TITLE: Tohoku Bubble Chamber Helium Dewar Analysis

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Introduction

The helium dewars used on the Tohoku Bubble Chamber magnets were intended to meet the requirements of the Laboratory's Room Temperature Pressure Vessel Safety Standard, 14.1. Certain fabrication and design details do not meet this standard, however. The purpose of this note is to fulfill the 14.1 requirement of an Engineering Note, and to present the information required in paragraph 5.6 of 14.1, "Extended Engineering Note for Exceptional Vessels", dealing with the integrity and ramifications of the exceptional aspects of these dewars.

The following information will be presented:

I. Determination of Maximum Allowable Working Pressure (Internal and External) of Inner Vessel.

These calculations use the rules of the ASME Boiler and Pressure Vessel Code, Section VIII, Div. 1 and Div. 2, modified as required by 14.1 for in-house vessels. Those vessel details not meeting the requirements of 14.1 are noted, and an appropriate analysis applied.

II. Determination of Maximum Allowable Working Pressure (Internal and External) of Vacuum Vessel.

These calculations use the rules of Section VIII, Div. 1 as modified by 14.1

III. Verification of Weld Strength.

These calculations use the rules of Div. 1 as modified by 14.1.

IV. System Venting Verification.

V. Pressure Test Results.

VI. Exceptional Vessel Details.

This is a summary and discussion of those aspects of the helium dewars which do not meet 14.1 requirements.

I. Determination of Maximum Allowable Working Pressure (Internal and External) of Inner Vessel

The components of the inner vessel are shown in Fig. 1.

A. Internal MAWP of Inner Vessel

1. Cylindrical Shell 1

The appropriate formula is given in UG-27 of Div. 1.

R = 24 in.

t = 0.375 in.

E = 0.70 from Table UW-12 for double weld butt joint, circumferential stress calculation

E = 0.65 from Table UW-12 for single welded butt joint, longitudinal stress calculation

S = 15000 psi

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 162 \text{ psid circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 307 \text{ psid longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 1 is 162 psid.

2. Cylindrical Shell 2

This calculation will consider only the pressure loading of this shell. A complete analysis with weight and pressure loading is presented later in this part. The applicable formula is given in UG-27.

R = 4.31 in.

t = 0.148 in.

E = 1 for circumferential stress calculation

E = 0.60 for groove weld in shear, longitudinal stress calculation

S = 12800 x 0.80 as required by UW-12 for non-radiographed vessels, circumferential stress calculation

S = 12800 for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 344 \text{ psid} \qquad \text{circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 535 \text{ psid} \qquad \text{longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 2 is 344 psid.

3. Cylindrical Shell 3

This calculation will consider only the pressure loading of this shell. A complete analysis with weight and pressure loading is presented later in this part. The applicable formula is given in UG-27.

R = 5.375 in.

t = 0.165

E = 1 for circumferential stress calculation

E = 0.65 for single welded butt joint, longitudinal stress calculation

S = 12800 x 0.80 for circumferential stress calculation

S = 12800 psi for longitudinal stress calculation

Then

$$P = \frac{S\ E\ t}{R+0.6t} = 309 \text{ psid} \qquad \text{circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 517 \text{ psid} \qquad \text{longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 3 is 309 psid.

4. Cylindrical Shell 4

This calculation will consider only the pressure loading of this shell. A complete analysis with weight and pressure loading is presented later in this part. The applicable formula is given in UG-27.

$R = 6.375$ in.

$t = 0.165$ in.

$E = 1$ for circumferential stress calculation

$E = 0.65$ for single welded butt joint, longitudinal stress calculation

$S = 12800 \times 0.80$ for circumferential stress calculation

$S = 12800$ psi for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 261 \text{ psid} \qquad \text{circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 435 \text{ psid} \qquad \text{longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 4 is 261 psid.

5. Cylindrical Shell 5

This shell forms the top hat of the dewar, and contains several openings. Only one of these openings is greater than 3 in. pipe size, and therefore reinforcement is necessary according to UG-36(c)(3)(a).

The required area of reinforcement in any plane through the shell taken parallel to and containing the centerline of the opening is

1) $A = d t_r F$

where

d = finished diameter of opening

t_r = thickness of shell required for pressure

F = factor to compensate for the variation in pressure stress on different planes = 1.00

The area available for reinforcement is calculated from UG-40. This is the area in a plane through the shell taken parallel to and containing the centerline of the opening and extending one opening diameter on either side of the opening centerline.

2) $A = d(t - t_r)$

where

d = finished diameter of opening

t = actual shell thickness = 0.188"

t_r = thickness of shell required for pressure

Combining 1) and 2) gives

3) $t_r = t / 2 = 0.094$ in.

This thickness is then used in the formula of UG-27

R = 8 in.

t = 0.094 in.

E = 1 for circumferential stress calculation

E = 0.55 for fillet weld, longitudinal stress calculation

S = 12800 x 0.80 for circumferential stress calculation

S = 12800 psi for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 119 \text{ psid} \qquad \text{circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 166 \text{ psid} \qquad \text{longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 5 is 119 psid = 119 psig since there is no vacuum loading on this member.

6. Torispherical Head 1

This head contains no openings which require reinforcement. There is a tee section assembly plate welded into an opening in the head at the apex of the crown. This plate is greater in thickness than the head, and is attached by full penetration welds. Therefore, it will not affect the MAWP of the head. The strength of the plate weld is considered in Part III of this note.

The appropriate formula for torispherical heads under internal pressure is given in UG-32.

$$t_{min} = t_{actual} - t_{tol} = 0.375 - 0.031 = 0.344$$

$S = 15000 \times 0.8$ (required by UW-12)

$E = 1$ seamless head

$L = 48$

then,

$$P = \frac{S\ E\ t}{0.885L + 0.1t} = 97 \text{ psid}$$

Therefore, the MAWP (internal) of torispherical head 1 is 97 psid.

7. Torispherical Head 2

Torispherical head 2 contains a 8.625 in. central opening at which cylindrical shell 2 attaches. This opening must be reinforced according to UG-37. There are no additional reinforcing elements attached to the head; therefore, all reinforcement must come from the head thickness which is not required to resist the pressure loading.

It was demonstrated for cylindrical shell 5 above that the thickness of the shell available to resist pressure loading after consideration of opening reinforcement is

$t_r = t/2$

For torispherical heads, $t = t_{min}$, where t_{min} is the thinnest portion of the head.

Then,

$t_r = 0.344/2$

$t_r = 0.172$

The formula for torispherical heads under internal pressure is given in UG-32

$t = 0.172$ in.

$S = 15000 \times 0.8$ (required by UW-12)

$E = 1$ seamless head

$L = 48$ in.

then

$$P = \frac{S\ E\ t}{0.885L+0.1t} = 49 \text{ psid}$$

This calculation does not consider the effects of the weight loading on the shell/head junction, which will tend to further reduce the MAWP under Div. 1 rules. Therefore, the MAWP of torispherical head 2 is less than 49 psid as calculated by Div. 1 rules.

8. Flat Head 5

(NOTE: Flat heads 1-4 and the flange to which flat head 5 is bolted are treated in the finite element analyses of inner vessel assemblies presented later in this note).

The appropriate formula is given in UG-34. To use this formula, it is necessary to calculate the total bolt load in lbs, W, using the procedure of Appendix 2.

The required bolt load for the operating conditions is

$$W_{m1} = 0.785\ G^2P + (2b \times 3.14\ GmP) \quad \text{UG-34(c)(2)}$$

where

G = diameter at location of gasket load reaction = 17 in.

P = pressure = 70 psid = 70 psig (There is no vacuum load on this member)

b = effective gasket seating width = 0.03125 (Table 2-5.2)

m = gasket factor = 0 (for o-rings, Table 2-5.1)

Then,

$$W_{m1} = 0.785\ (17)^2(85) = 15880 \text{ lbs}$$

The minimum initial bolt load required for gasket seating is

$$W_{m2} = 3.14\ bGy$$

where y = minimum design seating stress for gasket = 0 (for o-rings, Table 2-5.1)

b = as above

G = as above

Then,

$$W_{m2} = 0$$

Then the total bolt load is $W = W_{m1} + W_{m2} = 19284$ lbs. Note: That this reduces to bolt preload = maximum pressure load.

The required head thickness for an unstayed flat head is given by UG-34 as

$$t = d(CP/SE + 1.9\ Wh_g/SEd^3)^{1/2}$$

where

d = diameter of head = 17 in.

C = factor to account for means by which head is attached = 0.3

S = maximum allowable stress in tension of head = 15000 psi

E = efficiency of welds in head = 1 (seamless head)

W = total bolt load = 19284 lbs

h_g = gasket moment arm = 0.375 in

P = pressure = 70 psid = 70 psig

Then,

$$t = 0.738$$

The actual head thickness is 1.0 in.

Therefore, the MAWP of flat head 5 is greater than 70 psid = 70 psig.

The stresses in the bolts attaching the flat head to the flange must be checked. Twenty-four 5/16-18 UNC 304 stainless steel bolts are used. The root diameter stress area of a single bolt is 0.0454 in^2. Then, the stress in a single bolt is

$$\sigma_b = \frac{15880}{24(0.0454)}$$

$$\sigma_b = 14570 \text{ psi}$$

The maximum allowable stress in tension is 15000 psi for annealed 304 stainless steel bolts from Div. 1 with the 14.1 derating. Therefore, the number and size of the bolts is adequate.

9. Finite Element Analyses of Inner Vessel Assemblies

It is clear from the calculation of MAWP for torispherical head 2 that the use of Div. 1 rules as modified by 14.1 produces an MAWP for the inner vessel which is unacceptably low given the possibility of large pressure rises during a magnet quench.

A new approach will now be taken which will show by detailed finite element stress analysis that the dewar can be safely operated at a higher pressure than indicated by the Div. 1 calculations.

The finite element method will be applied to the following three inner vessel assemblies:

Neck Assembly

This is defined to consist of torispherical head 2, cylindrical shells 2, 3, and 4, and flat heads 1, 2, and 3. Division 1 calculations have already been presented for the cylindrical shells and torispherical head based on pressure loading alone. Although Div. 1 rules are available for the flat heads (see UG-39 and Appendix 14) these calculations have not been done in favor of modeling the entire assembly including all loadings, i.e. pressure, weight and thermal gradient due to conduction.

Top Hat/Flange Gusset Assembly

This is defined to consist of cylindrical shell 5, the top hat flange, and the 12 gussets which brace the flange to the shell. There are no rules presented in Section VIII for the design of a gussetted flange, nor is such a design prohibited. Therefore, the finite element analysis is appropriate for this assembly.

Top Hat/Vacuum Vessel Gusset Assembly

This is defined to consist of cylindrical shell 5, cylindrical shell 7, flat head 4, and the 12 gussets which connect these components. There are no rules presented in Section VIII for the design of a gussetted shell/head assembly, nor is such a design prohibited. Therefore, the finite element analysis is appropriate for this assembly.

The finite element results were evaluated according to Section VIII, Div. 2, Appendix 4, Article 4-1, "Design Based on Stress Analysis". These rules allow stresses to be categorized according to origin, extent, and self-limiting characteristics. Stresses in each category must be limited in value to a multiple of S_m, the maximum allowable stress intensity for the material. The approach taken in the following analysis is to generate the stresses which result from a pressure loading of 85 psid (plus weight loading and thermal gradient where appropriate), then to calculate the value of S_m necessary to allow these loads. This S_m will then be compared to the maximum allowable stress in tension as established for the material by strict interpretation of Div. 1 and 14.1, and any deviations will be discussed. (The comparison of stress intensity to allowable stress in tension is legitimate, basically involving a conceptual jump from the maximum normal stress failure theory of Div. 1 to the more elaborate, but still conservative, maximum shear stress failure theory of Div. 2).

9a. Finite Element Analysis of Neck Assembly

An axisymmetric finite element model was made using the ANSYS four node area element STIF42. This model is shown in Fig. 2. The weight of the dewar, contents, and the unbalanced pressure force was applied at the outer diameter of torispherical head 2. The weight of the copper nitrogen shield was applied to cylindrical shell 4 at junction 2. A simple support was used on the outer diameter of flat head 3. This is conservative, since the actual construction allows a moment constraint to develop at the connection of flat head 3 and flat head 4.

The bellows in the dewar neck is assumed to have failed, and 85 psid is applied to the appropriate surfaces as shown in Fig. 2. This is a worse case.

In addition to mechanical loads, thermal stresses were also considered. Two modes were investigated: 1) operation, and 2) venting. During normal operation it is assumed that all of torispherical head 2 up to junction 4 will be at 4.4 K. During venting, the entire neck is assumed to be 4.4 K up to junction 3. The gradients are calculated by ANSYS based on conduction alone. The resulting nodal temperatures are input to the structural models to calculate stresses.

The stress results for the neck assembly are summarized in Table II for the venting thermal gradient, which produced the highest stresses. Figs. 3-6 show the stress intensities in the components. Table III compares the required S_m for each component to the maximum allowable stresses defined in three ways: 1) full Div. 1 allowable stress, 2) Div. 1 allowable stress times 0.8 as required by UW-12 for non-radiographed vessels <u>or</u> 14.1 for in-house vessels, and 3) Div. 1 allowable stress times both the UW-12 derating <u>and</u> the 14.1 derating. As can be seen from the table, cylindrical shell 2 and 4 exceed all but the full Div. 1 allowable stresses. The effective derating from Div. 1 allowable stress is 13400/16000 = 0.84 for cylindrical shell 2, and 14400/16000 = 0.9 for cylindrical shell 4. The safety factors for these components in terms of ultimate tensile strength (75000 psi minimum from Table UHA-23) are 5.2 for cylindrical shell 2, and 5.6 for cylindrical shell 4. The reason these safety factors appear to comply with 14.1 is that the cylindrical shells are welded pipe, and the full Div. 1 allowable for these shells already reflects a derating of 0.8. (There are, in fact, three 0.8 factors which are used if 14.1 and Div. 1 are strictly interpreted, i.e., UW-12, 14.1, and the welded pipe derating).

The neck assembly is judged to be good for 85 psid for the following reasons.

1. Safety factors based on ultimate tensile strength are greater than 5 for all components.

2. The determining stresses in cylindrical shells 2 and 4 are the primary

local membrane and secondary stresses, which are very unlikely to cause failure due to their ability to redistribute by localized yielding.

3. The Div. 1 calculations for internal MAWP presented previously used both the 14.1 derating and the UW-12 derating, as well as the derating implicit in the Div. 1 allowables for welded pipe, and showed the cylindrical shells to be capable of withstanding more than 85 psid. These were primary membrane stress calculations, and are verified by the ANSYS results.

4. The weld efficiency of 0.65 for the butt welds by which shells and heads are attached has not been addressed by the finite element work. It is felt that the safety factors are sufficient to account for this efficiency. (See Part III for Div. 1 calculations of weld strength under primary loadings).

Therefore, the MAWP of the helium dewar neck assembly is 85 psid.

9b. Finite Element Analysis of Top Hat/Flange Gusset Assembly

Previous analyses of the dewar top hat flange indicated that this flange was inadequately sized for 85 psid according to Div. 1 rules. Twelve 1/4 in. stainless steel gussets were installed around the flange to brace it against the top hat shell (cylindrical shell 5).

A finite element model was made using eight node solid ANSYS STIF45 elements (Fig. 7). This model is a 30 degree sector. Symmetry boundary conditions were applied as shown in the figure. The lower edge of the top hat shell was constrained in the axial direction only. The bolt loading resulting from a pressure of 85 psid on flat head 5 was applied as concentrated loads on nodes at the bolt locations. An internal pressure of 85 psid was applied to the top hat shell. A gap 1 in. long was created between the gusset and top hat shell to more accurately model the skip weld used to attach these parts.

