TOHOKU (30-Inch) BUBBLE CHAMBER MAGNET VACUUM SHELL ANALYSIS

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Introduction

The vacuum shell for this magnet was originally designed as a four sided hollow torus for use with the old 30-Inch Chamber. This shell was designed by "hand" calculations. The new Tohoku chamber required slightly more room. To accomodate this change the inner wall was remade at a larger diameter, and a conical piece was added creating a five sided torus. Slightly lower stresses are found in this new shell because it is smaller and more "circular" in cross section. Figure 1 shows the basic overall dimensions.

Two different ANSYS calculations were run to determine stresses more accurately than those previously calculated. The major loading is either external vacuum or internal pressure loading arising from a failure in the helium system. All electromagnetic loads are transferred from the magnet through the vacuum shell to the iron. Axial loads and the vertical decentering loads do not have any significant effect on the vacuum shell stress levels. The horizontal loads, however, can change the stress distribution and must be evaluated. These horizontal loads are transferred to a post which is welded to the vacuum shell base plate and pinned to the vacuum shell top plate. This pin is welded into the top plate. See Fig. 6, 14 and 15 for the shape of the post and its location on the displacement plot.

A simplified axisymmetric ANSYS model was first run for both internal and external pressure using the axisymmetric conical shell element, SFIF11. Figure 2 to 4 are from this model. This will be compared to and serve as a check on the more complex 3D shell model. See Figs. 5 to 12. Quadrilateral shell elements, SFIF63, and 3D elastic beam element, STIF4, for the posts were selected for this second model. The following load cases were run for the 3D model.

1. 44,000 lb preload on both posts.
2. 37,000 lb decentering force on only the right post.
3. 1 atm internal pressure.
4. 1 atm external pressure.
5. 1 atm external pressure + 37,000 lb decentering force.

It will be shown that neither preload nor magnetic decentering loads have any significant effect upon the high stress regions of the vacuum shell. Therefore, external load plus preload or external load plus the decentering force is virtually the same case. Likewise, there is little diference if internal pressure is considered instead of external pressure. Preload from the inconel

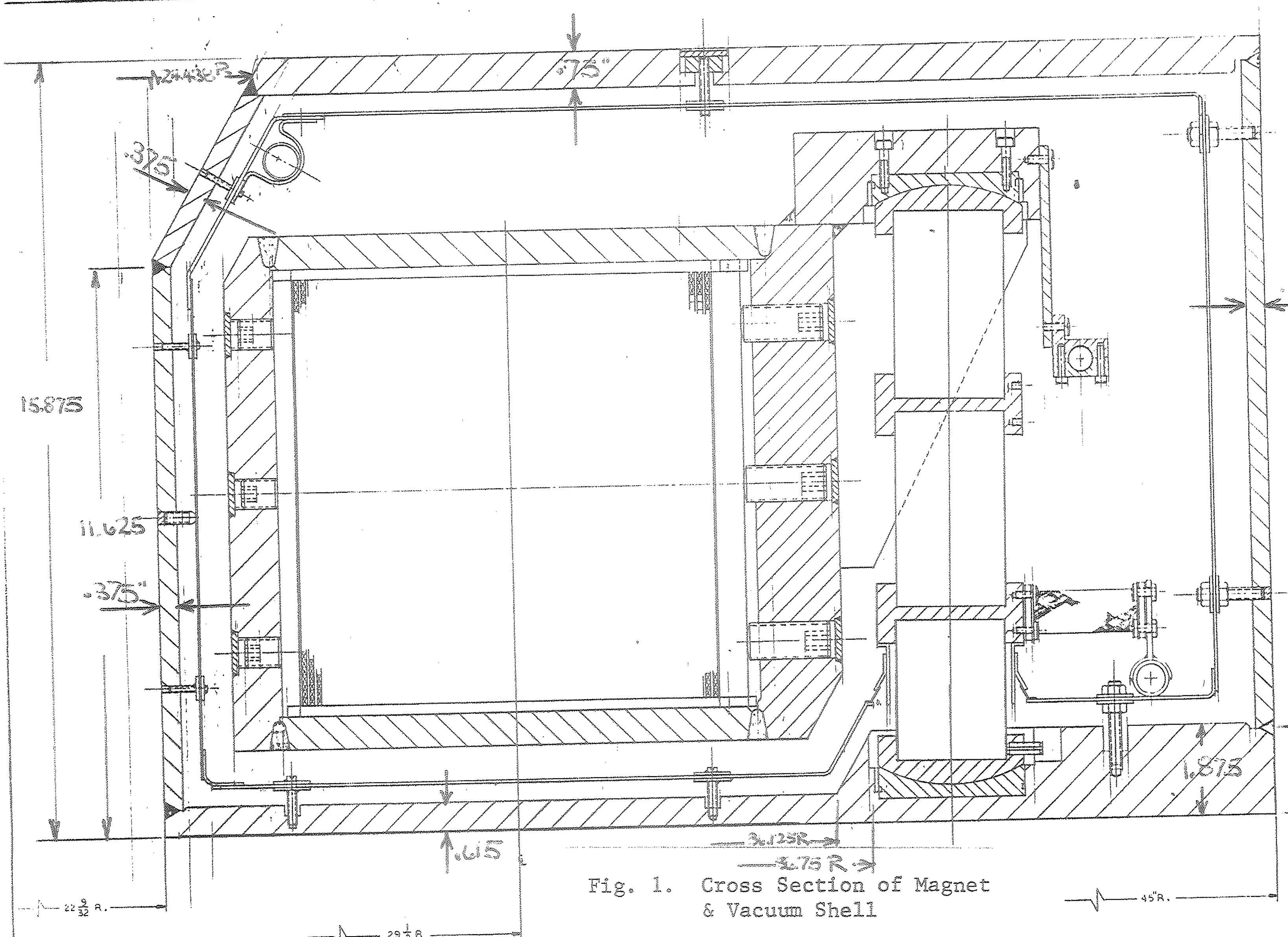


Fig. 1. Cross Section of Magnet & Vacuum Shell

REV.	DESCRIPTION	DRAWN	DATE
		APPD.	DATE

ITEM NO	PART NO.	DESCRIPTION OR SIZE	QTY. REQ.
		PARTS LIST	

UNLESS OTHERWISE SPECIFIED

FRACTIONS	DECIMALS	ANGLES
±	±	±

1. BREAK ALL SHARP EDGES 1/64 MAX.
2. DO NOT SCALE DWG.
3. DIMENSIONING IN ACCORD WITH ANSI Y14.5 STD's.
√ MAX. ALL MACHINED SURFACES

ORIGINATOR		
DRAWN	C.P.G.	1-6-82
CHECKED	C. [illegible]	
APPROVED	[illegible]	
USED ON		
MATERIAL-		

FERMI NATIONAL ACCELERATOR LABORATORY
UNITED STATES DEPARTMENT OF ENERGY

RESEARCH SERVICES
30" BUBBLE CHAMBER COIL CONVERSION
CROSS SECTION

SCALE	FILMED	DRAWING NUMBER	REV.
FULL			

600 screws in the posts is greatly reduced upon cooldown. Some slight load always is present due to Belleville washers. Distortion figures for the 1 atm case are shown for reference. The vacuum system is in common with the dewars and has one 6", one 2", and two 1" diameter pop off reliefs which is much greater combined area than that used for LHe system.

Both the 3D and the axisymmetric models are constrained in the axial and radial directions at the bottom plate's outer edge. No rotational constraints are imposed at this location. These constraints approximate the welded tabs with their bolts extending into the iron yoke. The 3D model also has the appropriate boundary conditions necessary for 1/2 model symmetry. Both models use a 0.75" thick element(s) without pressure which rigidly connects the midplane of bottom plate to the welded connection of the outer cylindrical wall. Coupled restraint equations (CERIG in ANSYS) model the welded connection of the post to the centerline of the bottom plate. At all other locations welds were close enough to shell midplane to make little difference.

Assumptions and Additional Information

1. Vessel is made of 304 stainless steel.

2. Welds are 308L stainless steel.

3. Section VIII Div. 2 is used as the basis with a Fermilab extra safety factor of 0.8 applied to the basic stress intensity limit $S_m = 20.0 \times 0.8 = 16.0$ ksi.

Comparison of Models

The most highly stressed region in this vacuum shell occurs in the inner wall at the junction with the base plate. The following chart lists stresses and deflections for both models subjected to internal pressure and the 3D model subjected to external pressure + 37,000 lb loading on the right post.

Top and bottom refer to the two sides of the shell elements. For these two models the bottom side is on the inside. σ_z is in axial direction and σ_θ is in the hoop direction. Total refers to combined bending and direct or tensile stresses. S.I. is the stress intensity. Peak stress in the two models agrees very closely while there is an 18% difference in deflection. It is also very apparent from this table that the decentering load has little impact on the peak stress levels.

