LOFT TECHNICAL REPORT LTR 1210-3 Rev. A
RE & C REPORT RE-E-76-066 Rev. A
SEPTEMBER 14, 1977

STRESS ANALYSIS OF LOFT CONTAINMENT
VESSEL ATTACHMENTS FOR THE MAINSTEAM
AND FEEDWATER PIPING SUPPORT STRUCTURES

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STRESS ANALYSIS OF LOFT CONTAINMENT
VESSEL ATTACHMENTS FOR THE MAINTEAM
AND FEEDWATER PIPING SUPPORT STRUCTURES

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**LOFT TECHNICAL REPORT
LOFT PROGRAM**FORM EG&G-229
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ABSTRACT

The LOFT Containment Vessel attachments for the Mainstream and Feedwater Piping Support Structures, shown on Ref. 6, have been analyzed for operating and faulted loading conditions. The same attachments were analyzed in LTR 1120-3, Ref. 9, with the only major difference between this report and Ref. 9 being that faulted conditions have also been considered in this report.

The loading used in this analysis is from LTR 115-13, Sup. A, Ref. 5. Reference 5 contains the analysis of the piping support structures. This report contains the analysis of the connections to the containment vessel for the most current design and loading. Also contained in this report is the analysis of the piping supports up to the point where they are analyzed in LTR 115-13.

The loading included LOCE, LOCA, SSE, and thermal loads. The containment vessel was analyzed to the criteria of the 1965 ASME Code, Ref. 1, and the piping supports were analyzed to the criteria of the 1974 ASME Code, Ref. 2, as specified in the design specification, Ref. 10.

SUMMARY

A linear elastic static stress analysis of the LOFT Containment Vessel Attachments for the Mainsteam and Feedwater Piping Support Structures has been completed. Seismic, operational, faulted, and thermal loading were obtained from LTR 115-13, Sup. A, Reference 5. The computer model used to obtain the loads in Reference 5 is shown in Figure 1 of this report.

The stress levels obtained in the containment vessel and the supports are well within the allowable limits set in the 1965 and 1974 ASME Codes.

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1.0 DESCRIPTION OF SYSTEM

The Mainsteam and Feedwater Piping Support Structure is a frame structure built out of steel beams which are supported off of the LOFT Blowdown Suppression Tank Shielding Structure and the LOFT Containment Vessel. This structure provides support to the piping systems for seismic, operational, faulted, and thermal loading.

The supports off of the LOFT Containment Vessel consist of six- or twelve-inch diameter carbon steel pipes, SA 333, welded to the containment vessel shell, which is carbon steel plate, SA 516, one and three sixteenths-inches thick.

2.0 ANALYSIS

A Bijlaard Analysis, Reference 7, was performed to determine the stresses in the containment vessel shell due to the attachments. These stresses were combined with those caused by an internal pressure differential of forty psi, and the discontinuity stresses due to a temperature difference between the vessel shell and the attachment. The stresses due to this difference in temperatures was determined using conservative formulas found in Subsection NB of the ASME Code, Reference 2.

3.0 RESULTS

All of the stresses found in the containment vessel shell were below the allowable values set in the 1965 ASME Code, Reference 1.

The newly designed attachments, whose design had changed since their analysis in LTR 115-13, Sup. A, all met the criteria specified in the 1974 ASME Code, Reference 2.

4.0 REFERENCES

1. American Society of Mechanical Engineers, Boiler and Pressure Vessel Code, 1965 Edition, including addenda through Summer 1965.
2. American Society of Mechanical Engineers, Boiler and Pressure Vessel Code, Section III, "Nuclear Power Plant Components", 1974.
3. American Society of Mechanical Engineers, Boiler and Pressure Vessel Code, Section VIII, "Unfired Pressure Vessels", 1965.
4. R. G. Rahl, "LOFT Feed and Steam Lines Isolation Snubbers", LOFT Technical Report LTR 115-13, December 22, 1975.
5. R. G. Rahl, "LOFT Feed and Steam Lines Isolation Snubbers", LOFT Technical Report LTR 115-13, Sup. A, September 22, 1977.
6. EG&G Idaho Drawings, Numbers: 207325 Rev. E, 207365 Rev. G, 207275 Rev. D, 207381 Rev. E, 207607 Rev. E, and 207609 Rev. C.
7. Welding Research Council Bulletin #107, August 1965, Third Revised Printing, April 1972.
8. American Institute of Steel Construction, Manual of Steel Construction, Seventh Edition, 1970.
9. D. P. Finicle, "Stress Analysis of LOFT Containment Vessel Attachments for the Mainsteam and Feedwater Piping Support Structures", LOFT Technical Report LTR 1210-3 (RE-E-76-066), November 19, 1976.
10. EG&G Idaho Specification, "Containment Vessel Attachments for the Mainsteam and Feedwater Piping Support Structures Design Specification", ES 60190, Rev. A, November 4, 1976.

CW=Containment Wall

Steam Line
Parallel To Y-Axis

CW
30

Feed Line
Parallel To X-Axis

CW 48

CW 49

35
CW

37
CW

2
CW

#-Node Number

COMPUTER MODEL FROM LTR 11/5-13

FIGURE 1

by R.G. RAHL

LTR 1210

3

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APR 21 1902

FIGURE 4

ATTACHMENT AT
NODES 2, 35, 37 & 30

△ 306.87
(25'-6.9)

209.5
(17'-5.5) △ 6

35'-0" R
REF

CONTAINMENT
VESSEL SHELL

15.3

35°

1.00

31°

13.5

6

20

21

18

24

25

26

31

4

APPENDIX A
CALCULATIONS

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SUBJECT ANALYSIS OF CONTAINMENT ATTACHMENTS FOR DATE _____
MAIN STEAM & FEEDWATER SNUBBER SUPPORT STRUCTURE
 PREPARED BY D. FINIGLE CHECKED _____ WORK REQUEST _____

PURPOSE 3. The purpose of this report is to show the structural adequacy of the attachments to the ^{LOFT} containment vessel, which support the MAIN STEAM & FEEDWATER SNUBBER SUPPORT STRUCTURE. There are a total of 8 attachments to the vessel, all of which are circular attachments welded to the containment vessel wall.

The purpose of this revision is to include some additional loading (LOCA + SSE) that was not considered in the original LTR 1210-3. Also an oversight in the S_m value for the containment vessel, will be corrected. Per the 1965 ASME Code, SECTION III, CLASS A stress analysis must be performed, but SEC VIII Allowable stress values (S_m) must be used. The conclusions for the LOCE or SSE loads, found in LTR 1210-3 still remain the same.

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ANALYSIS:

The maximum of LOCA or SSE loading will be considered as an operational loading condition and the 1965 ASME Code, ref 1, will be used as the acceptance criteria and the LOCA + SSE will be considered as a faulted loading condition and the 1974 ASME Code, ref 2, will be used for the acceptance criteria.

The 1974 ASME Code References Appendix F, ref 2, for the faulted Allowables. Appendix F gives the allowable limits for primary stresses; P_m , P_r & P_b , as the lesser of

$$\left\{ \begin{array}{l} 2.4 \text{ Sm} \\ .7 \text{ Su} \\ .7 \text{ Sh} \end{array} \right\} \text{ for mat of } \begin{array}{l} \text{tbl } 1-1.2 \\ \text{" " " " } 1-1.1 \end{array}$$

WORK REQUEST

$S_{\text{min}} = 38,000$
 $S_{\text{min}} = 70,000$
 $S_{\text{min}} = 17,500 \text{ psi}$

} from pg 100, ref 3

H. -39

loads from reference 5 are listed on the following page.