The resulting maximum stress intensity in the top hat shell is 18900 psi. This occurs at the junction of the shell and the lower bottom edge of the gusset as shown in Fig. 8. This is a local primary membrane plus secondary stress, and is limited to 3 S_m. The required S_m is then 6300 psi, which is well below the maximum allowable stress for the material given by Div. 1 and modified by 14.1. The local primary membrane component alone is 7500 psi. This requires an S_m of 5000 psi, which again is well below the allowable.

The maximum stress intensity in the gusset is 9200 psi. This occurs at the bottom of the gusset near the top hat shell as shown in Fig. 9. This stress is less than the maximum allowable stress for the material even if this stress is classified as a primary stress. Low stresses throughout the gusset are expected since it is acting as a very deep beam in this configuration.

Evaluation of flange stresses must be done using the allowable flange design stresses given in Appendix 2 of Div. 1. This Appendix considers the various stress components (longitudinal, radial, and tangential) and establishes allowable stresses as multiples of the maximum allowable stress for the flange material. The lowest allowable stress for flange stresses is the maximum allowable stress for the material, or 15000 psi for the flange (no derating for non-radiography is required by UW-12(c) for flanges). Since the highest stress intensity found in the flange is 7000 psi (Fig. 9), it is clear that the flange stresses are all within acceptable limits.

The strength of the fillet welds by which the gussets are attached to the shell and flange must be addressed. The nodal force output of ANSYS for nodes at the interface of the gusset, shell and flange was used to calculate the maximum shear stress in the fillet throats[1].

Figure 10 shows the forces acting on the welds. The attachment welds to the top hat shell will be considered first. All fillet welds are assumed to have a 3/16 in. leg.

The external moment acting on the weld in Fig. 10 about A-A is

$$M = 1.0\ (517 + 188) = 705\ \text{in}\cdot\text{lbs}$$

The resultant normal force is

$$F = 517 - 188 = 329\ \text{lbs}$$

which acts to put the welds in tension.

The moment of inertia of the four fillet welds about A-A is (assuming 3/16 fillet legs)

$$I_{AA} = 4[(3/16)(1.0)^3 / 12 + 1.0\ (3/16)(1)^2](0.707)$$

$$I_{AA} = 0.57\ \text{in}^4$$

The maximum bending stress in the throat of the weld caused by the moment is

$$\sigma_b = \frac{Mc}{I} = \frac{705\ (1.5)}{0.57}$$

$$\sigma_b = \pm\ 1855\ \text{psi}$$

The direct stress resulting from the normal force is

$$\sigma_d = \frac{P}{A}$$

$$\sigma_d = \frac{329}{4(0.707)(3/16)(1.0)}$$

$$\sigma_d = 620\ \text{psi}$$

Then the maximum normal stress is

$$\sigma_n = \sigma_d + \sigma_b = 2475 \text{ psi.}$$

The average shear stress in the fillets is

$$\tau = \frac{P}{A}$$

$$\tau = \frac{(632 + 730)}{4(0.707)(3/16)(1.0)}$$

$$\tau = 2568 \text{ psi}$$

The maximum shear stress in the fillet welds is then

$$\tau_{max} = [(\sigma_n/2)^2 + \tau^2]^{1/2}$$

$$\tau_{max} = 2850 \text{ psi}$$

This result may be compared with a second method

$$\tau_{max} = \frac{\text{resultant force in bottom weld}}{\text{throat area}}$$

$$= \frac{(517^2 + 730^2)^{1/2}}{2 \times 0.707 \times 3/16} = 3370 \text{ psi}$$

The maximum shear stress allowed in the throat of a fillet weld is limited by UW-15 to 0.49 x maximum allowable stress for the shell material, or 0.49 x 12800 x 0.80 (from UW-12) = 5018 psi. Since 2850 5018, the welds attaching the gussets to cylindrical shell 5 meet the strength requirements of Div. 1 as modified by 14.1.

The welds attaching the gusset to the flange must also be considered. Figure 10 shows the total normal force on this weld to be 1362 lbs. Assuming failure is by shear in the fillet throat, the nominal fillet throat shear is

$$\tau = \frac{P}{A}$$

$$\tau = \frac{1362}{2(3/16)(0.707)(1.5)}$$

$$\tau = 3425 \text{ psi}$$

This shear stress is less than the allowable of 0.49 x 15000 x 0.80 (from UW-12) = 5880 psi. Therefore, the welds attaching the gussets to the flange meet the strength requirements of Div. 1 as modified by 14.1, and the MAWP of the tophat/flange gusset assembly is 85 psid.

9c. Finite Element Analysis of Tophat/Vacuum Vessel Gusset Assembly

Preliminary analyses of the dewar tophat/vacuum vessel junction indicated that this structure was inadequately sized for 85 psid according to Div. 1 rules. Therefore, twelve 3/8 in. stainless steel gussets were installed around the junction (flat head 4) which braced it to the vacuum shell neck (cylindrical shell 7) and the tophat shell (cylindrical shell 5). A finite element analysis was then performed to verify the suitability of this gusseting.

The finite element model is made of eight node solid ANSYS STIF45 elements (Fig. 11). This model is a 15 degree sector. Symmetry boundary conditions were applied as shown in the figure. The lower edge of the vacuum shell neck was constrained in the axial direction only. The longitudinal stress resulting from a pressure of 85 psid was applied to the tophat shell as shown and a pressure of 85 psid was applied to the inner surface of this shell. A pressure of 15 psid was applied to the outer surface of the vacuum shell neck. The welds which attach the gusset to the shells are continuous fillets, so the modeling of weld gaps was not necessary.

The resulting stress intensities and locations within the vessel components are summarized in Table IV. As can be seen, the required S_m for each component is well below the allowable stress even if it is assumed that all stresses are primary membrane stresses. Figure 12 shows the stress intensity through the most highly stressed region of the gusset.

The strength of the fillet welds by which the gussets are attached to the shells must be considered. Figure 13 shows the forces on the welds as calculated by ANSYS. The attachment weld to the vacuum shell neck will be considered first. All fillet welds are assumed to have a 3/16 in. leg.

The area of the throat of the welds to the vacuum shell neck is

$$A = 5.5(0.707)(2)(3/16)$$

$$A = 1.46 \text{ in}^2$$

The normal stress in the weld is

$$\sigma_n = \frac{264}{1.46}$$

$$\sigma_n = 181 \text{ psi}$$

The shear stress in the weld is

$$\tau = \frac{1192}{1.46}$$

$$\tau = 816 \text{ psi}$$

Then the maximum shear stress in the weld is

$$\tau_{max} = [(\sigma_n/2)^2 + \tau^2]^{1/2}$$

$$\tau_{max} = 821 \text{ psi}$$

The shear stress is well below the allowable of 0.49 x 12800 x 0.80 (from UW-12) = 5018 psi.

Therefore, this weld is adequate.

The next weld considered is between the gusset and flat head 4. The area of the throat of this weld is

$$A = 1.0(0.707)(3/16)(2)$$

$$A = 0.265 \text{ in}^2$$

The normal stress in the weld is

$$\sigma_n = \frac{720}{0.265}$$

$$\sigma_n = 2717 \text{ psi}$$

The shear stress in the weld is

$$\tau = \frac{464}{0.265}$$

$$\tau = 1751 \text{ psi}$$

Then the maximum shear stress in the weld is

$$\tau_{max} = [(\sigma_n/2)^2 + \tau^2]^{1/2}$$

$$\tau_{max} = 2216 \text{ psi}$$

This shear stress is well below the allowable of 0.49 x 15000 x 0.80 (from UW-12) = 5880 psi.
Therefore, this weld is adequate.

The last weld to be considered is the one between the gusset and the tophat shell. The area of the throat of this weld is

$$A = 3.5(0.707)(3/16)(2)$$

$$A = 0.93$$

The normal stress in the weld is

$$\sigma_n = \frac{200}{0.93}$$

$$\sigma_n = 215 \text{ psi}$$

The shear stress in the weld is

$$\tau = \frac{472}{0.93}$$

$$\tau = 507 \text{ psi}$$

Then the maximum shear stress in the weld is

$$\tau_{max} = [(\sigma_n/2)^2 + \tau^2]^{1/2}$$

$$\tau_{max} = 518 \text{ psi}$$

This shear stress is well below the allowable of 0.49 x 12800 x 0.80 (from UW-12) = 5018 psi. psi. Therefore, this weld is adequate, and the MAWP of the tophat/vacuum vessel gusset assembly is 85 psid.

Conclusion

The MAWP (internal) of the inner vessel is 85 psid.

B. External MAWP of Inner Vessel

1. Cylindrical Shell 1

The appropriate procedure is given in UG-28

$L = 31.75 + 2\ (1.5 + 1/3 \times 8) = 40.1$ in. UG-28(b)(1)

$t = 0.375$ in. $L/D_o = 0.84$

$D_o = 0.375$ in. $A = 0.0012$ (Fig. 5-UGO-28.0)

$D_o/t = 128$ $\qquad$ $B = 9500$ (Fig. 5-UHA-28.1)

Then,

$$P = \frac{4B}{3(D_o/t)}$$

$P = 99$ psid

Therefore, the MAWP (external) of cylindrical shell 1 is 99 psid.

2. Cylindrical Shell 2

The appropriate procedure is given in UG-28

$L = 12.375$ in.	$D_o/t = 58.3$
$t = 0.148$ in.	$L/D_o = 1.43$
$D_o = 8.625$ in.	$A = 0.002$
	$B = 11500$

Then,

$$P = \frac{4B}{3(D_o/t)}$$

$P = 263$ psid

Therefore, the MAWP (external) of cylindrical shell 2 is 286 psid.

3. Cylindrical Shell 3

The appropriate procedure is given in UG-28

$L = 8.125$ in.	$D_o/t = 65$
$t = 0.165$ in.	$L/D_o = 0.76$
$D_o = 10.75$ in.	$A = 0.004$
	$B = 13000$

Then,

$$P = \frac{4B}{3(D_o/t)}$$

$P = 267$ psid

Therefore, the MAWP (external) of cylindrical shell 3 is 267 psid.

4. Cylindrical Shell 4

The appropriate procedure is given in UG-28

$L = 9.3$ in. $\quad D_o/t = 77$

$t = 0.165 \quad L/D_o = 0.73$

$D_o = 12.75$ in. $\quad A = 0.0025$

$B = 11500$

Then,

$$P = \frac{4B}{3(D_o/t)}$$

$P = 199$ psid

Therefore, the MAWP (external) of cylindrical shell 4 is 199 psid.

5. Cylindrical Shell 5

This shell contains an opening in excess of 3 in. in diameter which requires reinforcement. UG-37(c)(1) requires that the area of reinforcement A = 1/2 d t_r F where t_r is the minimum thickness for external pressure. The wall thickness in the shell is partitioned accordingly assuming a worse case of no reinforcement contribution from the neck and F=1.

$$t_a = t_r + 0.5\ t_r$$

where t_a is the actual thickness = 0.188 and t_r is the required thickness for external pressure.

$$t_r = 0.667\ t_a = 0.126"$$

This thickness will be used with the procedure of UG-28.

$L = 7$ in. $\quad D_o/t = 127$

$t = 0.126 \quad L/D_o = 0.44$

D_o = 16 in. $\quad$ A = 0.0024

B = 12000

Then,

$$P = \frac{4B}{3(D_o/t)}$$

P = 126 psid

Therefore, the MAWP (external) of cylindrical shell 5 is 126 psid.

6. Torispherical Head 1

The appropriate procedure is given in UG-33. This procedure requires that two calculations be performed, and the lesser of the two pressures chosen.

From UG-33(a)(1)

L = 48 in

$t = t_{min} = t_{nom} - t_{tol} = 0.344$ in.

S = 15000 psi

E = 1

Then,

$$P = \frac{S\ E\ t}{1.67(0.885L+0.1t)}$$

P = 73 psid

From UG-33(a)(2)

R_o = 48 $\quad$ $A = 0.125/(R_o/t) = 0.0009$

$t = t_{min} = 0.344$ $\quad$ B = 8800 (Fig. 5-UHA-28.1)

$R_o/t = 139$

Then,

$$P = \frac{B}{(R_o/t)}$$

$P = 63$ psid

Therefore, the MAWP (external) of torispherical head 1 is 63 psid.

7. Torispherical Head 2

This head contains a large central opening which must be reinforced (see internal MAWP calculation for this head). Using UG-37(c)(1) and the same procedure for calculating required thickness as cylindrical shell #5, $t_r = 0.667$ $t_a = 0.344 \times 0.667 = 0.229"$. This thickness will now be used with the procedures of UG-33(a).

From UG-33(a)(1),

$L = 48$ in $\qquad t = 0.229$ in

$S = 15000 \qquad E = 1$

Then,

$$P = \frac{S\ E\ t}{1.67(0.885L+0.1t)}$$

$P = 48$ psid.

From UG-33(a)(2)

$R_o = 48 \qquad A = 0.125/(R_o/t) = 0.0006$

$t = 0.229 \qquad B = 7100$

$R_o/t = 210$

Then,

$$P = \frac{B}{(R_o/t)}$$

$P = 34$ psid

Therefore, the MAWP (external) of torispherical head 2 is 34 psid.

8. Flat Heads 1-5

Flat heads do not require consideration of stability. The MAWP (external) for these heads is the same as the MAWP(internal), or 85 psid.

9. Tophat Gussets

The gussets on the top hat and vacuum shell are not necessary for external pressure loading. Under external pressure they serve to brace cylindrical shell 5, which cannot see an external pressure greater than 15 psid, and is in fact rated at 147 psid external.

Conclusion

The MAWP (external) of the inner vessel is 34 psid.

II. Determination of Maximum Allowable Working Pressure (Internal) and External) of Vacuum Vessel

The components of the vacuum vessel are shown in Fig. 14.

A. Internal MAWP of Vacuum Vessel

1. Cylindrical Shell 6

The appropriate formula is given in UG-27 of Div. 1

$R = 27$ in.
$t = 0.375$
$E = 0.70$ for double weld butt joint, circumferential stress calculation
$E = 0.65$ for single weld butt joint, longitudinal stress calculation
$S = 15000$ psi

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 145 \text{ psid circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 272 \text{ psid longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 6 is 145 psid.

2. Cylindrical Shell 7

The appropriate formula is given in UG-27 of Div. 1.

R = 7 in.
t = 0.188
E = 1 for circumferential stress calculation
E = 0.60 for single welded butt joint, longitudinal stress calculation
S = 12800 x 0.80 = 10240 psi for circumferential stress calculation
S = 12800 psi for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 271 \text{ psid} \qquad \text{circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 417 \text{ psid} \qquad \text{longitudinal stress}$$

Therefore, the MAWP (internal) of cylindrical shell 7 is 271 psid.

3. Torispherical Head 3

This head contains a 8.635 in. diameter hole, which requires reinforcement according to UG-36. It was shown previously in this note that the amount of head thickness available for pressure loading after consideration of hole reinforcement is $t_{min}/2$. Therefore, this thickness is used in the formula of UG-32(e) to determine the pressure rating of the head.

L = 54

$t = t_{min}/2 = 0.344/2 = 0.172$ in.

S = 15000 x 0.80 = 12000

E = 1 (seamless heads)

Then,

$$P = \frac{S\ E\ t}{0.885L+0.1t}$$

P = 43 psid

Therefore, the MAWP (internal) of torispherical head 3 is 43 psid.

4. Torispherical Head 4

This head contains two openings which require reinforcement. These openings are spaced closer than two times their average diameter, which causes the reinforcement areas to overlap. UG-42 contains rules for the reinforcement of multiple openings under these circumstances.

The critical plane for reinforcement contains the centerlines of both openings. The overlapping area may be proportioned between openings according to the ratio of their diameters. The result is that the metal available for reinforcement of the 14 in. diameter hole is confined to a circle of 10 in. radius about the opening centerline. The metal available for reinforcement of the 6.635 in. diameter hole is confined to a circle of 6 in. radius about the opening centerline.

