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AEC Research and Development Report
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SM-1 reactor vessel cover and flange stress analysis

Contract No. AT[30-1]-2639
with U. S. Atomic Energy Commission
New York Operations Office

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ALCO PRODUCTS, INC.
NUCLEAR POWER ENGINEERING DEPARTMENT

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SM-1 REACTOR VESSEL COVER AND FLANGE STRESS ANALYSIS

By:
Professor M. F. Sayre, Consultant

Approved By:
M. H. Dixon, Project Engineer

Issued: February 19, 1962

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LIST OF ILLUSTRATIONS

1.0 SUMMARY

The maximum stress calculated for the SM-1 reactor vessel closure studs occurs during operation at full power. This value is 27,180 psi of which 19,800 psi is tension and 7380 psi bending. This stress does not include a stress concentration factor for effect of threads. It was conservatively assumed the studs were initially tightened to a code allowable stress of 20,000 psi as specified in the ASME Code[1] rather than the lesser stress obtained by the normal operating procedure.

The maximum calculated stress occurs at the outside surface of the cover at point C, Fig. 1, where the stress ranges from 318 psi in tension to 99,660 psi in compression. The alternating stress is 50,000 psi. According to the Navy Code[2] for a stress range of 50,000 psi, the cover material can safely undergo a maximum of 1600 cycles.

It was estimated that the SM-1 will go through approximately 900 startup and shutdown cycles during a 20-yr life period,[3] so the calculated stress is regarded as safe.

For a transient condition of 30°F/hr during heat-up, approximate temperature differences between the inside and outside surfaces of the cover were obtained. Temperature differentials between the inside and outside surfaces of the cover are increased by roughly 10% above the steady state condition. More exact calculations of the transient stresses did not appear necessary since they would be not more than 10% greater than the steady state thermal stress.

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2.0 INTRODUCTION

This report presents results of a stress analysis of the cover and flange region of the SM-1 reactor vessel. The work was done under Item 6.9 of the FY'61 Program for Engineering Support and Development of Army PWR Power Plants.*

The vessel was originally designed and built to meet requirements of the ASME Boiler and Pressure Vessel Code, Section VIII. However, the code calculations are primarily designed for sizing the vessel, i.e. selecting adequate wall thicknesses and flange dimensions. They do not tell the designer what the actual stresses in the vessel are, although they result in a safe vessel design.

Recently, it has become necessary to define the vessel stresses in considerable detail. An investigation of SM-1 closure stud failures, Item 1.2 of the FY'61 program plan, required an accurate knowledge of the stud stresses in order to interpret the results of tests that were run to examine whether stress corrosion was the cause of the stud failures. The analysis of the cover-stud-flange region reported here provides the necessary stud stresses.

The severe cyclic operation of the SM-1 plant imposed by the Army operator training program has caused concern that the plant might be subject to strain-cycling fatigue failure. A second purpose of the analysis, then, was to provide the detailed stresses required for a strain-cycling analysis of the vessel cover and flange, usually the areas of highest stress.

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3.0 PRESSURE AND THERMAL STRESSES IN SM-1 REACTOR COVER AND FLANGE

3.1 DESCRIPTION OF COVER AND FLANGE

The SM-1 reactor cover, Fig. 1, has a shallow central dome of 26 in. inside radius and 2-7/8 in. nominal thickness including the stainless steel overlay. Through a transition section, this merges into a flat, on top of which is welded a massive bull ring. This is held down to the upper flange of the reactor vessel by 16 studs of 2-17/32 in. diam reduced section, which screw into tapped holes in the upper vessel flange.

An octagonal section stainless steel gasket is located almost in line with the inner face of the bull ring. There is a nominal clearance of 7/16 in. between cover and flange after the cover has been tightened down.

The upper face of the dome is covered with thermal insulation, which in turn is covered and protected from the shield water by a 1/2 in. thick stainless steel dished head, welded to the bull ring. This dished head forms an important strength element in resisting rotation of the bull ring.

The reactor cover operates immersed in water at a nominal temperature of 125°F. This water penetrates the gap between upper flange and cover as far as the gasket and fills the clearance space around the bolts in the cover. The reactor shell is insulated, and the outer face and part of the upper face of the flange is protected from contact with water by an air gap.

With pressurized water within the reactor at 420°F, the outer face of the gasket and adjacent metal faces become hot enough to cause the unpressurized water outside to boil. This steam moves outward in the gap between cover and flange, and also upward in the gap surrounding the studs and becomes condensed as it meets colder metal or colder water. This boiler-condenser action combines with heat transfer by conduction through the metal and with heat loss by convection to the surrounding water to determine temperature gradients through the flange and the bull ring.

3.2 METHOD OF STRESS COMPUTATION

In computing stresses, the gasket has been assumed to act as a hinge connecting cover and flange, while the cover bull ring and vessel flange act as semi-rigid members rotating about this gasket. The bull ring and flange are sufficiently rigid so that internal bending can be neglected. In order to obtain valid deflection figures, the stainless steel overlay has been included as part of the thickness. Elastic behavior has been assumed.

