OFFSHORE POWER SYSTEMS

TITLE: ANALYSIS OF DUKE MCGUIRE CONTAINMENT SHELL
TO DETERMINE RESPONSE OF A CRITICAL PANEL
TO UNIFORM INTERNAL PRESSURE

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ABSTRACT

The Duke McGuire steel containment shell may experience uniform internal pressure loads associated with a postulated "hygrogen burn" which are significantly in excess of the maximum design pressure for the vessel (15.0 psi). The maximum pressure of 15.0 psi is consistent with an allowable membrane stress intensity (maximum principal stress difference based on Tresca yield criteria), in the unstiffened shell, limited to approximately 50 percent of minimum yield strength (by ASME B & PV Code, Section III). Various considerations such as code safety factor, actual yield strength, Von Mises yield criterion, effects of longitudinal and circumferential stiffeners, plasticity and effects of large displacements suggest that the shell can sustain significantly higher pressures at its limit of functional capability. That functional limit may be based on either:

1. Gross plastic instability of the structural system
2. Limited radial deflections or
3. Limited ductility

This report presents results of the McGuire containment shell analysis that was conducted to provide estimates of the shell's functional capability.

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1.0 INTRODUCTION

The purpose of this report is to present results of the Duke McGuire containment shell analysis that was conducted to provide estimates of the shell's functional capability. The configuration of the McGuire containment shell is contained in Reference 1.

Section 2.0 contains a discussion of material characteristics employed in the analysis and how they compare to those utilized in calculations based on the ASME B & PV Code, Section III rules. Section 3.0 contains a description of the finite element model and Section 4.0 presents the analysis results.

Microfiche of the computer calculations supporting Section 4.0 can be found in a companion document, Reference 2. Documentation of the computer program used to conduct the analysis can be found in Reference 3.

2.0 MATERIAL CHARACTERISTICS

The material characteristics addressed in evaluating the function capability of the McGuire containment shell under internal pressure loading signify a relaxation of the conservatisms implicit in the ASME B & PV Code, Section III design pressure calculations. Differences arise in prescription of material yield strength at temperature, stress or strain limits and yield criteria.

2.1 Material Yield Stress

The ASME B & PV Code, Section III prescribed minimum yield strength at temperature for SA 516 Grade 60 material is 32 KSI. The yield strength of the SA 516 Grade 60 material used in this analysis is 42.1 KSI based on actual test results. A Young's Modulus, E, of 27.9 x 10^6 psi was used to characterize elastic material behavior.

2.2 Stress or Strain Limits

The ASME B & PV Code, Section III stress limits are based on maintaining certain factors of safety against material first yield as well as material ultimate strength. The resulting allowable membrane stress intensity for SA 516 Grade 60 material is approximately 50 percent of the specified material minimum yield strength. The analysis of functional capability reported herein is based on an elastic-perfectly plastic material representation and allows for material strains into the plastic range. Functional capability may be limited by a predicted total strain in terms of yield strain (ductility ratio).

2.3 Yield Criteria

The ASME B & PV Code, Section III design pressure calculations are based on comparing "stress intensities" (principal stress differences) to a percentage of yield strength (or a more conservative percentage of ultimate strength). This relates to margin against yield with yielding defined by the Tresca yield criterion. The analysis of functional capability is based upon a Von Mises yield criterion (maximum distortion strain energy). In the case of an unstiffened containment shell under uniform pressure, the maximum Von Mises "equivalent stress" would be 15 percent lower than the corresponding maximum ASME "stress intensities". Therefore, with all other things being equal, applying a Von Mises yield criterion will result in a predicted shell capability as high as 15 percent more than that predicted by applying the maximum shear stress (Tresca) yield criterion.

3.0 FINITE ELEMENTS ANALYSIS

The axisymmetry of the shell, the relatively uniform spacing of the circumferential & longitudinal stiffeners and the uniformity of the applied pressure loading suggested that a valid analysis, incorporating appropriate boundary conditions, could be conducted for a critical panel (between circumferential and longitudinal stiffeners). A constant shell thickness of 0.75 inches and circumferential stiffeners on 10'-0" centers with 6 1/8" X 1/2" longitudinal stiffeners (on 3° centers) were addressed for this critical panel of the 690 inch radius cylindrical shell (Figure 1). Two axes of symmetry within the panel allowed for reducing the physical bounds of the finite element model to 1/4 of the panel (Figure 2) by imposing boundary conditions to reflect the symmetry of the structure and loading. The finite element model is shown in Figure 3.

3.1 Boundary Conditions

The conditions of symmetry are enforced by imposing the proper boundary conditions. The boundary conditions imposed on the model shown in Figure 3 are:

a) Vertical displacement (UZ)=0 for nodes at elevation Z=0.0 inches.

b) Vertical displacements (UZ) of nodes at elevation Z=60.0 inches coupled.

c) Rotation about tangential axes (ROTY)=0 for nodes at elevations Z=0.0 inches and Z=60.0 inches.

d) Tangential displacement (UY)=0 for nodes at $\theta=0^{\circ}$ and $\theta=1.5^{\circ}$.

e) Rotation about vertical axis (ROTZ)=0 for nodes at $\theta=0^{\circ}$ and $\theta=1.5^{\circ}$.

f) Rotation about radial axes (ROTX)=0 for nodes along Z=0.0 inches, Z=60.0 inches, $\theta=0^{\circ}$ and $\theta=1.5^{\circ}$.

3.2 Applied Loads

The applied loading consisted of uniform internal pressure on the panel elements as well as a corresponding uniform vertical load applied at elevation Z=60.0 inches to address longitudinal stresses (uplift from the vessel head) on the circumferential boundaries of the model.

