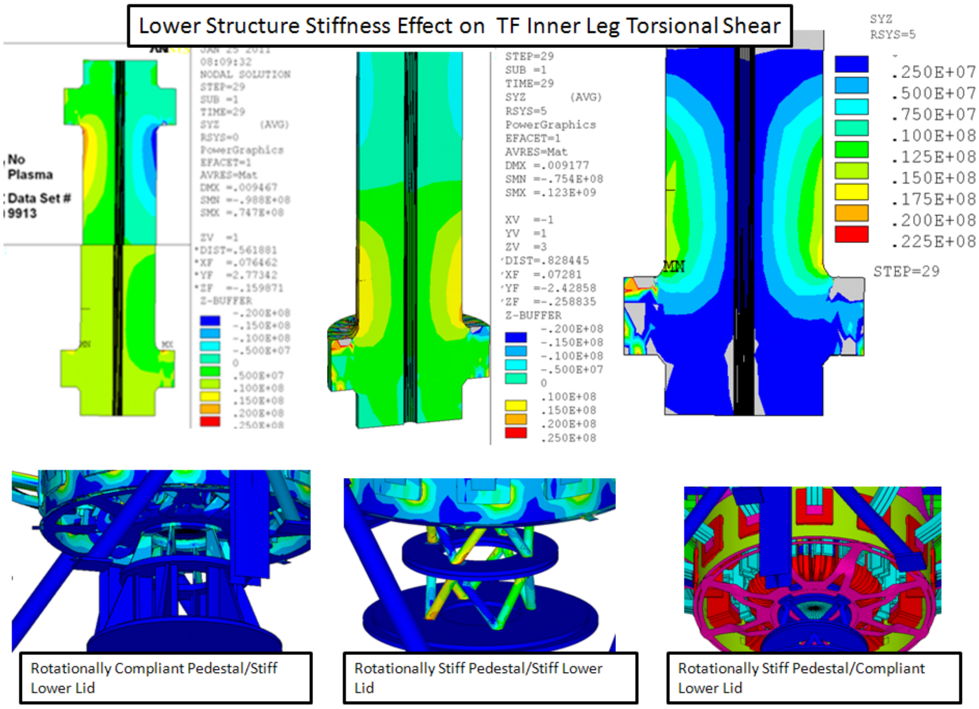
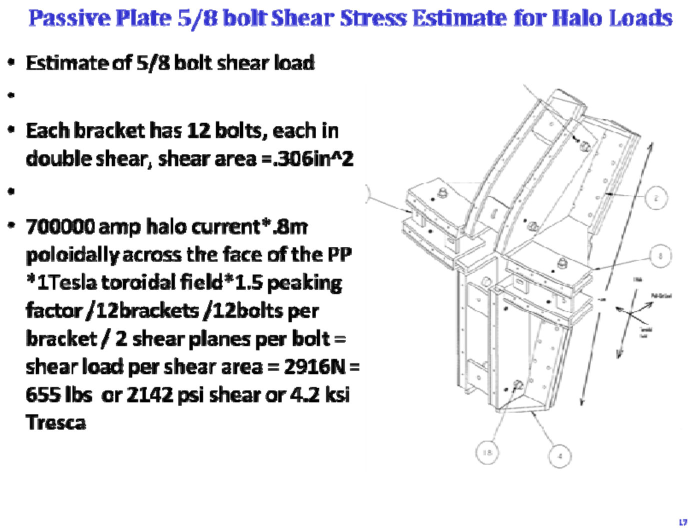


Figure 7.2-1 Comparison of TF Inner Leg Torsional Shear for Various Lower Assembly Stiffnesses

8.0 Halo Loads

Halo loads on the upper end of the centerstack casing are transferred to the bellows and then through the ceramic break to the vessel. The upper lid is not in this load path. Halo currents load the passive plates and the centerstack casing. Casing loads are transmitted to the pedestal directly. The passive plate loads are transferred to the vessel and then to the foundation through the braced legs, and/or through the umbrella structure to the lower lid, then to the pedestal. Had the bent spokes of the lower lid been retained, they would have introduced a bending compliance to the lateral stiffness of lid. Most of the passive plate loading would have been transferred to the vessel braced columns, reacted internally within the vessel or resisted by inertia of the vessel and appurtenances. With the stiffer, flat lower lid, more of the halo loads will be reacted through the vessel structures and less through pedestal and vessel legs. Halo loads are not considered in the finite element analysis of the spoked lids, but simplified calculations are presented below. Upper and lower design loads for the centerstack casing halo loads were provided by Art Brooks [9].

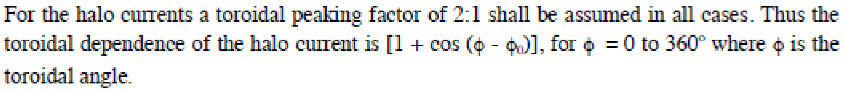


Passive Plate 5/8 bolt Shear Stress Estimate for Halo Loads

- Estimate of 5/8 bolt shear load
-
- Each bracket has 12 bolts, each in double shear, shear area =.306in^2
-
- 700000 amp halo current*.8m poloidally across the face of the PP *1Tesla toroidal field*1.5 peaking factor /12brackets /12bolts per bracket / 2 shear planes per bolt = shear load per shear area = 2916N = 655 lbs or 2142 psi shear or 4.2 ksi Tresca

Figure 8.0-1 Figure from reference [8] calculating the Halo Current Loading on the Passive Plates

Inductive disruption loads on the vessel and passive plates are basically axisymmetric. Halo current loads can develop lateral loads in both the centerstack casing and in the vessel that will appear in connections between the outer vessel structure and the inner casing. These connections consist of the supports down to the floor - the pedestal and vessel legs, the spoked lid and the bellows/ceramic break. Ideally, the bellows and ceramic break should see little of this load.