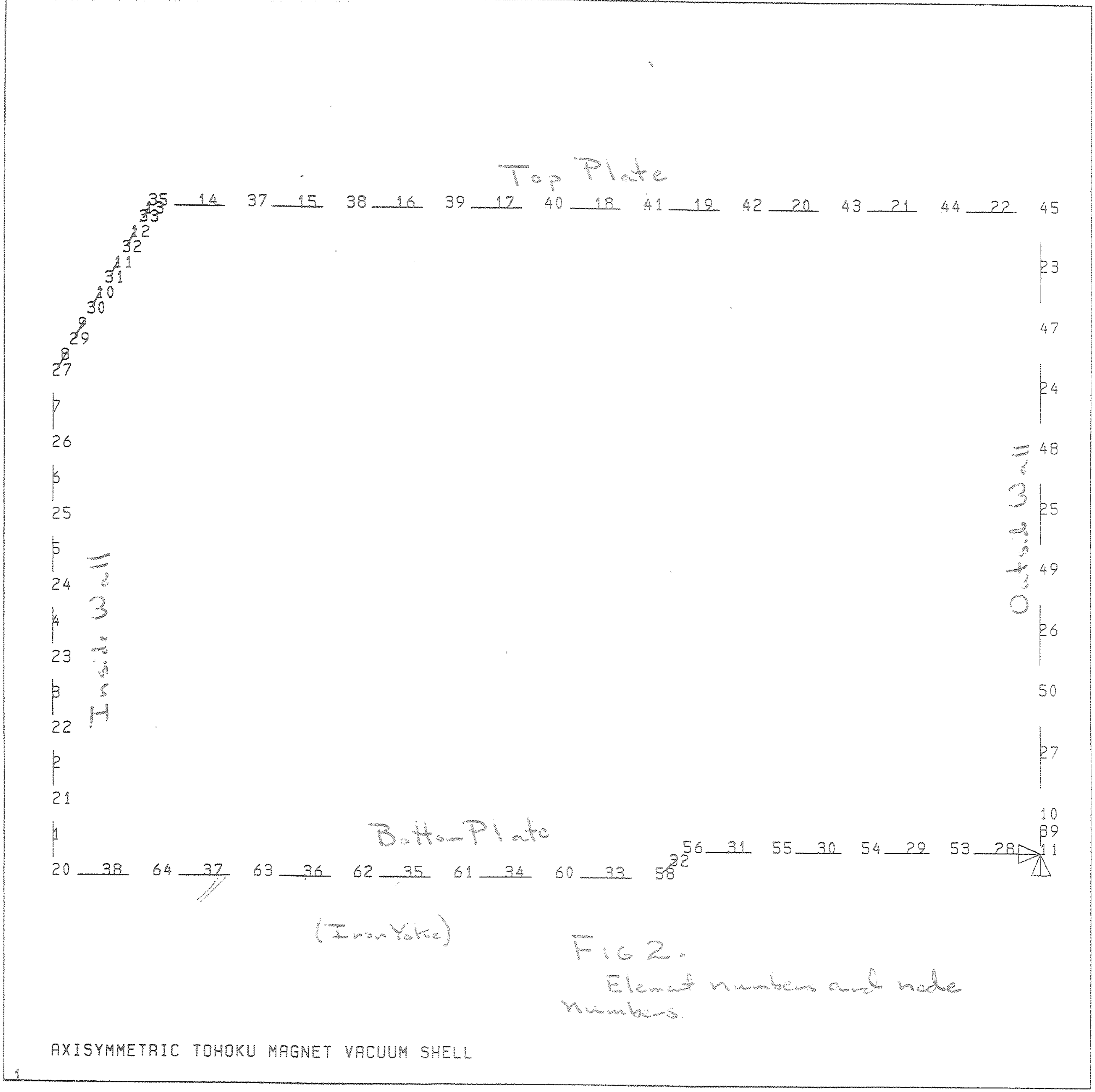
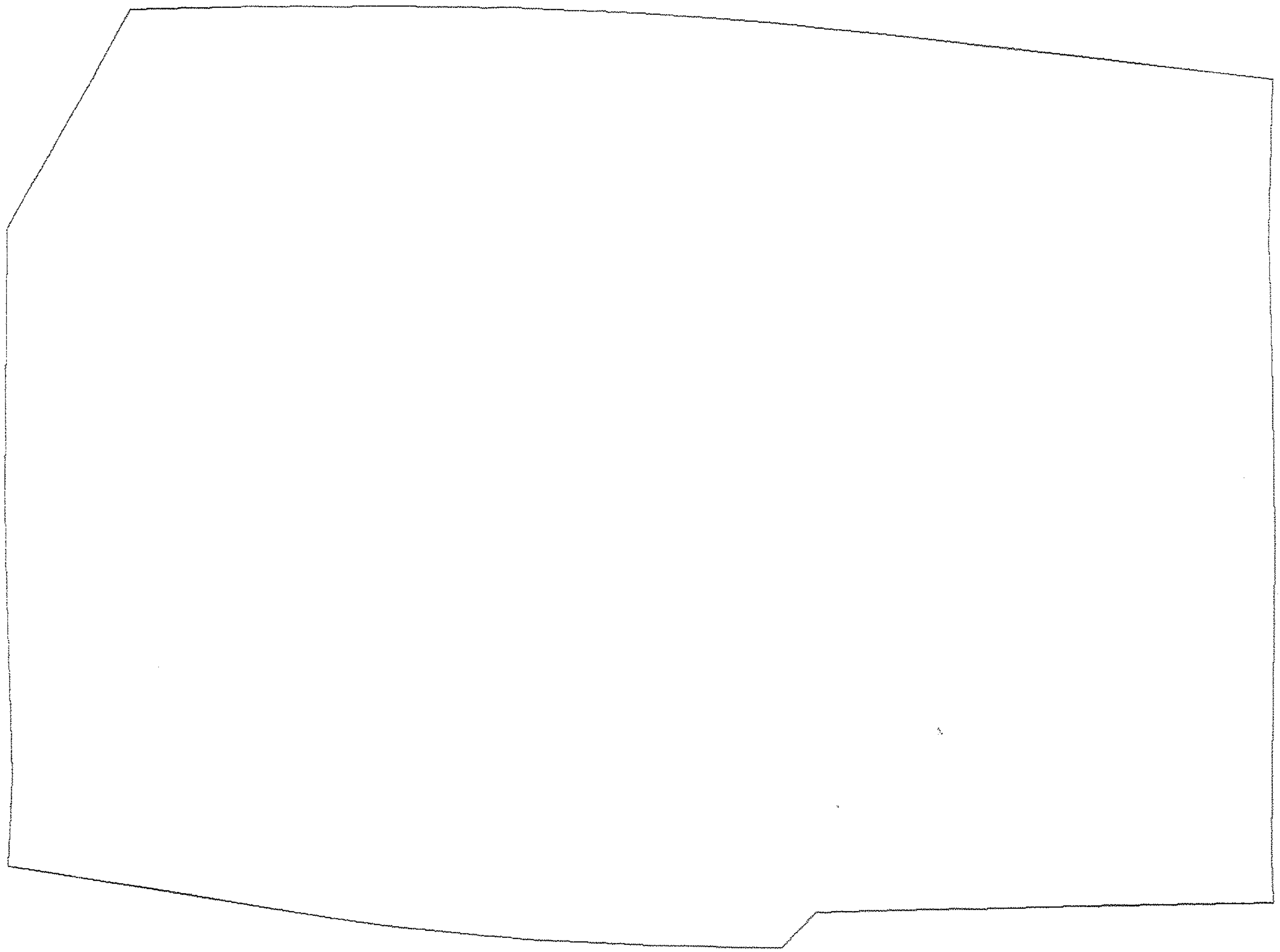


