R & D ASSOCIATES
Post Office Box 9695
Marina del Rey,
California 90291

COMMISSION
CORRESPONDENCE

25 July 1980

Nuclear Regulatory Commission
M/S 1149
Washington, D. C. 20555

Attention: Commissioner Victor Gilinsky

Dear Victor:

Our critique of the Ames analysis of the Sequoyah containment is enclosed. Please call if you have any questions.

Sincerely,

Harmon

Harmon W. Hubbard

HWH/dl

Enclosure: "Sequoyah Containment Analysis," July 1980, (2 cys.).

8008080082

SEQUOYAH CONTAINMENT ANALYSIS

1. INTRODUCTION

This letter report is in response to a request from the U.S. Nuclear Regulatory Commission to review and critique the ultimate strength analyses of the Sequoyah containment.

The description of the containment vessel and the analysis for review were provided in the NRC Information Report dated 22 April 1980, Ref. SECY-80-107A. The tasks requested in the work statement were as follows:

1. To what extent are the assumptions in the analyses conservative?
2. To what extent is the calculated ultimate strength conservative?
3. What are the uncertainties in the analyses, methods, and models?
4. To what extent is there assurance of no gross leakage from the vessel at stresses above the design stress and yield stress?
5. How would the analyses and results be altered if the stresses are caused by ignition/detonation of 300-600 kg of hydrogen distributed uniformly and nonuniformly in the containment?
6. To what extent can distributed ignition sources mitigate the effects of hydrogen?

This report will cover the first four tasks of the work statement. A report on the hydrogen problem, tasks 5 and 6, will be issued separately. A preliminary briefing of the analyses conducted by RDA was given to Commissioner Gilinsky and Dr. J. Austin at RDA on 18th July 1980.

2. BACKGROUND - SEQUOYAH CONTAINMENT VESSEL DESIGN

The containment vessel for Sequoyah is a low-leakage, free-standing steel structure consisting of a cylindrical wall, a hemispherical dome, and a bottom liner plate encased in concrete. Figure 1 shows the outline and configuration of the containment vessel.

The structure consists of side walls measuring 113 feet 8-5/8 inches in height from the liner on the base to the spring line of the dome and has an inside diameter of 115 feet. The bottom liner plate is 1/4 inch thick, the cylinder varies from 1-3/8 inch thickness at the bottom to 1/2 inch thick at the spring line and the dome varies from 7/16 inch thickness at the spring line to 15/16 inch thickness at the apex.

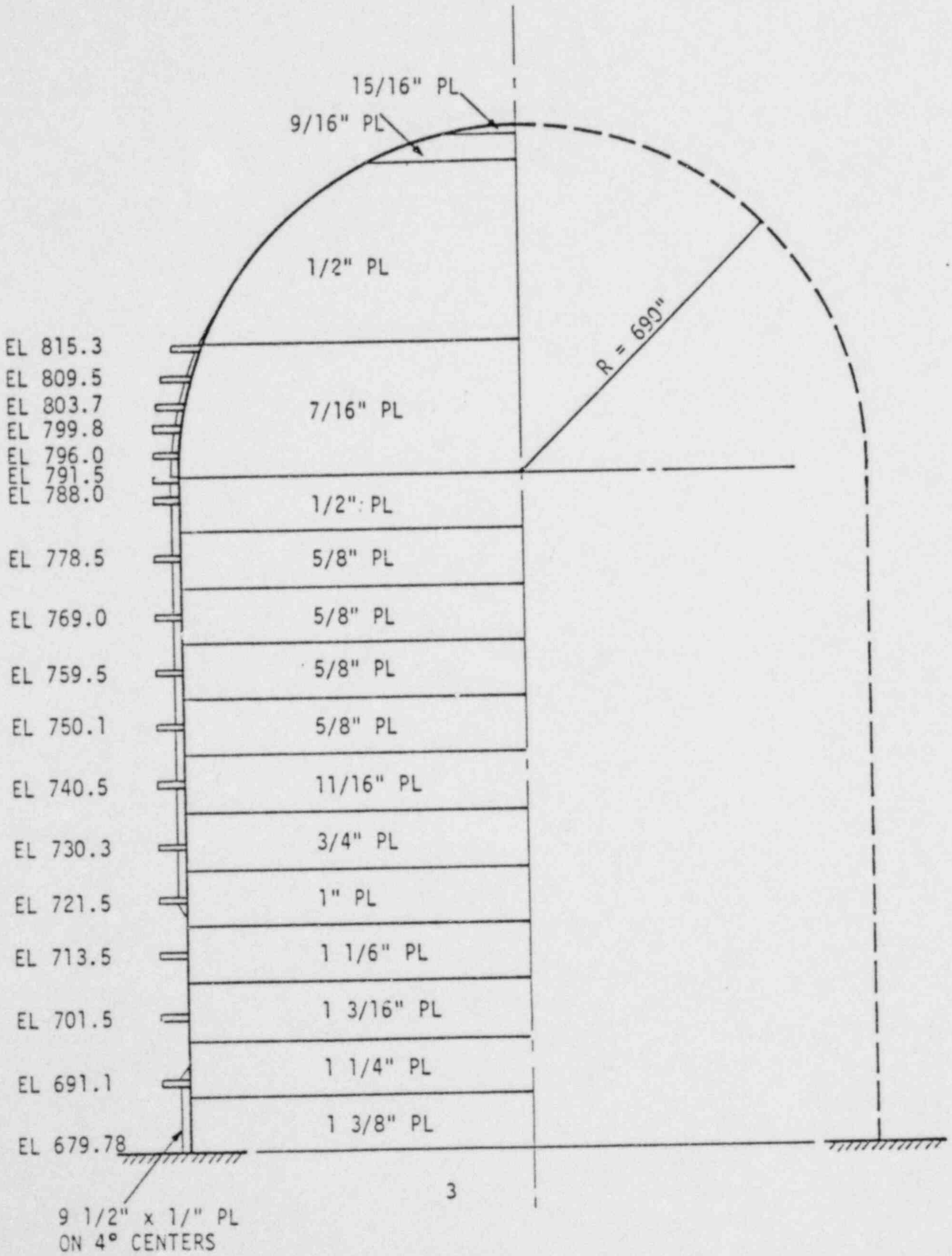
The containment vessel is provided with both circumferential and vertical stiffeners on the exterior of the shell. These stiffeners are required to satisfy design requirements for expansion and contraction, seismic forces, and pressure transient loads. The circumferential stiffeners were installed on approximately 20-foot centers during erection to insure stability and alignment of the shell. Vertical stiffeners are spaced at 4 degrees and other locally stiffened areas are provided for penetration, etc., as required

