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**THE EFFECT OF GAMMA HEATING ON THE
APPR-1 PRESSURE VESSEL — NEW DESIGN**

By
H. R. Kroeger

July 1956

**Advanced Scientific Techniques
Research Associates
Milford, Connecticut**

Technical Information Service Extension, Oak Ridge, Tenn.

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THE EFFECT OF GAMMA HEATING
ON THE APPR-1 PRESSURE VESSEL
- NEW DESIGN -

H. R. Kroeger

ASTRA
Box 163
Milford, Connecticut

July 1956

Work performed under Contract No. AT(11-1)-318.

For Alco Products, Inc.

I. SUMMARY AND CONCLUSIONS

This report examines the effect on pressure vessel stress of reducing the inside diameter of the vessel from the previous specified size (see ASTRA 415E2.1). Although a reduction in diameter decreases the pressure stress it exposes the vessel to a higher incident gamma energy and the resulting thermal stress increases. As a result the total stress is relatively insensitive to diameter.

Standard practice for a welded pressure vessel indicates a maximum allowable design stress of 16600 (for ASTM-A 212-B steel). For a 40.5 inch inside diameter pressure vessel (41 inches I.D. neglecting the clad) the required wall thickness, based on analyses contained herein, is 2.6 inches. If boron loaded stainless steel were to be used for the thermal shield the wall thickness could be reduced to 2.3 inches.

The substitution of a canned lead slab of equal weight, for the present thermal shield would allow the pressure vessel diameter to be reduced to 39.5 inches. This change in combination with the added efficiency of lead as a thermal shield material reduces the required wall thickness to 2.1 inches.

For the general APPR-1 configuration the pressure stress represents by far the largest portion of the total vessel stress. The pressure stress can be reduced by increasing the wall thickness, by reducing the vessel diameter, or by lowering the design pressure.

An alternative approach would be to accept a higher pressure stress and specify a steel with a higher allowable design stress.

For reactor vessels either a reduction in diameter or an increase in wall thickness unfortunately tends to increase the thermal stress.

If the requirement of minimum overall weight is added to the above considerations the design of an optimum system becomes somewhat more complex. Several obvious guides can be established:

1. The thermal shield should be positioned as close to the core as is possible without adversely affecting the nuclear characteristics.
2. The thermal shield should be as thin as possible (a heavy shield material) and should be designed to minimize secondary gamma generation.
3. The pressure vessel should be of a high strength steel which still retains favorable properties i.e. low coefficient of expansion and high thermal conductivity.
4. The clad material for the pressure vessel should be boron loaded to minimize the generation of gammas (and heat) in the pressure vessel wall.

II. INCIDENT RADIATION ENERGY

The purpose of this analysis is to evaluate the stresses in the wall of a 40.5 inch I.D. pressure vessel designed for APPR-1. Because of the similarity to the original design analysis (ASTRA 415E2.1) many short cuts can be adopted. For complete understanding this report should be read with that report at hand.

A. The Primary Gammas

The primary gammas for the original design were attenuated through 10.625 inches of water and 2.125 inches of stainless steel before reaching the pressure vessel. For this design the water attenuation distance is reduced to 7.25 inches.

$$I_{9.25} = \frac{S_v B R^2 F(\theta, b)}{2(a + z)}$$

$$a = 9.25 \text{ inches} = 23.5 \text{ cm.}$$

Using other data directly from ASTRA report no. 415E2.1

Group	z	(a + z)	$\tan \theta (= \frac{h}{2(a + z)})$	θ
1	12.2	35.7	0.785	38°
3	14.2	37.7	0.743	37°
5	14.7	38.2	0.733	36°
7	14.9	38.4	0.730	36°

$$b = (0.842 \mu_w (a - t)) + \mu_c z + \mu_{sst} \quad (\text{where } t = 2.125 \text{ in} = 5.4 \text{ cm})$$

Group	$(0.842 \mu_w(a - t))$	μ_{cz}	Sum	μ_{sst}	b
1	1.067	1.96	3.027	2.62	5.647
3	0.594	1.30	1.894	1.53	3.424
5	0.457	1.16	1.617	1.31	2.927
7	0.396	1.12	1.516	1.25	2.766

Group	B_w	B_{ss}	B_{Total}
1	5.7	3.5	20
3	2.3	1.87	4.3
5	1.83	1.52	2.78
7	1.63	1.39	2.26