Fig 2.
Element numbers and node numbers

ANSYS
84/10/30
15.3856
PLOT NO. 2
PREP7 ELEMENTS
ENUM=1
NNUM=1
TDBC=1

ORIG SCALING
ZV=1
DIST=12.3
XF=33.6
YF=7.91



ANSYS
84/10/30
15.3958
PLOT NO. 4
POST1
STEP=1
ITER=1
DISPLACEMENT

ORIG SCALING
ZV=1
DIST=12.3
XF=33.6
YF=7.91
DMAX=.0384
DSCA=32

Fig 3.
1 atm internal pressure

AXISYMMETRIC TOHOKU MAGNET VACUUM SHELL

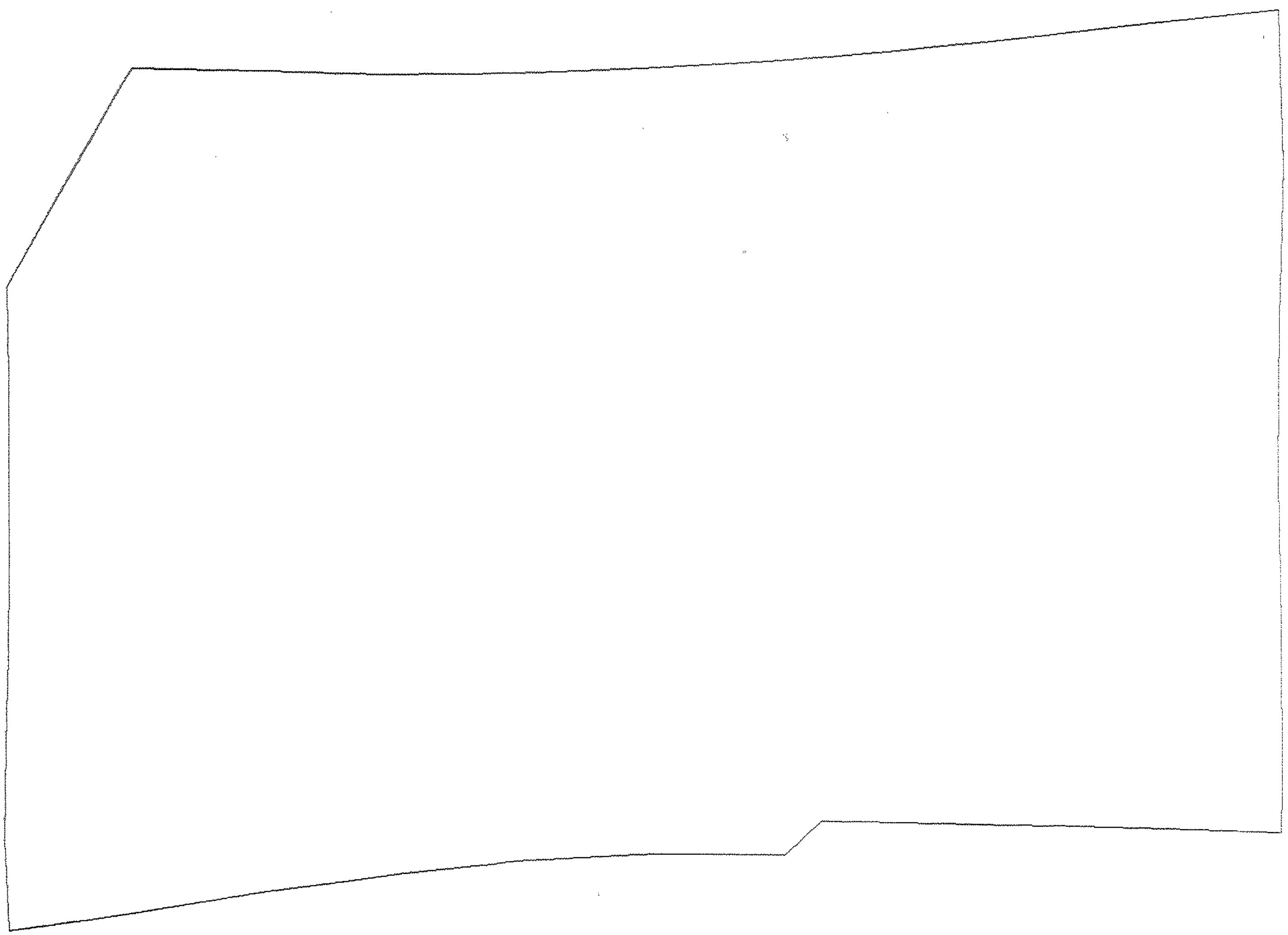


Fig 4

1 ATM EXTERNAL PRESSURE

ANSYS
84/10/30
15.3967
PLOT NO. 5
POST1
STEP=9999
ITER=1
DISPLACEMENT

ORIG SCALING
ZV=1
DIST=12.3
XF=33.6
YF=7.91
DMAX=.0384
DSCA=32

ANSYS
84/11/21
17.3172
PLOT NO. 4
PREP7 ELEMENTS
TDRC=1
RDSC=1

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
XF=-.0313
YF=22.3
ZF=7.88

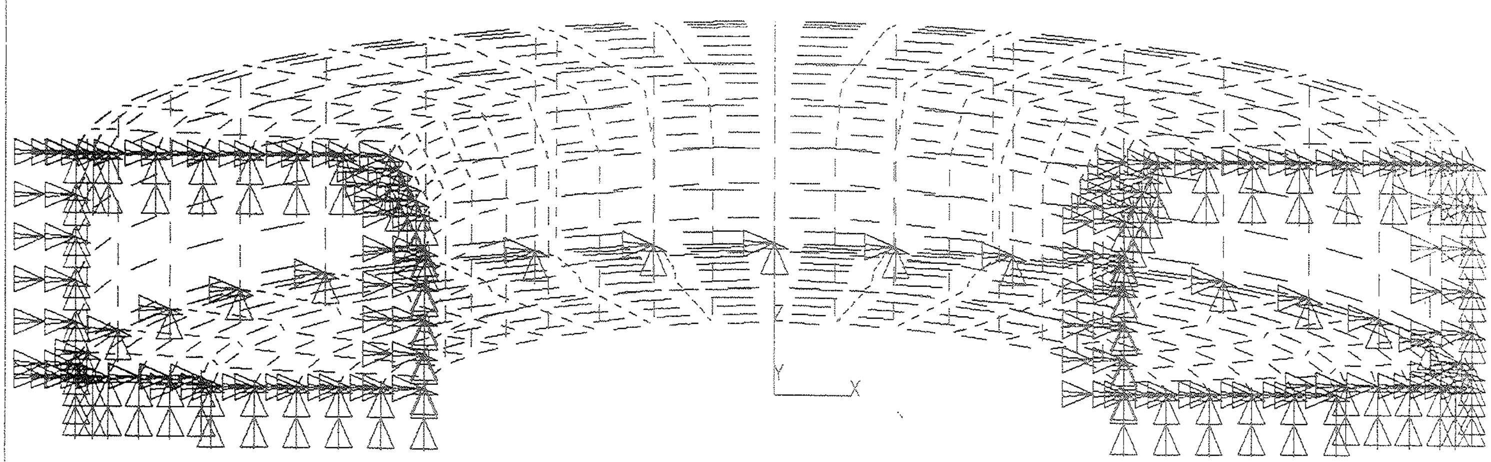
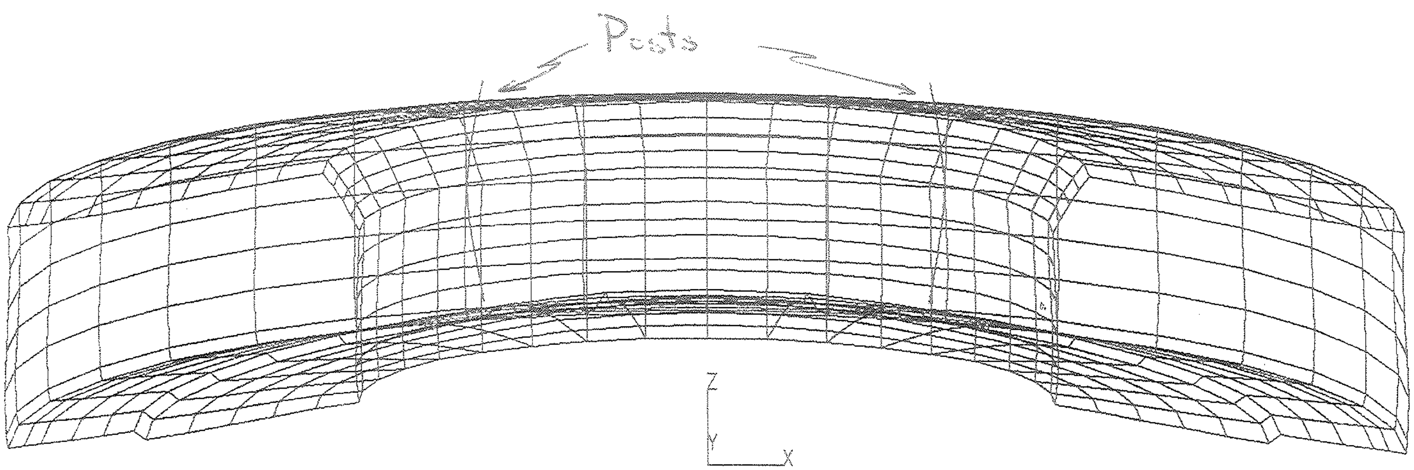


FIG 5 3D Model
Showing Translational and
Rotational Boundary Conditions

TOHOKU MAGNET VACUUM SHELL

1



ANSYS
84/11/21
17.7661
PLOT NO. 1
POST1
STEP=1
ITER=1
DISPLACEMENT

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
YF=22.3
ZF=7.92
DMAX=.0157
DSCA=313

