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APPLIED MECHANICS BRANCH

LOFT REACTOR VESSEL 290° DOWNCOMER STALK
INSTRUMENT PENETRATION FLANGE STRESS ANALYSIS

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DEPARTMENT OF ENERGY

IDAHO OPERATIONS OFFICE UNDER CONTRACT EY-76-C-07-1570

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LOFT TECHNICAL REPORT LTR 1119-39

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LOFT REACTOR VESSEL 290° DOWNCOMER STALK
INSTRUMENT PENETRATION FLANGE STRESS ANALYSIS

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PSE	FE&OB Mgr.	
RSB <i>ORM</i>	PP&CSB Mgr.	ABSTRACT

The LOFT Reactor Vessel 290° Downcomer Stalk Instrument Penetration Flange Stress Analysis has been completed using normal operational and blowdown loading. A linear elastic analysis was completed using simplified hand analysis techniques. The analysis was in accordance with the 1977 ASME Boiler and Pressure Vessel Code, Section III, for a Class 1 component. Loading included internal pressure, bolt preload, and thermal gradients due to normal operating and blowdown (LOCE).

I certify that this stress report (LTR 1119-39, dated June 6, 1978) demonstrates that the applicable stress limitations are satisfied when the flange is subjected to the loading conditions specified in the Design Specification. This applies to the requirements of NCA 3350 of Section III of the ASME Code.

Signed D. P. Finicle
D. P. Finicle 6-6-78

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LOFT REACTOR VESSEL 290⁰ DOWNCOMER STALK INSTRUMENT PENETRATION FLANGE STRESS ANALYSIS

I. SUMMARY

A linear elastic static analysis of the Instrument Penetration Flange has been completed. The flange was analyzed for 1000 heatup-cooldown cycles and 260 LOCE cycles. The flange was found to be structurally adequate for the specified loadings. Corrosion effects were not considered in this analysis.

II. INTRODUCTION

The LOFT Reactor Vessel 290⁰ Downcomer Stalk Instrument Penetration Flange became necessary to prevent leakage of reactor coolant. The flange replaces the Reactor Vessel Nozzle Capping Assembly after a short period of time. The design life of the flange is 1000 heatup-cooldown cycles and 260 LOCE cycles.

III. DESCRIPTION OF SYSTEM

The LOFT Reactor Vessel 290⁰ Downcomer Stalk Instrument Penetration Flange is a three-inch thick plate which is bolted to the 290⁰ downcomer stalk by six bolts. The flange is penetrated by three conax fittings, which allow the instrumentation leads to pass through, and by three ΔP tubes. The entire flange is fabricated from Inconel 600. The flange is sealed by two K-seals located inside the bolt circle. The mating flange is a raised-face type with an outer bearing diameter of 6 inches. The purpose of the flange is to prevent any leakage of reactor coolant.

IV. LOADING

The Instrument Penetration Flange is loaded by internal pressure, bolt preloads, and thermal gradients through the thickness. Internal pressure and

design cycles were obtained from the design specification^[8]. Bolt preload values were obtained from M. Hoover of LOFT, and thermal gradients were obtained from LTR 1119-37^[7]. The maximum operating pressure is 2250 psi with a design value of 2500 psi. The design operating temperature is 650°F. The maximum bolt preload is 17,300 lb/bolt and the maximum thermal gradient is 217°F through the thickness.

V. ANALYSIS

The structural analysis consisted of hand analysis techniques to demonstrate the structural adequacy of the instrument penetration flange. Stress levels and rules for analysis from the ASME Code^[1] were followed. The stresses in the flange (ignoring any effects of holes) were calculated for the pressure, preload, and thermal loadings. Then it was demonstrated that adequate hole reinforcement was provided per Ref. 1. Also, it was shown that the exemption from fatigue analysis rules in NB-3222.4 (Ref. 1) were met. Since the holes were adequately reinforced and fatigue did not need to be evaluated, the peak stresses in the flange around the holes were not evaluated. This procedure is in accordance with NB-3331^[1]. The stresses that were calculated for a solid plate were then compared with limits given in Ref. 1. In calculating the stresses in the flange, thick shell equations were used for pressure loading. The stresses were compared with results obtained using thin shell theory. The thick shell theory gave results that were five percent higher than thin shell theory. Since the results were relatively close, thin shell theory was used to determine the stresses in the flange due to bolt preload and temperature gradients through the thickness. Formulas based on thin shell theory are more readily available for bolt preload and temperature gradient loading.