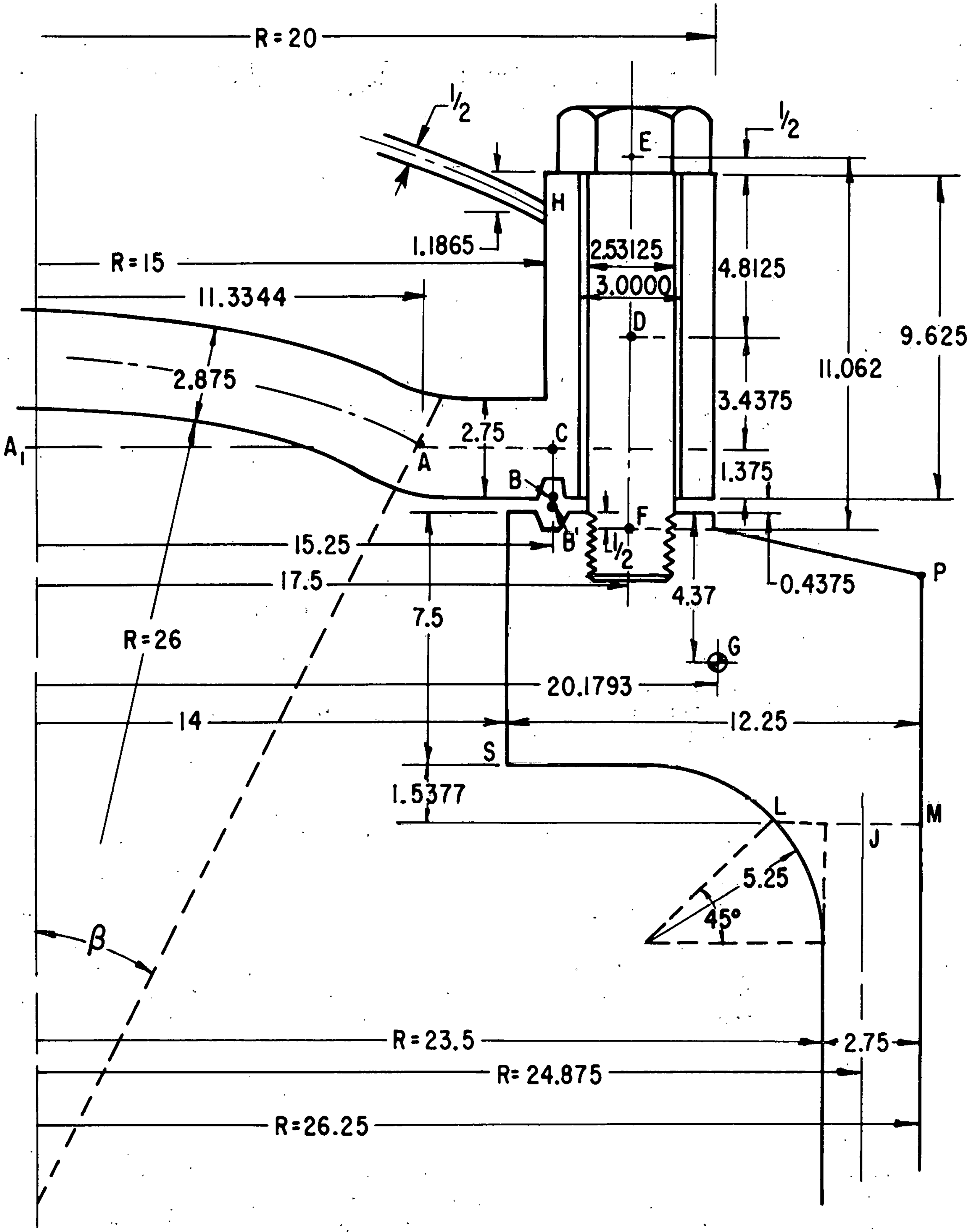

Figure 1. SM-1 Reactor Vessel Cover, Flange and Stud Region

Appendix B to this report contains derivation of equations for deflections and stresses in the cover, the studs, and in the upper flange of the vessel. In these equations,

P_E = stud tension, lb/in. of stud circle circumference,

the tension per bolt = $\dfrac{35\pi P_E}{16}$, and

p = internal fluid pressure, psi.

The coefficient of

$$P_E \frac{35\pi}{16}$$

represents the effect of steady state thermal stresses.

3.3 DISCUSSION AND RESULTS

Heat reaches the bull ring only through a narrow bottle neck, 2-3/4 in. wide, and is lost to the surrounding water over the top, bottom, and outside face of the ring. The temperature calculations are summarized in Fig. 2. For purposes of stress computation, these have been conventionalized into a linear variation from 255°F across the bottom to 133°F at the top, with a mean of 194°F. Similarly, the flange has been conventionalized into a mean of 372.3°F, with a linear variation of 13.35°F per inch.

In line with these temperatures, the free expansion of the centroid of the bull ring is

$$(194-125)\ 6.7 \times 10^{-6} \times 17.5 = 0.00809 \text{ in.}$$

The angle of rotation of the vertical centerline of the bull ring is,

$$\frac{255-133}{9.625}\ 6.7 \times 10^{-6} \times 17.5 = 0.001486 \text{ radians, counterclockwise.}$$

For the flange, the free expansion is,

$$(372.3 - 125)\ 7 \times 10^{-6} \times 20.1793 = 0.03488 \text{ in.}$$

and the angle of rotation is

$$13.35 \times 7 \times 10^{-6} \times 20.1793 = 0.001885 \text{ radians, counterclockwise.}$$

High precision is not claimed for individual temperature values shown in Fig. 2, but due to the averaging effect, large changes would be needed to produce appreciable changes in the computed thermal stresses.