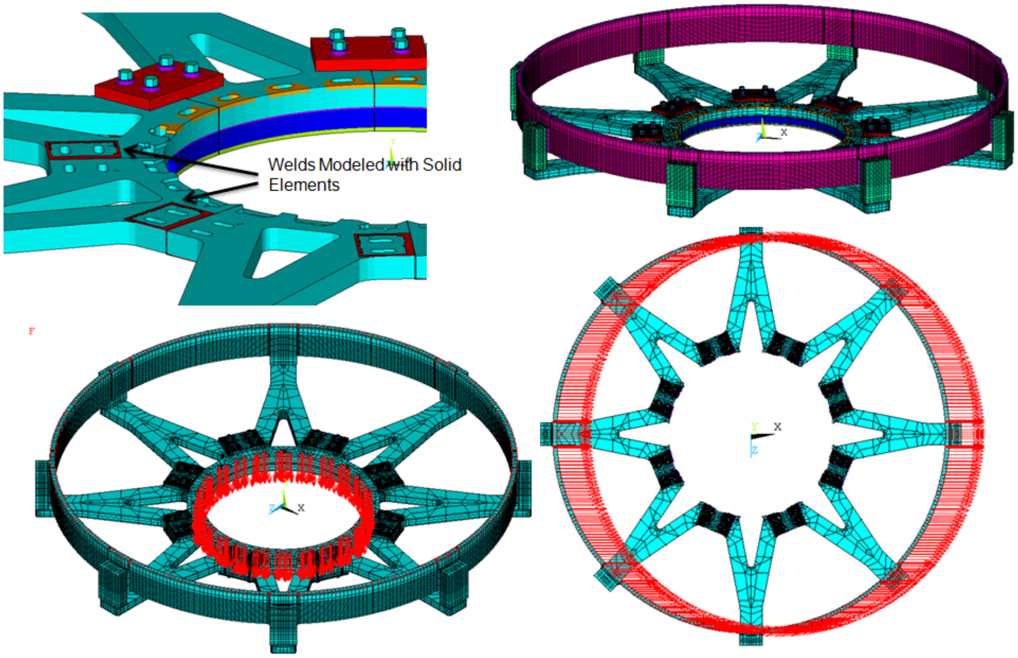


For the halo currents a toroidal peaking factor of 2:1 shall be assumed in all cases. Thus the toroidal dependence of the halo current is $[1 + \cos(\phi - \phi_0)]$, for $\phi = 0$ to 360° where ϕ is the toroidal angle.

The net load is the integral of cos^2 or .5, so the net load on the vessel and potentially by the spoked lid being transferred to the pedestal is 0.5*700000amps*.8m*1T *.2248=63,000 lbs. At the lower lid with 8 spokes, this would be split over 8 outer one inch bolts or 7875 lbs per bolt. At the outer umbrella structure rim, the lower spoked lid has been designed for 9000*36 /8*13.5/40 = 13,700 lbs. Concurrence of a worst torque and a worst halo disruption should be considered a faulted case. The one inch bolts at the end of the spokes (in the flat Lower Spoked Lid Design) have a yield of 80 ksi and would have a yield shear capacity of 80*pi/4/2 = 31,000 lbs.

8.1 Lower Spoked Lid FEA Analysis with Torque and Halo Loading

The lower spoked lid is an assembly of segments that rely on frictional bolted connections. The lateral loading from the halo currents in the outer structures, the passive plates and vessel will develop some load that will be reacted through the lower spoked lid. Some will actually go through the bellows, and some will go through the load path that included the vessel support columns, floor and pedestal. The cyclic symmetry model described in the section 6.1.1 friction study was repeated eight times and the outer lugs and umbrella structure ring segment added.



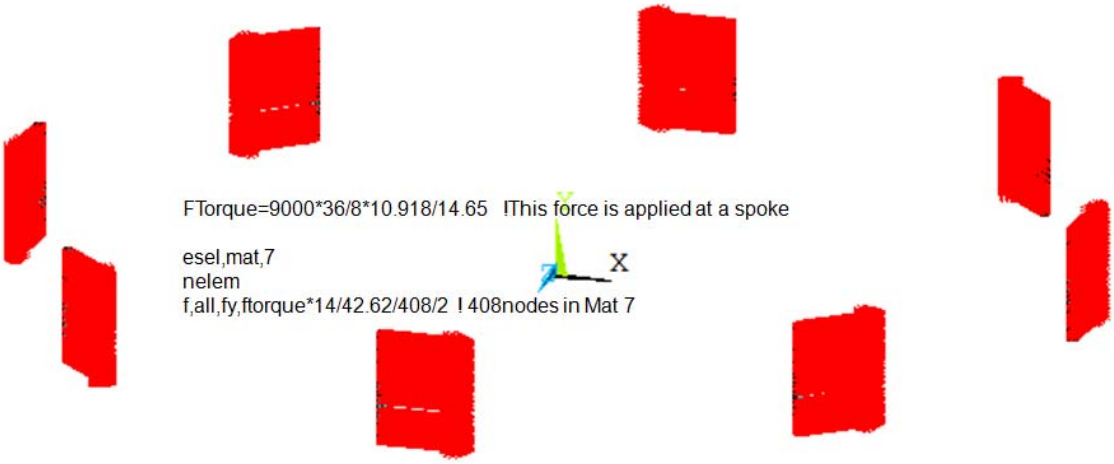
All Loads Shown Including the Inner Bolt Circle Bolt Preload and the Halo Nodal Forces at the Umbrella Perimeter

Hub Bolt Preload Forces Not Shown to Allow Viewing of the Halo Nodal Forces at the Umbrella Perimeter

Figure 8.1-1 Loading in the Spoked Lid Halo Load Analysis

The 63,000 pound lateral loading discussed in section 8.0 was applied uniformly to the outer ring that models the umbrella structure.

```
f,all,fx,fhalo/1664       ! 1664 nodes in Mat 41, fhalo=63000 lbs.
```