The 14 in. diameter hole is considered first. The area required for reinforcement is given by UG-37

$$A = dt_rF$$

with $d = 14$ in.
$F = 1$
t_r = required minimum head thickness for pressure to be determined

Then,

$$A = 14t_r$$

The available area for reinforcement is

$$A = (20-14)(t_A - t_r)$$

where t_A = actual thickness of head. Then,

$$A = 6(0.344 - t_r)$$

Equating the required and available areas gives

$$t_r = 0.103 \text{ in.}$$

This calculation is repeated for the 6.635 in. diameter opening, giving

$$t_r = 0.153$$

Then, the procedure of UG-32(e) shall be used with $t = 0.103$

$$L = 54$$

$$t = t_{min} = 0.103$$

$$S = 15000 \times 0.80 = 12000$$

$E = 1$ (seamless head)

Then,

$$P = \frac{S\ E\ t}{0.885L+0.1t}$$

$P = 26$ psid

Alternately one could argue that the two openings can be replaced by a single opening whose diameter encompasses both openings as per UG-42(c). Then the head would have a pressure rating of 43 psid, the same as torispherical head #3.

Conclusion

The more conservative method is chosen. The MAWP (internal) of vacuum vessel is 26 psid = 26 psig.

B. External MAWP of Vacuum Vessel

1. Cylindrical Shell 6

The applicable procedure is given in UG-28(c).

$L = 36$ in.	$D_o/t = 144$
$t = 0.375$ in.	$L/D_o = 0.67$
$D_o = 54$ in.	$A = 0.0015$
	$B = 9800$

Then,

$$P = \frac{4B}{3(D_o/t)}$$

$P = 91$ psid

2. Cylindrical Shell 7

The applicable procedure is given in UG-28(c).

$L = 10$ in.	$D_o/t = 74$
$t = 0.188$	$L/D_o = 0.71$

D_o = 14 in. A = 0.0035

B = 12800

Then,

$$P = \frac{4B}{3(D_o/t)}$$

P = 231 psid

3. Torispherical Head 3

This head contains a large central opening which must be reinforced (see internal MAWP calculation for this head). Using UG-37(c)(1) and the same procedure as for torispherical head 2 under external pressure, t_r = 0.67 t_a = 0.229".

This procedure requires that two calculations be performed and the lesser of the two pressures chosen.

From UG-33(a)(1)

L = 54
t = 0.229
S = 15000 psi
E = 1

Then,

$$P = \frac{S\ E\ t}{1.67(0.885L+0.1t)}$$

P = 43 psid

From UG-33(a)(2)

R_o = 54 A = $0.125/(R_o/t)$ = 0.00054

t = 0.229 B = 6800

R_o/t = 236

Then,

$$P = \frac{B}{(R_o/t)}$$

$$P = 29 \text{ psid}$$

Therefore, the MAWP (external) of torispherical head 3 is 29 psid.

4. Torispherical Head 4

This head contains two openings which require reinforcement and whose reinforcement areas overlap. (See internal MAWP calculation for this head). This head also supports the weight of the inner vessel and its contents and the nitrogen shield, for a total weight loading of 1676 lbs.

The weight loading tends to add to the vacuum loading, decreasing the stability of the head. This effect is difficult if not impossible to calculate. For the purposes of this calculation, weight loading will be ignored. A discussion of the effects of weight on vessel components will be presented in Part VI of this note.

The two openings are treated as a single opening with a diameter which encompasses both of them as allowed per UG-42(c). The external pressure rating is, therefore, identical to that of torispherical head #3 which is 29 psid. Note that psi x area of the head = 2290 lbs > total applied dead weight. Thus, a more than adequate safety margin has been provided for stability. Therefore, the MAWP (external) of torispherical head #4 is 28 psid.

Conclusion

The MAWP (external) of the vacuum shell is 28 psid.

III. Verification of Weld Strength

The welds on the helium inner vessel must meet the requirements of UW-2(b) for cryogenic service. The general Div. 1 requirements of UW-13, UW-15, UW-16, and UW-18 also apply.

The welds must be sized adequately for the MAWP of the vessel regardless of whether the weld detail itself is permitted by Div. 1. (See Part VI for a discussion of non-Code weld details). In determining weld strength, the allowable stresses for groove weld tension and shear, and fillet weld shear as given in UW-15 must be used. To simplify the calculation of stresses in each weld, the relationship between MAWP, allowable stress, and weld dimension for acceptable welds will be developed.

For groove welds in tension:

d = diameter of opening or O.D. of pipe

S_A = allowable stress

t_1 = thickness of weld normal to load

F = pressure force on weld = $(\pi d^2/4)$MAWP

A = cross sectional area of weld in tension = πdt_w

Then, the tensile stress in the weld is

$$\sigma_w = F/A$$

$$\sigma_w = \frac{(\pi d^2/4)\text{MAWP}}{\pi dt_1}$$

$$\sigma_w = \frac{d(\text{MAWP})}{4t_1}$$

For acceptable welds,

$$\sigma_w \leq S_A$$

1) $$\frac{d(\text{MAWP})}{4t_1} \leq S_A$$

For groove welds in shear: (d, S_A, and F as before)

t_2 = thickness of weld parallel to weld

A = cross sectional area in shear = πdt_2

Then, for an acceptable weld,

2) $$\frac{d(\text{MAWP})}{4t_2} \leq S_A$$

For fillet welds in shear (d, S_A, and F as before)

t_3 = fillet leg x 0.707

A = cross sectional area in shear = πdt_3

Then, for an acceptable weld,

3) $$\frac{d(MAWP)}{4t_3} \leq S_A$$

The results of the weld strength calculations for the welds shown in Fig. 15 are summarized in Table V. Those welds which must resist additional non-pressure loading have this loading included in the MAWP used in the calculation.

(NOTE: Indication of openings in tophat shell [cylindrical shell 5] in Fig. 15 is schematic only).

IV. System Venting Verification

Due to the coupling of the dewars and magnet cryostats and the use of a common vacuum space, the subject of system venting is discussed in a separate note by W. Craddock.

V. Pressure Test Results

The internal vessel was pressure tested to 93 psig (95 psig for dewar B) while subjected to a vacuum loaad of 15 psi. Required Div. 1 pneumatic test pressure = (85 psid x 1.25) = 106 psid = 91 psig + vacuum. The vacuum vessel was tested to only 24 psig (internal) limited by the pressure rating of the connected magnet vacuum shell.

VI. Exceptional Vessel Details

There are three areas in which the 30-inch helium dewars do not comply with all of the requirements of 14.1. These are A) thickness of torispherical head 4 for vacuum and weight loading, B) opening reinforcement and stresses in the neck assembly, and C) welding.

A. Thickness of Torispherical Head 4 for Vacuum and Weight Loading

The Div. 1 external pressure calculation for this head showed an MAWP (external) of 16 psid. In addition to the vacuum loading, this head must also support the weight of the dewar, contents, and nitrogen shield. This is a total weight of 1680 lbs. This weight loading tends to add to the vacuum loading, and its effect on the stability of the head is very difficult to estimate. Div. 1 makes no provision for the inclusion of non-pressure external loads in the calculation of shell thickness under external pressure; however it is clear from the vacuum only calculation that the head does not meet Div. 1 (and hence 14.1) requirements if weight loading is considered. In arguing for the acceptability of this head, the following points should be considered:

1. The Div. 1 calculations for shells under external pressure result in a safety factor of approximately four.[2]

2. The head thickness available for pressure was greatly reduced due to the reinforcement requirements of the head opening. Since reinforcement need only be considered for one opening diameter on either side of the opening centerline (UG-40), the bulk of the head has a thickness of 0.375 inches available for external loads, although Div. 1 rules make no provision for considering this effect in the calculations.

3. If the simple-minded approach is taken of converting the weight loading into a pressure applied across the 14 in. diameter opening, then this pressure is 11 psid, and the head would require an external pressure rating of 27 psid to account for all loads. If a safety factor of 4 is available from the Div. 1 rating of 16 psid, then this approach would still result in a safety facctor of 2.4.

4. UG-42(c) allows two openings to be treated as a single opening with a diameter enclosing both openings. UG-36(b)(2) permits any size opening in the head of a cylindrical vessel but recommends a reversed curve section for openings greater than 1/2 the inside head diameter. We are only 3/8" under this recommendation. Treated a single opening the external pressure rate of head #4 = head #3 = 33 psid.

Therefore, given that the pressure differential cannot exceed one atmosphere and that the Div. 1 calculation is conservative both in formulation and in the head thickness used, then it is felt that this head presents no unusual danger to personnel or equipment and should be exempted from the Div. 1 requirements for head thickness under external pressure.

B. Opening Reinforcement and Stresses in the Neck Assembly

The Div. 1 calculations presented in Part I of this note for torispherical head 2 (the top inner vessel head) showed that strict interpretation of the opening reinforcement requirements of UG-36 and the use of Div. 1 allowable stresses as derated by 14.1 gives an MAWP of 49 psid for the head. The subsequent finite element analysis, evaluated by the rules of Appendix 4 of Div. 2, demonstrated that the MAWP could be raised to 85 psid with a safety factor greater than 5 on the minimum specified tensile strength of the head material. (NOTE: The use of Appendix 4 does not imply a "Div. 2" design. This Appendix is used because it provides a rational way to evaluate stresses in a pressure vessel).

It is felt that the spirit of 14.1 is satisfied in view of the resulting safety factors in the neck assembly, although the stresses slightly exceed the maximum allowable values as derated by 14.1. (See Table III). The neck assembly should pose no unusual danger to personnel or equipment, and can withstand pressures well in excess of 85 psid should this be necessary during a magnet quench.

C. Welding

1. Weld Design

UW-2 of Div. 1 requires that all parts of the vessel operating at cryogenic temperatures shall use full penetration welds. This should probably be applied to the entire inner vessel including neck and top hat, since the venting of the system will cool these components concident with a high pressure. Several of the welds on the inner vessel do not meet this requirement.

The welds which are not full penetration are fillet welds. UW-13, UW-16, and UW-18 cover the requirements for fillet welds. Several welds do not meet these requirements.

The welds which do not meet the Div. 1 design requirements are shown in Table VI.

It can be seen from Table VI and Fig. 15 that welds 5, 11, and 23 are the only non-Code welds which are also strength welds between major pressure parts. It was realized early in the analysis of this vessel that these welds were inadequate. This led to the installation of the gussetting around the tophat. The analysis of the gussetting in Part I of this note shows that it is very effective in strengthening the tophat assembly. There is no doubt that adequate strength now exists for a pressure of 85 psid.

Welds 2, 6, 7, 8, and 9 are single fillet welds and are not allowed by the Code. Section III proves that they are adequately sized based on Code allowable stress values. These calculations are appropriate since the welds are only required to carry pressure loads on the nozzles, and are not required to transmit any reinforcement loads. See UG-41, UW-15 and L-7 example 6. All reinforcement is due to a derating of the MAWP.

The requirement of full penetration welds for cryogenic service is probably based on the superior fatigue resistance of such welds rather than superior strength (it is certainly possible to produce full strength fillet welds). The reduction of fatigue properties with temperature certainly justifies extra caution. The following observations can be made on the use of fillet welds in this dewar.

1. The dewar was designed by an experienced cryogenic designer who used weld details which are accepted practice in Laboratory dewar design.

2. The dewar will operate at a constant pressure well below 85 psid. Any fluctuations in pressure tending to fatigue the welds will be rare and will occur at much smaller stresses than were calculated for 85 psid.

3. Part III of this note demonstrates that all of the fillet welds with the exception of 5 and 11 meet the strength requirements of Div. 1. Welds 5 and 11 are strengthened by gussetting.

It is felt, therefore, that the use of non-Code weld details on this vessel does not endanger personnel or equipment, and that this vessel should be exempted from this requirement.

2. Welding Procedures and Qualifications

14.1 requires that all welding on the vessels be performed using a procedure which is qualified under the rules of Section IX of the Code, and by a welder qualified in the procedure under Section IX rules. All of the vessel welding with the exception of the longitudinal seams in cylindrical shells 1 and 6 was performed at Fermilab in the spring of 1982. Atached to this note is a copy of welding procedure ES-155001, and the welder qualification records for the two welders which worked on the vessel. The procedure covers the TIG welding of stainless steel, and applies to all welds, fillet or groove, on the vessel. The welder qualifications show both welders to have been qualified in this procedure in August, 1982. It appears that the 14.1 requirement for welding was met for welds done at the Laboratory. However, 14.1 requires a signed document from the welding foreman stating that the appropriate procedures and qualified welders were used. Such a document was not obtained. It is felt that the information presented here is persuasive enough to allow an exemption from the documentation requirement.

The longitudinal seam welds in cylindrical shells 1 and 6 were performed by an outside vendor according to Fermilab weld details. There is no record of procedures or qualifications of welders for this work. The following statements can be made concerning these welds.

1. The Fermilab weld detail is explicit and fully consistent with the Div. 1 requirements for butt welds.

2. Attached to this note is a letter from the vendor describing welding rods and heat numbers. Included also is a material certification sheet showing that the material meets ASME Code, Section II specification SA-240, with yield and tensiles strengths well above the specified minimums.

3. The welds affect the circumferential stress calculations for the shells. These calculations are presented in Part I and II of this note and show that cylindrical shells 1 and 6 have an MAWP (internal) of 162 psid and 145 psid, respectively. This is well above the MAWP of 85 psid for the inner vessel and 51 psid for the vacuum vessel.

It is felt that, given the material documentation, weld detail, and large MAWP of the shells, the welds on these shells can be safely exempted from the 14.1 welding requirements.

Conclusion

The 30-inch Bubble Chamber helium dewar inner vessels can be safely rated at 85 psid (internal). The details of these vessels which do not meet the requirements of 14.1 do not constitute an unusual danger to personnel or equipment. The MAWP (internal) of the vacuum shell is 26 psid.

References

1. Shigley, J.E., Mechanical Engineering Design, Third Edition, McGraw-Hill, 1977.

2. Compressed Gas Association, "Insulated Tank Truck Specification CGA-341".

Table I
Component Material, Dimensions, and Maximum Allowable Stress

Component	Material	Dimensions	Maximum Allowable Stress***
Cylindrical shell 1	304 SS	48 O.D. x 31-3/4 lg x 3/8 wall	15000
Cylindrical shell 2*	304 SS	8-5/8 O.D. x 12-3/8 lg x 0.148 wall	12800
Cylindrical shell 3*	304 SS	10-3/4 O.D. x 8-1/8 lg x 0.165 wall	12800
Cylindrical shell 4*	304 SS	12-3/4 O.D. x 9-5/16 lg x 0.165 wall	12800
Cylindrical shell 5*	304 SS	16 O.D. x7 lg x 0.188 wall	12800
Cylindrical shell 6	304 SS	54 O.D. x 36 lg x 3/8 wall	15000
Cylindrical shell 7*	304 SS	14 O.D. x 10 lg x 0.188 wall	12800
Torispherical head 1**	304 SS	48 O.D. x 3/8 wall	15000
Torispherical head 2**	304 SS	48 O.D. x 3/8 wall	15000
Torispherical head 3**	304 SS	54 O.D. x 3/8 wall	15000
Torispherical head 4**	304 SS	54 O.D. x 3/8 wall	15000
Flat head 1	304 SS	10-3/4 O.D. x 8-3/8 I.D. x 11/16 thk	15000
Flat head 2	304 SS	12-3/4 O.D. x 10-1/2 I.D. x 11/16 thk	15000
Flat head 3	304 SS	15 O.D. x 12-1/2 I.D. x 1/2 thk	15000
Flat head 4	304 SS	15-5/8 O.D. x 13-5/8 I.D. x 1/2 thk	15000
Flat head 5	304 SS	19 O.D. x 1 thk	15000
Flange	304 SS	19 O.D. x 15-5/8 I.D. x 1/2 thk gusseted	15000

*   welded pipe or tube
** standard ASME flanged and dished head
*** stress allowables are Div. 1 allowables x 0.8 as required by 14.1.