VI. RESULTS

Maximum stress in the flange due to internal pressure is 10,930 psi; due to temperature gradients through the thickness is 23,660 psi; and due to bolt preload is 7150 psi. The stresses due to internal pressure are primary while the stresses due to preload and temperature gradients are secondary.

VII. CONCLUSIONS

A static analysis has been completed of the LOFT Reactor Vessel 290⁰ Downcomer Stalk Instrumentation Penetration Flange. The results show that the design configuration, shown in Figures 1, 2, 3, 4, and drawing 208572, Ref. 9, is adequate to withstand the specified loading.

VIII. REFERENCES

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2. Raymond J. Roark, Formulas for Stress and Strain, McGraw-Hill Book Company, 4th Edition, 1965.
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5. Timoshenko and Goodier, Theory of Elasticity, McGraw-Hill Book Company, Third Edition, 1970.
6. R. E. Peterson, Stress Concentration Design Factors, John Wiley & Sons, Inc., 1966.
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8. EG&G Specification ES 60246, "LOFT Reactor Vessel 290⁰ Downcomer Stalk Instrument Penetration Flange" to be released.
9. EG&G Drawing, 208572, LOFT Reactor Vessel 290⁰ Downcomer Stalk Instrument Penetration Flange Assembly, February 27, 1978.

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FIGURE 1

INSTRUMENT PENETRATION FLANGE

~PLAN VIEW~
INSTRUMENT PENETRATION FLANGE

FIGURE 3

~ SWAGELOCK ATTACHMENT DETAILS
FOR AP TUBES ~

FIGURE 4

APPENDIX A
HAND CALCULATIONS

CALCULATION WORK SHEETPage A-1 of PagesSubject REACTOR VESSEL NOZZEL FLANGE ASSEMBLY Date Prepared By D. P. FINICLE Checked Work Request

The reactor vessel flange Assembly is shown in figure 1. The flange is 3" thick and sees internal pressure and some thermal loading. ^{± bolt preload.} It appears that the flange is structurally adequate, so simplified conservative analysis techniques will be used to verify its structural adequacy.

The outer ring of the flange assembly is identical to the flange analyzed in LTR 1119-38, ref 3. The loading considered in ref 3 is almost identical to that in this analysis.

To determine the stresses due to internal pressure, Thick plate equations will be used.

Design Conditions

 $p = 2500 \text{ psi}$ $T = 650^\circ \text{F}$

Bolt Torque = 265 ft-lb.

Heatup-Cooldown Cycles 1000

LOCE - 260 cycles

Max. $\Delta T = 217^\circ \text{F}$ for LOCE from ref 7

} Design Spec. ref 8

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DETERMINE STRESSES DUE TO PRESSURE

From Timoshenko's Theory of Elasticity, ref 5, pg 385.

$$\sigma_r = q \left[\frac{2+\nu}{8} \frac{z^3}{c^3} - \frac{3}{8} \frac{2+\nu}{5} \frac{z}{c} + \frac{3(3+\nu)}{32} \frac{a^2 z}{c^3} \right]$$

@ r=0

$$\sigma_z = q \left(-\frac{z^3}{4c^3} + \frac{3}{4} \frac{z}{c} - \frac{1}{2} \right)$$

$$\tau_{rz} = -\frac{3q\nu}{8c^3} (c^2 - z^2)$$

$a = 4.6'' \sim$ radius of the bolt circle $q = 2500$ psi
 $c = 2.15/2 = 1.08''$ } See fig. 2
 $\nu = .31$

@ r=0
 at the center of the plate (z=c)

$$\sigma_r = 2500 \left[\frac{2+.31}{8} - \frac{3}{8} \frac{2+.31}{5} + \frac{3(3+.31)}{32} \left(\frac{4^2 \cdot 1.08}{1.08^3} \right) \right] = 10930 \text{ psi}$$

$$\sigma_z = 2500 \left(-\frac{1}{4} + \frac{3}{4} - \frac{1}{2} \right) = 0$$

$$\tau_{rz} = 0$$

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@ $r \approx 1'$ (where ^{LARGE} holes are) see Fig 3.
 $\bar{z} = 0$