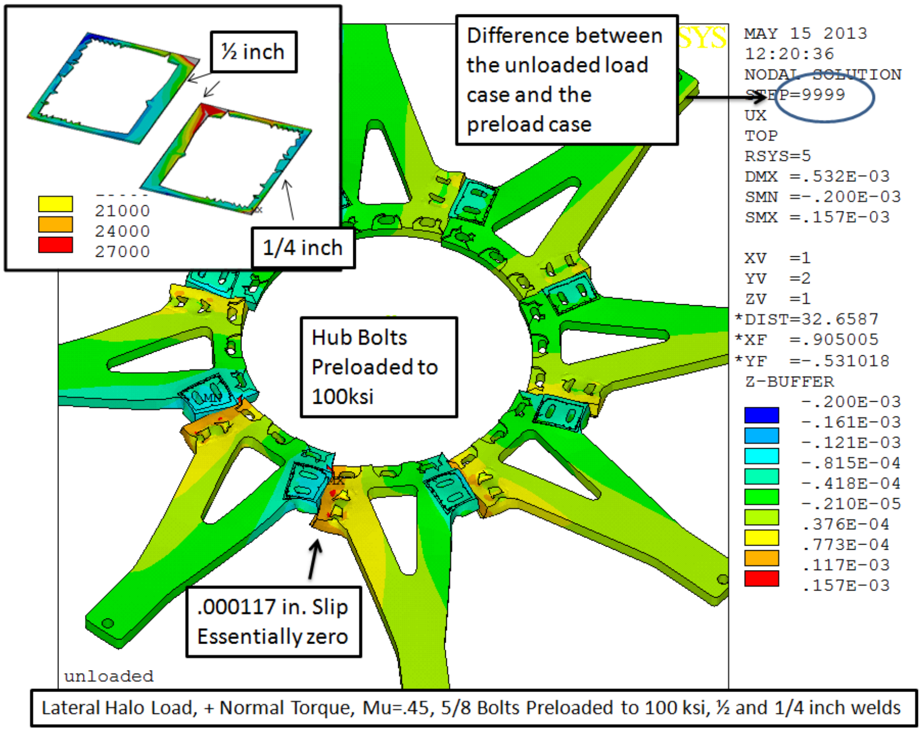


```
FTorque=9000*36/8*10.918/14.65   !This force is applied at a spoke

esel,mat,7
nelem
f,all,fy,ftorque*14/42.62/408/2  ! 408nodes in Mat 7
```

Figure 8.1-2 Torsional Loading in the Spoked Lid Halo Load Analysis

Elements modeling the pinned connection between the spokes and blocks welded to the umbrella structure were used to apply the torque. This is shown in Figure 8.1-2


½ inch

1/4 inch

Difference between the unloaded load case and the preload case

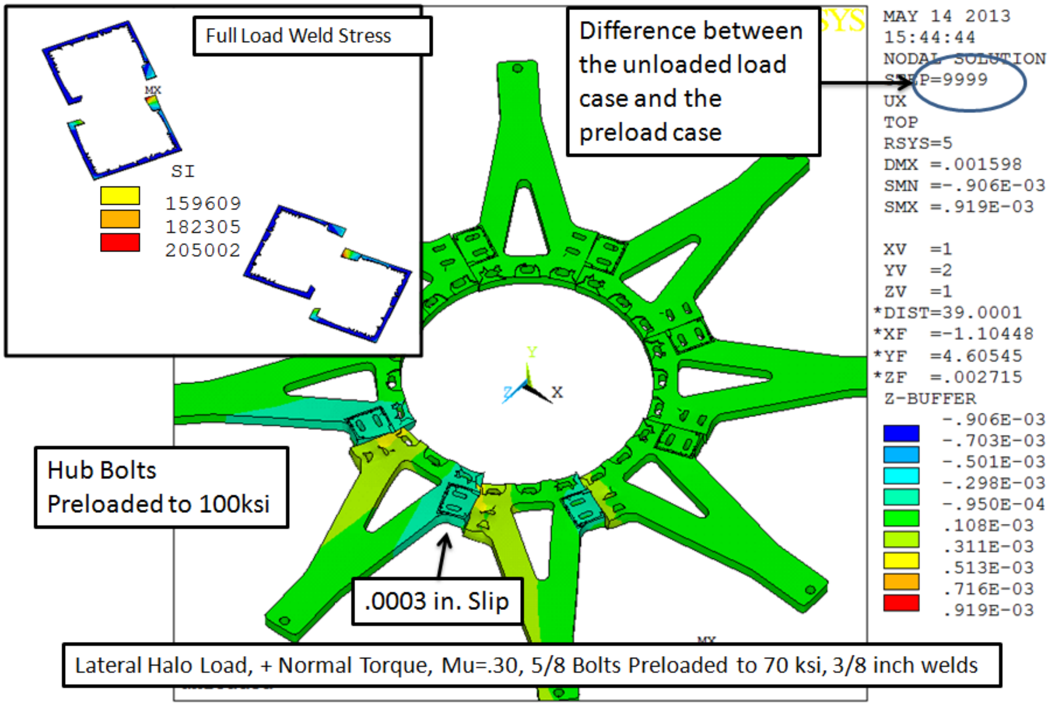
Hub Bolts Preloaded to 100ksi

.000117 in. Slip Essentially zero

Lateral Halo Load, + Normal Torque, Mu=.45, 5/8 Bolts Preloaded to 100 ksi, ½ and 1/4 inch welds

Figure 8.1-3 "Final" Radial Displacement loaded with Torque and Halo Loads

A series of analyses was performed with increasing bolt preloads and larger welds to eliminate slippage and produce a reasonable weld stress. The difficult loading to sustain is the lateral halo load.