Group	S_v	$2(a + z)$	B	R^2	$F(\theta, b)$	$I_{9.25}$
1	2.31×10^{13}	71.4	20	784	0.00155	7.86×10^{12}
3	7.42×10^{11}	75.4	4.3	784	0.0167	5.54×10^{11}
5	4.57×10^{11}	76.4	2.78	784	0.0275	3.58×10^{11}
7	3.94×10^{11}	76.8	2.26	784	0.0323	2.94×10^{11}

B. THE SECONDARY GAMMAS

1. Capture Gammas

The incident gammas at the pressure vessel wall resulting from capture gamma generation in the flow separator and the thermal shield will differ from those in the original design only by the attenuation factor. For the water capture gammas the change in the number and thickness of the slabs must also be considered.

a. Flow separator

Group	$I_{0.125}$	Attenuation	$I_{0.25}$
1	4.78×10^{10}	$2.45/47.8$	2.44×10^9
3	8.66×10^{10}	$9.3/86.6$	9.3×10^9
5	1.31×10^{11}	$1.5/13.1$	1.5×10^{10}
7	4.12×10^{11}	$5/51.2$	5.0×10^{10}

b. Thermal shield

Group	$I_{8.5}$	Attenuation	$I_{9.25}$
1	4.72×10^{10}	$2.45/2.82$	4.1×10^{10}
3	1.42×10^{11}	$9.3/10.5$	1.26×10^{11}
5	2.38×10^{11}	$1.5/1.7$	2.55×10^{11}
7	7.77×10^{11}	$5/5.6$	6.94×10^{11}

c. Water capture (2.2 MEV)

Slab	I_1	I_3	$I_{6.5}$	Attenuation	$I_{9.25}$
1	4.14×10^{11}			$1.86/16.7$	4.62×10^{10}
2		1.86×10^{12}		$1.86/11.6$	2.98×10^{11}
3			1.73×10^{12}	$1.86/6.3$	5.11×10^{11}

Slab 4 may be computed from the expression

$$I = \frac{\phi C e^{-\mu t}}{2(A - \mu)} (1 - e^{-t(A - \mu)})$$

$$\text{where } \phi = 3.3 \times 10^{12}$$

$$C = 0.0126$$

$$\mu = 0.0391$$

$$t = 0.75 \text{ in.} = 1.91 \text{ cm}$$

$$e^{-Ax} = \cancel{0.0} = \frac{3.52 \times 10^{12}}{3.3 \times 10^{12}} = 1.07$$

$$Ax = -0.068$$

$$A = \frac{-0.068}{1.91} = -0.0356$$

For slab 4

$$I_{9.25} = \frac{3.3 \times 10^{12} \times 0.0126 \times e^{-0.0391 \times 1.91} (1 - e^{-1.91(-0.0747)})}{2(-0.0356 - 0.0391)}$$

$$I_{9.25} = \frac{1.65 \times 10^{12} \times 0.0126 \times 0.928}{-0.0747} (1 - 1.153)$$

$$I_{9.25} = 3.95 \times 10^{10}$$

The total incident gammas due to water capture are

$$I_{9.25} = (0.492 + 2.98 + 5.11 + 0.395) \times 10^{11} = 8.977 \times 10^{11}$$

d. Pressure vessel

Capture gammas generated in the pressure vessel are proportional to the incident thermal neutron flux.

Group	$I_{12.75}$	Flux Ratio	$I_{9.25}$
1	4.4×10^{10}	$3.52/2$	7.75×10^{10}
5	6.4×10^{10}	"	1.13×10^{11}
7	1.74×10^{11}	"	3.06×10^{11}

2. Inelastic scatter gammas

The gammas due to inelastic scattering may be estimated in the same manner as the capture gammas above.

a. Flow separator

$$I_{9.25} = 2.38 \times 10^{11} \times \frac{2.45}{47.8} = 1.22 \times 10^{10}$$

b. Thermal shield

$$I_{9.25} = 1.23 \times 10^{11} \times \frac{2.45}{2.82} = 1.07 \times 10^{11}$$

c. Pressure vessel

Inelastic scatter gammas (equivalent incident) are proportional to the incident fast flux

$$I_{9.25} = 1.72 \times 10^{10} \times \frac{7 \times 10^{11}}{1.7 \times 10^{11}} = 7.1 \times 10^{10}$$

It should be noted that the above estimates of capture and inelastic scatter gammas in the pressure vessel are not quite precise since it is expected that the wall thickness will change slightly.