The design of the containment vessel was to the requirements of the ASME Code, Section III, Subsection B. The Code includes cases 1177-5, 1290-1, 1330-1, 1413, 1431, and the Winter 1968 Addenda.

The following pressures and temperatures were used in the design of the vessel:

Overpressure test (1)	13.5 psig
Maximum internal pressure (2)	12.0 psig at 200°F
Design internal pressure (2)	10.8 psig at 220°F

Figure 1. Sequoyah Containment Vessel



Leakage rate test pressure	12.0 psig
Design external pressure	0.5 psig
Lowest service metal temperature	30° F
Operating ambient temperature	120° F
Operating internal temperature	120° F

(1) 1.25 times design internal pressure as required by ASME Code, UG-100(b).

(2) See Paragraph N-1312(2) of Section III of the ASME Code which states that the "design internal pressure" of the vessel may differ from the "maximum containment pressure" but in no case shall the design internal pressure be less than 90 percent of the maximum containment internal pressure.

The steel plate used is to ASME specifications SA-516 grade 60 with a yield stress of 32,000 psi, an ultimate stress of 60,000 psi and a Young's Modulus, E, of 28 x 10^6 psi at 70°F. For the above code, the maximum shear stress criterion yields an equivalent maximum membrane principal stress, in the hoop direction, given by:

$$\text{hoop stress} = \frac{PR}{t} = \text{allowable stress, where } P = 10.8 \text{ psi}$$
$$R = 690 \text{ in.}$$

(the given allowable stress in the 1977 version of the code is 16,500 psi (i.e., approximately 1/2 the yield stress)).

Hence, $$t = \frac{10.8 \times 690}{16,500} = 0.452 \text{ in.}$$

Thus, the minimum plate thickness of 1/2 inch satisfies the basic code requirements.

Originally the vessel was designed with only seven ring stiffeners and local vertical stiffeners at penetration regions. Detailed buckling analysis and seismic excitation analysis showed, however, that additional rings and vertical stiffeners would be required and the final configuration of Figure 1 resulted. It should be noted that the longitudinal, or meridional, stresses in a cylindrical membrane are only half of the hoop stress and hence do not contribute to the maximum shear criterion of the ASME Code. Further the dome stresses are all of the same type ("meridional" as opposed to "hoop") and hence with the plate thicknesses used the dome membrane stresses are much less than the critical cylindrical stresses.

3. THE ANALYSIS OF A SHELL WITH RING AND STRINGER STIFFENERS

The application of rings and stiffeners to a membrane structure is well known in aircraft structural anlaysis and must be treated with caution since local bending stresses can be induced. It was noted that the analysis provided in the reference document SECY-80-107A used a "smearing" technique whereby the rings and longitudinal stiffeners (or "stringers") are smeared out over the membrane thickness thereby increasing the effective thickness of the membrane and hence its pressure capability. It is well known, however, in aircraft structural analysis that in general this cannot be done. The problem is succinctly described in the following extract from "Analysis and Design of Flight Vehicle Structures," E. F. Bruhn, Purdue University, Tri-State Offset Company, 1965. (Library of Congress Card #64-7896).

Because of functional requirements over and above those of a simple pressure vessel, the pressurized cabin shell of an airplane has a number of stress analysis problems peculiar to its configuration. Several of the more general of these will be considered here.

To stabilize the shell wall in transmitting heavy tail loads through the fuselage, longitudinal stringers are added. These same stringers will also help to carry the meridional pressure loads. The skin and stringers must, of course, have equal <u>strains</u> in the longitudinal directions but, because the skin is in a two-dimensional state of stress, they cannot have equal longitudinal <u>stresses</u>: hence the following analysis.

Let the meridional (longitudinal) stresses in the skin and stringers be S_M and S_L, respectively. S_t will be the tangential (hoop) stress in the skin. We have

$$S_t = \frac{PR}{t}$$

If N is the total number of stringers, each of cross sectional area A_L, then equilibrium longitudinally requires

$$P \pi R^2 = 2 \pi R t S_M + NA_L S_L.$$

The condition of equal longitudinal strain in the skin and stringers yields

$$E \varepsilon = S_L = S_M - \mu S_t$$

where μ is Poisson's ratio (= .27 for steel).

Solving these three equations one finds

$$S_t = \frac{PR}{t}$$

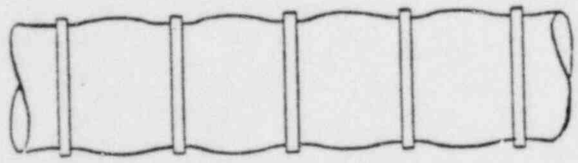
$$S_M = \frac{PR}{2t} \frac{(1 + 2\mu\alpha)}{(1 + \alpha)} = \frac{PR}{2t} \frac{(1 + 0.54\,\alpha)}{(1 + \alpha)}$$

$$S_L = \frac{PR}{2t} \frac{(1 - 2\,\mu)}{(1 + \alpha)} = \frac{PR}{2t} \frac{0.46}{(1 + \alpha)}$$

where $\alpha = NA_L/2\pi Rt$ is the ratio of total stringer area to skin area. A little study will show that $t(1 + \alpha)$ is a sort of "effective shell wall thickness": it is the result of taking all the cross sectional area (skin plus stringers) and distributing it uniformly around the perimeter. On this basis, the results are a little disappointing: the stringers are carrying only 40% of the stress one might expect if the net longitudinal load ($P \pi R^2$) were distributed evenly over the entire cross sectional area ($2 \pi Rt (1 + \alpha)$). Thus the meridional skin stresses are reduced by the factor $(1 + .6 \alpha)/(1 + \alpha)$ from what they would be without the stringers.

