TITLE: TOHOKU BUBBLE CHAMBER MAGNET LN_2 BUFFER DEWAR STRESS ANALYSIS

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Introduction

The LN_2 buffer 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 fullfil 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, modified as required by 14.1 for in-house vessels.

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

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

III. Calculation of Weld Strength, Opening Reinforcement and Support Effects.

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

IV. System Venting Verification.

A schematic and calculations are presented to show adequate system reliefs.

V. Pressure Test Results.

VI. Exceptional Vessel Details.

Calculations and discussion of those aspects of the LN_2 buffer dewars which do not meet 14.1 requirements.

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

The dimensions and allowable stresses for dewar components are given in Table I. Figure 1 shows the component locations.

A. Internal MAWP of Inner Vessel

1. Cylindrical Shell 1

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

R = 10 in
t = 0.188 in
E = 1 for circumferential stress calculation
E = 0.60 single welded joint without backing strip for longitudinal stress calculation
S = 12800 x 0.8 as required by UW-12 of Div. 1 for non-radiographed vessels, circumferential stress calculation
S = 12800 for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R + 0.6\ t} = 190 \text{ psid} \qquad \text{circumferential stress}$$

$$P = \frac{2\ S\ E\ t}{R - 0.4\ t} = 291 \text{ psid} \qquad \text{longitudinal stress}$$

Therefore, the MAWP (internal) of the shell is 190 psid.

2. Torispherical Heads 1 and 2

$t_{min} = t_{nom} - t_{tol} = 0.188 - 0.032 = 0.156$

S = 15000 x 0.8 = 12000
(The 0.8 factor is required by UW-12 for nonradiographed vessels)

E = 1.0 (seamless heads)

L = 20" inside spherical radius

Then,

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

$P = 106$ psid

3. Cylindrical Shell 2

This shell is under pressure and weight loading. The weight loading consists of: 2 heads x 30 lbs each; 1 shell x 20 lbs; and 67 liters of LN_2 = 120 lbs. This weight produces a longitudinal membrane stress through the shell thickness of

$$\sigma_W = \frac{60 + 20 + 120}{\pi \times 2.5 \times 0.035}$$

$$\sigma_W = 720 \text{ psi}$$

The junction of the shell with torispherical head 2 experiences bending stress due to the weight and pressure loading, as shown in Fig. 2(a). The force resulting from pressure can be calculated assuming an internal pressure of 60 psi. Then

$$F_p = 60 \times \pi \times 1.25^2$$

$$F_p = 294 \text{ lbs}$$

The total force which produces bending at the junction is then 60 + 20 + 120 + 294 = 494 lbs.

To calculate the bending stresses in the shell resulting from this loading, the structure is modeled as shown in Fig. 2(b). The displacements θ_1 and θ_2 can be expressed as a function of the moments M_1 and M_2 and force P through the use of formulas from Ref. 1.

For the shell, relevant constants are:

$$\lambda = [3(1-\nu^2)/(R^2t^2)]^{0.25} = 0.94$$

$$D = Et^3/12(1-\nu^2) = 118$$

For the head, relevant constants are:

$$D = Et^3/12(1-\nu^2) = 18250$$

$$b/a = 1.25/10 = 0.125$$

$$P = \frac{494}{2\ \pi\ (10)} = 7.86 \text{ lb/in}$$

The angular deflection θ_1 can be expressed in terms of M_1 using Table 30, case 10 of Ref. 1. $\lambda L > 6$ so we have a long cylinder.

1) $\theta_1 = \frac{M_1}{D\lambda} = 0.009\ M_1$

The moment M_1 is the only loading which affects θ_1.

The angular deflection θ_2 is a function of M_2 and P. The superposition of cases 5c and 1k from Table 24 of Ref 1 can be used, as indicated in Fig. 2. From case 5c

$$R_1 = \frac{K_{Q_b} M_2}{a}$$

$$R_1 = \frac{1.0937\ M_2}{10}$$

$$R_1 = 0.10937\ M_2$$

$$(\theta_2)_{M_2} = \frac{K_{\theta_b} M_2\ a}{D}$$

$$(\theta_2)_{M_2} = \frac{0.1142\ (10)\ M_2}{18250} = 6.257\ (10^{-5})\ M_2$$

From case 1k,

$$(\theta_2)_{R_1+P} = \frac{(R_1 + P)\ a^2}{DC_7}\ (aC_9/b - L_9)$$

with $C_7 = 3.583$

$C_9 = 0.190$

$L_9 = 0$

$$(\theta_2)_{R_1+P} = 2.324\ (10^{-3})(R_1 + P)$$

Then, substituting for R_1 and adding the contribution to θ_2 from M_2,

2) $\theta_2 = 3.1675\ (10^{-4})\ M_2 + 0.01827$

Equations 1 and 2 can be solved since $\theta_1 = \theta_2$ (compatability) and $M_1 = -M_2$ (equilibrium). The result is

$$M_1 = 1.96\ \frac{\text{in·lbs}}{\text{in}}$$

The bending stress in the shell is

$$\sigma_{bend} = \frac{6M_1}{t^2} = 9600 \text{ psi}$$

This value of bending stress is a worse case since the angular deflection is overestimated. The angular deflection is actually smaller because the head is not a flat plate.

This bending moment also creates membrane and hoop stresses which are a maximum at the neck/head junction.

$$\sigma_{\theta memb} = \frac{2\ M_1\ \lambda^2\ R}{t} = 124 \text{ psi}$$

$$\sigma_{\theta bend} = \nu\ \sigma_{axial\ bending} = 2880 \text{ psi}$$

Thus from dead weight and pressure induced bending stresses, the following are obtained

$$\sigma_{\theta max} = 124 + 2880 = 3004$$

$$\sigma_{axial\ max} = 9600 + 720 = 10360 \text{ psi}$$

Use the standard formulas for obtaining the pressure rating of the inner neck.

For circumferential stress,

$S = 12800 \times 0.8 = 10240$ as required by UW-12 of Div 1

$E = 1$

$R = 1.25$

$t = 0.035$

Then,

$$P = \frac{S\ E\ t}{R + 0.6t} = \frac{(10240 - 3004) \times 1 \times 0.035}{1.25 + 0.6 \times 0.035} = 199 \text{ psid}$$

For longitudinal stress,

$S = 12800$

$E = 0.55$ for a double full fillet lap joint

$$P = \frac{2\,S\,E\,t}{R-0.4t} = \frac{2(12800 - 10380) \times 0.55 \times 0.035}{1.25 - 0.4 \times 0.035} = 75.4 \text{ psid}$$

Thus the neck is good for greater than 60 psid as the head junction up to this point in the analysis. The top of the neck has a much smaller bending moment so that this analysis is valid for the entire length.

Conclusion

The maximum allowable working pressure (internal) of the inner vessel based on shell and head thickness is 60 psid. Flat heads 1 and 2 were not considered in establishing this MAWP. These heads are covered in Part VI as vessel exceptions.

B. External MAWP of Inner Vessel

1. Cylindrical Shell 1

The applicable procedure is found in UG-28 of Div. 1

$L = 6 + 2\ (1.5 + 1/3 \times 3.313) = 11.21$ in. UG-28(b)(1)

$t = 0.188$ in.

$D_o = 20$ in.

$L/D_o = 0.561$

$D_o/t = 106 > 10$ UG-28(c)(1)

$A = 0.0019$ (Fig. 5-UGO-28.0)

$B = 11500$ (Fig. 5-UHA-28.1)

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

$P = 145$ psid

2. Torispherical Heads 1 and 2

The applicable procedure is found in UG-33 of Div. 1. This procedure requires calculating by two methods, then choosing the more conservative result.

From UG-33(a)(1),

Using UG-32(e) with

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

$E = 1$

$L = 20$ in

$t = t_{nom} - t_{tol} = 0.156$ in.

then

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

$P = 63$ psid

From UG-33(a)(2) with

$R_o = 20$ in

$t = 0.156$ in.

$A = 0.125/(R_o/t) = 9.8 \times 10^{-4}$ (Fig. 5-UHA-28.1)

$B = 9000$ (Fig. 5-UHA-28.1)

Then,

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

$P = 70.2$ psid

The external MAWP of the torispherical heads is 63 psid.