Fig. 6 Displacement Plot

44000 LB PRELOAD ON EACH POST

ANSYS
84/11/21
17.7786
PLOT NO. 4
POST1
STEP=2
ITER=1
DISPLACEMENT

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
YF=22.3
ZF=7.92
DMAX=.00913
DSCA=540

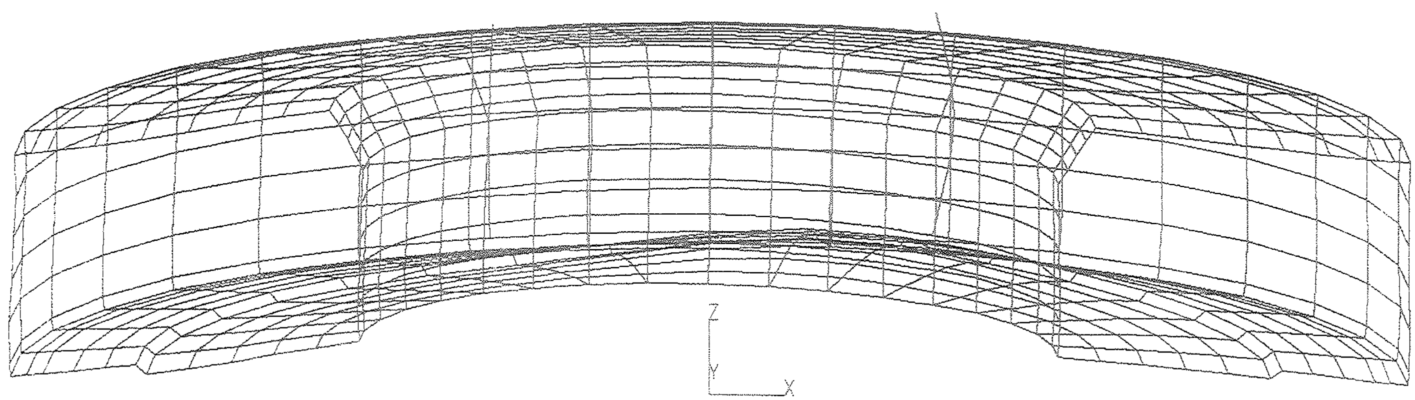


Fig. ? Displacement Plot

37000 LB FORCE ON RIGHT POST

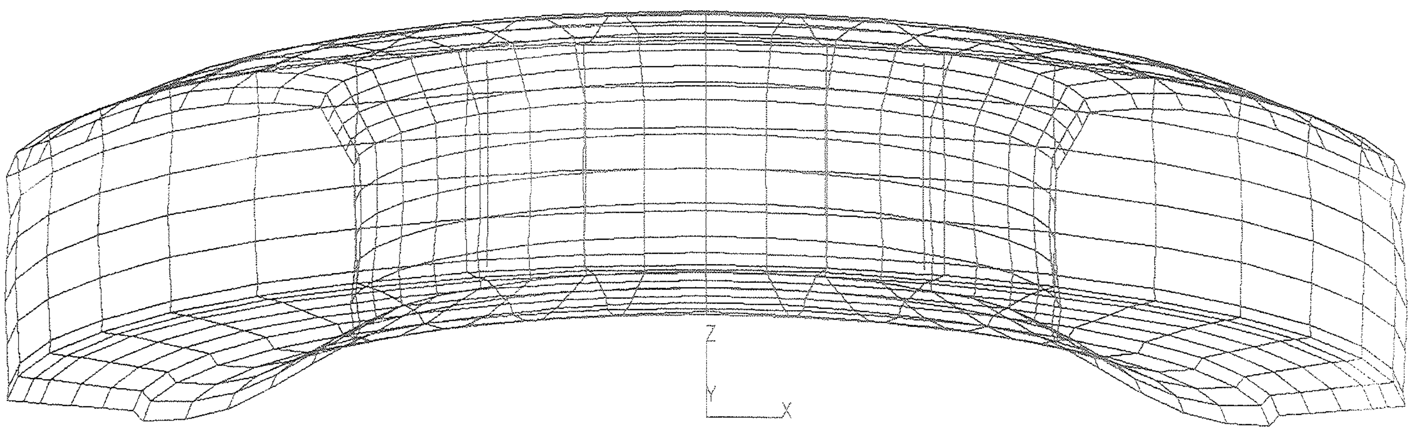


Fig 8 Displacement Plot

1 ATM INTERNAL PRESSURE

ANSYS
84/11/21
17.7969
PLOT NO. 5
POST1
STEP=3
ITER=1
DISPLACEMENT

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
YF=22.3
ZF=7.92
DMAX=.0438
DSCA=113

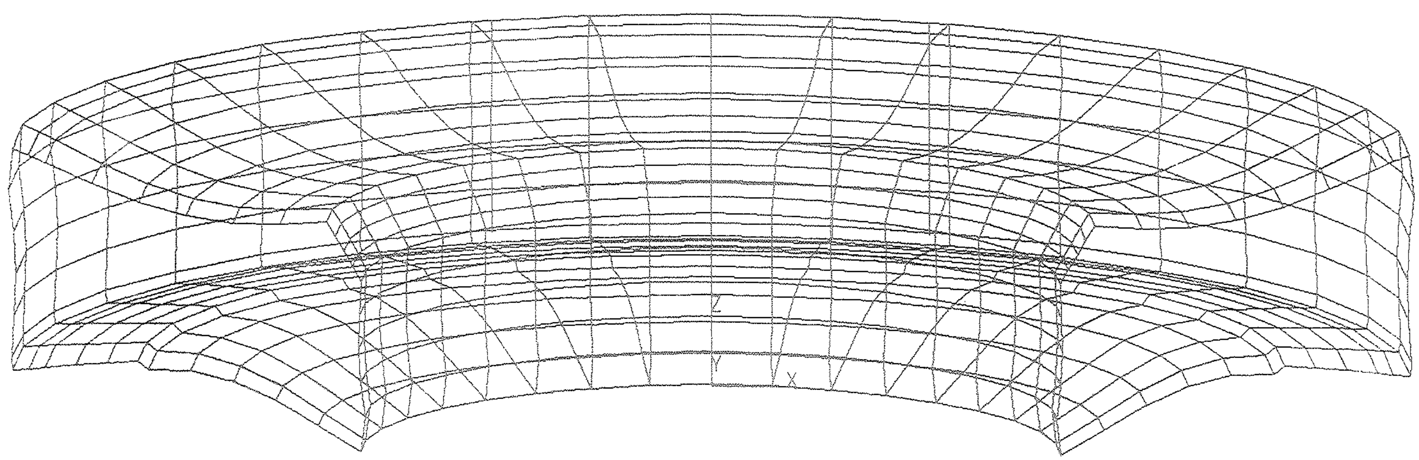


Fig 9. Displacement Plot

1 ATM EXTERNAL PRESSURE

ANSYS
84/11/21
17.8319
PLOT NO. 9
POST1
STEP=9999
ITER=1
DISPLACEMENT

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
YF=22.3
ZF=7.92
DMAX=.0438
DSCA=113

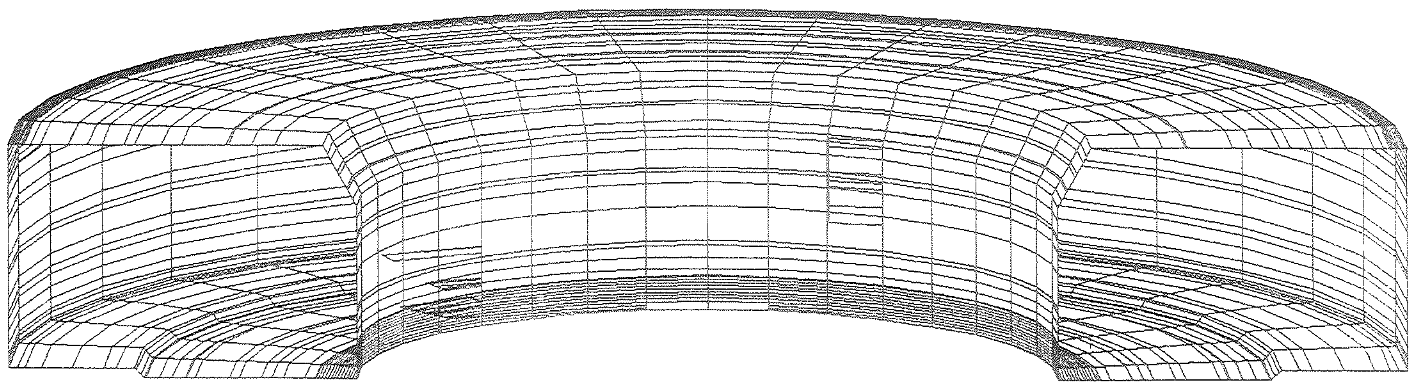


Fig. 10

Stress Intensity Plot
(Stress on inner surface
of shell elements)