Because of the necessity for transmitting various concentrated loads from within the cabin and from the wings and tail to the main shell and because it is also necessary to provide some lateral restraint which will stabilize the stringers and skin against an overall instability failure, the pressurized fuselage of an airplane contains a considerable number of rings and frames distributed along the length of the shell. These rings are seldom, if ever, spaced closely enough such that they can be considered effective in carrying a part of the hoop stresses (in the way the stringers were effective in carrying part of the meridional stress). Rather, they act more like widely spaced restraining bands having the effect shown exaggerated in Figure 2.

Figure 2. Restraining rings along a pressurized tank. The action is representative of a fuselage with widely spaced rings inside.



It is obvious that the rings in this case will produce secondary bending stresses in the skin and hence may have a detrimental effect on the simple membrane stress system.

Equally harmful are the tensile loadings developed in the rivets joining the skin and rings. (End of Extract)

4. STRINGER EFFECTIVENESS

Following the method of Section 3 above and Figure 3 illustrates the application of the longitudinal stiffeners to the Sequoyah vessel. In calculating the meridional stresses an "effective" pressure is used, which is the internal pressure of the container less that pressure which is needed to support the structural weight above the section under consideration. Thus, at the critical 1/2 inch plate section (top of the cylinder) a dome weight of about 550,000 lb has to be supported and this is equivalent to an internal pressure of about 0.37 psi, and the internal pressure has to exceed this value before a meridional tension stress can be achieved. At the base the equivalent pressure to offset the overall weight of the container (about 2.3 million lb) is 1.54 psi.

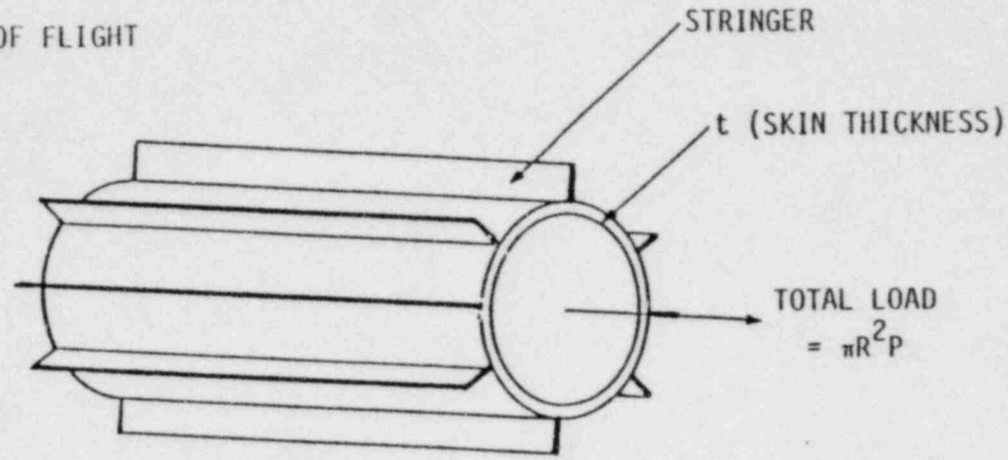
It is seen from Figure 3 that the stringers are stressed to only about 40% of the amount of the meridional stress in the membrane. Of the total longitudinal load the membrane carries 93% and the stringers only 7%. It is therefore clearly incorrect to assume that the stringer cross sectional area can be "smeared" out fully over the membrane - the smearing technique can be used but by using about 40% of the stringer cross sectional area.

5. RING STIFFENER EFFECT

The analysis of thin walled cylinders with ring stiffeners is treated in detail in "Beams on Elastic Foundation" by M. Hetenyi (University of Michigan Press 1946) pages 83-84. Figure 4 shows the results of this analysis applied to the cylindrical section of the Sequoyah vessel. It is seen that the ring stiffeners have to be spaced very much closer than 80 inches to have any appreciable reduction on the membrane

Figure 3. Stringer Effectiveness

- REF "ANALYSIS AND DESIGN OF FLIGHT VEHICLE STRUCTURES" BRUHN



TOTAL STRINGER C.S. AREA $= A$, STRESS $= S_L$

SKIN STRESS (MEMBRANE): S_t (HOOP) $= PR/t$, S_M (MERIDIONAL) = ?

LONGITUDINAL EQUILIBRIUM: $\pi R^2 P = 2\pi R t S_M + A S_L$

EQUAL LONGITUDINAL STRAIN $\epsilon = \frac{S_L}{E}$ (STRINGER)

$= \frac{S_M - \mu S_t}{E}$ (SKIN) ($\mu = 0.3$, POISSONS RATIO)

SOLUTION: $S_M = \frac{PR}{2t}\left[\frac{1 + 0.6a}{1 + a}\right]$

$S_L = \frac{PR}{2t}\left[\frac{0.4}{1 + a}\right]$ $\qquad a = \frac{A}{2\pi R t}$

$R = 690"$ $t = 1/2"$

$P = 12$ PSI (EFFECTIVE)

