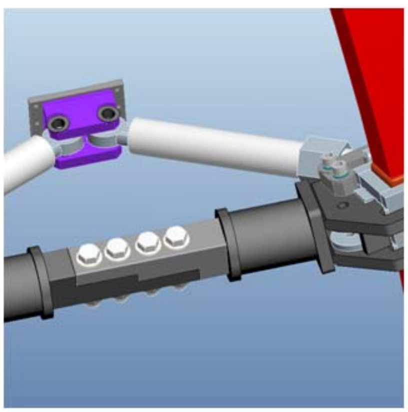
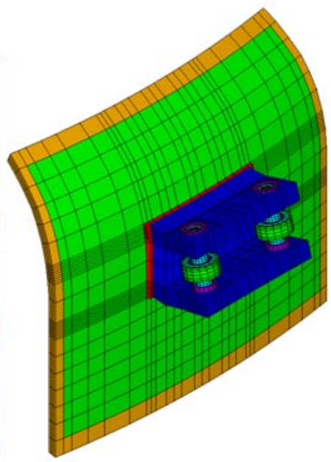
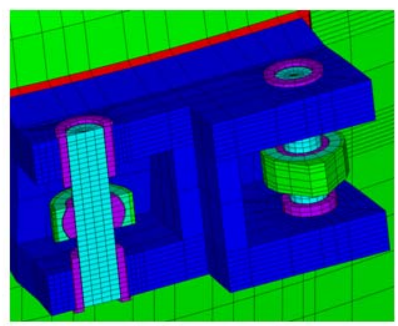
NSTX Upgrade

TF Strut to Vessel Knuckle Clevis Connection

NSTXU-CALC-132-09-00 Rev 1

July 2012

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PPPL Calculation Form

Calculation # **NSTXU-CALC-132-09-00** ___

(ENG-032)

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

To qualify the local attachment detail of TF outer leg support truss to the knuckle of the vacuum vessel

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

Included in Section 5 in the body of the calculation

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

This is a qualification of a small part of the outer leg support system. The concept for this support has gone through a number of iterations. The existing clevis attachment had a large offset to the pin centerline which produced a large prying moment in addition to the shear on the clevis. Concepts were developed that limited the load into the clevis with unwieldy soft spring concepts. These increased loads at the aluminum blocks. Concepts were developed to increase the load carrying capacity of the clevis, and reposition the centers of action of the applied loads such that the primary loading on the clevis resulted mainly in shear at the vessel surface . The last design is welded directly to the vessel wall after removal of the existing pad. This design and analysis results for this design are discussed in section 9. It has substantially increased the clevis load carrying capacity. It replaces a concept that was intended to preserve the ability to un-bolt the clevis and remove PF4 if needed. This employed extensions on either side of a bolted block that transferred shear to the vessel surface. This concept is discussed in section 19. There are many other concepts included in the calculation which illustrate the evolution of the design and the mechanics of the intersection of the truss rods that carry the TF OOP loading from the coils to the vessel shell.

Loads at the attachment varied depending on the attachment and truss concept. At the CDR it was assumed that the design load for normal operation was 20000lbs, based on CDR and PDR versions of the outer leg calculation [1]. 20000lbs at the TF clamp, became about 30,000 lbs when resolved to the vessel surface. For the FDR a rigid link is utilized that provides significant support to the outer leg at the knuckle elevation. This stiffened up the connection and as of June 2011, the design loads are 37,000lbs at the vessel surface, with 5000 lbs radial tension load due to shared TF bursting loads with the ring. A vertical load of 1403 lbs is also reported in [1]. The higher (37 kip) total clevis load which resolves to a strut load of 27 kips, has been used to size the struts, clevis and pins. When the ball end is tightly fit between the clevis plates, 3/4 inch pins meet the static allowables, but don't quite pass the fatigue allowables. In the present design , which uses 1 inch pins, there is a gap between the clevis plates and the ball end bushing. Bending of the pin is reduced by tightly fitting the pin in bushings which also must be tightly fitted into the clevis plates. The fixity at the plates reduces bending at the middle of the pin length, at the rod end. Some of the offset moment is carried by the plates by contact compressive stresses. These were excessive and resulted in the use of 718 sleeves. The block was increased in size In order to improve the stresses in the clevis plates due to the proximity of the pins and bushings to the clevis plate edge. Local stresses in the clevis plate were still above the fatigue limit. Using epoxy to bond the sleeve to the clevis plates alleviates the peak stress but stress analysis including the bond layer indicates a very good epoxy strength is needed. All this works if the pins, sleeves, and plates are tightly fitted. It was assumed the fit would survive welding of the clevises to the vessel. Initial welding of the clevises to the vessel shell did not distort the pin and sleeve fit-up.

The spherical ball ends are a catalog item and it is assumed that rod ends of an adequate rating for the fatigue loading, were chosen.

Calculation (Calculation is either documented here or attached)

See the body of the following document

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

The most recent (April 2012) welded concept has a geometry that eliminates the offset moment on the clevis. The intersection of the line of action of the two truss links is at the vessel surface. This makes the load at the vessel wall, predominantly shear (37,000 lb) with a relatively small (5000 lb) radial load. The latest (Aprilv2012) design is acceptable if the pins, sleeves, and ball ends have a tight tolerance. Even for the ideal fit, the sleeves should be bonded to the clevis plates. The peak stresses for the un-bonded installation (70 ksi Sig1 and 100ksi Tresca) around the hole would be in excess of the fatigue allowable of 40 ksi. In July 2012 the mesh was refined a bit and local stresses are higher but the character of the behavior of the analysis hasn't changed. The April version is 1/4 inch larger on three faces than the March 2012 version to alleviate "tear-out" clevis stresses at the hole for the pin and bushing. The model was run with an epoxy layer bonding the sleeve to the clevis plate. Most of the epoxy sees less than 7ksi tension. There are local small spikes of 20 ksi, and an edge that will probably crush from the compression. Shears are 5 to 20ksi The clevis plate stresses are acceptable with the bonded sleeve. Given the uncertainty in the epoxy performance, the clevis holes should be on the fatigue inspection list.

. The new weld is passes static criteria. A three sided weld was analyzed because PF4 interferes with making the weld at the bottom of the (upper) clevis. The recommended weld is a 3/8 groove backed with a 3/8 fillet. Fatigue evaluations were acceptable based on a uniform distribution of stress in the weld perimeter. This geometry is similar to the PF4/5 support pad and has higher stresses at the corners of the rectangular pads. Consequently these should be added to the inspection list.

2 inch OD Rods or 2 inch sch 160 pipes are acceptable to take the compressive load in the struts without buckling . There is one area where the clevis and the vessel support I Beam support bracket interfered. A "special" bent strut was investigated. Solid bars with the same OD as the 2 inch pipe was tried and did not pass. An even heavier section was needed. Instead the straight struts are retained and special clevis and "chair" vessel supports, with appropriate clearances, are used. The spherical ball ends have been specified. These will have threads exposed to the cyclic stresses and have been designed to have large thread diameters at the ID of the struts to reduce the cyclic stress. In section 11, pin fit-up was studied. zero clearance in the pin fit supported the pin adequately to reduce the mid span moment in the pin. Much of the moment support at the ends of the pin was lost with a .003 inch diametral clearance. This implies press fit for the 718 sleeves after a final reamed alignment of the clevis holes. and a press fit for the pin in both the sleeves. .0005 inch diametral Interference fit of the bushing improves the stress modestly, but this is not planned. Instead a 3M weld

The concept with added extension pieces welded to the vessel shell -shown in section 19 of the calculations is also adequate to accept the normal operating scenario loads.]

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

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Table of Contents

Revision Status

Revision	date	description
Original Issue, Rev 0	1-2011	
Rev 1	March 2012	Added Multiple TF Clevis Concept Changes
Rev 1		Added March 2012 Clevis
Rev1	Aug 2012	Added Bonded and Un-Bonded Sleeve Analyses

3.0 Executive Summary:

This is a qualification of a small part of the outer leg support system. The concept for this support has gone through a number of iterations. The chosen attachment has been sized and shaped to accept only shear loading and a relatively small tensile loading, and has been found acceptable for expected OOP loads that will be imposed on the vessel knuckle region by the TF outer leg support truss.

The most recent (April 2012) welded concept is acceptable if the pins, sleeves, and ball ends have a tight tolerance. Even for the ideal fit, an attempt should be made to bond the sleeves to the clevis plates. The peak stresses (70 ksi Sig1 and 100ksi Tresca) around the hole are in excess of the fatigue allowable of 40 ksi. In July 2012 the mesh was refined a bit and local stresses are higher but the character of the behavior of the analysis hasn't changed. The April version is slightly larger than the March 2012 version to alleviate "tear-out" clevis stresses at the hole for the pin and bushing. The model was run with an epoxy layer bonding the sleeve to the clevis plate. Most of the epoxy sees less than 7ksi tension. There are local small spikes of 20 ksi, and an edge that will probably crush from the compression. Shears are 5 to 20ksi The clevis plate stresses are 34.8 ksi with the bonded sleeve, below the 40 ksi fatigue limit. Given the uncertainty in the epoxy performance, the clevis holes should be on the fatigue inspection list.

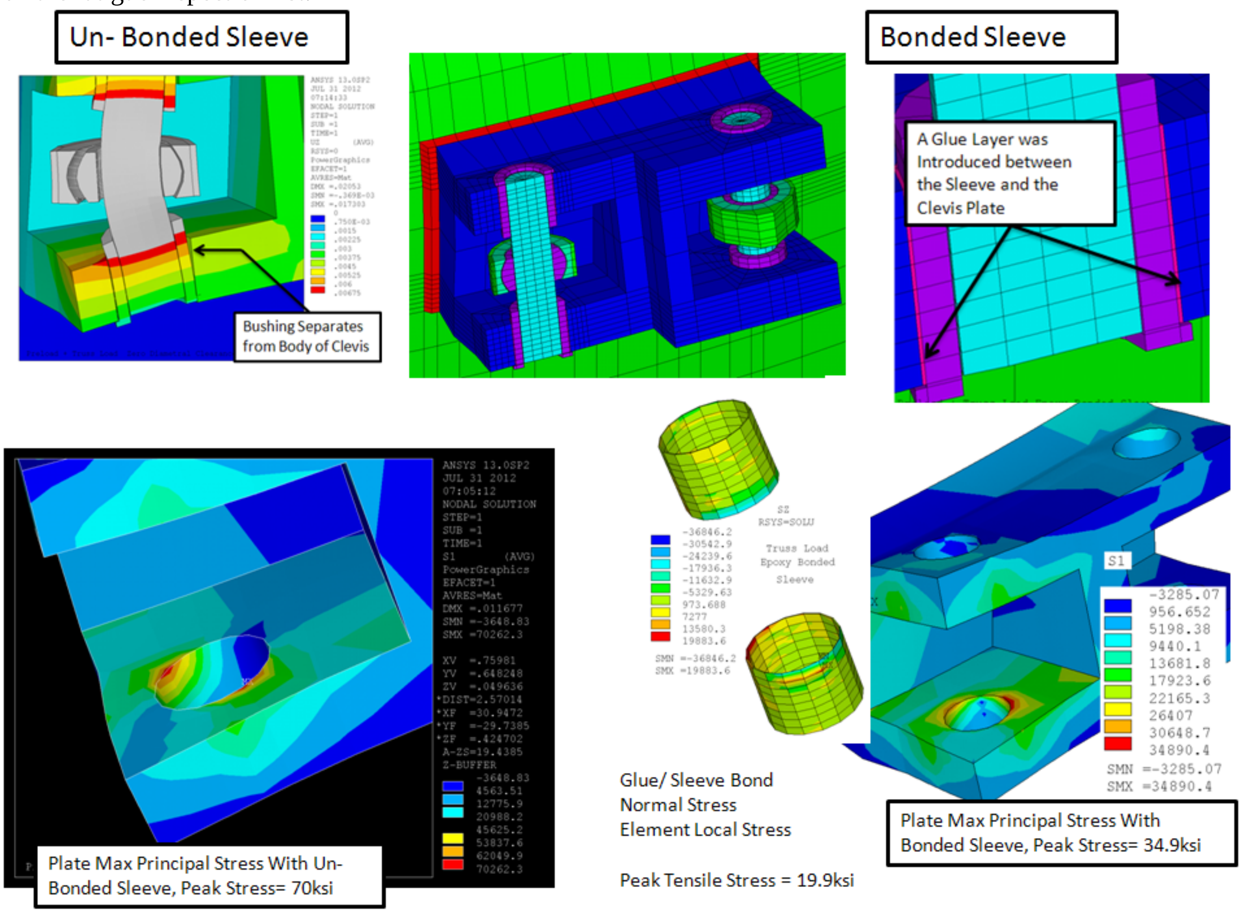


Figure 3.0-1 April 2012 Design, July Analysis Bonded and U-Bonded Sleeve

Final designs were governed by the intent to intersect the lines of action of the truss loads at the surface of the vessel to avoid applying a moment on the weld pattern. This required an interplay between the toroidal separation and the radial location of the pin centerlines. The April 2012 design minimizes the moments applied to the weld pattern., Pin bending stress has been a critical element in the design of the clevis. Alignment of the rods was expected to require some vertical position adjustment of the centerlines of the ball ends. This increased the span of the pin. Simple assessments of 3 point support of the pin produced excessive stresses. Fit-up between the pin and the ball end bushing and clevis were intended to reduce the effective span and add fixity to the ends of the pin causing the bending moment in the center to be reduced. This required a tight fit between the pin and clevis which in turn added bearing stress to the clevis - which was then fixed by adding a 718 insert which in-turn reduced the clevis edge distance from the hole to the metal edge. - This in-turn resulted in addition of .25 inches of material on three sides of the clevis. The added material improved the stress but it still did not satisfy fatigue allowables. An interference fit improves the stress but it still does not pass the fatigue allowable. Use of the 3M weld bond adhesive does not increase the capacity sufficiently. The 2000 psi shear capacity adhesive is not adequate to support the 22,000 lbs tension in the strut. It will de-bond.

Weld stresses are acceptable in terms of static and fatigue allowables, but inspections of the welds at the corners of the square pad are recommended. A three sided weld was analyzed because PF4 interferes with making the weld at the bottom of the (upper) clevis. The recommended weld is a 3/8 groove backed with a 3/8 fillet.

The existing clevis attachment had a large offset to the pin centerline which produced a large prying moment in addition to the shear on the clevis. Concepts were developed that limited the load into the clevis, and concepts were developed to increase the load carrying capacity of the clevis. Loads at the attachment varied depending on the attachment and truss concept.

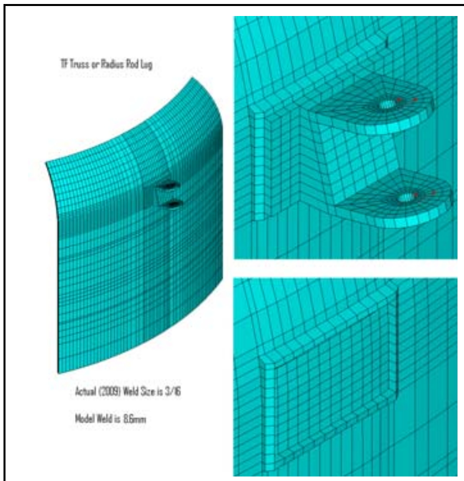


Figure 3.0-2 Existing Clevis Details

The existing clevis attachment bolting and 3/16 fillet welds are insufficient to support the upgrade truss/radius rod loads with the offset the present clevis design imposes. . Welding the bolted clevis to the pad and increasing the weld size to 3/8 inch meets the static stress limits. Further analysis and possible re-enforcement was needed to satisfy fatigue limits. Once welding was considered, improvements in the clevis were also considered. One concern is that the existing bolts will gall when attempts are made to remove them. This is not expected (based on conversations with Eric Perry) but if they do gall than they can be ground off and the welded clevis welded over the bolts. In addition to the welded concept, other concepts is evaluated here beginning on section 12. These discussions are retained as back-ups in case access or interferences make welding difficult, and to illustrate the design evolution and the mechanics that contributed to the design evolution.

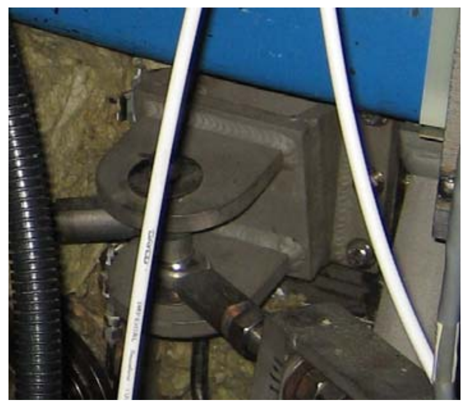


Figure 3.0-3 Photo of one of the Existing Clevises (as of 2011)

In the appendices, some of the calculations and presentation material are included to provide an understanding of the history that led to the present design choice. The weakness of the existing clevis produced a variety of design solutions that were more difficult and were not chosen. Prior to the CDR a diamond truss assembly was investigated, but only worked for up-down symmetric OOP loads and was impossible to install around the existing diagnostics, wave guides and service lines. At the PDR, a solution that employed compliant trusses to limit loading into the clevis was presented.. This design used first, a coiled spring and then a Belleville spring stack. Off-loading the OOP loading from the vessel was thought necessary to limit stresses at the mid-plane port ligaments.

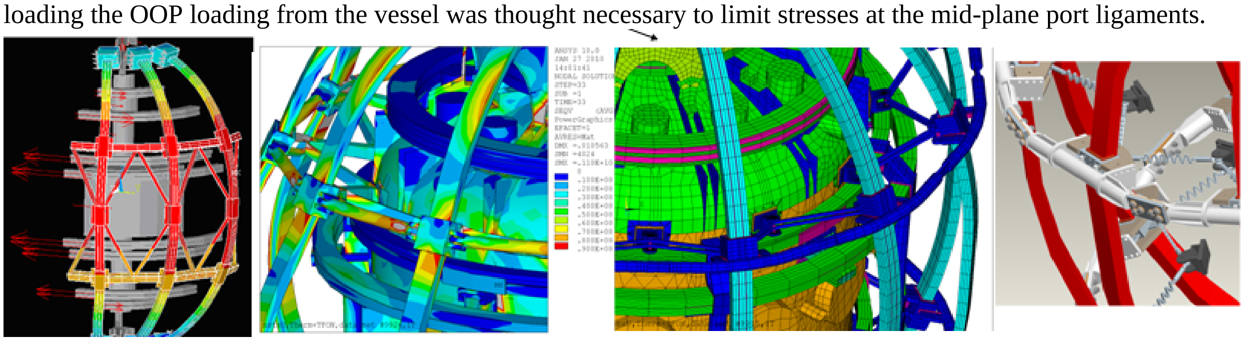

Diamond Truss Pinned Ring Rigid Truss Rigid Ring to Existing Clevis Soft Springs to Existing Clevis

Figure 3.0-4 Early TF Outer Leg Support Concepts

However more detailed analysis showed adequate capacity at the equatorial plane and the spring truss was dropped. Options that used the existing clevis pads as shear keys - with no tensile capacity were judged to have a precarious purchase on the pad, and this concept was never considered seriously. A concept which converted the PF 4 and 5 support to take the TF OOP load was also considered and dropped. Some of the evaluations of this are included in Appendix B.

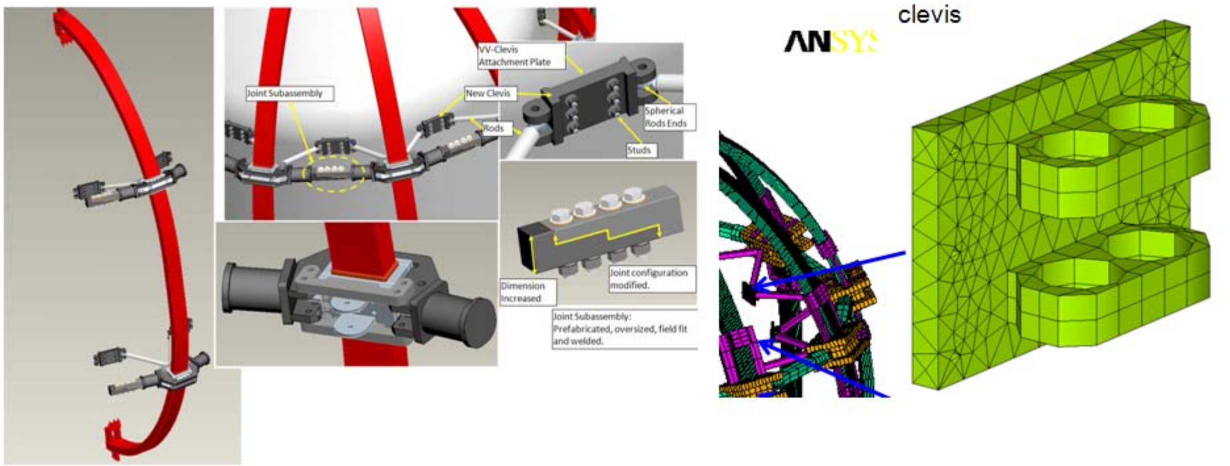


Figure 3.0-5 FDR TF Outer Leg Support

An early version of the knuckle clevis is shown in the middle. Truss loads imposed a moment on this concept because of the width of separation to the "ears". The modeling employed in Ref 1 is shown at right. Preliminary results from this analysis show a truss shear load of 75 KN or 17,000 lbs Just based on the distribution of OOP loads in the upper outer leg of the TF the load should be around 20,000 lbs at the TF clamp. With the 10% headroom, it becomes 22,000 lbs, and resolving it from the ring radius to the vessel shell increases it to 30,000 lbs. Estimates of this load later went up to 37,000lbs[1]. As in a truss, the diagonal struts should be alternating between tension and compression. The strut loads should just be the shear load divided by 2* cos(truss angle). There is a smaller (~5000lb) radial load superimposed on the strut alternating tensions and compressions. This increases the rod tension to 27,000 lbs and this is the load used to size the strut spherical ball end, and clevis. . 2 inch sch 80 pipes are needed to take the compressive load in the struts. "Special" bent struts were investigated to clear vessel support brackets/chairs. Solid bars with the same OD as the pipes were tried and did not pass. Instead, special vessel support brackets/chairs were designed.[10]

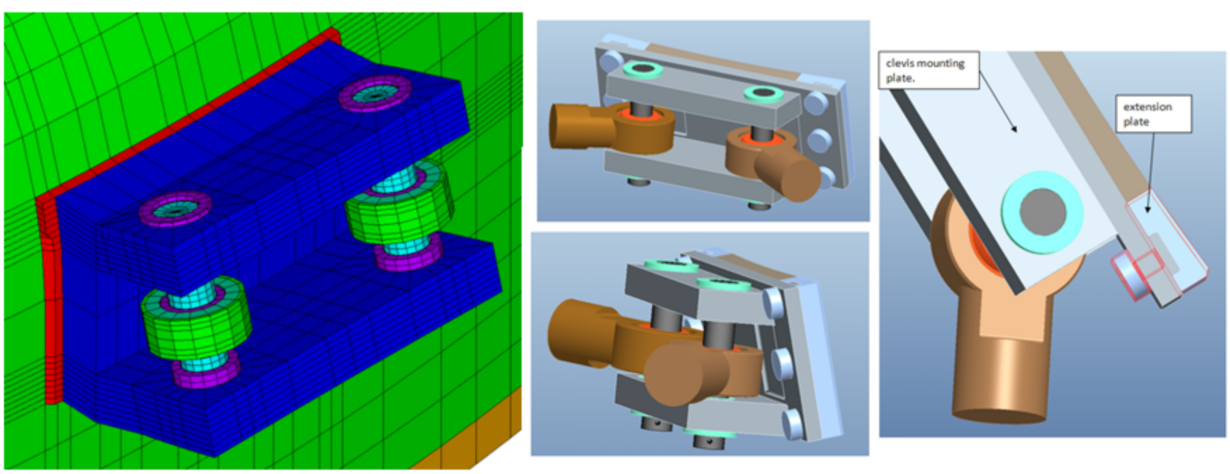


Figure 3.0-4 Clevis Details in the Late 2011(Right) and March 2012 Design (left)

4.0 DCPS Algorithm

As required for input to the machine simulator described in the DCPS Requirements Document [9], The DCPS algorithms will be supplied for loading in the calculation for the outer TF support structures, ref [1]. A simplified approach for the clevis would be to scale the loads from the OOP torque computed in the design point spreadsheet. This is the upper half outer leg torque from spreadsheet - based on the equation in ref [6]. The shear load limit at this writing is 37,000 lbs. Derived from Scenario #79. The reported stresses can be scaled by the calculated torque for the currents being checked by the DCPS divided by the torque for equilibrium # 79 Charlie's revision or new version of the DCPS requirements document[12] has some important changes. The planned disruption and shut-down look-aheads, have been removed, and the effect of passive structures has been ignored. I talked with Charlie about the TF outer leg summations in the spreadsheet. As of March 7 2012, Charlie had not updated the TF torque sums for the disruption currents. He provided the new torque values in March 7 2012. The disruption torque is lower than the normal outer leg torque. -See the discussion in Appendix G, Ref [11]. The DCPS stress multipliers may remain scaled based on the TF outer leg upper half torque divided by the EQ 79 torque. There is no fatigue margin in the clevis pin, so the OOP torque must be maintained below the EQ 79 value - or fatigue cycle counting must be implemented.

5.0 References

[1] Analysis of TF Outer Leg, Han Zhang, Calculation Number NSTXU-CALC-132-04, and Preliminary Results shown in the March 15 NSTX progress meeting

[2] NSTX-CALC-13-001-00 Rev 1 Global Model – Model Description, Mesh Generation, Results, Peter H. Titus March 2011

[3] Analysis of Existing and Upgrade PF4/5 Coils and Supports – With Alternating Columns. NSTX-CALC-12-05-00 Rev 0 P. Titus March 2011

[4] NSTX Structural Design Criteria Document, NSTX_DesCrit_IZ_080103.doc I. Zatz

[5] NSTX Design Point Sept 8 2009 http://www.pppl.gov/~neumeyer/NSTX_CSU/Design_Point.html

[6] OOP PF/TF Torques on TF , R. Woolley, NSTXU CALC 132-03-00

[7] NSTX TF Outer Leg Clamp Pin Assembly NSTX-CALC-132-12 Rev 0 November 2011 Peter Rogoff,

[8] National Spherical Torus Experiment NSTX CENTER STACK UPGRADE GENERAL REQUIREMENTS DOCUMENT NSTX_CSU-RQMTS-GRD Revision 4 September 15, 2011

[9] DIGITAL COIL PROTECTION SYSTEM (DCPS) REQUIREMENTS DOCUMENT (DRAFT), NSTX-CSU-RD-DCPS for the National Spherical Torus Experiment Center Stack Upgrade, February 5, 2010 R. Woolley

[10] NSTX-U CALC 12-10-00 "Redesigned Vessel Support Bracket", Peter Rogoff, March 2012

[11] March 7 email from C. Neumeyer with Post Disruption Torque Additions to the Design Point Spreadsheet Appendix G

[12] National Spherical Torus Experiment NSTX CENTER STACK UPGRADE, Coil Protection System Requirements Document Revision 0 February 1, 2012 Charles Neumeyer

6.0 Input

6.1 Criteria

From the Criteria Document, Ref 4:

• When considering bearing stresses in pins and similar members, the Sy value at temperature is applicable, except that a value of 1.5 Sy may be used if no credit is given to bearing area within one pin diameter from a plate edge.

• The average primary shear stress across a section loaded under design conditions in pure shear (e.g., keys, shear rings, screw threads) shall be limited to 0.6 Sm. The maximum primary shear under design considerations, exclusive of stress concentration at the periphery of a solid circular section in torsion, shall be

Coil and structural criteria are outlined in "NSTX Structural Design Criteria Document", Zatz[2].
Fatigue requirements are based on the Rev 4 GRD, recently revised in September 2011 [13]. The pertinent section is excerpted below.

b. Number of Pulses

For engineering purposes, the number of NSTX pulses, after implementing the Center Stack Upgrade, shall be assumed to consist of a total of 20,000 pulses based on the pulse spectrum given in Table 2-4 which allows for pulsing at various duty cycles coordinated per section 2.4 a.

Table 2-4 - NSTX CSU Pulse Spectrum

Performance	60%	75%	90%	100%	
B_t	0.6	0.75	0.9	1	T
I_p	1.2	1.5	1.8	2	MA
$T_{pulse}=T_{flat-Ip}$ (sec)					Total pulses
3	200	1800	1200	1000	4200
3.5	200	1800	1200	1000	4200
4	200	1800	1200	1000	4200
4.5	200	1800	1200	500	3700
5	200	1800	1200	500	3700
				Total	20000

Figure 6.1-1 Snapshot of the Rev 4 General Requirements Document [6]

With a factor of 20 on life, this would require a life of 4e5 (400,000) in a SN evaluation.

6.2 Drawing Excerpts (Existing Design)

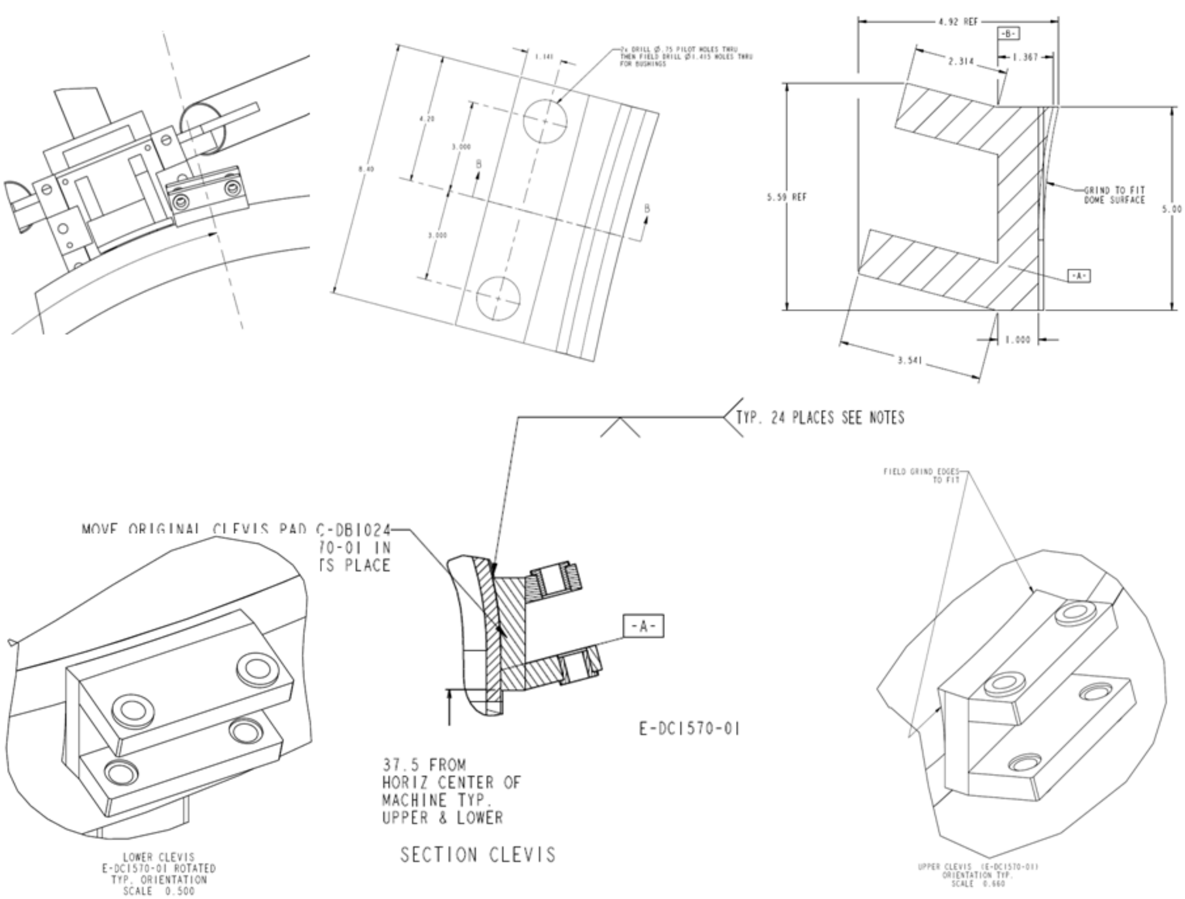


Figure 6.2-2 Clevis Details March 2012

1/4 inch was added to three sides of the clevis plates in April 2012 to improve the pin hole edge distance.