The above estimates are summarized below in terms of gamma flux $\times 10^{-11}$.

Group (Energy)	1	3	5	7
Primary Gammas	78.6	5.54	3.58	2.94
Capture Gammas				
Flow Separator	0.024	.093	.15	.50
Thermal Shield	0.41	1.26	2.55	6.94
Water	--	8.98	--	--
Pressure Vessel	--	.775	1.13	3.06
Inelastic Scatter				
Flow Separator	0.122	--	--	--
Thermal Shield	1.07	--	--	--
Pressure Vessel	0.71	--	--	--
<hr/>				
Total	80.936	16.648	7.41	13.44
Mev/cm ² -sec	80.94	50.0	37.0	94.1
μ_e	0.206	0.176	0.178	0.185
Mev/cm ³ -sec	16.7	8.8	6.6	17.4
Total				49.5
Btu/ft ³ -hr				76700
Btu/in ³ -hr				44.4

III. COMBINED STRESSES

A. PRESSURE STRESS

The general expression for the tangential stress in a thick walled pressure vessel is:

$$S_{TP} = p \left(\frac{a^2}{b^2 - a^2} \right) \left(\left(\frac{b}{x + a} \right)^2 + 1 \right)$$

where p = design pressure - psi

a = inside radius - inches

b = outside radius - inches

x = point of interest in the wall

It is obvious that the maximum pressure stress occurs when x = 0, i.e. at the inside surface of the pressure vessel.

For this analysis

$$p = 1400$$

$$a = 20.5 \text{ (clad is neglected)}$$

$$b = 22.5, 22.75, 23$$

For b = 22.5

$$\begin{aligned} S_{TP} &= 1400 \left(\frac{20.5^2}{22.5^2 - 20.5^2} \right) \left(\frac{22.5^2}{20.5^2} + 1 \right) \\ &= 1400 \left(\frac{420}{506 - 420} \right) \left(\frac{506}{420} + 1 \right) \\ &= 1400 \times 4.88 \times 2.205 = 15100 \text{ psi} \end{aligned}$$

For b = 22.75

$$\begin{aligned} S_{TP} &= 1400 \left(\frac{420}{518 - 420} \right) \left(\frac{518}{420} + 1 \right) \\ &= 1400 \times 4.28 \times 2.23 = 13370 \text{ psi} \end{aligned}$$

For $b = 23$

$$S_{TP} = 1400 \left(\frac{420}{529 - 420} \right) \left(\frac{529}{420} + 1 \right) \\ = 1400 \times 3.85 \times 2.26 = 12200 \text{ psi}$$

The above values are plotted in Figure 1

B. THERMAL STRESS

The thermal stress at any point (x) in a thick walled cylinder depends on the temperature distribution in the wall and can be computed from the following expression. The maximum thermal stress is the tangential component at the inside surface.

$$S_{TT} = \left(\frac{\alpha E}{1 - \nu} \right) \left(\frac{1}{(x + a)^2} \right) \int_0^x T(a + x) dx \\ + \left(\frac{\alpha E}{1 - \nu} \right) \left[\left(1 + \left(\frac{a}{x + a} \right)^2 \right) \left(\frac{1}{b^2 - a^2} \right) \int_0^c T(a + x) dx - T \right]$$

for $x = 0$

$$S_{TT} = \left(\frac{\alpha E}{1 - \nu} \right) \left(\frac{2}{b^2 - a^2} \right) \int_0^c T(a + x) dx$$

where α = coef. of thermal expansion = 7.95×10^{-6}

E = Young's modulus = 23.3×10^6

ν = Poissons ratio = 0.3

a = inside radius = $\frac{20.5}{12} = 1.71'$, $a^2 = 2.92 \text{ FT}^2$

b = outside radius

c = wall thickness = $(b - a)$

T = temperature difference in the wall

FIGURE 1

HRK 7-9-56

In terms of x

$$T = \frac{Q_0}{k \mu_e^2} (1 - e^{-\mu_e x} - x \mu_e e^{-\mu_e x})$$

where Q_0 = incident energy

k = thermal conductivity

= 26.3 Btu/ft²-hr-F°/ft

μ_e = energy absorption coef.