3. Cylindrical Shell 2

Using the same procedure as for cylindrical shell 1,

$L = 14$ in	$L/D_o = 5.6$
$t = 0.035$ in	$D_o/t = 71$
$D_o = 2.5$ in	$A = 0.0004$
	$B = 5500$

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

P = 103 psid

Conclusion

The maximum allowable working pressure (external) of the inner vessel based on shell and head thickness is 63 psid. Flat heads 1 and 2 were not considered in establishing this MAWP. These heads are covered in Part VI as vessel exceptions.

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

A. Internal MAWP of Vacuum Vessel

1. Cylindrical Shell 3

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

R = 12 in

t = 0.25 in

E = 1 for circumferential stress calculation

E = 0.60 for single welded joint without backing strip for longitudinal stress calculation

S = 12800 x 0.8 as required by UW-12, circumferential stress calculation

S = 12800 for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R+0.6t} = 211 \text{ psid (circumferential stress)}$$

$$P = \frac{2\ S\ E\ t}{R-0.4t} = 323 \text{ psid (longitudinal stress)}$$

2. Torispherical Heads 3 and 4

The applicable formula is given in UG-32 of Div. 1

$t_{min} = t_{nom} - t_{tol} = 0.25 - 0.032 = 0.218$ in

$L = 24$ in

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

$E = 1$ (seamless heads)

then,

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

$P = 123$ psid

3. Cylindrical Shell 4

From UG-27, with

$R = 2$ in

$t = 0.120$ in.

$E = 1$ for circumferential stress calculation

$E = 0.55$ for double fillet attachment weld at torispherical head 4, longitudinal stress calculation

$S = 12800 \times 0.80$ as required by UW-12, circumferential stress calculation

$S = 12800$ psi for longitudinal stress calculation

Then,

$$P = \frac{S\ E\ t}{R + 0.6t} = 593 \text{ psid}$$

$$P = \frac{2\ S\ E\ t}{R - 0.4t} = 865 \text{ psid}$$

Conclusion

The maximum allowable working pressure (internal) of the vacuum vessel based on shell and head thickness is 123 psid. Flat heads 1 and 2 were not considered in establishing this MAWP. These heads are covered in Part VI as vessel exceptions.

B. External MAWP of Vacuum Vessel

1. Cylindrical Shell 3

The applicable procedure is found in UG-28 of Div. 1

$L = 8.5 + 2\,(1.5 + 1/3 \times 4) = 14.17$ in. UG-28(b)(1)

$t = 0.25$ in.

$D_o = 24$ in.

$L/D_o = 0.59$

$D_o/t = 96 > 10$ UG-28(c)(1)

Factor A = 0.0025 (Fig. 5-UGO-28.0)

Factor B = 12000 (Fig. 5-UHA-28.1)

Then,

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

P = 167 psid

2. Torispherical Heads 3 and 4

The applicable procedure is found in UG-33 of Div. 1. This procedure requires calculating by two methods, then choosing the more conservative result.

From UG-33(a)(1)

Using UG-32(e) with

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

$E = 1$

$L = 24$

$t = t_{nom} - t_{tol} = 0.218$ in

then,

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

P = 74 psid

From UG-33(a)(2)(e) with

R_o = 24 in

t = 0.218 in.

$A = 0.125/(R_o/t) = 0.0011$ (Fig. 5-UHA-28.1)

$B = 9200$ (Fig. 5-UHA-28.1)

then,

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

P = 83 psid

The external MAWP of the torispherical heads is 74 psid.

3. Cylindrical Shell 4

Using the same procedure as for cylindrical shell 4,

L	= 12 in	L/D_o	= 3
t	= 0.120 in	D_o/t	= 33.3
D_o	= 4 in	A	= 0.0025
		B	= 12000

Then,

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

P = 480 psid

Conclusion

The maximum allowable working pressure (external) of the vacuum vessel based on shell and head thickness is 74 psid. Flat heads 1 and 2 were not considered in establishing this MAWP. These heads are covered in Part VI as vessel exceptions.

III. Calculation of Weld Strength, Opening Reinforcement and Support Effects

A. Calculation of Weld Strength (see Fig. 1 for weld location)

There are no weld design details for the welds on this vessel (see Part VI for a discussion of vessel welding as relates to 14.1 requirements). All weld calculations presented here will deal only with the connections of pipes to shells as required by UW-15 of Div. 1. The full penetration groove welds by which the torispherical heads attach to the cylindrical shells do not require analysis since their efficiency appeared in the formulas for calculating shell size. See sections UG-41(b), UG-45(b), UW-15, and appendix L7 example 6.

1. Attachment Weld 1

Type: Double fillet [permitted by UW-16(c)]

Purpose: Attachment of cylindrical shell 2 to torispherical head 2

Dimensions: Assumed 0.0625 in. leg

Length of weld = 2.5 x $\pi/2$ x 2 = 7.85 in. (Note: length of weld considered is on one half of the opening as required by code)

Shear area = 0.0625 x 0.707 x 7.85 = 0.347 in^2

Shear force at 60 psid + dead weight = 60 x π x 2.5^2 x 0.25 = 295 lbs + 200 = 495

Strength of weld = strength of weakest component x shear efficiency x 0.8 (from UW-12) = 12800 x 0.49 x 0.8 = 5018 psi

Nominal shear stress in weld = shear force/shear area = 495/0.347 = 1427 psi

Since 1427 < 5018, the weld is adequately sized.

2. Attachment Weld 2

Type: Double fillet

Purpose: Attachment of cylindrical shell 4 to torispherical head 4

Dimensions: Assumed 0.0625 in. leg

Length of weld = 4 x $\pi/2$ x 2 = 12.5 in.

Shear area = 0.0625 x 0.707 x 12.5 = 0.552 in^2

Shear force = vacuum load + weight of inner vessel and contents = 15 x π x 4^2 x 0.25 + 200 = 388 lbs

Strength of weld = 12800 x 0.49 x 0.8 = 5018 psi

Nominal shear stress in weld = 388/0.552 = 706 psi

Since 706 < 5018, the weld is adequately sized.

3. Attachment Weld 3

Type: Double fillet

Purpose: Attachment of 1/2 in. dia. 304 stainless steel welded tube to torispherical head 1

Dimensions: Assumed 0.0625 in. leg

Length of weld = $1/2 \times \pi/2 \times 2 = 1.57$ in.

Shear area = $0.0625 \times 0.707 \times 1.57 = 0.0695$ in^2

Shear force at 60 psid = $60 \times \pi \times 0.5^2 \times 0.25 = 12$ lbs

Strength of weld = $12800 \times 0.49 \times 0.8 = 5018$ psi

Nominal shear stress in weld = 12/0.0695 = 172 psi

Since 172 < 5018, the weld is adequately sized.

4. Attachment Weld 4

This weld is a single fillet weld attaching a 3/4 in. dia. 304 stainless steel welded tube to torispherical head 1. This type of attachment is prohibited by UW-16(e). See Part VI for a discussion of this exception. The strengrth calculation will be performed to demonstrate adequate weld size.

Dimension: Assumed 0.0625 in. leg

Length of weld = $3/4 \times \pi/2 = 1.18$ in.

Shear area = $0.0625 \times 0.707 \times 1.18 = 0.052$ in^2

Shear force at 60 psid = $60 \times \pi \times 0.75^2 \times 0.25 = 26.5$ lbs

Strength of weld = $12800 \times 0.49 \times 0.8 = 5018$ psi

Nominal shear stress in weld = 26.5/0.052 = 510 psi

Since 510 < 5018, the weld is adequately sized.

5. Attachment Weld 5

This weld is a single fillet weld attaching a 1.5 in. dia. 304 stainless steel welded tube to torispherical head 3. This type of attachment is prohibited by UW-16(e). See Part V for a discussion of this exception. The strength calculation will be performed to demonstrate adequate weld size.

Dimension: Assumed 0.0625 in. leg

Length of weld = 1.5 x $\pi/2$ = 2.35 in.