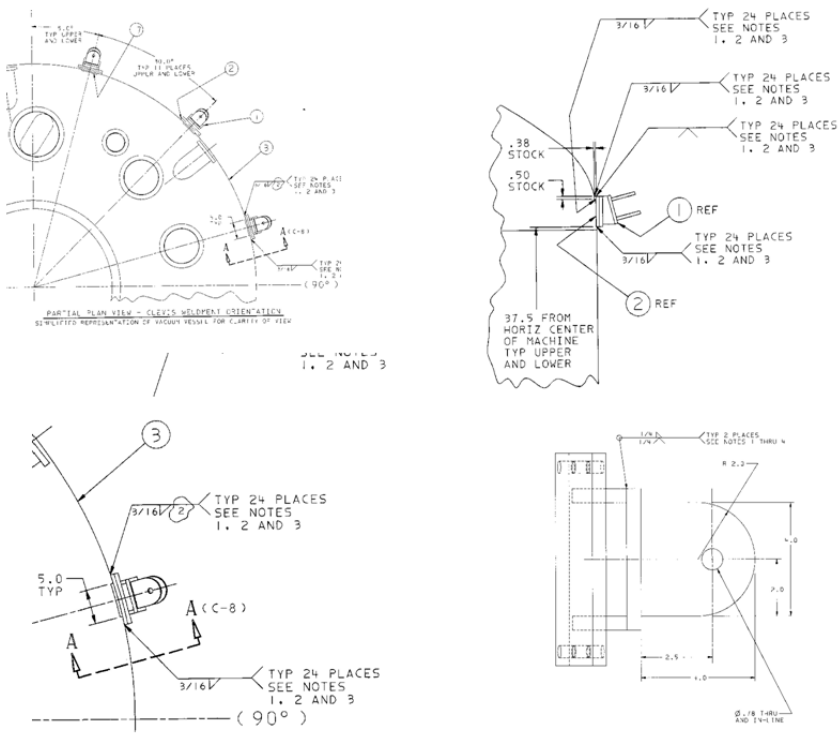


Figure 6.2-2 Clevis Details for the Original NSTX Clevis

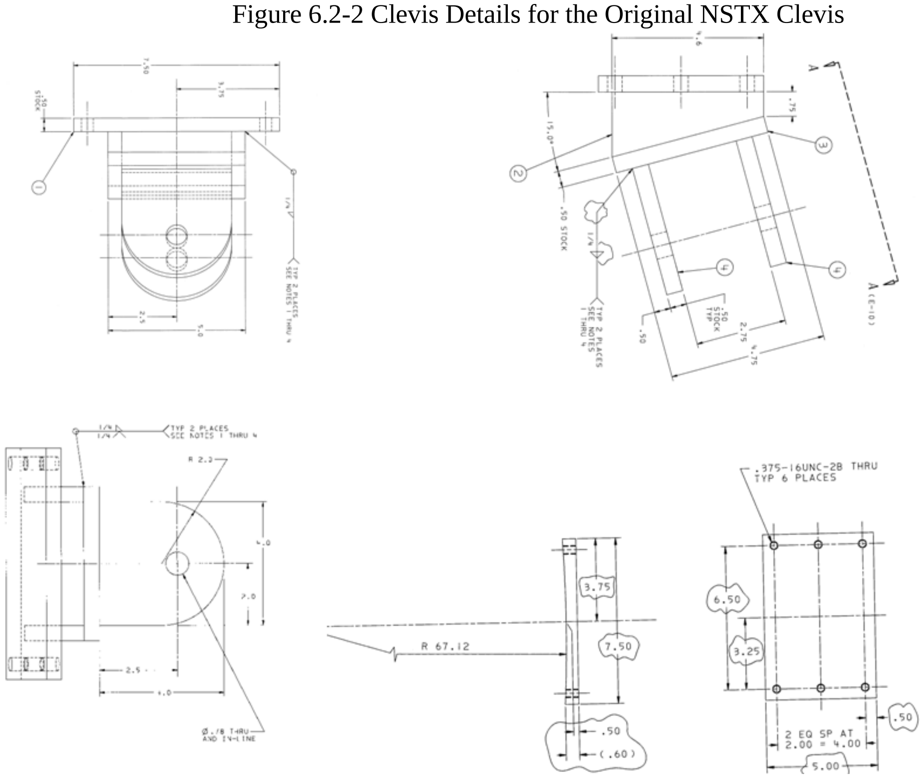


Figure 6.2-3 Clevis Details for the Original NSTX Clevis

Note:
(1) VV Clevis PAD is not welded all around.
(2) Large gap between clevis and vessel.
(3) Weld region (underneath pad) cannot be inspected. This is a source of fatigue failure.
(4) Welds need to be rated accordingly.

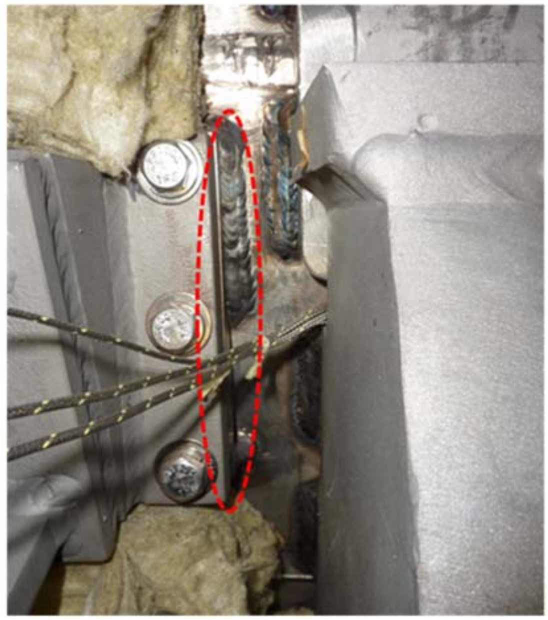
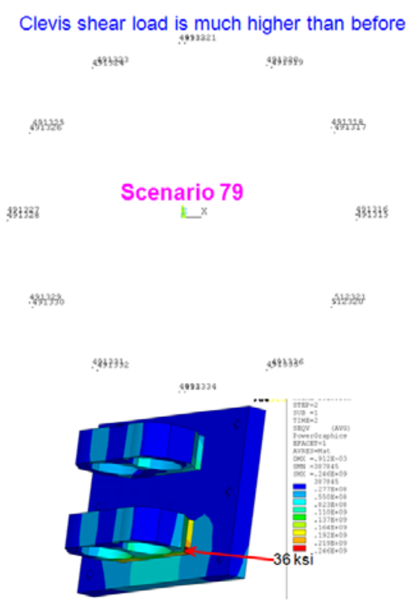


Figure 6.2-4 Clevis Details for the Original NSTX Clevis Where it Interferes with the Vessel I Beam Support Column "Chair" or bracket

7.0 Loads

The OOP Load at the TF Outer Leg is some fraction of the net TF outer Leg OOP load. For scenarios that produce up-down symmetric loading, the upper half of the outer leg sum is split between the umbrella structure and the TF clevis which is the subject of this calculation. Loads utilized in this analysis come from H. Zhang's analysis of the outer legs structural support, ref [1]



Cylindrical coordinate system

NODE	FX	FY	FZ
491315	-45966.	68168.	-11996.
491316	63083.	92613.	16418.
491317	-47387.	70275.	-12367.
491318	61972.	90982.	16129.
491319	-47256.	70081.	-12332.
491320	61821.	90760.	16089.
491321	-47223.	70033.	-12324.
491322	60156.	88316.	15656.
491323	-45109.	66897.	-11772.
491324	61028.	89596.	15883.
491325	-44169.	65503.	-11527.
491326	62344.	91528.	16226.
491327	-46104.	68373.	-12032.
491328	59583.	87474.	15507.
491329	-41922.	62171.	-10940.
491330	66018.	96921.	17182.
491331	-50283.	74571.	-13122.
491332	60398.	88671.	15719.
491333	-50770.	75292.	-13249.
491334	55240.	81099.	14377.
491335	-43581.	64632.	-11373.
491336	60829.	89304.	15831.
512320	-44957.	66671.	-11732.
512321	61226.	89886.	15935.

Max shear load: 163KN (previous requirement 5000 lbs=22 KN)
Max radial load: 24096 N (5428 lbs)
Max vertical load: 6242 N (1406 lbs)

Figure 7.0-1 Loads from Ref [1]

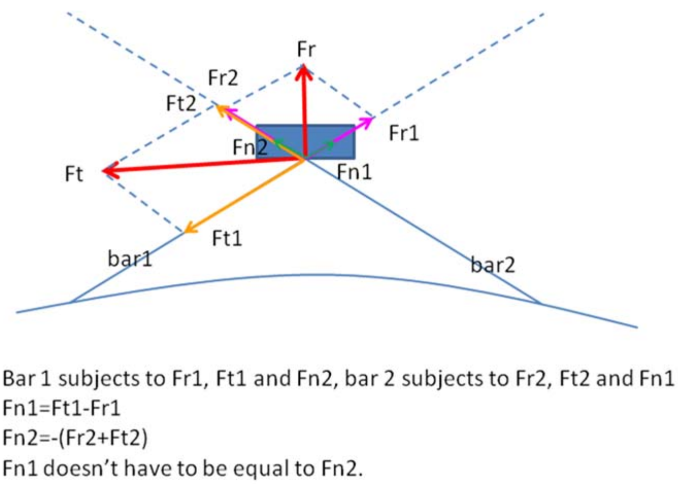


Figure 7.0-2 Rod Loads Provided by Han Zhang from Ref [1] -Including the Radial Component

Currently (June 2011) the design loads is 37,000lbs at the vessel surface, with 5000 lbs radial tension load due to shared TF bursting loads with the ring. A vertical load of 1403 lbs is also reported in [1] The higher (37 kip) load has been used to size the struts, clevis and pins.

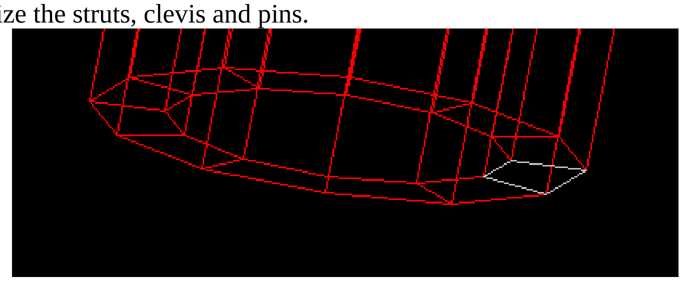


Figure 7.0-3 Area of the Strut in the Global Model

From the Strut modeling in [2], Run34, tarea = 1.1e-4 for 1/12 of the strut area

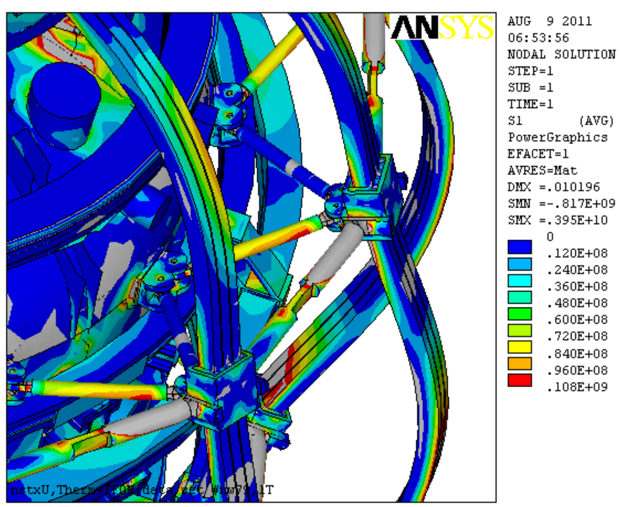


Figure 7.0-4 StrutMax Principal Stress from the Global Model[2]

From [2], run34 The Tensile Strut load is:1.1e-4*12*84e6*.2248 = 24925 lbs

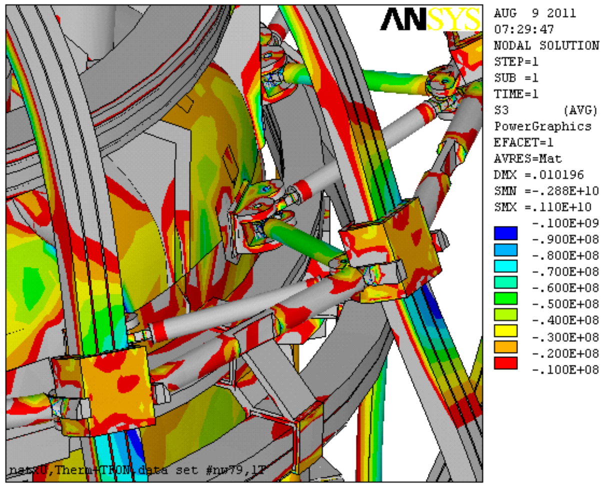


Figure 7.0-4 Strut Min Principal Stress from the Global Model[2]

From [2], run34 The Compressive Strut load is:1.1e-4*12*-60e6*.2248 = 17804 lbs. These calculations are presented as a check of the ref [1] loads which are larger and are used in this calculation to qualify the clevis.

The loads utilized for this calculation are based on the equilibria in the design point spreadsheet. These have been updated based on a new DCPS document [11] that produced a new set of post disruption currents and a new net TF outerleg torque. The outer led disruption torque is less than the normal operating torque (See C. Newmeyer's March 7 email in Appendix G, Ref 10)

8.0 Materials and Allowables

718 Typical Mechanical Properties At Room Temperature:

Ultimate Tensile Strength	Yield Strength (0.2 % offset)	Elongation in 50 mm (2")	Elastic Modulus (Tension)
MPa ksi	MPa ksi	%	GPa 106 psi
1240 180	1036 150	12	211 30.6

1/3 Ult=60ksi 2/3 yield=100 ksi Sm=60ksi, Bending Allowable = 90 ksi

The allowed shear stress is .6*sm = 36 ksi

Design Life = 20,000 Full Power Pulses, With a factor of 20, The requirement is 400,000 cycles which yields a 95 ksi allowable
At 20,000 cycles the criteria based on 2*stress yields 160/2 = 80 ksi

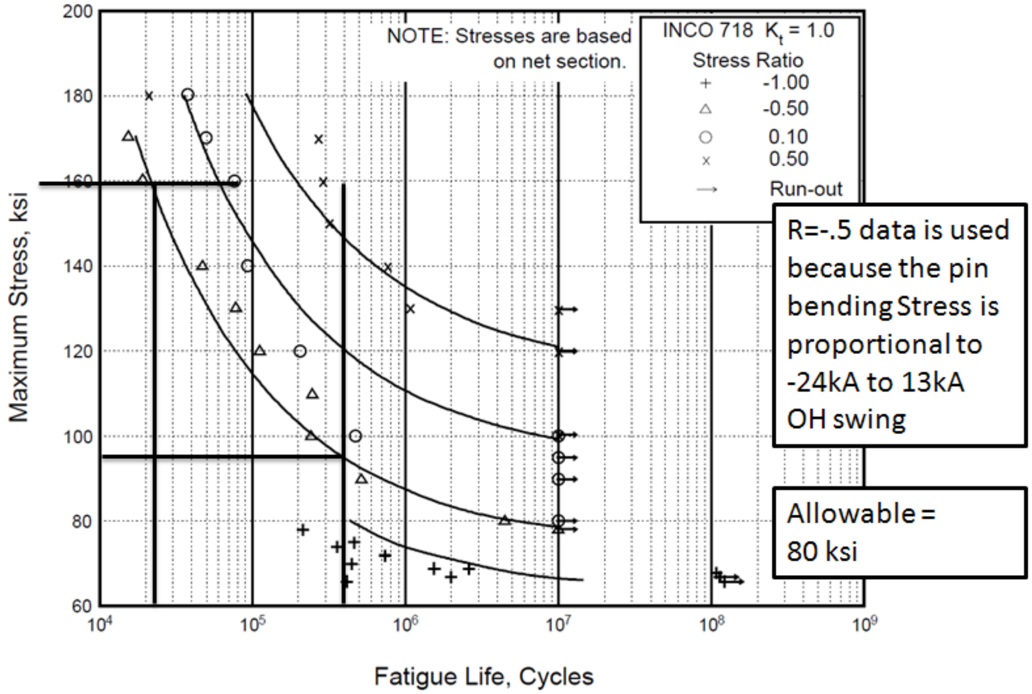


Figure 6.3.5.1.8(f). Best-fit S/N curves for unnotched Inconel 718 bar and plate at room temperature, longitudinal direction.

Figure 8.0-1 Best Fit S/N curves for unnotched Inconel 718 bar and plate at room temperature, longitudinal direction

ASTM A193 Bolt Specs from Portland Bolt.com

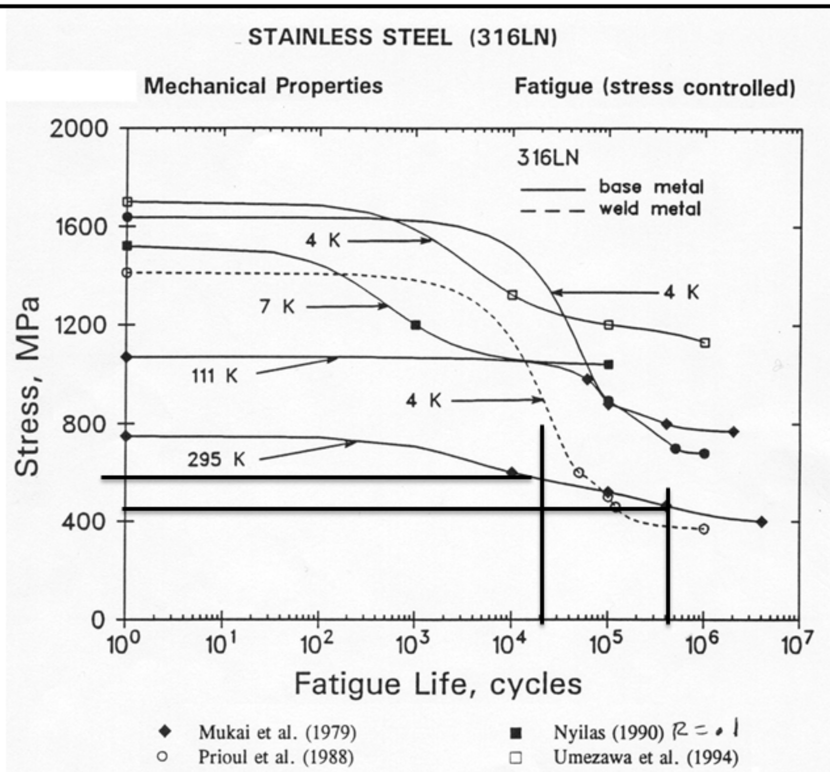
B8M	Class 1 Stainless steel, AISI 316, carbide solution treated.
B8	Class 2 Stainless steel, AISI 304, carbide solution treated, strain hardened
B8M	Class 2 Stainless steel, AISI 316, carbide solution treated, strain hardened

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

Grade	Size	Tensile ksi, min	Yield, ksi, min	Elong, %, min	RA % min
	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

Design Life = 20,000 Full Power Pulses, With a factor of 20, The requirement is 400,000 cycles which yields a 420 MPa =60.9ksi
At 20,000 cycles the criteria based on 2*stress yields 550 MPa/2 = 275MPa = 40 ksi



R=.1 data is used because the Contact related stress in the plate is not reversed

Allowable = 40 ksi

Figure 8.0-2 SN Curves for 316 Stainless Steel

Weld Allowable

From the NSTX Criteria:

For welds in steel, the design Tresca stress shall be the lesser of:

2/3 of the ***minimum*** specified yield if the weld at temperature, or

1/3 of the ***minimum*** specified tensile strength of the weld at temperature.

From the AISC Criteria:

Reference and Weld	Rod or weld wire	Parent Material	Allowable Stress (Exclusive of Weld Efficiency)
AISC Stress on cross section of full penetration Welds		All	Same as Base material
AISC Shear Stress on Effective Throat of fillet weld	AWS A5.1 E60XX	A36 -	21 ksi

For shear on an effective throat of a fillet, For 304 Stainless, the weld metal is annealed, or the base metal in the heat effected zone is annealed. and Estimate 241*21/36 = 140 MPa = 20 ksi (without weld efficiency)
This is consistent with NSTX Criteria of 2/3 yield or 2/3 of 30ksi for annealed 304
With a weld efficiency of .7 the allowable is 14ksi, or 96 MPa
For fillets divide weld area by sqrt(2)

Figure 8.0-3 Static Weld Allowable

American Welding Society | Structural Welding Code—Stainless Steel | AWS D1.6/D1.6M:2007 An American National Standard

Table 2.1
Allowable Stresses (see 2.3.2)

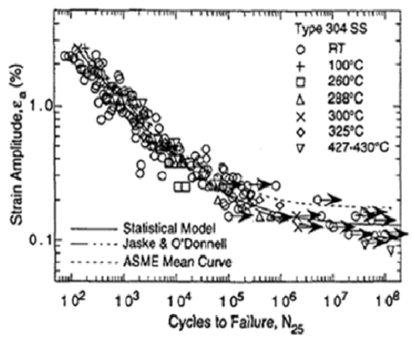
Stress in Weld		Allowable Stress[a,b,c,d]
CJP Groove Welds		
Tension normal to the effective area		The lesser of values for base metal or filler metal.
Compression normal to the effective area		The lesser of values for base metal or filler metal.
Tension or compression parallel to the axis of the weld		Same as for base metal.
Shear on the effective area		0.30 × nominal tensile strength of filler metal, except shear stress on base metal shall not exceed 0.40 × yield strength of base metal
PJP Groove Welds		
Tension normal to the effective area		0.30 × nominal tensile strength of filler metal, except tensile stress on base metal shall not exceed 0.60 × yield strength of base metal
Compression normal to the effective area	Joint not designed to bear	0.5 × nominal tensile strength of weld metal, except compression stress on adjacent base metal shall not exceed 0.60 × yield strength of base metal
	Joint designed to bear	The lesser of values for base metal or filler metal
Tension or compression parallel to the axis of the weld		Same as for base metal
Shear parallel to the axis of the weld		0.30 × nominal tensile strength of filler metal, except shear stress on base metal shall not exceed 0.40 × yield strength of base metal
Fillet Welds		
Shear on effective area of weld		0.30 × nominal tensile strength of filler metal, except shear stress on base metal shall not exceed 0.40 × yield strength of base metal
Tension or compression parallel to the axis of the weld		Same as for base metal

Figure 8.0-3b Static Weld Allowable

For a fillet the allowable stress, according to AWS, would be .4*30,000 12,000 psi on the effective throat area, assuming the annealed property of the vessel in the heat effected zone is 30,000 psi

Fatigue:
for a nominal 60,000 cycles, the strain range allowable is ~.175%
For 20 on life, or 1200,000 cycles, the strain range is .15%

Strain Amplitude = 109/200000 = .05%
For 2 on stress or 20 on life the strain allowable is .00175/2 or fro a modulus of 200e9 the allowed stress is 175 Mpa. For a stress concentration of 4, the allowed nominal weld stress is 43.75 Mpa = 6345psi



From Tom Willard's Collection of SST Fatigue Data
"Estimation of Fatigue Strain-Life Curves for Austenitic in Light Water Reactor Environments Stainless Steels", Argonne Nat. Lab. 1998

Figure 8.0-4 Weld Fatigue Allowable

If the FEA modeling represents the local weld stress concentration, well, the fatigue allowable is 175 MPa (25.4ksi), and for simple line load calculations the allowable is 6345 psi to allow for a concentration factor of 4.

9.0 Welded Clevis

9.1 April 2012 Welded Clevis

The February 2012 version of the welded block had "tear-out" stresses around the pin hole of 66 ksi - See section 9.2. These were improved by addition of 1/4 inch of material at both net sections of the plate eye.

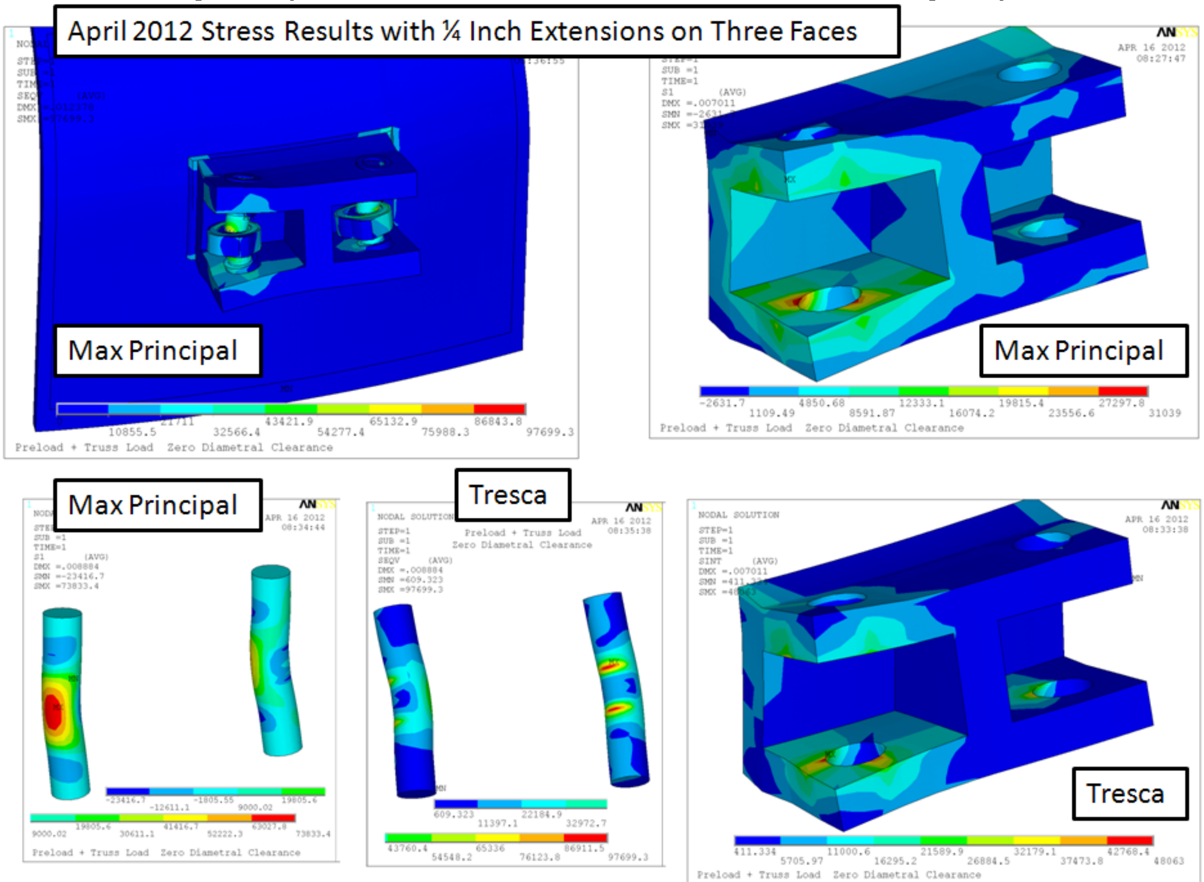


Figure 9.1-1 Clevis Plate Stress, April 2012 Geometry

A case with a .0005 inch diametral interference between the bushing and clevis was run. It reduced the peak operating stress in the clevis from 48ksi to 42.2 ksi and reduced the magnitude of the stress range to 36.8 ksi.

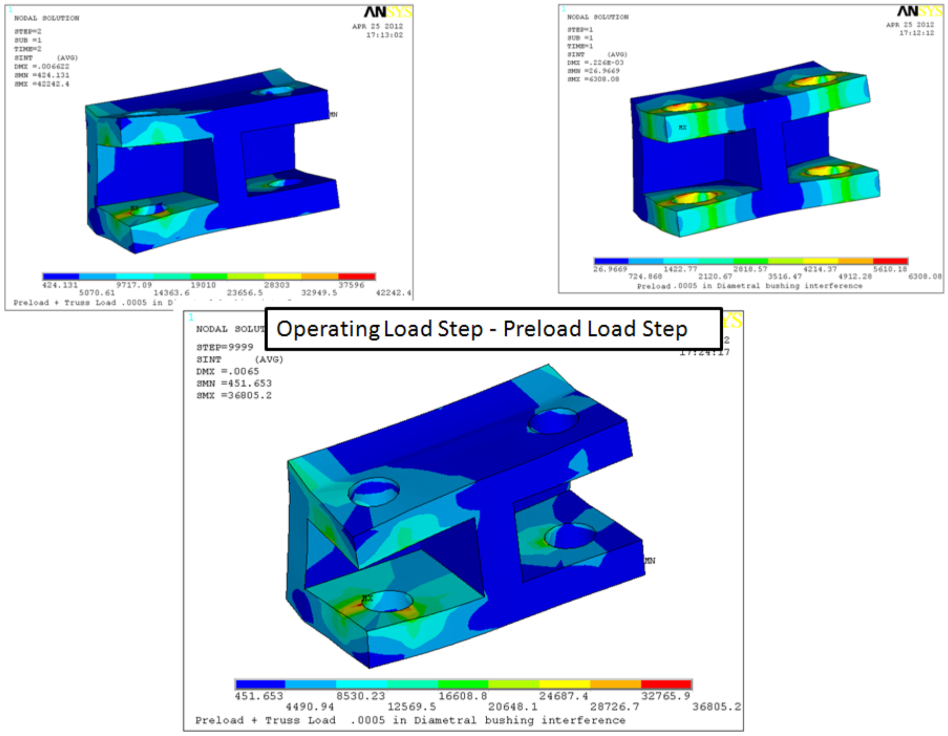


Figure 9.1-2 Clevis Plate Stresses with an interference fit.