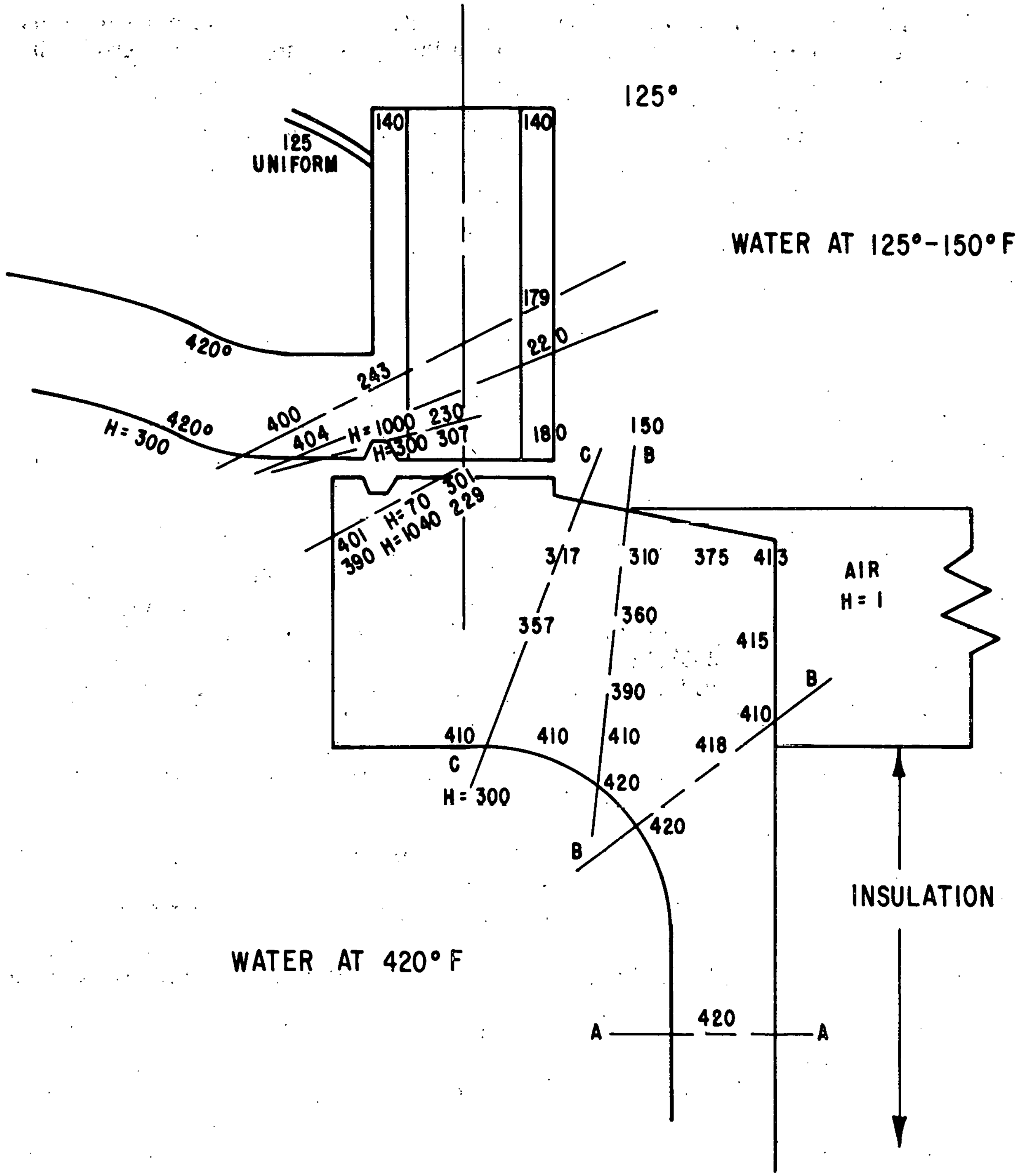

Figure 2. SM-1 Steady State Temperature Distribution - Reactor Vessel Cover and Flange Region

3.3.1 Stresses in Reactor Cover and Bolts

Stresses at critical points have been computed for the three following cases:

A - Reactor unpressurized. Studs tightened as called for by ASME Unfired Pressure Vessel Code(1), paragraph UA47.

Total Load $\frac{\pi}{4} G^2p + 2\pi bGmp$, for p = 1200 psi.

B - Pressure raised to 1200 psi. Stud setting unchanged. Unit at 125°F.

C - Steady state thermal stresses, with pressurized water at 1200 psi, 420°F. Stud setting unchanged.

As a result of these steps, the following changes occur:

Case	Rotation, Radians x 10^6		Gasket		Stud Tension	
	Bull Ring,	Flange	ΔR inches	Shear lb/inch	lb/Bolt	lb/Inch of Stud Circle
A	- 337	+ 181	-.00113	+ 724	63,500	9,244
B	- 235	+ 133	-.00007	+15,116	68,200	9,920
C	+2147	+1480	+.02591	- 989	100,000	14,560

Application of the internal pressure produces a large shear force on the gasket, but only slightly increases the stud tension. The steady state temperature gradients cause large rotations of the bull ring and flange, and increase the stud tension by over 50%. Gasket shear drops almost to zero.

Stresses in the reactor cover at the points of interest are summarized below. As shown by Fig. 1, these points are: (1) the edge of the spherical portion of the cover (Point A), (2) the portion of the cover at the gasket (Point C), (3) the junction with the 1/2 in. thick dished head (Point H), and the studs (Points E and F). The equations on which these are based are given in Appendix B so that computations may be made for other values of pressure and of stud tension. The letters T and C after the stresses refer to tension and compression respectively.

The highest computed stud stress was 27,180 psi. It occurred at the lower end (F) of the reduced section. This does not include any stress concentration factor for effect of threads; however, it does conservatively assume that the studs were initially tightened exactly as called for by the ASME Code. (1)

The greatest computed stresses occur in the cover at point C, Fig. 1, in line with the gasket. The drawings show on the upper face a 90° re-entrant angle at the inside of the bull ring, with no fillet radius. On the lower face, the cover notched 1/2 in. deep for the gasket, leaving a net metal thickness of 2-1/4 in. As long as the radial stresses on the lower face remain compression, they will

TABLE 1
STRESS SUMMARY

I. Stress (psi) in Section at Point A (2-3/4 in. Thickness)

Loading	Direct	Bending	Combined At	
			Top	Bottom
Case A	437 C	650	213 T	1,087 C
Case B	4,710 T	3,770	8,480 T	940 T
Case C	12,100 C	41,400	53,500 C	29,300 T

II. Stresses (psi) in Section at Point C (2-3/4 in. Thickness)

Loading	Direct	Bending	Combined At	
			Top	Bottom
Case A	710 C	794	84 T	1,504 C
Case B	3,865 T	18,930	15,065 C	22,795 T
Case C	13,000 C	56,100	69,100 C	43,100 T

III. Stresses (psi) in Section at Point C
(Treated as 2-1/4 in. thickness to gasket groove)

Loading	Direct	Bending	Combined At	
			Top	Bottom
Case A	867 C	1,185	318 T	2,052 C
Case B	4,720 T	28,260	23,540 C	32,980 T
Case C	15,860 C	83,800	99,660 C	67,940 T

IV. Stresses (psi) in Weld to 1/2 in. Thick Dished Head at Point H

Loading	Direct	Bending	Combined At	
			Top	Bottom
Case A	1,078 C	9,672	10,750 C	8,494 T
Case B	1,138 C	9,890	11,028 C	8,752 T
Case C	4,375 C	30,800	35,175 C	26,425 T

V. Stud Stresses (psi) in Section at Point E

Loading	Direct	Bending	Combined	
			Left	Right
Case A	12,590 T	2,800	9,790 T	15,390 T
Case B	13,500 T	2,040	11,460 T	15,540 T
Case C	19,800 T	2,760	22,560 T	17,040 T

VI. Stud Stresses (psi) in Section at Point F

Loading	Direct	Bending	Combined	
			Left	Right
Case A	12,590 T	6,450	19,040 T	6,140 T
Case B	13,500 T	4,570	18,070 T	8,930 T
Case C	19,800 T	7,380	12,420 T	27,180 T

pass through the gasket metal, and the effective thickness remains 2-3/4 in. This is true for stud load; but with internal pressure and particularly with thermal stress, the bending moment reverses direction, and the gasket notch will tend to open. The effective thickness, becomes 1/2 in. less, and stress concentrations occur at each re-entrant angle of the notch. The gasket also will tend to exert a wedging action.