Full Load Weld Stress

Difference between the unloaded load case and the preload case

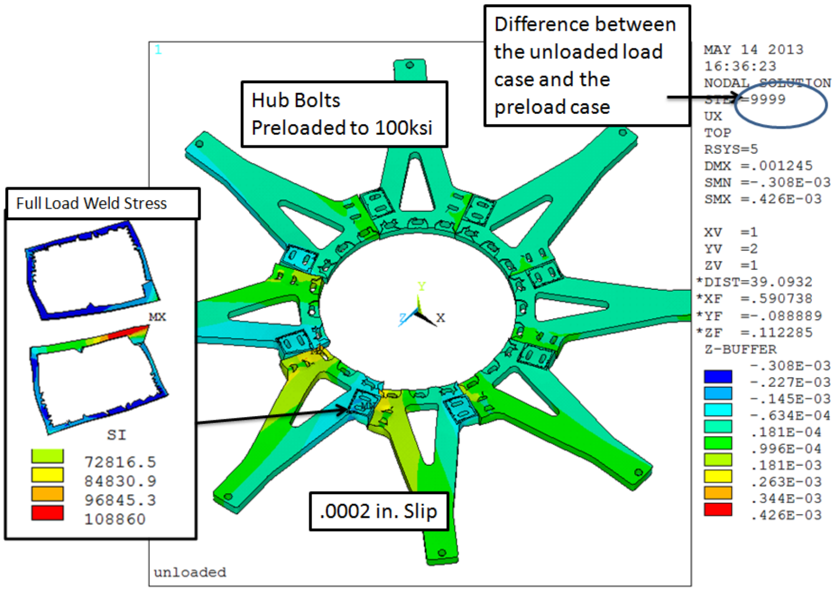
Hub Bolts Preloaded to 100ksi

.0003 in. Slip

Lateral Halo Load, + Normal Torque, Mu=.30, 5/8 Bolts Preloaded to 70 ksi, 3/8 inch welds

Figure 8.1-4 Radial Displacement with the Halo Loads and Torque Loads Removed (Insufficient Weld)

Figure 8.1.4 shows a permanent set or slippage of .0003 inches as a result of the halo loading. This is based on a friction coefficient of .3. A high friction coefficient coating is planned for these joints. An increased weld cross section is also possible.



Lateral Halo Load, + Normal Torque, Mu=.35, 5/8 Bolts Preloaded to 70 ksi, 3/8 inch welds

Figure 8.1-5 Radial Displacement with the Halo Loads and Torque Loads Removed (Insufficient Weld)

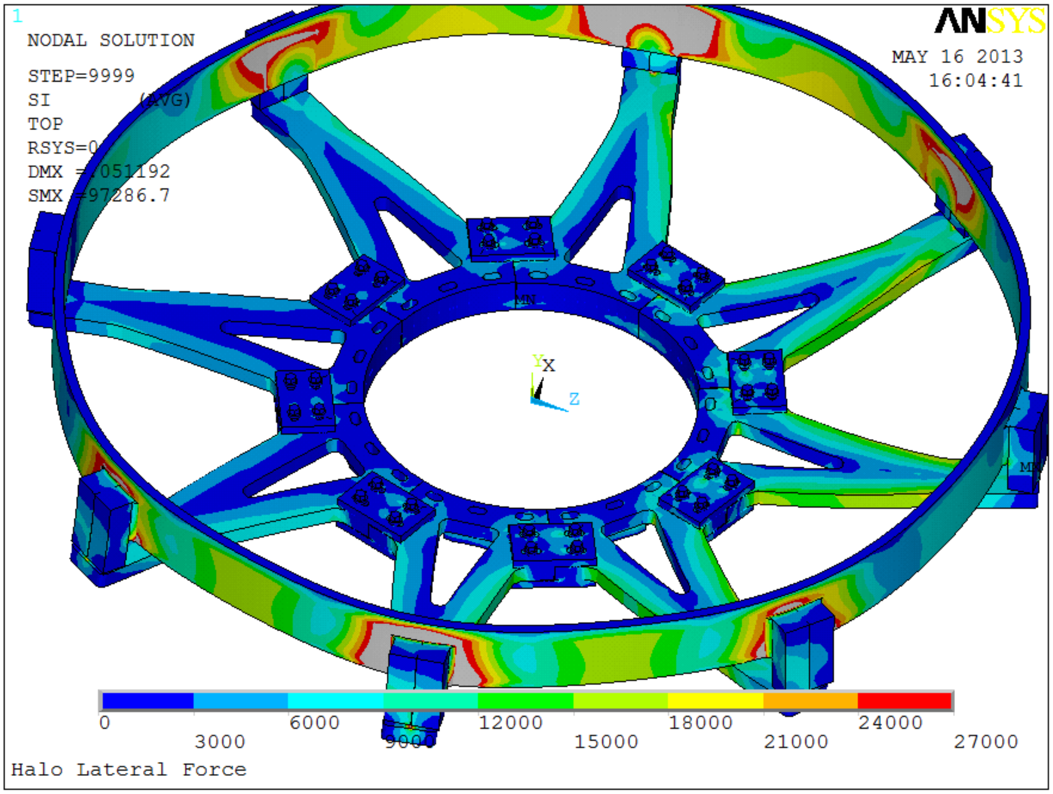


Figure 8.1-6 Stress With Machine Torque and Halo Lateral Loads Applied (Differenced with the Preload)

The stress due to halo lateral loading is dominated by the preload stresses around the bolts and the flexure of the ring that models the umbrella structure. The ring is fictitious, and the umbrella structure analyses should be consulted for this component. The stresses in figure 8.1-6 are the differential stresses with respect to the preload load case. The spoked lid components are stressed below 124 MPa (18 ksi) – below the 300 MPa fatigue allowable, even though the halo loading has a much lower cycle count than the normal pulse requirement of 30,000 cycles.

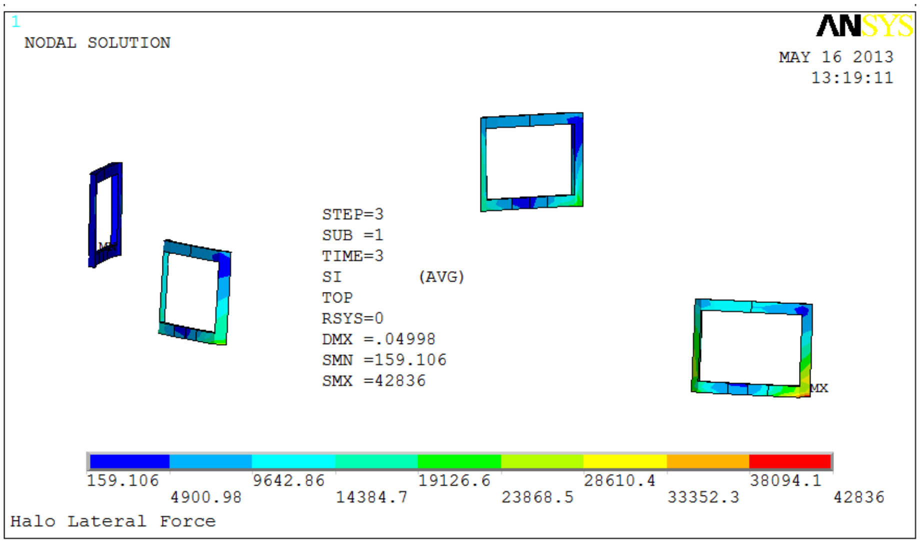


Figure 8.1-7 Lower Flat Spoked Lid Lug to Umbrella Shell Weld – Halo Disruption Loading

The local peak in the corner is a common stress condition in the NSTX square corner welds. In this case, it is for disruption halo current loading and many fewer cycles of loading are expected. A static allowable is appropriate for this weld. The bulk of the weld is below 30 ksi. Table 3.3-2 lists 241 MPa, or 35 ksi, for a static weld bending allowable.