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WORK REQUESTED:

MECHANICAL LOADS *

lbs. & in-lbs

Node #	Beam #	P	M _c	M _L	M _T	V _c	V _L
48	40	2521	-0-	45116	63	-0-	153
49	41	2469	-0-	28291	604	-0-	1681
30	50	2286	4267	10421	3466	1117	422
2	4	5258	92291	709	426	3824	1213
35	6	5400	100143	623	374	3938	1266
37	44	1023	2538	18672	9728	840	297

LOADS DUE TO THERMAL EXPANSION **

lbs: £ m - lbs

NODE #	BEAM #	P	M _C	M _L	M _T	V _C	V _L
48	40	423	- 0 -	25566	272	0	204.7
49	41	1351	- 0 -	8701.	289.7	0	74.8.
30	50						
2	4						
35	6	2227	66475	12603	7573	1523	338
37	44						

LOCA + SSE ***

lbs & in - lbs

Node #	Beam #	P	M _c	M _L	M _T	V _c	V _L
48	40	8811	-0-	159050	143	-0-	388
49	41	8850	-0-	103155	1703	-0-	6253
30	50	3436	8012	26130	8692	2060	1086
2	4	3936	91999	138	83	3078	1302
35	6	4297	107615	234	140	3325	1358
37	44	8180	3608	116350	81500	6770	2495

***These loads are from App. B, LTR 115-13, sup. A, ref. 5

* These are the maximum of L_{PCE} or SSE , from ref. 5

** These loads are from App. B, LTR 115-13, ref 4

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The configuration at nodes 48 & 49 is shown in fig 2. Nodes 48 & 49 actually were modeled where the 2 24" DIA Pipes connect to the frame (see fig 2). The 24" DIA Pipes are drawn in phantom on fig 2. Since the computer model did not actually model the configuration as it now is, some adjustments must be made to the applied loads.

SKETCH 1

The loading of pipe supports will be determined based on their location and the relative stiffnesses of the members. As an easy way of keeping track of the 3 attachments, they will be numbered # as shown in sketch 1.

STRESS ANALYSIS FOR OPERATING CONDITIONS

SSE alone must meet normal operating allowables.

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MECHANICAL LOADS ON ATTACHMENTS 1, 2 & 3.

The loads on each pipe will be as follows.

① Attachment ①

$$P^{①} = P_{49} = 2469$$

P at Node 49, from pg 4

$$P^{②} = 0$$

$$P^{③} = P_{48} = 2521\#$$

$$M_c^{①} = M_c^{②} = M_c^{③} = 0$$

$$M_L^{①} = \frac{M_{L48} + M_{L49}}{3} = M_L^{②} = M_L^{③} = \frac{45116 + 28291}{3} = 24470 \text{ "#}$$

$$M_T^{①} = \frac{M_{T48} + M_{T49}}{3} = M_T^{②} = M_T^{③} = \frac{63 + 604}{3} = 222 \text{ "#}$$

$$V_c^{①} = V_c^{②} = V_c^{③} = 0$$

$$V_L^{①} = \frac{V_{L48} + V_{L49}}{3} = \frac{153 + 1681}{3} = 611.16 = V_L^{②} = V_L^{③}$$

MECH. LOADS ON ATTACHMENT ①

MECH. LOADS ON ATTACHMENT ②

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MECH LOADS ON ATTACHMENT (3)

FROM PG 2, THE THERMAL LOADS ARE AS FOLLOWS.

$$P^{(1)} = P_{49} = 1351 \text{ lb}$$

$$P^{(2)} = 0$$

$$P^{(3)} = P_{48} = 423 \text{ lb}$$

$$M_L^{(1)} = M_L^{(2)} = M_L^{(3)} = 0$$

$$M_L^{(1)} = \frac{M_{L48} + M_{L49}}{3} = M_L^{(2)} = M_L^{(3)} = \frac{25566 + 8701}{3} = 11420 \text{ in-lb}$$

$$M_T^{(1)} = \frac{M_{T48} + M_{T49}}{3} = M_T^{(2)} = M_T^{(3)} = \frac{272 + 2897}{3} = 1056 \text{ in-lb}$$

$$V_C^{(1)} = V_C^{(2)} = V_C^{(3)} = 0$$

$$V_L^{(1)} = \frac{V_{L48} + V_{L49}}{3} = \frac{204 + 748}{3} = 320 \text{ lbs} = V_L^{(2)} = V_L^{(3)}$$

THERMAL LOADS AT THE CONTAINMENT SHELL

ATTACHMENT (1)

$$P = 1351$$

$$M_L = 22230 \text{ in-lb}$$

$$M_T = 1056 \text{ in-lb}$$

$$V_L = 320 \text{ lb}$$

ATTACHMENT (2)

$$M_L = 22230 \text{ in-lb}$$

$$M_T = 1056$$

$$V_L = 320 \text{ lb}$$

ATTACHMENT (3)

$$P = 423 \text{ lb}$$

$$M_L = 22230 \text{ in-lb}$$

$$M_T = 1056 \text{ in-lb}$$

$$V_L = 320 \text{ lb}$$

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COMBINED MECHANICAL + THERMAL LOADS

ATTACHMENT ①

$$P = 2469 + 1351 = 3820 \text{ lbs.}$$

$$M_1 = 45121 + 22230 = 67350.14$$

$$M_T = 222 + 1056 = 1278 \text{ " \#}$$

$$V_1 = 611 + 320 = 931\#$$

ATTACHMENT (2)

$P = 0 -$

$$M_1 = 45121 + 22230 = 67350, \text{ " \#}$$

$$M_T = 222 + 1056 = 1278 \text{ " \#}$$

$$V_L = 611 + 3.20 = 931 \text{ \#}$$

ATTACHMENT (3)

$$P = 2521 + 423 = 2944 \text{ lb.}$$

$$M_L = 4512.1 + 22230 = 67350.1 \text{ kg}$$

$$M_T = 222 + 1056 = 1278 \text{ g}$$

$$V_1 = 611 + 320 = 931 \#$$

From inspection of the above loads, Attachment ① has the highest loading for Mechanical plus Thermal and Attachment ③ has the highest loading for Mechanical loads alone.

At the location of attachment 1, Elevation $\approx 4813'$ the containment vessel is a cylindrical shell of 35' radius and a thickness of $1\frac{3}{16}$ inches.

Attachment ① is 16" dia. Sch 80 Pipe attached with a full penetration groove weld.

A Bjlaard Analysis ^{ref. [2]} of the containment vessel will be performed.

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Page 8 is the chart containing the Bylaard Analysis. Parameters for the attachment are as follows.

$$R_m = 35 ft = 420 \text{ inches}$$

$$r_o = 6.625/2 = 3.31 \text{ inches}$$

$$T = 1 \frac{3}{16}'' = 1.19 \text{ inches}$$

$$\gamma = \frac{R_m}{T} = \frac{420}{1.19} = 353$$

$$\beta = .875 \frac{r_o}{R_m} = .875 \frac{3.31}{420} = .0069$$

Attachment ① will be analyzed for Mechanical plus thermal loading. If the stress level in the attachment and the shell are below the levels allowed for mechanical loads alone, then attachment 3 will not be analyzed.