3.3 Analysis

The analysis was conducted using the ANSYS Revision 3 (Update 67J1) computer program (Reference 3). The entire model was composed of ANSYS STIF 48 - Plastic Triangular Shell elements with panel and circumferential stiffener flange elements having both bending and membrane stiffness capability and other stiffener elements having membrane stiffness only. The static analysis addressed elastic-perfectly plastic (no strain hardening) material behavior with yielding based on a Von Mises yield criterion. Large displacement theory was invoked with convergence aided by the incorporation of stress stiffening.

4.0 ANALYSIS RESULTS

A linear elastic static analysis was first conducted at an internal pressure of 15 psig. As expected, the structure remained completely elastic with a maximum Von Mises equivalent stress, anywhere in the modelled structure of 21.1 KSI (corresponding to panel plate bending at the longitudinal stiffener-circumferential stiffener discontinuity-location (A) of Figure 2.) Based on a yield strength of 42.1 KSI, the analysis indicated that the structure would remain completely elastic up to an internal pressure of approximately 30 psig. The maximum "membrane" (midsurface) stress in the panel at 15 psig internal pressure was 10.7 KSI (Von Mises equivalent), as shown in Table 1. In an identical unstiffened shell the maximum "membrane" (midsurface) stress at 15 psig internal pressure would be 11.95 KSI (Von Mises equivalent). The results tabulated in Tables 1, 2 and 3 indicate that the panel shares "hoop load" with the circumferential stiffener by means of panel action and longitudinal stiffener bending.

Large displacement, inelastic static analysis was conducted for internal pressures ranging from 30 psig to 50 psig, in steps of 5 psig; from 52 psig to 68 psig, in steps of 2 psig; and at 69 psig. The analysis for an internal pressure of 30 psig was done for a single iteration only, conducted for the purpose of "starting" the large displacement formulation. As such, computed results at 30 psig do not include large displacement effects.

The large displacement and plasticity convergence criteria employed for internal pressures up to and including 64 psig were as follows:

a) Plasticity Ratio = .01, ratio of incremental plastic strain within an iteration to total elastic strain.

b) Large Displacement Increment = .001 inches, incremental displacement within an iteration.

At 66 psig internal pressure, the final iteration was characterized by a plasticity ratio of .029 and a large displacement increment of .002 inches.

At 68 psig internal pressure, the final iteration was characterized by a plasticity ratio of .011 and a large displacement increment of .001 inches.

At 69 psig internal pressure, the structure becomes unstable. The final iteration was characterized by a plasticity ratio of .038 and a large displacement increment of .011 inches.

Examination of the element strain data indicated that all panel and circumferential stiffener elements were in the plastic strain region confirming that the structure was no longer capable of resisting incremental hoop loads.

Figure 4 contains plots of internal pressure load vs. radial displacement at locations A, B, C, and D of Figure 2. The basic data points for the plots were pressure loads of 32 psig through 68 psig, extrapolated back to zero displacement at zero load. The plots illustrate a load-displacement curve that is linear and stable up to 60 psig and stable up to 68 psig. Table 4 contains a tabulation of significant stresses in the circumferential (ring) stiffener at a load of 68 psig, indicating that limited portions of each cross-section of stiffener were still within elastic limits. At an internal pressure load of 68 psig, Figure 4 indicates a maximum radial displacement of approximately 1.56 inches, and examination of the computer output reveals that the maximum "equivalent" total strain, anywhere in the structure (panel), is .0104 (approximately seven times yield strain), a bending strain occurring in the local region of the stringer/ring stiffeners discontinuity. The maximum strain in the longitudinal stiffener is .0019 (approximately 1.25 times yield strain) and the maximum strain in the web of the circumferential stiffener is .0022 (approximately 1.5 times yield strain). The maximum strain in the flange of the circumferential stiffener is .0015 (approximately yield strain).

The results reported herein suggest that basing functional capability of the McGuire containment shell, under uniform pressure loading, on gross plastic instability of the structure (i.e. unstable load-deflection curve) renders a limit of 68 psig.

5.0 REFERENCES

1. Duke Power Company McGuire Nuclear Station Units 1 & 2, Drawing No. MC 1042-1, "Reactor Building Units 1 & 2 Containment Vessel Cylinder Plate Layout and Penetration Location".

2. OPS Document TD36A00 "Analysis of Duke McGuire Containment Shell to Determine Response of Critical Panel to Uniform Internal Pressure (RP35A99) Microfiche, September, 1980.