$$\sigma_r = q \left[\frac{2+\nu}{8} \cdot \frac{r^3}{c^3} - \frac{3(3+\nu)}{32} \cdot \frac{r^2 z}{c^3} - \frac{3}{8} \cdot \frac{2+\nu}{5} \cdot \frac{z}{c} + \frac{3(3+\nu)}{32} \cdot \frac{a^2 z}{c^3} \right]$$

$$\tau_r = 2500 \left[\frac{2+.31}{8} - \frac{3(3+.31)}{32} \frac{(1.08)}{1.08^3} - \frac{3}{8} \frac{2+.31}{5} + \frac{3(3+.31)}{32} \frac{4^2 1.08}{1.08^3} \right]$$

$$\tau_r = 10265 \text{ psi}$$

$$\sigma_7 = 0$$

$$\zeta_{r2} = 0$$

① $r \approx 1''$
 $z = 0$

$$D_T = 0$$

$$\sigma_2 = 2500 \left(-\frac{1}{2} \right) = \underline{\underline{-1250 \text{ psi}}}$$

$$\sigma_{rz} = \frac{-3(2500)1}{8(1.08)^3} (1.08^2) = \underline{\underline{870 \text{ psi}}}$$

NOTE: T_R is tension on the top of the flange
compression on the bottom (inside)

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COMPARING THESE RESULTS WITH THIN PLATE THEORY

from pg 387, ref 5

@ the center of the plate i.e. @ $z=0$

$$\sigma_r = \frac{3(3+\nu)}{32} \frac{\omega^2 r^2}{c^3} \eta$$

$$F_r = \frac{3(3 + .31)}{32} \frac{4^2(1.08)}{1.08^3} 2500 = 10640 \text{ psi}$$

hooking @ Pg 2, $\sigma_r = 10930$.

Therefore thick plate theory gives a maximum stress that is 5% larger than thin plate theory.

DETERMINE STRESSES DUE TO THERMAL GRADIENTS

The worst case for thermal stresses would be if the plate edges were fixed. This fixity is almost impossible to achieve but it will produce conservative results. This fixity will be used for determining the stresses in the center of the plate but not at the edges. At the edges it is believed that the fixity is not adequate to generate much of a moment. The maximum $\Delta T = 216.6^\circ F$, from LTR 1119-37, ref 7.

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from Raint, ref 4, case 15, pg 366
the mark on the center of the
plate is

$$M_c = -YD \frac{(1+\nu)^2}{2(1-\nu^2)} \Delta T \cdot (a^2 - r_o^2)$$

$$r_0 = 0$$

$$\gamma = 7.45 \times 10^{-6} \text{ m/m}$$

$\Delta T = 250^\circ \text{F}$

$$E = 28.9 \times 10^6 \text{ psi}$$

4 = 31

3.0"

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{28.9 \times 10^6 (3)^3}{12(1-.31^2)}$$

$$D = 7.194 \times 10^7$$

$$M_c = \frac{-(7.95 \times 10^{-6}) 7.194 \times 10^7 (1 + .31)^2 217}{2(3)} = 35496 \text{ ft}$$

$$\sigma_c = \frac{6M}{t^2} = \frac{6(35496)}{(3)^2} = \underline{\underline{-23660 \text{ psi}}}$$

Compression on the outside (hot face)
Tension on the inside (cold face)

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DETERMINE STRESSES DUE TO BOLT PRE-LOAD

Bolt Pre-load & The moment applied to the plate are the same as those used in ref 3.

$$F_{\text{PRELOAD}} = 17305 \text{ \#/BOLT}$$

$$\text{MOMENT ARM} = 1.0''$$

$$\text{MOMENT / CIRC. OF PLATE} = \frac{17305(6)1}{2\pi(3)} = 5508 \text{ ''-lb/''}$$

from pg 365, ref 4

for $r_o = a$

$$M_c = -M_o l_g$$

$$l_g = \frac{1}{2} \left[1 + \nu + (1 - \nu) \left(\frac{r_o}{a} \right)^2 \right]$$

$$l_g = \frac{1}{2} [1.31 + (-.69)] = 1$$

$$\therefore M_c = -M_o$$

$$\sigma_{\text{bending}} = \frac{6M}{t^2} = \frac{6(5508)}{(2.15)^2} = 7150 \text{ p.s.i.}$$

NOTE: Compression on the inside face (bottom)
 Tension on the top face (outside)

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The stresses that have been calculated, do not consider the effects of the holes in the plate. The ASME Code does not require that these stresses be calculated if:

- 1) The Code Stress limits for a Solid Plate can be met
- 2) The Reinforcing Requirements of NB 3300 can be met.