Using the 2-1/4 in. thickness at point C in the cover, and ignoring stress concentration effects, the calculated stress at the top surface ranges from 318 psi in tension after the cover is bolted down, to 99,660 psi in compression at steady state operating conditions (refer to Summary III). The alternating stress intensity is

$$\frac{318 + 99,660}{2} = 50,000 \text{ psi.}$$

From the Navy Code the mean stress in this case is zero. Thus, the alternating stress of 50,000 psi is less than the Navy Code allowable alternating stress of 66,000 psi for 500 cycles.

At the bottom surface of point C, the computed stresses range from 2052 psi in compression to 67,940 psi in tension. If stress concentration effects are ignored, the alternating stress intensity is

$$\frac{2052 + 67,940}{2} = 35,000 \text{ psi.}$$

The mean stress in this case is 7,000 psi. Plotting this point on Fig. 3 shows that this stress condition is safe for more than 2500 cycles.

Figures 3 and 4, fatigue diagrams for A-212 Gr. B and 304 stainless steel, show the mean and alternating stresses plotted for points A, C and H.

The worst condition occurs at the top surface of point C which falls within the 500 cycle range. In order to find the maximum allowable number of cycles for an alternating stress of 50,000 psi, reference is made to the Navy Code. For 500 cycles of operation, the assigned usage factor is 0.4. The usage factor is the ratio of the number of times a certain operation will occur to the number of cycles required to produce an unsafe condition. From the Design Fatigue Strength Curve for A-212, in the Navy Code, the number of cycles to produce an unsafe condition is 4000 for an alternating stress of 50,000 psi. Applying the assigned usage factor of 0.4 gives 1600 cycles as the allowable number of cycles during the vessel lifetime.

The number of startup and shutdown cycles for the SM-1 over its 20-yr life has been estimated to be 900[3] which is well below the allowable 1600 cycles corresponding to the 50,000 psi alternating stress.

If stress concentration effects are included the margin of safety will be less.