3. ANSYS Engineering Analysis User's Manual Update No. 1, July, 1979.

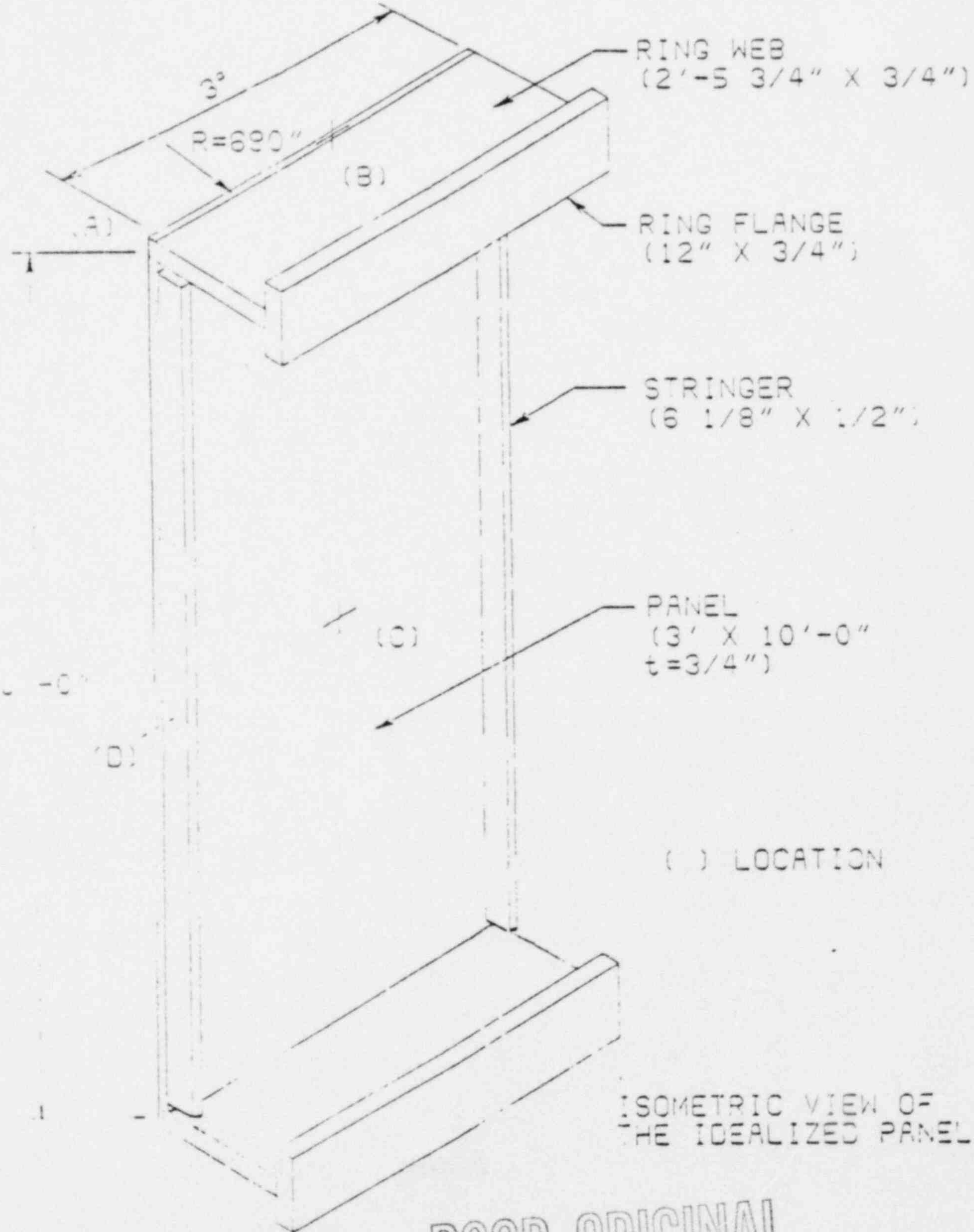

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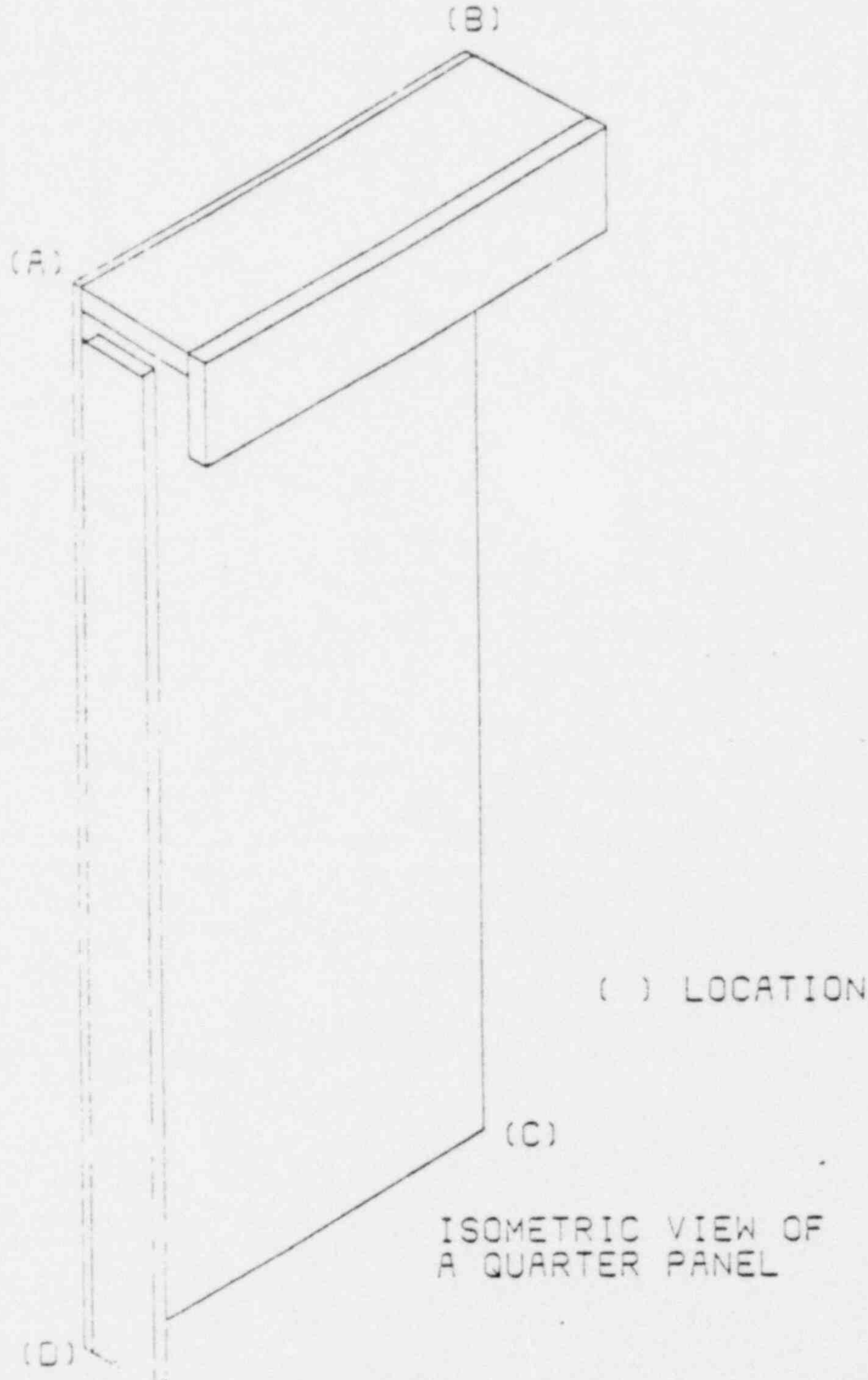
FIGURE 1