MEETING REINFORCING CRITERIA NB 3300 - ref. 1

From NB 3333

dia. opening $\approx 1''$
dia. head $\approx 10''$

$$\frac{d'ia}{2} = 5''$$

1225 $\therefore$ hole is not to bin

$$A = .5 d t_r$$

d = from NB 3332 = 1"

$t_r =$ From NB 3221.1 & NB 3221.3 = thickness required to meet primary stress limits for a solid plate.

primary
Maximum Stress on Pg 2 is 10930 psi

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bending stresses $\approx \frac{6M}{t^2} = 10930$.

$$t = 2.15''$$

$$\therefore M \approx 8420 \text{ " - \#}$$

Allowable Stress $P_L + P_b \leq 1.5 S_m$

$$\{S_m = 23.3 \text{ ksi}$$

$$1.5 S_m = 34.95 \text{ ksi}$$

↳ from NA for Inconel 600
SB 166.

$$\frac{6M}{t^2} = \frac{6(8420)}{t^2} = 34950$$

$t_{REQ'D} = 1.20''$

$$A = .5 d t_r = .5 (1) 1.20 = .60 \text{ in}^2 \leftarrow \text{Area of Reinforcement Req'd.}$$

This reinforcing area must be provided within the limits of Reinforcement, NB-3334, wet 1.

NB 3334 the larger of a or b below.

a) dia = $\sqrt[3]{}$ actual diameter.

b) $.62/2 + 2.15 = 2.46'' \leftarrow \text{governs}$

The above says that the reinforcement must be within a radius of 2.46" of the ϕ of the hole.

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using a req'd thickness of 1.20" and an actual thickness = 2.15" ~ see fig 2, to get the req'd reinforcement area = .60 in², a thickness on each side of each hole = .32", See sketch below

The reinforcing is more than adequate for the large holes. By inspection, it is adequate for the smaller holes in the center, see figure 3.

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The exemption from fatigue requirements, found in NB 3222.4(d), must be met before the stress effects due to the holes in the flange can be ignored. This requirement is in addition to the reinforcing requirement that was previously shown satisfied.

NB 3222.4(d) ref 1.

$$(1) \quad 3S_m = 3(23.3) = 69.9 \text{ ksi}$$

from fig I-9.2, ref 1

Number of Cycles = 4000 cycles. $\gg 1260$

 $\therefore (1) \text{ is OK.}$

(2) Service Pressure fluctuation per the design spec? pressure fluctuation of 100 psi
10⁵ cycles
 $\frac{1}{3} \times \text{Design Press.} \times \frac{S_a}{S_m}$

$$S_m = 23.3 \text{ ksi @ } 650^\circ\text{F}$$

Significant Pressure Fluctuations

$$= D.P. \times \frac{1}{3} \times \frac{S}{S_m}$$

$$S_n @ 10^6 \text{ Cycles} = 25 \text{ ksi}$$

$$= 2500 \left(\frac{1}{3} \right) \frac{25000}{23300} = 894 \text{ psi}$$

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The pressure fluctuates more than 894 psi only
 $1000 + 260 = 1260$ cycles.

$S_a @ 1260 \text{ cycles} = 100 \text{ ksi}$

$$\therefore \frac{1}{3} \times D.P. \times \frac{S_a}{S_m} = \frac{1}{3} \times 2500 \times \frac{100}{23.3} = 3576 \text{ psi}$$

Max. Service Pressure fluctuation = 2289 psi

$3576 > 2289 \therefore (2) \text{ is OK.}$

(3) TEMPERATURE DIFFERENCE

$$\frac{S_a}{2E_d}$$

$$S_a @ 1000 \text{ cycles} = 110,000 \text{ psi}$$

$$E = 30.5 \times 10^6 \text{ psi @ } 290^\circ \text{ F}$$

$$\alpha = 7.91 \times 10^{-6} \text{ } ^\circ\text{C}^{-1} \text{ } @ 290^\circ\text{F}$$

$$\text{TEMP DIFFERENCE} = \frac{110,000}{2(30.5)(7.91)} = 227^{\circ}\text{F}$$

→ Max. AT thru the thickness during startup & shutdown is 60°F , Per. N.E. FACE

•• (3) is ok

(4) Max Temperature fluctuation during Service
 $\approx 60^\circ\text{F}$

This is not significant $6^\circ < 5.7^\circ$ see pg. 11a.