1 ATM INTERNAL PRESSURE

ANSYS
84/11/21
17.0217
PLOT NO. 7
POST1
STEP=3
ITER=1
STRESS PLOT
SI
BOTTOM

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
YF=22.3
ZF=7.92
HIDDEN
DMAX=.0438
MX=16401
MN=349
INC=1000

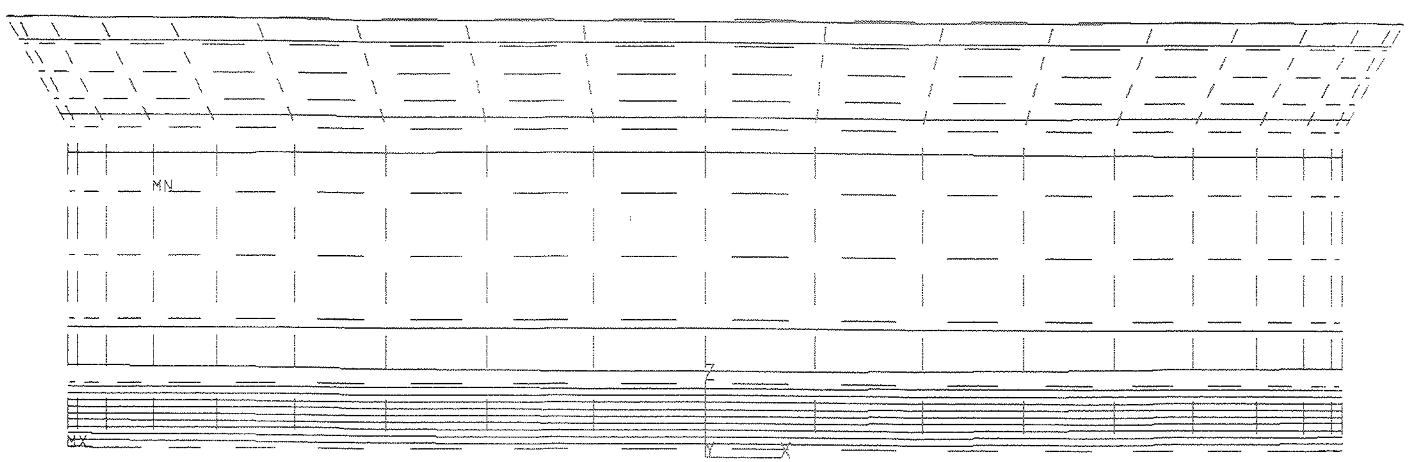


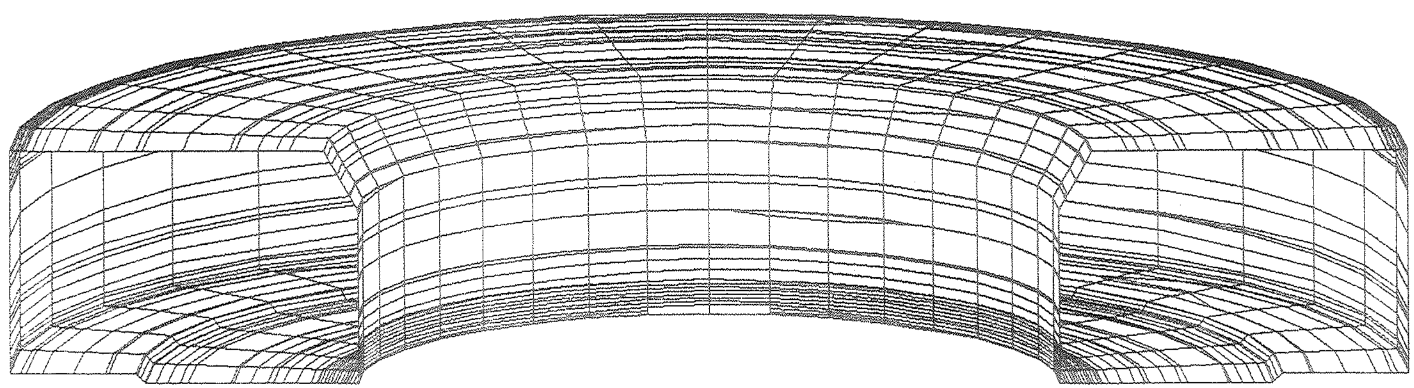
Fig. 11

Stress Intensity Plot for inner surface of only the inner cylinder and cone

1 ATM INTERNAL PRESSURE

ANSYS
84/11/21
17.8247
PLOT NO. 8
POST1
STEP=3
ITER=1
STRESS PLOT
SI
BOTTOM

ORIG SCALING
YV=-1
DIST=27.1
YF=12.3
ZF=7.91
DMAX=.0421
MX=24125
MN=719
INC=2000



ANSYS
84/11/21
17.8597
PLOT NO. 12
POST1
STEP=9999
ITER=1
STRESS PLOT
SI
TOP

ORIG SCALING
YV=-1
ZV=.2
DIST=49.3
YF=22.3
ZF=7.92
HIDDEN
DMAX=.0404
MX=15779
MN=522
INC=1000

Fig. 12

Stress Intensity Plot on
Outside Surface

1 ATM EXTERNAL PRESSURE + 37000 LB LOAD

Peak Stress Summary Comparison
(all pressure in psi)

Quantity	Axisymmetric Model 1 atm Internal Pressure	3D model 1 atm Internal Pressure	3D Model 1 atm External Pressure + 37,000 lb Decentering Load
Axial Deflection	0.0353	0.0416	-0.372
σ_z(Total,TOP)	-23570	-22990	23500
σ_z(Total,BOT)	23104	24110	-24670
σ_z(bending)	-23337	-23550	24085
σ_z(tensile)	-233	560	-585
σ_θ(Total,TOP)	-6711	-6801	6805
σ_θ(Total,BOT)	7291	7550	-7900
σ_θ(bending)	-7001	-7176	7353
σ_θ(direct)	290	374	-548
S.I.(TOP)	23585	22990	23500
S.I.(BOT)	23104	24120	24670

Note that these stress levels are obtained by averaging only adjacent elements with the same thickness. This gives us real peak stress levels. See for example the difference in peak stress levels between Figures 10 and 11.

Comparison to ASME Section VIII Division 2 Pressure Vessel Code

Inner Cylinder at Bottom Plate Junction:

The inner wall is nominally 3/8" but the V butt weld at this corner is only 5/16" deep. To approximate stresses in this weld, multiply bending stresses by $(6/5)^2 = 1.44$ and tensile stresses by $6/5 = 1.20$ as found in the table above. This weld as well as all other corner welds in this vacuum vessels are prohibited by the ASME Code Division 2 [AD-413.2(d)] and also were not radiographed. Stresses from the 3D 1 atm internal pressure column will be used. Evaluate stress levels using Appendix 4 of Division 2.

In the axial or z direction,

bending stress = 23550 x 1.44 = 33912

direct or normal stress = 560 x 1.20 = 672

σ_z (Total,I.D.) = -33912 + 672 = -33240

σ_z (Total,O.D.) = 33912 + 672 = 34584

In the hoop or θ direction,

bending stress = 7176 x 1.44 = 10330

direct or normal stress = 374 x 1.20 = 448

σ_θ (Total, I.D.) = -9882

σ_θ (Total, O.D.) = 10778

In the radial direction

σ_r (I.D.) = 0

σ_r (O.D.) = -14.7

σ_r (avg) = -7

Table 4-120.1 categorizes stresses for typical cases. For a cylindrical shell at the junction with a head or flange, stresses are classified as primary local, P_L, and secondary, Q. Assume a more conservative requirement that local membrane stresses are actual general membrane streses.

P_m(SI) = 672 - -7 = 679 < S_m = 15000

$P_L + P_b$ + Q(SI) = 34584 - -14.7 = 34600 < 3 S_m = 45000

Since virtually all the stress at this location arises from the pressure loads, the vacuum shell is actually rated for 45000/34600 x 14.7 = 19 psig. Hoop stress for a free floating cylinder is given by σ_θ = Pr/t = 14.7 x 22.47/0.3125 = 1060 psi compared to the 448 psi hoop stress predicted by ANSYS. This small (absolute not percentage) difference shows that the inner cylinder is a truly self supporting component, and that bending stresses arise from discontinuity rather than basic equilibrium requirements. It is, therefore, correct to categorize these total overall stresses as $P_L + P_b + Q$ instead of $P_L + P_b$. If one were to require an additional equivalent weld efficiency factor of 0.65 as used in Division 1, the combination of $P_L + P_b + Q$ would not be acceptable. This extra safety factor was not used for the following reasons.