ISOMETRIC VIEW OF
A QUARTER PANEL

FIGURE 2

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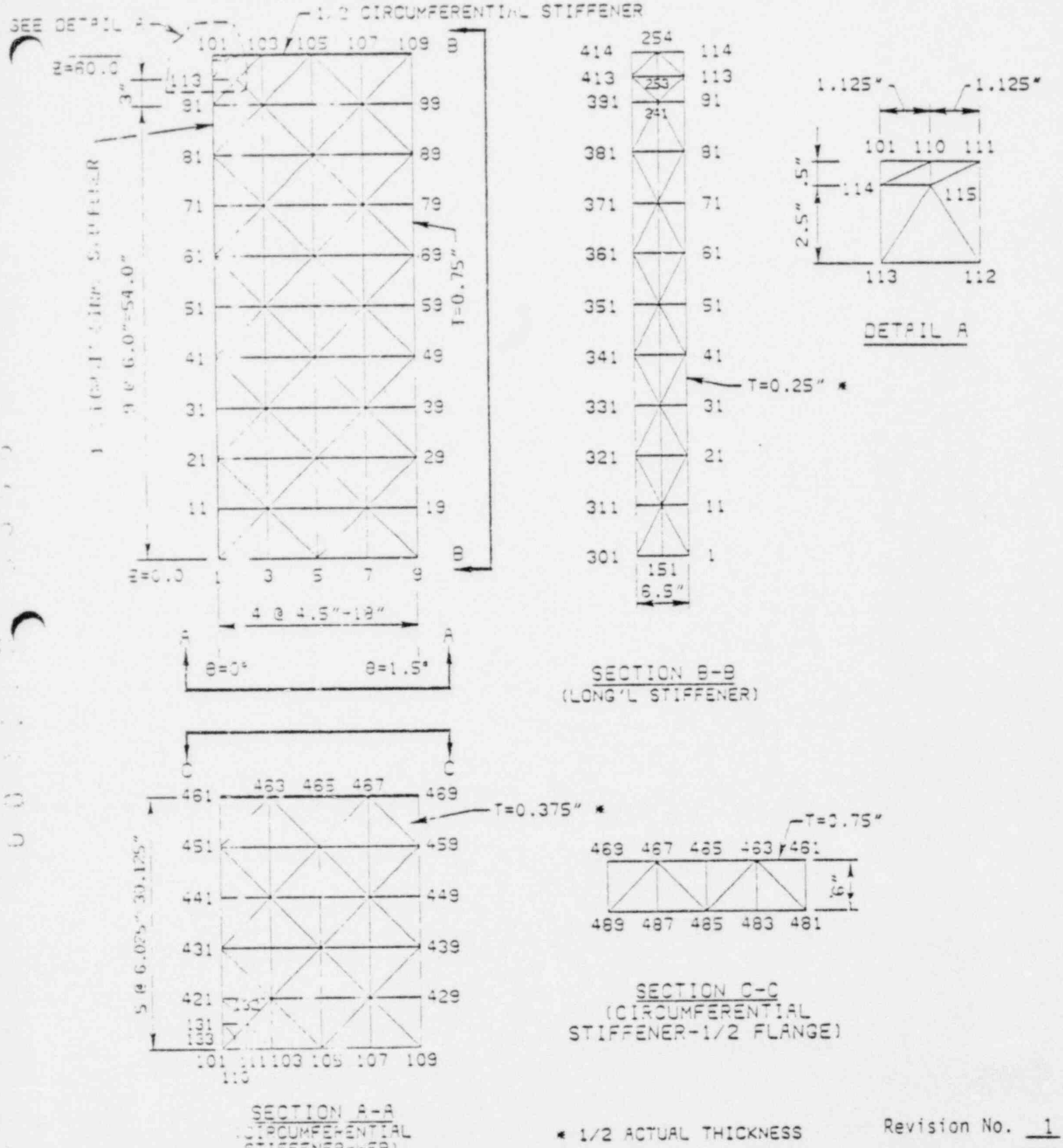
ALL ELEMENTS ANSYS STIF 48
"PLASTIC TRIANGULAR SHELL"