Stresses in Shell due to Internal Pressure = 40 psi

$$\sigma_\phi = \frac{Pr}{t} = \frac{40(35)12}{1.19} = 14120 \text{ psi}$$

$$\sigma_x = \frac{Pr}{2t} = 7060 \text{ psi}$$

From pg 10, the maximum membrane plus bending stress occurs at point A_u. Adding the stresses due to internal pressure yields

$$\sigma_\phi = 12340 + 14120 = \underline{\underline{26460 \text{ psi}}}$$

$$\sigma_x = 14900 + 7060 = \underline{\underline{21960 \text{ psi}}}$$

Table 5—Computation Sheet for Local Stresses in Cylindrical Shells

ATTACHMENT ①

1. Applied Loads*

Radial load, $P = 3870$ lb.
 Circ. Moment, $M_c = 67350$ in. lb.
 Long. Moment, $M_L = 1278$ in. lb.
 Torsion Moment, $M_T =$
 Shear Load, $V_c =$
 Shear Load, $V_L = 931$ lb.

2. Geometric Parameters

$R_m = 353$
 $H = (0.875) \frac{P}{R_m} = 0.0068$

Stress Concentration due to:
 a) membrane load, $K_n = 1.0$
 b) bending load, $K_b = 1.0$

*NOTE: Enter all force values in accordance with sign convention

CYLINDRICAL SHELL

From Fig.	Read curves for	Compute absolute values of stress and enter result	STRESSES - if load is opposite that shown, reverse signs shown							
			Au	AL	Bu	BL	Cu	CL	Du	DL
3C	$\frac{H\phi}{P/R_m}$ 64	$K_n \left(\frac{H\phi}{P/R_m} \right) \cdot \frac{P}{R_m T}$ 489	-490	-490	-490	-490	-490	-490	-490	-490
1C	$\frac{H\phi}{P}$ 24	$K_b \left(\frac{H\phi}{P} \right) \cdot \frac{6P}{T^2}$ 3284	-3900	+3900	-3900	+3900	-3900	+3900	-3900	+3900
3A	$\frac{H\phi}{M_c/R_m^2\beta}$ —	$K_n \left(\frac{H\phi}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T}$ —								
1A	$\frac{H\phi}{M_c/R_m\beta}$ —	$K_b \left(\frac{H\phi}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2}$ —								
3B	$\frac{H\phi}{M_L/R_m^2\beta}$ 24	$K_n \left(\frac{H\phi}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T}$ 1115	-1120	-1120	+1120	+1120				
1B or 1B-1	$\frac{H\phi}{M_L/R_m\beta}$ 107	$K_b \left(\frac{H\phi}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2}$ 7130	-7130	+7130	+7130	-7130				
Add algebraically for summation of ϕ stresses, $\Sigma \phi$			-12640							
4C	$\frac{H_n}{P/R_m}$ 60	$K_n \left(\frac{H_n}{P/R_m} \right) \cdot \frac{P}{R_m T}$ 458	-460	-460	-460	-460	-460	-460	-460	-460
2C	$\frac{H_n}{P}$ 18	$K_b \left(\frac{H_n}{P} \right) \cdot \frac{6P}{T^2}$ 2913	-2920	+2920	-2920	+2920	-2920	+2920	-2920	+2920
4A	$\frac{H_n}{M_c/R_m^2\beta}$ —	$K_n \left(\frac{H_n}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T}$ —								
2A	$\frac{H_n}{M_c/R_m\beta}$ —	$K_b \left(\frac{H_n}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2}$ —								
4B	$\frac{H_n}{M_L/R_m^2\beta}$ 7.0	$K_n \left(\frac{H_n}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T}$ 325	-325	-325	+325	+325				
2B or 2B-1	$\frac{H_n}{M_L/R_m\beta}$ 11	$K_b \left(\frac{H_n}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2}$ 1171	-11200	+11200	+11200	-11200				
Add algebraically for summation of H stresses, ΣH			-14900							
Shear stress due to Torsion, M_T										
Shear stress due to load, V_c										
Shear stress due to load, V_L										
Add Algebraically for summation of shear stresses, $\Sigma \tau$										
COMBINED STRESS INTENSITY, S										
1) When σ_ϕ & σ_n have like signs			$S = \frac{1}{2} [\sigma_\phi + \sigma_n + \sqrt{(\sigma_\phi - \sigma_n)^2 + 4\tau^2}]$							
2) When $\tau = 0$			$S = \text{largest of } \sigma_\phi, \sigma_n = \sigma_\phi - \sigma_n $							
3) When σ_ϕ & σ_n have unlike signs			$S = \sqrt{(\sigma_\phi - \sigma_n)^2 + 4\tau^2}$							

$N_1/(M_L/R_m^2\beta)$ so determined by (C_L) from Table 8 (see para. 4.3).

4.2.2.5.2: When considering bending moment (M_1) : $\beta = K_L \sqrt{\rho_1 \beta_1}$, where K_L is given in Table 8.

4.3 Calculation of Stresses

4.3.1 STRESSES RESULTING FROM RADIAL LOAD, P .4.3.1.1 Circumferential Stresses (σ_ϕ):

Step 1. Using the applicable values of β and γ

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∴ Maximum Stress Intensity = 26760 psi ^{from pg 9}
which is $P_L + P_B + Q$.

The Containment Vessel shell is SA-516 G270 Carbon Steel which has a $S_m = 17,500$ psi, from Section VIII, ASME Code, ref [3]

from Fig N-414, pg 17, ref [1]

$$P_L < 1.5 S_m = 1.5(17,500) = 26,250 \text{ psi}$$

$$P_L + P_B + Q < 3.0 S_m = 3.0(17,500) = 52,500 \text{ psi}$$

$$P_L + P_B + Q = 26,760 < 52,500$$

From pg 10, it can be seen that the membrane stress ^{intensity} due to mech + Thermal loading is 1610 psi. Only the mechanical contribution to the membrane stress goes to make-up P_L . Since the value of 1610 psi is larger than the actual contribution to P_L made by the mechanical loads, it will be conservative to use this value.

MAXIMUM MEMBRANE STRESS.

$$P_m = 14,120 \text{ psi} \sim \text{due to internal pressure}$$

$$P_m < S_m = 17,500 \text{ psi}$$

$$P_L = 14,120 + 1620 = 15,740 \text{ psi}$$

$$P_L < 1.5 S_m = 26,250$$

∴ Attachments 1, 2, & 3 Meet the Operating Allowables of ref [1] for LOCE + Thermal loading. See note top of pg 12.

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NOTE: Fatigue has not been considered in this analysis because there are no peak (F) stresses in the shell and since $P_L + P_B + Q < 3S_m$ this gives a fatigue life ≈ 50000 cycles, from ref 1, LORT design life ≈ 260 cycles.

ANALYZE THE ATTACHMENT PIPES "6" Sch 80"

From pg 6, The max loads on the pipe are

$P = 3820 \text{ lbs}$

$$A = 8.405 \text{ m}^2$$

$M_L = 67350 \text{ " \#}$

$$I = 40.5 \text{ in}^4$$

$$M_r = 1278$$

$$S = 12.22$$

$$V_1 = 931 \#$$

$$A_{\text{pipe}} = \frac{P}{A} + \frac{M}{S} = \frac{3820}{8.405} + \frac{67350}{12.22} = 454 + 5511 = 5965 \text{ psi}$$

$$\tau = \frac{T_c}{J} + \frac{2V}{A} = \frac{1278(3.31)}{2(40.5)} + \frac{2(931)}{8.405} = 52.2 + 221 = 274 \text{ psi}$$

Pipe is SA 333 GR6, $F_y = 30,000$

$$S_m = 16.5 \text{ ksi} \quad \text{from Tbl I-10.1, pg 463, ref [2]}$$

$F_b = .66 F_y$ } from pg 371, App. XVII, ref 2.

$$F_b = .66(30) = 19,800 \text{ psi}$$

5965 LL 19800

∴ The Pipes (16" Sch 80) are OK

∴ Attachments ① ② & ③, see sketch 1, pg 5,
are structurally adequate

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ANALYSIS OF MAIN STEAM HANGER H39 FOR OPERATING CONDITIONS

This hanger was not analyzed in ref [4], because it was not designed at the time of the analysis, but the loading on this hanger was determined to be 2818 lbs axial force, in ref [5]. The hanger assembly is shown in figure 3. The attachment to the containment vessel is with a 12" sch 60 steel pipe welded to the vessel with a 1/2" fillet weld.