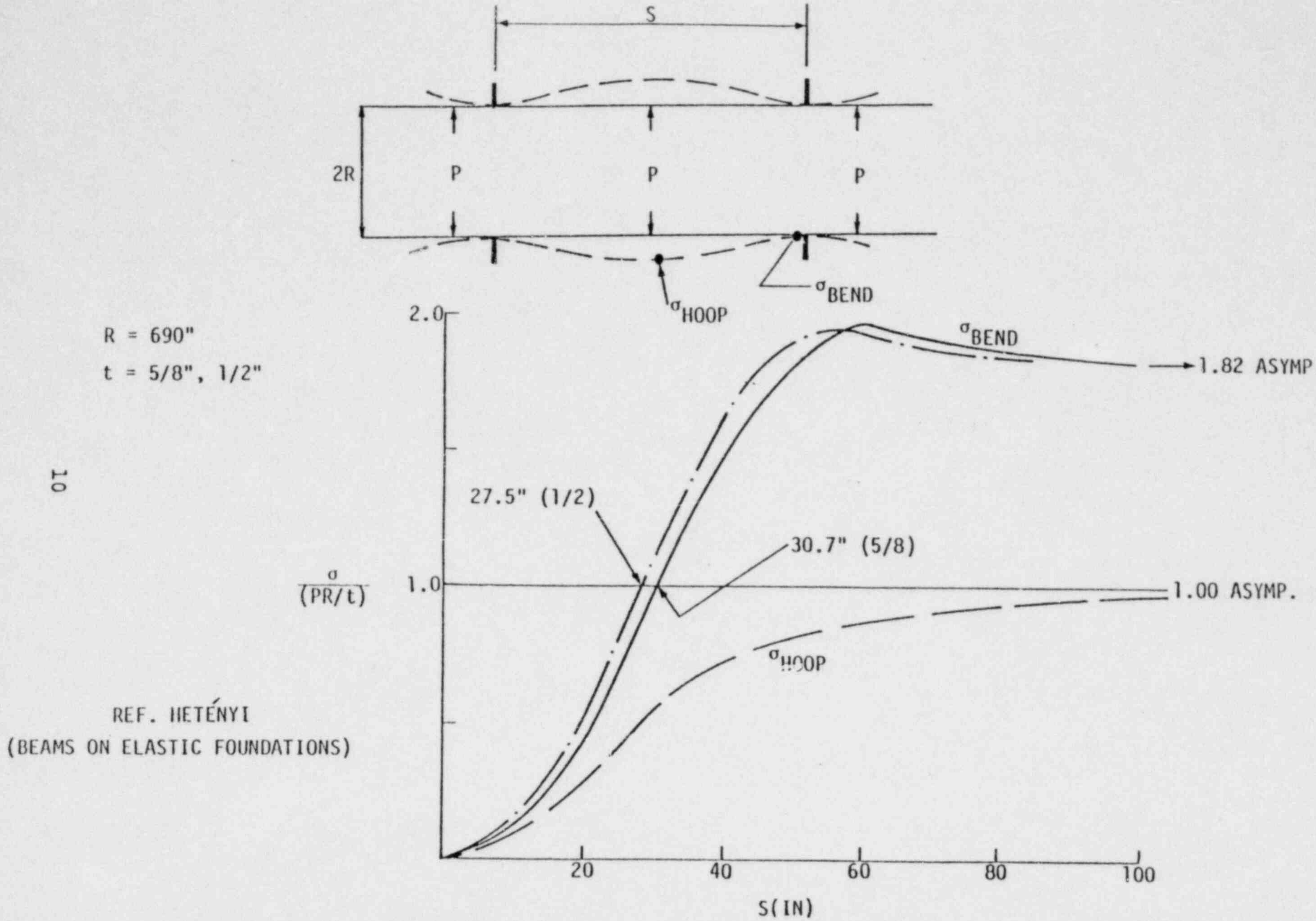
$A = 90 \times 4.75 = 427.5 \text{ in}^2$

$a = 0.197$

$\boxed{PR/2t = 8280 \quad S_M = 7235 \quad S_L = 2767}$

<u>NOTE</u>: SKIN CARRIES 16.8 MILLION LB (93%) STRINGERS 1.2 MILLION LB (7%) OF TOTAL LOAD OF 18 MILLION LB.

Figure 4. Ring Stiffener Effect



hoop stress. Further local bending stresses at the attachment to the ring which are greater than the unmodified hoop stress are generated when the ring spacing is in excess of 27.5" to 20.7" (respectively for 1/2" and 5/8" plate). Since the actual design ring spacings are at 10 ft two conclusions may be drawn:

a. Membrane hoop stresses in a considerable region between the ring stiffeners is for practical purposes not influenced by the ring stiffeners.

b. A local bending stress at the ring attachment to the shell is induced and this stress is some 80% higher then the simple membrane hoop stress.

Thus, the critical region for hoop stress will be the 1/2 inch plate midway between the two rings. (This occurs between rings at elevations 778.5 and 788.0 shown in Figure 1). This section has the upper 2/3 of 1/2 inch plate and the lower 1/3 of 5/8 inch plate, and hence the mid-section area of criticality is in the 1/2 inch plate). In this case the critical internal pressure may be calculated as follows:

yield stress (= 32,000 psi) = PR/t
(R = 690 in., t = 1/2 in.)
giving P = 23.2 psi

This corresponds to the Boiler Code Max Shear Stress Criterion for yield. If ultimate strength is used then this pressure would be scaled up in the ratio of ultimate to yield stresses (60,000 to 32,000 psi) giving a value of 43.5 psi.

The corresponding longitudinal stress would be half the hoop stress in a simple unstiffened cylinder. As shown in Section 4, the membrane longitudinal stress is reduced by a factor of 0.87 due to the presence of the stringers. An alternative method to the minimum shear stress method of the Boiler Code is to use Von Mises criteria which determines the critical stress as a function of both the hoop stress (σ_t) and the longitudinal or meridional stress (σ_M). This is given by:

$$\sigma_{crit} = \sqrt{\sigma_M^2 + \sigma_t^2 - \sigma_M\sigma_t}$$

In this case $\quad \sigma_M = 0.5 \times 0.87\sigma_t = 0.435\ \sigma_t$

Hence $\quad \sigma_{crit} = 0.868\sigma_t$

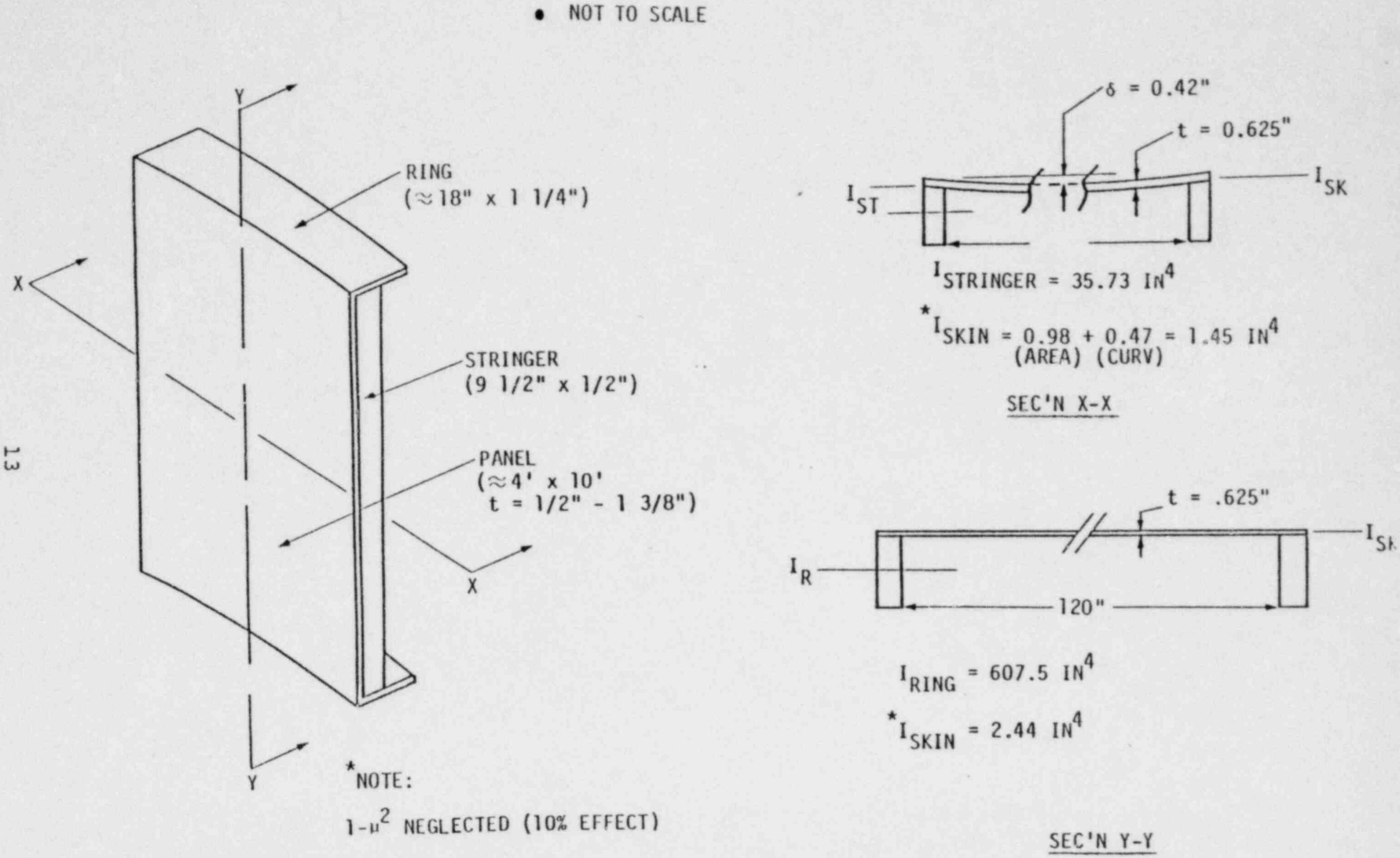
Hence, for the Von Mises criteria the critical pressures corresponding to yield and ultimate stresses are respectively 26.8 and 50.3 psi.

6. ALTERNATIVE PANEL ANALYSES

An alternative approach, in order to determine local stress regions induced by the rings and stringers, is to consider the cylinder to be a number of rectangular panels framed by ring sectors and stringer sections as shown in Figure 5. Thus, the cylinder is composed of a number of panels approximately 4 ft by 10 ft as shown with thicknesses varying from 1/2 in. to 1 3/8 in. A comparison of the bending stiffness of the panel and the rings and stringers is shown in Figure 5. The cross sectional moment of inertia about the bending axis is a measure of the stiffness of a beam. In the case of a panel bending as a beam there is an additional term due to a Poisson's Ratio (μ) contribution. This is, however, only a 10% effect (proportional to $1 - \mu^2$, and $\mu = 0.27$) and is neglected in calculating the moment of inertia of the panel.

From Figure 5 it is seen that in bending about the XX axis the stringers are over twenty times as stiff as the skin, even though the skin is curved across the bending axis thereby increasing its effective moment of inertia by some 50%. For bending about the longitudinal axis YY the relative stiffness

Figure 5. Panel Arrangement



is even higher (about 250 to 1). The analysis of Figure 5 were carried out for a 5/8 inch thick skin. The relative stiffness will be even higher for a 1/2 inch thick skin since the skin moment of inertia involves a $\bar{t}^3$ term.

It is clear from these considerations that an analysis of the skin as a panel held rigidly at the boundaries should be made (i.e., encastré edges). The legitimacy of this encastré assumption is strengthened when one considers that adjacent panels help in keeping the ring and stringer edges from twisting. For example, symmetry in the cross section across a stringer in the XX direction ensures that the stringer cannot twist for panel bending in about the YY axis.