Shear area = 0.0625 x 0.707 x 2.35 = 0.104 in^2

Shear force = 15 x π x 1.5^2 x 0.25 = 26.5 lbs

Strength of weld = 12800 x 0.49 x 0.8 = 5018 psi

Nominal shear stress in weld = 26.5/0.104 = 254 psi

Since 254 < 5018, the weld is adequately sized.

6. Attachment Weld 6

Type: Double fillet

Purpose: Attachment of 2 in. dia. 304 stainless steel welded tube to torispherical head 3

Dimensions: Assumed 0.0625 in. leg

Length of weld = 2 x $\pi/2$ x 2 = 6.25 in.

Shear area = 0.0625 x 0.707 x 6.25 = 0.275 in^2

Shear force = 15 x π x 2^2 x 0.25 = 47 lbs

Strength of weld = 12800 x 0.49 x 0.8 = 5018

Nominal shear stress in weld = 47/0.275 = 170 psi

Since 170 < 5018, the weld is adequately sized.

7. Attachment Weld 7

This weld is a single fillet weld attaching a 2" pipe size Fike rupture disc assebly to torispherical head 4. This type of attachment is prohibited by UW-16(e). See Part VI for a discussion of this exception. The strength calculation will be performed to demonstrate adequate weld size.

Dimenson: Assumed 0.0625 in. leg

Length of weld = 2.375 x $\pi/2$ = 3.73 in.

Shear area = 0.0625 x 0.707 x 3.73 = 0.165 in^2

Shear force = 15 x π x 2.375^2 x 0.25 = 66 lbs

Strength of weld = 12800 x 0.49 x 0.8 = 5018 psi

Nominal shear stress in weld = 66/0.165 = 400 psi

Since 400 < 5018, the weld is adequately sized.

B. Opening Reinforcement

Openings in shells and heads are required to be reinforced under the rules of UG-36 of Div. 1. UG-36(c)(3) states that connections attached in accordance with Div. 1 rules and not larger than 3 inches in pipe size (in vessel shells or heads 3/8 in thick or less) need not be reinforced. Three attachments, while not larger than 3 in pipe size, are not attached according to Div. 1 rules, and reinforcement of their openings is treated in Part VI of this note.

Only one opening exceeds 3 in pipe size. This is the opening in torispherical head 4 to accomodate cylindrical shell 4. Calculation of reinforcement will be done using UG-37 rules.

To calculate the thickness of the head available for reinforcement it is necessary to determine the thickness required for the vacuum and weight loads. Weight loading must be considered since the entire weight of inner vessel and contents is transmitted to the head through the cylindrical shell.

The weight loading will be treated conservatively by converting it to an equivalent pressure, then applying this pressure to the entire head in addition to the vacuum load.

Equivalent pressure = weight of inner vessel + contents cross sectional area of cylindrical shell 4.

= 200/(π x 4^2 x 0.25)

= 16 psid

The total pressure which the head must withstand is then 15 + 16 = 31 psid. The thickness required for this pressure is found by trial and error using the procedure of UG-33. It is found that a thickness of 0.12 in is necessary to withstand the pressure of 31 psid on the convex side of the head. To prove this, (recalling that the lesser pressure as calculated by two means is to be used),

From UG-33(a)(1),

t = 0.12
L = 24
S = 15000 x 0.8 = 12000

$E = 1$

Then,

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

$P = 41$ psid

From UG-33(a)(2),

$t = 0.12$ in.

$R_o = 24$ in.

$A = 0.0006$

$B = 7000$

Then,

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

$P = 35$ psid

The minimum thickness of the head was previously established as 0.218 in. The amount of thickness available for reinforcement is 0.218 - 0.12 = 0.098.

The amount of area required for reinforcement of an opening in a head under external pressure is given in UG-37(c). The required area is

$A = dt_rF \times 1/2$

where $d = 4$ in

$t_r = 0.12$ in

$F = 1$

Then $A = 0.24\ in^2$

The amount of available reinforcement is calculated according to UG-40, by considering the area of a cross section through the head taken parallel to and containing the centerline of the opening. The cross section extends one shell diameter (4 in) on either side of the opening centerline. The area available for reinforcement is

$A = 4 \times 0.098 = 0.392\ in^2$

Since 0.392 > 0.24, the opening is adequately reinforced.

C. Consideration of Support Effects

The nitrogen dewars are supported on four supports welded to torispherical head 3. Each support has an area of contact with the head of 0.5 in^2. The total weight supported by the support is the weight of inner vessel plus contents (200 lbs) and the weight of the vacuum vessel (2 heads @ 30 lbs each plus 2 shells @ 50 lbs = 110 lbs). The total weight is then 310 lbs.

The support legs produce two types of failure consideration: stress and elastic stability. The rules of Div. 1 do not cover in detail support schemes (see non-mandatory Appendix G), suggesting good engineering practice. Stresses in the head at the attachment point will therefore be estimated by procedures from Ref. 1, and the difficult issue of stability will be considered from an empirical viewpoint.

1. Stresses in head in region of support

An appropriate procedure is given in Table 31, case 2, of Ref. 1. Each support will be considered to be far enough from adjacent supports and the head knuckle region that this procedure will apply.

r_o = radius required to produce 0.5 in^2 area = 0.4 in

t = 0.218 in

P = 310/4 = 77.5 lbs

$r'_o = r_o$

R_2 = 24 in

$\mu = r'_o\,[12\,(1-\nu^2)\,/\,(R_2^2\,t^2)]^{1/4} = 0.8$

B = 0.181

C = 0.429

Maximum membrane stress under center load is

$$\sigma_m = -BP\,(1-\nu^2)^{1/2}/t^2$$

$$\sigma_m = 281 \text{ psi}$$

Maximum bending stress under center of load is

$$\sigma_b = -CP\,(1+\nu)/t^2$$

$$\sigma_b = 900 \text{ psi}$$

These values of stress are well below the allowable of 12800 x 0.80 for the head.

2. Consideration of Elastic Stability

The effects of the support loads on the elastic stability of the head are very difficult to calculate. Few concentrated load sphere buckling solutions are available, and all are subject to suspicion due to the large effect of geometric and loading irregularities.

The most persuasive argument which can probably be made for the adequacy of the support system is also the least technical. As of this writing, the empty dewars have been used several times by personnel to gain access to other parts of the system. This use has consisted of standing on the top vacuum head (torispherical head 4). It is estimated that the heaviest of these individuals weighs approximately 200 lbs. Thus, since 120 lbs must be credited to LN_2 inventory weight, the dewar supports have been "load tested" to 80 lbs over the operational external loading, or approximately 125% of operational load.

Conclusion

Although there are other support options which are more easily analyzed, this support system is consistent with the requirements of Div. 1, and has been demonstrated to be safe.

IV. System Venting Verification

The 30" Bubble Chamber LN_2 dewars use the following reliefs (refer to schematic, Fig. 3):

1. 533T-8M-20 Circle Seal, 1" set 20 psig (RV-06-N).

2. 3/4" Fike rupture disc, set 50 psig (RD-04-N).

3. D559B-6M-30 Circle Seal, 3/4" set at 30 psig RV-07-N. This last valve is not really part of vessel but protects the LN_2 dewar from overpressurization from the supply.

Refering to the document "Maximum Pressure in the Tohoku Bubble Chamber Magnet System", it was shown that the worse case pressure rise would be 40 psig or 55 psid. Division 1 venting rules require that the vessel not be pressurized above 110% of the MAWP or 1.1 x 60 = 66 psid. Therefore, the relief valves are sufficient.

V. Pressure Test Results

For a MAWP = 60 psid, the Division 1 pneumatic test pressure is 60 x 1.25 = 75 psid. Both vessels were tested to 60 psig + vacuum = 75 psid. In fact LN_2

dewar B was tested to 66 psig = 81 psid. The vacuum vessel was internally pressure tested to 24 psig = 24 psid as part of the overall system.

VI. Exceptional Vessel Details

There are two areas in which the LN_2 dewars do not fully satisfy 14.1 requirements. These are A) welding, and B) minimum shell thickness. The reasons for non-compliance in these areas will be discussed and it will be shown that the dewars are believed to be safe for their intended use.