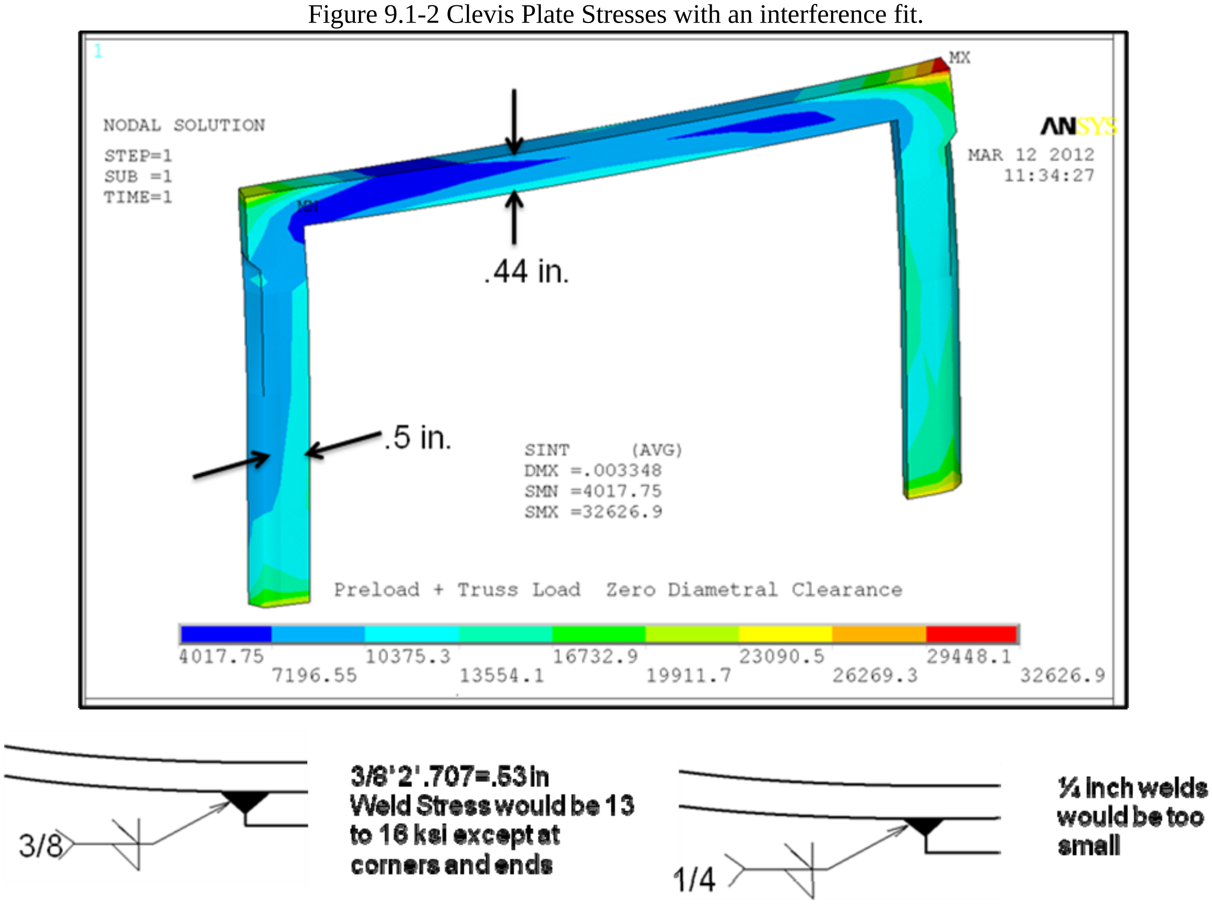


Figure 9.1-3 Clevis Weld Stress

From Figure 8.0-3 , The static weld allowable is 14 ksi. with no inspection, and 20 ksi with liquid penetrant inspection. Based on figure 8.0-4 and depending on how well the FEA captures the weld stress concentration, the weld allowable is 6 ksi to 25 ksi. From the contours in Figure 9.0-5, much of the weld would pass the fatigue criteria, but the corners and ends would not. This is similar to most of the pad-to-vessel welds and these end points and corners should be added to the inspection list.

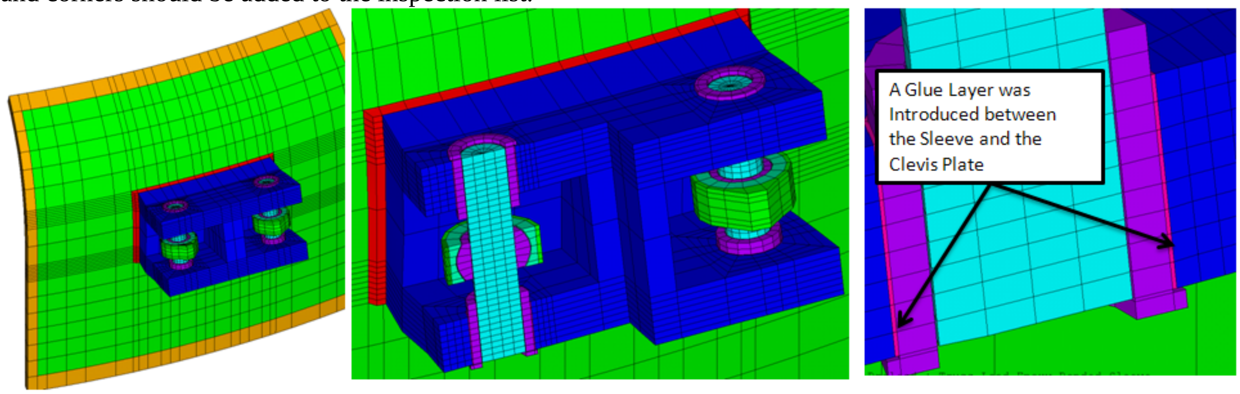


Figure 9.1-4 July 2012 Analysis with a Refined Mesh Around the Hole.

The April design of the clevis block is slightly larger than the March 2012 version to alleviate "tear-out" clevis stresses at the hole for the pin and bushing. In July 2012 the mesh was refined a bit with more elements around the holes. Local stresses are higher but the character of the behavior of the analysis hasn't changed. The model shown in figure 9.1-4 was run with and without the glue layer. Gap elements are used when the glue is not present. The results showed that an un-bonded sleeve does not contribute to the tensile net sections of the clevis plate. separation of the sleeve and the clevis plate occurs at the back side of the sleeve. This is shown in figure 9.1-5.

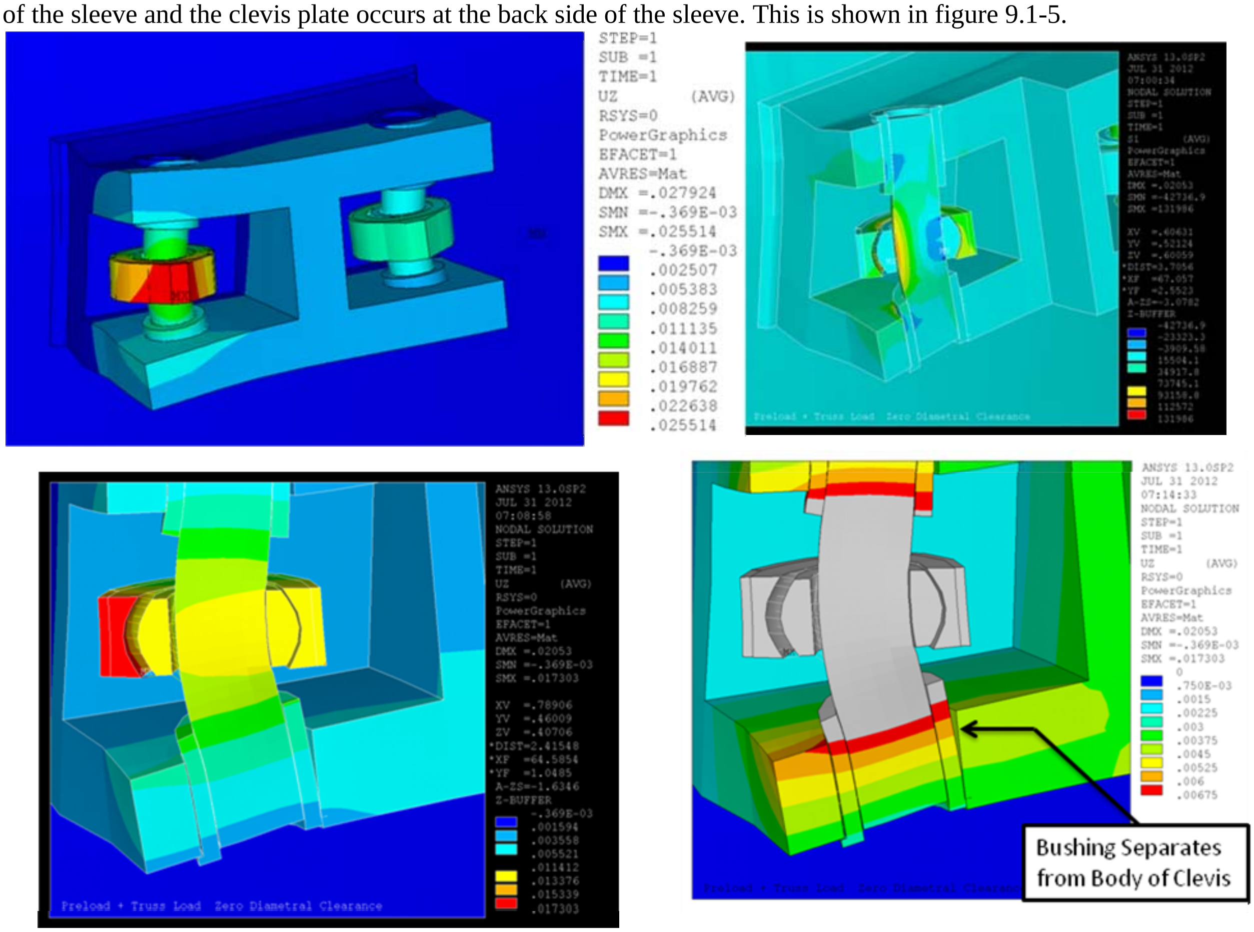


Figure 9.1-5 July 2012 Analysis with a Refined Mesh Around the Hole, and Un-bonded Sleeve

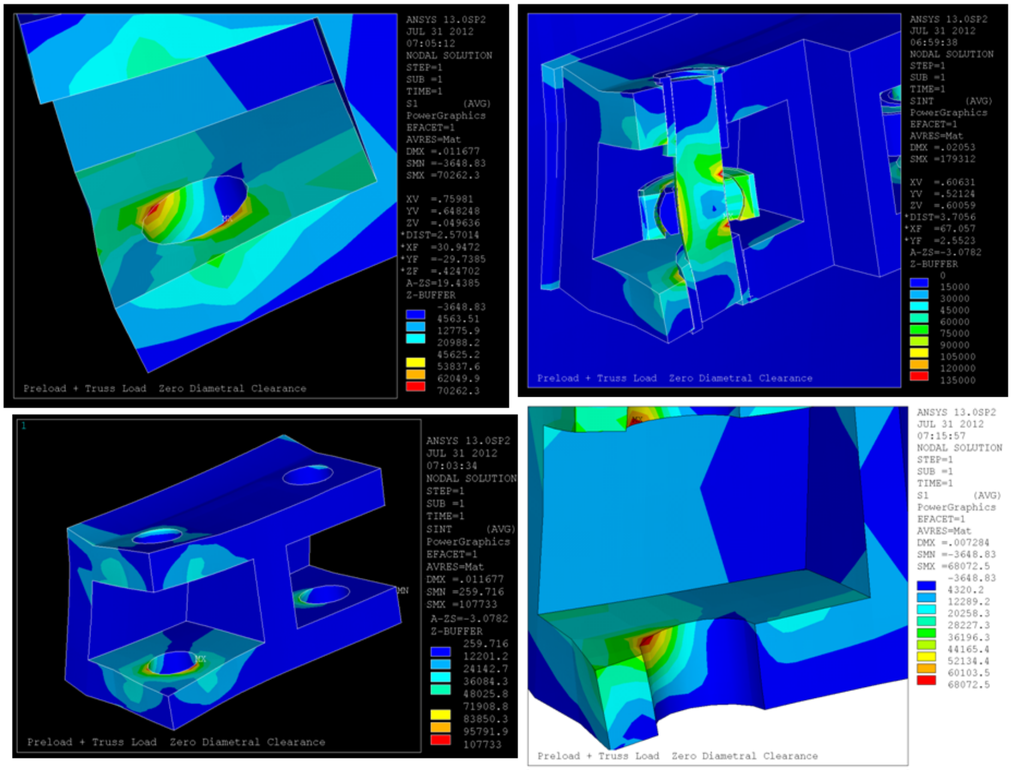


Figure 9.1-6 July 2012 Analysis with a Refined Mesh Around the Hole, and Un-bonded Sleeve

With the refinement in the mesh and the un-bonded sleeve, the stresses in the clevis plate are above the fatigue allowable of 40 ksi. Note that the peak stresses are localized, and most of the net sections have considerably lower stresses.

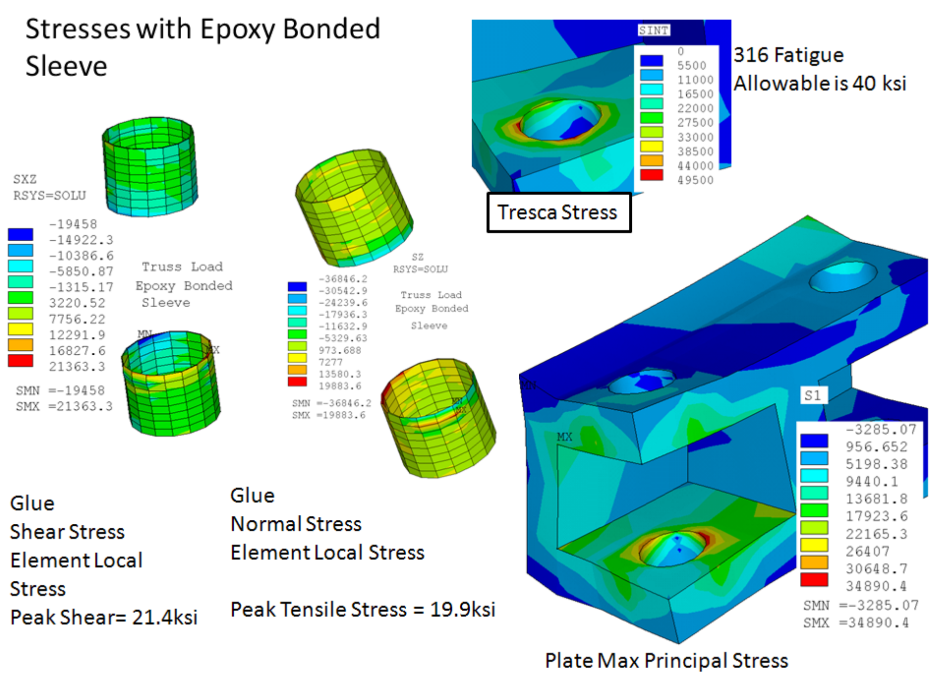
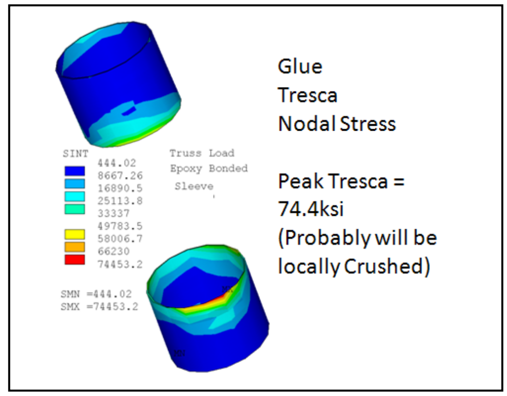


Figure 9.1-7 July 2012 Analysis with a Refined Mesh Around the Hole, and Bonded Sleeve

The most recent (April 2012) welded concept is acceptable if the pins, sleeves, and ball ends have a tight tolerance. In Figure 9.1-7 the results for the glue layer are shown.. The model was run with an epoxy layer bonding the sleeve to the clevis plate. Most of the epoxy sees less than 7ksi tension. There are local small spikes of 20 ksi, and an edge that will probably crush from the compression. Shears are 5 to 20ksi The clevis plate stresses are 34.8 ksi with the bonded sleeve, below the 40 ksi fatigue limit. Given the uncertainty in the epoxy performance, the clevis holes should be on the fatigue inspection list.



9.2 Late Feb 2012 Welded Clevis

This design (Late Feb/ March 2012) is welded directly to the vessel wall after removal of the existing pad.

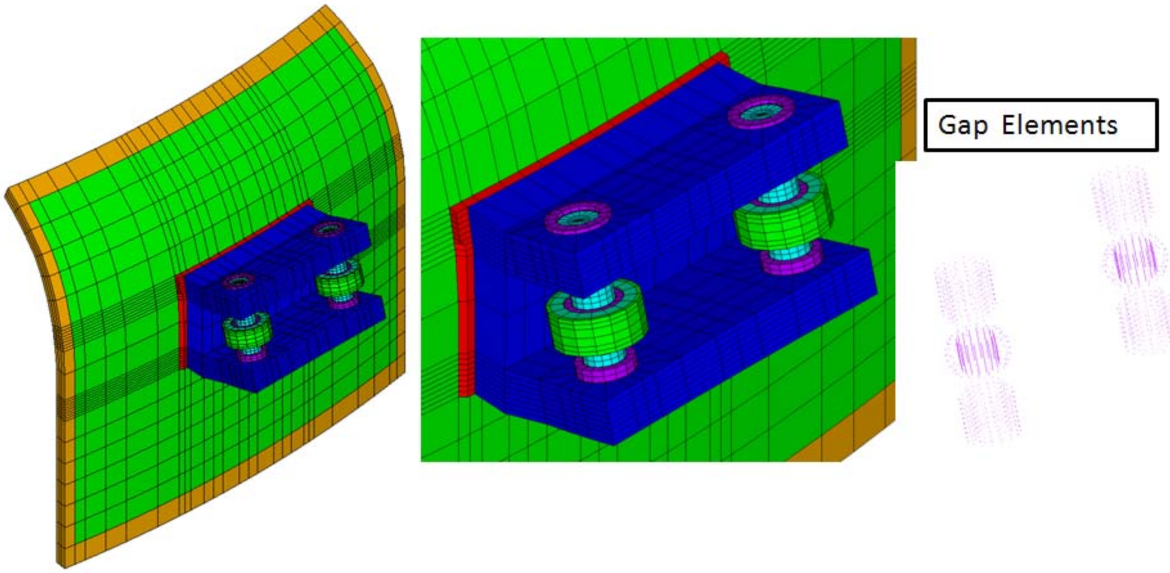


Figure 9.2-1 Model of the TF Clevis Design as of March 2 2012

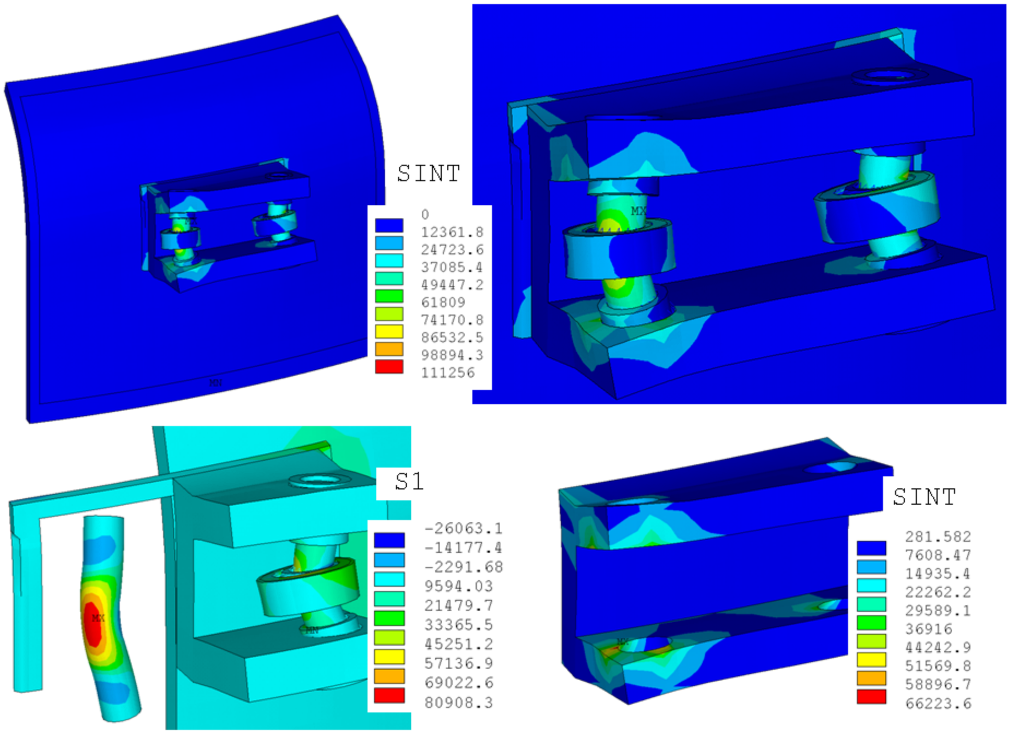


Figure 9.2-2 March 2012 Stress Results

Because of the high stress in the clevis plate because of the proximity of the pin and bushing to the edge of the clevis plate, the block was increased in size.

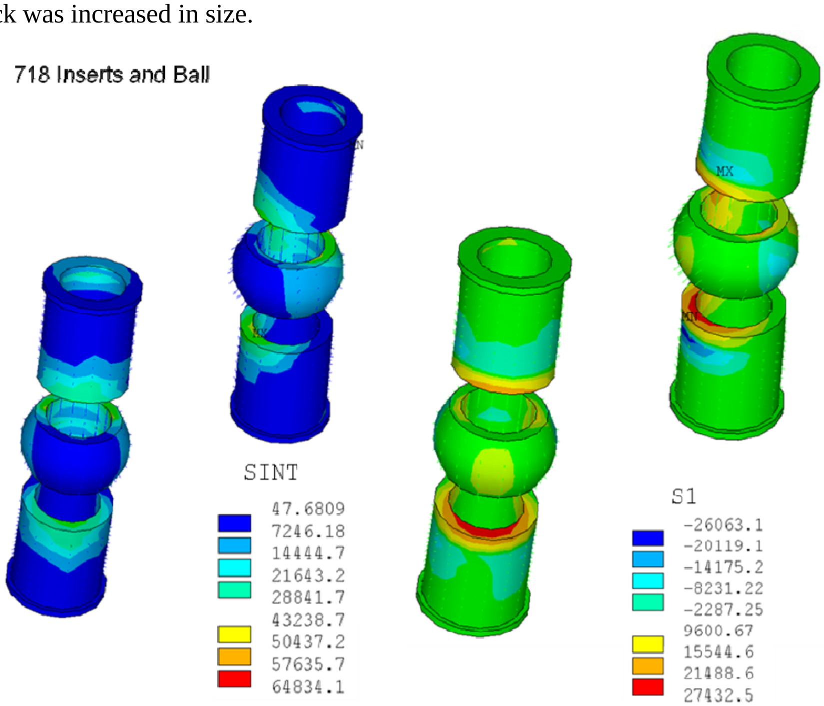


Figure 9.2-3 Bushing and Ball End Stress

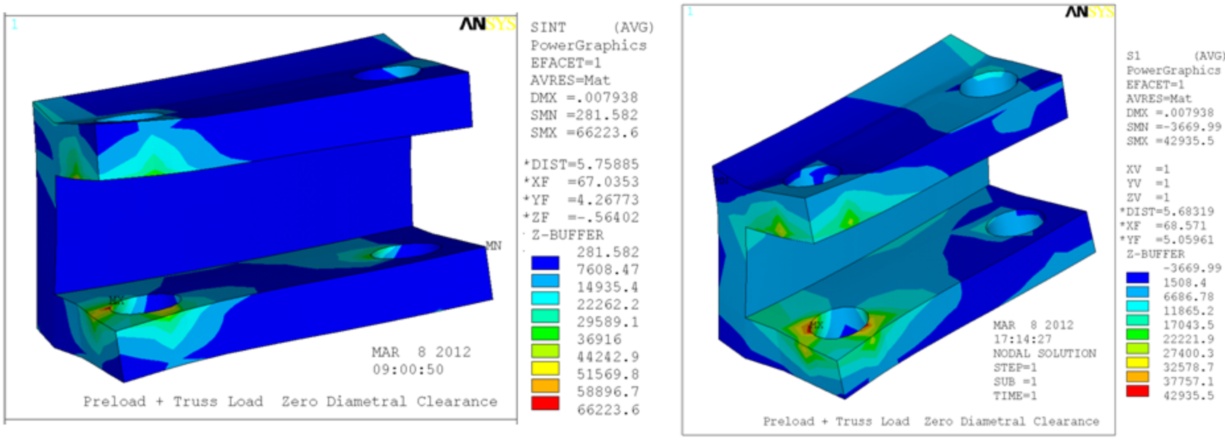


Figure 9.2-4 Clevis Plate Stress March 2012 Block Geometry

The clevis plates are stressed above the Tresca based of 40 ksi (Figure 8.0-2), A lot of this is compressive bearing stress between the sleeve and plate. The max principal stress, which is a better indication of the propagation of a fatigue failure, is 43 ksi. Both of these stresses are a bit high. H. Zhang (the calculation checker) found an error in the allowable. It had been 60.9 ksi based on a factor of 20 on life and the 2 on stress had not been considered. The fatigue based on 2 on stress is 40 ksi. - The stress state of the clevis block was discussed with Mark Smith and Tom Willard and the block was increased in size to improve the pin hole edge distance. The Tresca is still a bit above the allowable, but the max principal is 31 ksi

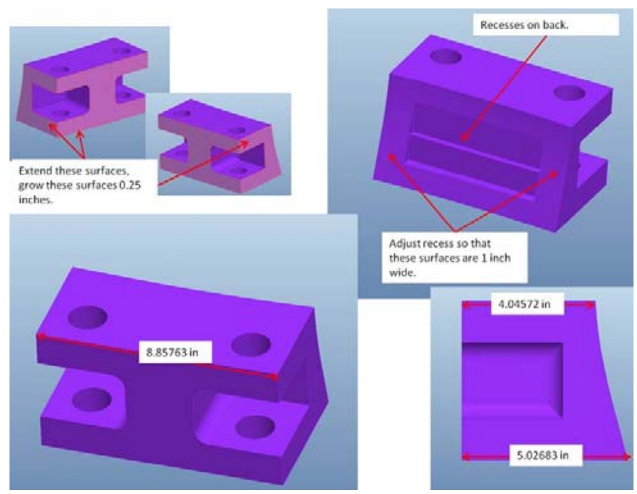


Figure 9.0-5 Clevis Plate April 2012 Geometry

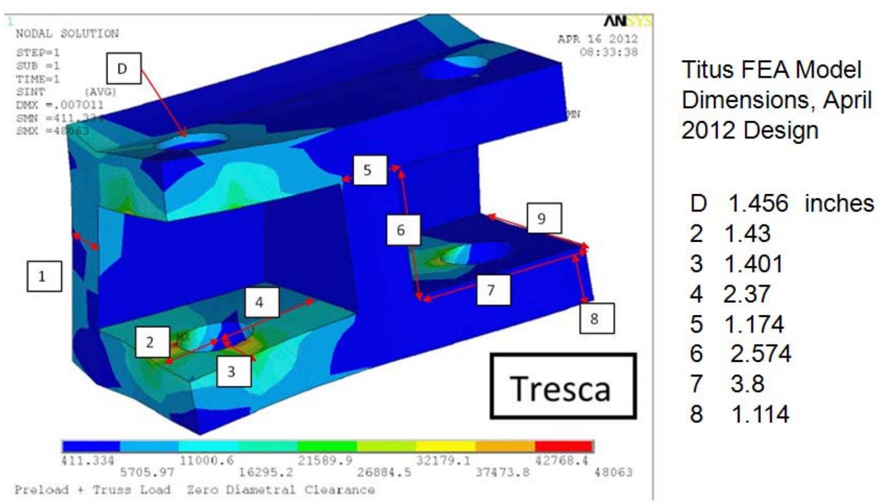


Figure 9.2-6 Clevis Plate April 2012 Geometry, As Analyzed

10.0 Earlier Welded Clevis

A number of concepts have been evaluated to reinforce or replace the present clevis hardware. These will be presented after the final design configuration. The preferred option is a simple concept in which a clevis plate with a refined geometry is welded to the existing pad. The shape and sizing of the clevis is chosen to eliminate moments applied with respect to the surface of the vessel. This loads the attachment to the vessel only in shear - no bolt or weld tension is required. A couple of concepts produce no, or little moments at the vessel surface. The welded concept is presented first and then a mechanical concept is described and analyzed. The qualification of the mechanical concept is included in case it is needed in one or more of the 24 locations needing upgrade.

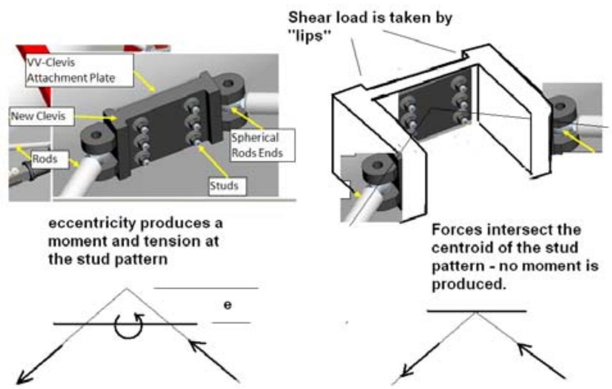
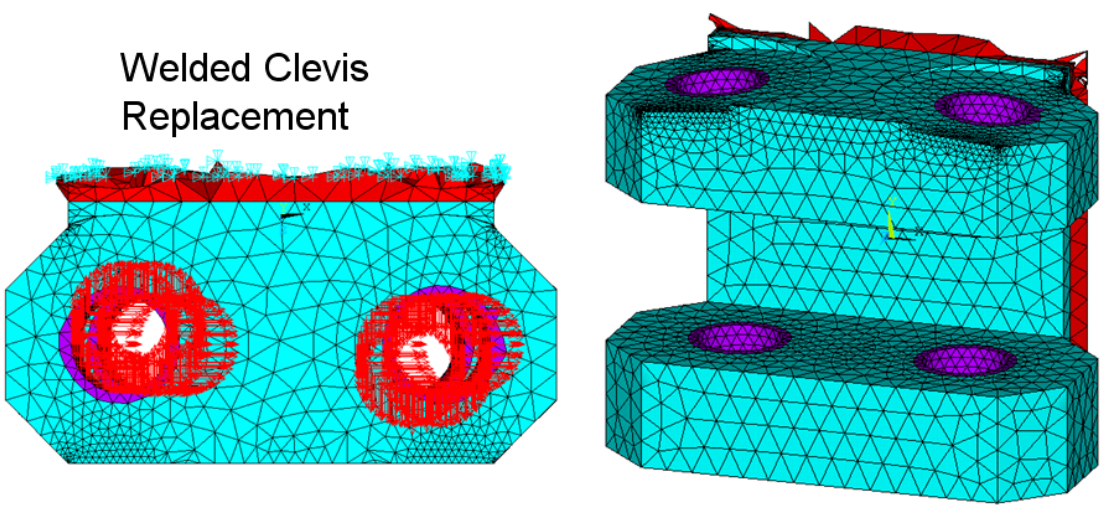


Figure 9.0-1 Mechanics and geometry that eliminates the offset moment on the clevis assembly



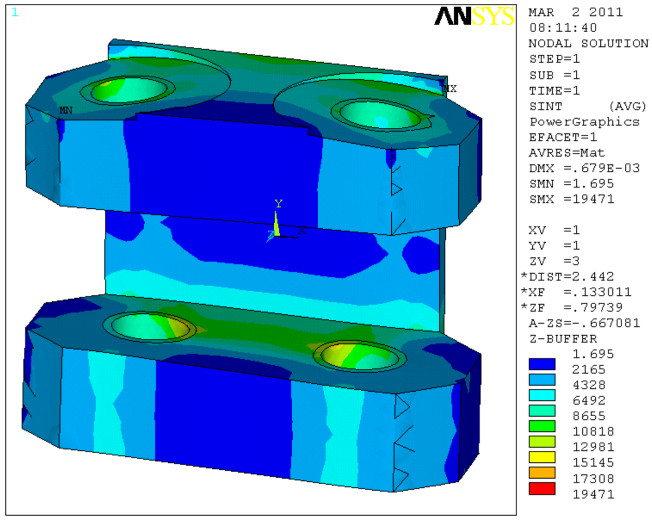


Figure 9.0-2 Figure Stresses based on a 20,000 lb Shear Load

The stress will scale by 37/20, so the peak stress is 36013psi, and away from the local contact is: ~15ksi*37/20 = 27 ksi

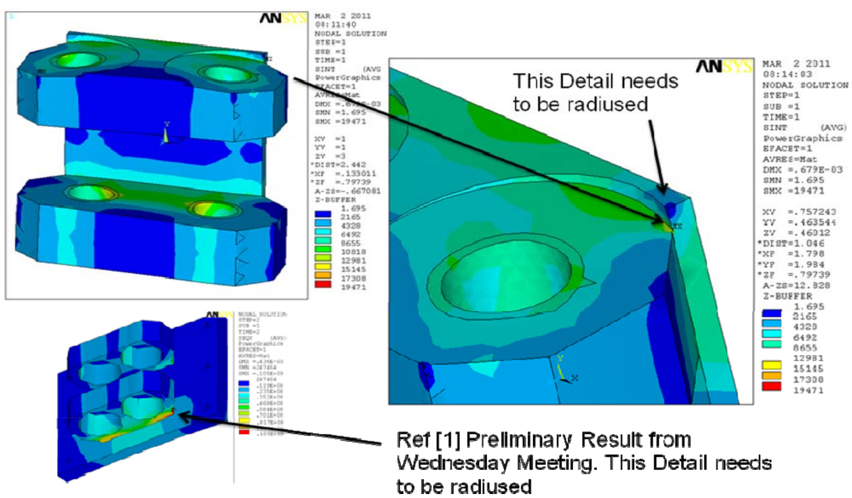


Figure 9.0-3

11.0 Clevis Pin Analysis

A simple treatment of the pin stress assuming no fixity at the ball end bushing or clevis plates is shown in figure 11.0-1. If the pins can be tightly fitted, the stress drops to 80 ksi The clevis pin is currently (April 2012) a 1 inch diameter pin that is loaded in bearing bending and shear. The bending stress was a function of the separation of the clevis plates, and the fixity assumed for the fit in the clevis plates and the rod end ball bushing.

