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PDX VACUUM VESSEL STRESS ANALYSIS

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Summary

A stress analysis of PDX vacuum vessel is described and the summary of results is presented. The vacuum vessel is treated as a toroidal shell of revolution subjected to an internal vacuum. The critical buckling pressure is calculated. The effects of the geometrical discontinuity at the juncture of toroidal shell head and cylindrical outside wall, and the concavity of the cylindrical wall are examined. An effect of the poloidal field coil supports and the vessel outside supports on the stress distribution in the vacuum vessel is determined. A method evaluating the influence of circular ports in the vessel wall on the stress level in the vessel is outlined.

Introduction

The vacuum vessels of Tokamak experimental devices are toroidal shells of either circular or other shaped cross-section. Due to poloidal field coils placed inside the vacuum vessel[1] of the Poloidal Divertor Experiment (PDX), the vacuum vessel requires a cross-section in the shape of a racetrack. The PDX vacuum vessel and its major dimensions is shown in Figure 1. The details of the vessel, its design, fabrication and assembly are described elsewhere.[2]

The objectives of the analysis of the vacuum vessel were:

a) to determine the dimensions, primarily the thickness of the vessel wall,

b) to find the effects of defects which might occur during the manufacture of the vessel,

c) to calculate the influence of large ports in the vessel walls on the stresses (not shown in Figure 1),

and

d) to evaluate the stress concentrations around the interior poloidal coil supports and around the outside vessel supports.

The vacuum vessel is subjected to two different types of loading:

1) The uniform external pressure due to the operating vacuum of 1.0E-8 Torr maintained in the vessel.
2) Non-uniform load such as concentrated loads from
    a) the supports of poloidal coils attached to the inner surface of the vessel,
    b) the outside foot supports and
    c) the accessory equipment and diagnostic instruments.

Two types of analyses are characteristic for the vacuum vessel. There are

a) the stability analysis, where the critical buckling pressure on the vessel is determined and

b) the stress and deflection analysis, where the stress distribution throughout the vessel and the deformed geometry of the vessel due to both uniform and non-uniform loading conditions are calculated.

The geometry of the vessel and the loading characteristics required a computer solution. A mathematical finite difference model of the vessel used in both the stability and stress analysis due to concentrated loads from the poloidal coil supports is shown in Figure 2.

The evaluation of stresses around the circular ports for the neutral beam injection, the heating, pumping and diagnostic equipment required a finite element model of a 72° segment of the vessel as shown in Figure 3. However, the results of this finite element analysis were not available at the time of writing of this paper, and will be reported later elsewhere.

Methods and Results of Analysis

Stability Analysis

The stability of circular toroidal shells subjected to external pressure was investigated elsewhere analytically[3,4] as well as experimentally.[5] The PDX vacuum vessel is a toroidal shell of a racetrack cross-section. The difficulties associated with the analytical solution of stresses in toroidal shells with non-circular cross-section resulted in the use of a numerical method. The computer program BOSOR4[6] using the finite difference technique was used in the analysis.

The computer model, see Figure 2, shows that the torus is represented by four sections: two cylindrical segments for the inner and outer walls and two toroidal segments for the upper and lower domes. There are four 2 inch x 3 inch flanges at the joints of toroidal and cylindrical segments. The flanges are needed for assembly purpose, but they also serve as structural stiffening rings. They are treated in the analysis as discrete elastic structures.

Several various thicknesses of the shell wall were considered from 0.25 inch to 0.5 inch. The critical buckling pressures were determined for the shells with and without supporting rings. The length of the cylindrical walls was varied as well. Additional rings supporting the outer cylindrical wall were also considered. The result of the analysis is shown in Figure 4. The critical buckling pressure is 127.5 psi in the case where the toroidal shell is 0.5 inch thick and the cylindrical walls are 61 inches long with four supporting rings. These parameters represent the final design of the vacuum vessel.

Stress and Deflection Analysis

The possibility of defects in the vessel due to the manufacturing process required an investigation of the influence these defects might have on stresses in the vacuum vessel. There is a geometrical discontinuity in the juncture of the upper dome and the outer cylindrical wall of 0.5 inch and a 0.5 inch concavity of the outer wall as shown in Figure 5. The evenly spaced foot (Figure 1) supports around the circumference are approximated by a continuous saddle support. This approximation will not influence the stresses in the vicinity of the discontinuity.

The finite difference linear stress analysis using the BOSOR4 computer program showed the maximum effective stress on the inner surface of the outer wall is 2211 psi and at the juncture where discontinuity exists is 1344 psi, see Figure 5.