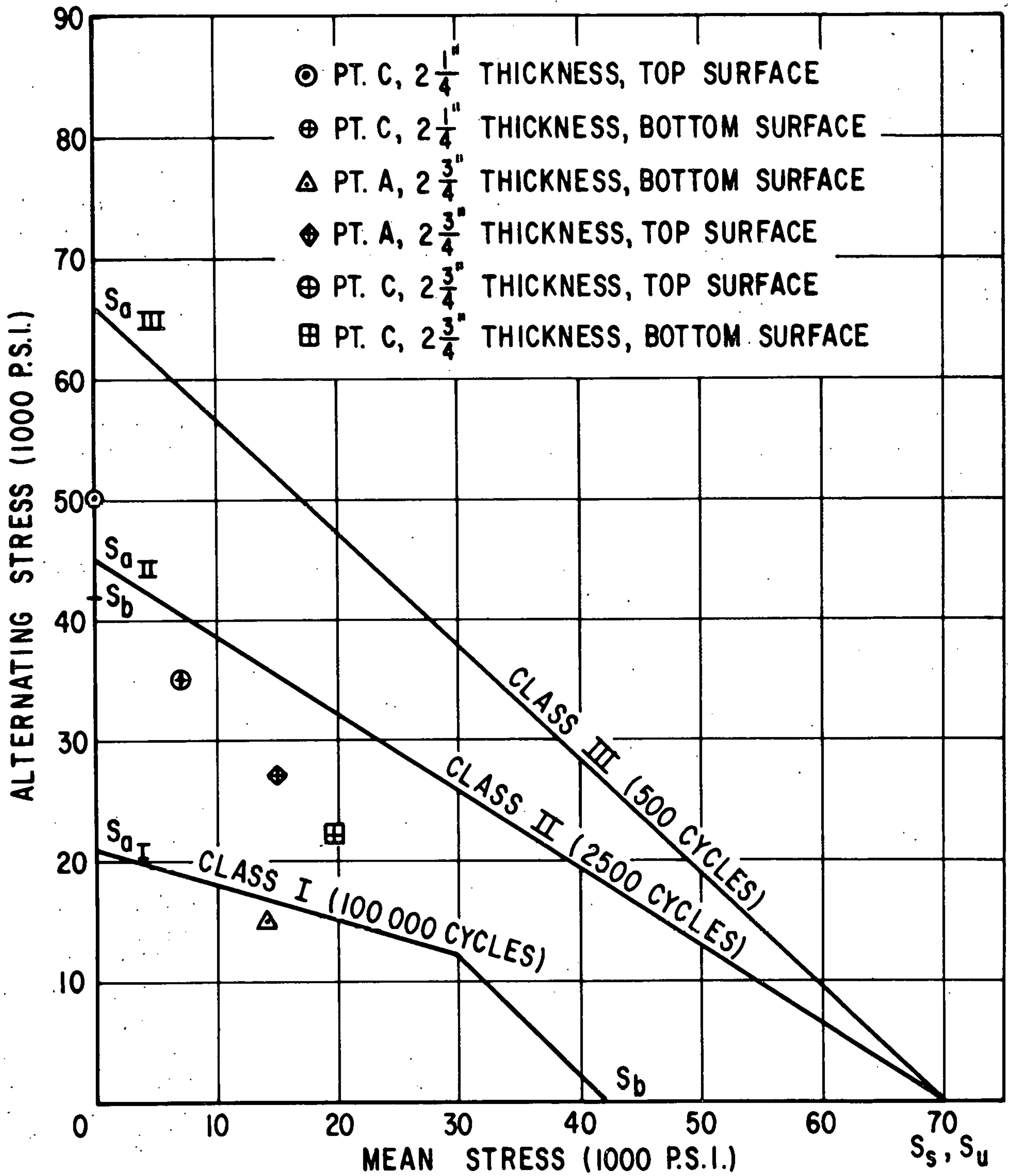

Figure 3. Fatigue Diagram (A-212 Gr B Material) - SM-1 Reactor Vessel

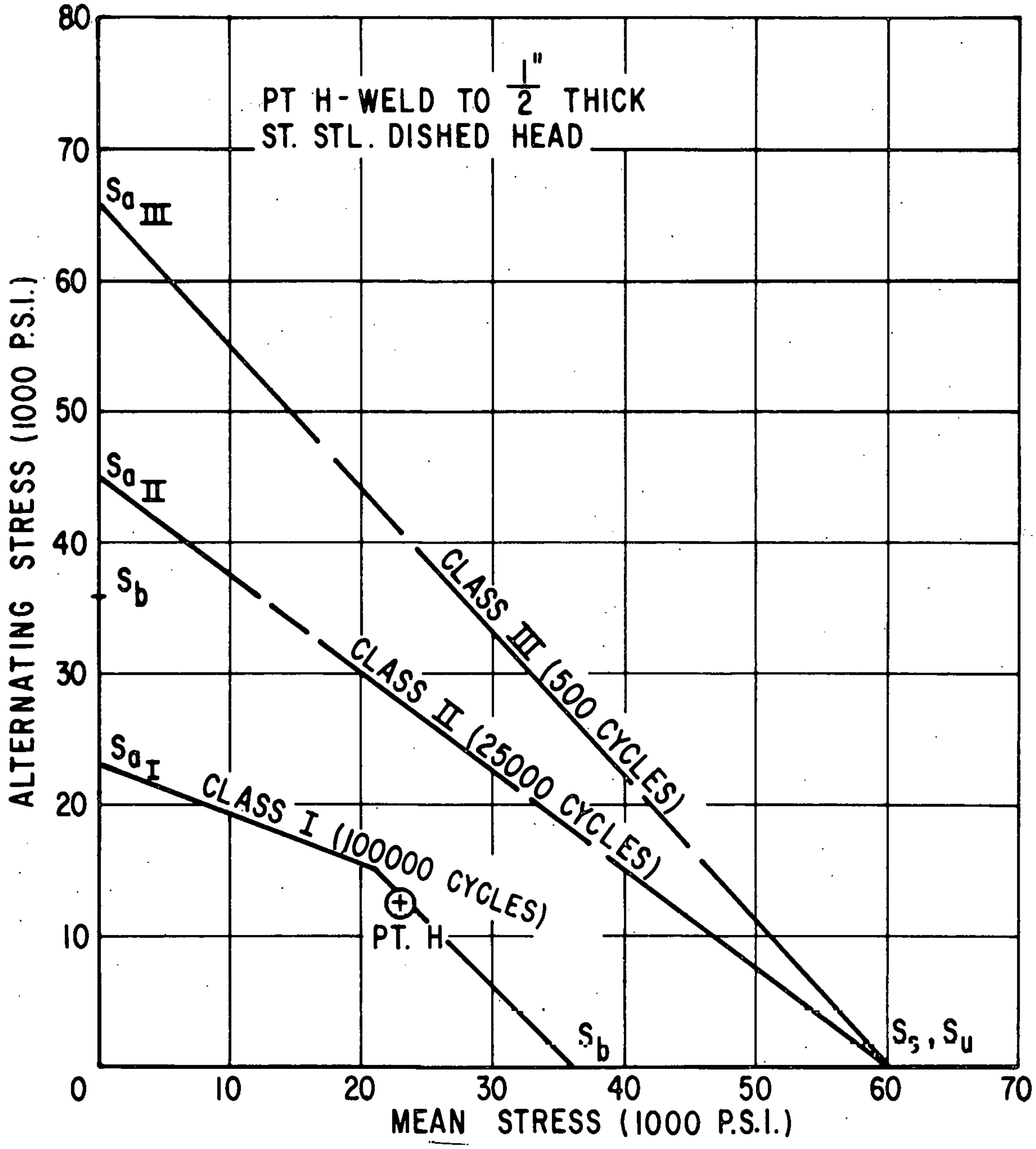

Figure 4. Fatigue Diagram (304 Stainless Steel Material) - SM-1 Reactor Vessel

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4.0 EFFECT OF THERMAL TRANSIENTS

4.1 METHOD OF COMPUTATION

Under steady state operating conditions, the cover temperatures will range from 420°F inside the gasket, to 320°F to 180°F across the lower face of the bull ring to 135-140°F on its upper face. The vessel flange temperatures will range from 420°F on the inside face, to 220°F to 320°F on the area exposed to outside cooling.

Due to the slow heat transfer through the metal during heating up periods, these temperature differentials are temporarily exceeded and stresses may be temporarily increased. For the normal startup rate of 30°F per hour, approximate values can be readily obtained using curves,[4] which are exact solutions of a semi-infinite slab subjected to a ramp temperature charge.

For a 7 in. thick steel slab, having an outer face insulated and inner face exposed to water which is rising in temperature at 30°F per hour, and assuming values for h of 300 (rapid circulation) or 30 (stagnant),

$$N_{Fo} = \frac{.0084t}{(7)^2} = 0.000171t_{\text{seconds}} = 0.616\ t_{\text{hours}}$$

$$N_{B_1}^{-1} = \frac{16.4 \times 12}{300 \times 7} = 0.094 \quad h = 300$$

$$\text{or} \quad \frac{16.4 \times 12}{30 \times 7} = 0.94 \quad h = 30$$

4.2 DISCUSSION AND RESULTS

From curves prepared by Alco,[4] the following information is obtained:

		Wall Temperature Rise (°F)					
		h = 300			h = 300		
Time	Temp. Rise °F	Inside	Out	Diff.	Inside	Out	Diff.
1/2 hr	15°	12.35	1.85	10.5	4.6	0.55	4.05
1 hr	30°	26.16	9.18	17.0	11.9	3.9	8.0
5 hr	150°	145.2	120.0	25.2	108.0	84.7	23.3
10 hr	300°	295.5	270.0	25.5	254.1	229.9	24.2

With stagnant water, the temperature of the wetted wall is slower to rise, but the differential between inside and outside face presently reaches about 25°F and thereafter remains constant. For a wall thickness of 2.75 in. and using the same procedure as above, the differential amounts to only 4°F, reached more rapidly.