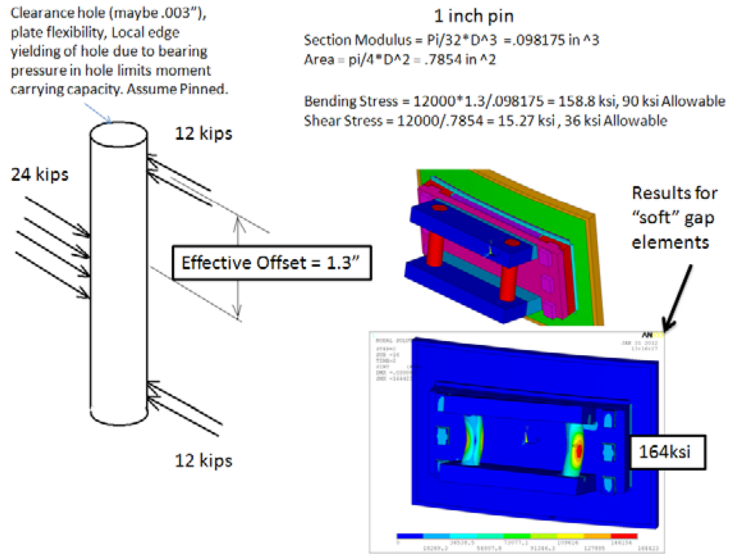


Figure 11.0-1 Clevis Pin Analysis Based on No fixity at the pin center or ends.

In order to qualify the current(April 2012) pin, a tight fit is needed at the clevis plates and bushings, and credit must be taken for the latest version of the GRD that requires a design life of 20,000 full power cycles rather than the original 30,000 cycles. The pin fatigue allowable of 80 ksi is developed in Figure 11.0-5. The pin stresses are shown in Figure 11.0-2. The 98 ksi stress is a contact compressive stress and is not on the tensile side of the pin. If a crack initiated here it would not propagate. The 74 ksi max principal stress is considered representative of fatigue loading, and this is within the allowable. Again this relies on a tight fit.

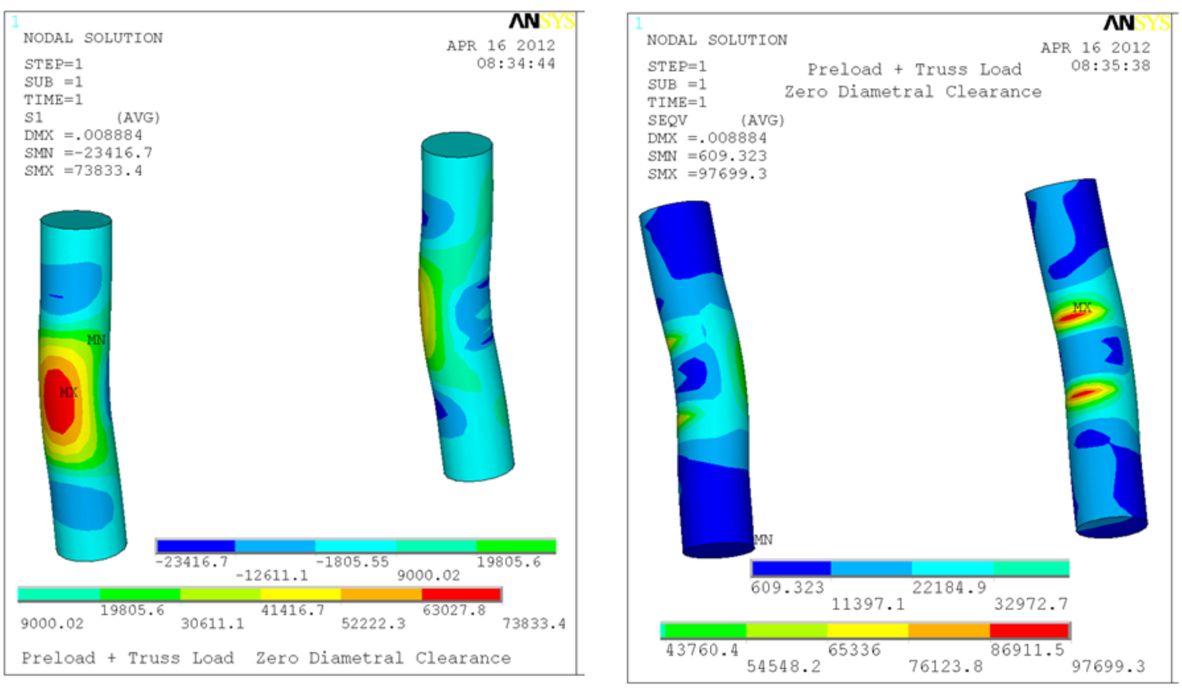


Figure 11.0-2 Clevis Pin Analysis Based on the April 2012 Clevis Configuration

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For 718, from section 8, Sm , the membrane allowable, is 60ksi and the allowed shear stress is .6*sm = 36 ksi, Ref[4] NSTX Criteria Doc. This is based on the average shear stress in the pin - as would be consistent with a membrane stress. Stress Calculations are included in the figure below. The project had used a 3/4 pin because it produced a more compact total assembly, and a larger 1 inch pin in a later design to reduce stresses.

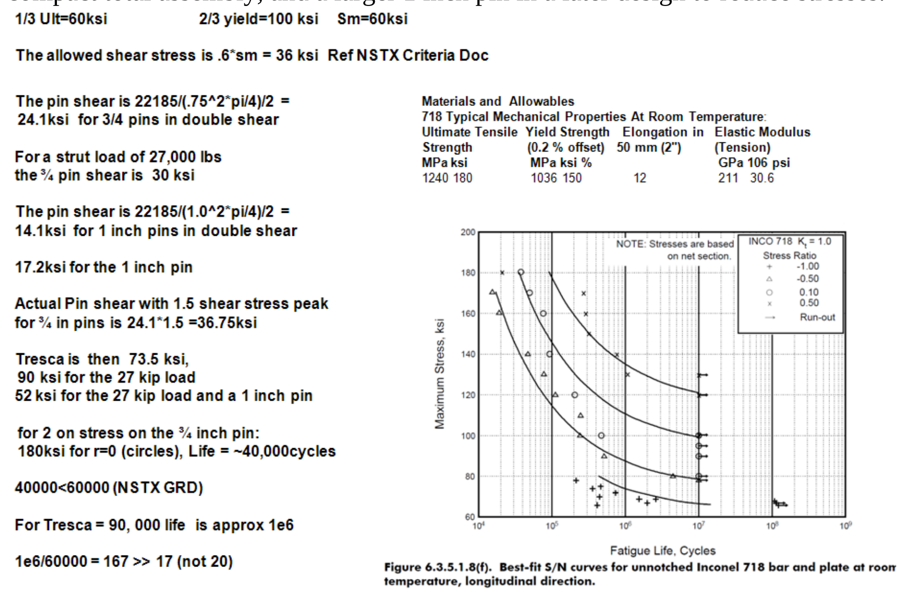


Figure 11.0-3 Clevis Pin Analysis, from and early sizing attempt.

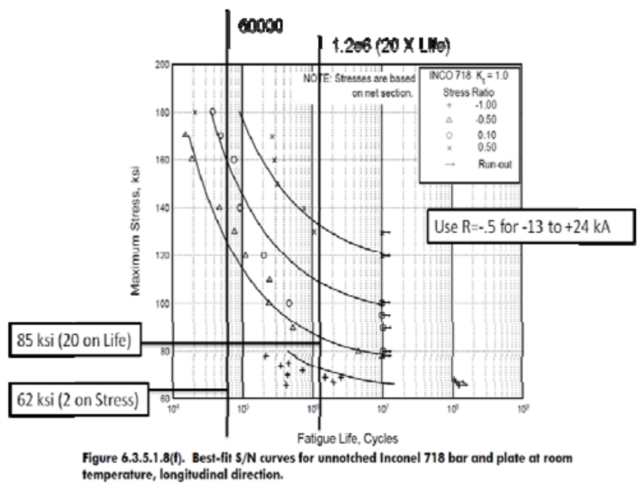


Figure 11.0-4 Clevis Pin Fatigue Allowable Based on Earlier GRD requirement of 30,000 Full Power Pulses

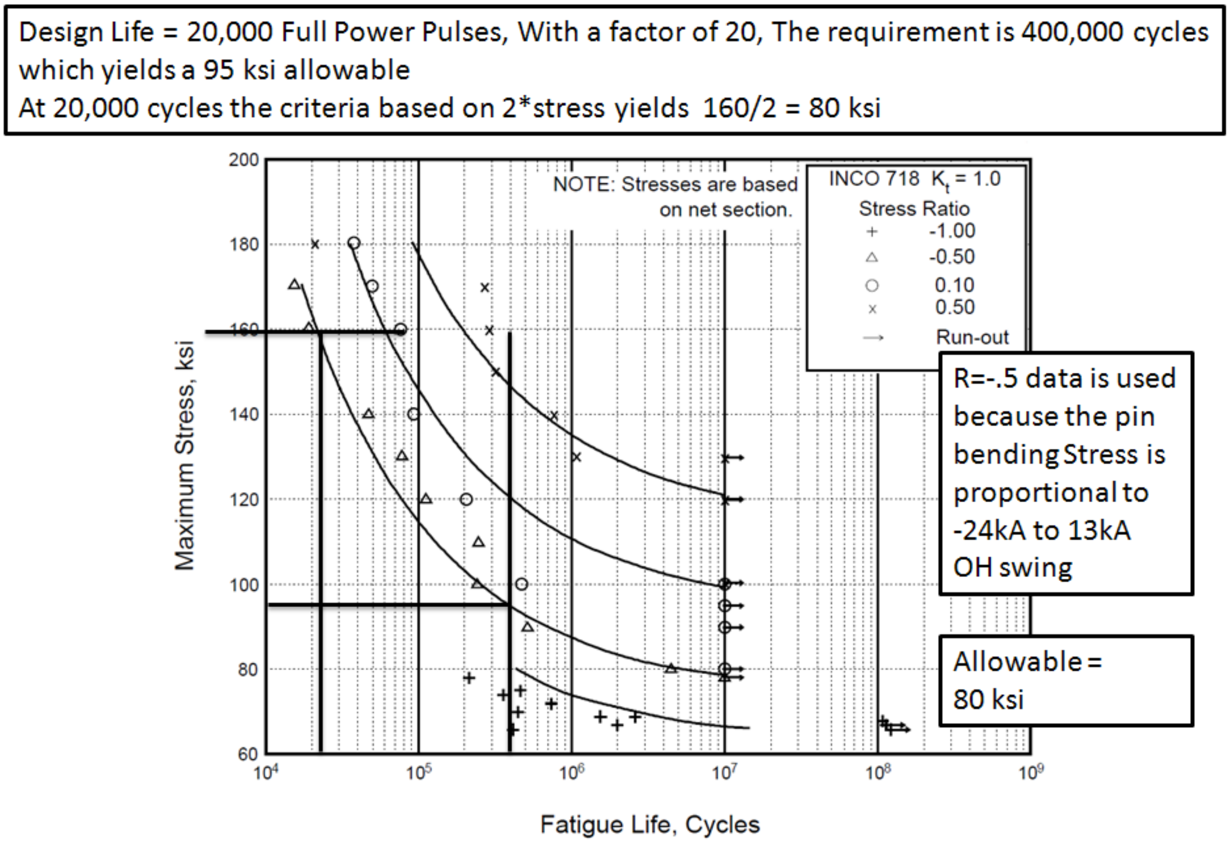


Figure 6.3.5.1.8(f). Best-fit S/N curves for unnotched Inconel 718 bar and plate at room temperature, longitudinal direction.

Figure 11.0-5 Clevis Pin Fatigue Allowable Based on the GRD requirement of 20,000 Full Power Pulses

In later designs, one inch pins were used and the design as of Feb 10 2012 had clevis plates separated to the point where the pin bending was excessive. The effect of the pin-fit was examined. If the pins have zero clearance and the plates and ball end are fully elastic, then the estimate the pin stress, provided by M. Smith is correct. but at 103ksi it still violates both the static and fatigue the allowable for the 718 pin.

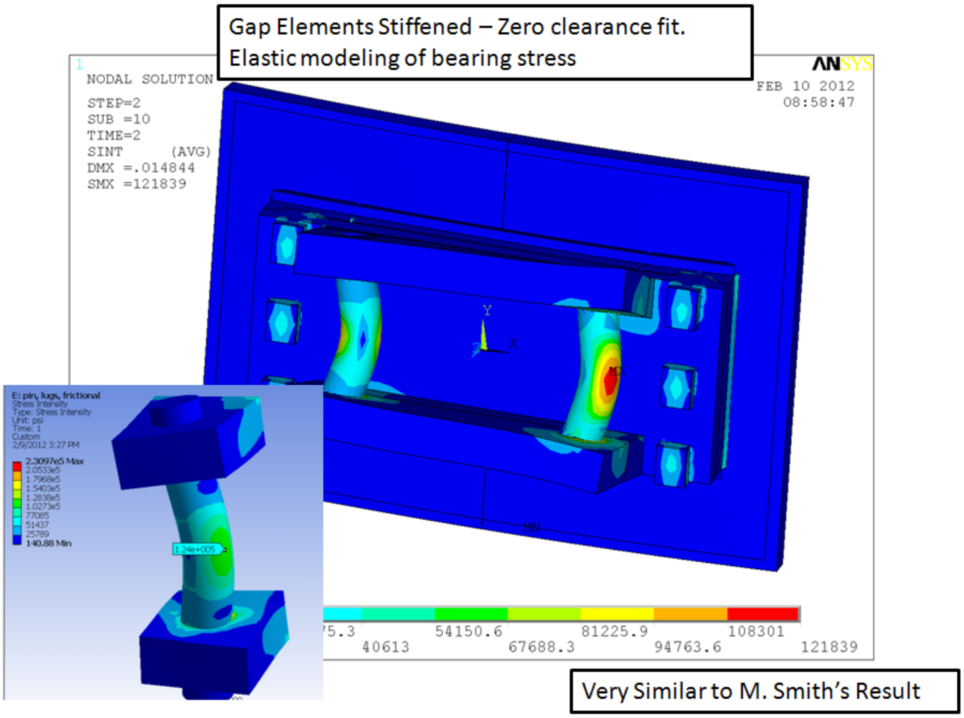


Figure 11.0-5 Study of the Effect of Pin Fit-Up

If a clearance is modeled i.e. .003" on the diameter so that the pins could be removed, and if you consider local contact yielding (I get a 170 ksi local bearing stress at the edge of the plate holes) then the assumption of moment support at the plates and probably at the ball end bushing, is not correct. The bending stress in the pin is then 160 ksi and the plate supports for the pins must be brought closer together.

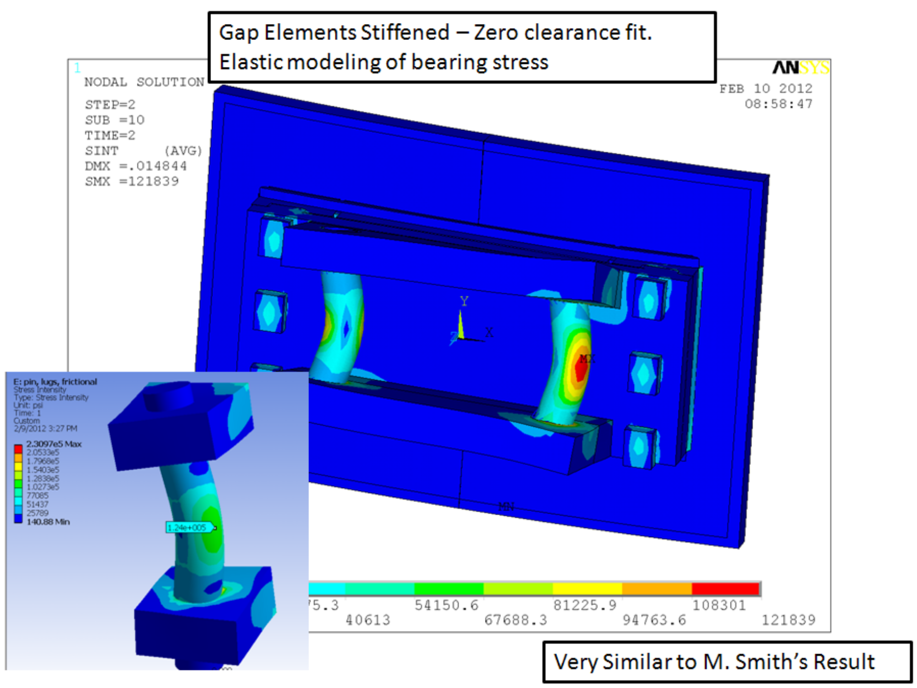


Figure 11.0-6 Study of the Effect of Pin Fit-Up

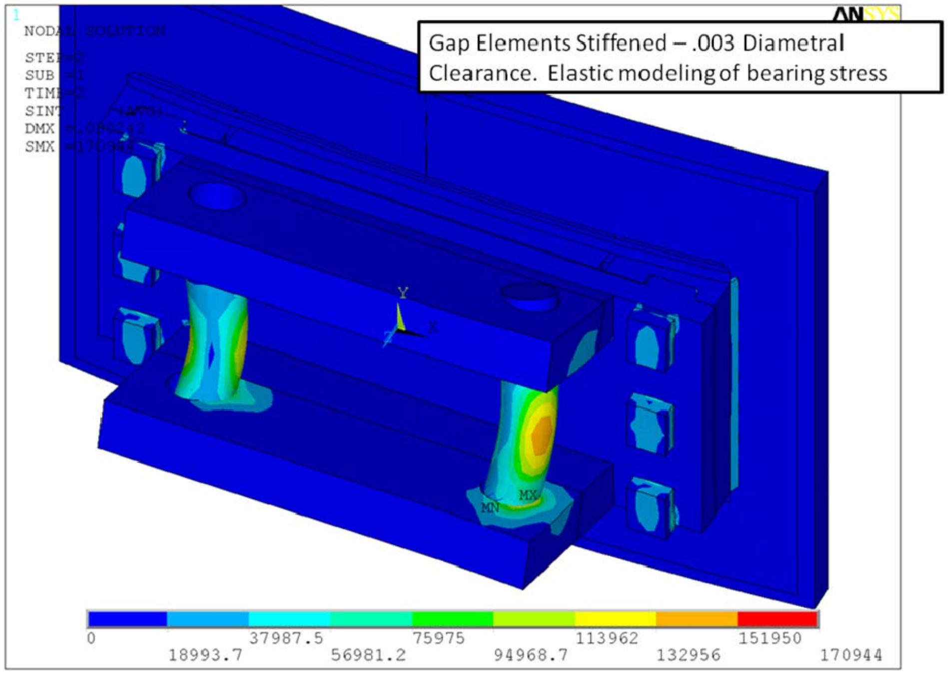


Figure 11.0-7 Study of the Effect of Pin Fit-Up (loose Fit)

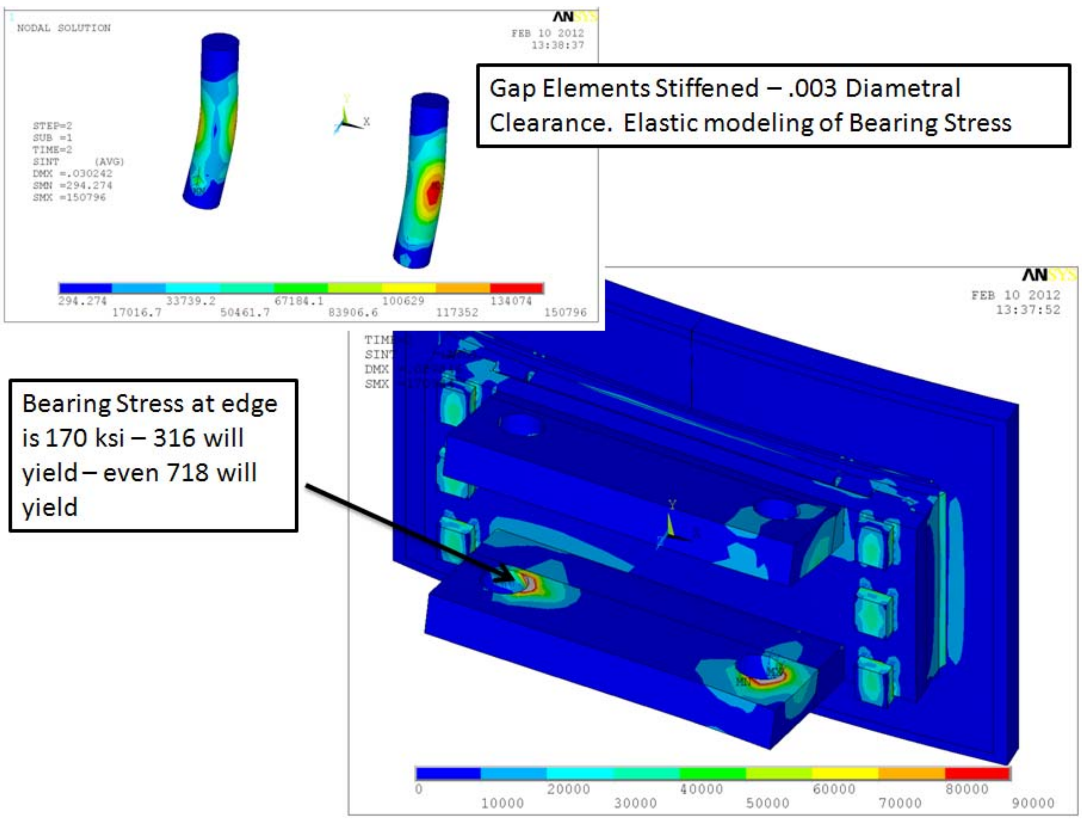


Figure 11.0-8 Study of the Effect of Pin Fit-Up (loose Fit)
Local Bearing stresses are significant

12.0 Weld Stress for the Welded Clevis Design

Welds were assessed using a finite element model and these analyses are shown in figure 11.0-1 and 2 Hand calculations are presented in figure 11.0-3 and these include the latest loads and thoughts on how the loads are taken. The weld stress allowable is 14 ksi with only visual inspection, and 20 ksi with penetrant inspection.

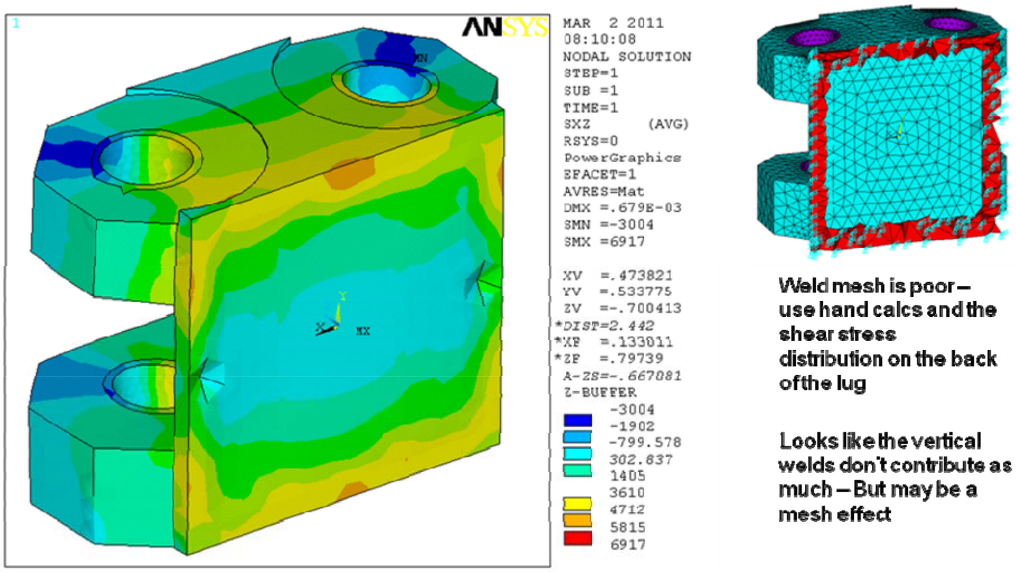


Figure 12.0-1

Full Perimeter			Full Perimeter			Only Horizontal Legs	
Welded Pad Width	3.375		Welded Pad Width	3.375		Welded Pad Width	3.375
Welded Pad Height	3.5		Welded Pad Height	3.5		Welded Pad Height	3.5
Weld Fillet	0.15		Weld Fillet	0.25		Weld Fillet/Groove	0.5
Weld Perimeter	14.35		Weld Perimeter	14.75		Weld Length	7.75
Weld Stress	13142.18		Weld Stress	7671.469		Weld Stress	7300.269

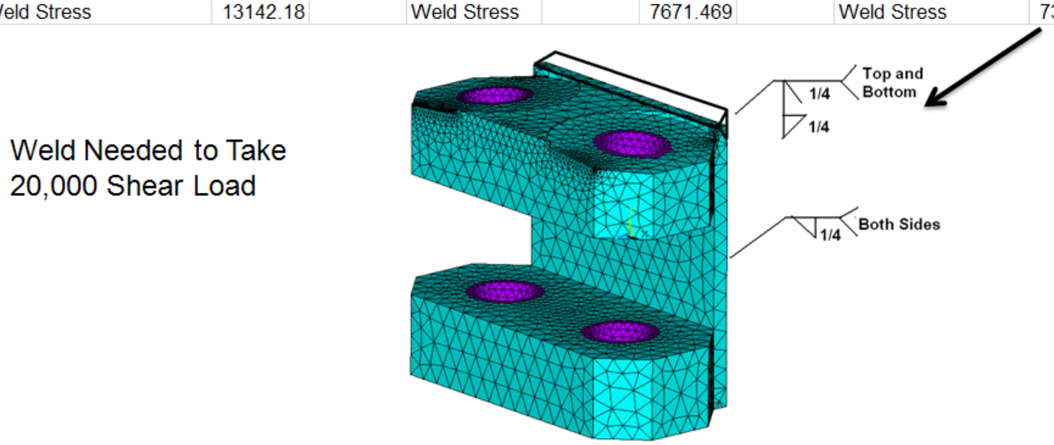


Figure 12.0-2

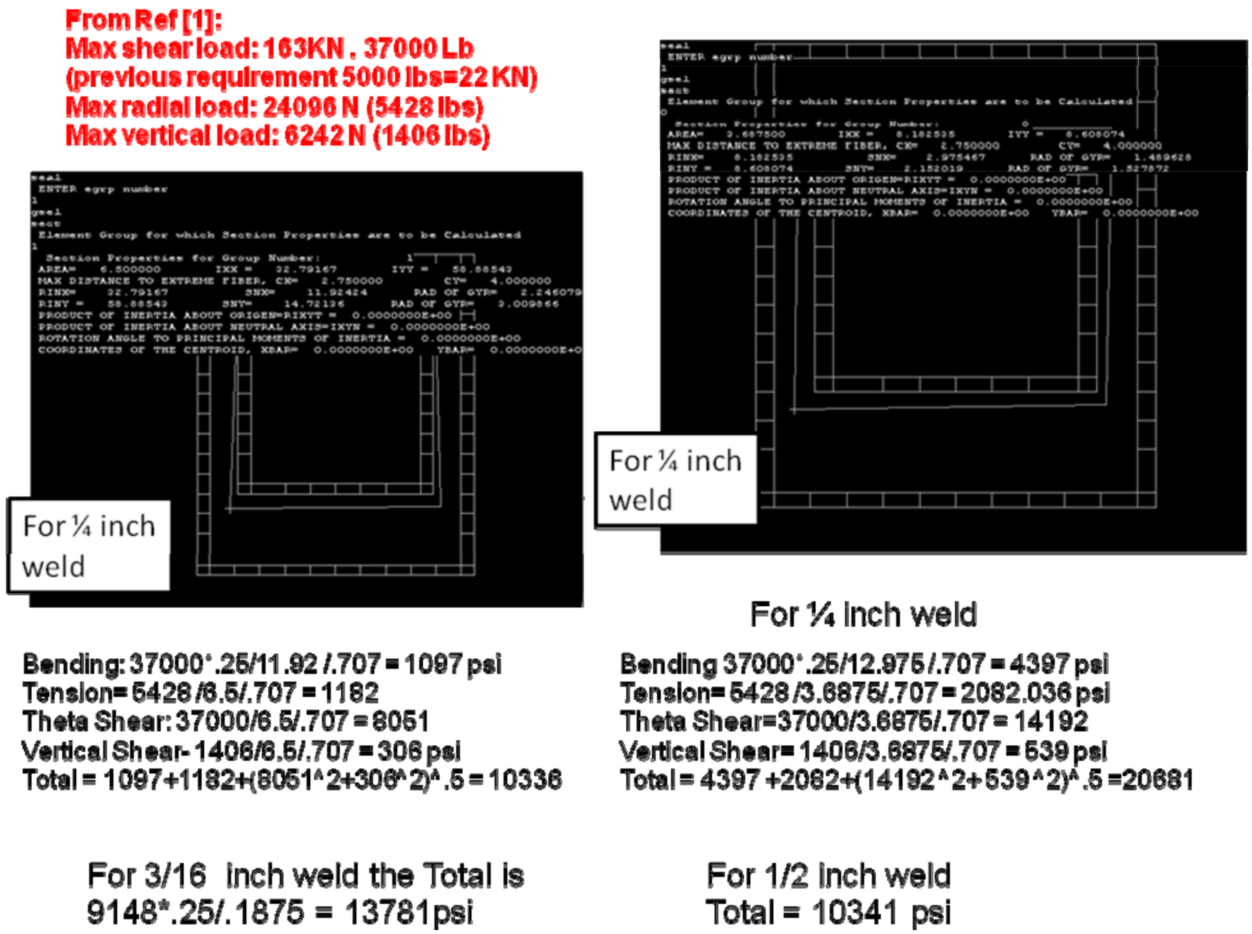


Figure 12.0-3 Hand Calculations based on Weld Pattern Section Modulus

From Figure 6.1-2 , The weld allowable is 14 ksi. with no inspection, and 20 ksi with liquid penetrant inspection

For the existing 3/16 weld (actually the effective size for larger poorly shaped welds) and for a proposed effective 1/2 inch weld -1/4 inch Jgroove+1/4 inch fillet, the weld stresses are within static allowables.