Table II
Finite Element Results for Neck Assembly
(Venting Thermal Gradient)

Component	Location	Stress Category**	ANSYS Stress Intensity	Required S_m
Flat Head 3*	--	--	--	--
Cylindrical Shell 4	Junction 1	P_L	12600	8400
	Junction 1	P_L+Q	40115	13400
	Away from Junction 1	P_m	3200	3200
	Junction 2	P_L	5080	3400
	Junction 2	P_L+Q	23083	7700
Flat Head 2*	--	--	--	--
Cylindrical Shell 3	Junction 2	P_L	4000	2700
	Junction 2	P_L+Q	5000	1700
	Away from Junction 2	P_m	2200	2200
	Junction 3	P_L	4300	2900
	Junction 3	P_L+Q	5100	1700
Flat Head 1*	--	--	--	--
Cylindrical Shell 2	Junction 3	P_L	8600	5700
	Junction 3	P_L+Q	14206	4800
	Away from Junction 3	P_m	2400	2400
	Junction 4	P_L	21585	14400
	Junction 4	P_L+Q	35887	12000
Torispherical Head 2	Away from Junction 4	P_m	5600	5600

*Stresses in the flat heads are well below any reasonable maximum allowable stress intensity for primary stresses (see Figs. 3-5).

**P_L = primary local membrane stress. Allowable is 1.5 S_m (Div. 2)

P_L+Q = primary local membrane + secondary stress. Allowable is 3 S_m (Div. 2)

P_m = primary membrane stress. Allowable is S_m (Div. 2)

Table III
Comparison of Maximum Required S_m in Neck Assembly with Various Allowable Stresses

Component	Max Required S_m	Maximum Allowable Stress Div. 1 (Full)	Div. 1 (UW-12or14.1)	Div. 1 UW-12&14.1)
Cylindrical Shell 4	13400	16000	12800	10240
Cylindrical Shell 3	2900	16000	12800	10240
Cylindrical Shell 2	14400	16000	12800	10240
Torispherical Head 2	5600	18800	15000	12000

Table IV
Finite Element Results for Tophat/Vacuum
Shell Gusset Assembly

Component	Location	Stress* Category	Stress Intensity	Required S_m
Cylindrical Shell 7	Junction with bottom of gusset	P_m	4000	4000
Flat Head 4	Junction with Cyl.Shell 5 and gusset	P_m	3300	3300
Cylindrical Shell 5	Junction with Flat Head 4	P_m	7000	7000
Gusset	Junction with Flat head 4	P_m	4400	4400

*All stresses are classified as primary membrane stresses to demonstrate compliance with primary stress requirements although some stresses may be secondary.

Table V
Summary of Weld Strength Calculations

Weld (Fig.15)	d	MAWP (psid)	t_i (i=1,2,3)	Type of Stress*	$\frac{d(MAWP)}{4t_i}$ (psi)	S_A** (psi)	Comments
1	8.625	15	3/8	g.s.	86	7200	good
2	0.50	85	0.044	f.s.	241	5880	good
3	2.38	85	0.044x2	f.s.	575	5880	good
4	6.635	15	3/8	g.s.	66	7200	good
5	16	85	0.044	f.s.	7727	5018	(1)
6	1.125	85	0.044	f.s.	2173	5018	good
7	1.50	85	0.044	f.s.	724	5018	good
8	0.50	85	0.044	f.s.	241	5018	good
9	4.50	85	0.044	f.s.	2173	5018	good
10	15.0	--	--	--	--	--	(2)
11	15.624	85	0.044	f.s.	7545	5018	(3)
12	14.0	26***	3/8	g.s.	242	6144	good
13	8.625	110.5***	3/8	g.s.	6.35	6144	good
14	12.75	99***	0.165	g.t.	1912	7577	good
15	10.75	101***	0.165	g.t.	1645	7577	good
16	54	15	3/8	g.t.	540	8880	good
17	48	85	3/8	g.t.	2720	8880	good
18	48	85	3/8	g.t.	2720	8880	good
19	54	15	3/8	g.t.	540	8880	good
20	8.625	110.5***	0.148	g.t.	1610	7577	good
21	10.75	101***	0.165	g.t.	1645	7577	good
22	12.75	99***	0.165	g.t.	1912	7577	good
23	14.0	26***	0.044	f.s.	2068	5018	good
24	7.0	85	3/8	g.s.	397	7200	good

* g.s. - groove shear
g.t. - groove tension
f.s. - fillet shear

** This is the lowest allowable stress for the components being joined multiplied by 0.8 (UW-12) and the appropriate factor from UW-15.

*** Reflects external loading

(1) This calculation does not include the strengthening effects of the gussets.
(2) This weld is not required for mechanical or pressure loads. It serves as a seal between helium and vacuum spaces.
(3) This calculation does not include the strengthening effects of the gussets.

Table VI
Welds Not Meeting Code Requirements

Weld No. (Fig. 15)	Weld Type*	Div. 1 Requirements Violated
2	S.F.	UW-2, UW-16(e)
3	D.F.	UW-12
5	S.F.	UW-2, UW-13(f)
6	S.F.	UW-2, UW-16(e)
7	S.F.	UW-2, UW-16(e)
8	S.F.	UW-2, UW-16(e)
9	S.F.	UW-2, UW-16(e)
10	S.F.	UW-2, UW-13(f)
11	S.F.	UW-2, UW-13(f)
23	S.F.	UW-2, UW-13(f)

*S.F.-single fillet
D.F.-double fillet

Table I
Component Material, Dimensions, and Maximum Allowable Stress

Component	Material	Dimensions	Maximum Allowable Stress***
Cylindrical shell 1	304 SS	48 O.D. x 31-3/4 lg x 3/8 wall	15000
Cylindrical shell 2*	304 SS	8-5/8 O.D. x 12-3/8 lg x 0.148 wall	12800
Cylindrical shell 3*	304 SS	10-3/4 O.D. x 8-1/8 lg x 0.165 wall	12800
Cylindrical shell 4*	304 SS	12-3/4 O.D. x 9-5/16 lg x 0.165 wall	12800
Cylindrical shell 5*	304 SS	16 O.D. x7 lg x 0.188 wall	12800
Cylindrical shell 6	304 SS	54 O.D. x 36 lg x 3/8 wall	15000
Cylindrical shell 7*	304 SS	14 O.D. x 10 lg x 0.188 wall	12800
Torispherical head 1**	304 SS	48 O.D. x 3/8 wall	15000
Torispherical head 2**	304 SS	48 O.D. x 3/8 wall	15000
Torispherical head 3**	304 SS	54 O.D. x 3/8 wall	15000
Torispherical head 4**	304 SS	54 O.D. x 3/8 wall	15000
Flat head 1	304 SS	10-3/4 O.D. x 8-3/8 I.D. x 11/16 thk	15000
Flat head 2	304 SS	12-3/4 O.D. x 10-1/2 I.D. x 11/16 thk	15000
Flat head 3	304 SS	15 O.D. x 12-1/2 I.D. x 1/2 thk	15000
Flat head 4	304 SS	15-5/8 O.D. x 13-5/8 I.D. x 1/2 thk	15000
Flat head 5	304 SS	19 O.D. x 1 thk	15000
Flange	304 SS	19 O.D. x 15-5/8 I.D. x 1/2 thk gusseted	15000

* welded pipe or tube
** standard ASME flanged and dished head
*** stress allowables are Div. 1 allowables x 0.8 as required by 14.1.

Table II
Finite Element Results for Neck Assembly
(Venting Thermal Gradient)

Component	Location	Stress Category**	ANSYS Stress Intensity	Required S_m
Flat Head 3*	--	--	--	--
Cylindrical Shell 4	Junction 1	P_L	12600	8400
	Junction 1	P_L+Q	40115	13400
	Away from Junction 1	P_m	3200	3200
	Junction 2	P_L	5080	3400
	Junction 2	P_L+Q	23083	7700
Flat Head 2*	--	--	--	--
Cylindrical Shell 3	Junction 2	P_L	4000	2700
	Junction 2	P_L+Q	5000	1700
	Away from Junction 2	P_m	2200	2200
	Junction 3	P_L	4300	2900
	Junction 3	P_L+Q	5100	1700
Flat Head 1*	--	--	--	--
Cylindrical Shell 2	Junction 3	P_L	8600	5700
	Junction 3	P_L+Q	14206	4800
	Away from Junction 3	P_m	2400	2400
	Junction 4	P_L	21585	14400
	Junction 4	P_L+Q	35887	12000
Torispherical Head 2	Away from Junction 4	P_m	5600	5600

*Stresses in the flat heads are well below any reasonable maximum allowable stress intensity for primary stresses (see Figs. 3-5).

**P_L = primary local membrane stress. Allowable is 1.5 S_m (Div. 2)

P_L+Q = primary local membrane + secondary stress. Allowable is 3 S_m (Div. 2)

P_m = primary membrane stress. Allowable is S_m (Div. 2)

Table III
Comparison of Maximum Required S_m in Neck Assembly with Various Allowable Stresses

Component	Max Required S_m	Maximum Allowable Stress Div. 1 (Full)	Div. 1 (UW-12or14.1)	Div. 1 UW-12&14.1)
Cylindrical Shell 4	13400	16000	12800	10240
Cylindrical Shell 3	2900	16000	12800	10240
Cylindrical Shell 2	14400	16000	12800	10240
Torispherical Head 2	5600	18800	15000	12000

Table IV
Finite Element Results for Tophat/Vacuum
Shell Gusset Assembly

Component	Location	Stress* Category	Stress Intensity	Required S_m
Cylindrical Shell 7	Junction with bottom of gusset	P_m	4000	4000
Flat Head 4	Junction with Cyl.Shell 5 and gusset	P_m	3300	3300
Cylindrical Shell 5	Junction with Flat Head 4	P_m	7000	7000
Gusset	Junction with Flat head 4	P_m	4400	4400

*All stresses are classified as primary membrane stresses to demonstrate compliance with primary stress requirements although some stresses may be secondary.

Table V
Summary of Weld Strength Calculations

Weld (Fig.15)	d	MAWP (psid)	t_i (i=1,2,3)	Type of Stress*	$\frac{d(MAWP)}{4t_i}$ (psi)	S_A** (psi)	Comments
1	8.625	15	3/8	g.s.	86	7200	good
2	0.50	85	0.044	f.s.	241	5880	good
3	2.38	85	0.044x2	f.s.	575	5880	good
4	6.635	15	3/8	g.s.	66	7200	good
5	16	85	0.044	f.s.	7727	5018	(1)
6	1.125	85	0.044	f.s.	2173	5018	good
7	1.50	85	0.044	f.s.	724	5018	good
8	0.50	85	0.044	f.s.	241	5018	good
9	4.50	85	0.044	f.s.	2173	5018	good
10	15.0	--	--	--	--	--	(2)
11	15.624	85	0.044	f.s.	7545	5018	(3)
12	14.0	26***	3/8	g.s.	242	6144	good
13	8.625	110.5***	3/8	g.s.	6.35	6144	good
14	12.75	99***	0.165	g.t.	1912	7577	good
15	10.75	101***	0.165	g.t.	1645	7577	good
16	54	15	3/8	g.t.	540	8880	good
17	48	85	3/8	g.t.	2720	8880	good
18	48	85	3/8	g.t.	2720	8880	good
19	54	15	3/8	g.t.	540	8880	good
20	8.625	110.5***	0.148	g.t.	1610	7577	good
21	10.75	101***	0.165	g.t.	1645	7577	good
22	12.75	99***	0.165	g.t.	1912	7577	good
23	14.0	26***	0.044	f.s.	2068	5018	good
24	7.0	85	3/8	g.s.	397	7200	good

* g.s. - groove shear
g.t. - groove tension
f.s. - fillet shear

** This is the lowest allowable stress for the components being joined multiplied by 0.8 (UW-12) and the appropriate factor from UW-15.

*** Reflects external loading

(1) This calculation does not include the strengthening effects of the gussets.
(2) This weld is not required for mechanical or pressure loads. It serves as a seal between helium and vacuum spaces.
(3) This calculation does not include the strengthening effects of the gussets.

Table VI
Welds Not Meeting Code Requirements

Weld No. (Fig. 15)	Weld Type*	Div. 1 Requirements Violated
2	S.F.	UW-2, UW-16(e)
3	D.F.	UW-12
5	S.F.	UW-2, UW-13(f)
6	S.F.	UW-2, UW-16(e)
7	S.F.	UW-2, UW-16(e)
8	S.F.	UW-2, UW-16(e)
9	S.F.	UW-2, UW-16(e)
10	S.F.	UW-2, UW-13(f)
11	S.F.	UW-2, UW-13(f)
23	S.F.	UW-2, UW-13(f)

*S.F.-single fillet
D.F.-double fillet

G.O.C[illegible]LSON Inc
Producers of Stainless Steel, Nickel Alloys and Titanium
THORNDALE, PA. 19372

TES[illegible]

GOC #	DATE
26345	2-18-83

CUSTOMER ORDER NUMBER	T/R CODE	CUSTOMER MARK
G2982	M	G2982

SOLD TO

ALL METAL MFG
9925 S INDUSTRIAL DR
BRIDGEVIEW, IL60455

TYPE AND SPECIFICATIONS

304 HRAP ASME SA 240, ASME CODE SECT. II, 1980 ED.
SUMMER 1982 ADD.

ITEM	DESCRIPTION	HEAT	TEST	YIELD STR. P.S.I 2% OFFSET	TENSIL STR. P.S.I.	% ELONG IN 2"	% RED OF AREA	BRINELL HARDNESS	BEND TEST	
1	.3750 X 31.7500 X 149.6250 SHEAR SQ & ACCURATE TOL WIDTH: + OR - 1/16 LENGTH + OR - 3/32	Q0006 - D7A	TT	42200	89100	60	61	157		A
2	.3750 X 36.0000 X 168.5000 SHEAR SQ & ACCURATE TOL WIDTH: + OR - 1/16 LENGTH + OR - 3/32	Q0006 - D7A	TT	42200	89100	60	61	157		A

'83 APR -6 A11:12

RE[illegible]
FERM[illegible]
CONTRACTS

**** LADLE ANALYSIS ****

ITEM	QTY	HEAT		C	MN	P	S	SI	CR	NI	N
1	1	Q0006	D7A	.061	1.34	.024	.004	.49	18.32	8.38	.033
2	2	Q0006	D7A	.061	1.34	.024	.004	.49	18.32	8.38	.033

R. A. Pugh Cert Analyst 2/24/83

Sworn to and subscribed before me this ________ day of ________ I hereby certify the above figures are correct as contained in the records of the corporation

INDUSTRIAL
METAL FABRICATING
STEEL - ALUMINUM
S LESS - OTHER ALLOYS

ALL METAL MANUFACTURING COMPANY

9925 SOUTH INDUSTRIAL DRIVE
BRIDGEVIEW, ILLINOIS 60455

CHICAGO PHONE A/C (312) 582-0400
LOCAL PHONE A/C (312) 599-7600

March 17, 1983

'83 APR -6 A11:12

Fermi National Accelerator
Laboratory
P.O. Box 500
Batavia, Illinois 60510

REC
FERM
CONT

Attention: Mr. R. A. Lauer

Gentlemen:

Per your request following is a list of welding rod and heat numbers used to fabricate the items on your P.O. #91936 dated 2-7-83.

The (2) L He Dewar-vacuum shells, dwg. 2771-MC-156258 Rev. None were given a root pass using 1/16 dia. 308L "Airco" rod, heat #J9147 and finish welded with 1/8 dia. 308L "McKay" D.C.T. rod, heat #43511.

The (2) L He Dewar-inner Dewar shells, dwg. 2771-MC-156259 Rev. None were given a root pass using 1/16 dia. 316L "Alloy Rods" heat #X33003 and finish welded with Kryo-Kay 316L welding rod which was furnished by Fermi-Lab.

Also attached is a material certification sheet covering the plate material used to fabricate the above items.