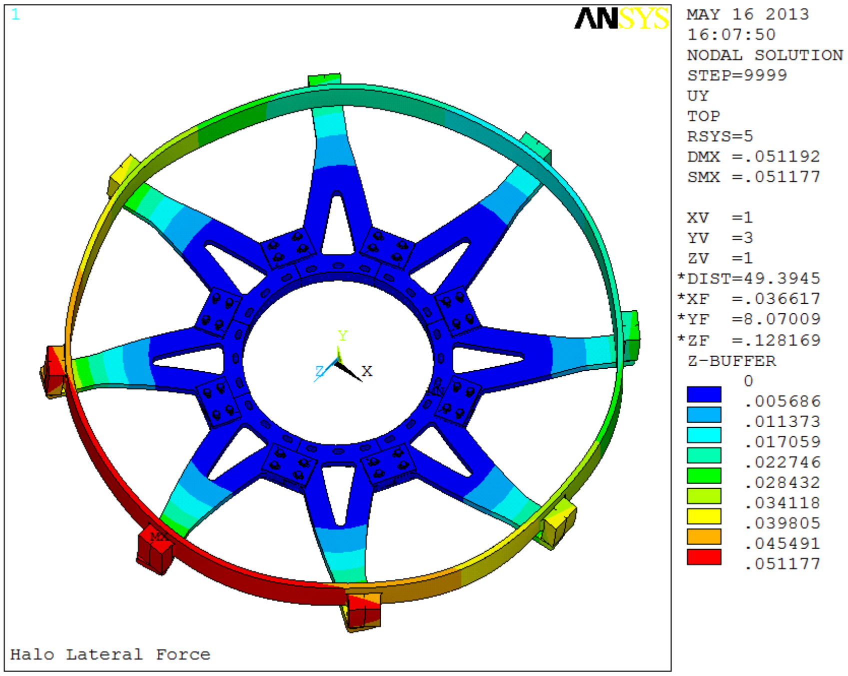


Figure 8.1-8 Lower Flat Spoked Lid Theta Displacement – Halo Disruption Loading

Appendix A
Original Flex Plate

A flex plate or cover or "lid" was intended as the structure that extends from a connection to the TF central column flags to the outboard edge of the umbrella structure. These details are only concepts in the drawings as of 2010, and a simple representation of the plate is included in the global model (Figure 3). The flex plate must allow the relative motions of the central column which is fixed vertically at the lower end by connections to the pedestal and to the lower TF flag extensions. The upper connections between the outer rim of the umbrella structure and the TF flags must allow the full vertical expansion of the central column. This is 9 mm at the elevation of the connection. The lid/flex plate is intended to bend and absorb the vertical motions elastically. Bending stresses develop at the ID and OD of the plate which produce prying moments at the bolt circles.

The prying moments, or Mb inner and outer (in Figure 2), are the bending stress multiplied by the plate section modulus or on a per perimeter length basis, the moment is the stress times (t^2)/6.

Figure 1 Bruce Paul's Model of the Lid/Flex Region

At the outboard bolt circle, the stress is about 150 MPa (Figure 3) and the moment is 150 MPa *(5/8/39.37)^2/6 = 6300 N-m/m. If there were bolts every 20cm, then the prying moment would be 6300*.2 = 1260 N-m and if the distance from the bolt centerline to the edge of the plate were 10 cm, the bolt load would be 12600 N or 3000 lbs. In the global model, the inner edge is pinned, due to a plate element to solid transition. In actuality, it will probably be a bolted connection. For design purposes, the inner flex can be considered as having 150 MPa bending as well as the contribution from the outer diameter of the flex.

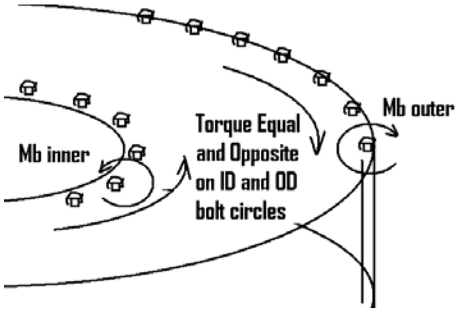


Figure 2 Torques and Moments

The original CDR concept employed a relatively thin (5/8 inch) disk or diaphragm.

Lid/Flex/Diaphragm Stresses with Access Ports

Access ports were added to the flex (figure 1) and the flex was re-analyzed (Figure 4). The stresses went up a bit, from 150 to 180 MPa. This would have been acceptable in terms of stress but more extensive access was desired. The lower "flex" or cover (Figure 5) needed to be segmented and to have more extensive access to service connections. Servicing requirements led to the spoked lid concept.