A Bijlaard analysis ^{ref [7]} of the Containment Vessel shell will be performed.

$$\left. \begin{array}{l} \sigma_{\phi} = 14120 \text{ psi} \\ \sigma_x = 7060 \text{ psi} \end{array} \right\} \begin{array}{l} \text{from pg 9} \\ \text{for internal pressure} \end{array}$$

from pg 14, the maximum stresses ^{for mechanical + thermal loads} have

$$\sigma_{\phi} = 3230 \text{ psi}$$

$$\sigma_x = 2490 \text{ psi}$$

combined stresses

$$\sigma_{\phi} = 14120 + 3230 = 17350 \text{ psi}$$

$$\sigma_x = 7060 + 2490 = 9550 \text{ psi}$$

$$P + P_B + Q = 17350 \text{ psi} < 3 S_m = \underbrace{52500 \text{ psi}}_{\text{pg 11}}$$

Table 5—Computation Sheet for Local Stresses in Cylindrical Shells

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HANGER 39

14/

1. Applied Loads*

Radial load, $P = 2018$ lb.
 Circ. Moment, $M_c =$ in. lb.
 Long. Moments, $M_L =$ in. lb.
 Torsion Moment, $M_T =$ in. lb.
 Shear Load, $V_c =$ lb.
 Shear Load, $V_L =$ lb.

3. Geometric Parameters

$$\gamma = \frac{R_m}{T} = 353$$

$$\beta = (0.875) \frac{r_o}{R_m} = 0.033$$

Stress Concentration due to:

a) membrane load, $K_n = 1.0$
 b) bending load, $K_b = 1.0$

*NOTE: Enter all force values in accordance with sign convention

CYLINDRICAL SHELL

From Fig.	Read curves for	Compute absolute values of stress and enter result	STRESSES - if load is opposite that shown, reverse signs shown							
			Au	AL	Bu	BL	Cu	CL	Du	DL
3C	$\frac{N\phi}{P/R_m} = 64$	$K_n \left(\frac{N\phi}{P/R_m} \right) \cdot \frac{P}{R_m T} = 360$	-360	-	-	-	-	-	-	-
1C	$\frac{M\phi}{P} = 24$	$K_b \left(\frac{M\phi}{P} \right) \cdot \frac{6P}{T^2} = 2866$	-2870	-	-	-	-	-	-	-
3A	$\frac{N\phi}{M_c/R_m^2\beta} =$	$K_n \left(\frac{N\phi}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T} =$								
1A	$\frac{M\phi}{M_c/R_m\beta} =$	$K_b \left(\frac{M\phi}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2} =$								
3B	$\frac{N\phi}{M_L/R_m^2\beta} =$	$K_n \left(\frac{N\phi}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T} =$								
1B or 1B-1	$\frac{M\phi}{M_L/R_m\beta} =$	$K_b \left(\frac{M\phi}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2} =$								
Add algebraically for summation of ϕ stresses, $\sigma_\phi =$			3230							
4C	$\frac{N_x}{P/R_m} = 60$	$K_n \left(\frac{N_x}{P/R_m} \right) \cdot \frac{P}{R_m T} = 338$	-340	-	-	-	-	-	-	-
2C	$\frac{M_x}{P} = 18$	$K_b \left(\frac{M_x}{P} \right) \cdot \frac{6P}{T^2} = 2149$	-2150	-	-	-	-	-	-	-
4A	$\frac{N_x}{M_c/R_m^2\beta} =$	$K_n \left(\frac{N_x}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T} =$								
2A	$\frac{M_x}{M_c/R_m\beta} =$	$K_b \left(\frac{M_x}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2} =$								
4B	$\frac{N_x}{M_L/R_m^2\beta} =$	$K_n \left(\frac{N_x}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T} =$								
2B or 2B-1	$\frac{M_x}{M_L/R_m\beta} =$	$K_b \left(\frac{M_x}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2} =$								
Add algebraically for summation of X stresses, $\sigma_x =$			2490							
Shear stress due to Torsion, M_T										
$\tau_{\phi x} = \tau_{x\phi} = \frac{M_T}{2\pi r_o^2 T}$										
Shear stress due to load, V_c										
$\tau_{x\phi} = \frac{V_c}{\pi r_o T}$										
Shear stress due to load, V_L										
$\tau_{x\phi} = \frac{V_L}{\pi r_o T}$										
Add Algebraically for summation of shear stresses, $\tau =$										
COMBINED STRESS INTENSITY, S										
1) When σ_ϕ & σ_x have like signs			$S = \frac{1}{2} \left[\sigma_\phi + \sigma_x + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2} \right]$							
2) When $\tau = 0$			$S = \text{largest of } \sigma_\phi, \sigma_x \text{ or } \sigma_\phi - \sigma_x $							
3) When σ_ϕ & σ_x have unlike signs			$S = \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}$							

$N_L/(M_L/R_m^2\beta)$ so determined by (C_L) from Table 8 (see para. 4.3).

4.2.2.5.2: When considering bending moment (M_L) : $\beta = K_L \sqrt{\beta_1 \beta_2}$ where K_L is given in Table 8.

4.3 Calculation of Stresses

4.3.1 STRESSES RESULTING FROM RADIAL LOAD, P.

4.3.1.1 Circumferential Stresses (σ_ϕ):

Step 1. Using the applicable values of β and γ

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from inspection, membrane stresses are OK,
Analyze Welded Connection

.5" Fillet weld around the outside of
pipe

$$\text{load/inch} = \frac{2818}{2\pi(6.38)} = 70.3 \text{ lb.}$$

$$\tau_{\text{weld}} = \frac{70.3}{.707(.5)} = 198 \text{ psi} \quad \text{very low}$$

NOTE: **

Although the stresses in the fillet weld are low, fillet welds are not allowed on the pressure boundary. per Telecon with J. McQuarv on 11-19-76, this weld is being changed to a full penetration weld. Since the fillet weld was adequate, the full pen. weld is OK

Axial Stress in $2\frac{1}{2}$ " Sch 80 Pipe Stunt. ASTM A106
 $A = 2.254 \text{ in}^2$ $I = 1.924 \text{ in}^4$

$$\tau = \frac{P}{A} = \frac{2818}{2.254} = 1250 \text{ psi}$$

check Buckling per AISC* ref[8]

$$r = \sqrt{\frac{I}{A}} = \sqrt{\frac{1.924}{2.254}} = .92 \text{ in}$$

$$k = 1.2 \sim \text{pg 5-138, ref [5]}$$

$$L = 100 \text{ in}$$

$$\frac{kL}{r} = \frac{1.2(100)}{.92} = 130.4$$

$$F_y = 35 \text{ ksi}$$

* Subsection NA of ASME Code does not address column type buckling so guidelines from AISC ref[8] will be followed