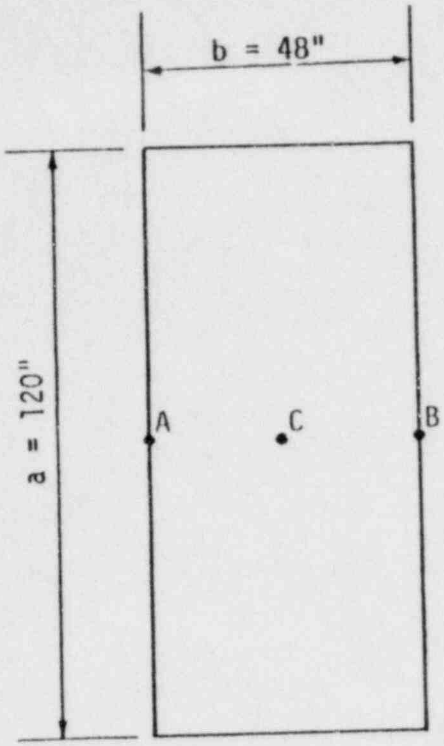
Two flat plate analysis have been carried out following the methods of "Formulas For Stress and Strain" - R. J. Roark, 5th Edition McGraw-Hill Book Co. (1975), pages 392 and 408.

a. Simple flat plate analysis

This is presented in Figure 6. For an encastré edged plate Table 8a on page 392 of the reference volume gives a value for the maximum bending stress at A & B (the midpoints of the long sides) as $\sigma = 0.5\ Pb^2/t^2$. For the plate under consideration this gives initial stresses for yielding at a pressure of 6.94 psi. At this pressure the inner plate fibers at A & B will just begin to yield in tension, and the outer plate fibers in these locations will be compressed to a stress of 32,000 psi. At a value of about 1.5 times this pressure (or 10.4 psi) yielding will occur through the entire plate section at A & B. (This is known as a "plastic hinge"). Ultimate yielding stresses of the surface fibers at A & B will be reached at a pressure of 13.0 psi.

The table, referenced above, shows that the stress at the midpoint of the plate (C in Figure 6) is half that occurring at A & B, and is in the opposite sense (i.e., tensile on the outside, compression on the inside). However the plate is not

Figure 6. Flat Plate Analysis



- REF: ROARK "FORMULAS FOR STRESS AND STRAIN" PAGE 392

MAX. σ_M (AT A, B) $\left(\begin{array}{l} b = 48" \\ t = 1/2" \end{array}\right)$

$$= \frac{0.5\ Pb^2}{t^2}$$

FOR σ = 32,000 PSI σ = 60,000 PSI

P = 6.94 PSI P = 13.0 PSI

CENTER $\sigma_C = 0.5\ \sigma_M$

FOR σ = 32,000 PSI σ = 60,000 PSI

P = 13.9 PSI P = 26.0

NOTE: FULL PLASTIC HINGE DEVELOPS AT A, B
AT 1.5 x 6.9 = 10.4 PSI

a truly "flat" plate and the analysis of Section 4 is more appropriate to the center of the plate which is mainly subject to the hoop tension. There would undoubtedly be some complex combination of bending stresses due to the ring and stringer constraints coupled with the hoop and meridional membrane stresses. A careful analysis with a finite element code would be required to resolve this point and this is beyond the scope of this review.

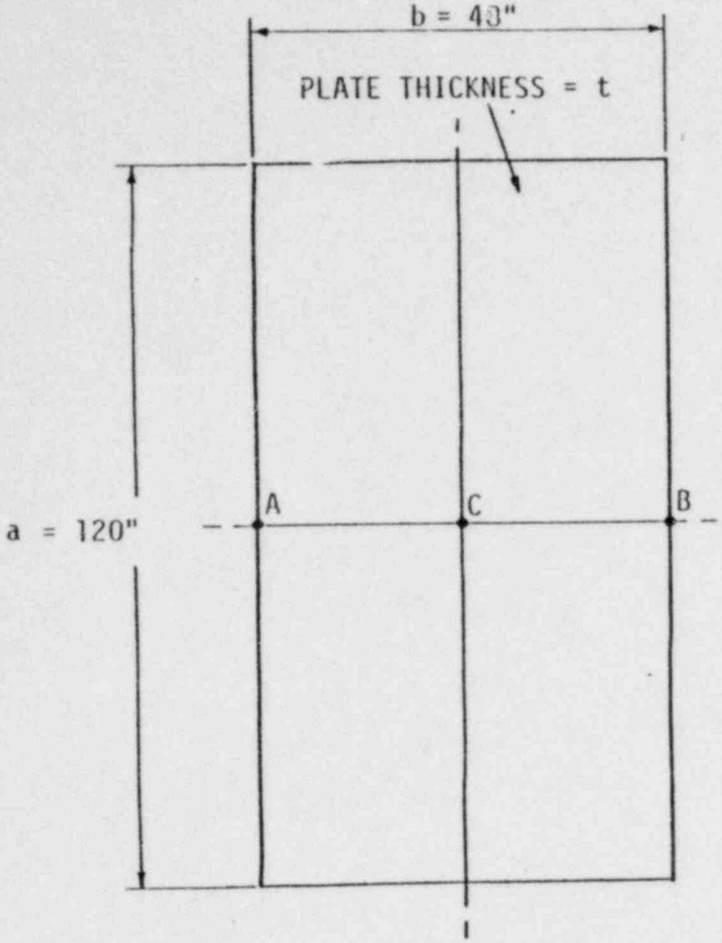
b. Large deflection plate analysis ("quilting" effect)

The analysis of (a) assumes a flat plate and makes no allowance for the finite deflections of the plate. The formula of page 408 of the referenced work makes allowance for the plate deflection. These results are summarized in Figure 7. Again maximum stresses occur at the midpoints of the long sides. The resulting stress is a combination of bending and membrane stresses. Yielding (at 32,000 psi stress) of the inner fibers at A & B begins at an internal pressure of 7.8 psi. Only 6 1/2% of the total stress is due to the membrane contribution.