Upper Flex Plate/Diaphragm Replaces the Gear Tooth Connection

Hot Central Column, Cold Vessel

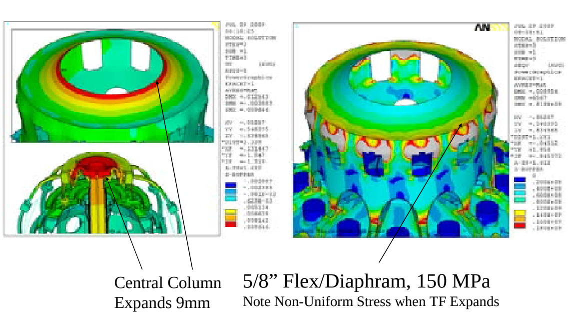
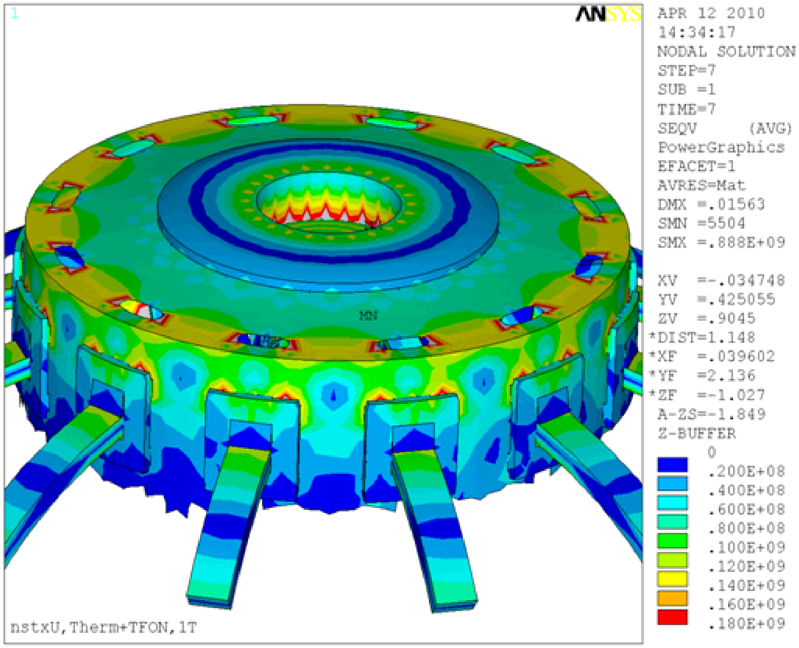
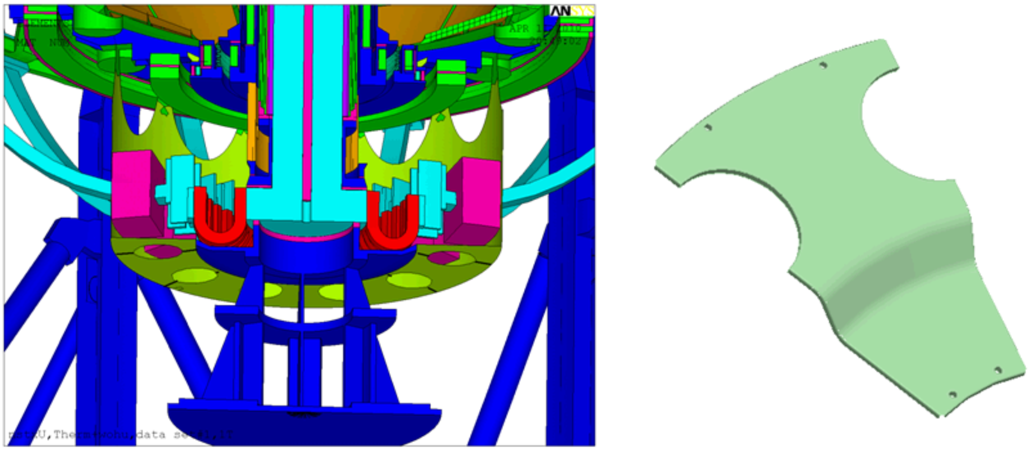


Figure 3 CDR Description of the Lid/Flex, Showing Vertical Displacement due to Centerstack Temp rise