for gammas

Then for the APPR-1 configuration where $x = 0$

$$S_T = \left(\frac{\infty E}{1 - \nu} \right) \left(\frac{Q_0}{k \mu_e^2} \right) \left(\frac{2}{b^2 - a^2} \right) \left[a \left(c - \frac{1}{\mu_e} + e^{-\mu_e c} \left(\frac{1}{\mu_e} - \frac{\mu_e c^2}{2} \right) \right) \right. \\ \left. + \left(\frac{c^2}{2} - \frac{1}{\mu_e^2} + e^{-\mu_e c} \left(\frac{\mu_e c + 1}{\mu_e^2} - \frac{\mu_e c^3}{3} \right) \right) \right]$$

The absorption of radiation energy in the pressure vessel wall is shown graphically in Figure 2. Values were obtained from the expression

$$Q = Q_0 (e^{-\mu_e x} f(R))$$

where $f(R)$ is the geometrical factor

$$\text{for a cylinder} = \left(\frac{a}{a + x} \right)$$

For Group 1 gammas $\mu_e = 0.206 \text{ cm}^{-1}$, $a = 1.71$

$$Q_0 = 16.7 \times 10^{11} \times 1.55 \times 10^{-8} = 25900 \text{ Btu/ft}^3\text{-hr}$$

FIGURE 2

x (cm)	μ_{e^x}	$e^{-\mu_{e^x}}$	f(R)	Q
0	0	1	1	25900
2.54	0.523	0.593	0.953	14600
5.08	1.047	0.351	0.910	8260
7.62	1.570	0.208	0.878	4740

Other energy groups were calculated in a like manner. An effective value of the energy absorption coefficient (A) can be computed from the total energy attenuation shown in Figure 2.

$$e^{-Ax} = \frac{15600}{76700} = 0.2035$$

$$Ax = 1.59$$

$$A = \frac{1.59}{0.25} = 6.36 \text{ ft}^{-1}$$

The effective Q_0 for the pressure vessel only may also be read from Figure 2.

The thermal stress now is obviously a function of pressure vessel wall thickness only.

$$\frac{\alpha E}{1-\nu} = \frac{7.95 \times 23.6}{0.7} = 278$$

$$\frac{Q_0}{k A^2} = \frac{67500}{26.3 \times 6.36^2} = 63.5$$

$$S_{TT} = 17650 \left(\frac{2}{b^2 - 2.92} \right) \left[1.71(c - 0.157 + e^{-6.36c}(0.157 - 3.18 c^2)) \right. \\ \left. + \left(\frac{c^2}{2} - 0.0247 + e^{-6.36c} \left(\frac{6.36c + 1}{6.36^2} - 2.12 c^3 \right) \right) \right]$$

$$\text{For } c = 2 \text{ in.} = 0.167 \text{ ft} \quad b = 1.875 \text{ ft} \quad b^2 = 3.52$$

$$e^{-6.36c} = e^{-6.36 \times 0.167} = e^{-1.06} = 0.3465$$

$$S_{TT} = \frac{17650 \times 2}{0.6} \left[1.71(0.167 - 0.157 + 0.3465(0.157 - 0.0884)) \right. \\ \left. + (0.0139 - 0.0247 + 0.3465(0.051 - 0.0046)) \right]$$

$$S_{TT} = 58800 (1.71 (0.01 + 0.3465 \times 0.069) + (-0.0108 + 0.3465 \times 0.046))$$

$$S_{TT} = 58800 (1.71 (0.01 + 0.24) + (-0.0108 + 0.016))$$

$$S_{TT} = 58800 (0.0581 + 0.0052)$$

$$S_{TT} = 58800 \times 0.0633 = 3720 \text{ psi}$$

$$\text{For } c = 2.5 \text{ in} = 0.208 \text{ ft} \quad b = 1.92 \text{ ft} \quad b^2 = 3.68 \text{ ft}^2$$

$$e^{-6.36 \times .208} = e^{-1.32} = 0.267$$

$$S_{TT} = \frac{17650 \times 2}{0.76} \left[1.71(0.208 - 0.157 + 0.267(0.157 - 0.138)) \right. \\ \left. + (0.0216 - 0.0247 + 0.267(0.0573 - 0.0191)) \right]$$