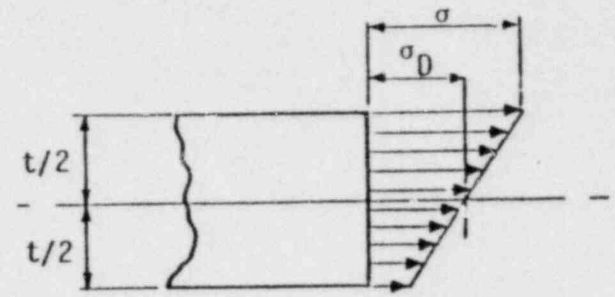
c. Comments on the maximum stress loading at A & B

The onset of yield could occur at the inner fibers at the midpoints of the long edges of the half inch plate sections at an internal pressure of 7.8 psi, assuming the more realistic "quilting" analysis. However, this is at local points only and full plastic hinging would not occur until about 11.7 psi. Even then local stress relief might well occur and for a "one-shot" pressurization it is not clear whether this would result in leakage. It would be a serious problem if many cycles of pressurization were encountered when cracking due to "LCF" (low cycle fatigue) might well occur. More serious, however, is the pure membrane stress induced in the 1/2 inch skin at 26.8 psi. This is a tension over the whole cross section of the panel and would occur over several inches of the vertical panel centerline.

Figure 7. Large Deflection Flat Plate Analysis ("Quilting Effect")



- REF: ROARK "FORMULAS FOR STRESS AND STRAIN" 5th EDITION, PAGE 408



STRESS FIELD ⟶ COMBINED TENSION AND BENDING

σ_D = DIAPHRAGM (MEMBRANE) STRESS AT A AND B

σ = TOTAL STRESS BENDING AND DIAPHRAGM AT A AND B

Y_{MAX} = OUT OF PLANE PLATE DEFLECTION AT C

$$\left.\begin{aligned} Y/t &= F_1(Pb^4/Et^4) \\ \sigma_D b^2/Et^2 &= F_2(Pb^4/Et^4) \\ \sigma b^2/Et^2 &= F_3(Pb^4/Et^4) \end{aligned}\right\}$$

DEFLECTION AND STRESS COEFFICIENTS

$E = 28 \times 10^6$ PSI

	t = 1/2"		t = 5/8"	
σ PSI	32,000	60,000	32,000	60,000
σ_D PSI	2100	5000	1500	4100
P PSI	7.8	15.9	11.8	23.2
Y_{MAX} INS	.24	.40	.20	.35

7. HOLD DOWN BOLT STRESSES

Figure 8 depicts the tension stress in the hold down bolts as the internal pressure is increased. The bolts are pre-stressed to a level of 25,000 psi and this bolt tension is not increased until the internal pressure overcomes the container weight as well as the preload tension. This occurs at an internal pressure of 17.3 psi. Increasing pressure will produce bolt yield stress at 64.5 psi and the ultimate bolt stress of 125,000 psi would be reached at an internal pressure of 77.1 psi. The latter, however, could not realistically be achieved since gross leakage would occur as soon as the bolts yield.

8. SUMMARY OF STRESS ANALYSES, CONCLUSIONS AND RECOMMENDATIONS--

Figure 9 summarizes the stress analyses described above together with the AMES "smeared" shell/stiffener analyses of SECY-80-107A.

The RDA analysis leads to the following conclusions.

a. The AMES analysis is optimistic.

1. The ring stiffeners are not amenable to the smearing technique--the spacing is such that the hoop stress in the mid-region between the rings is essentially unaffected.
2. The stringers are only partially amenable to smearing--the stringers only carry 40% of the load that would be expected with "equal" area effectiveness between membrane and stringers.
3. Having "smeared out" the rings and stringers they cannot be put back in to carry load. This leads to the rather surprising case of one of the rings being the "weak" element in the system.
4. The ultimate burst analysis is clearly incorrect--the hold down bolts would yield first.