Very truly yours,

ALL METAL MANUFACTURING COMPANY

Joseph Plachy

JP/hds

Enc:

Fermilab

WELDER QUALIFICATION TEST RECORD

Welder's Name CHAMDER SOOD, #3800 Ident. No. 11 Date March 11, 198

Welding Process(es) GTAW Type MANUAL

Test in Accordance with WPS No. 155001

Material Spec. Spec/Grade No. SA 213 304 to Spec/Grade SA 213 304

P No. 8 to P No. 8 Thick. .277 Dia. 6"

Filler Metal Spec. No. SFA 5.9 Class. No. ER 308 F No. 6

Backing No

Position 6-G Weld Progression Upward

Gas Type Argon Composition

Electrical Characteristics: Current DC Polarity Straight

Other Thickness Range Qualified: 0.062 - 0.554

FOR INFORMATION ONLY

Filler Metal Diameter and Trade Name

Submerged Arc Flux Trade Name

Gas Metal Arc Welding Shield Gas Trade Name

GUIDED BEND TEST RESULTS

Specimen No.	Type	Figure No.	Results
1	Face	QW 462.3a	Acceptable
2	Root	QW 462.3a	Acceptable
3	Face	QW 462.3a	Acceptable
4	Root	QW 462.3a	Acceptable

RADIOGRAPHIC TEST RESULTS
(FOR ALTERNATIVE QUALIFICATION BY RADIOGRAPHY)

Radiographic Results n/a

Test Conducted by IFR Engineering Test No. 008001-11

We certify that the statements in this record are correct and that the test welds were prepared, welded and tested in accordace with the requirements of Section IX of the ASME Code.

By: [signature]

3/11/83

Fermilab

WELDER QUALIFICATION TEST RECORD

Welder's Name L. Nelson Ident. No. 47 Date 9-16-82

Welding Process(es) GTAW Type Manual

Test in Accordance with WPS No. 155001

Material Spec. Spec/Grade No. SA 213 304 to Spec/Grade SA 213 304

P No. 8 to P No. 8 Thick. .277 Dia. 6"

Filler Metal Spec. No. SFA 5.9 Class. No. ER 308 F No. 6

Backing No

Position 6G Weld Progression Upward

Gas Type Argon Composition

Electrical Characteristics: Current DC Polarity Straight

Other Thickness Range Qualified: 0.062 - 0.554

FOR INFORMATION ONLY

Filler Metal Diameter and Trade Name 1/16; 3/32 Sandvick

Submerged Arc Flux Trade Name n/a

Gas Metal Arc Welding Shield Gas Trade Name n/a

GUIDED BEND TEST RESULTS

Specimen No.	Type	Figure No.	Results
1	Face	QW 462.3a	Acceptable
2	Root	QW 462.3a	Acceptable
3	Face	QW 462.3a	Acceptable
4	Root	QW 462.3a	Acceptable

RADIOGRAPHIC TEST RESULTS
(FOR ALTERNATIVE QUALIFICATION BY RADIOGRAPHY)

Radiographic Results n/a

Test Conducted by IFR Engineering Test No. 47445

We certify that the statements in this record are correct and that the test welds were prepared, welded and tested in accordace with the requirements of Section IX of the ASME Code.

By: James Forster

Date: 9/29/82

QW-482 WELDING PROCEDURE SPECIFICATION (WPS)

(See QW-201.1, Section IX, 1974 ASME Boiler and Pressure Vessel Code)

Company Name: FERMILAB

Welding Procedure Specification No.: FORMERLY WP900 ES-155001 Date: 10/76 Supporting PQR No.: 900-1-A

Revisions:

Welding Process(es): GTAW Type(s): MANUAL

JOINTS (QW-402)
Groove Design: V groove $37\frac{1}{2}^{0}$
Backing: Yes ____ No X
Backing Material (Type): not required
Other: ------

BASE METALS (QW-403)
P No. 8 to P. No. 8
Thickness Range: .062 to .56
Pipe Dia. Range: ------
Other: ------

FILLER METALS (QW-404)
F No. 6 Other
A No. 8 Other
Spec No. (SFA): 5.9
AWS No. (Class): ER-308L
Size of Electrode: 3/32
Size of Filler: 3/32 - 1/16
Electrode-Flux (Class): N/A
Consumable Insert: N/A
Other: AWS A5.12 EWTH2

POSITION (QW-405)
Position of Groove: 6G
Welding Progression: Bottom to top
Other:

PREHEAT (QW-406)
Preheat Temp.: 60^{0}F min.
Interpass Temp.: 350^{0}F max.
Preheat Maintenance: 60^{0}F min.
Other: ------

POSTWELD HEAT TREATMENT (QW-407)
Temperature: none required
Time Range: ------
Other: ------

GAS (QW-408)
Shielding Gas(es): Argon
Percent Composition (mixtures): 99.9% welding grade
Flow Rate: 15-25 CFH
Gas Backing: Argon Inert Gas
Trailing Shielding Gas Composition: ------
Other: ------

ELECTRICAL CHARACTERISTICS (QW-409)
Current AC or DC: DC Polarity: Straight
Amps (Range): 80 Amps - 100 Volts (Range): 8 - 12
Other: ------

TECHNIQUE (QW-410)
String or Weave Bead: Stringer
Orifice or Gas Cup Size: 5/16 to 1/2
Initial & Interpass Cleaning (Brushing, Grinding, etc): Grinding and/or Brushing
Method of Back Gouging: ------
Oscillation: Minimal*
Contact Tube to Work Distance: ------
Multiple or Single Pass (per side): Multiple
Multiple or Single Electrodes: Single
Travel Speed (Range): 2 to 4 inches per minute
Other: ------

*Bead width should not appreciably exceed gas cup orifice diameter.