* 1/2 ACTUAL THICKNESS

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FIGURE 3

FINITE ELEMENT MODEL OF THE QUARTER PANEL

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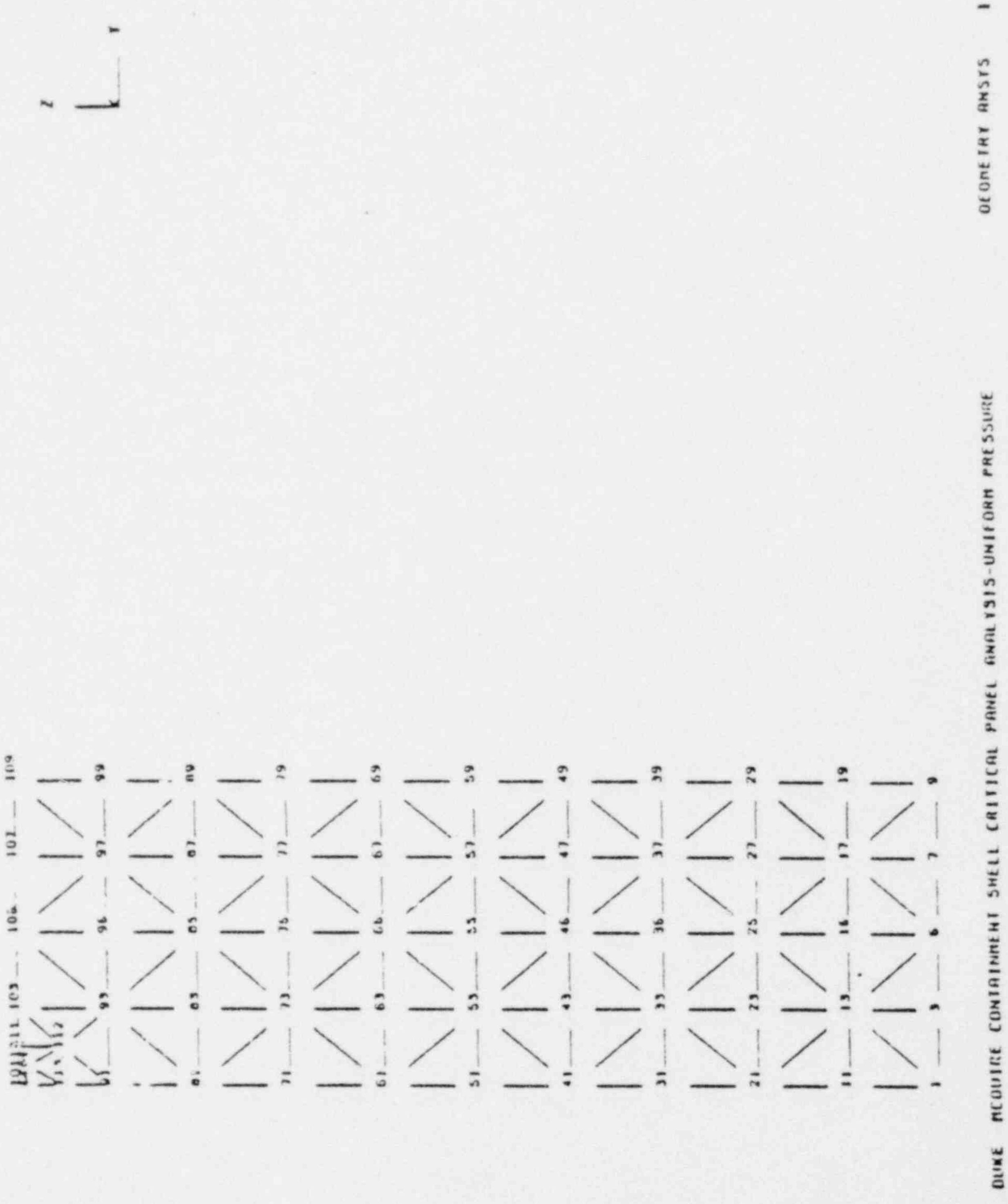

FIGURE 3A
ANSYS GEOMETRY - PANEL

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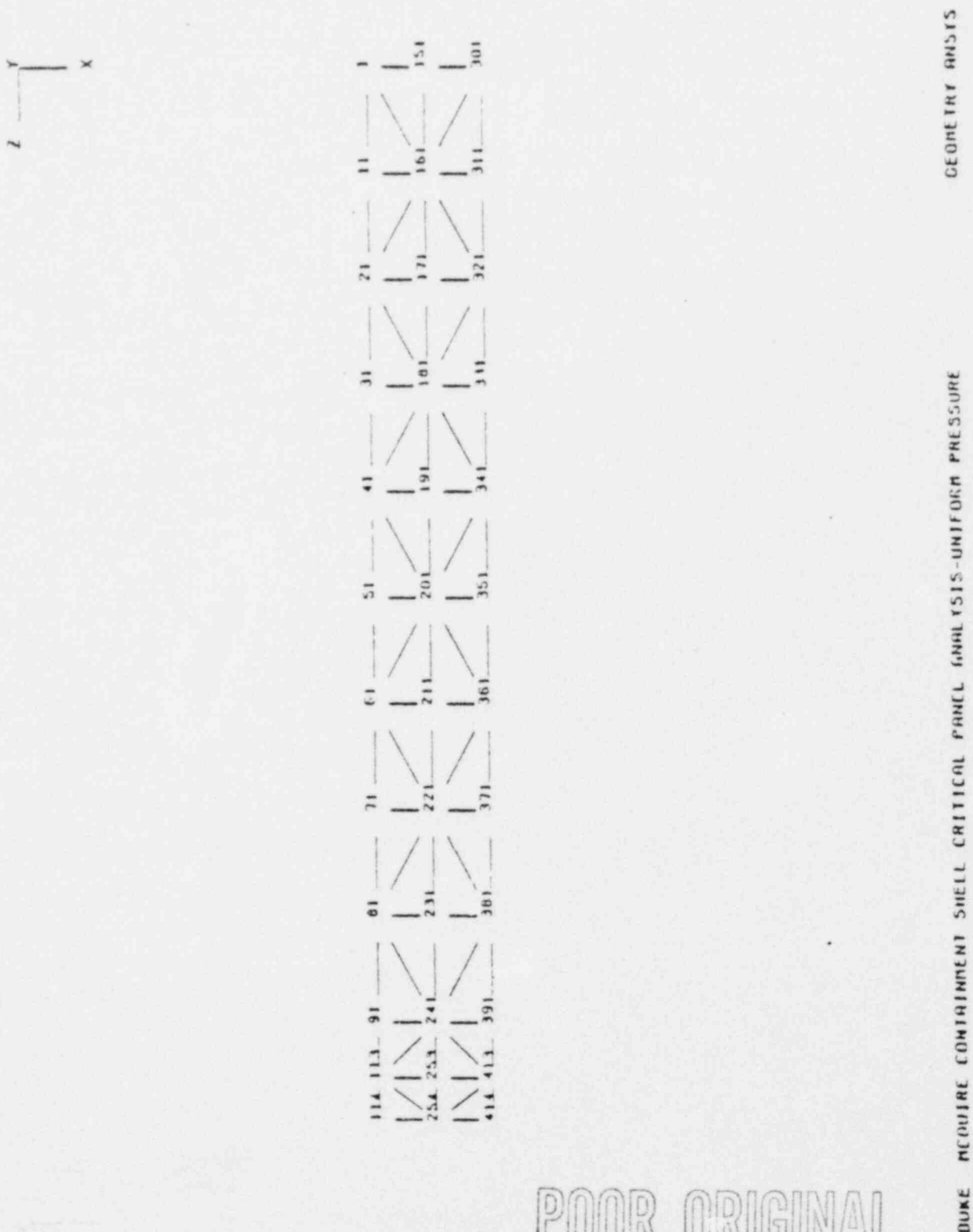