The effect of non-uniform concentrated loads which are applied on the shell at the supports of poloidal coils is also evaluated by using BOSOR4. The concentrated axial loads and moments are represented by a series expansion. The outside foot supports were approximated by a continuous saddle support. The work was not completed at the time of writing of this paper on the more exact modeling of the actual supports but will be reported later. The maximum effective stress at the cross-section where the concentrated loads are applied is 17,600 psi at the outer surface of the upper dome, see Figure 6. The deformed geometry at the plane of load application is also shown in Figure 6.

In order to evaluate the stresses around the circular ports, the analysis using the finite element technique was considered. The model of 72° segment was designed using thin shell elements. The flanges around the openings have stiffening effects and are simulated by 3-D beam elements (Figure 3). The analysis is not complete and the results will be published later. Some of the ports will be covered by flat cover plates and they are being modeled as plate elements. Non-uniform concentrated loads are applied at the flanges. Since loads are not expected to be very large, and the flanges have a large stiffening effect on the shell, we do not anticipate very large stress concentration around the ports.

Conclusion

The stress analysis of the PDX vacuum vessel had shown that:

a) the vessel is stable under the uniform vacuum pressure,

b) the effect of geometrical discontinuities on the stress level is small

c) the stresses level at the vicinity of the poloidal field coil supports due to the fault loading conditions is acceptable, and

d) the combined overall stresses are well within the allowable limits.

Acknowledgements

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References

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4. P.F. Jordan, Vibration and Buckling of Pressurized Torus Shells, AIAA Paper 66-445, Los Angeles, California, 1966.

5. E.G. Fishlowitz, Investigation of Elastic Stability of Circular Toroidal Shells Under Uniform External Pressure, NSRDC Report 3338, April 1972.

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Figure 1. PDX Vacuum Vessel.
754789

Figure 2. BOSOR4 Finite Difference Model of PDX Vacuum Vessel Stability Analysis.
754788

Figure 3. The Finite Element Model of 72^0 Segment of the Vacuum Vessel.
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Figure 4. The Critical Buckling Pressure of PDX Vacuum Vessel.
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Figure 5. BOSOR4 Model and Results of the Stress Analysis of PDX Vacuum Vessel with Geometrical Discontinuities for the Vacuum Vessel of 14.7 psi.
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Figure 6. The Deformed Cross-Section of the PDX Vacuum Vessel at the Plane of Application of Poloidal Coils Support Loads.
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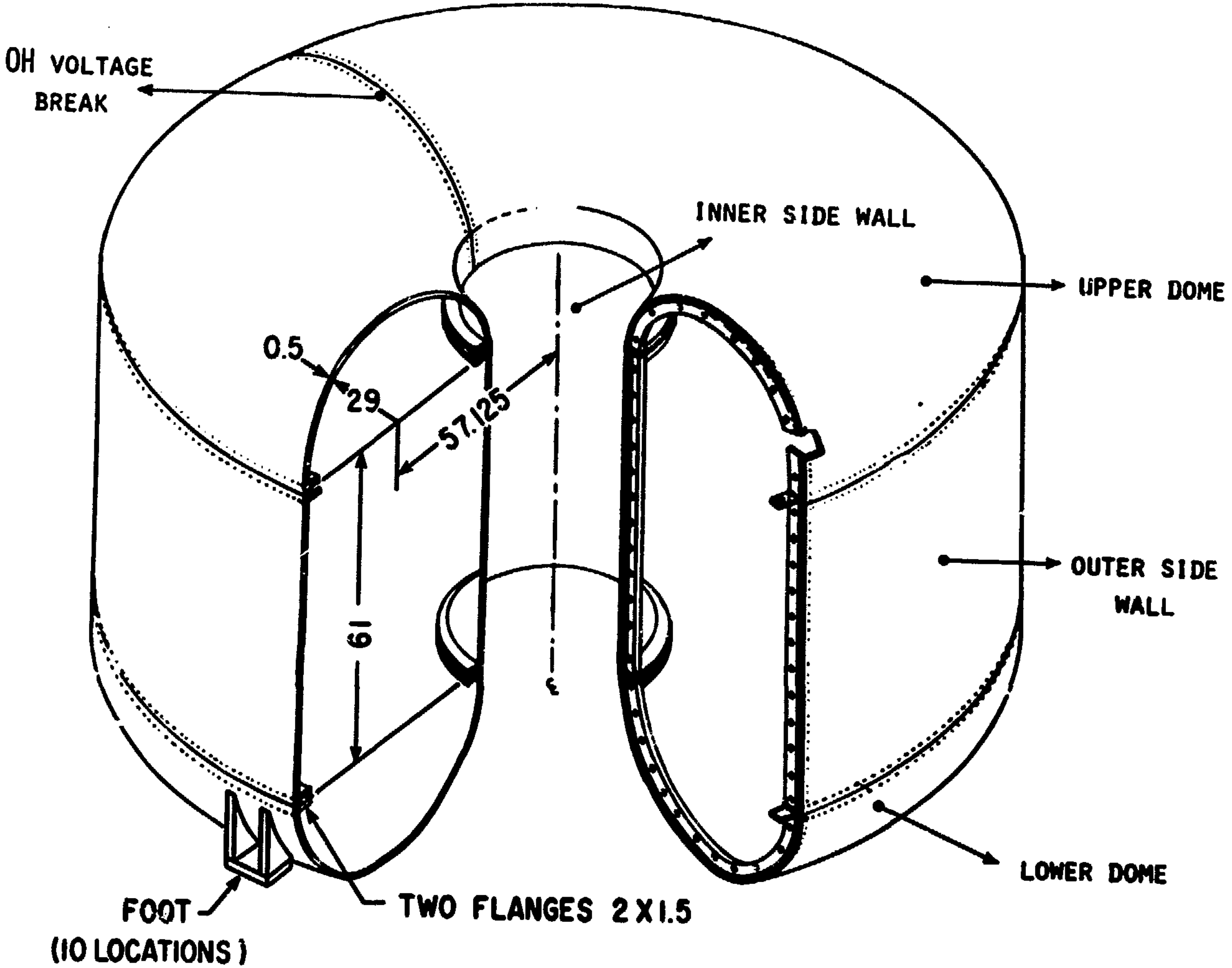
OH VOLTAGE BREAK
INNER SIDE WALL
UPPER DOME
0.5
29
57.125
19
℄
OUTER SIDE WALL
LOWER DOME
FOOT (10 LOCATIONS)
TWO FLANGES 2 X 1.5