A. Welding

1. Documentation of welding procedures and welding qualifications

Section 14.1 requires that welding in Fermilab shops shall be done according to a Welding Procedure Specification (WPS) qualified under Section IX of the ASME Boiler and Pressure Vessel Code, and by a welder qualified to the process of the procedure under Section IX rules. It is further required that a document signed by a welding foreman or his designee stating that welding was made per an appropriate procedure and by a qualified welder be submitted as part of the Engineering Note. The following statements can be made concerning these requirements relative to the LN_2 dewars:

1. It seems clear from discussions with the dewar's design engineer that it was intended that the 14.1 requirements for welding would be met.

2. There exists a memo from the chairman of the 30-inch B.C. Conversion, Cryo Safety Panel stating that his understanding was that an appropriate procedure (ES-155001) and qualified welder were being requested for vessel welding by the design engineer.

3. The original construction log for these dewars cannot be found. According to the memory of the welder and technicians involved, these dewars were constructed in October-November, 1982. There is attached to this note a document showing that the welder performing the TIG welds on the stainless steel vessel was qualified in procedure ES-155001 in August, 1982. A copy of this procedure is also attached. The welder involved recalls this procedure as the one which was applied.

4. A signed document from the welding foreman or his designee stating that the welds on these vessels were done according to an appropriate procedure and by a qualified welder was not obtained.

It seems that the actual fabrication requirements as they relate to welding procedures and welder qualification have been met. As it seems inappropriate to request a document from the welding foreman two years after completion of the work, it is hoped that the information presented here is persuasive enough to allow an exception to the document requirement of 14.1.

2. Geometry of major vessel welds

Another aspect of the vessel welding which is suspect from the standpoint of Div. 1 rules is weld geometry. There are no weld design detail drawings to show that the fillet attachment welds were properly sized. The Div. 1 requirements for double fillet attachment welds are shown in Fig. UW-16(i). The minimum throat dimension is 0.7 t_{min}, where t_{min} is the thickness of the thinner member. The welder involved has said that such a geometry is standard in this type of weld, and that in the absence of a detail drawing, this is the minimum weld dimension which he would produce. In fact, it is very difficult to TIG weld a fillet of less than 1/16 in. along a leg, and it is this minimum dimension which was used in all of the weld strength calculations presented in Part III of this note.

No detail drawings exist for the single welded butt joints by which the heads attach to the torispherical shells. The welder and technicians involved recall prepping the heads with the appropriate v-groove, and separating the parts a small amount to ensure fusion at the root of the weld. The deposition of weld metal above the shell surface was kept to a minimum. At this point it should be recalled that 14.1 requires a reduction of allowable stress for in-house vessels because the Lab is not a "Code" shop with material control and independent third party inspection and testing. Given that it seems highly probable that the butt welds conform to Div. 1 geometry requirements, and that the design calculations used not only the 14.1 derating but also that required by UW-12 for non-radiographed vessels, then it is felt that the spirit of the 14.1 requirements for welding has been satisifed for all major parts of the vessel except the attachment welds used on flat heads 1 and 2.

3. Non-code welds on flat heads 1 and 2

The attachment of flat heads 1 and 2 represent a clear violation of Div. 1 rules. Flat head 1 is used to close the vacuum space between cylindrical shell 2 and cylindrical shell 4. The attachment of such heads is covered in Div. 1, Appendix 9, "Jacketed Vessels". Several attachment details are illustrated. None of these is exactly like the type of attachment used on the dewars. The detail of 9.5(d-1) is perhaps the most similar. 9-5(a) states that closures geometries other than those illustrated may be used if the requirements of UG-101 are met. UG-101 deals with proof testing of components. No proof testing was performed on this head attachment geometry; therefore this detail cannot be considered consistent with Div. 1 requirements.

The attachment of flat head 2 to flat head 1 is achieved by a single full fillet lap joint. UW-12 of Div 1 does not allow the use of this weld for this purpose. The probable reason is that the loading imposes considerable bending at the fillet, and Div. 1 recognizes the weakness of a single fillet weld in bending. Also, fillet welds present fatigue problems under cyclic loading conditions, and Div. 1 is strict on their use in such circumstances.

Although the weld details are not permitted by Div. 1, it can be demonstrated by finite element analysis that this closure scheme is adequate for an MAWP of 60 psid. The finite element model is shown in Fig. 4. This model is axisymmetric and includes the fillet welds, assuming minimum leg dimensions. The loading consists of 60 psi on the inner vessel components, and 15 psi on the atmospheric side of the vacuum shell. The total pressure differential across flat head 2 is 45 psid, and across cylindrical shell 2, 60 psid.

The refinement of the mesh is sufficient to calculate stresses around the fillets and in the shell/head junctions. Figure 5 shows a contour plot of the normal stresses in the meridional direction. The highest stress is seen to occur in the junction of flat head 1 and cylindrical shell 2. This stress is far below the maximum allowable stress of 12800 psi for the shell material. The stresses in the heads are well below the maximum allowable stress of 15000 psi for the head material.

The fillet welds must be considered from a shear stress (as opposed to normal stress) standpoint. Figure 6 shows the shear stress in the fillets, and shows these stresses to be far below the allowable stress in shear of 0.49 x maximum allowable stress of weakest welded component.

The nitrogen dewars are not subject to significant cyclic loadings; therefore, the fatigue failure of the closure welds is not considered.

The closure head finite element analysis shows these heads and their attachment scheme to comply with the safety factor requirements of Div. 1 as modified by 14.1. Although the attachment welds do not conform to Div. 1 rules, it is shown that the closures are more than adequate for dewar operation at 60 psid.

4. Single fillet attachment welds

Attachment welds 4, 6, and 7 (Fig. 1) do not comply with UW-16 of Div. 1. Single fillet welds may be used for necks and tubes up to and including 6 diameter only when welded on the inside of the vessel, or if on the outside, then only with the use of a welding groove cut in the shell. The three attachment welds in question are single fillets on the outside of the shell applied to a tube which extends through the shell. In this case, UW-16(d) requires that two fillets, inner and outer, be used.

In arguing for the acceptability of these welds the following points should be considered:

1. The UW-16 configurations are clearly stronger in terms of fatigue resistance and resistance to external mechanical forces or moments. However, two of the non-Div. 1 attachments occur on the vacuum vessel, which is in fact not required by Div. 1 to meet Div. 1 requirements. Hence, there is some indication that vacuum vessels are considered by Div. 1 to be more benign in their sensitivity to some details than other pressure vessels. The third weld occurs on a tube which is fully protected by the vacuum vessel from external mechanical loads.

2. The weld size calculations of Part III showed all welds to be greatly oversized with respect to the pressure loadings expected at the MAWP.

3. Attachment weld 4 on the inner vessel will not see enough thermal and pressure cycles to constitute a fatigue hazard. The non-Div. 1 welds on the vacuum vessel are even less of a fatigue consideration.

4. The attachment of tubes to vacuum vessels by single fillet welds on the outside of the vessel has been common (and successful) practice at the Laboratory.

Therefore, it is believed that the non-Div. 1 attachment welds do not affect the MAWP of the LN_2 dewar vessels.

B. Minimum Shell Thickness

The neck of the inner helium vessel (cyindrical shell 2) does not meet the minimum shell thickness requirements of Div. 1. UG-16 requires that all shells be a minimum of 1/16 in. The dewar neck is 0.035 in. In arguing for the exception of these vessels from this requirement, the following points should be considered:

1. The conduction heat load of the dewars is directly related to the thickness of the dewar neck. Compliance with this requirement would make efficient small dewar design in general more costly and difficult.

2. UG-16 excepts the inner pipe of double pipe heat exchangers from this requirement if they are 6 in. diameter or less. This indicates some consideration of the protection provided by the outer pipe, and perhaps of the heat transfer problems a large wall thickness can cause.

It is believed that this requirement is not a reasonable one from the standpoint of dewar design, and that the protection provided by the vacuum jacket in some way compensates for the thin wall used on a dewar neck. All stress calculations show this neck to be adequately sized for 60 psid under Div. 1 rules as modified by 14.1, and therefore the requirement of minimum wall thickness should be waived.

Conclusion

The vessel details presented here which do not meet the requirements of Div. 1 of Section VIII of the ASME Boiler and Pressure Vessel Code do not pose a danger to personnel or equipment during operation and do not adversely effect the MAWP of 60 psid calculated for the vessel. The vessel will normally operate well below the MAWP of 60 psid.