$$S_{TT} = 46500 (1.71(0.051 + 0.267 \times 0.019) + (-0.0031 + 0.267 \times 0.0382))$$

$$S_{TT} = 46500 (1.71(0.051 + 0.00507) + (0.0102 - 0.0031))$$

$$S_{TT} = 46500 (1.71 \times 0.05607 + 0.0071)$$

$$S_{TT} = 46500 (0.0959 + 0.0071)$$

$$S_{TT} = 46500 \times 0.1030 = 4790 \text{ psi}$$

For $c = 3 \text{ in} = 0.25 \text{ ft}$ $b = 1.96 \text{ ft}$, $b^2 = 3.84$

$$e^{-6.36 \times 0.25} = e^{-1.59} = 0.204$$

$$S_{TT} = \frac{17650 \times 2}{0.02} \left[1.71(0.25 - 0.157 + 0.204(0.157 - 0.199)) \right. \\ \left. + (0.03125 - 0.0247 + 0.204(0.0624 - 0.0332)) \right]$$

$$S_{TT} = 38400 (1.71(0.093 + 0.204(-0.042)) + (0.00655 + 0.204 \times 0.0292))$$

$$S_{TT} = 38400 (1.71 (0.093 - 0.0086) + (0.00655 + 0.00596))$$

$$S_{TT} = 38400 (1.71 \times 0.0844 + 0.0125)$$

$$S_{TT} = 38400 (0.1442 + 0.0125)$$

$$S_{TT} = 38400 \times 0.1567 = 6020 \text{ psi}$$

The above values of thermal stress are plotted in Figure 1.

These values added to the pressure stresses give a total stress (also plotted in Figure 1). A wall thickness of about 2.6 inches is indicated in order to meet the specified design stress of 16600 psi for a welded vessel.

IV. ALTERNATE CONFIGURATIONS

The requirement of minimum overall weight suggests the review of several alternate schemes.

A. The use of boron steel in the thermal shield would not change the weight of the shield and would reduce the energy incident on the pressure vessel due to capture gammas.

We may conservatively assume that the thermal shield may initially contain sufficient boron so that the effective boron content after 20 years is at least 1%.

For 1% boron the thermal flux attenuating constant is 2.57 rather than 0.689 for stainless steel.

The total capture gammas may be computed as in 415E2.1, and are equal to $I + I'$

$$I = \frac{\phi C e^{-\mu t}}{a(A - \mu)} (1 - e^{-t(A - \mu)})$$

$$\text{where } \phi = 2 \times 10^{13}$$

C =	<table><tr><th>Group</th><th>C</th></tr><tr><td>1</td><td>0.0125</td></tr><tr><td>3</td><td>0.0219</td></tr><tr><td>5</td><td>0.0329</td></tr><tr><td>7</td><td>0.1035</td></tr></table>	Group	C	1	0.0125	3	0.0219	5	0.0329	7	0.1035
Group	C										
1	0.0125										
3	0.0219										
5	0.0329										
7	0.1035										

$$t = 2 \text{ inches} = 5.08 \text{ cm}$$

$$A = 2.57$$

Group	$e^{-\mu t}$	$(A - \mu)$	$t(A - \mu)$	$e^{-t(A - \mu)}$	$(1 - e^{-t(A - \mu)})$	I
1	0.083	2.08	10.6	0.00002.5	0.99997	5×10^9
3	0.237	2.287	11.6			2.27×10^{10}
5	0.289	2.326	11.8			4.09×10^{10}
7	0.307	2.338	11.9			1.36×10^{11}

For I' $\phi' = 3.3 \times 10^{12} \times e^{-(2.57 \times 5.08)}$

$$= 3.3 \times 10^{12} \times 2.02 \times 10^{-6} = 6.68 \times 10^6$$

$$A = -2.57$$

Group	$e^{-\mu t}$	$(A - \mu)$	$t(A - \mu)$	$(1 - e^{-t(A - \mu)})$	I'
1	0.083	-3.06	-15.6	-5.95×10^6	6.74×10^9
3	0.237	-2.853	-14.5	-1.98×10^6	1.21×10^{10}
5	0.289	-2.814	-14.3	-1.63×10^6	1.84×10^{10}
7	0.307	-2.802	-14.2	-1.47×10^6	5.56×10^{10}

Group	I + I'	Attenuation	$I_{9.25}$
1	1.17×10^{10}	$2.45/2.82$	0.102×10^{11}
3	3.48×10^{10}	$9.3/10.5$	0.308×10^{11}
5	5.93×10^{10}	$1.5/1.7$	0.523×10^{11}
7	1.92×10^{11}	$5/5.6$	1.71×10^{11}

The above values for the capture gammas from a boron steel thermal shield, combined with other gamma sources gives the following results.