Time	Temp. Rise (°F)	Wall Temperature Rise °F					
		h = 300			h = 30		
		Inside	Out	Diff.	Inside	Out	Diff.
1/4 hr.	7.5	6.02	3.0	3.02	2.01	0.92	1.09
1/2 hr	15	13.2	9.6	3.60	5.73	3.90	1.83
1 hr	30	28.25	24.6	3.65	16.2	13.5	2.7
2 hr	60	58.2	54.5	3.7	43.8	40.3	3.5

The vessel flange has a major dimension of slightly over 7 in. If heated on one face only, and insulated on the other three faces, a 25°F transient differential would be expected. It is actually heated on two faces, and cooled over most of the top face, so that in the steady state condition, the lower face and inside face is 410-420°F, and the top face of 220°F to 320°F, with an average differential of 120°F. The transient would increase this by approximately 10%.

The temperature of the bull ring is determined mainly by the heat loss to the surrounding water. The temperature at the upper end never rises much above that of the water. During heat-up, the lower end warms up gradually so that during the startup the differential between top and bottom is likely to be lower rather than higher than the steady state condition. There will be a period when the lower end of the bull ring will lag in temperature rise behind the dome and inner ring of the cover. This will increase the direct compressive pressure at section C, but decrease the bending moment. From the Stress Summary III (Table 1) it can be seen that the net change in combined stress for the transient condition compared to the steady state condition will be small. Consequently, the steady state stress at C is still regarded as the maximum stress in the cover-flange region.

More exact computation of the transient stresses does not seem necessary, in view of their probable small amount, which is roughly 10% or less above steady state thermal stress.

5.0 CONCLUSIONS

The maximum stud stress is 27, 180 psi and it occurs during operation at full power. This value will be used in the evaluation of the closure stud failures under a separate subtask of the contract program.

The maximum calculated combined stress of 99, 660 psi occurred in the cover at point C, the outside surface of the junction between the dome and the bull ring, as a result of pressure and steady state temperature gradients.

The alternating stress of 50, 000 psi at this point could safely be applied for 1600 cycles. Since the maximum number of stress cycles expected has been estimated at only 900, the stress level is safe.

The estimated effect of a thermal transient of $30^{O}F/hr$ was to increase the steady state thermal stresses by not more than 10%. However, the change in the distribution of stress during a transient was such that the steady state combined stress at point C remained the maximum calculated value.

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6.0 REFERENCES

1. Boiler and Pressure Vessel Code: Section VIII, "Unfired Pressure Vessels," American Society of Mechanical Engineers, New York, 1959.

2. "Tentative Structural Design Basis for Reactor Pressure Vessels and Directly Associated Components," United States Department of Commerce Bulletin No. PB 151987 (Revised December 1, 1958).

3. Chittum, R. A., Knipe, R. K., McLaughlin, D. W., "Preliminary Stress Fatigue Analysis of the Primary System of the SM-1," APAE Memo-283, to be published February 1962.

4. Porter, F. P., "Temperature Stresses in a Flat Plate Due to Transient Heat Flow," Internal Report of Alco Products, Inc.

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APPENDIX A - DETAIL METHOD OF COMPUTATION

Separate the cover and flange into the following units as shown in Fig. 5:

1 - Dome, 2.875 in. thick, extending to point A.

2 - Inner Ring, extending from A to C, above ℄ of gasket. This is treated as an annulus in bending, and as a thick cylinder for radial movement.

3 - Outer Ring. This is treated as a ring in torsion and as a thick cylinder radially. Internal bending within the ring is ignored.

4 - Dished head, above insulation.

5 - Studs.

6 - Lower Flange. This is treated as is the outer ring (3).

7 - Shell, treated as a beam on an elastic foundation, attached to the lower flange.

8 - Gasket. This carries shear and compressive force. Bending resistance is ignored.

Steps: -

A - Combine (1) and (2) to find relation between forces and deflections at point C, in terms of $\triangle R_{B1}$ and ϕ_D.

B - Solve (4) for relation between radial force and radial movements at H.

C - Solve (5) for forces and moments in studs in terms of angular rotation of (3) and (6) about center point of gasket.

D - Write equilibrium equations for radial forces and moments for (3), in terms of $\triangle R_{B1}$ at center of gasket, and ϕ_D and ϕ_G, using B and C. Combine with A.

E - Write equations for shear and moment at junction of shell and lower flange, in terms of $\triangle R_{B1}$ and ϕ_G.

F - Write equations for equilibrium of lower flange in terms of ΔR_{B1}, ϕ_G and ϕ_D, using data from C and E.

G - Solve for shear at B^1 and ΔR_{B1}, using D and F.

All final equations will include a term with P_E - measuring effect of stud tension; a term in p - measuring effect of internal pressure; and a numerical term, measuring effect of temperature differentials throughout the reactor.