From Figure 8.0-4 Weld Fatigue Allowable If the FEA modeling represents the local weld stress concentration, well, the fatigue allowable is 175 MPa (25.4ksi), and for simple line load calculations the allowable is 6345 psi to allow for a concentration factor

For the existing weld, the nominal stress is 14 ksi, or a little over twice the allowed weld stress computed in the figure above. These calculations assume uniform shear around the perimeter weld. There will actually be a concentration at the corners of the weld pattern. This was the case with the PF4 and 5 bracket supports. and the expectation that this would potentially be a fatigue failure point led to inspections of these areas by Joe Winston. No indications of cracking were found. The corner areas of square welded pads should be included on the list of areas to check.

Note:
(1) VV Clevis PAD is not welded all around.
(2) Large gap between clevis and vessel.
(3) Weld region (underneath pad) cannot be inspected. This is a source of fatigue failure.
(4) Welds need to be rated accordingly.

Figure 12.0-4 Weld As-Builts

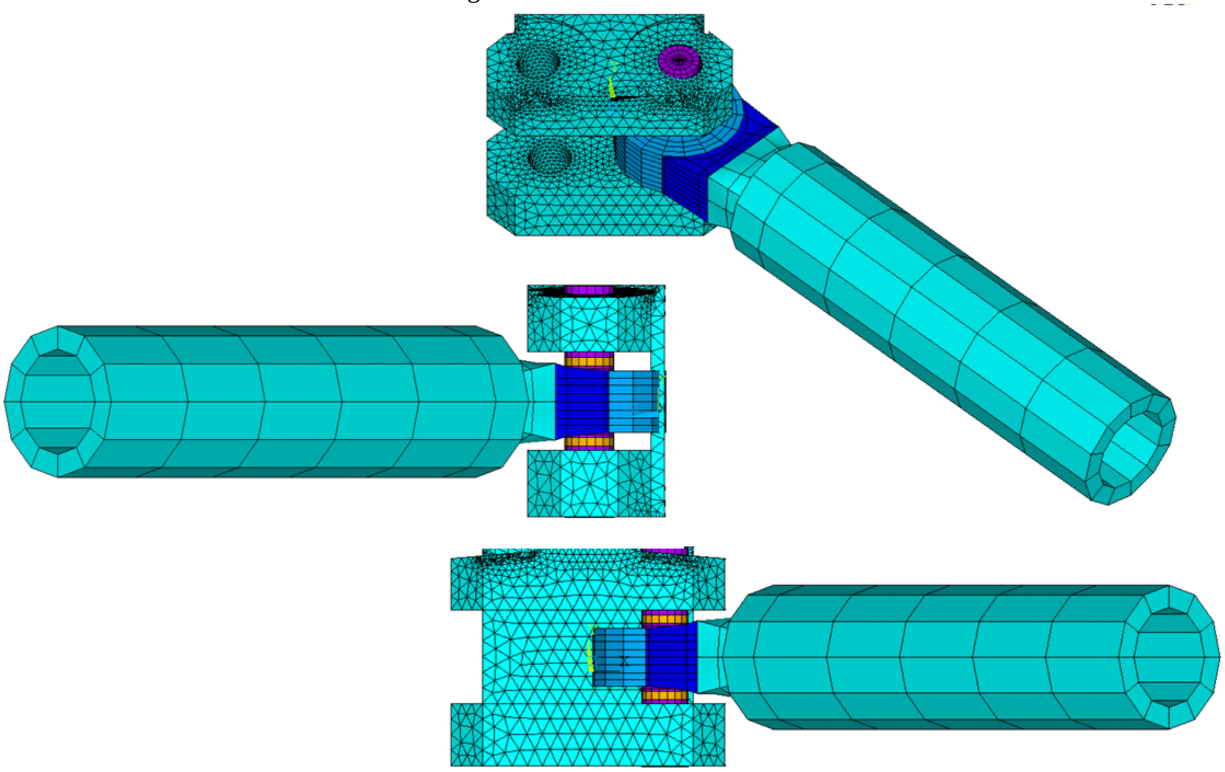


Figure 12.0-5 FEA Model/ Geometry Study to Show Clearance Issues with the Ball End

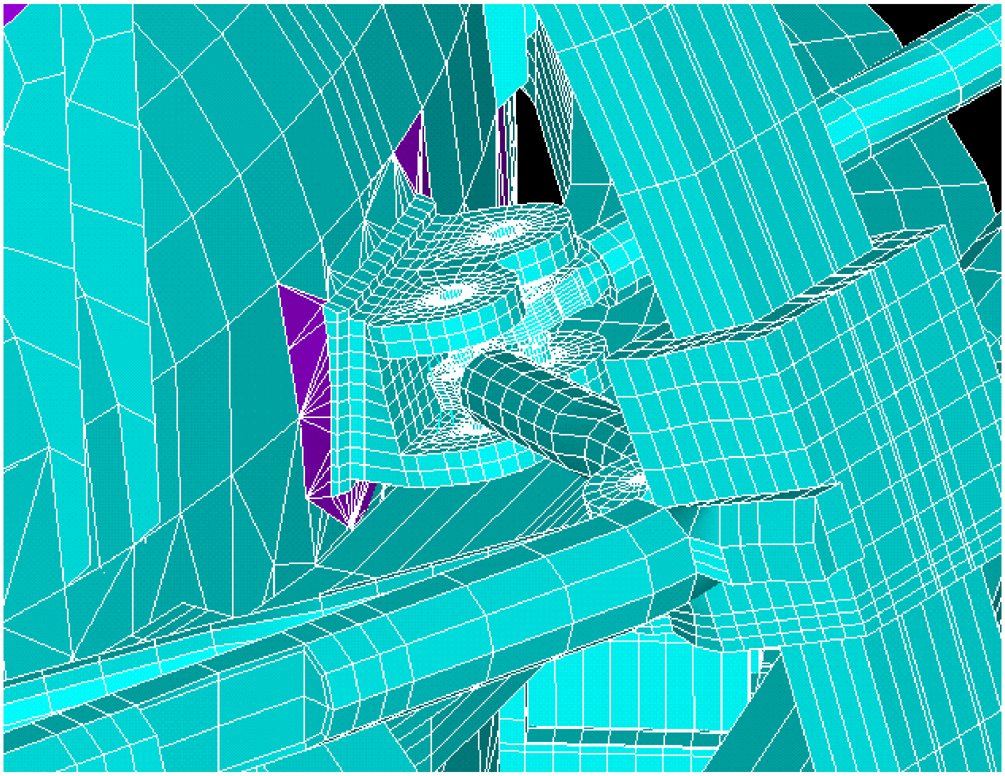


Figure 12.0-6 FEA Model/ Geometry Study to Show Clearance Issues with the Ball End

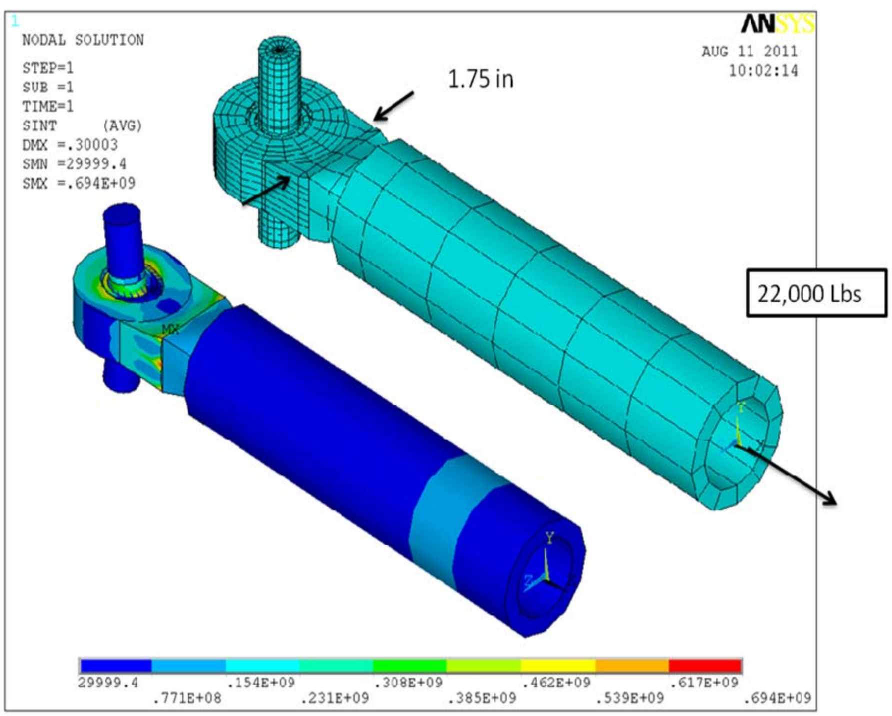


Figure 12.0-7 Analysis of the Ball End Detail

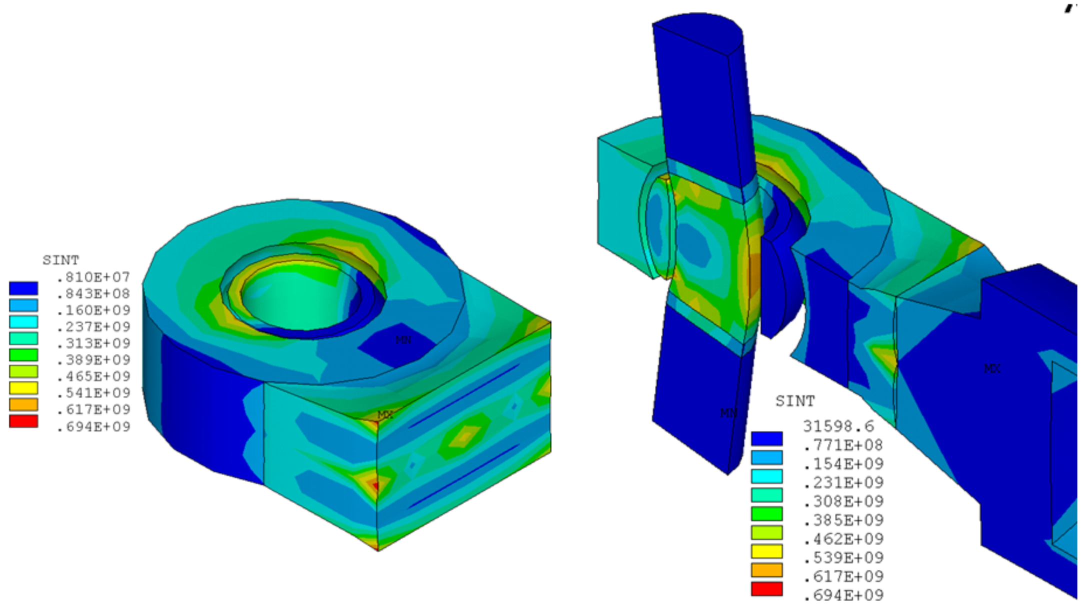
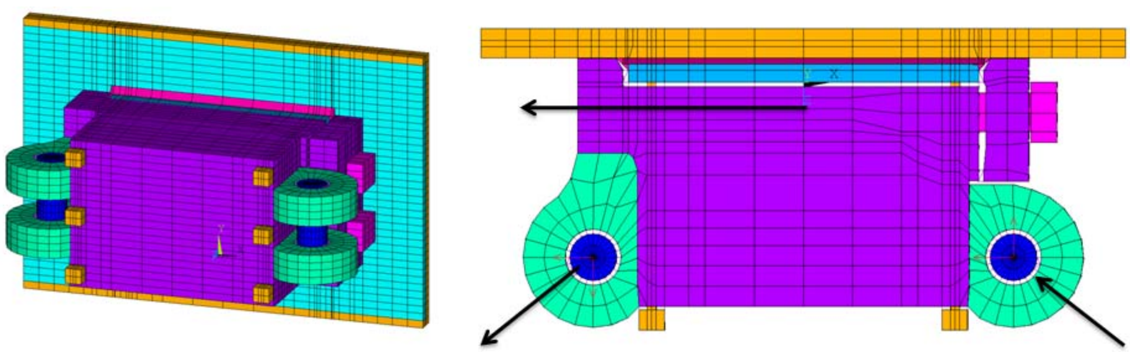


Figure 12.0-8 Analysis of the Ball End Detail

Figure 12.0-8 is an early indication of the pin bending issue that has effected all the designs up to the present configuration (Section 9)

13.0 Mechanical Attachment Employing Welded Studs and Clamped Shear Mechanism

The intention of this option is to provide a clevis geometry that develops only shear at the vessel surface, and then engage the existing pad as a "shear key" . One difficulty with this is the tolerancing on the size and positioning of the pads made it difficult to have a tight fit with the clamp. This was fixed with adjustable edge clamps. Another difficulty is that the edge of the pad that protrudes above the weld is small . This is all that is available to obtain a "purchase" by the clamp.



Clevis Detail with bolted edge clamp
Figure 13.0-1

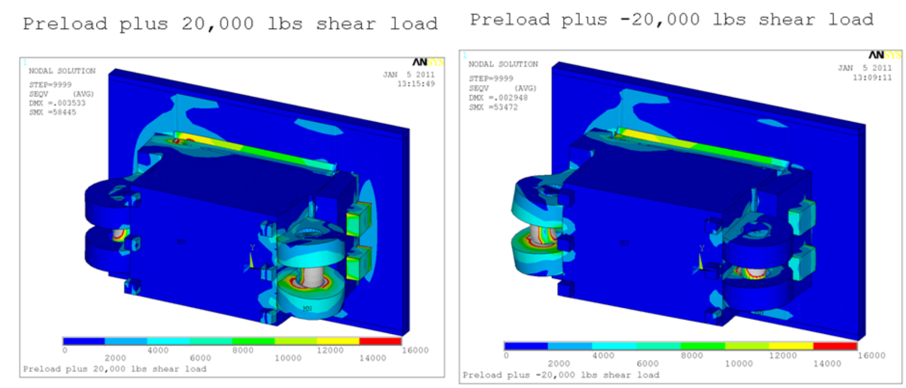


Figure 13.0-2

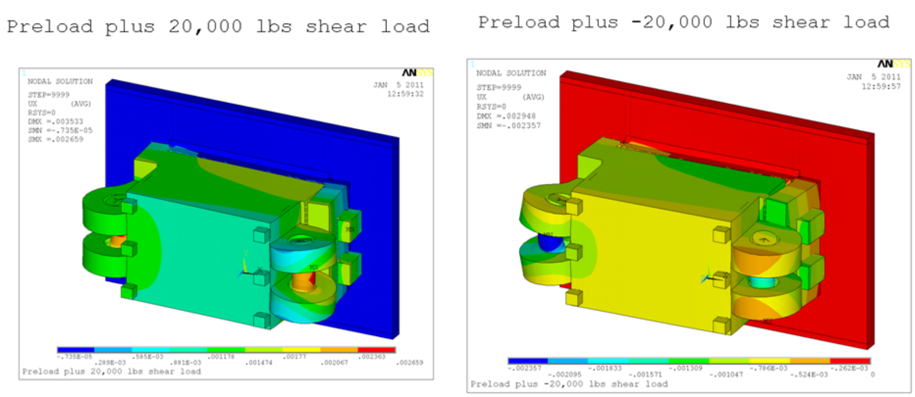


Figure 13.0-3

Clevis Pin Analysis for the Mechanically Attached Clevis

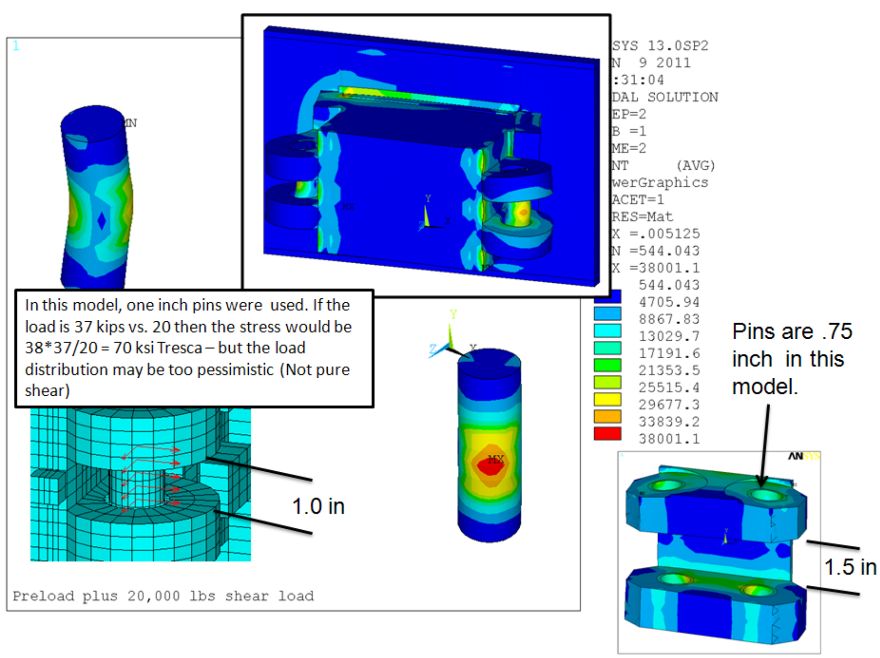


Figure 13.0-3 Clevis Pin Stress

Pin stress in this analysis is a consequence of pin bending due to the clearance in the clevis. The same tight fitting approach as is used for the welded clevis would be appropriate and would allow the use of 3/4 inch pins

14 Strut Analyses
14.1 Strut Buckling

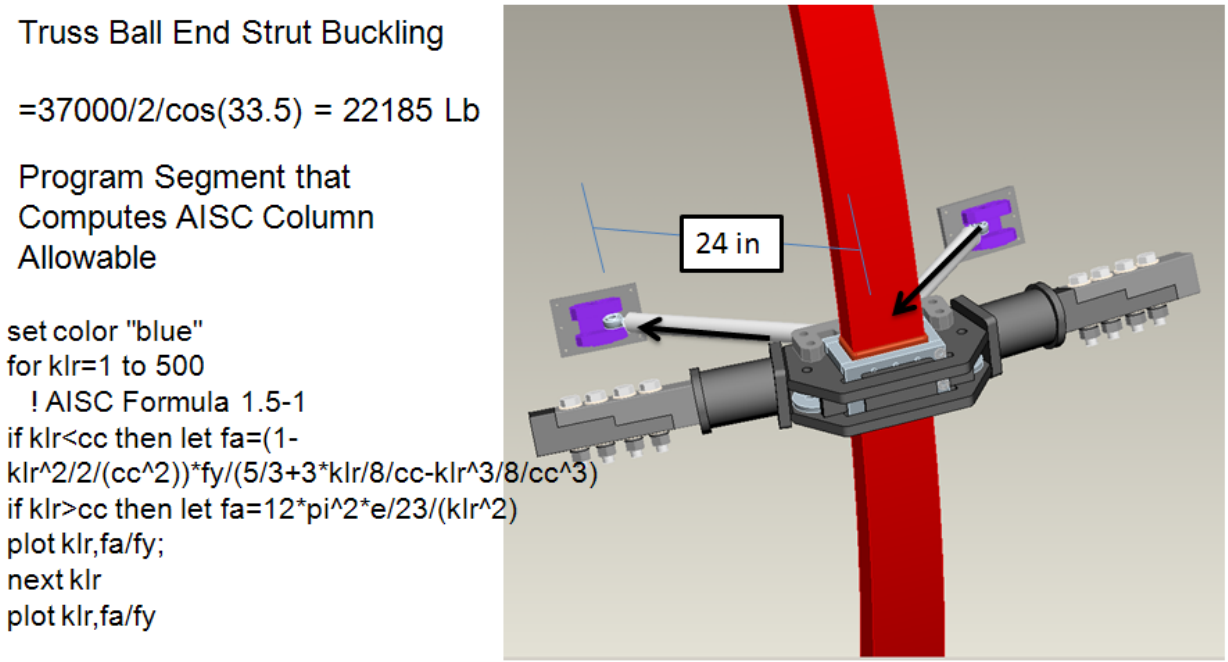


Figure 13.1-1

In this figure, the compressive load was conservatively calculated from the shear reaction load at the vessel surface without any credit for the radial tensile load, reported in [1]

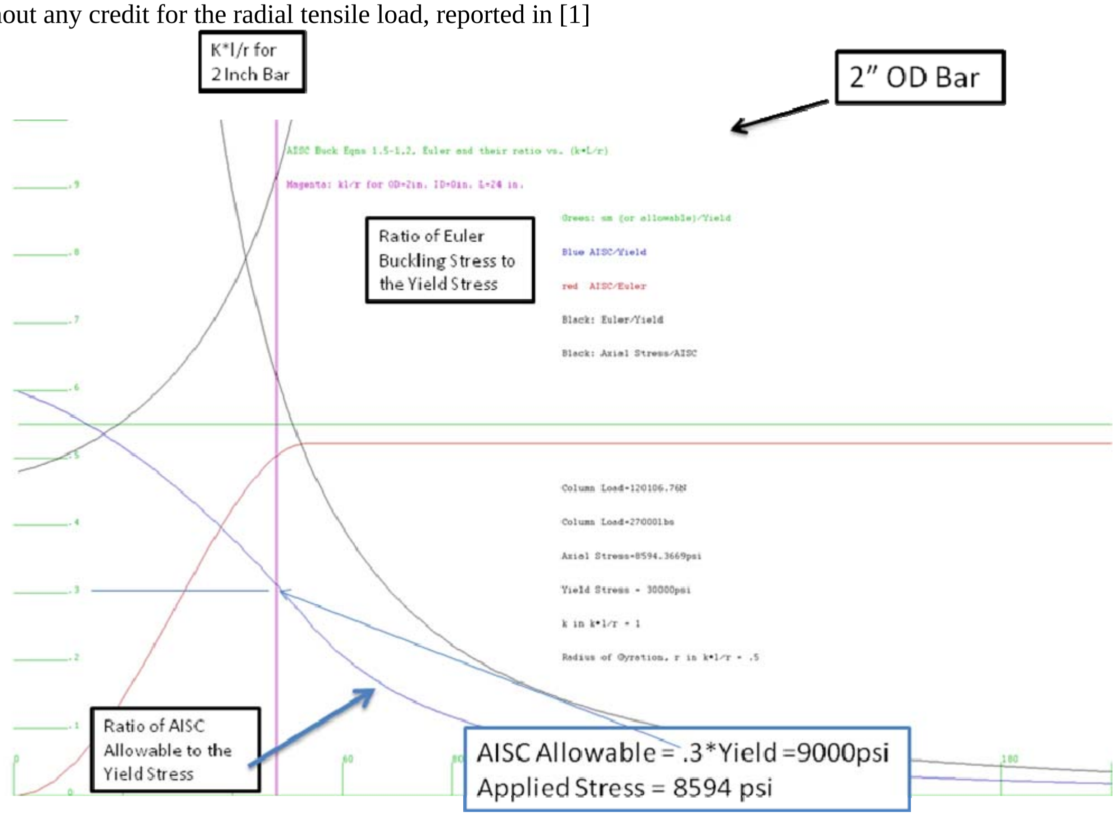


Figure 13.1-2 Current (July 13 2011) Strut

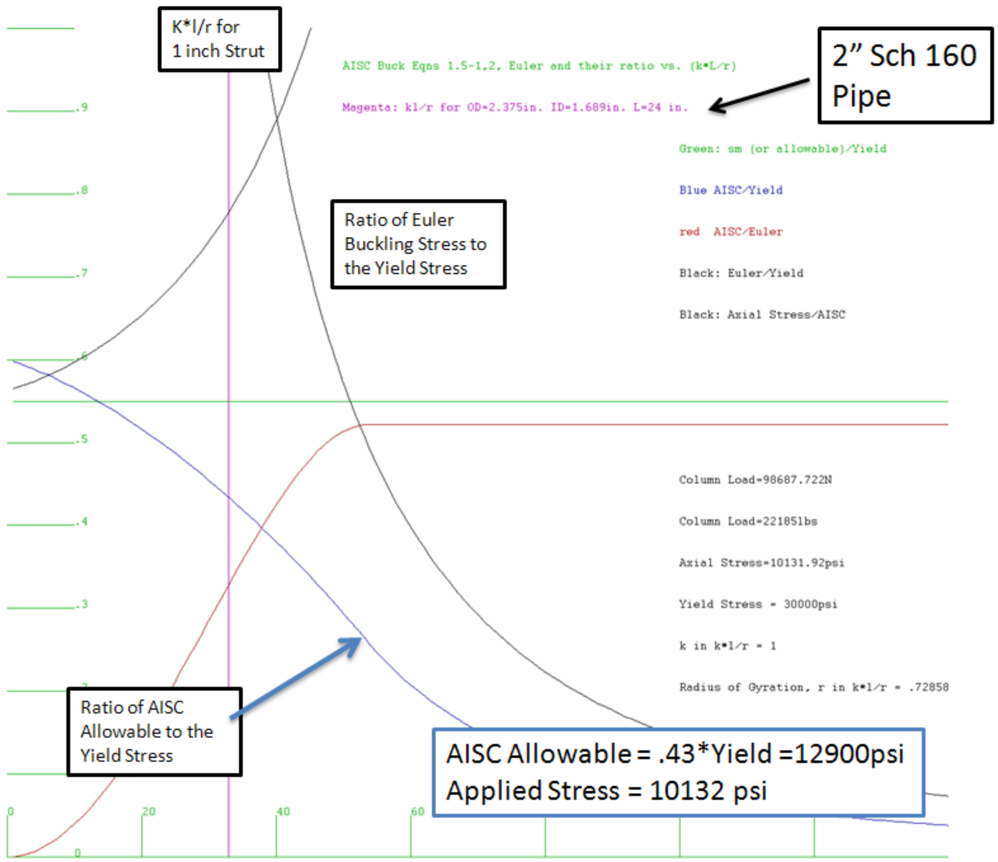


Figure 13.1-3 Qualification of 2" Sch 160 Pipe - Currently (July 13 2011) Not Chosen because the strut ends would have to have plugs welded in and a solid 2" OD rod can simply be drilled and machined to take the male spherical ball ends.

14.2 Bent Strut

There are two locations where the TF OOP struts intersect the vessel support bracket/chair. A special bent strut was investigated to clear the vessel support "chairs" The 2 inch schedule 160 pipe is overstressed with the bend. A solid bar with the same OD didn't do much better. The bent strut will have to have a larger OD to pass the stress criteria. Instead of a bent strut it was decided to design and analyze a special bracket with a cut-out. The concept is shown in figure 14.2-2 and is analyzed in calculation # NSTX-U CALC 12-10-00 "Redesigned Vessel Support Bracket", by Peter Rogoff [10]

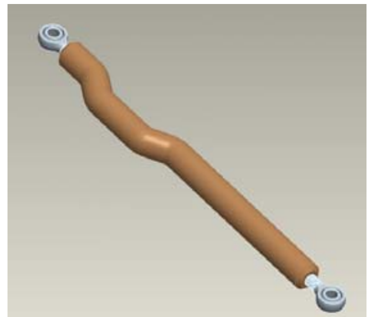


Figure 14.2-1

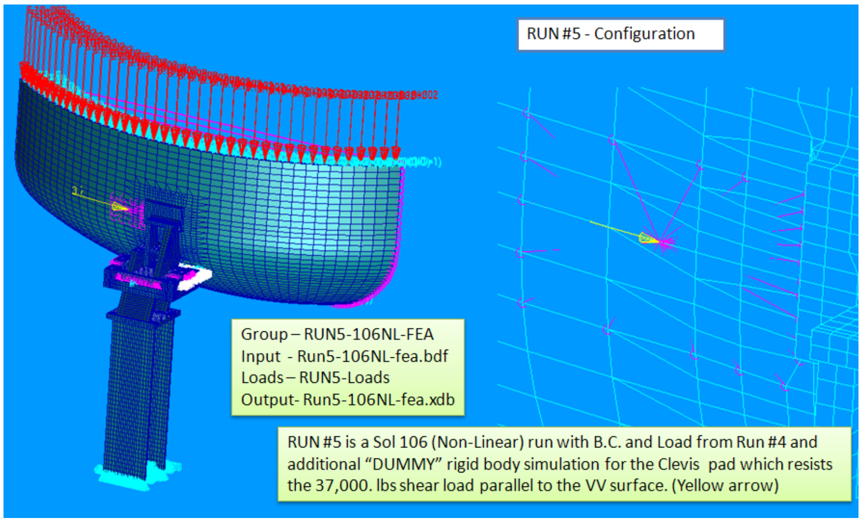


Figure 14.2-2 Special Vessel Support Bracket [10]

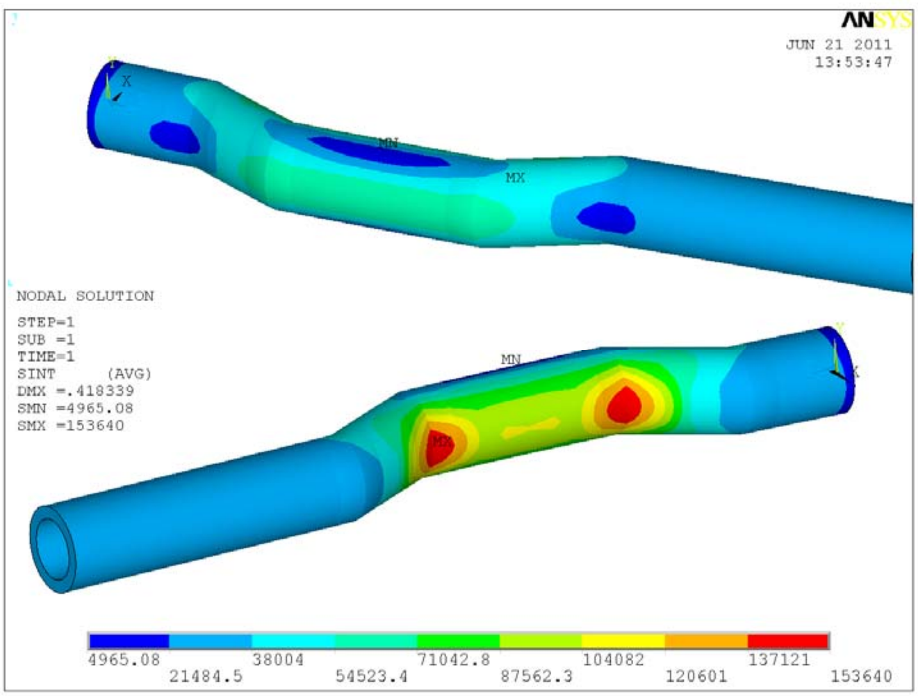


Figure 14.2-3 Two inch Schedule 160 Pipe

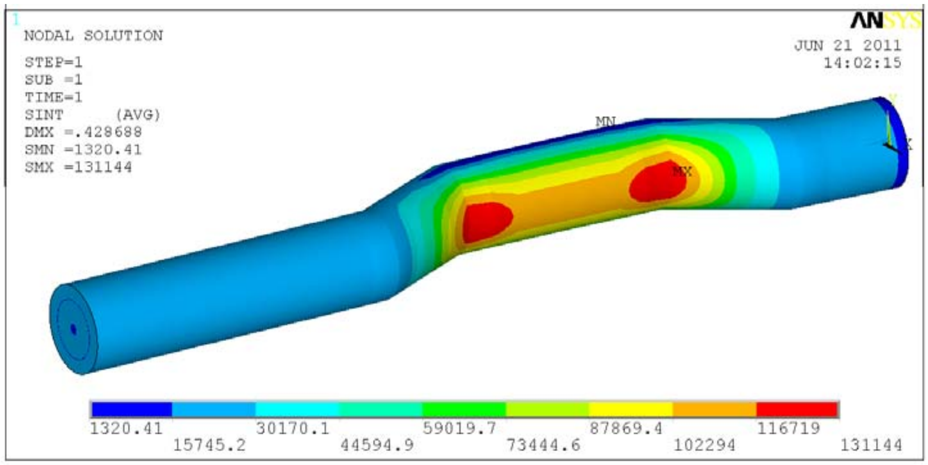
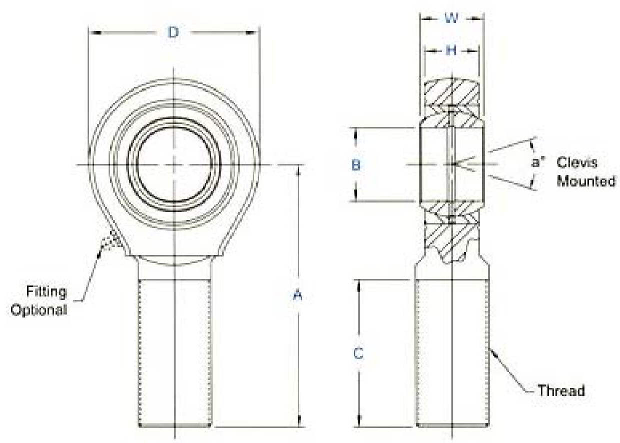


Figure 14.2-4 Solid Bar with same OD as 2 inch Schedule 160 Pipe

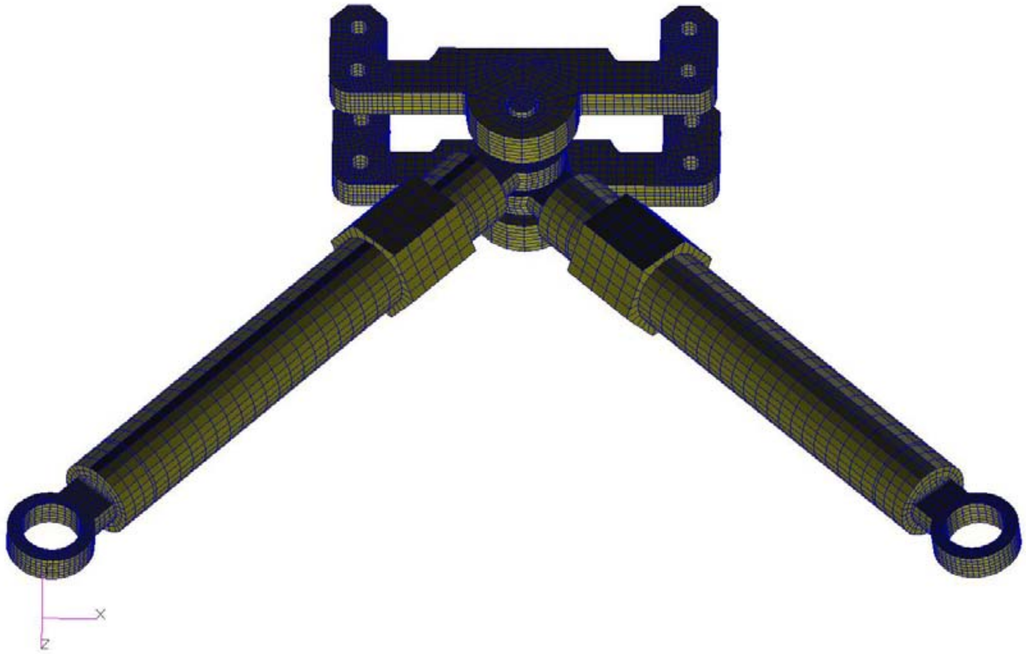
15.0 Spherical Ball End



This usual configuration of a ball end exposes the thread to the cyclic loading. in the strut. To improve this the diameter of the threaded end was increased to the ID of the pipe strut.