Group (Energy)	1	3	5	7
Total Photons ($\times 10^{-11}$)	80.63	15.70	5.38	8.21
Mev/cm ² -sec	80.63	47.1	26.9	57.4
μ_e	0.206	0.176	0.178	0.185
Mev/cm ³ -sec	16.6	8.3	4.8	10.6
Total				40.3
Btu/ft ³ -hr				62500

The thermal stress resulting from the above Q_0 can be found from previously calculated stresses and the ratio of the incident energies.

$$\frac{Q_0(B)}{Q_0} = \frac{62500}{76700} = 0.815$$

c	S_{TT}
2.0	$3720 \times 0.815 = 3030 \text{ psi}$
2.5	$4790 \times 0.815 = 3900 \text{ psi}$
3.0	$6020 \times 0.815 = 4900 \text{ psi}$

Figure 3 shows the stress relations for a 1% boron loaded thermal shield.

B. For a constant wall thickness, Figure 4 shows the relation of total stress and pressure vessel diameter. The relative flatness of these curves immediately leads to the suggestion that for minimum weight an even smaller (< 40.5 in.) diameter be considered. This could be accomplished in several ways, (a) substitute canned lead of equal weight for an all stainless steel thermal shield, (b) reduce the thermal shield diameter,

FIGURE 3

HRX 7-9-56

PRESSURE VESSEL STRESS AS A FUNCTION OF DIAMETER

FIGURE 4

WPK 7-12-56

or (c) some combination of (a) and (b).

(a) For equal masses, lead has a somewhat higher absorption coefficient and lower buildup factor than stainless steel. In addition secondary gamma generation is much lower.

If we assume the lead is canned in 1/8 inch stainless steel, the lead thickness for equivalent weight can be approximated from the expression

$$\rho_{pb} (R^2 - 17.75^2) = \rho_{Fe} (19.5^2 - 17.75^2)$$

$$R = \sqrt{\frac{7.91}{11.3} (65) + 315} = \sqrt{360.5} = 19 \text{ inches}$$

The approximate lead thickness is then 1 1/4 inches and the total thermal shield thickness would be 1 1/2 inches. This would allow the I.D. of the pressure vessel to be reduced to 39.5 inches.

As before the primary gammas can be estimated from the expression (from Astra report no. 415E2.1)

$$I_{8.75} = \frac{S_v B R^2 F(\theta, b)}{2(a + z)}$$

$$a = 8.75 \text{ inches} = 22.2 \text{ cm}$$

$$t_{ss} = 3/8 \text{ inch total} = 0.953 \text{ cm.}$$

$$t_{pb} = 1 1/4 \text{ inches} = 3.18 \text{ cm}$$

Group	z	(a + z)	$\tan \theta = \left(\frac{h}{2(a+z)} \right)$	θ
1	12.2	34.4	0.813	39°
3	14.2	36.4	0.77	37.5°
5	14.7	36.9	0.76	37°
7	14.9	37.1	0.755	37°

$$b = (0.842 \mu_w (a - t) + \mu_c z + \mu_{ss} t_{ss} + \mu_{Pb} t_{Pb})$$

Group	$(0.842\mu_w(a-t) + \mu_{cz})$	Sum	$\mu_{ss}t_{ss}$	$\mu_{Pb}t_{Pb}$	b	
1	1.067	1.96	3.027	0.466	2.54	6.033
3	0.594	1.30	1.894	0.270	1.50	3.664
5	0.457	1.16	1.617	0.232	1.56	3.409
7	0.396	1.12	1.516	0.221	1.66	3.397

Group	B_w	B_{ss}	B_{Pb}	B_{Total}
1	5.7	1.5	1.8	15.4
3	2.3	1.1	1.5	3.8
5	1.83	1.1	1.35	2.72
7	1.63	1.1	1.27	2.28

Group	S_v	$2(a+z)$	B	R^2	$F(\theta, b)$	$I_{8.75}$
1	2.31×10^{13}	68.8	15.4	784	0.00104	4.21×10^{12}
3	7.42×10^{11}	72.8	3.8	784	0.0128	3.88×10^{11}
5	4.57×10^{11}	73.8	2.72	784	0.0167	2.2×10^{11}
7	3.94×10^{11}	74.2	2.28	784	0.0168	1.6×10^{11}

The secondary gammas can be estimated much as before. A major reduction is expected in gammas from the thermal shield.