1. It is not required by Div. II.

2. Fermilab has an extra 0.8 derating.

3. The primary membrane stress is exceedingly low. Any local plastic deformation will greatly reduce the peak bending stress.

4. The vessel is fully pressure tested to 1.25 x 19 psig = 24 psig.

Outer Cylindrical Wall:

The outer wall has a much lower stress level than the inner wall bottom plate junction. The peak stress occurs at the junction with the top plate. At this location,

$P_L(SI) = 380 \times 1.20 = 460$ psi

$P_L + P_b + Q(SI) = 16230 \times 1.44 = 23370$ psi

Stress levels at this location are much less than the Code allows. The factors 1.20 and 1.44 are used again to estimate the stress in the less than full thickness welds.

Three large penetrations and a variety of much smaller openings exist in the outer cylindrical wall, and these must be considered. Figure 13 shows part of the outside wall. The 7" wide notch is for the helium interconnecting line and the 6-1/8" notch is for one at the vertical stainless steel support arms. The outer shell was installed over the arms and interconnecting line with the openings on the base plate side. Three-eights inch thick material was welded to the notches and base plate turning the U notch into a simple penetration. Three-eigths inch thick reinforcing plates were then welded over the openings.

A 6-5/8" O.D. x 0.109" wall pipe (6 IPS, SCH 5) extends out of the helium line opening, and 5-9/16" O.D. x 0.109" wall pipe (5 IPS, SCH 5) extends out of the two stainless steel arm openings. All other openings have relatively much more reinforcing and are not further considered.

The basic reinforcement requirement is given by $A = d\, t_r\, F$ in section AD-520 where d is the diameter of the opening and t is the minimum thickness required by Article D-2. F is an angle dependent factor with a value between 1.0 and 0.50. The openings are actually ellipsoidal with cylindrical pipes inserted through them. The method of reinforcing is difficult to analyze and Article D-2 cannot be really used. A simplified approximate analysis is chosen instead.

The openings are assumed to be cylindrical with 6.407" and 5.345" diameters. Refer to Fermilab Drawing No. 2771-MD-56496, (56497), and (56498) for reinforcing pad dimensions. AD-540.1(a)(1) limits the amount of reinforcement available to: 100% shall be on each side of the axis of the opening within the diameter of the opening. With this limitation the following areas from the reinforcing pads are available.

Helium Line	hoop direction	$A = (2.25 + 3.125) \times 0.375 = 2.02\ in^2$
	axial direction	$A = 4 \times 0.375 = 1.5\ in^2$
Vertical Line	hoop direction	$A = (1.375 + 2.75) \times 0.375 = 1.55\ in^2$
	axial direction	$A = 4 \times 0.375 = 1.5\ in^2$

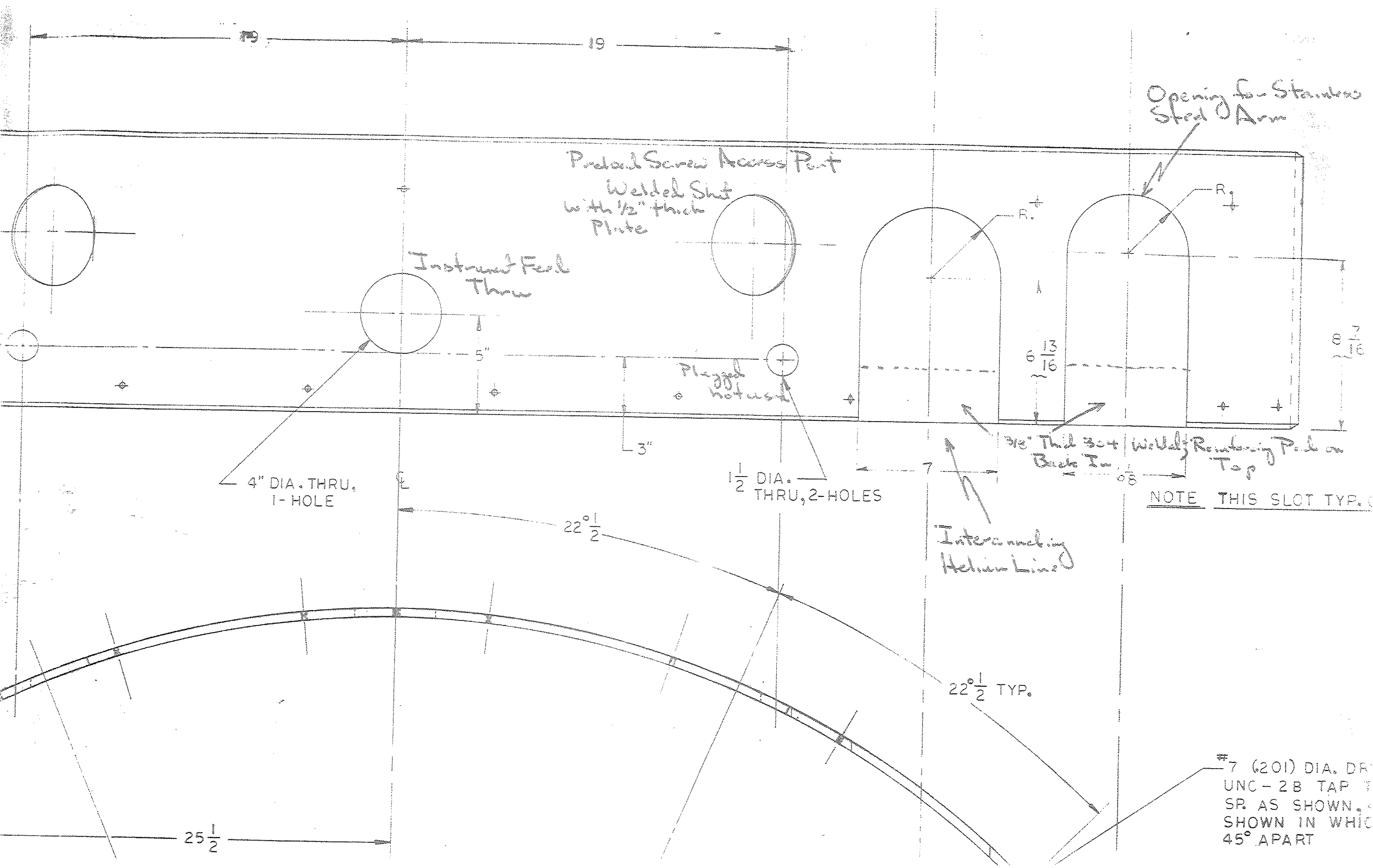


Fig 13. Outer Wall Showing Penetrations

No other limits to reinforcing apply to these pads. If the full 3/8" wall thickness were required, the following reinforcing areas would require

Helium Line	hoop	$6.407 \times 0.375 \times 1 = 2.40\ in^2$
	axial	$6.407 \times 0.375 \times 0.5 = 1.20\ in^2$
Vertical Arm	hoop	$5.345 \times 0.375 \times 1 = 2.0\ in^2$
	axial	$5.345 \times 0.375 \times 0.5 = 1.0\ in^2$

Using this criterion hoop reinforcement areas are not satisfied. If the highest stress values in the outer cylinder at the top plate junction were selected to represent the stress level in the entire shell, t_r (min required wall thickness) = (16230/45000) x 0.375 = 0.135". All reinforcing area requirements are easily satisfied. In fact the average stress in the outer wall is much lower. The following stresses from the axisymmetric model are found near the center of three openings.

σ_θ(direct) = 390

σ_z(direct) = 650

S.I. = 2600

Conclusion: Substantial extra reinforcing has been provided.

Horizontal Support Posts:

These posts are preloaded with 44,000 lbs which is the maximum that should ever be experienced. The loss of preload on cooldown is expected to be ~ 22,000 lbs (0.0014 strain). Preload was selected to be at least 1/2 the maximum electromagnetic horizontal decentering load (38000 lbs) plus cooldown load or 19000 + 22000 = 41000 lbs. With this preload the posts should never come loose.