Upper Lid/Flex Plate With Access Holes Hot Central Column, Cold Vessel

Figure 4 Upper Flex with access ports

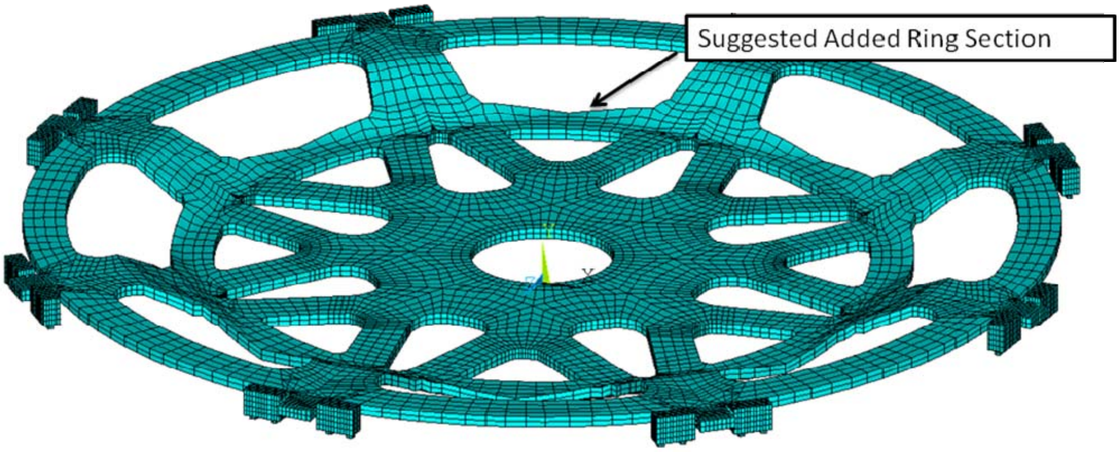


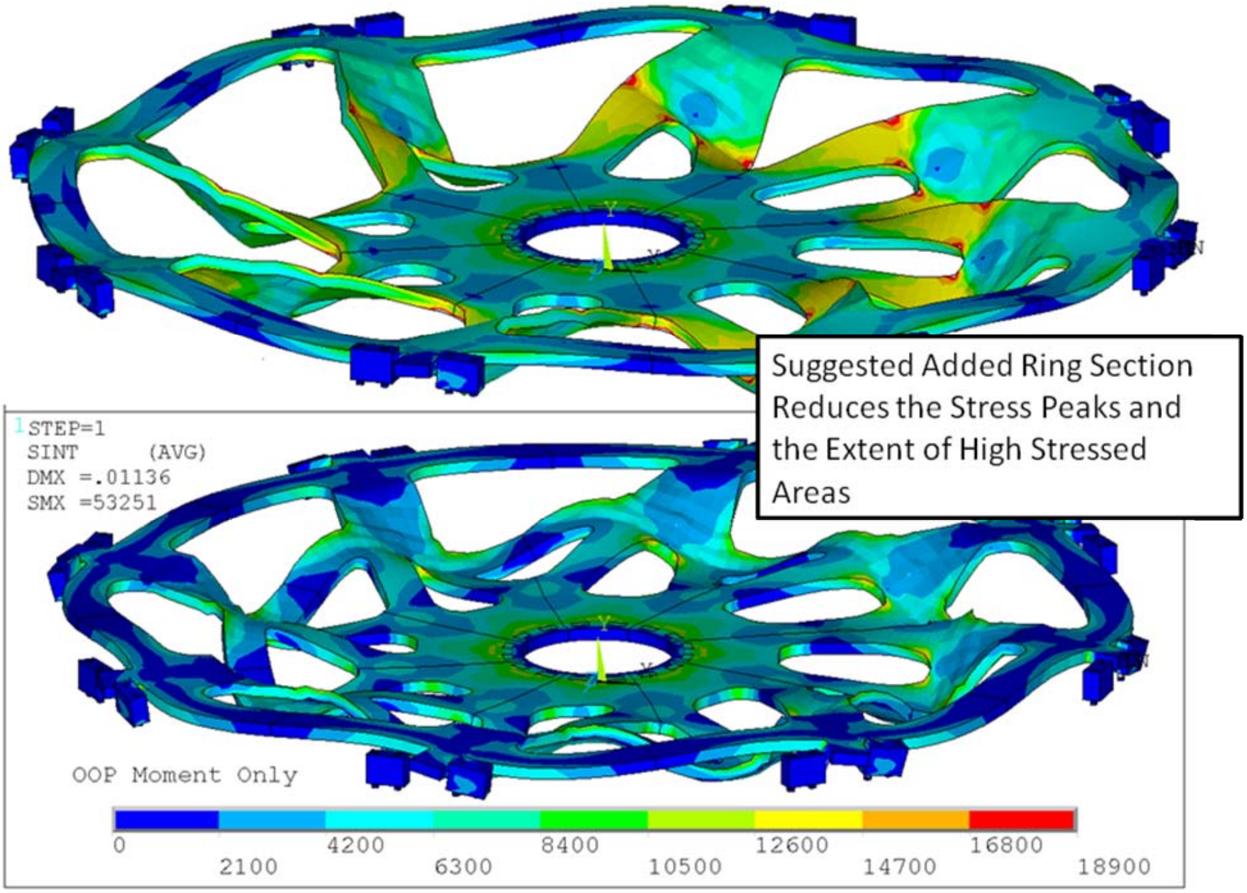
Lower Segmented Plate with Circular Ports for Access

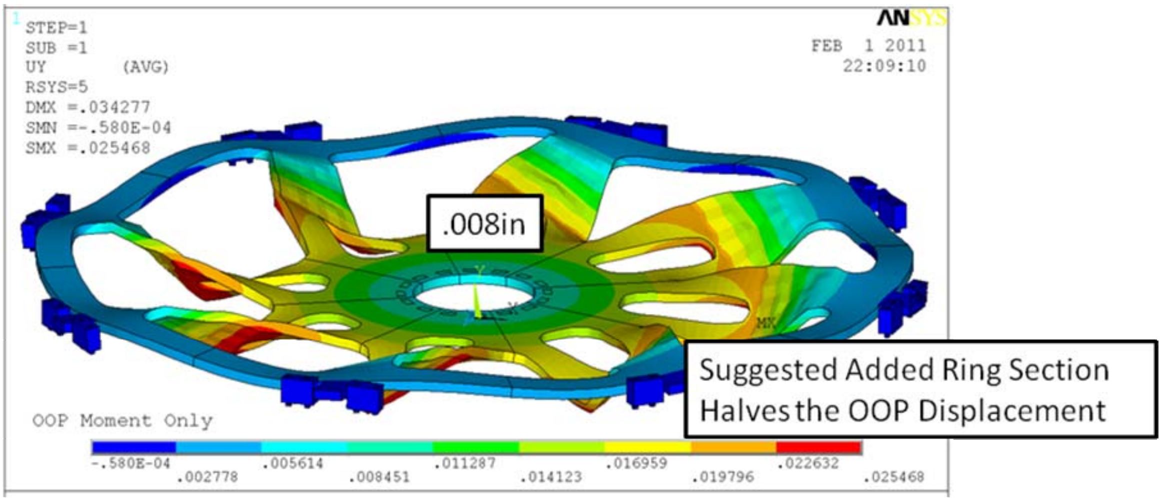
Figure 5 Lower Segmented Plate

Appendix B
Added Ring Section

The weakness of the bent spokes in the lower lid causes the torque to be shed to the pedestal, floor, and then to the vessel legs and vessel. If the lower spoked lid had to support the torque, the design would have to be stiffened substantially. In this Appendix, addition of a ring section was considered. It doubled the stiffness, and improved the stresses. The chosen design solution is to retain the torsionally compliant lid and rely on the torsional stiffness of the pedestal, floor, and vessel legs.