Gammas from the flow separator will be attenuated by 7.125 inches of water, 0.25 inch of stainless steel, and 1.25 inches of lead. Water attenuation can be estimated from Figure 11 of 415E2.1. The thermal shield attenuation can be found from the conventional expression

$$I/I_0 = f(R) B e^{-\mu x}$$

Group	f(R)	μx (S.S.+Pb)	$e^{-\mu x}$	B	I/I_0
1	0.85	3.01	0.050	2.7	0.115
3	0.85	1.77	0.170	1.65	0.238
5	0.85	1.79	0.167	1.5	0.213
7	0.85	1.88	0.152	1.4	0.182

Group	$I_{0.125}$	$ATT_{T.S.}$	ATT_w	$I_{8.75}$
1	4.78×10^{10}	0.115	0.223	1.23×10^9
3	8.66×10^{10}	0.238	0.307	6.32×10^9
5	1.31×10^{11}	0.213	0.319	8.9×10^9
7	4.12×10^{11}	0.182	0.337	2.52×10^{10}

Water capture gammas can be estimated similarly.

Slab	I_1	I_3	$I_{6.5}$	$ATT_{T.S.}$	ATT_w	$I_{8.75}$
1	4.14×10^{11}			0.21	0.33	2.87×10^{10}
2		1.86×10^{12}		0.21	0.47	1.84×10^{11}
3			1.73×10^{12}	0.21	0.87	3.16×10^{11}

We can assume the contribution of slab 4 is just twice that computed on page 9. i.e. 8×10^{10} . The total water gamma

contribution is then:

$$I_{8.75} = (0.29 + 1.84 + 3.16 + 0.8) \times 10^{11}$$

$$= 6.09 \times 10^{11}$$

The contribution of capture gamma from the thermal shield result primarily from the stainless steel can. An approximation can be easily obtained from the expression

$$I = \phi C t$$

for the inside steel $\phi = 2 \times 10^{13}$ and for the outside $\phi \approx 6.6 \times 10^{12}$. Gammas from the inside steel will be attenuated by 1 1/4 in. lead 1/8 in. steel and 3/4 in. of water. Outside steel gammas will be attenuated by 3/4 in. of water only

INSIDE: $\phi = 2 \times 10^{13}$ $t = 0.318$ cm

Group	C	I	ATT _{TS}	ATT _w	I _{8.75}
1	0.0125	8×10^{10}	~ 0.15	0.86	1×10^9
3	0.0219	1.4×10^{11}	~ 0.25	0.88	3.1×10^{10}
5	0.0329	2.1×10^{11}	~ 0.23	0.88	4.2×10^{10}
7	0.1035	6.6×10^{11}	~ 0.19	0.89	1.1×10^{11}

OUTSIDE: $\phi = 6.6 \times 10^{12}$ $t = 0.318$ cm

Group	I	ATT _w	I _{8.75}
1	2.6×10^{10}	0.86	2.24×10^{10}
3	4.5×10^{10}	0.88	4.05×10^{10}
5	6.9×10^{10}	0.88	6.1×10^{10}
7	2.2×10^{11}	0.89	1.96×10^{11}

For the lead the gamma contribution is all in the 7 Mev group - one photon per absorption.

$$I = \frac{\phi C e^{-\mu t}}{2(A - \mu)} (1 - e^{-t(A - \mu)})$$

$$\phi \approx 2 \times 10^{13}$$

$$C_{Pb} = 0.007$$

$$N = 0.52$$

$$A = \frac{1}{L} = \sqrt{3 \Sigma_s \Sigma_a}$$

$$A = \sqrt{3 \times 0.26 \times 0.007} = \sqrt{0.0055} = 0.074$$

$$I = \frac{2 \times 10^{13} \times 0.007 \times e^{-1.66}}{2(0.074 - 0.52)} (1 - e^{-3.18(0.074 - 0.52)})$$

$$I_{8.75} = 7 \times 10^{10} \times \frac{0.190}{(-0.446)} (1 - e^{1.42}) \times ATT.$$

$$I_{8.75} = 7 \times 10^{10} \times 0.426 \times 3.14 \times 0.86 = 8.1 \times 10^{10}$$