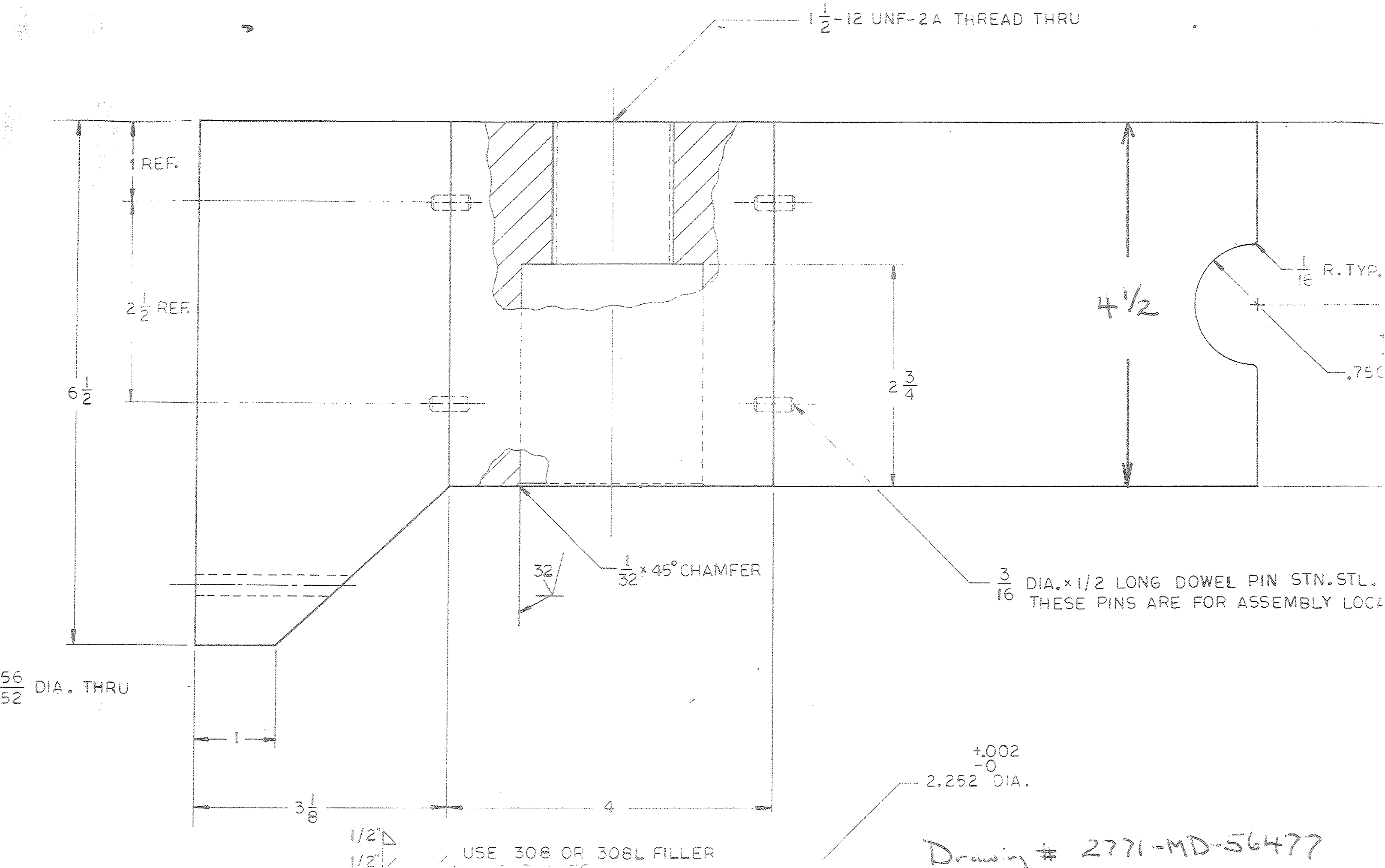
Stresses and Forces in Support Posts
(44,000 lb load)

Pinned End (TOP) Force	14,000 lbs
Fixed End (Base) Force	30,000 lbs
Maximum Bending Stress	17,600 psi
Bending Stress at Fixed End	2,465 psi
Bending Moment at Fixed End	34,720 in lbs

Half inch fillet welds were selected as shown in Figure 16. Their dimensions are given in the bottom half of this figure.

The welds on the bottom are in compression and were sized smaller to reduce distortion in the bottom half of this figure.

To simplify calculations assume all welds are 1/2" and are symmetrical about the neutral axis. Use the procedure found in Design of Welded Structure section 7.4 by Blodgett. The welds are treated as lines without area.