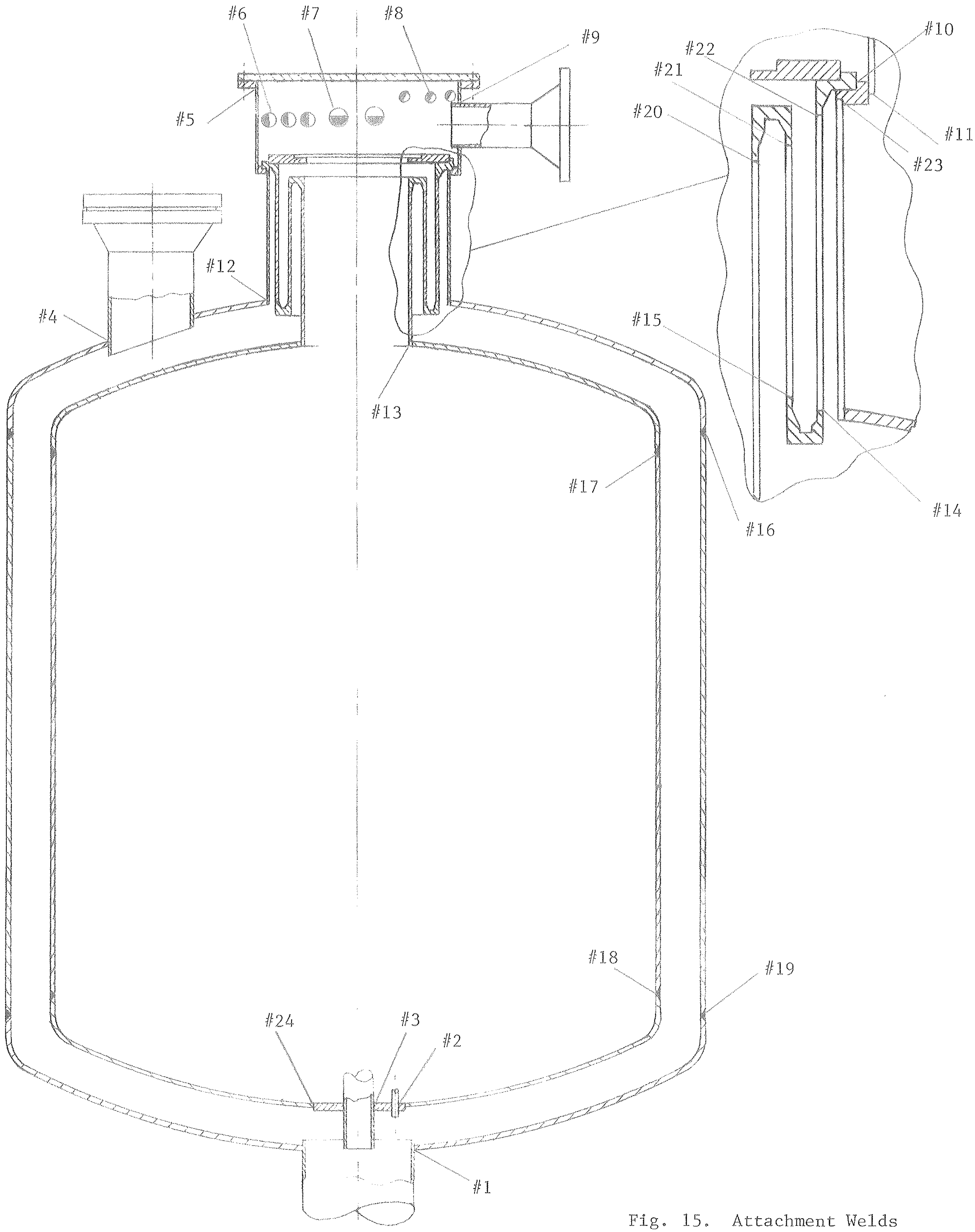


Fig. 15. Attachment Welds

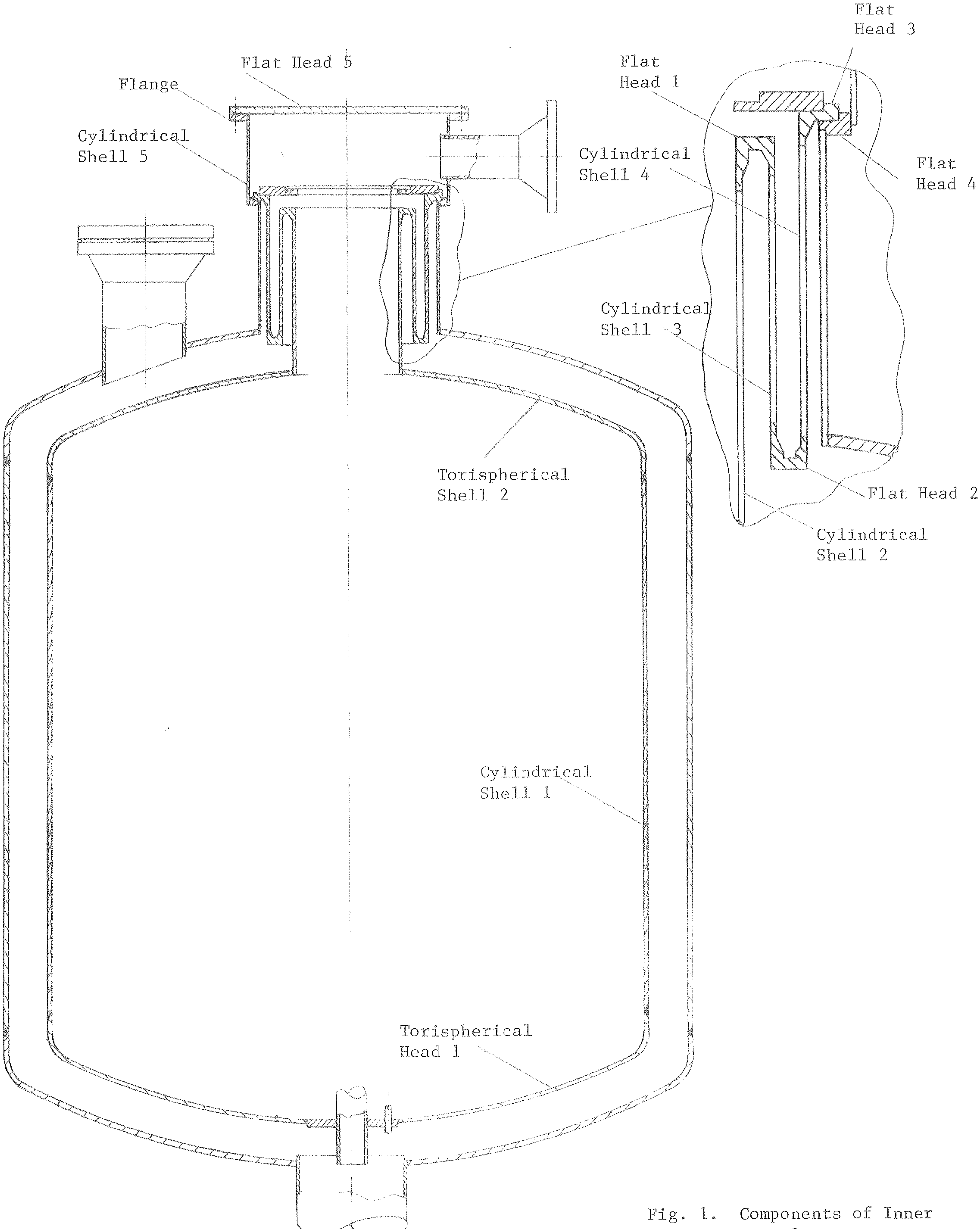


Fig. 1. Components of Inner Vessel

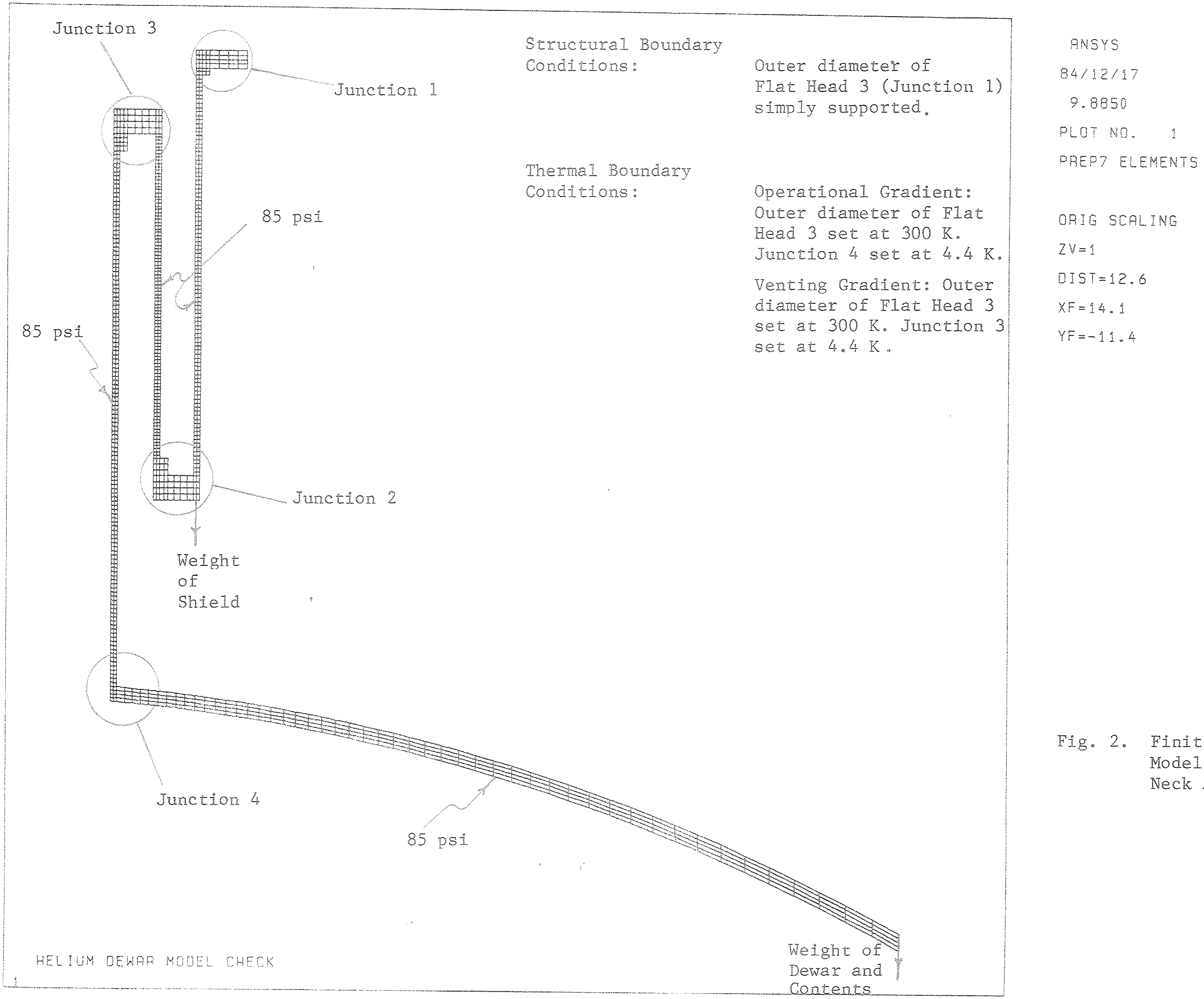
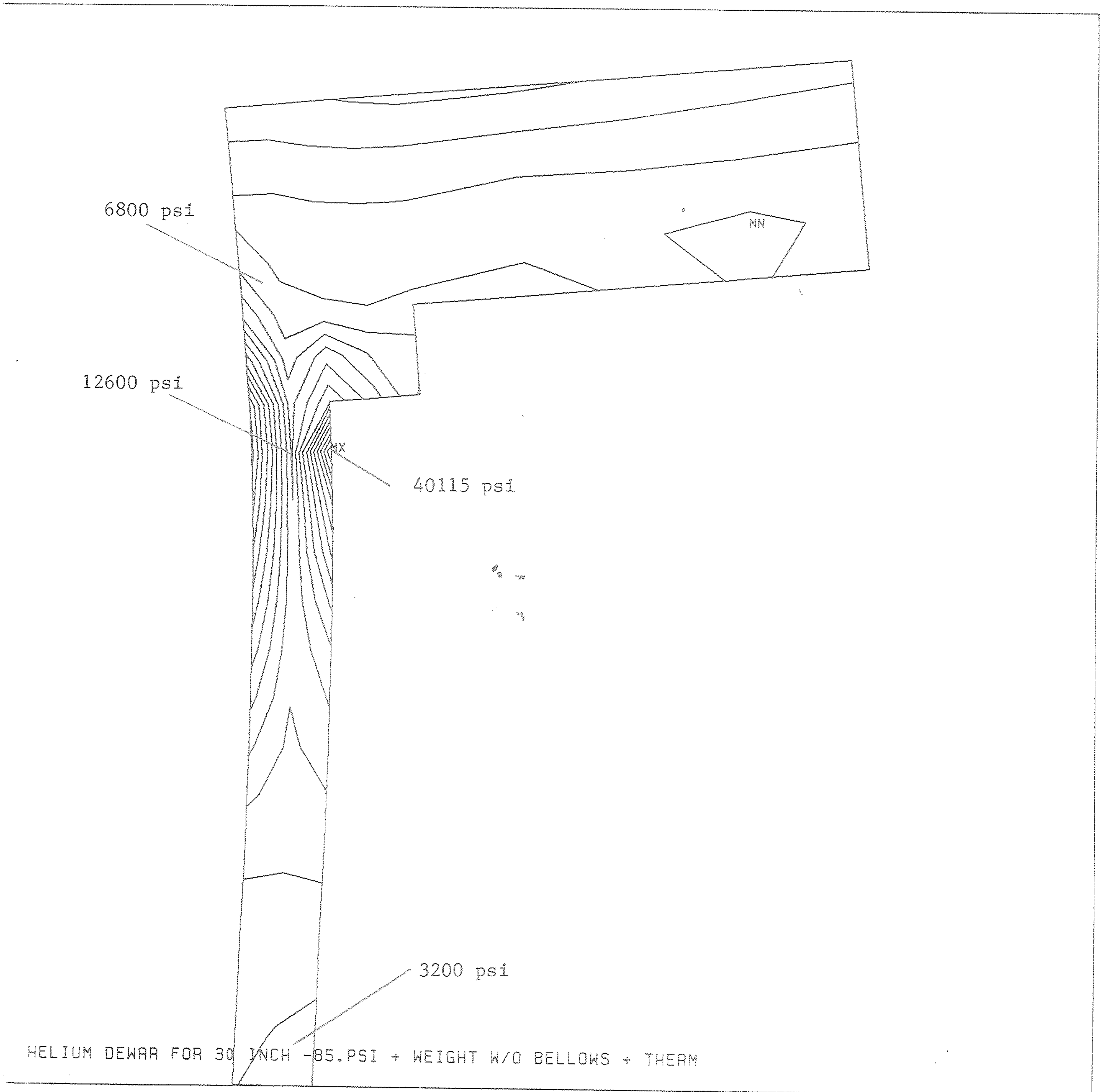


Fig. 2. Finite Element Model of He Dewar Neck Assembly



ANSYS
84/12/17
20.8606
PLOT NO. 1
POST1
STEP=1
ITER=1
STRESS PLOT
SI

ORIG SCALING
ZV=1
DIST=1.14
XF=6.85
YF=-1.03
EDGE
DMAX=.00762
DSCA=14.9
MX=40115
MN=1757
INC=2500

Fig. 3. Stress Intensity in Junction 1 of He Dewar Neck Assembly (Venting Thermal Gradient)

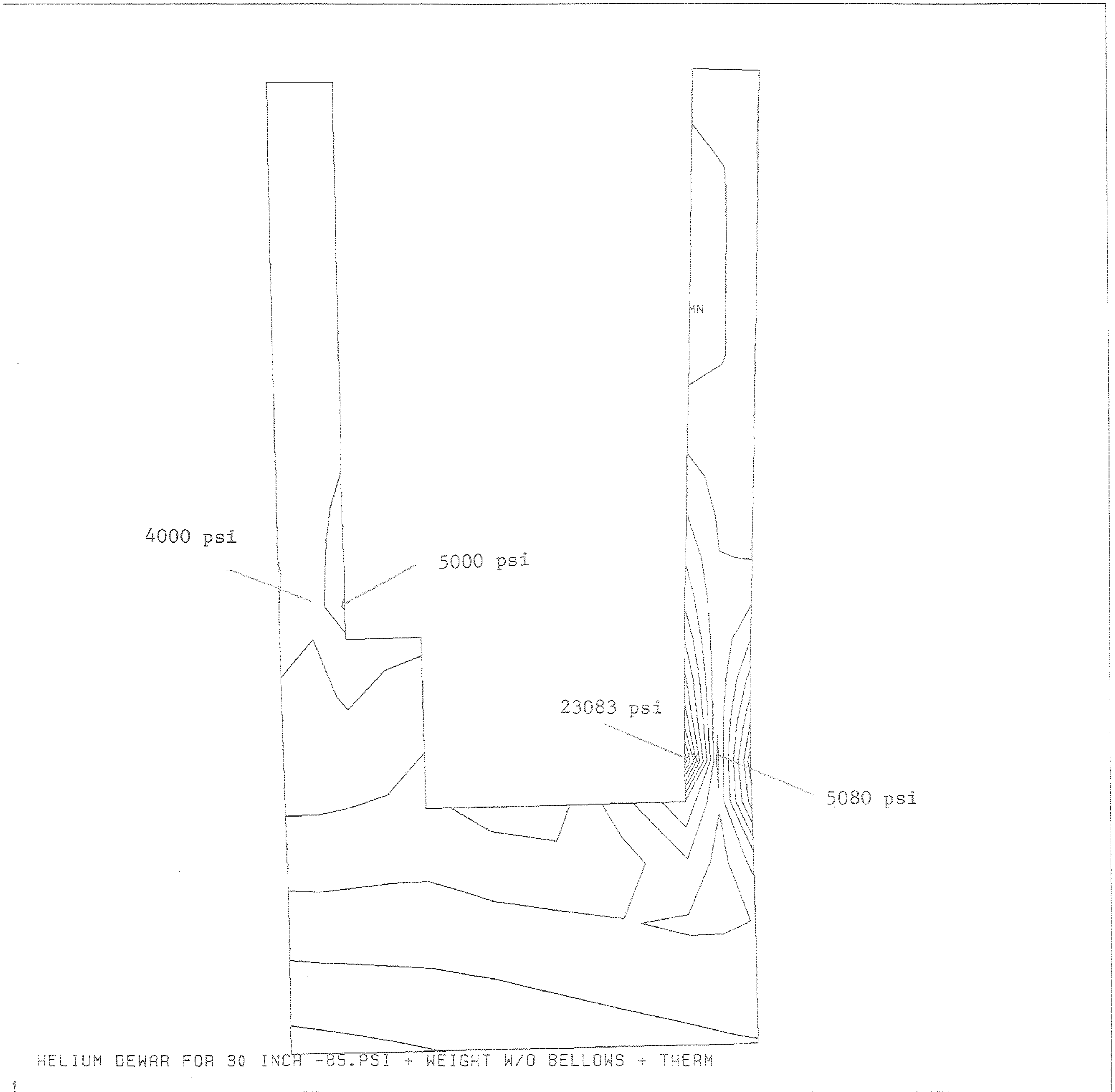


Fig. 4. Stress Intensity in Junction 2 of He Dewar Neck Assembly (Venting Thermal Gradient)

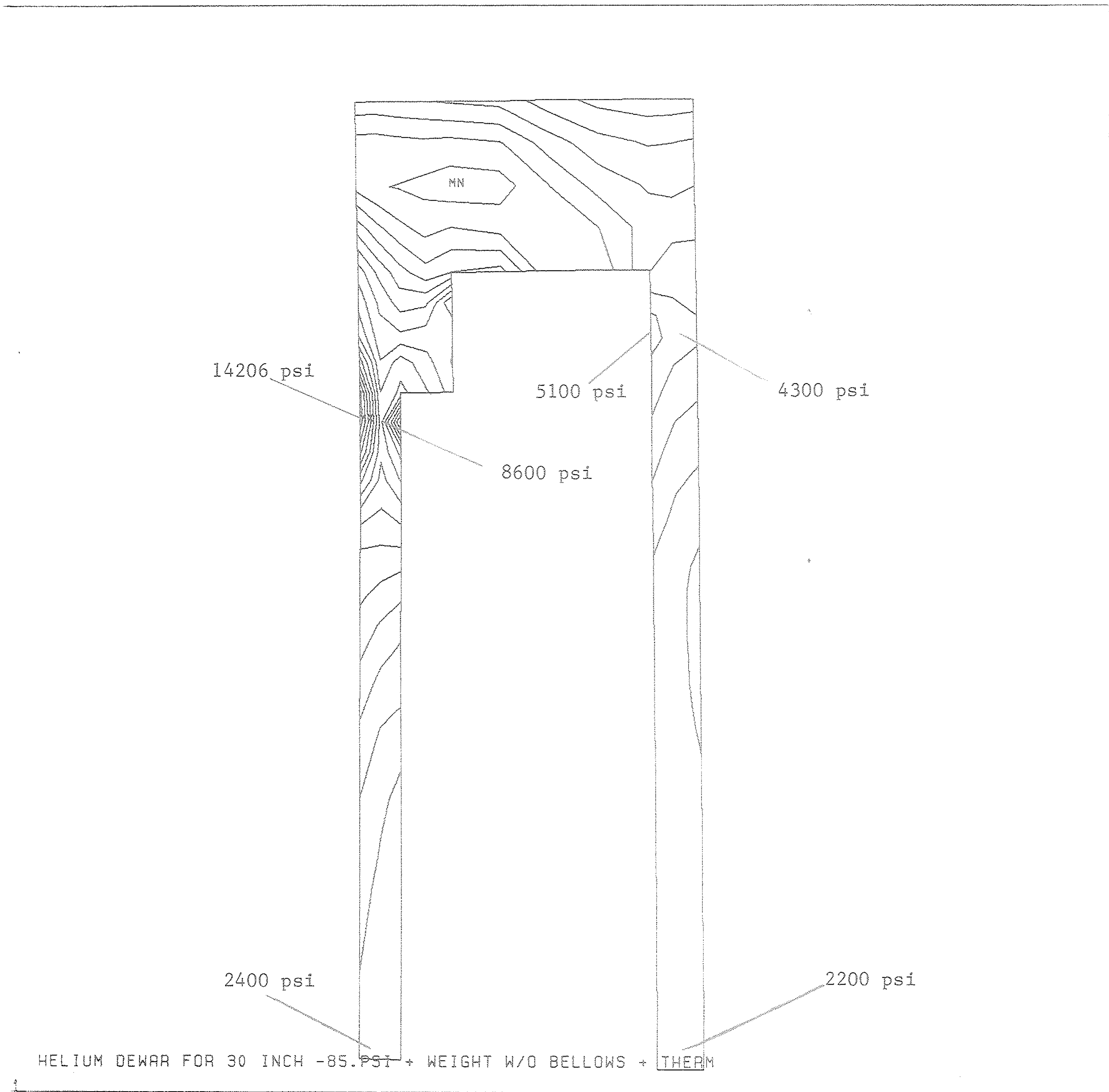


Fig. 5. Stress Intensity in Junction 3 of He Dewar Neck Assembly (Venting Thermal Gradient)

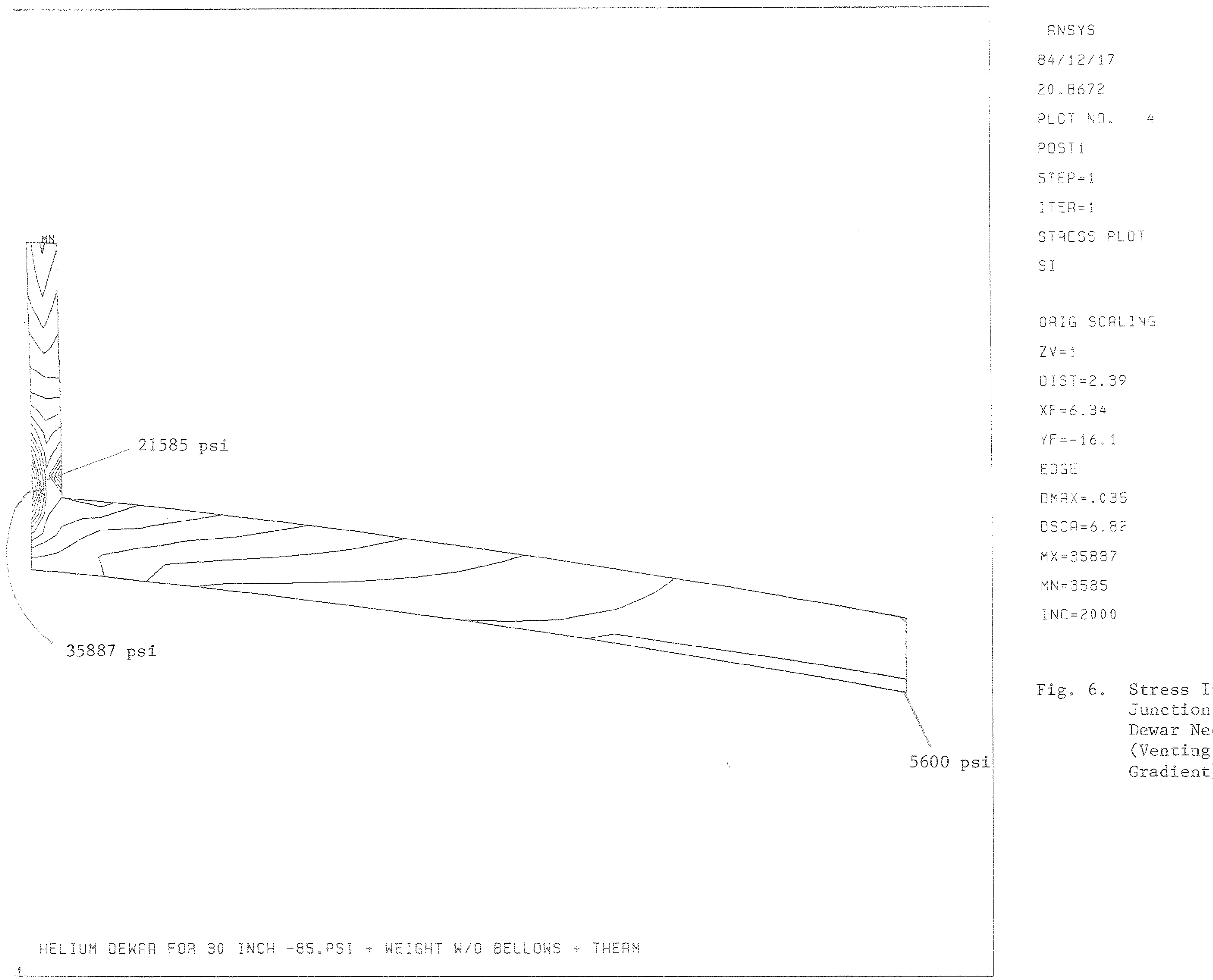


Fig. 6. Stress Intensity in Junction 4 of He Dewar Neck Assembly (Venting Thermal Gradient)