References

1. Roark, R.J., and Young, W.C., Formulas for Stress and Strain, 5th Edition, McGraw-Hill, 1975.

Table 1
Component Material, Dimensions and Allowable Stress

Component	Material	Dimensions	Maximum Allowable*** Stress (psi)
Cylindrical Shell 1*	SS304	20" OD x 0.188 wall	12800
Cylindrical Shell 2*	SS321	2-1/2" OD x 0.035 wall	12800
Cylindrical Shell 3*	SS304	24" OD x 1/4" wall	12800
Cylindrical Shell 4*	SS304	4" OD x 0.120 wall	12800
Torispherical Head 1**	SS304	20" dia x 3/16 wall	15000
Torispherical Head 2**	SS304	20" dia x 3/16 wall	15000
Torispherical Head 3**	SS304	24" dia x 1/4 wall	15000
Torispherical Head 4**	SS304	24" dia x 1/4 wall	15000
Flat Head 1	SS304	3-3/4 OD x 3/4 t	15000
Flat Head 2	SS304	3-5/8 dia x 3/8 t	15000

* Welded tube - efficiency of longitudinal tube weld is reflected in this value
** ASME flanged and dished head
*** Value is Div. 1 allowable stress multiplied by 0.8

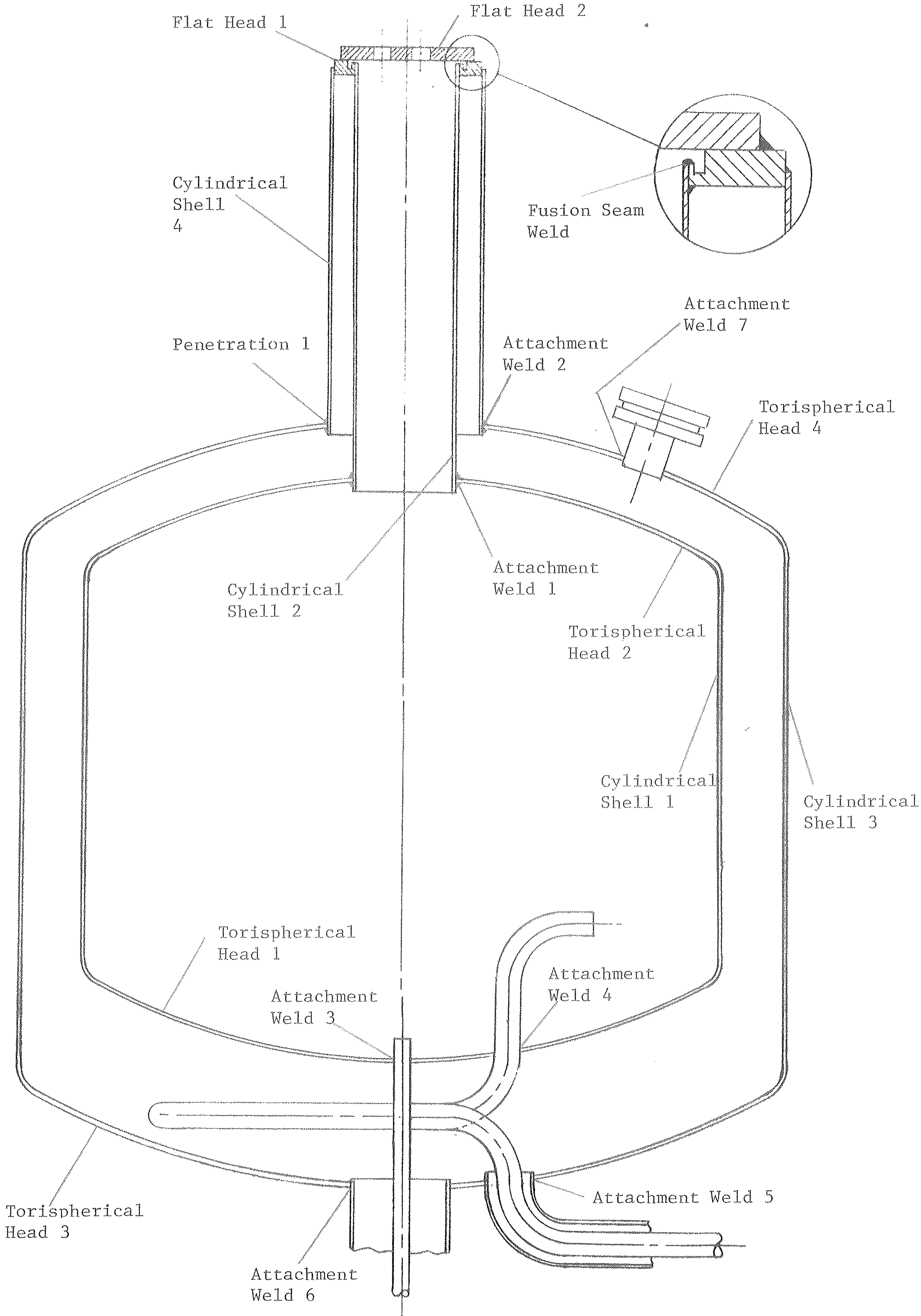


Fig. 1. LN_2 Buffer Dewar

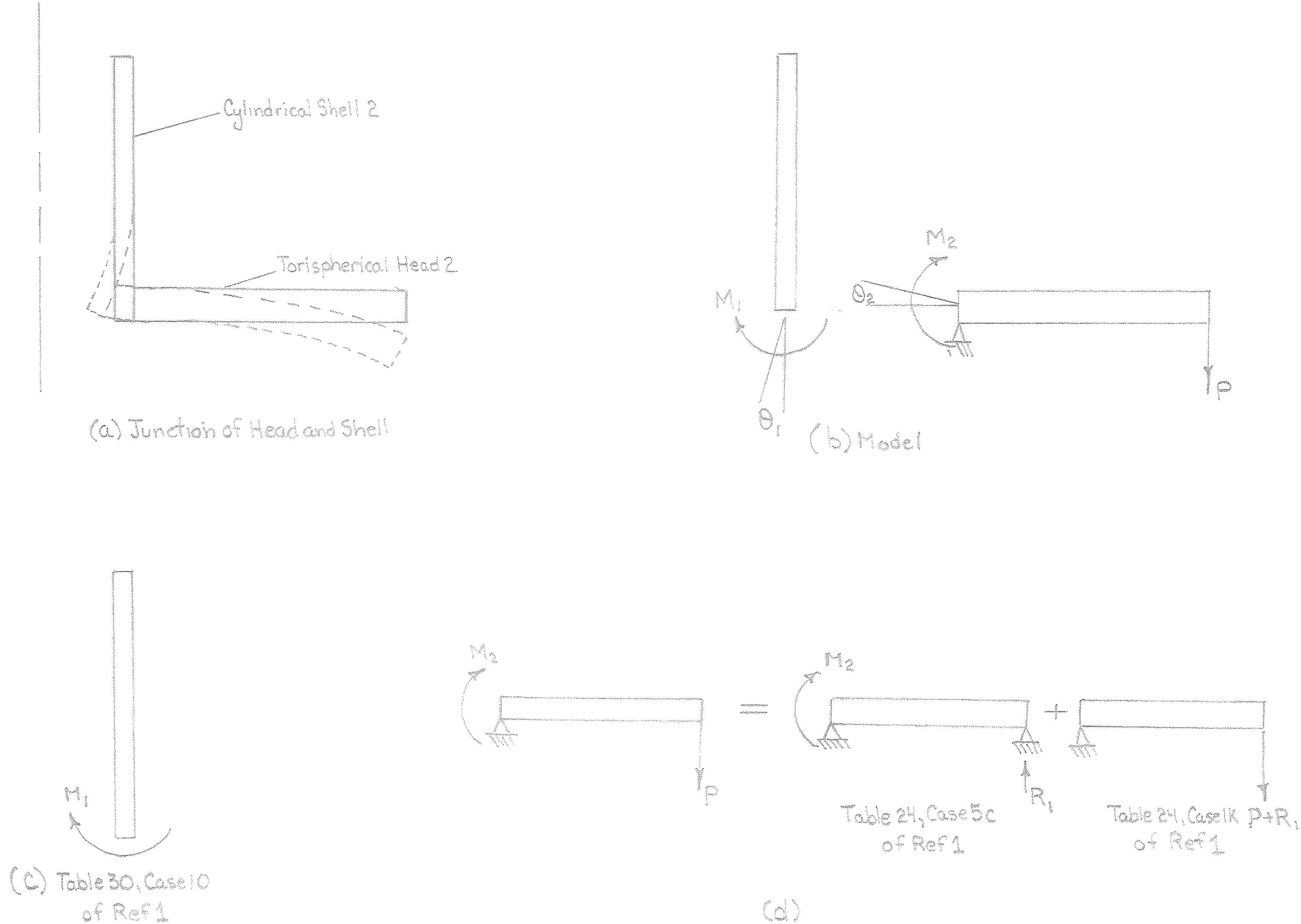


Fig 2. Modeling of Shell/Head Junction

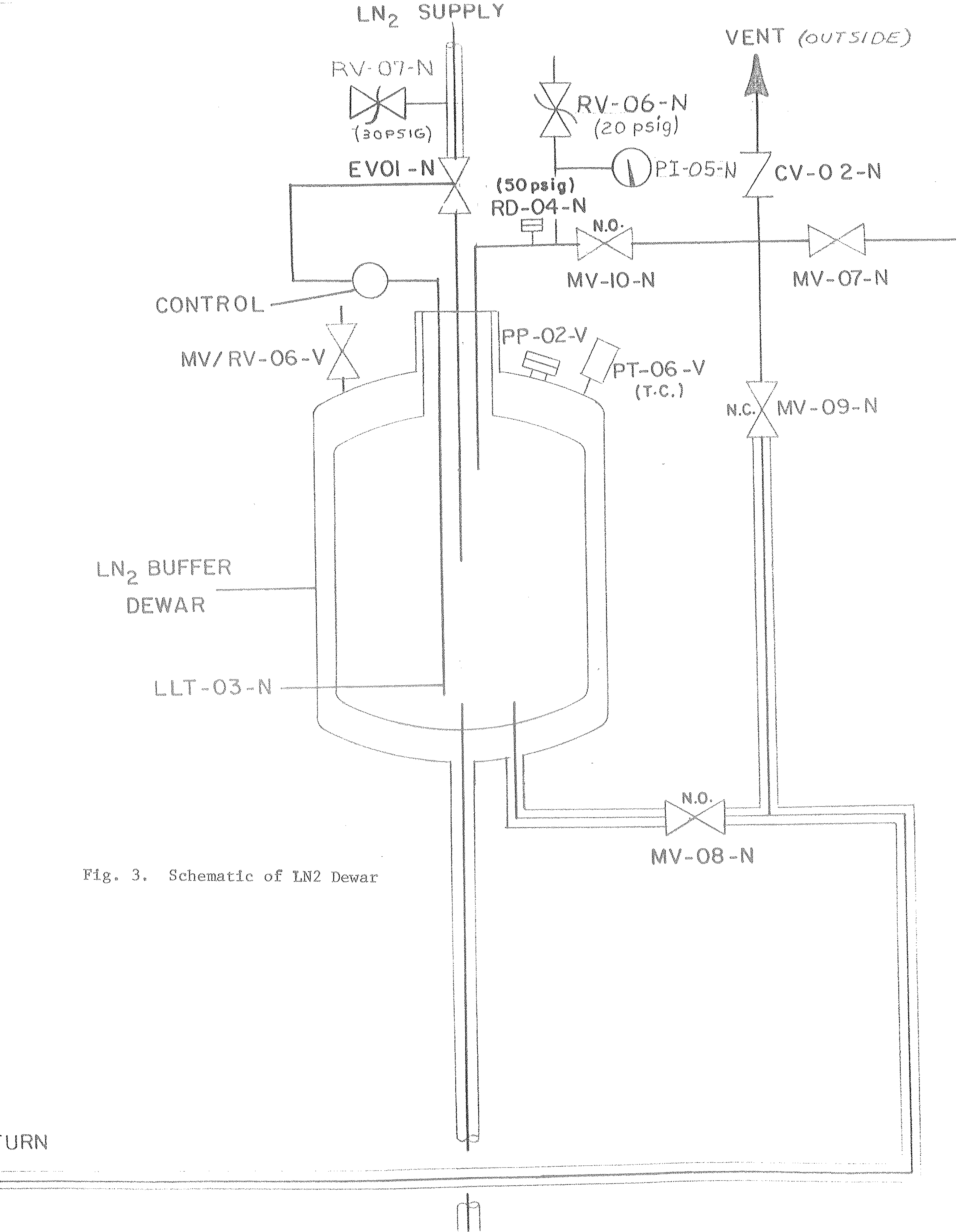
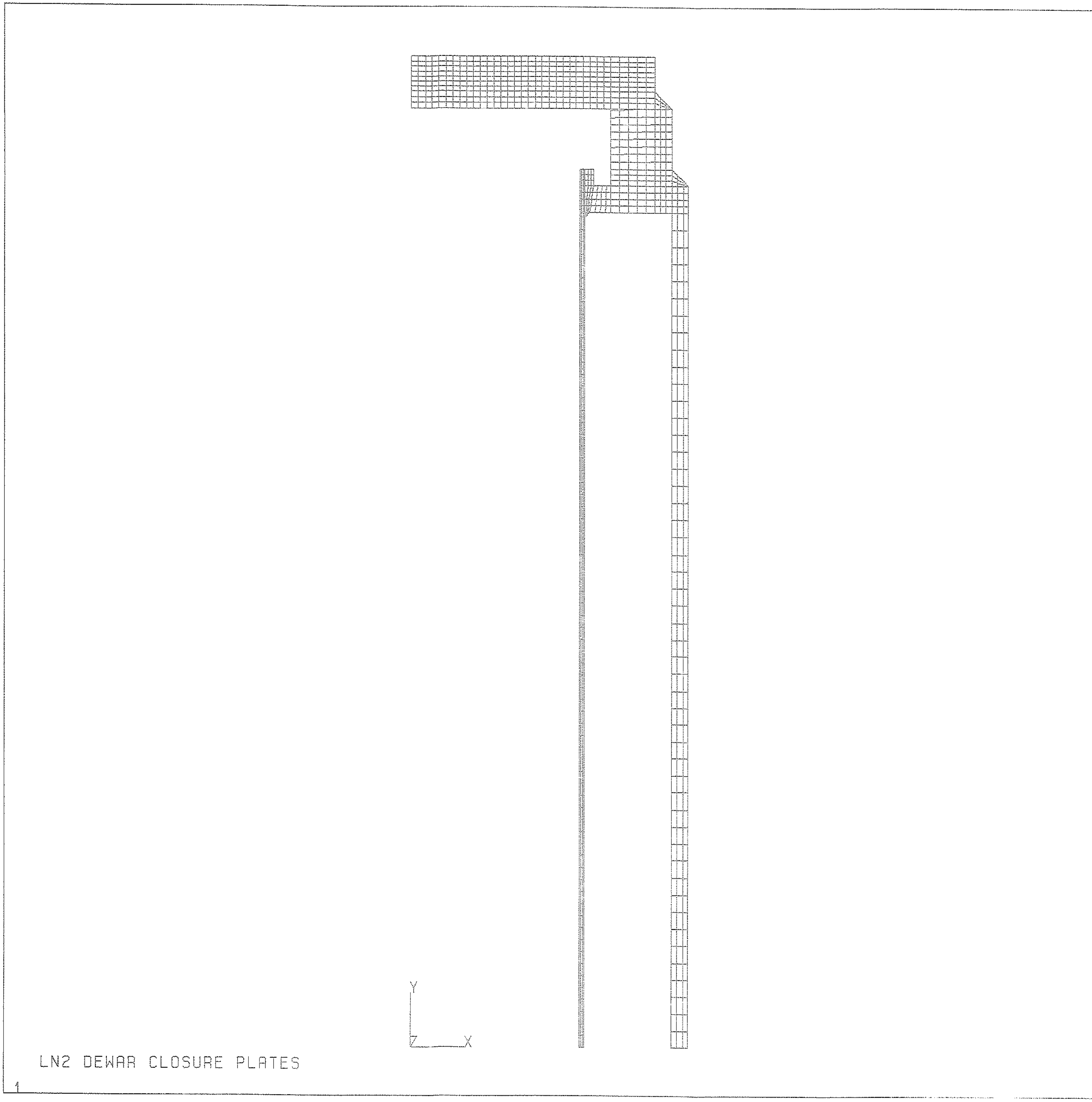