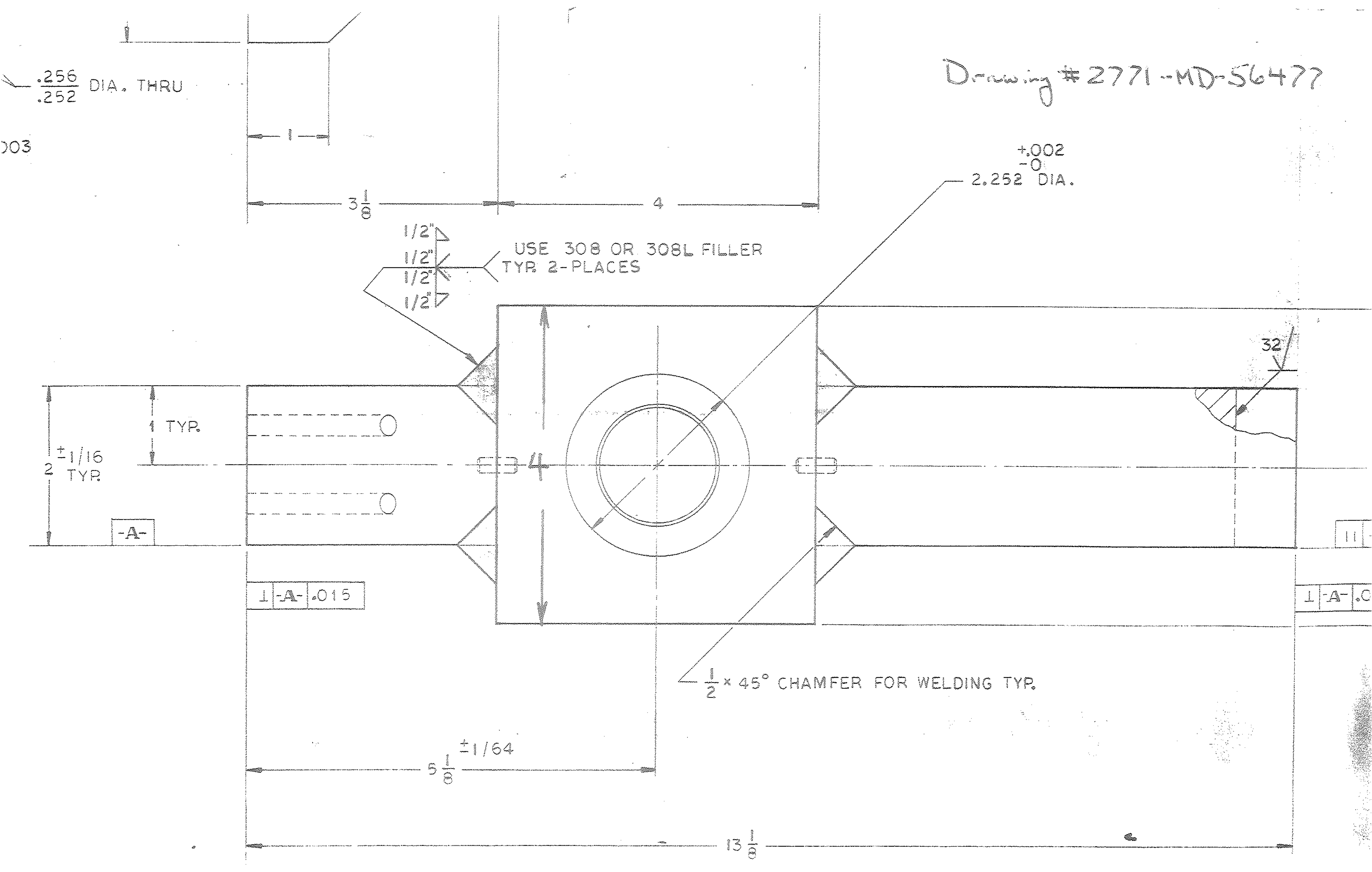


Drawing # 2771-MD-56477

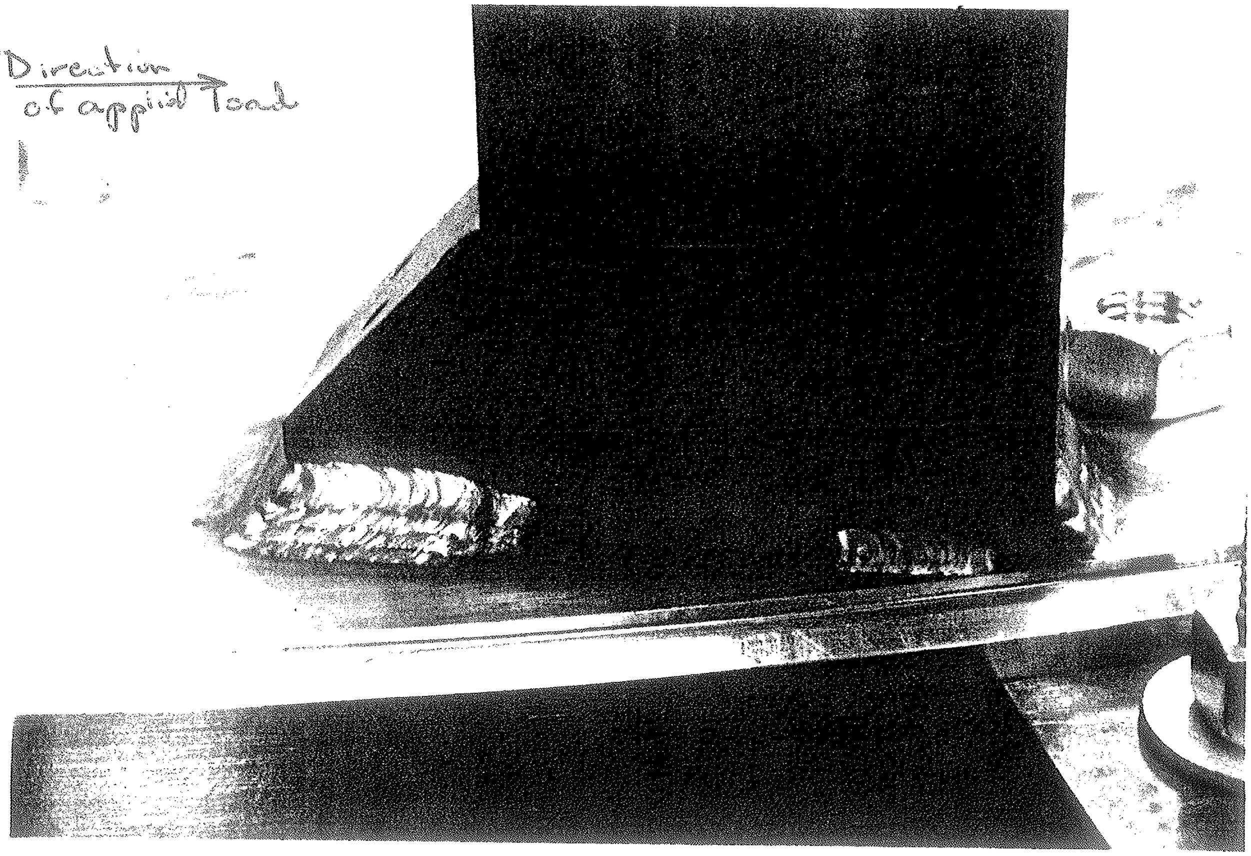
Fig. 14 Vacuum Shell Post

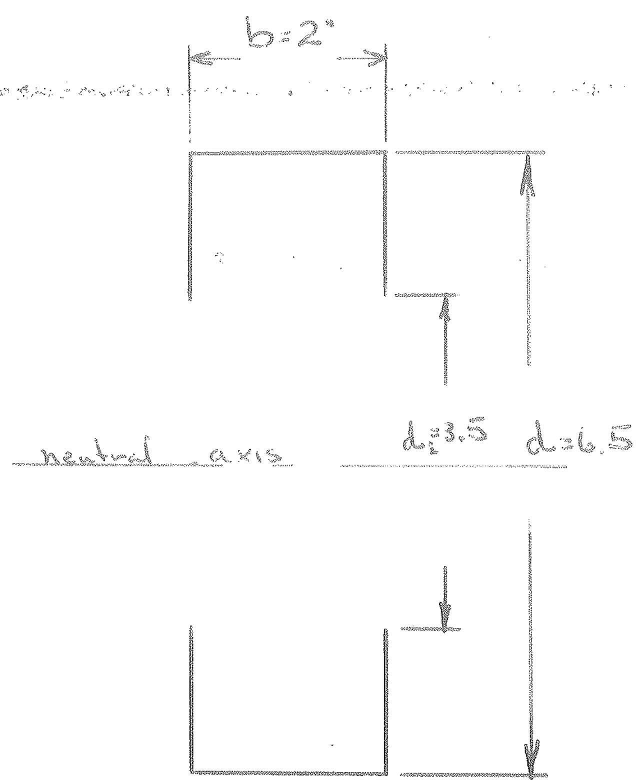
Fig 15 Vacuum Shell Post

Drawing #2771-MD-56477



Direction of applied load





Idealized model of welds used for calculating their stress levels. All welds are assumed to have a 1/2" leg

Fig 16. Welds for vacuum shell post

$$f = \frac{M}{S_w} \text{ (lbs/inch)}$$

where S_w is the section modulus, M is the bending moment, and f is the force per linear inch in the welds.

$$S_w = bd + \frac{d^2}{3} - \frac{d_1^{\ 2}}{3} = 23$$

Stress will only be calculated in the 2" long outside weld which is the most highly stressed location.

$$\text{Bending Force} = f_b = \frac{34720}{23} = 1510 \text{ lbs/inch}$$

$$\text{Shear Force} = f_s = \frac{30000}{2(2+2x1.5)} = 3000 \text{ lbs/inch}$$

$$\text{Resultant Force} = (f_b^{\ 2} + f_s^{\ 2})^{1/2} = 3360 \text{ lbs/inch}$$

$$\sigma_{max} = \frac{f_r}{0.707 \times h}$$

where h is the length of the leg of the weld = 0.5"

$$\sigma_{max} = 9500 \text{ psi}$$

Division 1 UW-15(b) uses a factor of 0.49 for determining maximum shear stress in a fillet weld. Using a Fermilab allowable of 15000 psi this weld is only good for 7500 psi and is over stressed by 2000 psi. Without the additional Fermilab factor of 0.8 this weld would be marginally acceptable according to ASME Code standards. The NBS LNG Handbook gives a value of 80000 psi for the ultimate shear strength of 304 stainless steel. Assuming 308 has the same strength, the safety factor is 8.4. This does not take into account that all other welds are stressed much lower. This weld is considered acceptable.

Buckling Under External Pressure

The two cylindrical shells are subject to axial buckling. Both vacuum shells have been previously evacuated and no evidence of buckling appeared. However, it is useful to make some estimates of the buckling load. Unfortunately, this torus is not easily analyzed. Table 35 in Roark and Young gives some approximate formulas for a cylinder subjected to buckling.

For a thin walled tube under uniform longitudinal compression use case #15.

$$q' = \frac{0.3\ Et}{r}\ \text{psi}$$

Using this formula the outer cylinder can withstand a compressive axial stress of 75000 psi. An examination of the axisymmetric model shows that both inner and outer cylinders have axial compressive direct (normal) stresses less than 1000 psi.

Use case #19b for a short tube under uniform lateral pressure with both ends held circular.

$$q' = 0.807\ \frac{Et^2}{\ell r}\ [(1-\nu^2)^{-3}\ t^2\ r^{-2}]^{1/4}$$

q' = 550 psi for the outer ring

Both of these formulas show that for the same stress the outer ring is more likely to buckle. Hence only the outer ring will be examined using Code design rules.

It is assumed that a standard cylindrical shell with a head is much more likely to collapse than our torus which has the inner wall as an "extra" support.

Using AD-310

t = 0.375"

D_o = 90"

L = 13.5"

L/D_o = 0.15

D_o/t = 240

From Figure 2-AGO-28.0, Appendix 2,

Factor A = 0.0032

From Figure 2-AHA-28.1, Appendix 2, for 304 stainless steel,

Factor B = 12500

$$P_a = \frac{4B}{3\ (D_o/t)} = 69\ \text{psi} > 14.7$$

The cylinder has plenty of thickness.

Because of the unusual shape, check also against the design requirements of AD-340, cylinders under axial compression.

Factor A = $0.125\ t/R_o = 0.00105$

Using Figure 2-AHA-28.1 again

Factor B = 9500 psi = maximum allowable axial compressive stress

Compressive axial stress (1000 psi for the O.D. cylinder) Factor B. Again buckling is no problem.

<u>Summary</u>

This vacuum vessel was not originally designed according to the ASME Pressure Vessel Code which in fact excludes vessels with less than a 15 psid difference. However, Section VIII Division 2 was selected as a basis for comparison of acceptability. The following conclusions are reached.

1. Welds are not of the type required by the Code nor are they radiographed.

2. Part of the weld on the support post may be slightly overstressed depending upon whether or not Fermilab extra 0.8 safety factor is required. A safety factor of 8.4 based on ultimate shear stress is calculated.

3. The peak stress in the shell occurs at the weld joining the inner cylinder with the base plate. Using Division 2 allowable stress intensities with Fermilab 0.8 safety factor, this weld is considered to have an acceptagble stress level.

4. Buckling is no problem.

5. The vessel should be considered safe for 1 atm external and 19 psig internal pressure. Reliefs for internal pressure are set at ~ 0 psig.

6. The vacuum shell has been fully pressure tested to 1.25 x 19 psig = 24 psig = 24 psid.