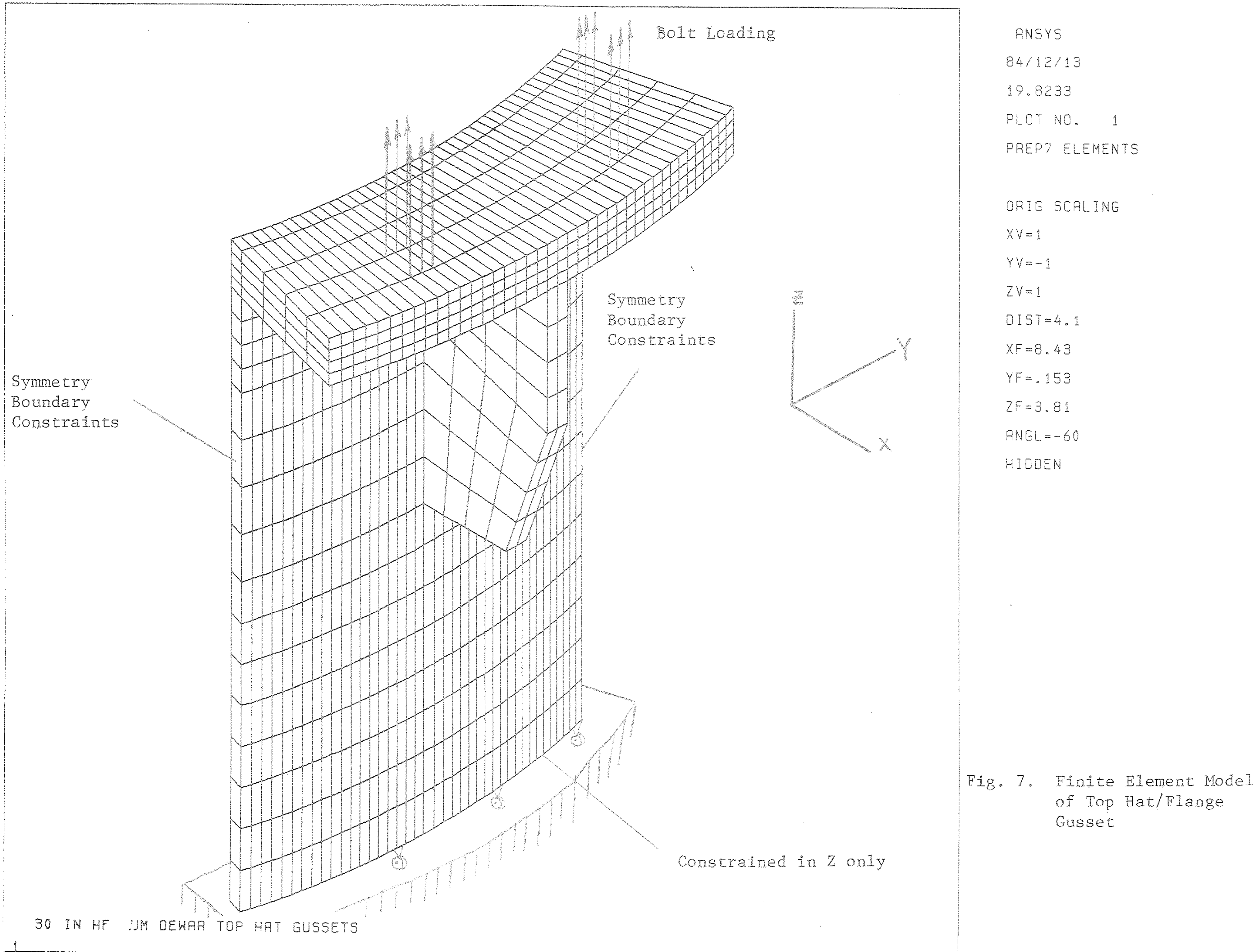


Fig. 7. Finite Element Model of Top Hat/Flange Gusset

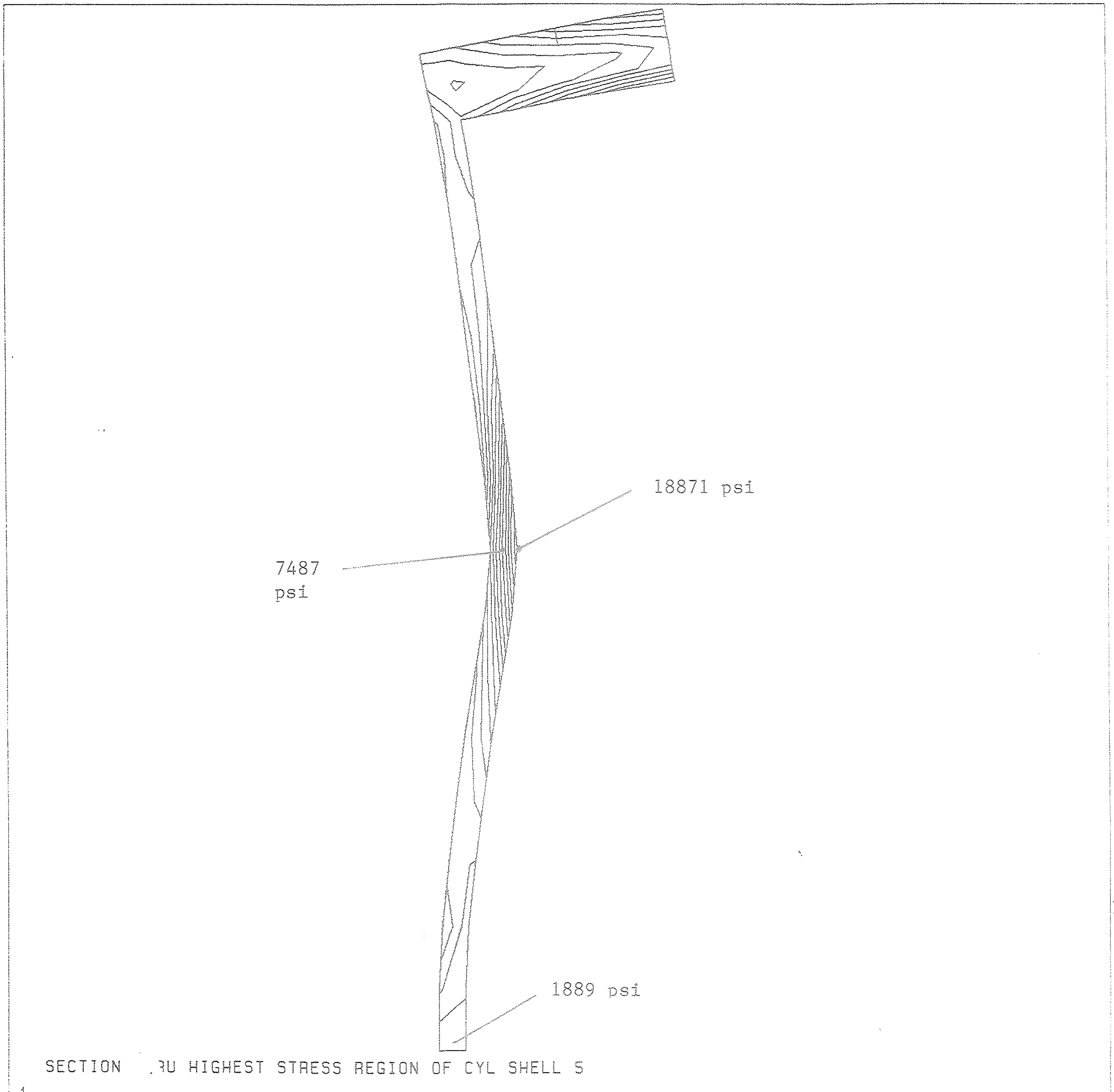


Fig. 8. Stress in Top Hat Shell due to .85 psid Internal Pressure

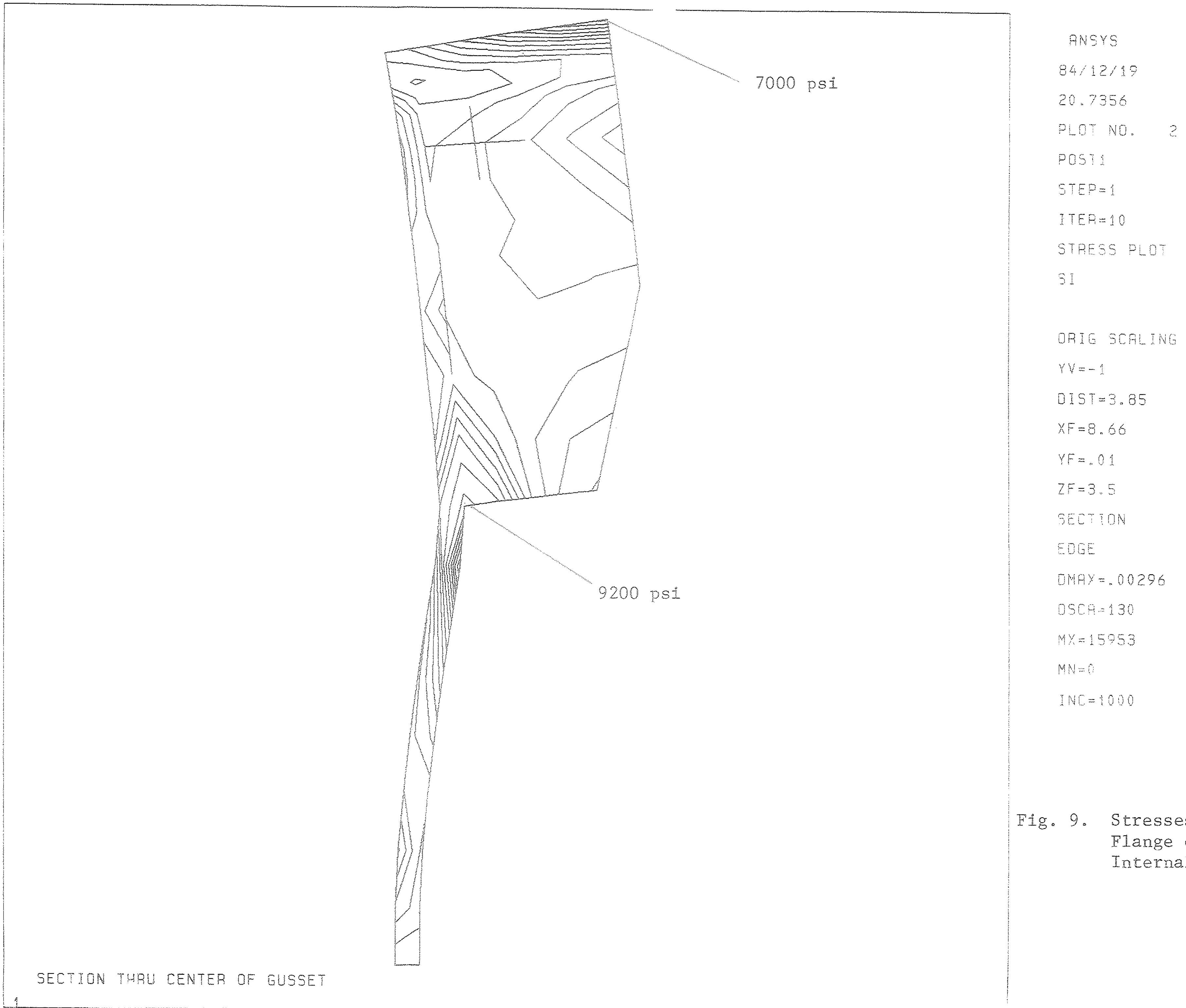


Fig. 9. Stresses in Gusset and Flange due to 85 psid Internal Pressure

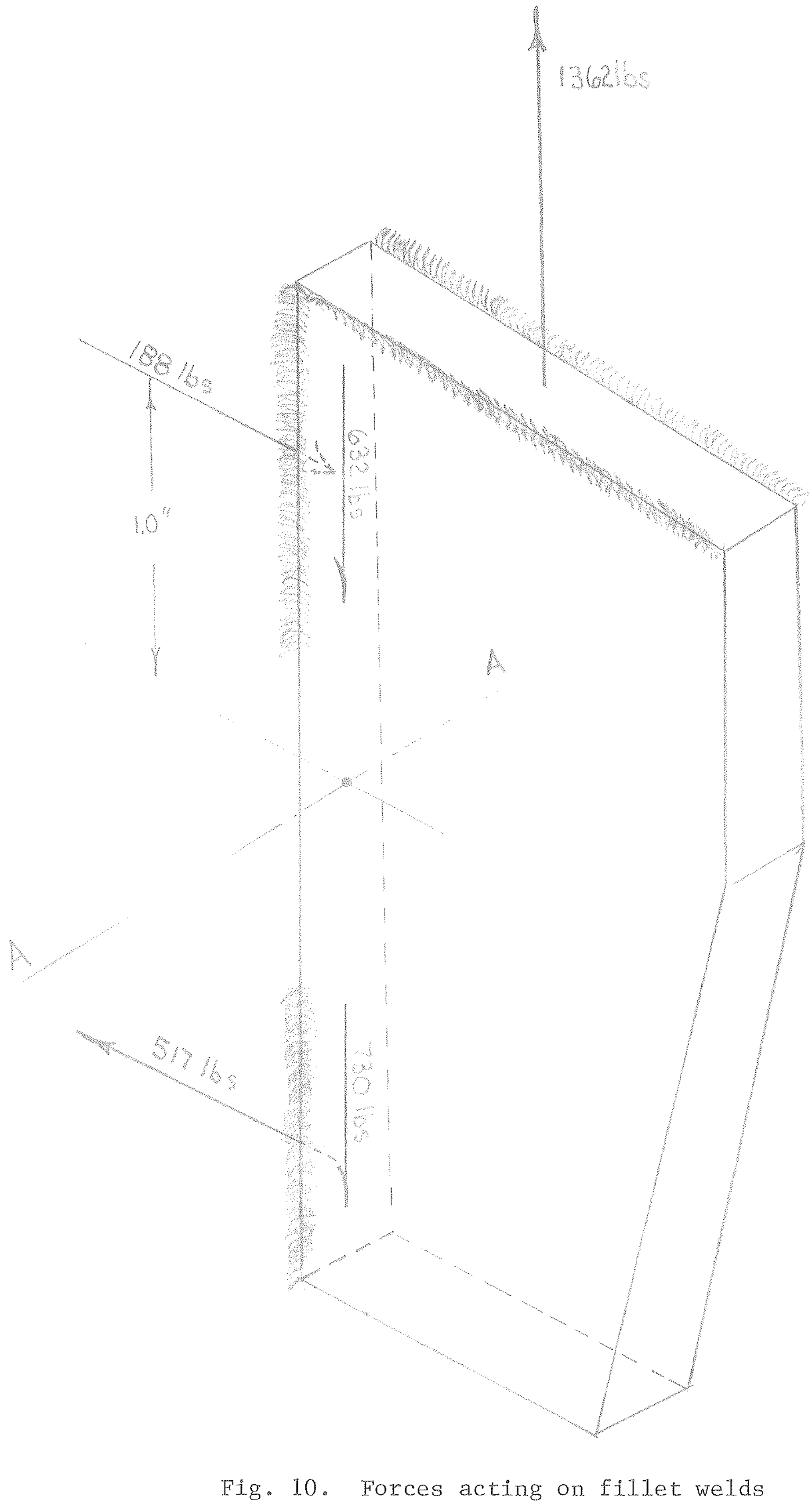


Fig. 10. Forces acting on fillet welds
in tophat/flange gussets