16.0 Strut Stiffness Study (by Pete Rogoff)

As shown in Figure 13.1-1 struts are used to connect the Vessel to the TF coil outer leg clamps. The latest strut design is presented here.



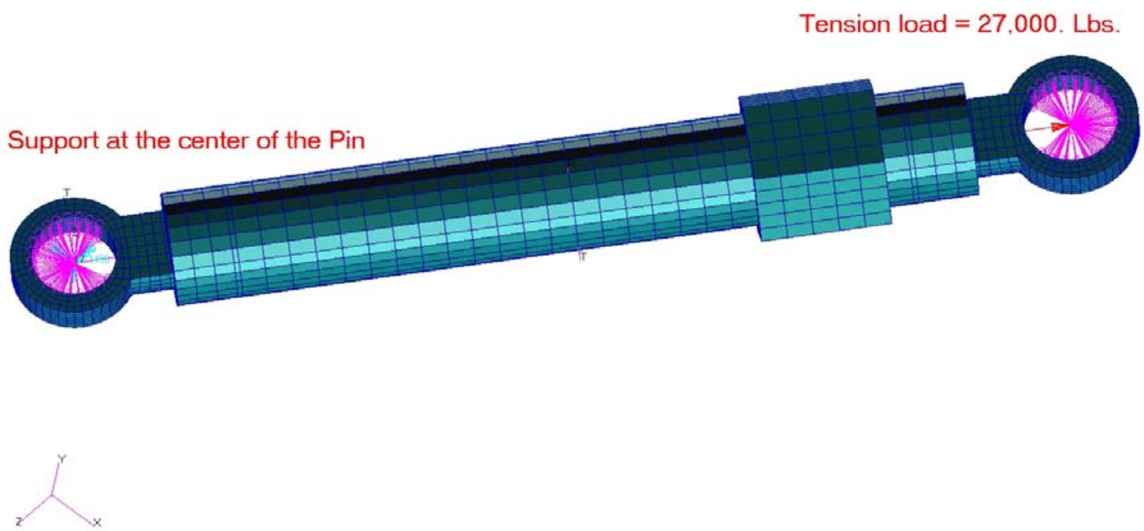
Note: A single 1.25 inch diameter In718 Pin is required at the coil clamp assembly. At the vessel supports 1.0 inch diameter pins are used.
General forces, through the coil/clamp structure, tend to stretch and rotate the assembly, putting one strut in tension while the other goes in the compression mode. Since the actual forces are carried through the single pin at the clamp side, it was prudent to calculate the possible strut spring rates for the given design. The actual spring rates, used in the Global models, should simulate the combined contributions of the Struts and the Single pin assembly of the present design.

Present complete ANSYS Global models predict the following forces:

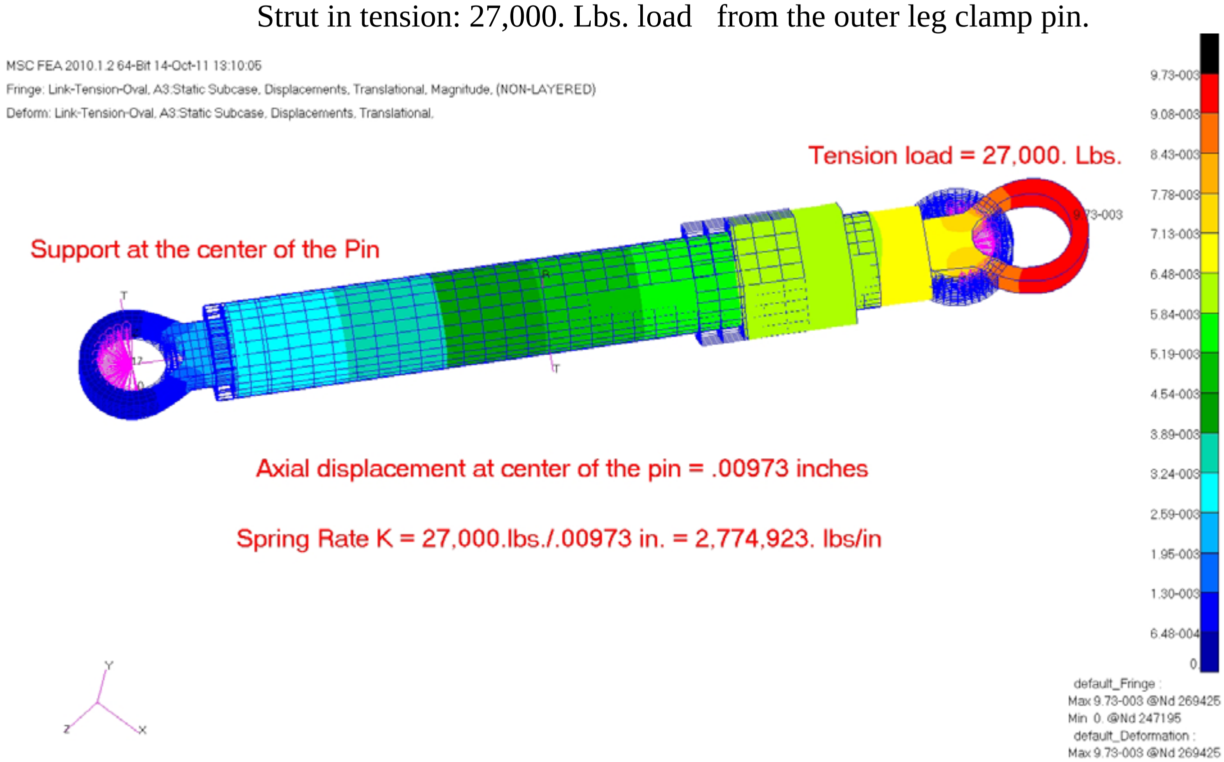
For the Strut in tension, Axial force = 27,000. Lbs.

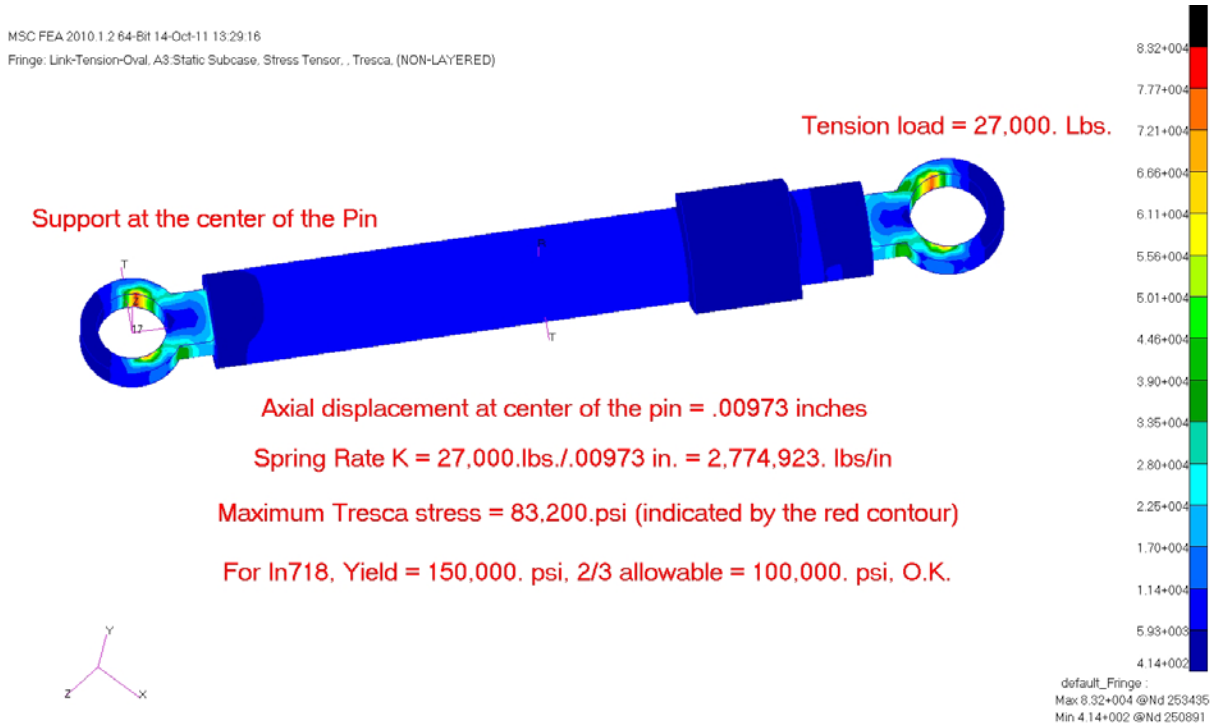
For the Strut in compression, Axial force = -15,000. Lbs.

These loads are also used in the actual coil clamp single pin design calculations which are the subject of the separate number (NSTXU-CALC-132-12-00). Therefore, calculating the stresses and the spring rates of the present strut design is important and is the subject of this Section.

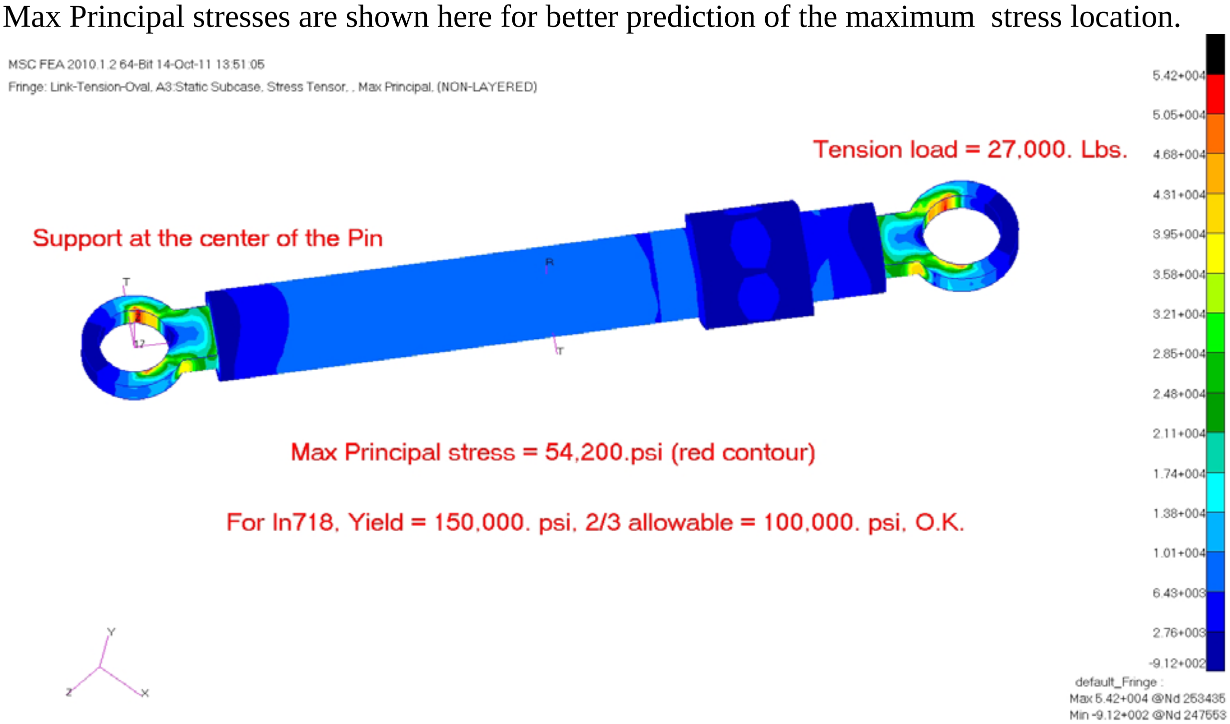


Strut in tension: 27,000. Lbs. load   from the outer leg clamp pin.

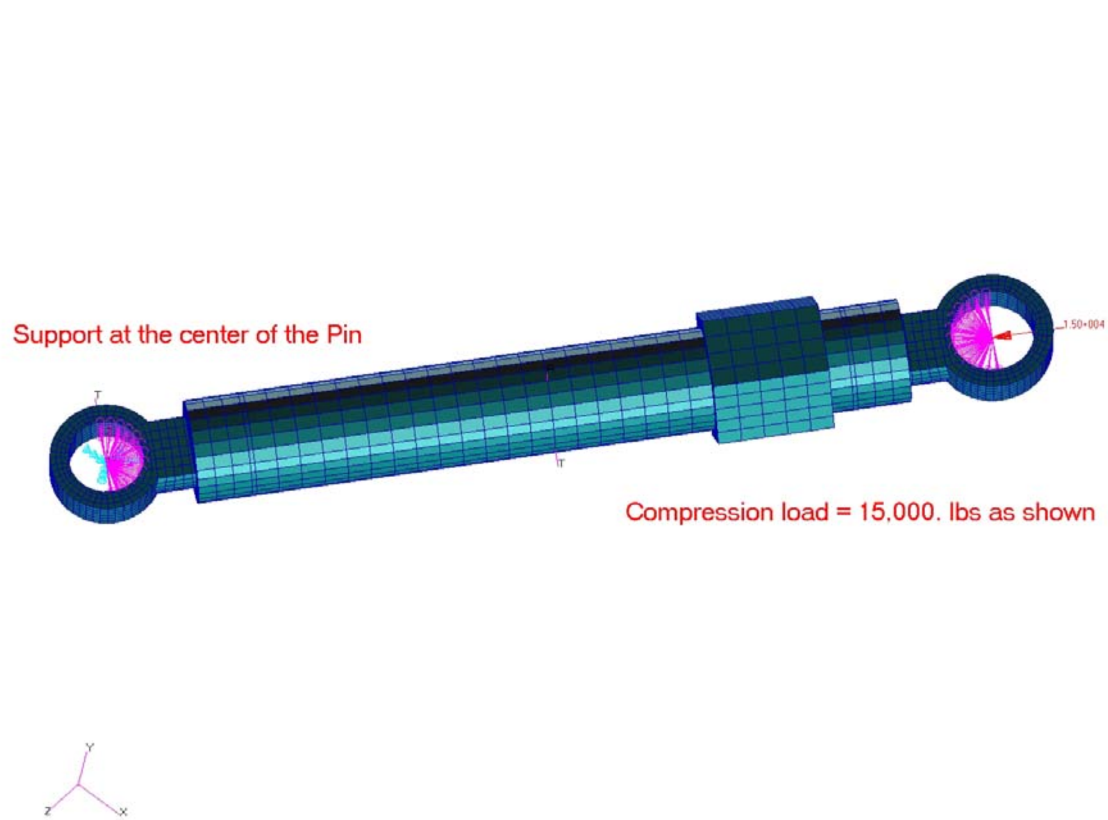




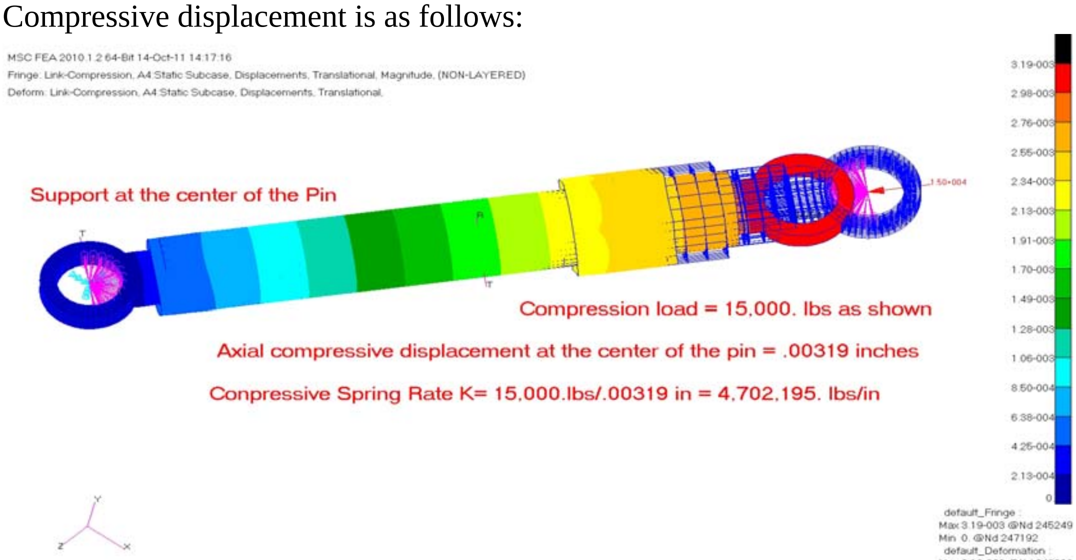
Max Principal stresses are shown here for better prediction of the maximum stress location.



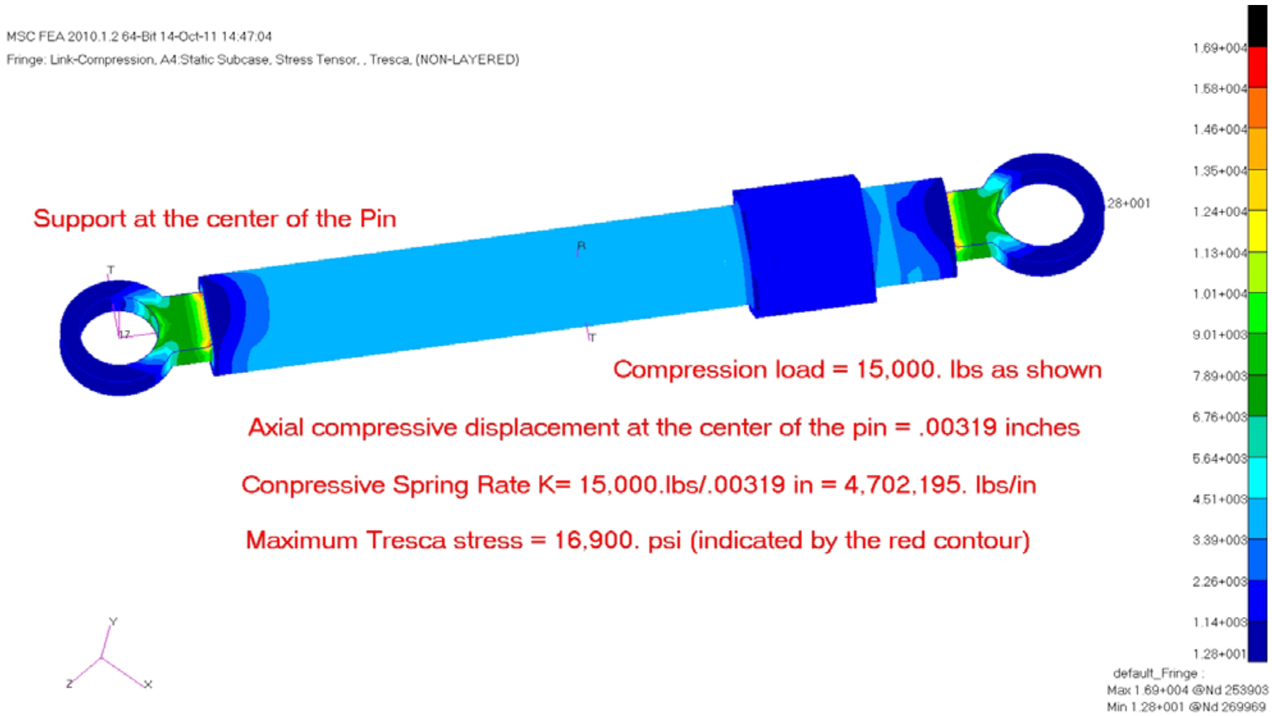
For compression calculations the simulation model is as follows:



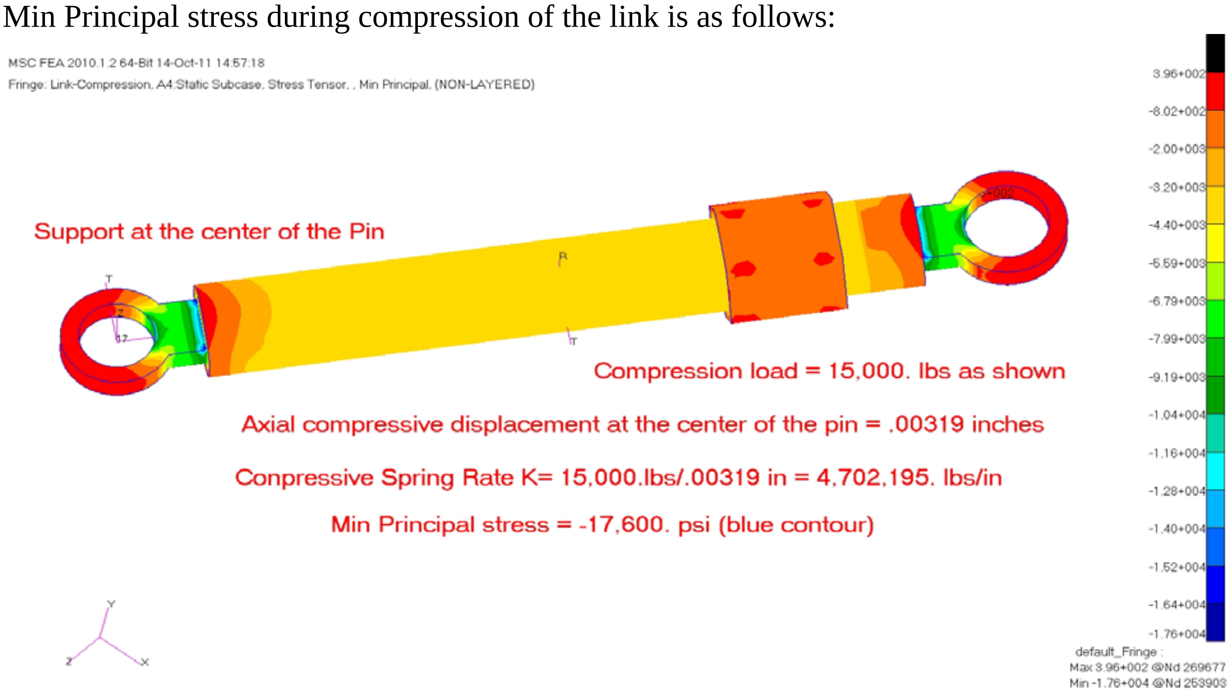
Compressive displacement is as follows:



Maximum Tresca stress during compression of the link is as follows:



Min Principal stress during compression of the link is as follows:



<u>Summary</u>

	Displacement Inches	Max Stress Tresca Psi	Max/ Min Principal Psi	Spring Rate Lbs/in
Tension	.00973	83,200	54,200	2,774,973
Compression	.00319	16,000	-17,600	4,702,195

Conclusion

The Link material is In718 with the Yield = 150,000.Psi and allowable of 100,000.Psi, this present design is adequate for the estimated load conditions. All the calculated stresses are well within the required allowable. It must be noted that, the calculated spring rates are not the total (actual) between the Vessel and the TF coil outer leg clamps. The actual connecting spring rate must include the bending effect of the 1.25 inch diameter pin at the coil clamp. Additional analyses will have to be performed for this condition if required.
Aurora Ball insets will be press fitted in to each end of the link. This process, based on the tolerance values will create a sort of preload as the hoop stress. This action will add stresses at critical locations. It is difficult to estimate these values. This is important in the link tension case, but about 17,000.Psi safety up to 2/3 yield allowable is available. So this may not be a problem.

17.0 Evaluation of the Existing Hardware for the Upgrade Loading

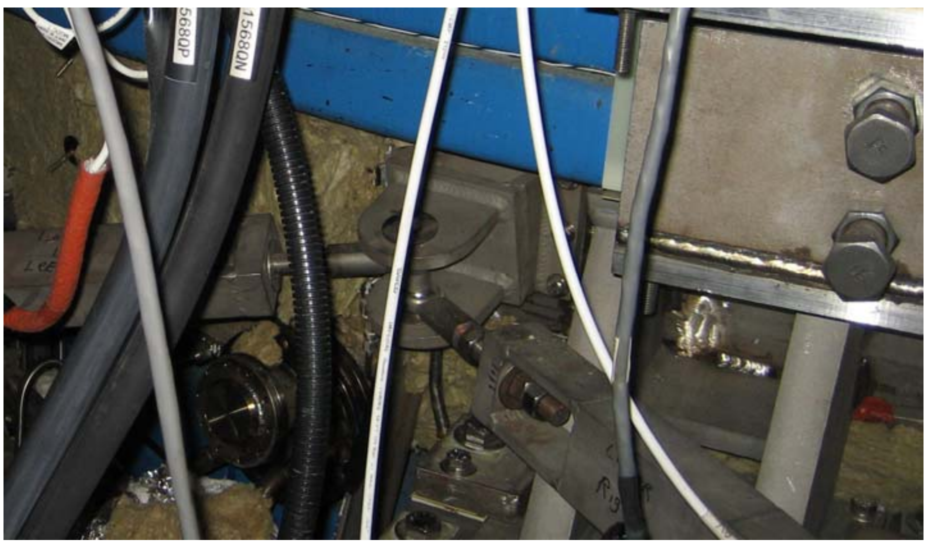
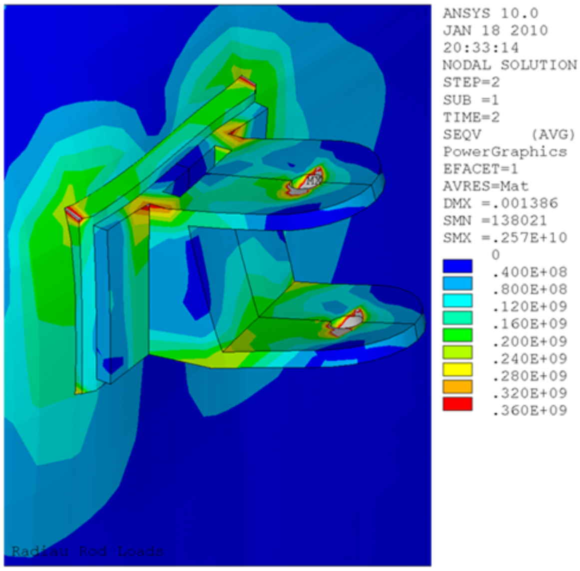
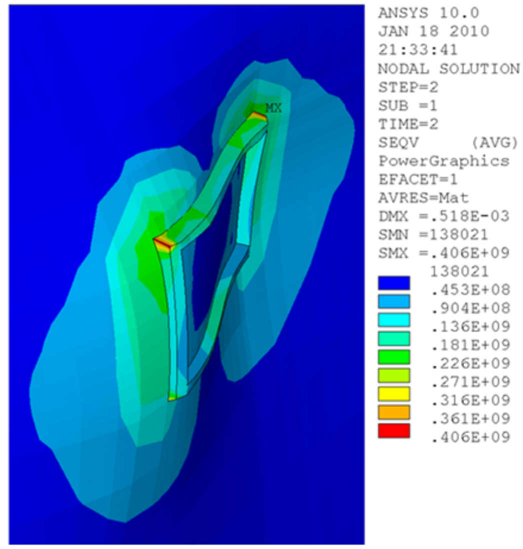


Figure 13.0-1One of the existing Clevis Attachments to the vessel

The truss or radius rod load was taken from Han Zhang's CDR analysis of the outboard legs, Reference [1].
For early analyses, C. Neumeyer provided a couple of sets of currents representing the worst up-down symmetric loading and the worst up-down asymmetric currents. For the symmetric currents, the max load in the truss/radius rod is 18.4 k lbs and min load is 4.5 k lbs. For the asymmetric current, max load in radius rods is 20.3 klbs and min load is 4 klbs. Max load in the ring (in the middle of the ring where connects to radius rod): 86 KN or 19.3klbs for the asymmetric PF current, and 80 KN or 18 klbs for the symmetric PF current."

These loads are derived from "worst case loads that Charlie Neumeyer provided in early 2009. The loads in the radius rods from the 96 scenarios in the global model [2] were also investigated This yielded 24000 lbs. The radius rod loads are reported at the TF outer leg. Global moment summations based on assumed load share between the umbrella structure, knuckle clevis and outer leg mid-plane, produce a somewhat higher load at the clevis radius. In this model, 30,000 lbs is used. In the radius rod design, the truss assemblies attached to the 12 clevises around the perimeter of the knuckle region of the vessel act to cancel the radial loads and only the tangential 30kip load remains, but this is offset from the surface of the vessel by about 4 inches.