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$$C_c = \sqrt{\frac{2\pi^2 E}{F_y}} = \sqrt{\frac{2\pi^2 29 \times 10^3}{35}} = 127$$

$$\frac{KL}{r} > C_c$$

$$\therefore F_a = \frac{12\pi^2 E}{23 \left(\frac{KL}{r}\right)^2} = \text{Allowable Compressive Stress}$$

$$F_a = \frac{12\pi^2 29 \times 10^6}{23 (130.4)^2} = 8782 \text{ psi}$$

$$1250 \text{ psi} < 8782 \text{ psi}$$

$\therefore$ Pipe Strut is Adequate

check Connection on End of Strut. (See fig. 3)

the ear assembly is connected to the pipe with a .38 fillet weld.

$$\tau_{\text{weld}} = \frac{2818}{.07(.38)\pi 2.875} = 1161 \text{ psi}$$

$$\tau_{\text{ALLOW}} \approx .4F_y = .4(35000) 14000 \text{ psi}$$

$\therefore$ Weld is OK

The entire Hanger H39 is Structurally Adequate

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The Remaining attachments (4 of them) are all 12" diameter round attachments. They are located at nodes 2, 30, 35 & 37 (see fig. 1)

ANALYSIS OF ATTACHMENTS AT NODES 2, 30, 35 & 37
FOR OPERATING LOADS

From inspection of the Mechanical loads listed on pg 4, it is obvious that ^{the} attachment at node 35 is the highest loaded attachment. The thermal loading for this node is listed on the bottom of page 4.

FOR NODE 35.

Mechanical Loads.

$P = 5400 \#$
 $M_c = 100143 \text{ "}\#$
 $M_h = 623 \text{ "}\#$
 $M_T = 374 \text{ "}\#$
 $V_c = 3938 \#$
 $V_h = 1266 \#$

Mechanical + Thermal loads

$P = 7627 \#$
 $M_c = 166600 \text{ "}\#$
 $M_h = 13230 \text{ "}\#$
 $M_T = 7947 \text{ "}\#$
 $V_c = 5461 \#$
 $V_h = 1604 \#$

ref [7]

A Bylaard Analysis of the Containment Vessel Shell is performed on page 18. for the mechanical plus thermal displacement loadings.

Table 5—Computation Sheet for Local Stresses in Cylindrical Shells

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ATTACHMENT @ NODE 35

MECHANICAL + THERMAL DISP. LOADS

1. Applied Loads*

Radial load, $P = 7627$ lb.
 Circ. Moment, $M_c = 16660$ in. lb.
 Long. Moments, $M_L = 13230$ in. lb.
 Torsion Moment, $M_T = 1247$ in. lb.
 Shear Load, $V_c = 546$ lb.
 Shear Load, $V_L = 1604$ lb.

2. Geometry

Vessel thickness, $T = 1.19$ in.
 Attachment radius, $r_o = 2.38$ in.
 Vessel radius, $R_m = 420$ in.

3. Geometric Parameters

$$\gamma = \frac{R_m}{T} = 354$$

$$H = (0.875) \frac{r_o}{R_m} = 0.0133$$

Stress Concentration due to:

- a) membrane load, $K_n = 1.0$
 b) bending load, $K_b = 1.0$

*NOTE: Enter all force values in accordance with sign convention

CYLINDRICAL SHELL

From Fig.	Read curves for	Compute absolute values of stress and enter result	STRESSES - if load is opposite that shown, reverse signs shown							
			Au	AL	Bu	BL	Cu	CL	Du	DL
3C	$\frac{N\phi}{P/R_m} = 66$	$K_n \left(\frac{N\phi}{P/R_m} \right) \cdot \frac{P}{R_m T} = 1007$	-1010	-	-	-	-1010	-	-	-
1C	$\frac{M\phi}{P} = .24$	$K_b \left(\frac{M\phi}{P} \right) \cdot \frac{6P}{T^2} = 7756$	-7760	-	-	-	-7760	-	-	+
3A	$\frac{N\phi}{M_c/R_m^2\beta} = 5.6$	$K_n \left(\frac{N\phi}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T} = 334$					-335	-	+	+
1A	$\frac{M\phi}{M_c/R_m\beta} = .103$	$K_b \left(\frac{M\phi}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2} = 13015$					-13020	-	+	-
3B	$\frac{N\phi}{M_L/R_m^2\beta} = 20$	$K_n \left(\frac{N\phi}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T} = 95$	-95	-	-	+				
1B or 1B-1	$\frac{M\phi}{M_L/R_m\beta} = .062$	$K_b \left(\frac{M\phi}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2} = 682$	-680	+	-	-				
Add algebraically for summation of ϕ stresses, $\sigma_\phi =$			9545				22125			
4C	$\frac{N_x}{P/R_m} = 66$	$K_n \left(\frac{N_x}{P/R_m} \right) \cdot \frac{P}{R_m T} = 1007$	-1010	-	-	-	-1010	-	-	-
2C	$\frac{M_x}{P} = .135$	$K_b \left(\frac{M_x}{P} \right) \cdot \frac{6P}{T^2} = 4362$	-4360	+	-	+	-4360	+	-	+
4A	$\frac{N_x}{M_c/R_m^2\beta} = 6.4$	$K_n \left(\frac{N_x}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T} = 382$					-380	-	+	+
2A	$\frac{M_x}{M_c/R_m\beta} = .022$	$K_b \left(\frac{M_x}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2} = 1234$					-1250	+	+	-
4B	$\frac{N_x}{M_L/R_m^2\beta} = 5.6$	$K_n \left(\frac{N_x}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T} = 27$	-30	-	+	+				
2B or 2B-1	$\frac{M_x}{M_L/R_m\beta} = .10$	$K_b \left(\frac{M_x}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2} = 1003$	-1000	+	-	-				
Add algebraically for summation of X stresses, $\sigma_x =$			6400				13580			
Shear stress due to Torsion, M_T				+	-	+	+	+	+	+
Shear stress due to load, V_c				+	-	-				
Shear stress due to load, V_L								+	+	+
Add Algebraically for summation of shear stresses, $\tau =$										
COMBINED STRESS INTENSITY, S										
1) When σ_ϕ & σ_x have like signs		$S = \frac{1}{2} \left[\sigma_\phi + \sigma_x + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2} \right]$								
2) When $\tau = 0$		$S = \text{largest of } \sigma_\phi, \sigma_x \text{ or } \sigma_\phi - \sigma_x $								
3) When σ_ϕ & σ_x have unlike signs		$S = \frac{1}{2} \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}$								

$N_i/(M_L/R_m^2\beta)$ so determined by (C_L) from Table 8 (see para. 4.3).

4.2.2.5.2: When considering bending moment (M_i) : $\beta = K_L \sqrt{\beta_1 \beta_2}$ where K_L is given in Table 8.