Figure 8. Hold Down Bolt Stresses

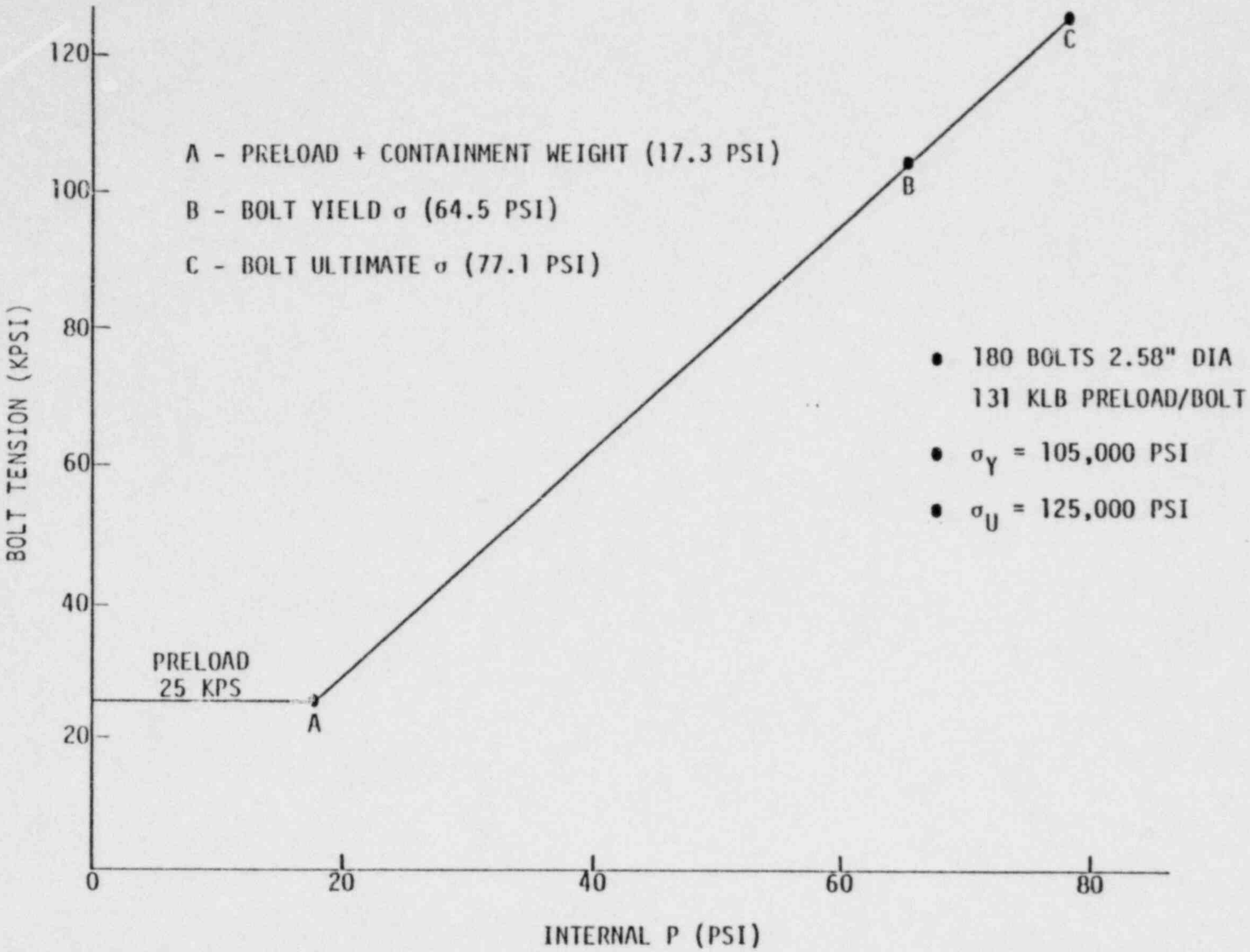


Figure 9. Sequoyah Containment Vessel - Summary of Stresses

	METHOD	CRITICAL ELEMENT	STRESSES	PRESSURE FOR YIELD STRESS PSI	PRESSURE FOR ULT. STRESS PSI
1	AMES "SMEARED" SHELL/STIFFENER ANALYSIS	1 RING STIFFENER 2 5/8" SKIN	PURE MEMBRANE (VON MISES)	1 35.6 2 38.6	1 66.7 2 72.4
2	RDA SHELL/STIFFENER ANALYSIS	1/2" SKIN	PURE MEMBRANE (BOILER CODE-MAX. SHEAR STRESS/ VON MISES)	23.2/26.8	43.5/50.3
3	RDA FLAT PLATE ANALYSIS (ENCASTRE' EDGES)	1/2" SKIN 1 MIDDLE OF LONG-EDGE 2 CENTER OF PLATE	PURE BENDING (BOILER CODE-MAX SHEAR STRESS) A) YIELD AT MAX FIBER B) FULL PLASTIC HINGE	1 A) 6.9 B) 10.4 2 A) 13.9 B) 20.85	1 13.0 2 26.0
4	RDA LARGE DEFLECTION FLAT PLATE ANALYSIS (ENCASTRE´ EDGES)	1/2" SKIN MIDDLE OF LONG-EDGE	COMBINED BENDING AND TENSION (BOILER CODE-MAX SHEAR STRESS) A) YIELD AT MAX FIBER B) FULL PLASTIC HINGE	A) 7.8 B) 11.7	15.9

MATERIAL ASME SA 516 GRADE 60

- o YIELD STRESS 32,000 PSI
- o ULTIMATE STRESS 60,000 PSI
- o E 28×10^6 PSI @ 70°F

b. The above four conclusions answer the first three tasks of the work statement. A preliminary answer to the fourth task--the question of leakage above the design point is given by the following summary of the panel/membrane analysis. Recommendations are also presented to refine these answers.

1. Onset of local yielding could occur at about 8 psi, but this is not considered a problem since local yielding could lead to stress relief. Full plastic hinging would not theoretically occur until 12 psi. This could lead to local cracking for a repeated pressurizing case (low cycle fatigue) but may not be important for a "one-shot" loading.

2. Gross membrane yielding could occur at about 27 psi. This corresponds to the ASME code value of 23 psi limit loading. It is interesting to note that an ela tic-platic analysis carried out by Sandia* gives a nominal failure pressure of 27 $\pm$ 3 psi.

It appears from this simplified analysis that the progression of events with increasing pressure, begins with pure bending resistance and small local elastic fiber deformations and progresses through combined bending and tensile resistance (quilting) with larger elastic deformations. Eventually local zones of plastic yielding will culminate in a state such that the final resistance mode is pure membrane tension in the skin material alone. This final state will only occur if the skin material is sufficiently ductile to avoid local rupture by tearing or cracking with the internal bending resistance nullified by yielding. Furthermore this final state will be reached independently of the properties of the stiffeners

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*"Report On Systems Analysis Task, Reactor Safety Study Methodology ApplicationsProgram, Sequoyah #1 Power Plant," Draft Report 1978, Asselin, Carlson, Gramond, Hickman, Fedele, Cybulskis and Wooton.

(from zero to infinitely stiff) so long as the spacing of the ring stiffeners is greater than about 60 inches for the 1/2 inch plate. The final state would then be pure membrane resistance with an equivalent longitudinal thickness which includes the partial effect of longitudinal stiffeners and with hoop thickness equal to the unmodified plate thickness. The resulting limit load pressure about 27 psi is thus probably a reasonable estimate of failure onset. The structure may fail locally below this value but will probably not survive much above this value whatever the properties of stiffeners as currently spaced.

Based on these analyses and conclusions it is recommended that further analyses and experimental verification be carried out:

a. A detailed finite element code analysis should be carried out to clarify the location, extent and profile of stress concentrations.

b. A full scale excastré panel should be pressurized to failure including a full strain gage and stress coat instrumentation. This would not be difficult or expensive since the panel size is only 10 ft by 4 ft, and the severity and effect of the local stress concentrations could be readily evaluated. The pressurization should be carried out in two stages.

   1. Up to 13.5 psi and back to zero (to simulate the containment acceptance pressure test). The panel should then be examined carefully for local deformations, etc. These would likely be shown up by stress coat or crack detection methods.

   2. Pressurization to failure with full instrumentation reading at selected pressure increments.