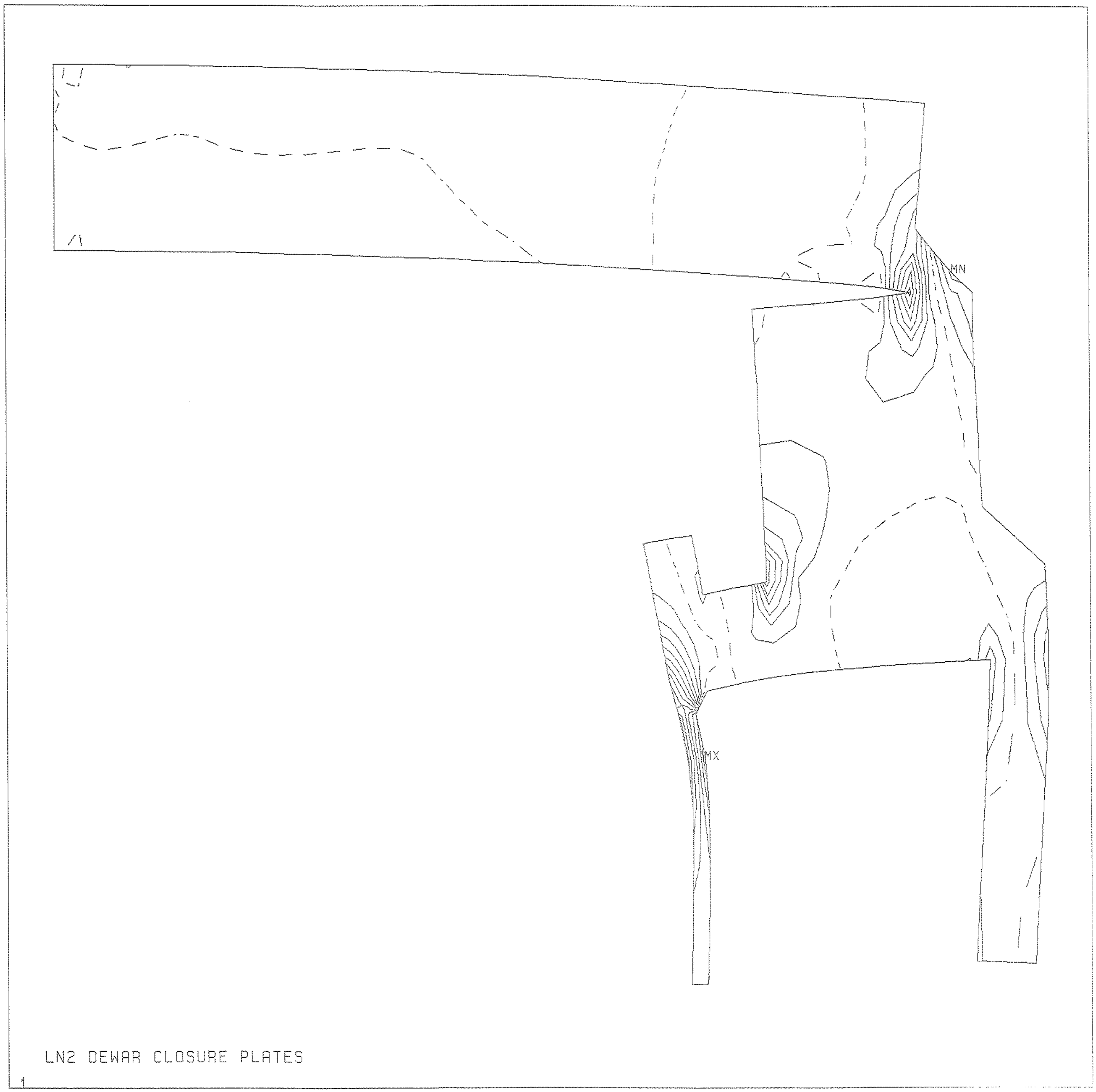
Fig. 3. Schematic of LN2 Dewar



ANSYS
84/12/10
8.6942
PLOT NO. 1
PREP7 ELEMENTS

ORIG SCALING
ZV=1
DIST=3.94
XF=1
YF=3.58

Fig 4. Finite Element Model of Flat Head/Shell Junction



ANSYS
84/12/10
11.1097
PLOT NO. 2
POST1
STEP=1
ITER=1
STRESS PLOT
SY

ORIG SCALING
ZV=1
DIST=1.1
XF=1
YF=6.29
EDGE
DMAX=.0000998
DSCA=1102
MX=2528
MN=-702
INC=200

Fig 5. Meridional Stresses in Flat Head/Shell Junction

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 37½°
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°F min.
Interpass Temp. 350°F max.
Preheat Maintenance 60°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.

FORM 1316 REV.

PITTSBURGH TESTING LABORATORY

ESTABLISHED 1881

PITTSBURGH, PA.

Order No. CH 5698

Date 8/5/82

PO #43643

PHYSICAL TEST REPORT OF WELDER PERFORMANCE QUALIFICATION TESTS

Client: Fermi National Accelerator Laboratory

P. O. Box 500, Batavia, Illinois 60510

Welder Name M. REYNOLDS Clock No. 3993 Stamp No. 48

Welding Process GTAW

Position (For vertical weld state whether upward or downward) 6G

(For Plate: Flat, horizontal, vertical, or overhead; For Pipe: Axis of pipe vertical, horizontal fixed or horizontal rolled).

In accordance with Procedure Specification No. ES155001 Formerly FNA-WP-900

Material – Specification SA312 to SA312 of P-No. 8 to P-No. 8

Diameter and Wall Thickness (if pipe) otherwise Joint Thickness 6" SCH 40 (.280 Wall)

Thickness Range this qualifies 1/16 to .560

FILLER METAL

Specification No. SAF 5.9

Describe Filler Metal ER-308L

Is Backing Strip Used? N/A

– For Information Only –

Filler Metal Diameter and Trade Name N/A Flux for Submerged Arc or Gas for Inert Gas Shielded Arc Welding N/A