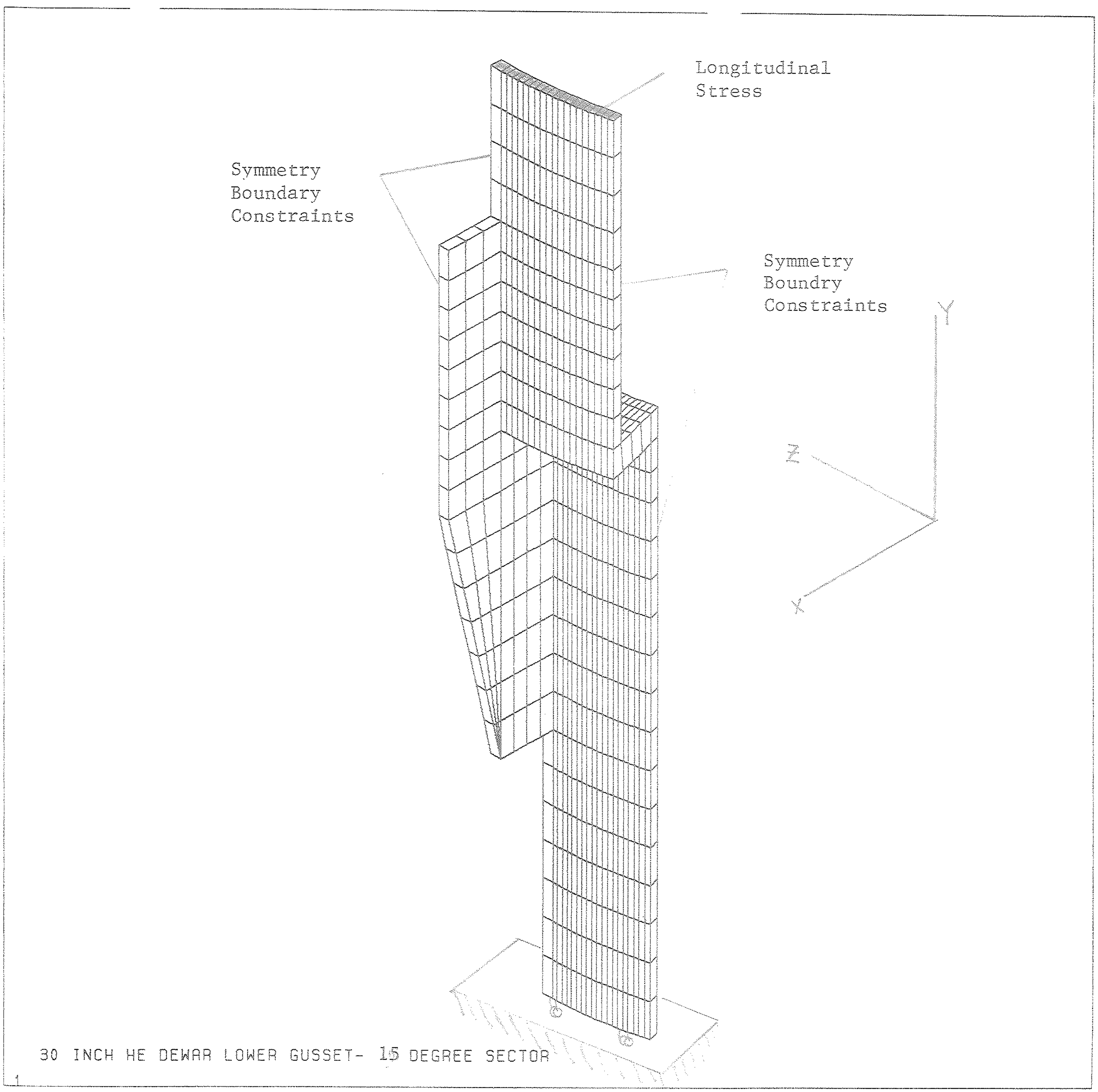


Fig. 11. Finite Element Model of Tophat/Vacuum Vessel Gusset

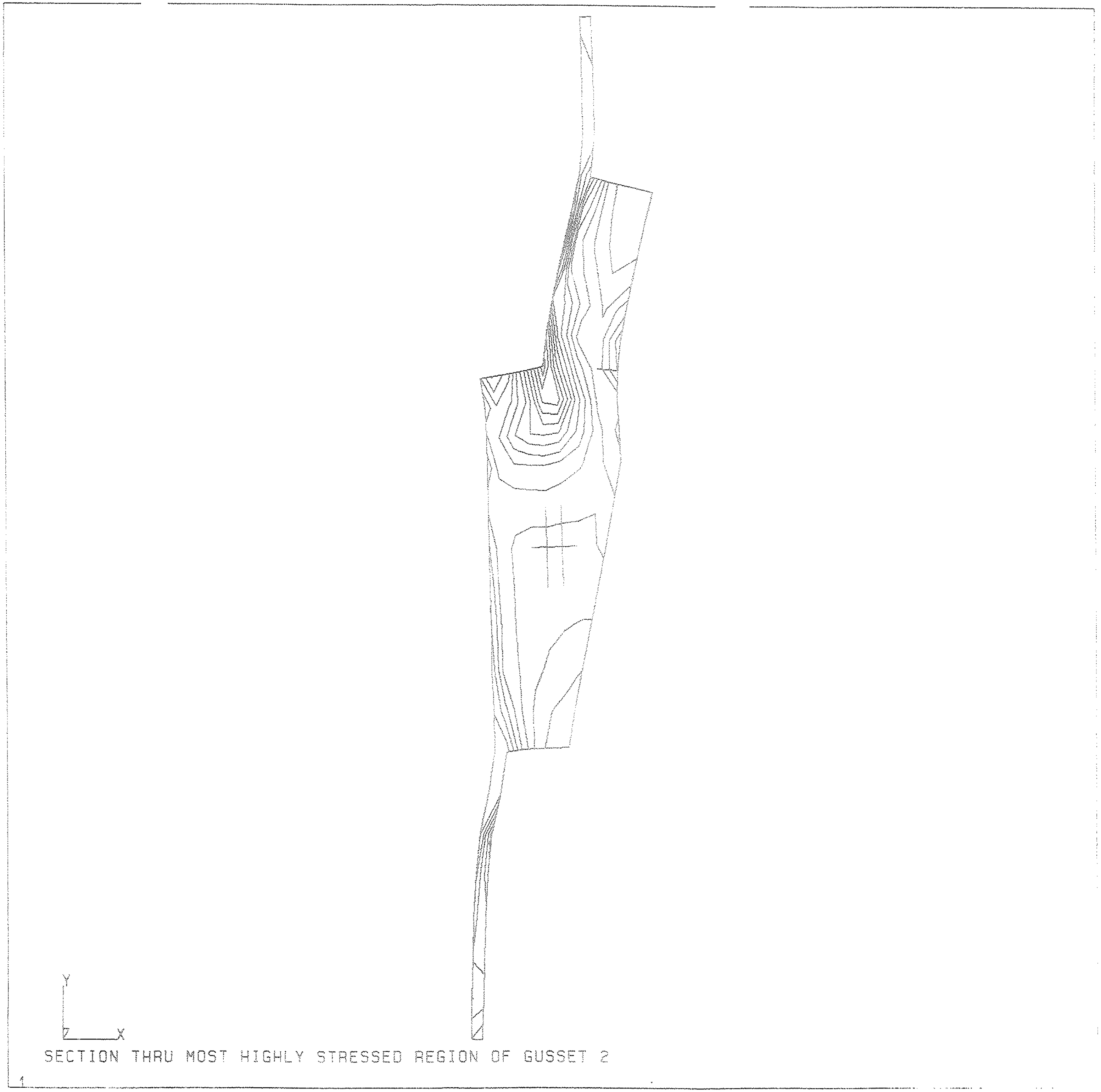


Fig. 12. Stress Intensity in Most Highly Stressed Region of Tophat/ Vacuum Shell Gusset

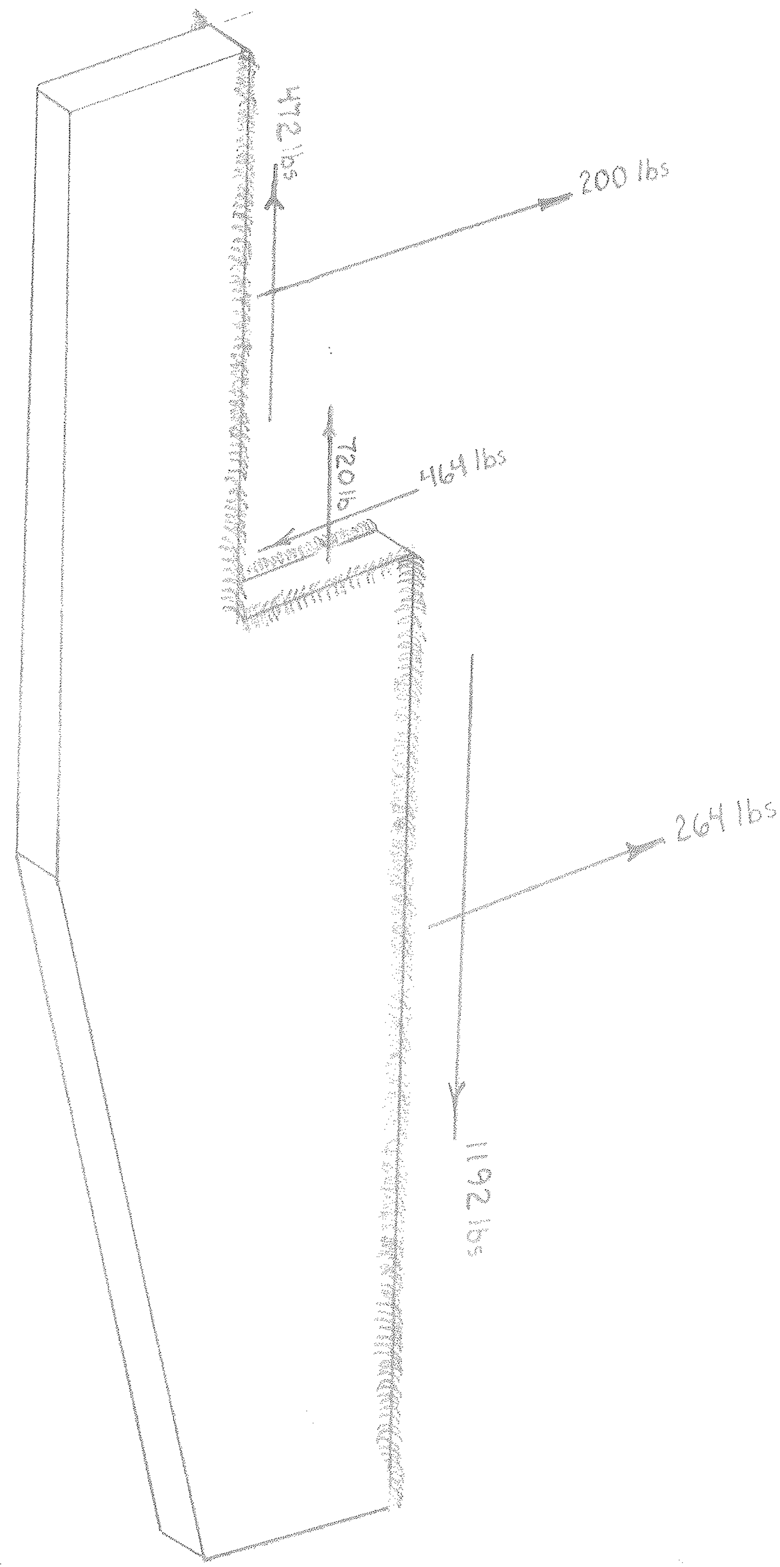


Fig. 13. Forces on Fillet Welds in Tophat/Vacuum Shell Gusset

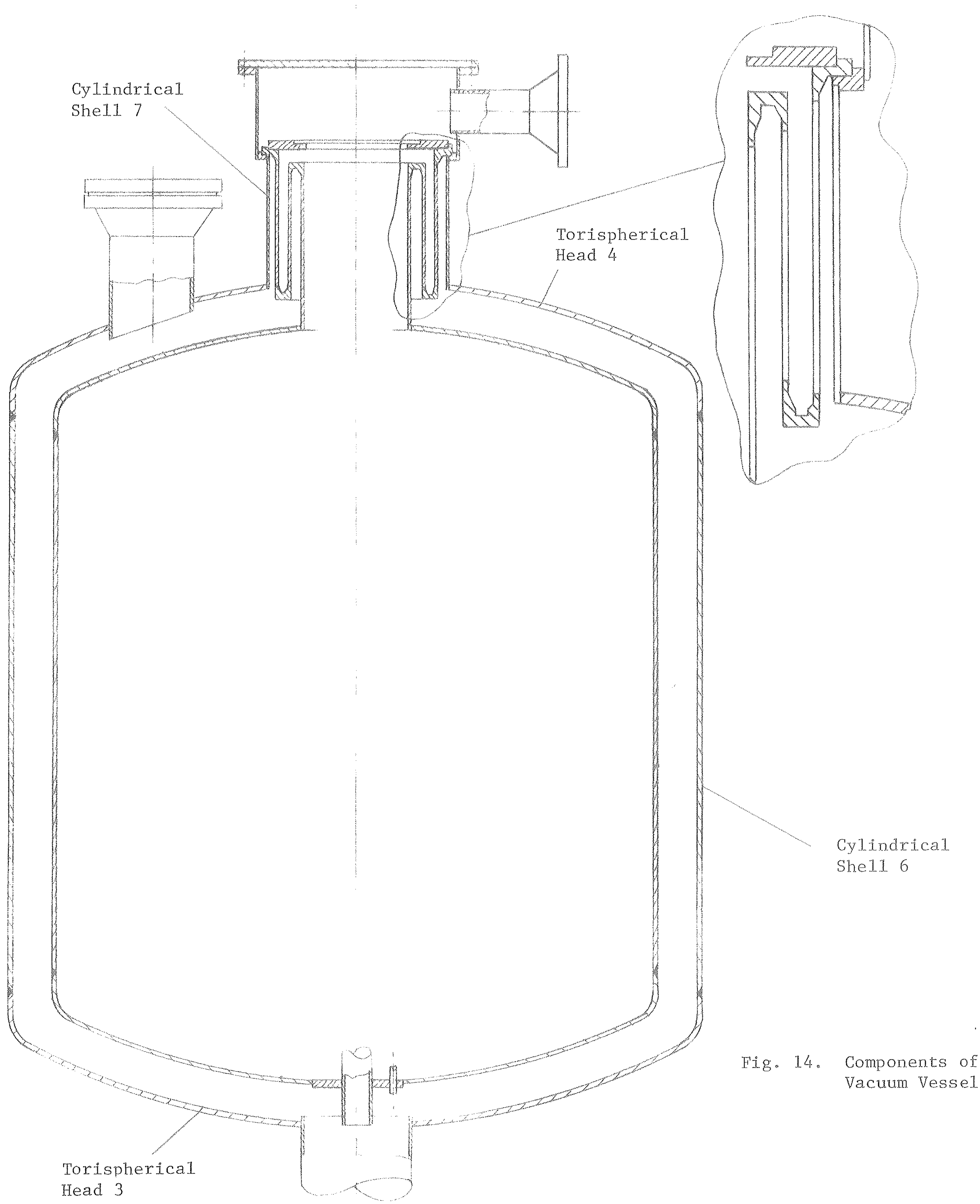


Fig. 14. Components of
Vacuum Vessel