FIGURE 3B

ANSYS GEOMETRY - LONGITUDINAL STIFFENER

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IMAGE EVALUATION
TEST TARGET (MT-3)

6″

IMAGE EVALUATION
TEST TARGET (MT-3)

6″

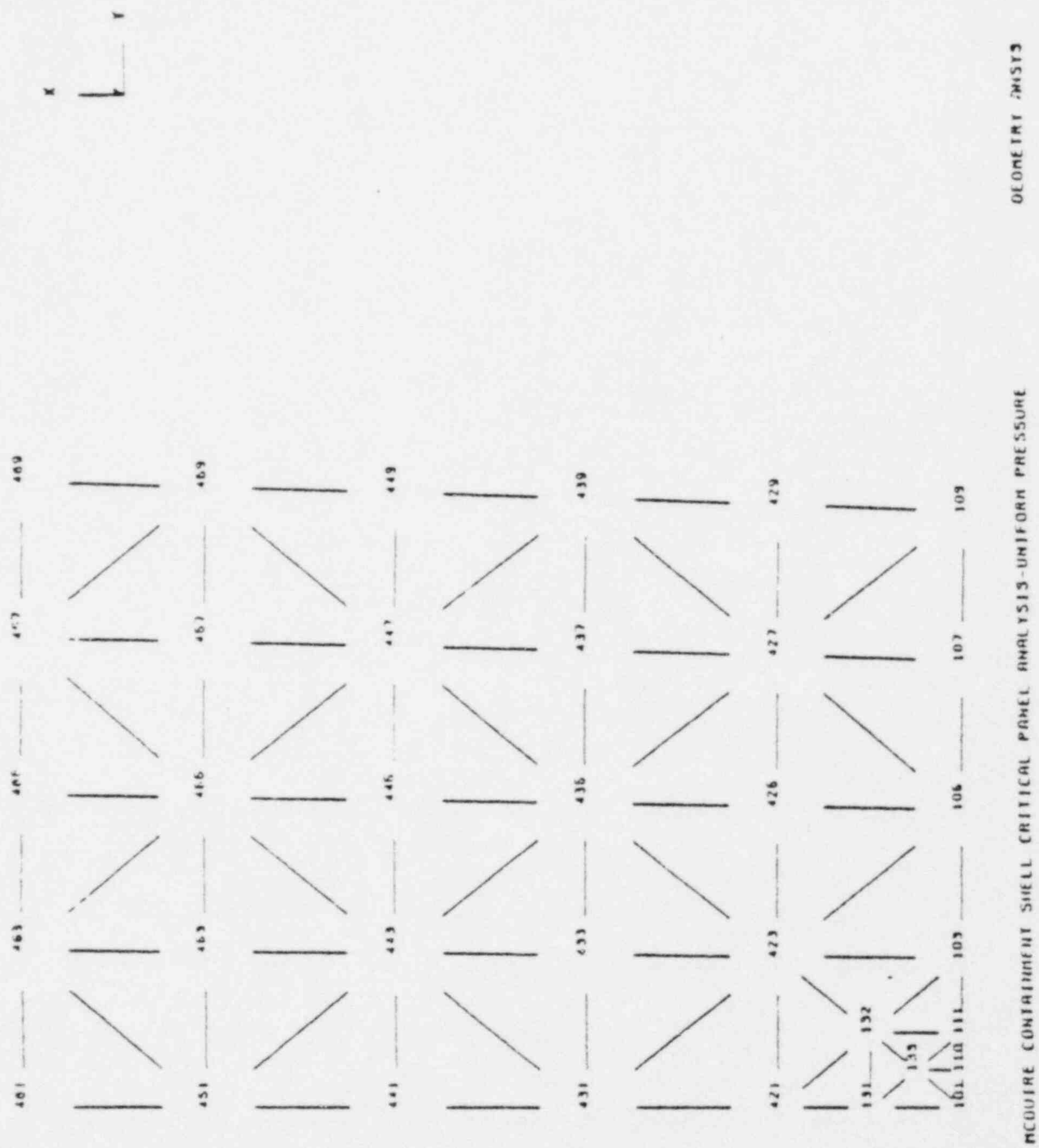

FIGURE 3C

ANSYS GEOMETRY - CIRCUMFERENTIAL STIFFENER WEB

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FIGURE 3D

ANSYS GEOMETRY - CIRCUMFERENTIAL STIFFENER FLANGE

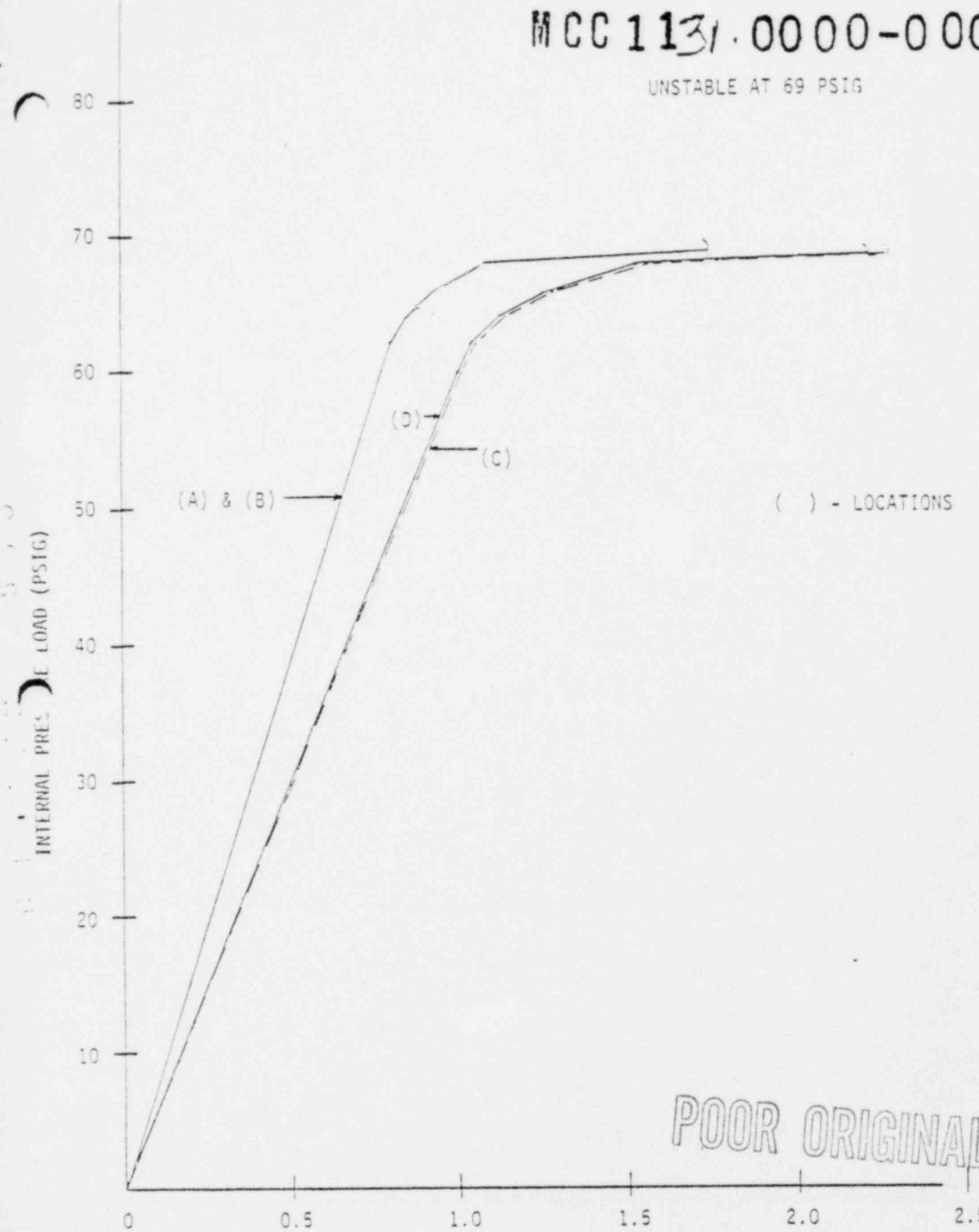


FIGURE 4

PRESSURE-DISPLACEMENT CURVES BASED ON GEOMETRIC & MATERIAL NON-LINEAR ANALYSIS

TABLE 1

STRESSES DUE TO 15 PSI INTERNAL PRESSURE LOAD

MEMBER	LOCATION	FIBER	STRESS INTENSITY (ksi)	VON MISES (ksi)
3/4" PANEL PLATE	CENTRAL REGION OF THE PANEL (C)	OUTER SURFACE MID SURFACE INNER SURFACE	13.4 12.3 11.3	11.6 10.7 9.7
	NEAR MID SPAN OF THE RING (B)	OUTER SURFACE MID SURFACE INNER SURFACE	8.1 9.7 11.3	7.2 8.6 10.9
	NEAR MID SPAN OF THE STRINGER (D)	OUTER SURFACE MID SURFACE INNER SURFACE	11.2 12.4 13.5	9.7 10.7 11.8
	NEAR STRINGER/ RING DISCONTIN-UITY (A)	OUTER SURFACE MID SURFACE INNER SURFACE	11.4 10.5 24.3	10.2 10.2 21.1