Cycles is only 1260 cycles

$$\Delta T_{\max \text{ Allow}} = \frac{S_u}{2EN} = 205^\circ\text{F}$$

$$E @ 541^\circ = 29.4 \times 10^6$$

$$\alpha @ 541^\circ = 2.30 \times 10^{-6}$$

$$650 - \frac{217}{2} = 541.5^\circ \text{F}$$

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Sig. temperature fluctuations $\sim \frac{S_a}{2Ed}$

$$\text{Sug. } \Delta T = \frac{28000}{(25.6 \times 8.23)^2} = 57^\circ$$

$$\begin{aligned} S_a &= 28000 \quad @ 10^6 \\ E &= 29.6 \times 10^6 \quad @ 525^\circ F \\ \alpha &= 8.23 \times 10^{-6} \quad @ 525^\circ F \end{aligned}$$

SA needs to be adjusted to account for the diff. in E value between Inconrel and E of the curve.

To account for this, the number of cycles will be modified

$$E @ 650^{\circ}F = 28.9 \times 10^6 \text{ psi}$$

cycles = $1260 \frac{26}{28.9} = 1133 \text{ cycles}$

E on Curves
Fig 1.9.2

$$S_a @ 1133 \text{ cycles} = 108000 \text{ psi}$$

$$\therefore \Delta T = \frac{108,000}{29.4 \times 8.30 \times 2} = 221^\circ \text{F}$$

from LTR 1119-37, ref 7, the max $\Delta T = 217^\circ F$

$217 < 221 \quad \therefore (4) \text{ is not}$

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(5) & (6) Not Applicable Because it is all internal 600 and there are no applied mechanical loads.

∴ The Requirements for Components Not Needing Cyclic Analysis have been Met.

∴ Since Fatigue evaluation is not required & the reinforcing rules of NB 3330 have been met, the general stresses in the plate will be considered and the stresses in the vicinity of the holes will not be considered per NB 3330.

SUMMARY OF STRESSES

The stresses calculated on pgs. 1 thru 6. are classified as follows.

Bending Due to Pressure $\sim P_b$

Bending Due to Thermal Gradient $\sim Q$

Bending Due to Preload. $\sim Q$

$$P_b = 10930 \text{ psi} < 1.5 S_m = 1.5 (23.3) = 34950$$

L pg. 2.

$$Q = 2,3660^* + 7150^* = 30,810 \text{ psi}$$

Lpg 5

2pg 6

* note: these stresses are of opposite sign but have been added for conservatism.

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$$P_L + P_B + Q = 10930 + 30810 = 41740 \text{ psi} < 3S_m = 3(23.3) = 69,900 \text{ psi}$$

$\therefore$ Stress limits of Sec III have been met, ref 1.

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STRESSES IN CONAX FITTINGS (See fig 2)

These fittings have a vendor rating higher than the design loading, therefore stresses in the fittings need not be evaluated. The effects of the holes in the flange have already been evaluated on the previous pages.

STRESSES IN THE WELD ON ΔP TUBES (See fig 1)

The DP tubes are .250" O.D with a wall thickness of .035". Pressure acts over .51"

The Pressure area = $\frac{\pi}{4} (.510)^2 = .20 \text{ in}^2$

The design internal pressure is 2500 psi.

The axial force in the tube = $.20 (2500)$
 $= 511 \text{ \#}$

Stresses in the weld. (Actually the ΔP ^{tubes} are held in a swagelok fitting & this fitting is welded onto the flange) See Fig 2.

The weld is a .09 fillet 0.500" in diameter

$$E_{\text{field}} = \frac{511}{0.7071 (\pi (.500)) \cdot .09} = 5109 \text{ psi}$$

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Allowable Stress in fillet weld

per NB 3356, ref. 1, the allowable stress must be reduced by one-half.

$$P_{m \text{ weld}} = 2\tau = 2(5109) = 10220 \text{ psi}$$

$$S_m = 23,300 \text{ psi} \sim \text{from pg A-10.}$$

$$P_m \leq .5S_m = .5(23300) = 11650 \text{ psi}$$

$$P_m = 10220 < 11650$$

∴ The fillet weld is Adequate