Above Information by: PTL ☐ Client ☒ Other

Preparation of specimen witnessed by PTL Yes ☐ No ☒

GUIDED BEND TEST RESULTS

TYPE AND FIGURE NO.	RESULT	FIGURE NO.	RESULT
Face	PASS	Face	PASS
Root	PASS	Root	PASS

Test Witnessed by Client Test No. 10691

per

Results of tests (do) (~~do not~~) meet requirements of ASME SECT IX

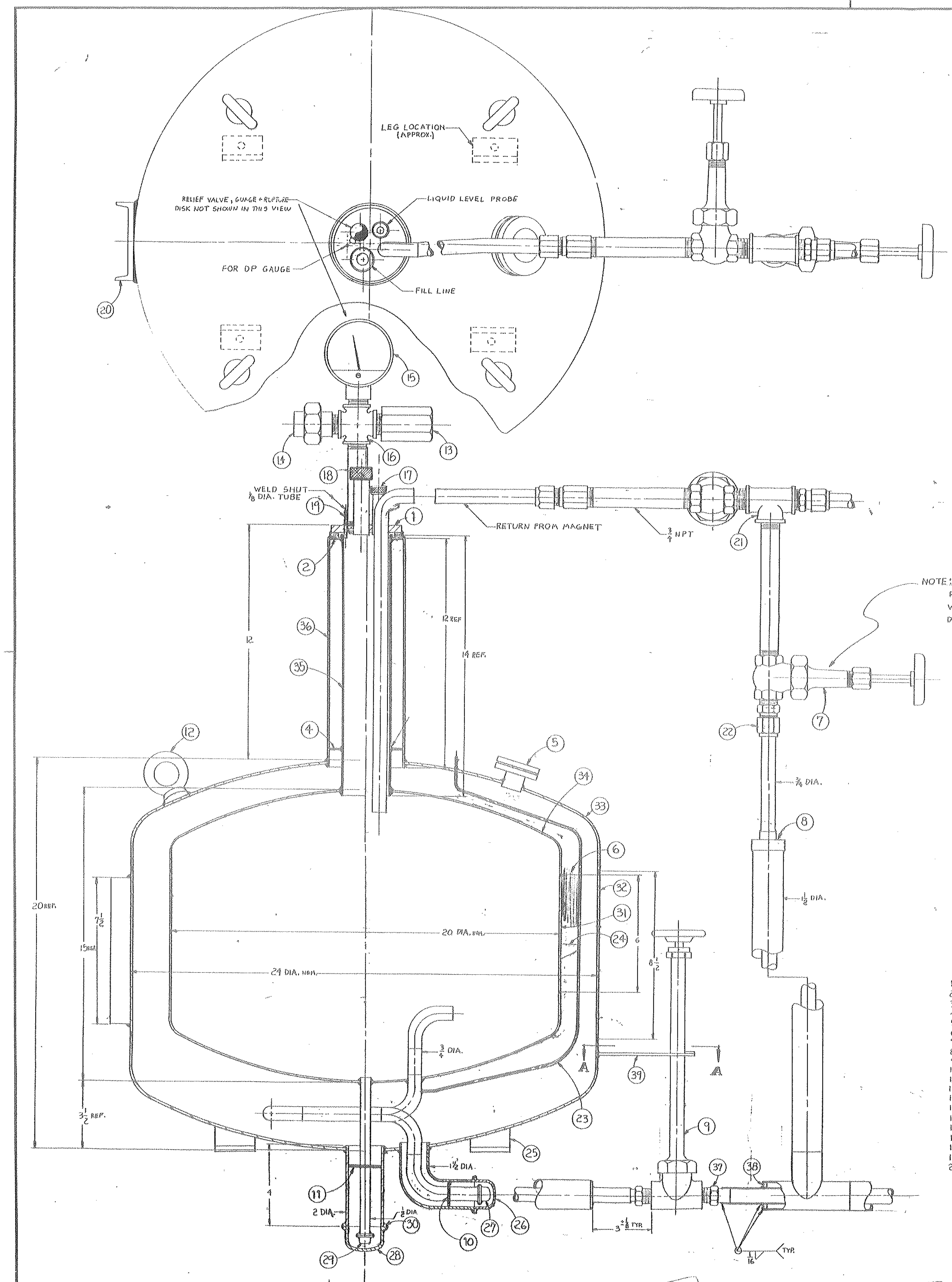
Remarks machined and tested by P.T.L.

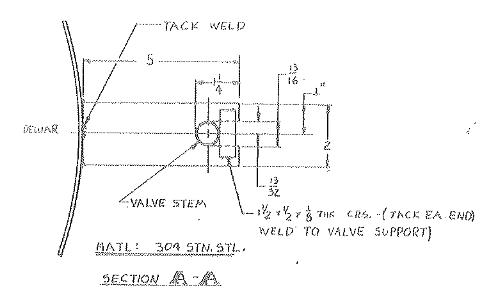
PITTSBURGH TESTING LABORATORY

dl

1cc: Client

REV.	DESCRIPTION	DRAWN	DATE
		APPD.	DATE





1. WELD INNER NECK TO INNER VESSEL TOP
2. WELD TUBES TO BOTTOM OF INNER VESSEL
3. WELD INNER VESSEL TOGETHER
4. WELD CONNECTING RING TO INNER TUBE
5. ATTACH G-10 SPIDER ON INNER NECK (WELD BEADS TO HOLD G-10 IN PLACE)
6. PLUG ALL TUBE ENDS + LEAK CHECK
7. COLD SHOCK
8. SUPERINSULATE INNER VESSEL
9. WELD OUTER NECK TO OUTER VESSEL TOP
10. WELD TUBES TO BOTTOM OF OUTER VESSEL
11. INSTALL SPIDER IN FILL LINE
12. WELD OUTER VESSEL TOGETHER
13. PLUG ALL TUBES + HOLES + LEAK CHECK
14. COLD SHOCK
15. INSTALL VACUUM FITTING
16. INSTALL LIQUID LEVEL PROBE
17. INSTALL RETURN FROM MAGNET
18. INSTALL BURST DISK ON VESSEL
19. INSTALL LIFTING HOOKS
20. INSTALL ALL OTHER VALVES + FITTINGS

ITEM NO.	PART NO.	DESCRIPTION OR SIZE	QTY. REQ.
39		VALVE SUPPORT (SEE SECT. 'A-A')	1
38	MB-156298	RETURN LINE FLANGE	2
37	PARKER 8-8FN-SS CONNECTOR	SOCKET END TO N.P.T. STRAIGHT	2
36		4" DIA. x .120 WALL STN. STL. TUBE	1
35		2 1/2" DIA. x .035 WALL STN. STL. TUBE	1
34		20" DIA. - 3/16" WALL - A.S.M.E. CODE HEAD STN. STL. - FLANGED + DISHED	1
33		24" DIA. - 1/4 WALL - A.S.M.E. CODE HEAD STN. STL. - FLANGED + DISHED	1
32		24" DIA. x 1/4" WALL STN. STL. TUBE	1
31		20" DIA. x .188 WALL STN. STL. TUBE	1
30	MB-	CAP FLANGES	8
29		1/2" DIA. STN. STL. CAP	1
28		2" DIA. STN. STL. CAP	1
27		3/4" DIA. STN. STL. CAP	1
26		1 1/2" DIA. STN. STL. CAP	1
25	MB-156160	SUPPORT LEG	4
24	MB-	TUBE BRACKET	1
23		1/8" ϕ x .035 WALL STN. STL. TUBE	36"
22		3/4" NPT - 3/4" ϕ TUBE "SWAGELOK"	3
21		3/4" x 3/4" x 3/4" NPT TEE	1
20		1" x 4" STN. STL. CHAN.	1
19		1/8" ϕ x .035 WALL STN. STL. TUBE	1 1/2"
18		5/8" ϕ STN. STL. VAC. COUPLING	1
17		3/8" ϕ STN. STL. VAC. COUPLING	1
16		3/4" x 3/4" NPT CROSS	1
15		PRESSURE GAUGE	1
14	"FIKE" CPV-UT	RUPTURE DISK - UNION TYPE 3/4" TO RUPTURE AT 50 P.S.I.	1
13	"CIRCLE SEAL"	INLINE TYPE RELIEF VALVE 3/4" FOR 20 P.S.I. RELIEF	1
12		STN. STL. PAD EYE	4
11		FILL LINE - SPIDER	1
10	MB-	LOWER LINE - SPIDER	1
9	"CRYOLAB" ES7-89-2TP62	EXTENDED STEM VALVE	1
8	MB-156075	BOTTOM RETURN LINE CAP	1
7	"GODDARD" P/N WCBN-6T	NON EXTENDED STEM 3/4" NPT BRS CRYOGENIC GATE VALVE	2
6		SUPERINSULATION LAYERS	25
5	"FIKE" ASSEMBLY P	2" CPV BT RUPTURE DISK ASSEM. TO RUPTURE AT 20 P.S.I.	1
4	MB-	DEWAR NECK - SPIDER	1
3		5/8" DIA. STN. STL. VACUUM COUP.	1
2	MB-156073	INNER + OUTER NECK CONN. RING	1
1	MB-	NECK - TOP PLATE	1

PARTS LIST

UNLESS OTHERWISE SPECIFIED			ORIGINATOR		
FRACTIONS	DECIMALS	ANGLES	DRAWN	P.J. CALAHAN	8-19-82
±	±	±	CHECKED		
			APPROVED		

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

USED ON

MATERIAL-

FERMI NATIONAL ACCELERATOR LABORATORY
UNITED STATES DEPARTMENT OF ENERGY

RESEARCH SERVICES
30" BUBBLE CHAMBER COIL CONVERSION
LN_2 DEWAR ASSEMBLY

SCALE	FILMED	DRAWING NUMBER	REV.
HALF		2771-ME-156145	