The resulting stresses in the sharp geometries of the attachment welds are high. The truss/radius rod clevis was modeled based on the original 2D NSTX drawings. Simple moment summations and spreadsheet calculations showed that the 3/8 inch attachment bolts were undersized for the upgrade loads. The FEA model was then built assuming the clevis assembly would be welded to the vessel pad. A perimeter of elements model the weld and the size is selected arbitrarily and then scaled to the actual or desired weld dimension.

18.0 Bake-Out Thermal Stresses

During Bake-Out, the Clevis is cooler than the vessel shell. It extends beyond the insulation. The existing clevis detail has survived many bake-outs. If the temperature gradient is assumed too steep, the thermal stresses in the weld are excessive. This was considered in more detail for a similar welded pad configuration used to support PF4 and 5. [3] The PF4/5 support pad was instrumented during a bake-out and the max delta T between the vessel shell and flange was noted and used in a thermal stress analysis. The temperature gradients are much more gradual and the thermal stresses are much lower. Stresses were acceptable. The Upgrade TF clevis weld is expected to behave similarly.

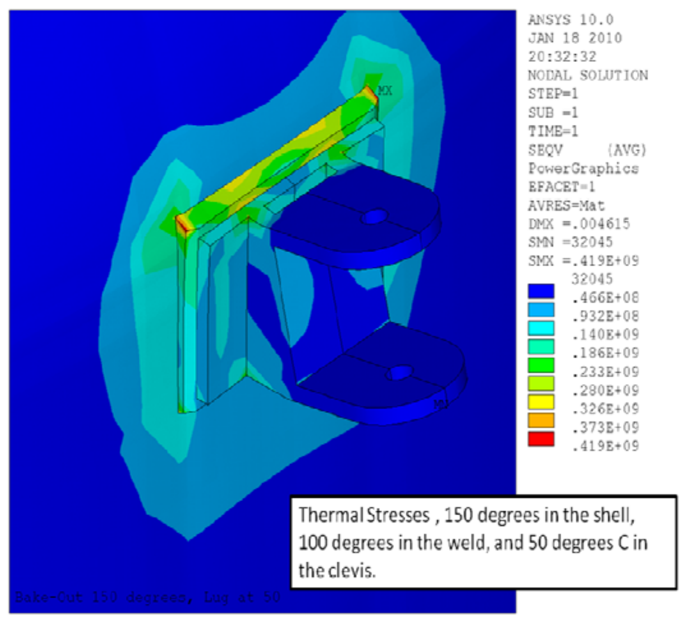


Figure 18.0-1 Bake-Out Thermal Stresses in the Existing Clevis

19.0 Late 2011 Early 2012 Design

This concept employs added groove plates on either side of the existing pad. These are initially positioned with the clevis as a fixture. Studs which have been shot onto the vessel wall are tightened to hold the components in alignment with good fit between the keys and grooves. A perimeter weld is applied. The clevis is then removed to allow the welds between the groove plates and existing pad to be made.

19.1 Model

The model is built from a 1/2 symmetry mesh which is reflected and extruded. The 2D mesh was built off of an iges file provided Mark Smith which was used for dimensions and was meshed outside ANSYS.

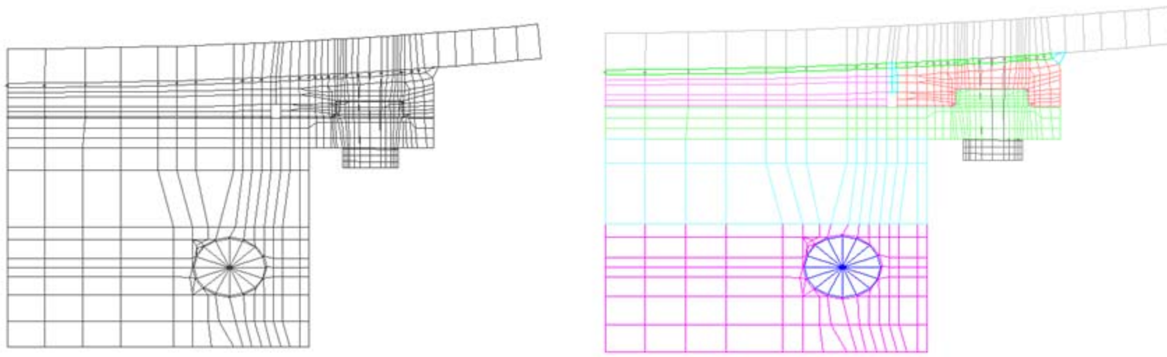


Figure 19.1-1 2D Mesh Used as the Basis for the Swept Mesh.

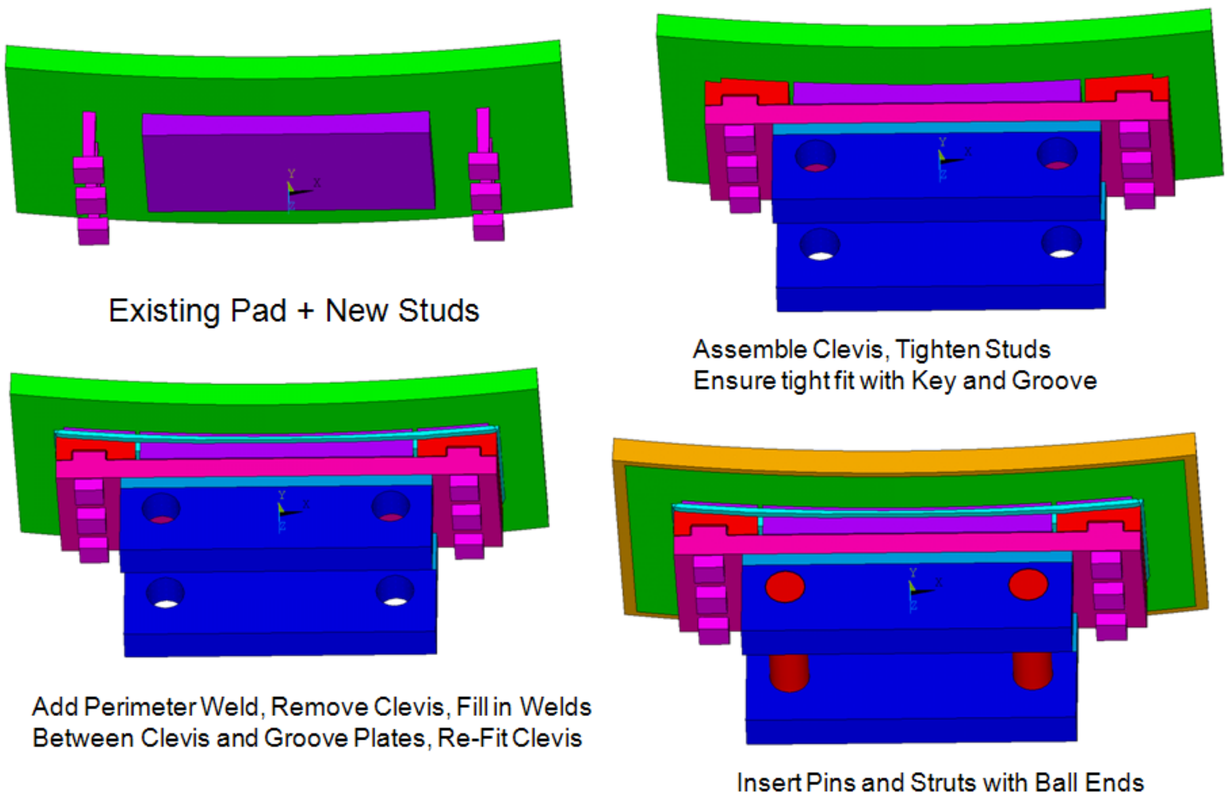


Figure 19.1-2 Bolted Clevis with Added Groove Plates

This concept has the lines of action of the struts intersect at the vessel surface. Inclined plates are also utilized to eliminate bending in the plates. This produces mostly shear at the vessel surface that is well reacted by the tight fit grooves and keys.

19.2 Plate and Assembly Results

This section reports results for the model with a 7100 lb stud preload. This may be too high for 1/2 inch studs. The analysis was run with 3623 lb stud load as well. This is discussed in section 9.3. The 316 stainless steel has fatigue allowable of 300 MPa or 43509 psi. The weld allowable is 14 ksi. Most of the clevis details do not approach the allowables - with the exceptions of the pin and notch/key corners.

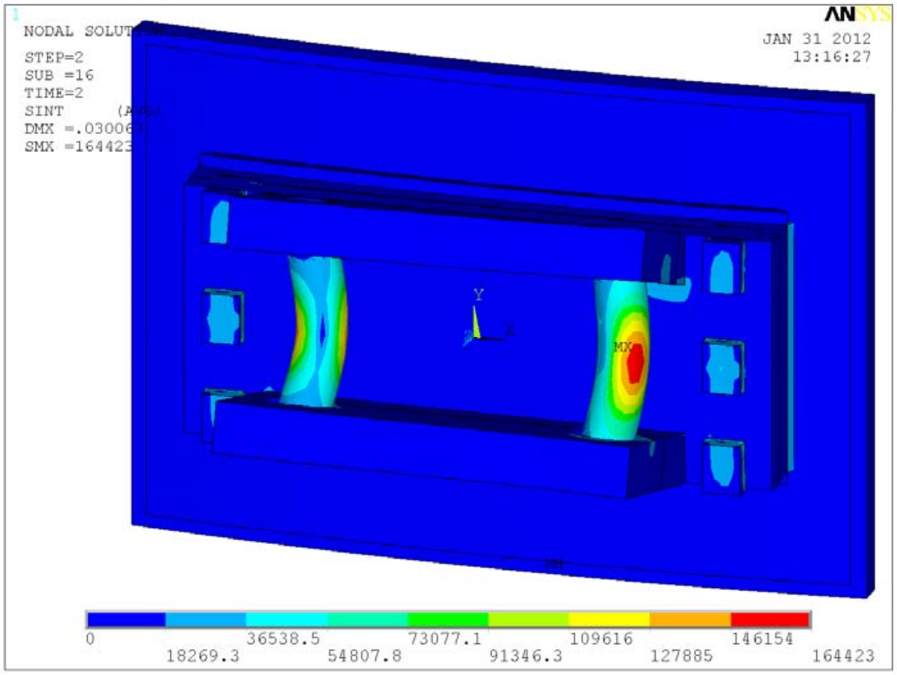


Figure 19.2-1 Bending Stress in the Pins

Figure 10.0-1 and Figure 13.0-3 show earlier clevis pin analyses. The allowable stress for the 718 pins is 90 ksi. The pin bending needs to be substantially reduced to satisfy the fatigue allowable. 164ksi is above yield for 718 of 150 ksi - so the pin as it is currently loaded will not pass a static allowable. For a ball end with close fitting clevis plates, the pin stress would be 75 ksi or less as shown in figures 10.0-1 and 13.0-3.

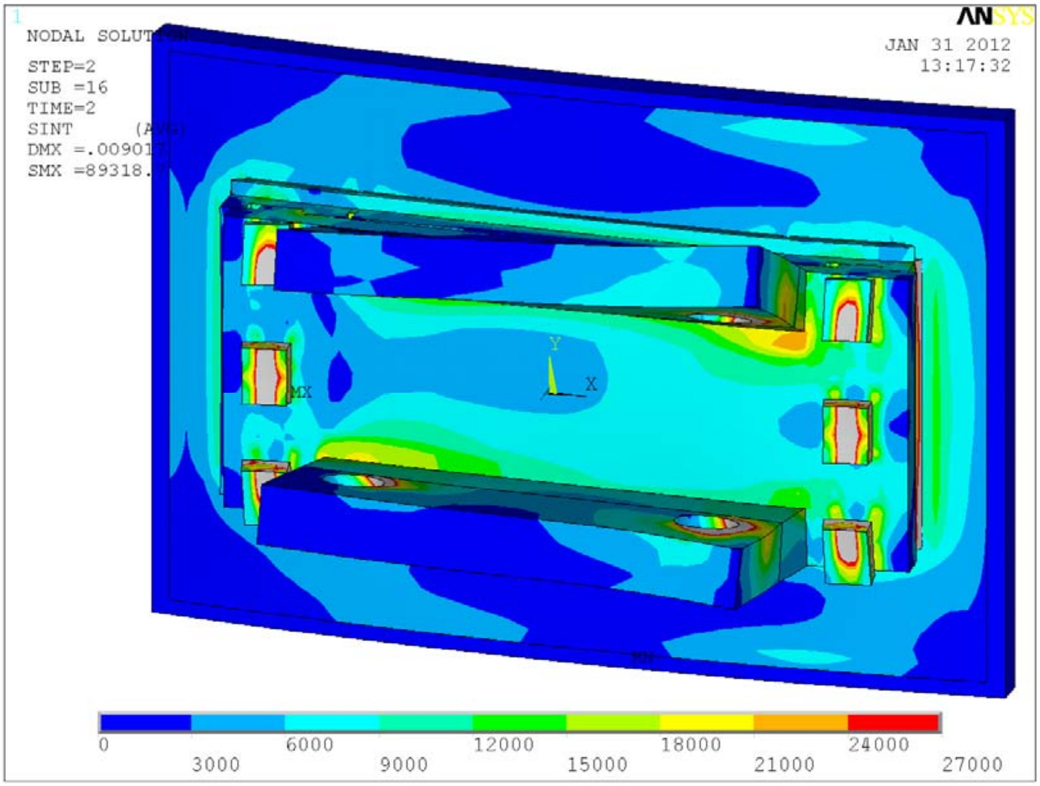


Figure 19.2-2 Stress in the Assembly with the pins Removed and the stress Contours Set at a Maximum of 27 ksi.

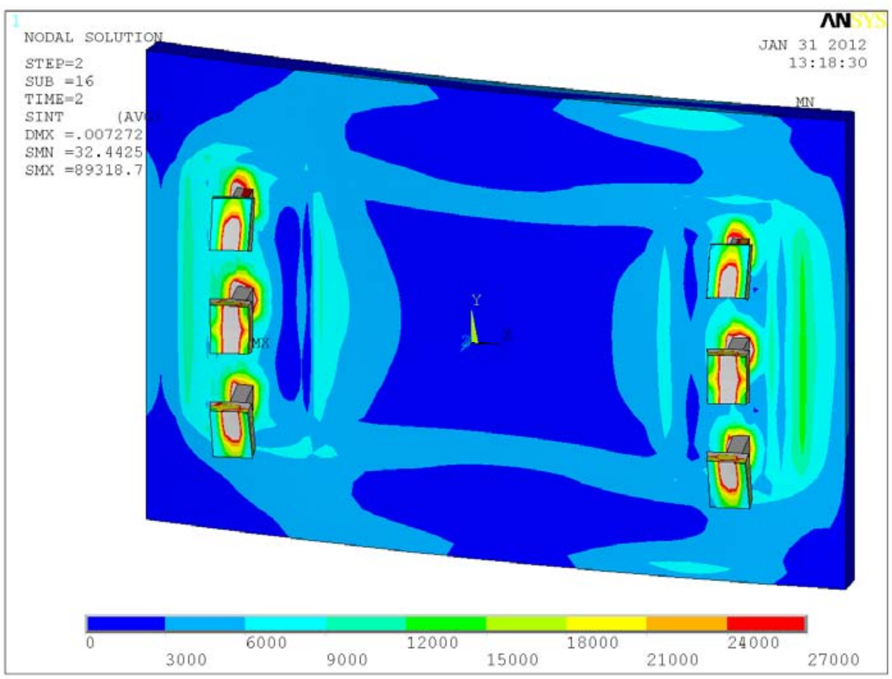


Figure 19.2-3 Stress in Vessel Wall With Contours Set at a Maximum of 27ksi

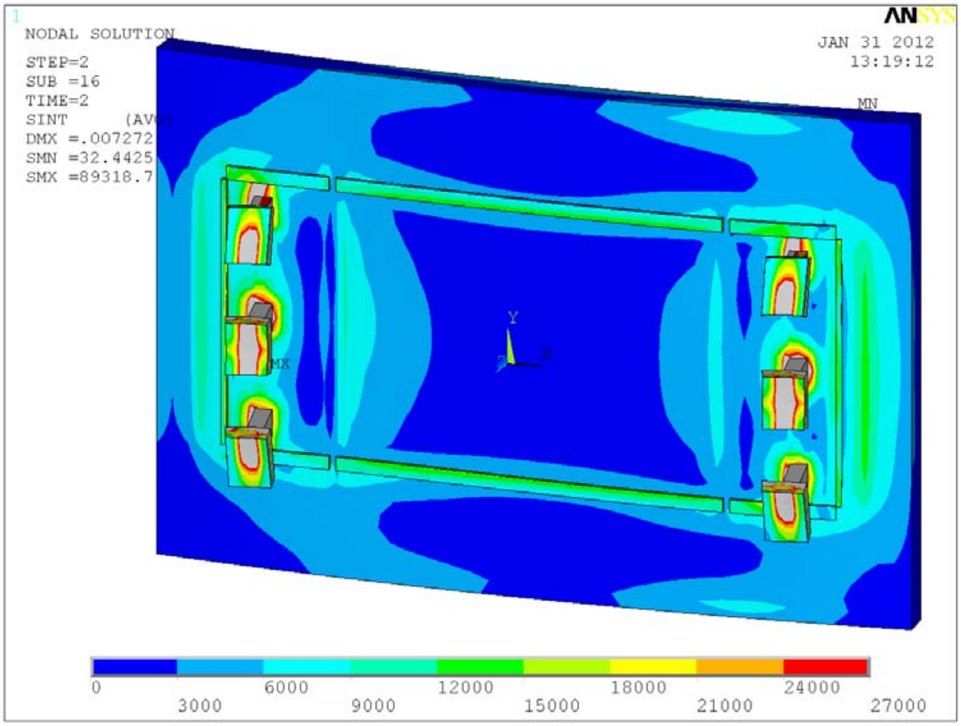


Figure 19.2-4 Stress in Vessel Wall and Weld With Contours Set at a Maximum of 27ksi

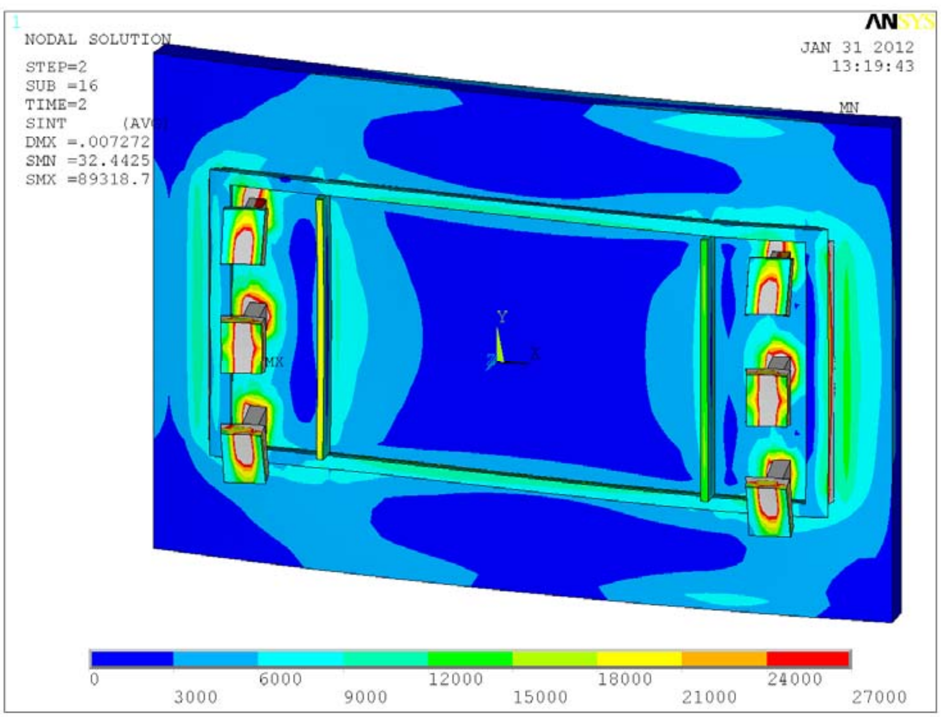


Figure 19.2-5 Stress in Vessel Wall and Weld With Contours Set at a Maximum of 27ksi

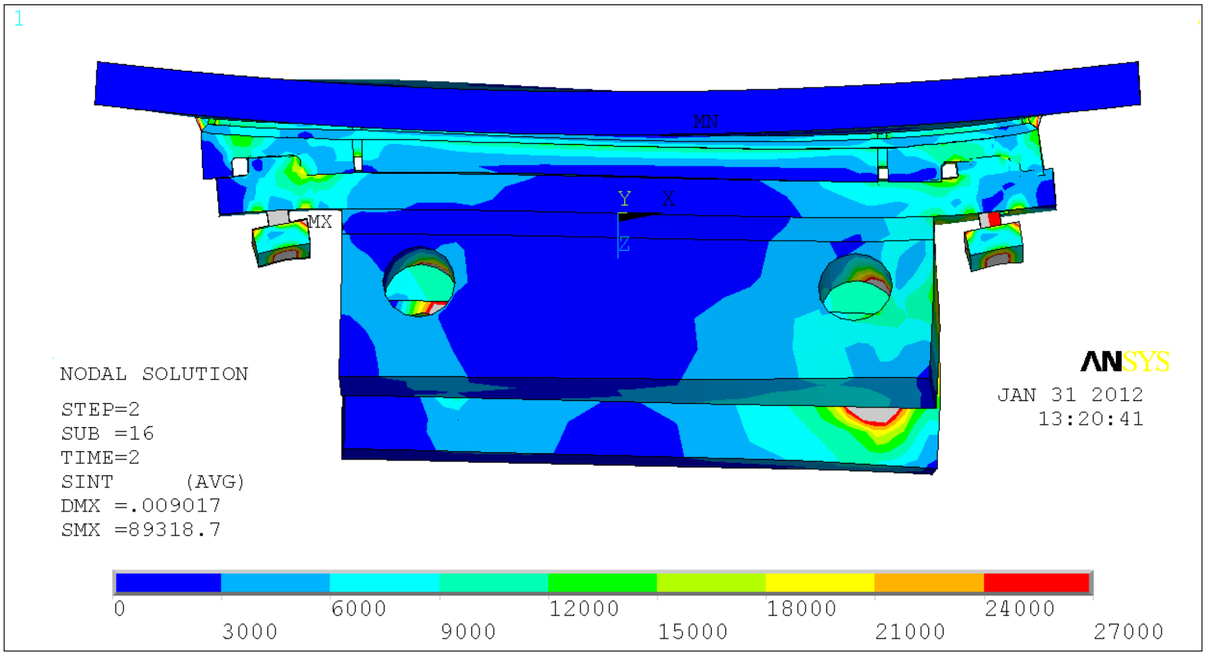


Figure 19.2-6 Stress in Vessel Wall With Contours Set at a Maximum of 27ksi

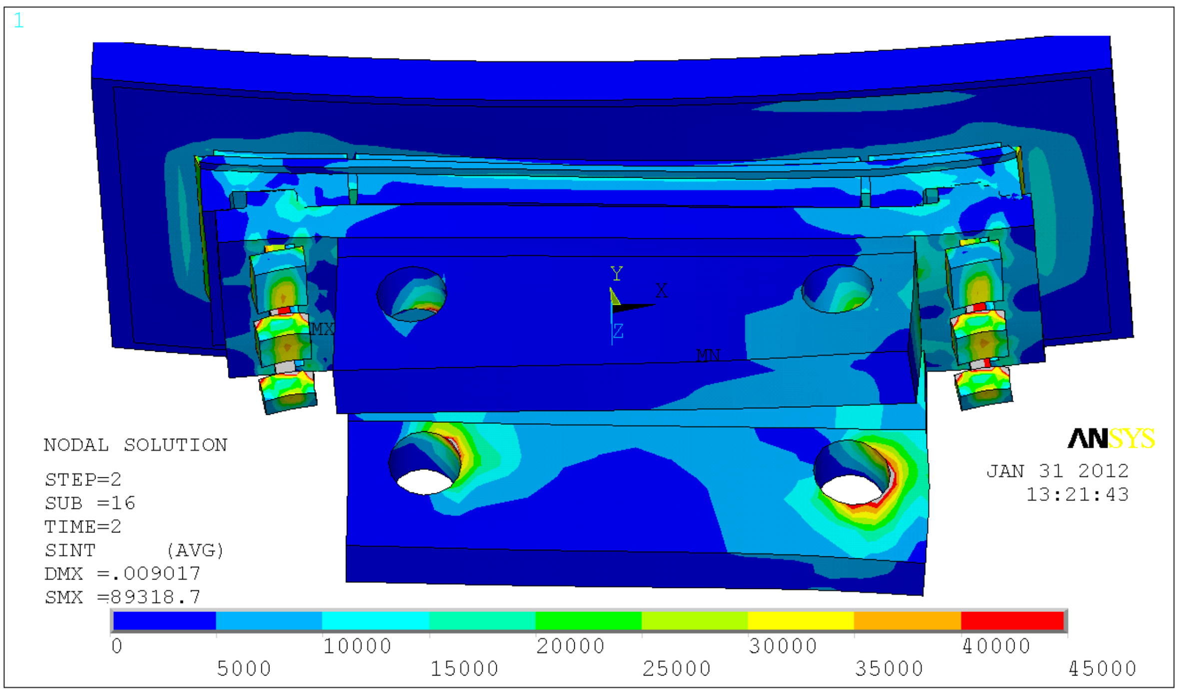


Figure 19.2-7 Stress in Vessel Wall With Contours Set at a Maximum of 45ksi

19.3 Stud Preload

Stud Preload is applied with interference in the gap elements under the head of the stud "head" or nut. The mesh was generated from a swept rectangular geometry so the bolts come out with a rectangular cross section Only the average axial stress has meaning. This is used to calculate the stud load. Two stud pretensions are presented. One with 7100 lbs and another with 3624 lbs. preload.

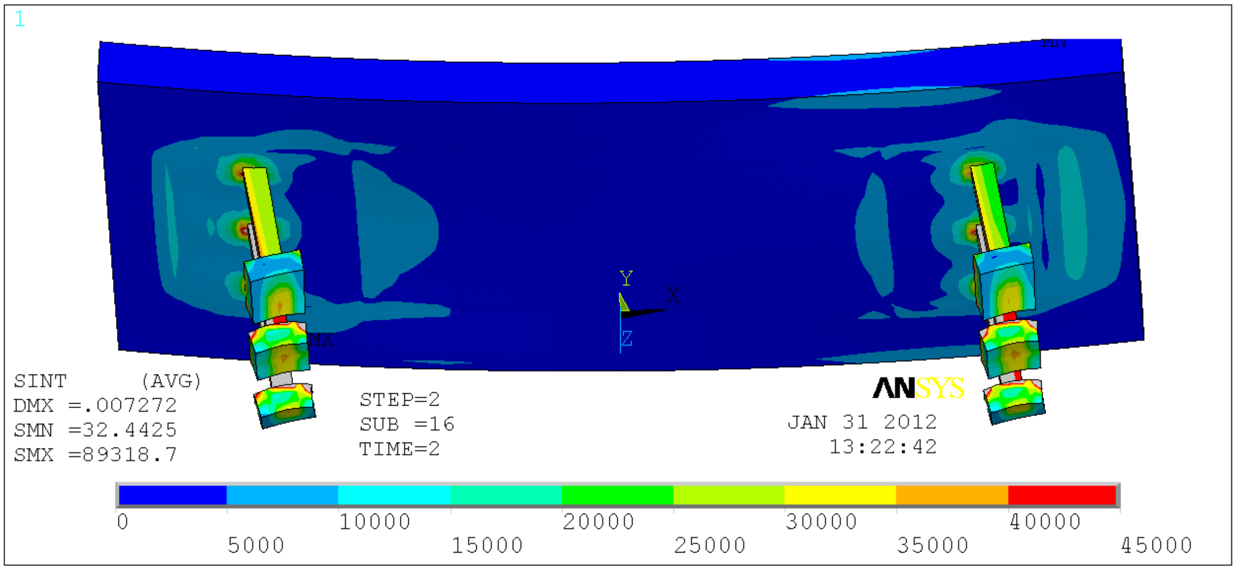


Figure 19.2-4 Stress in Vessel Wall With Contours Set at a Maximum of 45ksi, 7100 lb Stud Preload

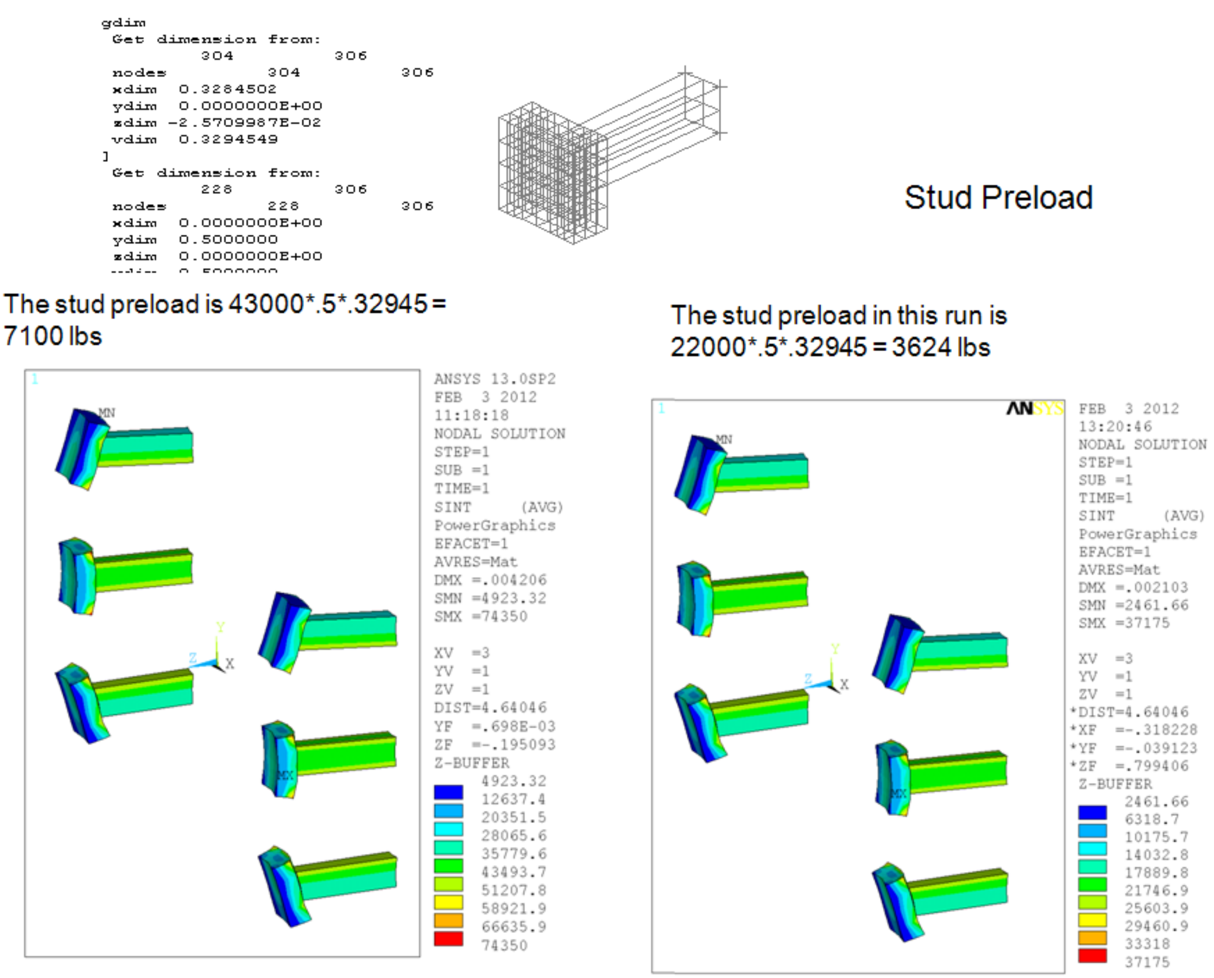


Figure 19.2-5 STUD preload

19.4 Peak Stress and Fatigue in the Tabs and Notches

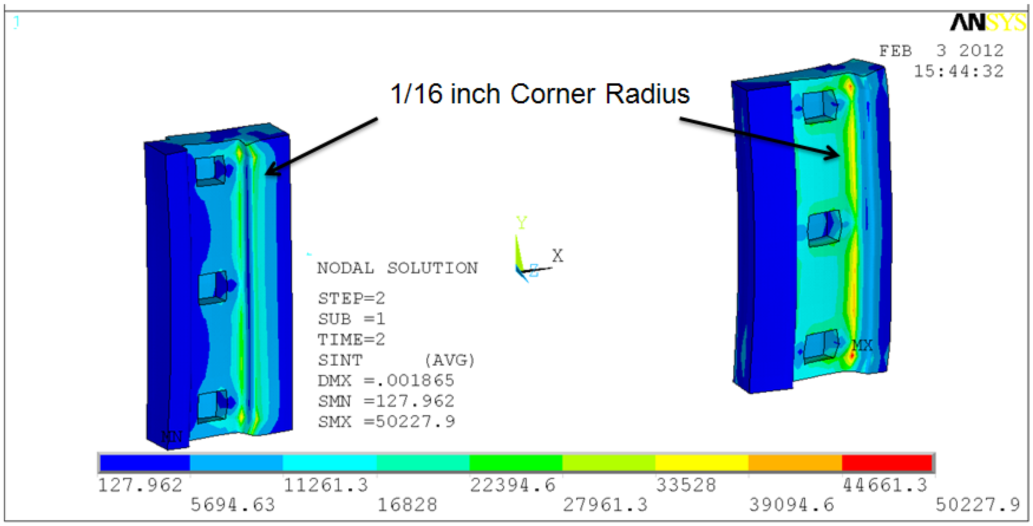


Figure 19.4-1 Stress at Root of Slot in the Extension Pieces, 3623 lb Stud Preload