4.3 Calculation of Stresses

4.3.1 STRESSES RESULTING FROM RADIAL LOAD, P.

4.3.1.1 Circumferential Stresses (σ_ϕ):

Step 1. Using the applicable values of β and γ

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FROM PG 18, it can be seen that the maximum stress is $\sigma_{\phi}^* = 22125 \text{ psi}$
 $\sigma_x^* = 13580 \text{ psi}$

Adding the pressure stresses from pg 9

$$\sigma_{\phi} = 22125 + 14120 = 36245 \text{ psi}$$

$$\sigma_x = 13580 + 7060 = 20640 \text{ psi}$$

$$\therefore P_L + P_B + Q = 36245 \text{ psi} < 3S_m = 52500 \text{ psi from pg 11}$$

Local Membrane stresses due to Mechanical Loads, from inspection of the membrane stresses on pg 18, it can be seen that the maximum membrane stresses are $\sigma_{\phi}^* = 1345 \text{ psi}$
 $\sigma_x^* = 1390 \text{ psi}$

although these stresses contain contributions due to thermal expansion, it is conservative to use them.

adding pressure stresses

$$\sigma_{\phi} = 1345 + 14120 = 15465 \text{ psi}$$

$$\sigma_x = 1390 + 7060 = 8450 \text{ psi}$$

$$\therefore P_L = 15465 \text{ psi} < 1.5S_m = 34950 \text{ psi}$$

The attachments to the containment vessel at nodes 2, 35, 37 & 30 do not overstress the containment vessel. The welds are full penetration groove welds which do not need analyzing.

The remainder of the attachment has already been analyzed in LTR 115.13, ref [5].

* See note on top of pg 20.

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* Note : The stresses found in the Bylaard Analysis are not exact because the attachments at nodes 2, 35, 37 & 30 do not join the vessel shell at 90° as can be seen from Figure 4. As can be seen from pg 16, the circumferential moment, M_c , makes the largest contribution to the maximum stress. The M_c value used is correct and only M_L & M_T would need to be transformed to a new coordinate system which would be normal to the vessel shell. This transformation was not performed because the results would only vary slightly from those obtained on pg 16.

LOCAL STRESSES DUE TO THERMAL HEATUP & GRADIENTS.

The local stresses due to a radial gradient and a difference in average temperature between the attachment and the vessel, have not been considered. These stresses will be calculated and if they are significant, they will be added to the stresses previously calculated on pages 1-17.

Per N.E. PACE, Thermal Analysis Branch Supervisor, a conservative estimate of the maximum difference in average temperature between the attachment and the containment shell is 30-40°F. He also says that the thermal gradient thru the vessel shell is negligible. (1-2°F)

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The ASME piping code gives a conservative way to calculate thermal stresses where two materials of different temperatures join.

from subsection NB of the ASME Code ref [5]
pg 136, Eq. 10.

Stress intensity due to temperature

$$S = \frac{1}{2(1-\nu)} E \Delta T_i + C_3 E_{ab} [\alpha_a T_a - \alpha_b T_b]$$

↗ This is equivalent to $P_L + P_B + Q$

where ΔT_i = temp difference between
inside & outside of shell

$T_{a(b)}$ = range of average temperature
side a(b) of the discontinuity

$$\begin{aligned} E_{\text{CONT VESSEL}} &= 29.9 \times 10^6 \text{ psi} \\ E_{\text{PIPE ATTACH}} &= 29.9 \times 10^6 \text{ psi} \end{aligned} \left. \vphantom{\begin{aligned} E_{\text{CONT VESSEL}} &= 29.9 \times 10^6 \text{ psi} \\ E_{\text{PIPE ATTACH}} &= 29.9 \times 10^6 \text{ psi} \end{aligned}} \right\} \text{Table I-6.0 ref [5]}$$

$$\begin{aligned} \alpha_{\text{CONT VESSEL}} &= 6.07 \times 10^{-6} \text{ in/in/}^\circ\text{F} \\ \alpha_{\text{PIPE ATTACH}} &= 6.07 \times 10^{-6} \text{ in/in/}^\circ\text{F} \end{aligned} \left. \vphantom{\begin{aligned} \alpha_{\text{CONT VESSEL}} &= 6.07 \times 10^{-6} \text{ in/in/}^\circ\text{F} \\ \alpha_{\text{PIPE ATTACH}} &= 6.07 \times 10^{-6} \text{ in/in/}^\circ\text{F} \end{aligned}} \right\} \text{Tbl I-5.0 ref [5]}$$

$$C_3 = 1.0 \text{ from Tbl NB 3683.2-1, ref [5]}$$

$$\nu = .3$$

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$$S = \frac{1}{2(1-0.3)} (29.9 \times 10^6) (6.07 \times 10^{-6}) (2^\circ\text{F}) + (1.0) (29.9 \times 10^6) (6.07 \times 10^{-6}) (40^\circ\text{F})$$

$$= 251 + 7259 = 7520 \text{ psi}$$

$P_L + P_B + Q = 7520$ ~ This stress should have been added to the $P_L + P_B + Q$ stresses found on pages 11, P 13 & P 19. It can be seen that when this is added to the stresses found on those pages that the stress level is well below allowable with the maximum

$$P_L + P_B + Q = 36245 + 7520 = 43760 < 3S_m = 52500 \text{ psi}$$

∴ All of the attachments to the containment vessel wall for the Main Steam and Feedwater Snubber Support Structure are Structurally Adequate and do not overstress the containment vessel wall for LOCE or SSE and thermal load cases. (operational loading)

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STRESS ANALYSIS FOR FAULTED CONDITIONS

For faulted loading conditions only primary stresses need to be considered, App. F, ref 2.

The LOCA + SSE LOADS ARE given on the bottom of page 4. The same methods used in the analysis for operating conditions will be used to determine the primary stresses due to the faulted loading. The attachments will be analyzed in the same order as they were in the previous section.

ANALYSIS OF ATTACHMENTS 1, 2 & 3, FAULTED LOADING

LOCA + SSE LOADS FROM PG 4

$$P_1 = P_{40} = 8250$$

Attachment

node 49

{ SEE PG. 6 }

$$P(2) = 0$$

$$P^{(3)} = P_{48} = 8611.16.$$

$$M_c^{(1)} = M_c^{(2)} = M_c^{(3)} = 0$$

$$M_L^{(1)} = \frac{M_{L48} + M_{L49}}{3} = 87.400 \text{ "lb} = M_L^{(2)} = M_L^{(3)}$$

$$M_T^{(1)} = \frac{M_{T48} + M_{T49}}{3} = 61.5 \text{ lb} = M_T^{(2)} = M_T^{(3)}$$

$$V_{C(1,2,3)} = 0$$

$$V_L^{(1)} = \frac{V_{L48} + V_{L49}}{3} = 221416 = V_L^{(2)} = V_L^{(3)}$$

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FAULTED LOADS ON ATTACHMENT.

①

FAULTED LOADS ON ATTACHMENT

②

FAULTED LOADS ON ATTACHMENT

3

* Note: Stresses due to thermal gradients do not need to be considered for faulted conditions

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FROM PG 4 THE THERMAL LOADS ARE AS FOLLOWS.*

$$P_{\text{atm}} = P_{\text{Hg}} = 1351 \text{ lb}$$

$$P_2 = 0$$

$$p^{(3)} = P_{4g} = 423 \text{ lb.}$$

$$M_c^{(1)} = M_c^{(2)} = M_c^{(3)} = 0$$

$$M_L^{(1)} = \frac{M_{L_{CB}} + M_{L_{49}}}{3} = M_L^{(2)} = M_L^{(3)} = \frac{25566 + 8701}{3} = 11420.67$$

$$M_T^{(1)} = \frac{M_{T48} + M_{T49}}{3} = M_T^{(2)} = M_T^{(3)} = \frac{272 + 2897}{3} = 1056 \text{ " \#}$$

$$V_c^{(1)} = V_c^{(2)} = V_c^{(3)} = 0$$

$$V_L^{(1)} = \frac{V_{L48} + V_{L49}}{3} = \frac{204 + 748}{3} = 320 \text{ lbs.} = V_L^{(2)} = V_L^{(3)}$$