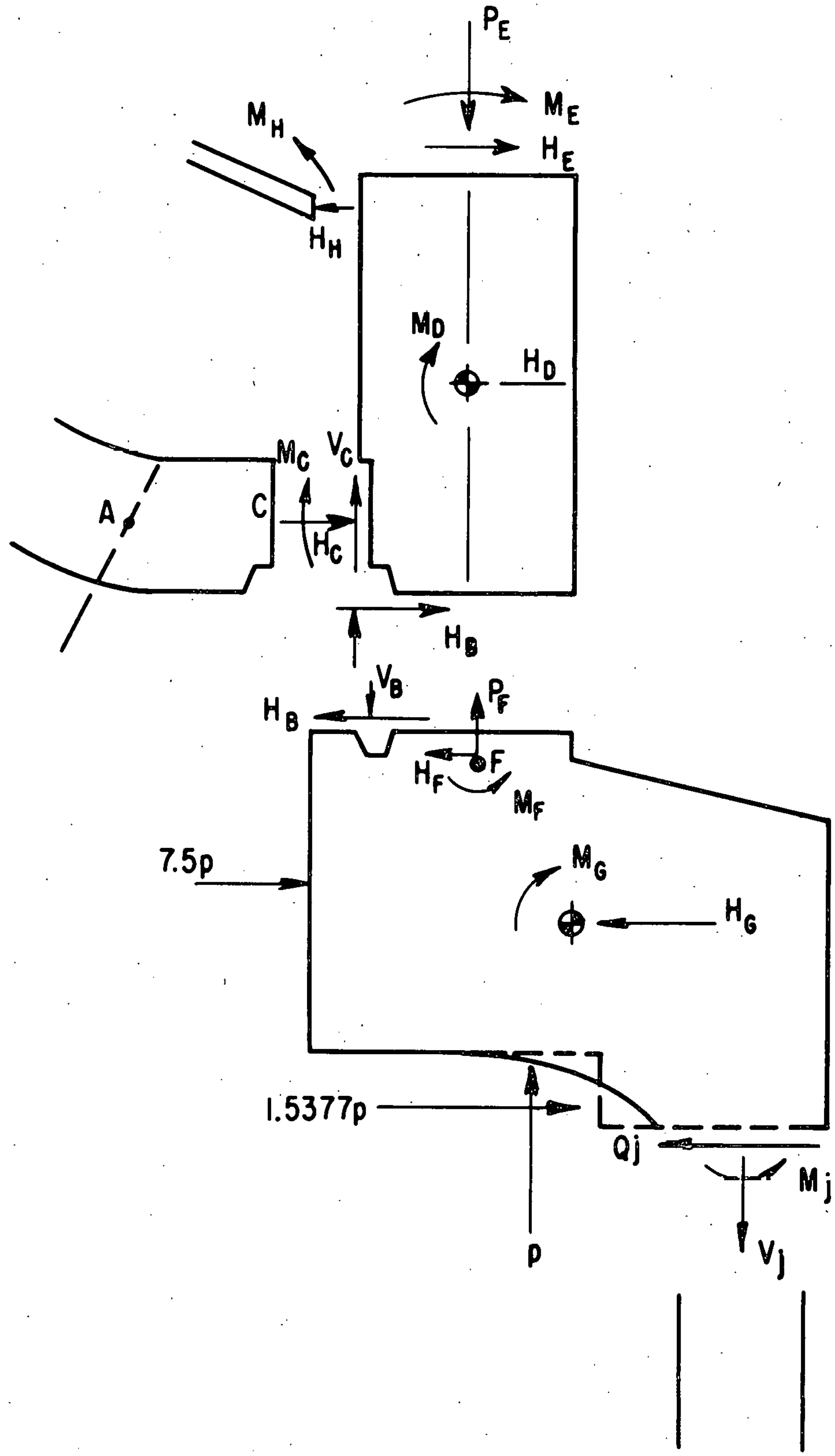

Figure 5. SM-1 Reactor Vessel Cover and Flange Free Body Diagram

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APPENDIX B - DATA SHEETS

SM-1 Reactor Vessel Cover, Flange and Stud Calculations

BOLT AND COVER STRESSES IN THE SM-1 REACTOR

BASIC EQUATIONS:

Elongation of Bolt $= 2.25(\phi_D - \phi_G)$

$$\phi_D - \phi_G = (.15754p + 1296.6 - .056178 P_E) \times 10^{-6}$$

Also $\Delta L = \frac{PL}{AE} = \frac{11.062}{5.04 \times 30 \times 10^6} \times 6.86 P_E = .5014 \times 10^{-6} \Delta P_E$

Whence: $\phi_D - \phi_G = .223 \times 10^{-6} \Delta P_E$, if nuts are not turned.

Bending Moments:

Upper end, $M_E = -.20207p - 1447.1 + .072058 P_E$

Lower end, $M_F = .453p + 3512.1 - .16153 P_E$

Shear, $H_E = H_F = .05922p + 448.24 - .02112 P_E$

Where: ϕ_D = angle of rotation, outer ring

ϕ_G = angle of rotation, Flange

p = internal pressure, psi

$6.86 P_E$ = bolt tension, lb./bolt

CASE A. REACTOR COLD AND WITHOUT PRESSURE

Bolts tightened up as per Boiler Code, to produce a force of 9244 lb. per inch at the bolt circle.

Then $\phi_D - \phi_B = -.056178 \times 9244 \times 10^{-6} = -518.6 \times 10^{-6}$

ΔL for bolts $= .5014 \times 10^{-6} \times 9244 = 4835 \times 10^{-6}$

Threads Turned $= (518.6 + 4835) \times 10^{-6} = .005353$ inches

Bolt Stresses, Tension $= \dfrac{9244 \times 6.86}{5.04} = 12,590$ psi

Bending Stress, $S = \dfrac{6.86 \times 2.531}{2 \times 2.014} M = 4.325 M$

$M_E = .072058 \times 9244 = +665$

Stress $= 4.325 \times 665 = 2800$ psi

$M_F = -.16153 \times 9244 = -1492$

Stress $= -1492 \times 4.325 = -6450$ psi

Combined at F:

$12,590 + 6450 = 19,040$ psi

CASE B. APPLY 1200 psi INTERNAL PRESSURE WITHOUT ALTERING BOLTS

P_E changes by ΔP_E

Then $\Delta(\phi_D - \phi_G) \times 10^6 = .15754 \times 1200 - .056178\, \Delta P_E$

$$= .223\, \Delta P_E$$

$$\Delta P_E = \frac{.15754 \times 1200}{.223 + .056178} = 676 \text{ lb per inch}$$

$9244 + 676 = 9920$ lb.

$$\text{Bolt Tension} = \frac{9920 \times 6.86}{5.04} = 13{,}500 \text{ psi}$$

$M_E = -.20207 \times 1200 + .072058 \times 9920 = +472.5$

Stress $= 472.5 \times 4.325 = 2040$ psi

$M_F = .453 \times 1200 - .16153 \times 9920 = -1057.4$

Stress $= -1057.4 \times 4.325 = -4570$ psi

Combined at F:

$13{,}500 + 4{,}570 = 18{,}070$ psi

CASE C. WITHOUT CHANGING INTERNAL PRESSURE OR BOLT SETTING, RAISE TEMPERATURE TO OPERATION CONDITION.

$$\Delta(\phi_D - \phi_G) \times 10^6 = 1296.6 - .056178 \Delta' P_E = .223 \Delta' P_E$$

$$\Delta' P_E = \frac{1296.6}{.223 + .056178} = 4640 \text{ lb per inch}$$

$$\Delta' P_E = 4640 + 9920 = 14560 \text{ lb.}$$

$$\text{Bolt Tension} = \frac{6.86 \times 14560}{5.04} = 19800 \text{ psi}$$

$$M_E = -.20207 \times 1200 - 1447.1 + .072058 \times 14560 = -639.6$$

$$\text{Stress} = 4.325 \times (-639.6) = -2760 \text{ psi}$$

$$M_F = .453 \times 1200 + 3512.1 - .16153 \times 14560 = 1705.7$$

$$\text{Stress} = 4.325 \times 1705.7 = 7380 \text{ psi}$$

Combined at F:

$$19800 + 7380 = 27{,}180 \text{ psi}$$

ALCO PRODUCTS INC.