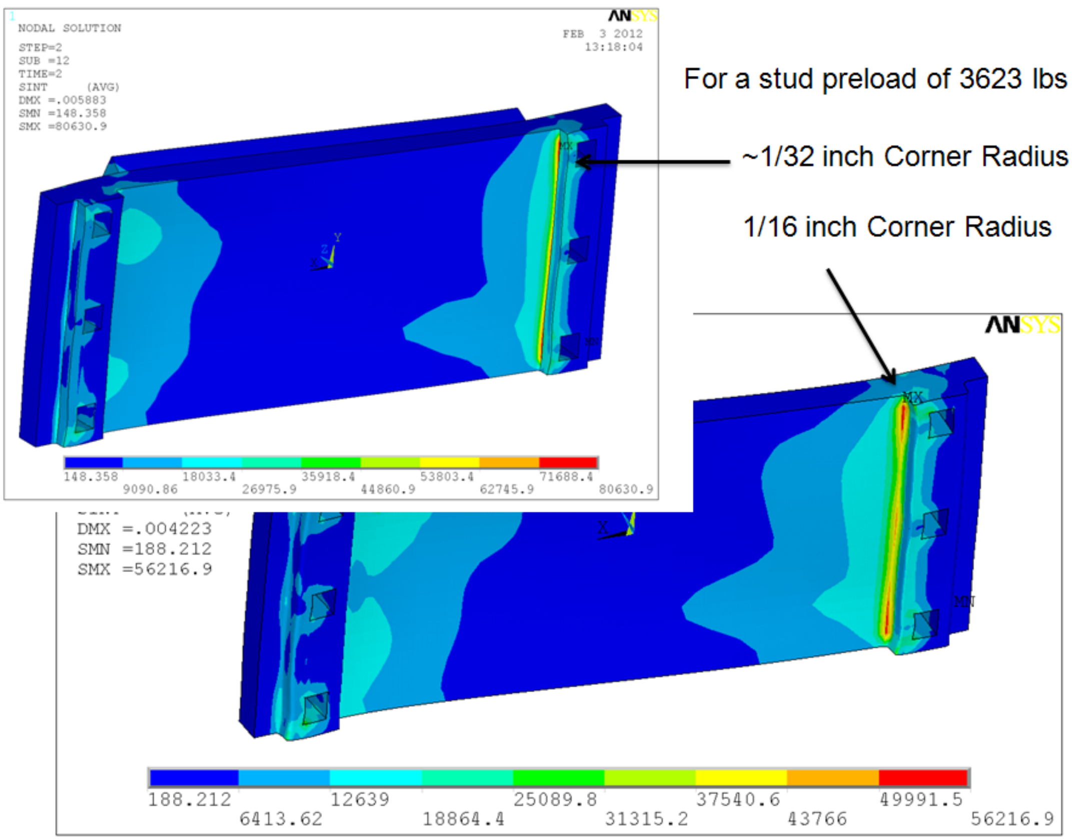
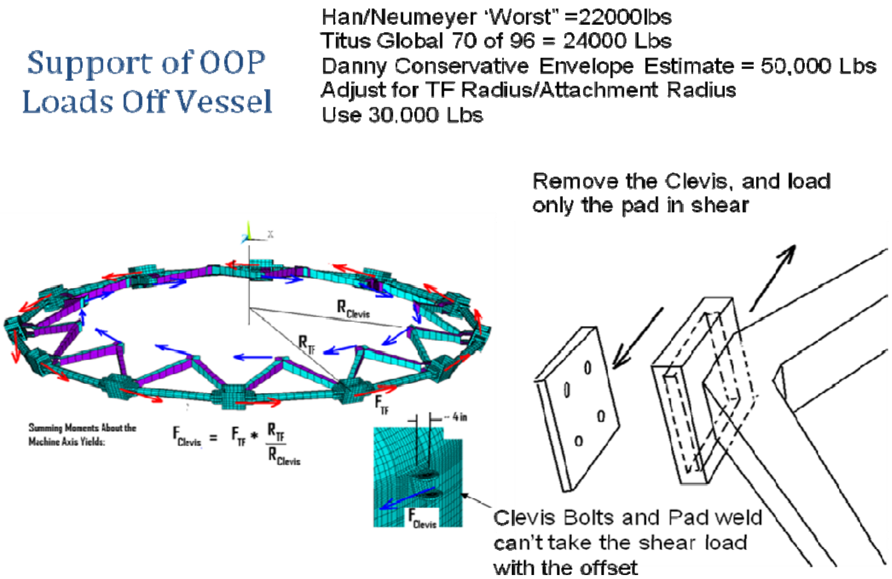


Figure 19.4-2 Stress at Root of the Tab in the Clevis Plate

Figure 8.0-2 shows an SN Curve for 316 Stainless Steel and plots an allowable of 300 MPa or 43509 psi to satisfy the fatigue criteria. With a 1/6th radius in the tab and notch radii, the stress is acceptable for much of the height of the tab and notch. 3/23 radius would be acceptable.

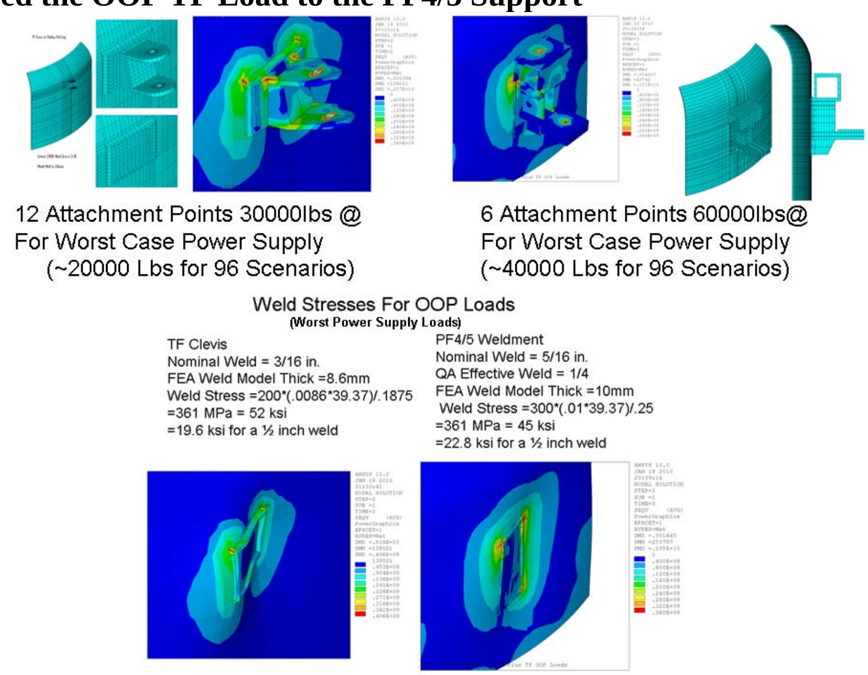
Appendix A
Options that used the existing clevis pads as shear keys



This scheme was attractive because it did not require disconnection during bake-out. It was rejected because the existing pads were not thick enough to be reliable shear keys, and it was judged undesirable to weld on the vessel.

Appendix B

Options that used the existing clevis pads Only for Vertical Support of the PF 4/5 system. and transferred the OOP TF Load to the PF4/5 Support



This concept was a major perturbation of the original support concepts for both the TF OOP loads and for the support of the PF4 and 5 coils. There is a substantial elevation difference between where the TF truss connections are and where the PF 4/5 bracket is connected to the vessel. The TF OOP loads imposed a large torque on the bracket which produced excessive weld stresses. This concept was rejected when the addition of the DCPS allowed use of the existing PF 4 and 5 support brackets to support the PF4 and 5 coils.

Appendix C

Pete,

Mike Bell gave approval for using high strength 440 series stainless for the rod ends.
He determined this based on magnetic permeability data from an ITER part he is familiar with, similar material.

To confirm this, arrangements are being made to have one of the 440 SS rod ends tested for permeability.
Barring some unforeseen result from this test, this is the plan.

So, the design will use a In718, 2 inch solid round with male threaded ends.
The rod ends will be female threaded, 440SS.

See preliminary drawings attached.

Pete,

For rod end, I believe the designer based the dimensions on stock parts, but maybe not.
I'll see.

There are clearance issues (fit up and assembly) which may not allow changing to the full thickness you show.
We'll implement as much as possible.

As far as the pins, 3/4 inch pins will be used for the VV clevis and 1 inch pins at the TFOL clevis.
Both will be In718.
Also, In718 bushings will be used to help strengthen both clevis.

Mark Smith

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Princeton Plasma Physics Laboratory
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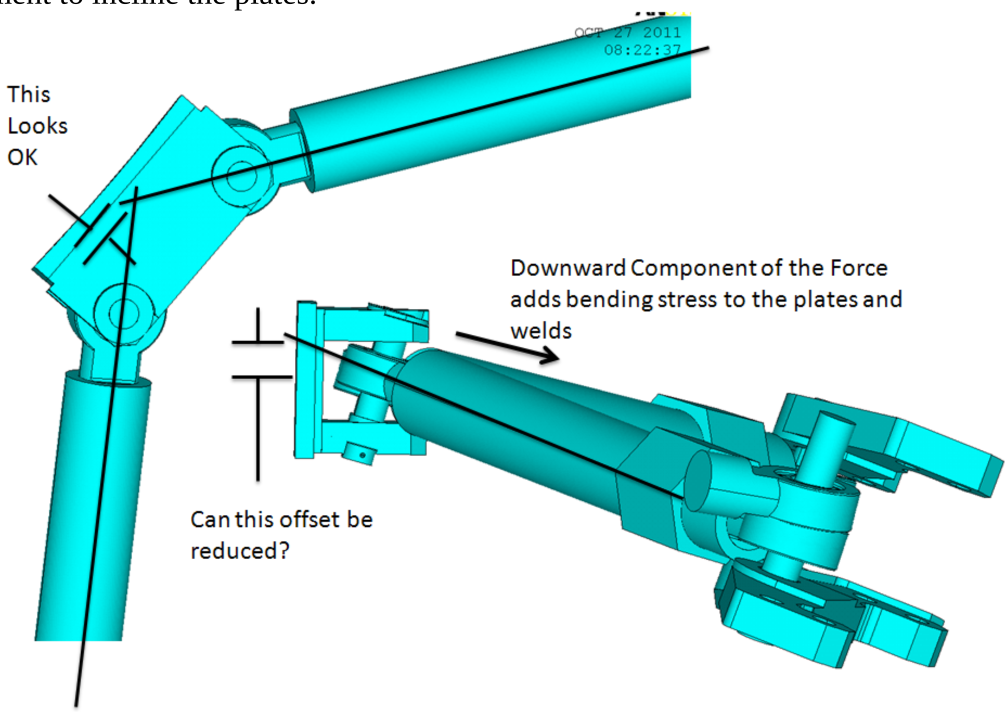
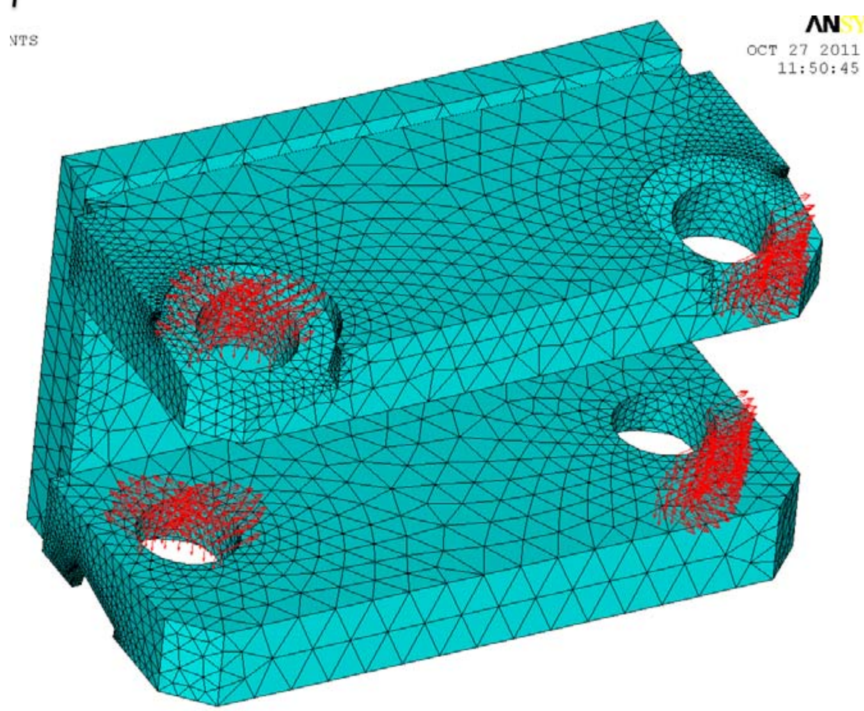
(609) 243-2778

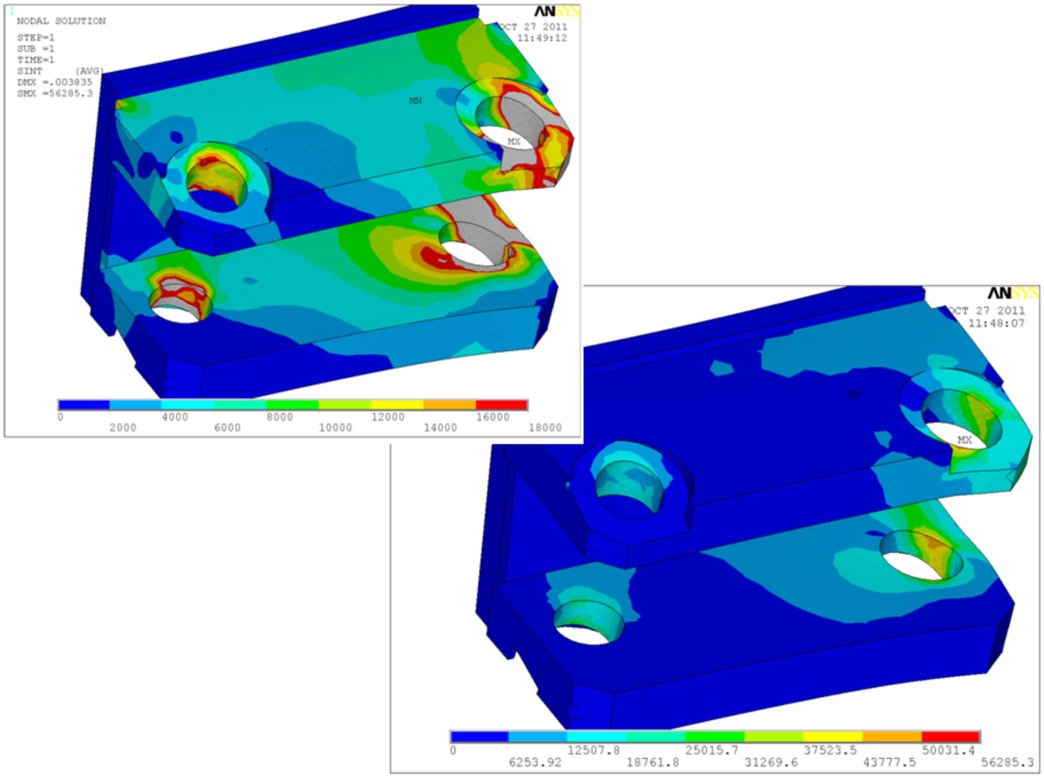
Appendic D

October 2011 Design

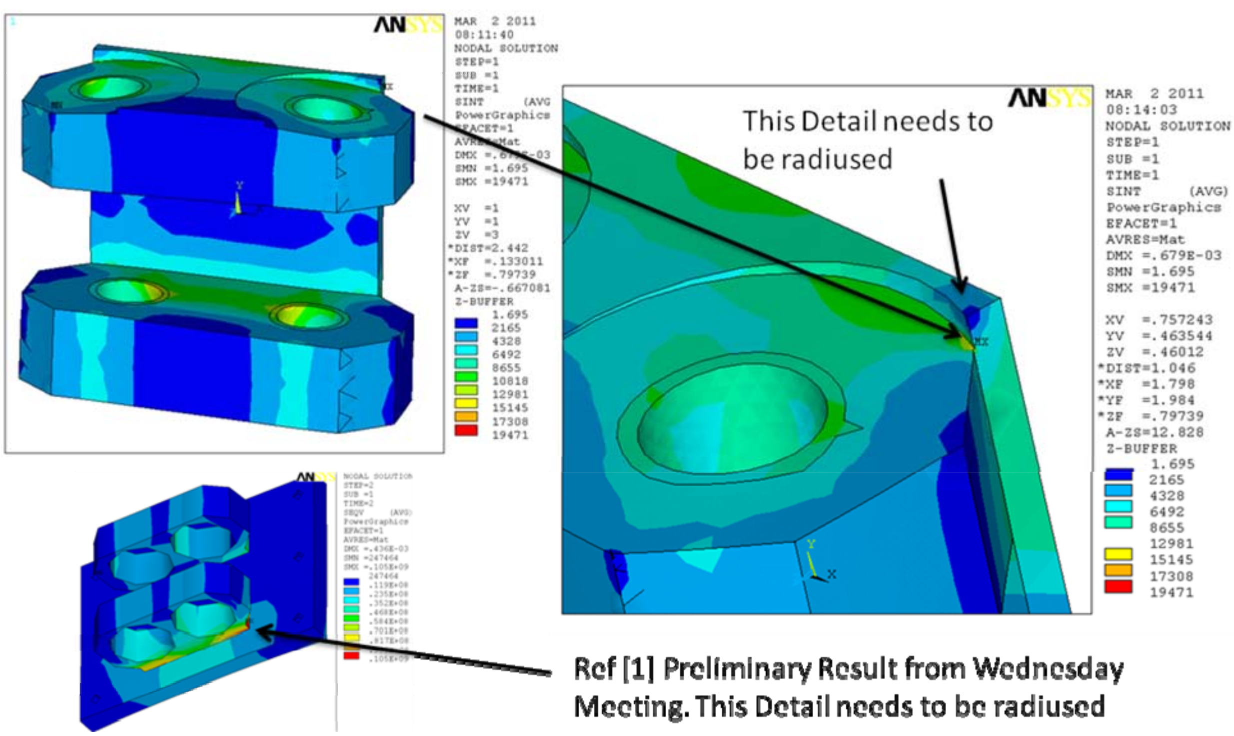
In this concept, there are two plates welded directly to the existing pad. The two separate plates are not as rigid as the other single machined clevis I think there will be some non-uniform loading on the existing pad. There is some un-necessary bending on the plates and welds because they are not inclined at the same angle as the struts. It would seem to be a simple improvement to incline the plates.





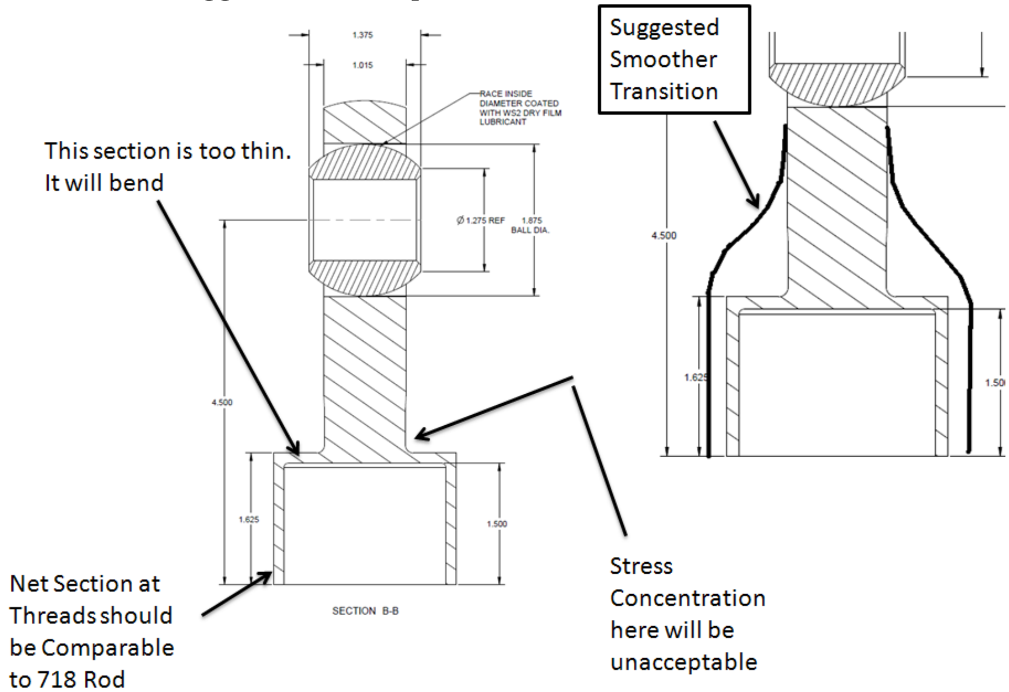


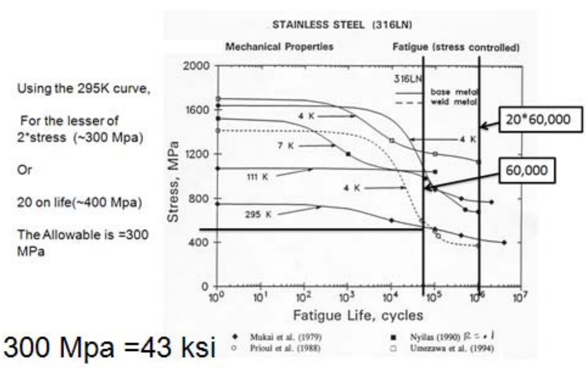
Previous Analysis showed some High Stresses in the Counterbores- These still need to be smoothed out

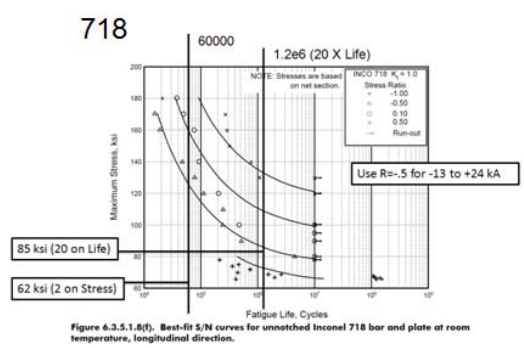


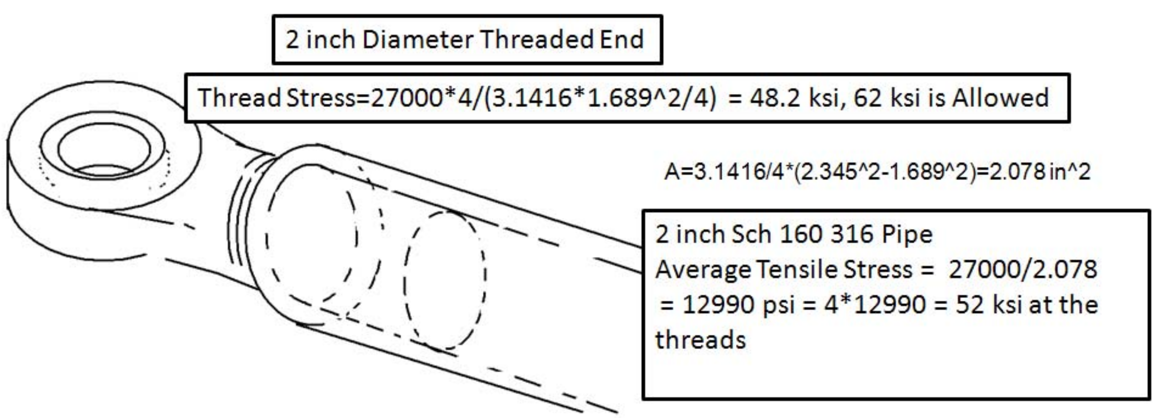
Ref [1] Preliminary Result from Wednesday Meeting. This Detail needs to be radiused

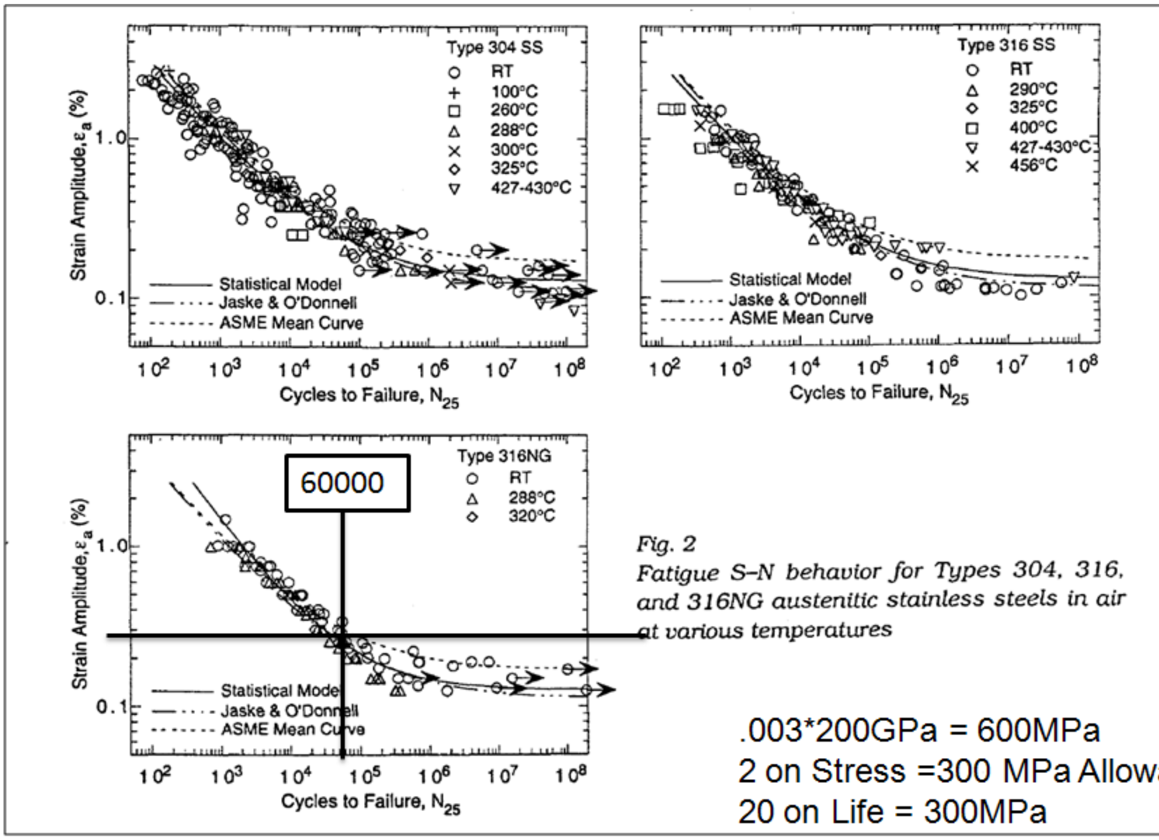
Appendix E
Suggestions for Improvements in the Strut Ball End





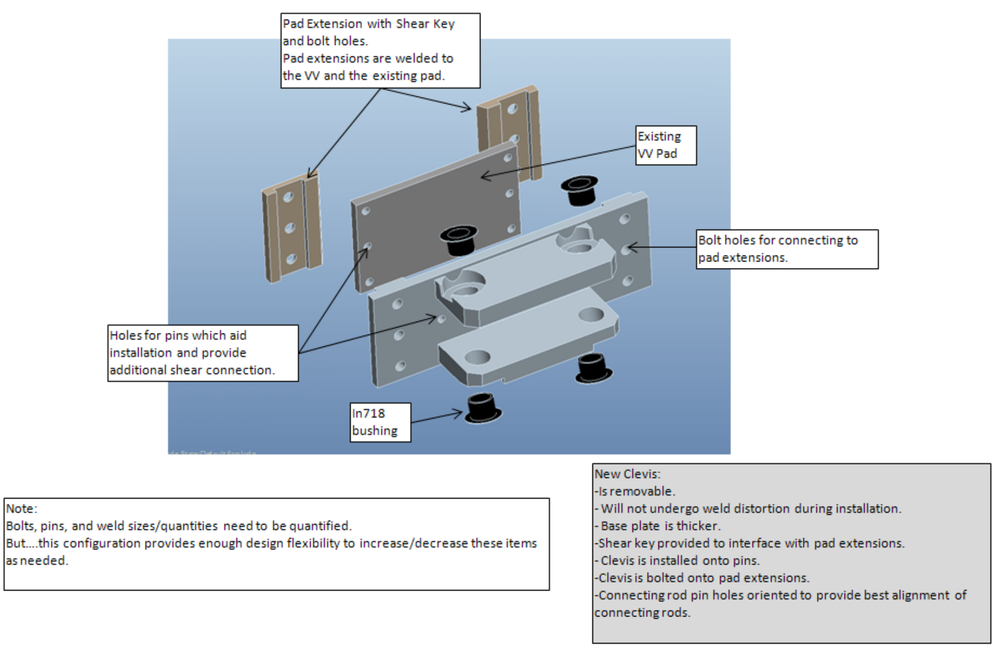






Appendix F

Late 2011 Early 2012 Concept with Stud Connected Extensions (Qualified in its final form in Section 19)



Appendix G
March 7 Email from C. Neumeyer on Post Disruption Torques

Pete, As we discussed a few days ago, I'm working on a revision to the DP spreadsheet to close out the checking exercise and I added the TF torque sums for the cases with plasma. Attached is a preliminary result. New entries are all the way on the right side in blue font. It seems that the presence of the plasma decreases the torque compared to the no-plasma case (which was the only case previously reported). And then, after disruption, the OH and PF currents experience a shift (according to the flux conserving solution) but the torque remains less than the no-plasma case. So, the case previously reported holds up as a "worst case". These results will be formally issued in the next few days. Ch