Thermal Loads At The Containment Shell

ATTACHMENT ①

$$P = 1351$$

$M_1 = 22230 \text{ "}\#$

$$M_T = 1056 \text{ " \#}$$

$V_L = 320 \#$

ATTACHMENT ②

$M_L = 22230''$

$$M_T = 1056$$

$$V_L = 320 \text{ \#}$$

ATTACHMENT (3)

PM 423#

$M_L = 22230'' \#$

$$M_T = 1056 \text{ " \#}$$

$V_1 = 320 \text{ ft}$

COMBINED (LOCA+SSE) + Thermal loads

ATTACHMENT ①

P = 10201#.

$$M_L = 104530'' \#$$

"M_T = 1671" #

$$V_1 = 2534 \text{ \#}$$

ATTACHMENT (2)

$M_1 = 184530 \text{ "H}$

$$M_T = 1.671 \text{ \#}$$

$$V_L = 2534.4$$

ATTACHMENT (3)

$P = 9234 \#$

$M_L = 184530''\#$

$$M_T = 167.1$$

$$V_1 = 2534\#$$

*NOTE: These loads are from the restraint of free end displacements. Stresses caused by thermal gradients thru the shell do not need to be considered for Faulted Cond.

CALCULATION WORK SHEET

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FROM INSPECTION OF THE LOADS ON PG 25
Attachment 1 will have the highest stresses.

$P = 10200 \text{ " \#}$
 $M_L = 184530 \text{ " \#}$
 $M_T = 1671 \text{ " \#}$
 $V_k = 2534 \text{ lb}$

Using the same procedure as used on pg 9 & 10.
Calculating only Membrane Stresses on pg 27.

From Pg. 27, $\sigma_{\phi \text{ MEMB}} = 4376 \text{ psi}$

$$\sigma_{x \text{ MEMB}} = 2120 \text{ psi}$$

from pg. 2, $2.4 S_m = 2.4(17500) = 42000 \leftarrow \text{governors}$
 $.7 S_u = .7(70000) = 49,000$

$$P_m + P_c + P_B = 4376 + \underbrace{14120}_{\text{pressure stresses from pg 8}} = 18500 < 42000$$

∴ The Shell is OK for Faulted loading

ANALYZE THE ATTACHMENT PIPES 6" Sch. 80

$$\sigma_{\text{pipe}} = \frac{P}{A} \pm \frac{M}{S} = \frac{10200}{8.405} \pm \frac{184530}{12.22} = 16314 \text{ psi}$$

$$\tau = \frac{T_c}{f} + \frac{2V}{A} = \frac{1671(3.31)}{2(40.5)} + \frac{2(2534)}{8.408} = 671 \text{ psi}$$

$$S.F = \frac{16314}{2} \pm \sqrt{\left(\frac{16314}{2}\right)^2 + 671^2}$$

$$= \underline{16355 \text{ psi}}$$

Table 5—Computation Sheet for Local Stresses in Cylindrical Shells

Attachment 1
FAULTED, LOADING

27/

1. Applied Loads

Radial load, $P = 10200$ lb.
Circ. Moment, $M_c = 124530$ in. lb.
Long. Moment, $M_L = 1167$ in. lb.
Torsion Moment, $M_T = 1167$ in. lb.
Shear Load, $V_c = 1167$ lb.
Shear Load, $V_L = 2534$ lb.

2. Geometric Parameters

$\gamma = \frac{R_m}{T} = 353$
 $H = (0.875) \frac{r_o}{R_m} = 0.0069$

2. Geometry

Vessel thickness, $T = 1.19$ in.
Attachment radius, $r_o = 3.31$ in.
Vessel radius, $R_m = 122$ in.

Stress Concentration due to:
a) membrane load, $K_n = 1.0$
b) bending load, $K_b = 1.0$

*NOTE: Enter all force values in accordance with sign convention

CYLINDRICAL SHELL

From Fig.	Read curves for	Compute absolute values of stress and enter result	STRESSES — if load is opposite that shown, reverse signs shown							
			Au	AL	Bu	BL	Cu	CL	Du	DL
3C	$\frac{N\phi}{P/R_m}$ 64	$K_n \left(\frac{N\phi}{P/R_m} \right) \cdot \frac{P}{R_m T}$	-1306	-1306	-1306	-1306	-1306	-1306	-1306	-1306
1C	$\frac{M\phi}{P}$ 24	$K_b \left(\frac{M\phi}{P} \right) \cdot \frac{6P}{T^2}$	-	+	-	+	-	+	-	+
3A	$\frac{M_c}{M_c/R_m^2 \beta}$	$K_n \left(\frac{M_c}{M_c/R_m^2 \beta} \right) \cdot \frac{M_c}{R_m^2 \beta T}$								
1A	$\frac{M_c}{M_c/R_m \beta}$	$K_b \left(\frac{M_c}{M_c/R_m \beta} \right) \cdot \frac{6M_c}{R_m \beta T^2}$								
3B	$\frac{N\phi}{M_L/R_m^2 \beta}$ 24	$K_n \left(\frac{N\phi}{M_L/R_m^2 \beta} \right) \cdot \frac{M_L}{R_m^2 \beta T}$	-3070	-3070	+3070	+3070				
1B or 1B-1	$\frac{M\phi}{M_L/R_m \beta}$ 24	$K_b \left(\frac{M\phi}{M_L/R_m \beta} \right) \cdot \frac{6M_L}{R_m \beta T^2}$	-	+	+	-				
Add algebraically for summation of ϕ stresses, σ_ϕ			$\sigma_\phi \text{ MEMB } 4376$							
4C	$\frac{N_x}{P/r_o}$ 60	$K_n \left(\frac{N_x}{P/r_o} \right) \cdot \frac{P}{R_m T}$	-1228	-1230	-1230	-1230	-1230	-1230	-1230	-1230
2C	$\frac{M_x}{P}$ 18	$K_b \left(\frac{M_x}{P} \right) \cdot \frac{6P}{T^2}$	-	+	-	+	-	+	-	+
4A	$\frac{M_x}{M_c/R_m^2 \beta}$	$K_n \left(\frac{M_x}{M_c/R_m^2 \beta} \right) \cdot \frac{M_c}{R_m^2 \beta T}$								
2A	$\frac{M_x}{M_c/R_m \beta}$	$K_b \left(\frac{M_x}{M_c/R_m \beta} \right) \cdot \frac{6M_c}{R_m \beta T^2}$								
4B	$\frac{N_x}{M_L/R_m^2 \beta}$ 7.0	$K_n \left(\frac{N_x}{M_L/R_m^2 \beta} \right) \cdot \frac{M_L}{R_m^2 \beta T}$	-850	-	+	+				
2B or 2B-1	$\frac{M_x}{M_L/R_m \beta}$ 11	$K_b \left(\frac{M_x}{M_L/R_m \beta} \right) \cdot \frac{6M_L}{R_m \beta T^2}$	-	+	+	-				
Add algebraically for summation of x stresses, σ_x			$\sigma_x \text{ MEMB } 2120$							
Shear stress due to Torsion, M_T		$\tau_{\phi x} = \tau_{x\phi} = \frac{M_T}{2\pi r_o^2 T}$ 15.6	+	+	+	+	+	+	+	+
Shear stress due to load, V_c		$\tau_{\phi\phi} = \frac{V_c}{\pi r_o T}$	+	+	-	-				
Shear stress due to load, V_L		$\tau_{xx} = \frac{V_L}{\pi r_o T}$					-	-	+	+
Add Algebraically for summation of shear stresses, τ										
COMBINED STRESS INTENSITY, S										
1) When σ_ϕ & σ_x have like signs			$S = \frac{1}{2} \left[\sigma_\phi + \sigma_x + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2} \right]$							
2) When $\tau = 0$			$S = \text{largest of } \sigma_\phi, \sigma_x \text{ or } \sigma_\phi - \sigma_x $							
3) When σ_ϕ & σ_x have unlike signs			$S = \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}$							

$N/(M_L/R_m^2 \beta)$ so determined by (C_L) from Table 8 (see para. 4.3).