The total gammas from the thermal shield are then:

Group	I	I _o	I _{Pb}	I _{8.75}
1	0.1 x 10 ¹⁰	2.24 x 10 ¹⁰	-	0.23 x 10 ¹¹
3	3.1 x 10 ¹⁰	4.05 x 10 ¹⁰	-	0.72 x 10 ¹¹
5	4.2 x 10 ¹⁰	6.1 x 10 ¹⁰	-	1.03 x 10 ¹¹
7	11.0 x 10 ¹⁰	19.6 x 10 ¹⁰	8.1 x 10 ¹⁰	3.87 x 10 ¹¹

Capture gammas from the pressure shell are proportional to the incident neutron flux.

Group	$I_{12.75}$	Flux Ratio	$I_{8.75}$
3	4.4×10^{10}	6/2	1.32×10^{11}
5	6.4×10^{10}	"	1.92×10^{11}
7	1.74×10^{11}	"	5.22×10^{11}

Inelastic scatter gammas are nearly comparable to those from stainless steel.

RECAPITULATING

Gamma Energy	1	3	5	7
Primary	42.1×10^{11}	3.88	2.2	1.6
Capt. Flow Separator	0.012	0.063	0.089	0.252
Capt. Thermal Shield	0.23	0.72	1.03	3.87
Capt. Water	--	6.09	--	--
Capt. Pressure Vessel	--	1.32	1.42	5.22
Inelastic Scatter	1.8	--	--	--
TOTAL	44.14	12.07	5.24	10.94
Mev/cm ² -sec	44.14	36.2	26.2	76.5
μ_e	0.206	0.176	0.178	0.185
Mev/cm ³ -sec	9.1	6.36	4.66	14.1
TOTAL				34.22
Btu/ft ³ -hr				53000

The pressure stress for 39.5 I.D. shell will be:

For a 2 in. wall

$$\begin{aligned} S_{TP} &= 1400 \left(\frac{20^2}{22^2 - 20^2} \right) \left(\frac{22^2}{20^2} + 1 \right) \\ &= 1400 \left(\frac{400}{84} \right) \left(\frac{484}{400} + 1 \right) \\ &= 1400 \times 4.76 \times 2.21 = 14700 \text{ psi} \end{aligned}$$

For a 2 1/2 in. wall

$$\begin{aligned} S_{TP} &= 1400 \left(\frac{400}{506 - 400} \right) \left(\frac{506}{400} + 1 \right) \\ &= 1400 \times 3.77 \times 2.295 = 11900 \text{ psi} \end{aligned}$$

For a 3 in. wall

$$\begin{aligned} S_{TP} &= 1400 \left(\frac{400}{529 - 400} \right) \left(\frac{529}{400} + 1 \right) \\ &= 1400 \times 3.1 \times 2.32 = 10100 \text{ psi} \end{aligned}$$

The thermal stresses for this case may be computed as before

$$\frac{Q_0}{k A^2} = \frac{46800}{26.3 \times 6.36} = 44.0$$

$$S_{TT}(2") = 278 \times 44 \left(\frac{2}{0.59} \right) (1.67 \times 0.034 + 0.0052)$$

$$= 12230 \times 3.39 \times (0.0567 + 0.0052)$$

$$S_{TT} = 12230 \times 3.39 \times 0.0619 = 2560 \text{ psi}$$

$$S_{TT}(2.5") = 12230 \left(\frac{2}{0.75} \right) (1.67 \times 0.0561 + 0.0071)$$

$$= 12230 \times 2.67 (0.0937 + 0.0071)$$

$$S_{TT} = 12230 \times 2.67 \times 0.101 = 3300 \text{ psi}$$

$$S_{TT}(3") = 12230 \left(\frac{2}{0.90} \right) (1.67 \times 0.0844 + 0.0125)$$

$$S_{TT} = 12230 \times 2.22 \times 0.1535 = 4170 \text{ psi}$$

The above results are shown graphically in Figure 5.

FIGURE 5