BY ______ DATE ______ SUBJECT ______ SHEET NO. 5 OF 9
CHKD. BY ______ DATE ______ JOB NO. ______

BENDING STRESSES IN COVER:

$$\Delta R_B = .95449 \times 10^{-6} p + .026551 - .12235 \times 10^{-6} P_E$$

$$H_B = 11.94565 p - 16469 + .078290\, P_E$$

$$\phi_D = .10601 \times 10^{-6} p + .002552 - .03651 \times 10^{-6} P_E$$

$$\phi_G = -.05153 \times 10^{-6} p + .001256 + .019668 \times 10^{-6} P_E$$

$$M_C = 13.53 \times 10^6\, \phi_D + 22.315 p - 3.1468 \times 10^6 \Delta R_B + 96499$$

$$H_C = 11.223 \times 10^6\, \phi_D - 6.950 p - 5.0713 \times 10^6 \Delta R_B + 151408$$

$$M_A = 1.0731\, M_c + .13059\, H_c - 24.140 p + 542$$

$$H_A = .6976\, H_c - 3.859 p + .1931\, M_A + 2895$$

$$V_B = \frac{17.5}{15.25} P_E - \frac{15.25}{2} P$$

@ A + C:

$$I/c = \frac{(2.75)^2}{6} = 1.260$$

ALCO PRODUCTS INC.

BY ______ DATE ______ SUBJECT ______ SHEET NO. 6 OF 9
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	BOLT LOAD ONLY	+ INTERNAL PRESSURE	+ STEADY STATE THERMAL STRESS
Bolt - P_E	9244	9920	14560 lb/in
PRESSURE P	0	1200	1200 psi
ϕ_D	-337.1×10^{-6}	-234.8	$+2147 \times 10^{-6}$
ϕ_G	$+181.5 \times 10^{-6}$	$+133.2$	$+1480 \times 10^{-6}$
$\phi_D - \phi_G$	-518.6×10^{-6}	-368.0	$+667 \times 10^{-6}$
ΔR_B	-1130×10^{-6}	-69.0	$+25915 \times 10^{-6}$ in.
H_B	724	+15116	−989 lb.
M_C	−1000	+23845	+70700 in.-lb.
H_C	+1950	−10625	+35,700 lb.
M_A	−818	−4755	+52,142 in.-lb/in.
H_A	+1202	−12958	+33245 lb/in.
V_B	+10608	+2232	+7559

STRESSES 2¾" Thickness

H_C	+710 C	−3865 T	+13000 C
M_C	−794	+18930	+56100
Σ	+1504	−22795	+69100
H_A	+437 C	−4710 T	+12100 C
M_A	−650	−3770	+41400
Σ	+1087	−8480	+53500

ALCO PRODUCTS INC.

BY ________ DATE ________ SUBJECT ______________________ SHEET NO. 7 OF 9
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STRESSES 2¼" Thickness:

	BOLT LOAD ONLY	+ INTERNAL PRESSURE	+ STEADY STATE TERMINAL STRESS
H_c	+867	−4720	+15860
M_c	−1185	+28260	+83800

ALCO PRODUCTS INC.

BY ______ DATE ______ SUBJECT ______ SHEET NO. 8 OF 9

CHKD. BY ______ DATE ______ JOB NO. ______

$$H_H = 2.5049 \times 10^6 \phi_D - .26818 \times 10^6 \Delta R_B' - 617$$

$$M_H = 1.8112 \times 10^6 \phi_D - .18314 \; 10^6 \Delta R_B' - 421$$

$$\Delta R_j = \Delta R_B + 9.256 \phi_G + .005873 - 6.081 \times 10^{-6} p$$

$$M_j = -2.759 \times 10^6 \Delta R_B - 43.296 \times 10^6 \phi_G + 130354 + 16.778 P$$

$$Q_j = .85728 \times 10^6 \Delta R_B + 10.694 \times 10^6 \phi_G - 40504 - 5.213 p$$

$$V_j = \frac{235}{2} \times \frac{23.5}{24.875} P \quad lb/in$$

$$A + 420°F, \; \Delta R_j = 24.875 \times 6.7 \times 10^{-6} (420 - 125) = .0492''$$

<u>Bolt load only:</u>

$$\Delta R_j = -1130 \times 10^{-6} + 9.256 \times 181.5 \times 10^{-6} = +560 \times 10^{-6}$$

$$M_j = -2.759(-1130) - 43.296(181.5) = -4730$$

$$Q_j = .85728(-1130) + 10.694(181.5) = +972$$

$$\left.\begin{aligned} H_H &= 2.5049(-337.1) - .26818(-1130) = -539 \\ M_H &= 1.8112(-337.1) - .18314(-1130) = -403 \end{aligned}\right\} \text{Bolt load}$$

$$\left.\begin{aligned} H_H &= 2.5049(-234.8) - .26818(-69) = -569 \\ M_H &= 1.8112(-234.8) - .18314(-69) = -412 \end{aligned}\right\} \text{Bolt load + Pressure}$$

ALCO PRODUCTS INC.

BY ______ DATE ______ SUBJECT ______
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Bolt Load + Pressure + Thermal Stresses =

$$\Delta R_j = 25915 \times 10^{-6} + 9.256 \times 1480 \times 10^{-6} + .005873 - 6.081 \times 1200 \times 10^{-6} =$$

$$= +.0382$$

$$M_j = -2.759 \times 25915 - 43.296 \times 1480 + 130354 + 16.778 \times 1200 =$$

$$= +15{,}700$$

$$Q_j = .85728 \times 25915 + 10.694 \times 1480 - 40504 - 5.213 \times 1200 =$$

$$= -9740$$

$$H_H = 2.5049(+2147) - .26818(+25915) - 617 = -2187$$

$$M_H = 1.8112(+2147) - .18314(+25915) - 421 = -1281$$