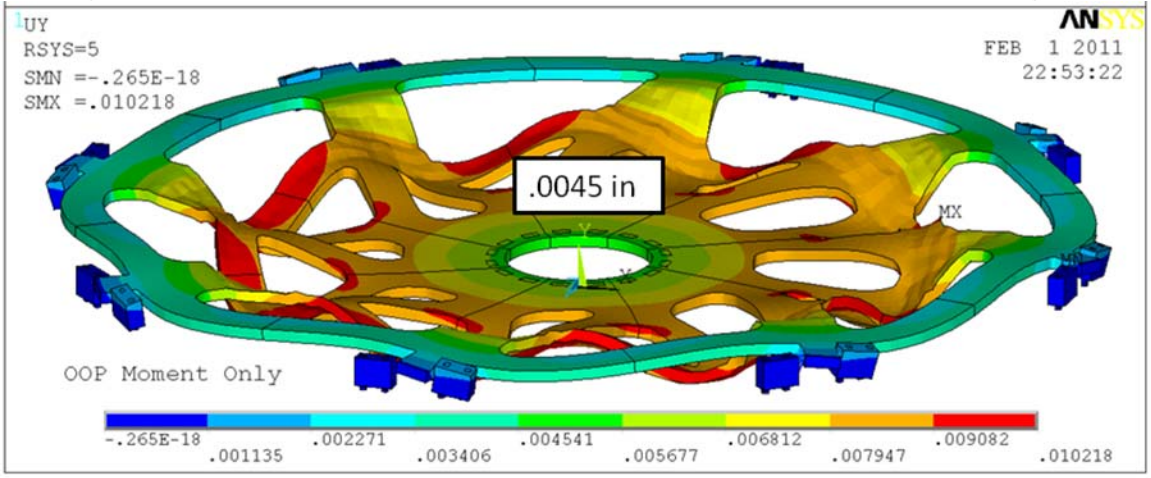

Suggested Added Ring Section


Suggested Added Ring Section Reduces the Stress Peaks and the Extent of High Stressed Areas



.008in

Suggested Added Ring Section Halves the OOP Displacement



.0045 in

Appendic C

Information on the frictional Connection between Crown and Spoked Lid

Friction Effects

Ali Zolfaghari [6] assumed 0.35 as a friction coefficient. The criteria document requires that we assume mu nominal minus 0.15. Steel on G-10 is usually assumed to be 0.3, so we will need a higher friction coefficient to qualify the joint. Phil Heitzenroeder has looked into this for NCSX. There are shim materials that we can place between the two surfaces to increase the friction coefficient. NCSX data is collected at:

http://ncsx.pppl.gov/NCSX_Engineering/Materials/index_Materials.htm

Another commercial site:

http://www.esk.com/en/products-brands/products/frictional-connection-elements/friction-enhancing-metal-shims.html

Criteria Document Content [2]:

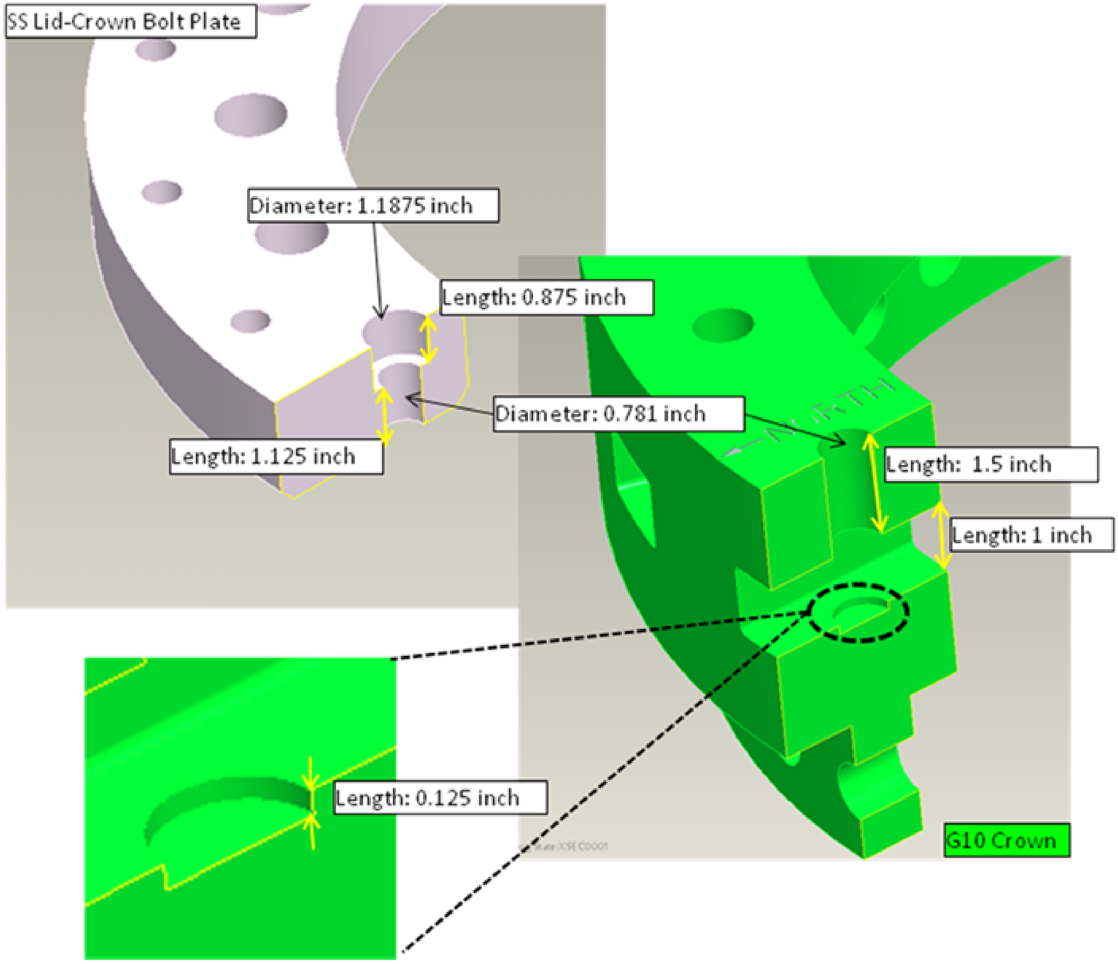
I-5.2.2 Coefficient of Friction

The allowable coefficient of friction (a) must always be determined in a conservative manner. Unlike stress, in some cases it is conservative to permit a coefficient of friction higher than the average measured value and, in some cases, lower than the measured value. The guidelines are

$$a_{min} = a - 0.15 \quad \text{but} \geq 0.02$$

$$a_{max} = a + 0.15$$

Friction values outside the range 0.1 - 0.4 require exceptional justification. The case of friction coefficient extremes must be considered as anticipated upset conditions in the design.


SS Lid-Crown Bolt Plate
Diameter: 1.1875 inch
Length: 0.875 inch
Diameter: 0.781 inch
Length: 1.125 inch
Length: 1.5 inch
Length: 1 inch
Length: 0.125 inch
G10 Crown

Appendix D Reference emails

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

Ref [10] 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