Fig. 1

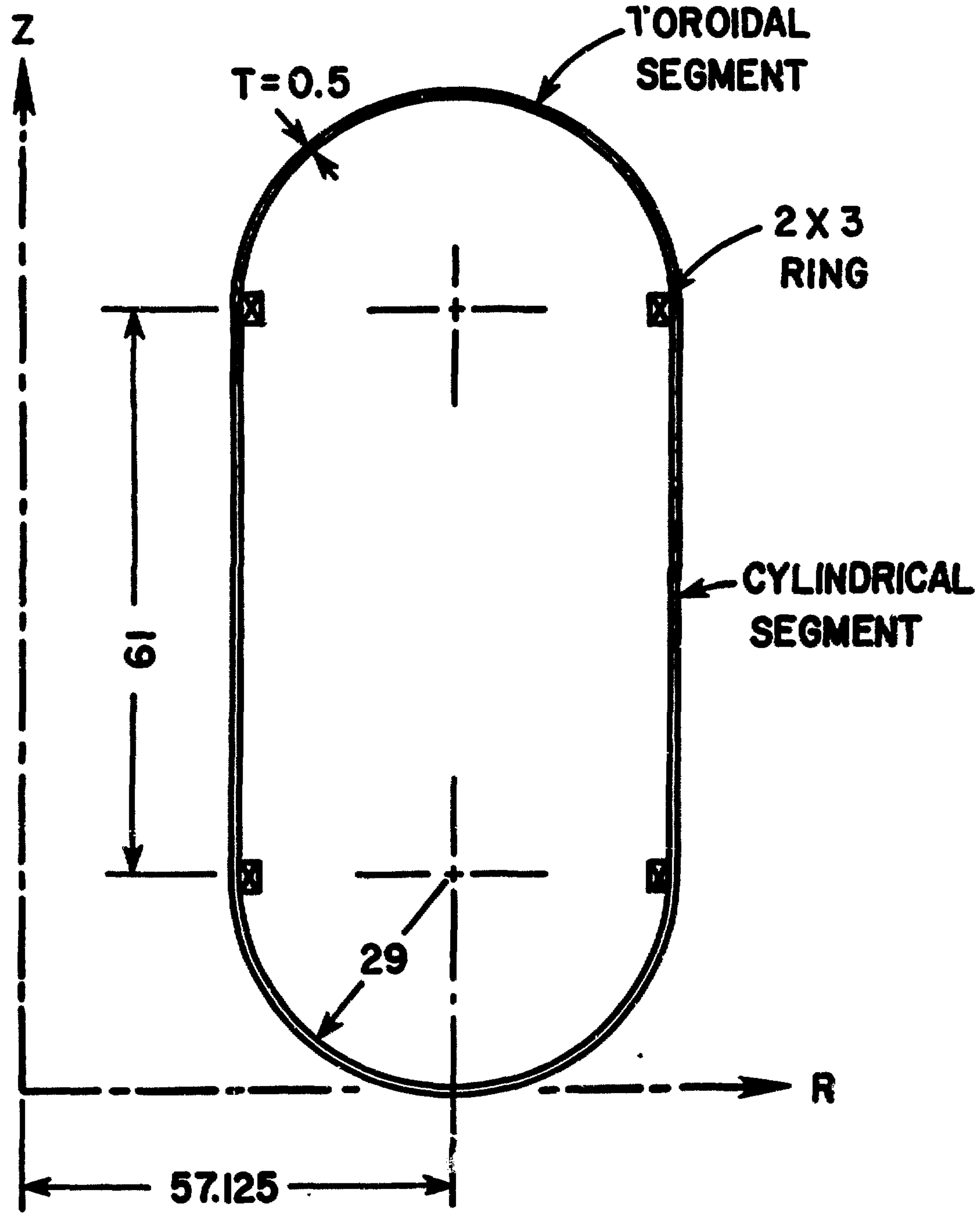
Z
T=0.5
TOROIDAL SEGMENT
2 X 3 RING
CYLINDRICAL SEGMENT
61
29
R
57.125