4.2.2.5.2: When considering bending moment (M_L) : $\beta = K_L \sqrt{\beta_1 \beta_2}$ where K_L is given in Table 8.

4.3 Calculation of Stresses

4.3.1 STRESSES RESULTING FROM RADIAL LOAD, P .

4.3.1.1 Circumferential Stresses (σ_ϕ):

Step 1. Using the applicable values of β and γ

See pg 12

∴ Attachments ① ② & ③, see sketch 1, pgs, are structurally adequate for faulted loading

From ref [5], the faulted loading on this hanger is 1992 lb. This is less than the operating load of 2818 lb analyzed for on pg 13, so this hanger is structurally adequate by inspection.

From inspection of the Local + SSE loads on pg 4, it is obvious that the Attachment at node 37 is the highest loaded.

	LOCA+SSE	THERMAL	LOCA+SSE+THERMAL
P	8180	-	8180 lb
M _C	3608	-	3608 "lb
M _L	116350	-	116350 "lb
M _T	81500	-	81500 "lb
V _C	6770	-	6770 lb
V _L	2495	-	2495 lb

CALCULATION WORK SHEET

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A Bylaard analysis will be performed on the containment vessel shell, like was done on page 18, but only membrane stresses will be calculated since they are the only stresses that need to be considered for faulted loading.

Parameters used on pg. 30 are from pg 18.

Stresses from pg 30.

$$\tau_{\phi \text{ MEMB}} = 2473 \text{ psi}$$

$$\tau_{x \text{ MEMB}} = 1347 \text{ psi}$$

$$p_{y,\phi} = 550 \text{ psi}$$

↗ The above stresses are very low
∴ The Attachments at N0PES 2, 30, 35, & 37 are OK.

The remainder of these attachments have been analyzed in LTR 115-13, sup. A, ref 5.

∴ All of the Containment Vessel Attachments are ok
For faulted loading

Table 5—Computation Sheet for Local Stresses in Cylindrical Shells

1. Applied Loads

Radial load, P 8100 lb.
Circ. Moment, M_c 3600 in. lb.
Long. Moments, M_L 16350 in. lb.
Torsion Moment, M_T 81500 in. lb.
Shear Load, V_c 5770 lb.
Shear Load, V_L 2435 lb.

2. Geometry

Vessel thickness, T 1.19 in.
Attachment radius, r_a 6.38 in.
Vessel radius, R_m 420 in.

3. Geometric Parameters

γ $\frac{R_m}{T}$ 354
 H $(0.075) \frac{r_a}{R_m}$ 0.0133

Stress Concentration due to:
a) membrane load, K_n 1.0
b) bending load, K_b 1.0
NOTE: Enter all force values in accordance with sign convention

CYLINDRICAL SHELL

From Fig.	Read curves for	Compute absolute values of stress and enter result.	STRESSES — if load is opposite that shown, reverse signs shown							
			A _v	A _L	B _u	B _L	C _u	C _L	D _u	D _L
3C	$\frac{N\phi}{P/R_m}$ 66	$K_n \left(\frac{N\phi}{P/R_m} \right) \cdot \frac{P}{R_m T}$	-1083	-	-	-	-1083	-	-	-
1C	$\frac{M\phi}{P}$ 24	$K_b \left(\frac{M\phi}{P} \right) \cdot \frac{6P}{T^2}$	-	+	-	+	-	+	-	+
3A	$\frac{N\phi}{M_c/R_m^2\beta}$ 5.6	$K_n \left(\frac{N\phi}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T}$					-7	-	+	+
1A	$\frac{M\phi}{M_c/R_m\beta}$ 103	$K_b \left(\frac{M\phi}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2}$					-	+	+	-
3B	$\frac{N\phi}{M_L/R_m^2\beta}$ 20	$K_n \left(\frac{N\phi}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T}$	-1330	-	+	+				
1B or 1B-1	$\frac{M\phi}{M_L/R_m\beta}$ 0.68	$K_b \left(\frac{M\phi}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2}$	-	+	+	-				
Add algebraically for summation of ϕ stresses, σ_ϕ			σ_ϕ MEMB. -2473							
4C	$\frac{N_x}{P/r_m}$ 66	$K_n \left(\frac{N_x}{P/r_m} \right) \cdot \frac{P}{R_m T}$	-1083	-	-	-	-1083	-	-	-
2C	$\frac{M_x}{P}$ 135	$K_b \left(\frac{M_x}{P} \right) \cdot \frac{6P}{T^2}$	-	+	-	+	-	+	-	+
4A	$\frac{N_x}{M_c/R_m^2\beta}$ 6.4	$K_n \left(\frac{N_x}{M_c/R_m^2\beta} \right) \cdot \frac{M_c}{R_m^2\beta T}$					-8	-	+	+
2A	$\frac{M_x}{M_c/R_m\beta}$ 0.62	$K_b \left(\frac{M_x}{M_c/R_m\beta} \right) \cdot \frac{6M_c}{R_m\beta T^2}$					-	+	+	-
4B	$\frac{N_x}{M_L/R_m^2\beta}$ 5.65	$K_n \left(\frac{N_x}{M_L/R_m^2\beta} \right) \cdot \frac{M_L}{R_m^2\beta T}$	-264	-	+	+				
2B or 2B-1	$\frac{M_x}{M_L/R_m\beta}$ 10	$K_b \left(\frac{M_x}{M_L/R_m\beta} \right) \cdot \frac{6M_L}{R_m\beta T^2}$	-	+	+	-				
Add algebraically for summation of x stresses, σ_x			σ_x MEMB. 1347							
Shear stress due to Torsion, M_T		$\tau_{\phi x} = \tau_{x\phi} = \frac{M_T}{2\pi r_o^2 T}$	+267	+	+	+	+	+	+	+
Shear stress due to load, V_c		$\tau_{\phi\phi} = \frac{V_c}{\pi r_o T}$	+283	+	-	-				
Shear stress due to load, V_L		$\tau_{x\phi} = \frac{V_L}{\pi r_o T}$					-	-	+	+
Add Algebraically for summation of shear stresses, τ			550							
COMBINED STRESS INTENSITY, S										
1) When σ_ϕ & σ_x have like signs			$S = \frac{1}{2} [\sigma_\phi + \sigma_x + \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}]$							
2) When $\tau = 0$			$S = \text{largest of } \sigma_\phi, \sigma_x \text{ or } \sigma_\phi - \sigma_x $							
3) When σ_ϕ & σ_x have unlike signs			$S = \sqrt{(\sigma_\phi - \sigma_x)^2 + 4\tau^2}$							

$N_x/(M_L/R_m^2\beta)$ so determined by (C_L) from Table 8 (see para. 4.3).

4.2.2.5.2: When considering bending moment (M_L) : $\beta = K_L \sqrt{\beta_1 \beta_2}$ where K_L is given in Table 8.

4.3 Calculation of Stresses

4.3.1 STRESSES RESULTING FROM RADIAL LOAD, P .

4.3.1.1 Circumferential Stresses (σ_ϕ):

Step 1. Using the applicable values of β and γ