TABLE 2

STRESSES DUE TO 15 PSI INTERNAL PRESSURE LOAD

MEMBER	LOCATION	FIBER	STRESS INTENSITY (ksi)	VON MISES (ksi)
RING	MID SPAN (B)	(IN WEB)		
		NEAR INNER EDGE	7.6	7.6
		NEAR OUTER EDGE	7.1	7.1
		(IN FLANGE)		
		OUTER SURFACE	7.8	7.0
		MID SURFACE	7.7	6.8
		INNER SURFACE	7.5	6.6
	RING-STIFFENER "JUNCTION" (A)	(IN WEB)		
		NEAR INNER EDGE	8.5	7.9
		NEAR OUTER EDGE	7.2	7.1
		(IN FLANGE)		
		OUTER SURFACE	7.8	7.0
		MID SURFACE	7.7	6.8
		INNER SURFACE	7.5	6.7

TABLE 3

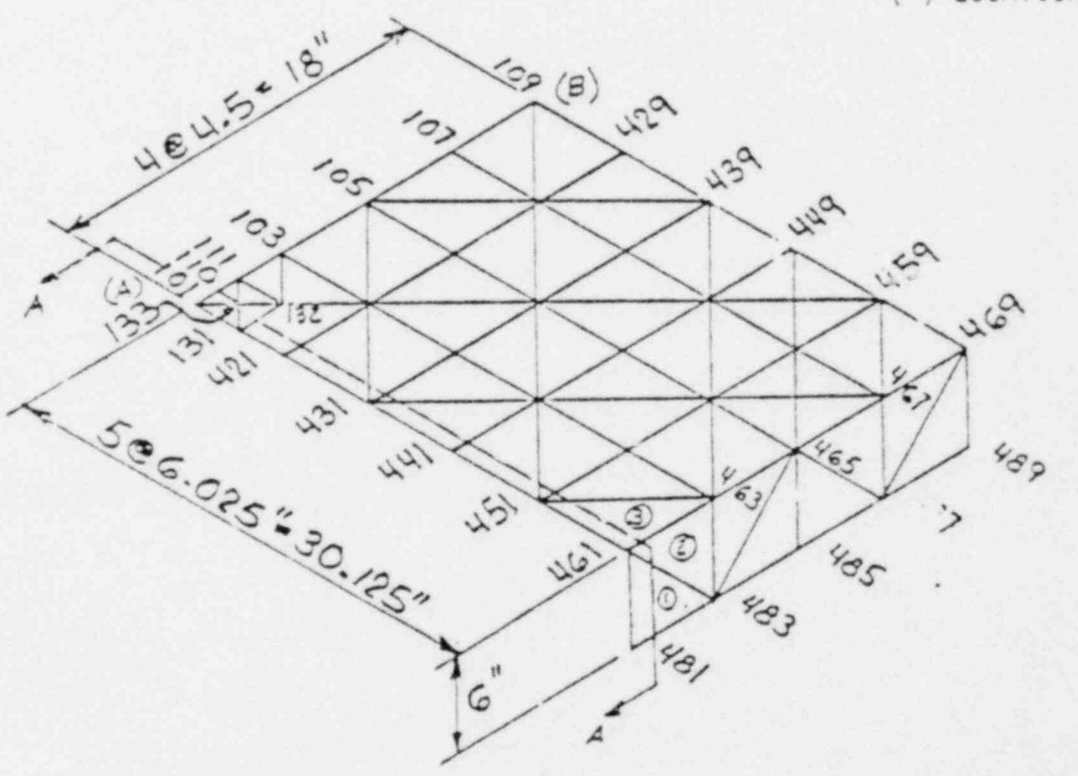
STRESSES DUE TO 15 PSI INTERNAL PRESSURE LOAD

MEMBER	LOCATION	FIBER	STRESS INTENSITY (ksi)	VON MISES (ksi)
STRINGER	MID SPAN (D)	NEAR INNER EDGE	8.1	7.6
		NEAR OUTER EDGE	12.6	12.2
	RING-STRINGER "JUNCTION" (A)	NEAR INNER EDGE	4.8	4.6
		NEAR OUTER EDGE	1.2	1.1

CIRCUMFERENTIAL STIFFENER CRITICAL
CROSSECTION (A-A) MIDSURFACE STRESSES
AT A UNIFORM PRESSURE LOAD OF
68 PSIG

ELEMENT NO. (see below)	HOOP STRESS (ksi)	VON MISES (ksi)
1	41.2	41.2
2	41.2	41.2
3	41.0	41.4

( ) LOCATION



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Professional Qualifications
of
Richard S. Orr
Chief Engineer of Structural Engineering
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February 9, 1981

My name is Richard S. Orr and my business address is 8000 Arlington Expressway, Jacksonville, Florida 32211. I am currently Chief Engineer of Structural Engineering for Offshore Power Systems. In this position I have the functional responsibility for the design of all structures assembled on the Floating Nuclear Plant. I have been assigned to this position since August of 1972. Prior to that, I performed the same function as Manager of Strucutral Engineering for Westinghouse Special Project Division.

From March, 1967 to September, 1971, I worked for Westinghouse Nuclear Energy Systems in the areas of structural review, plant arrangement, and development of new concepts. From October, 1962 to March, 1967 I worked for Rendel, Palmer & Tritton in London, England, where I spent two years in the analysis of prestressed concrete pressure vessels and two years on site during the construction of a power plant.

I earned a Master of Arts (first class) degree in Mechanical Sciences at Cambridge, England in 1962. I am a Professional Engineer registered in the states of Florida and New Jersey. I am Chairman of ACI Committee 349 which prepares code requirements for concrete nuclear safety related structures. I have been a member of code committees of the American Society of Mechanical Engineers preparing requirements for both steel and concrete containment vessels.