Fig. 2

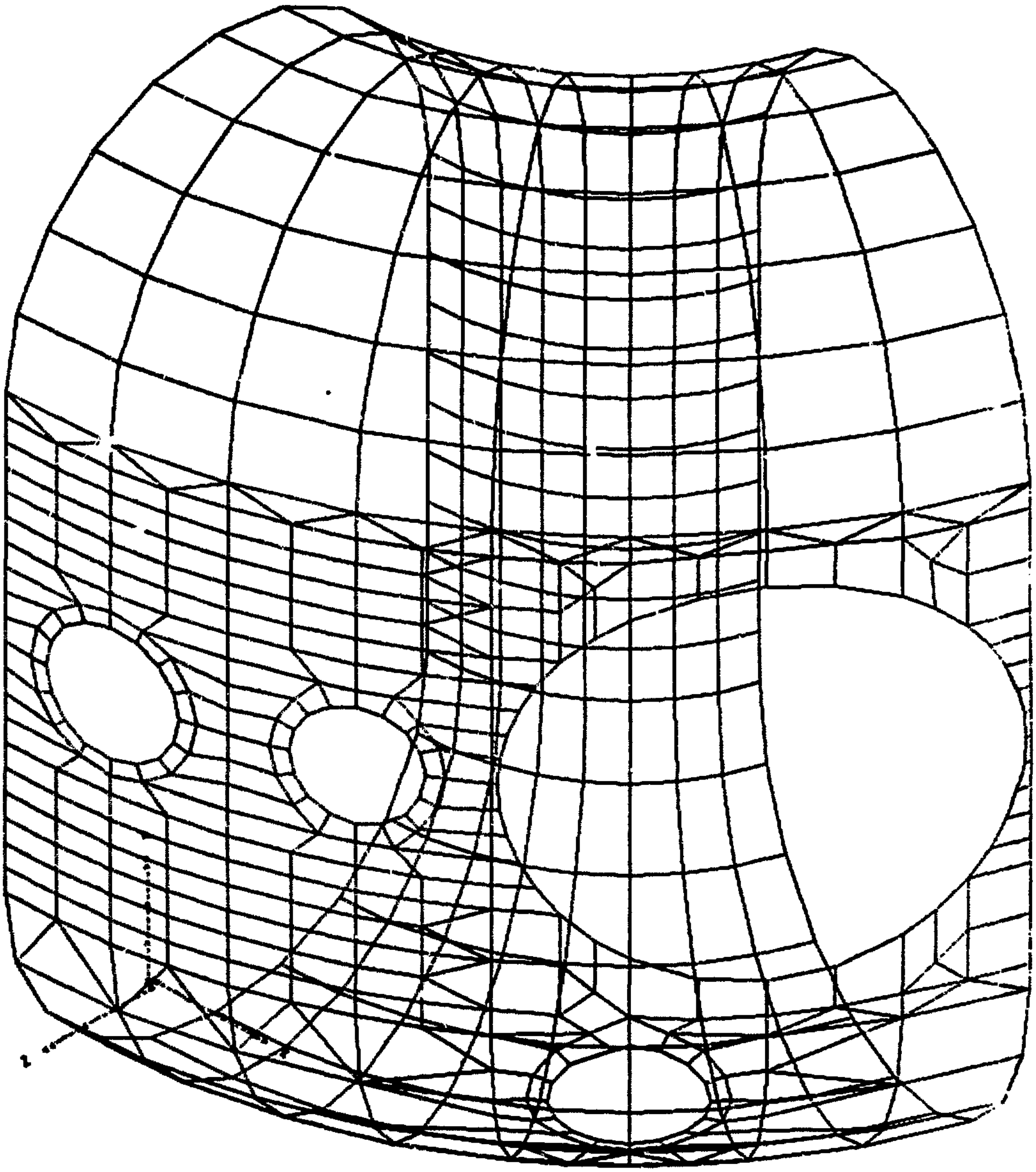

Fig. 3

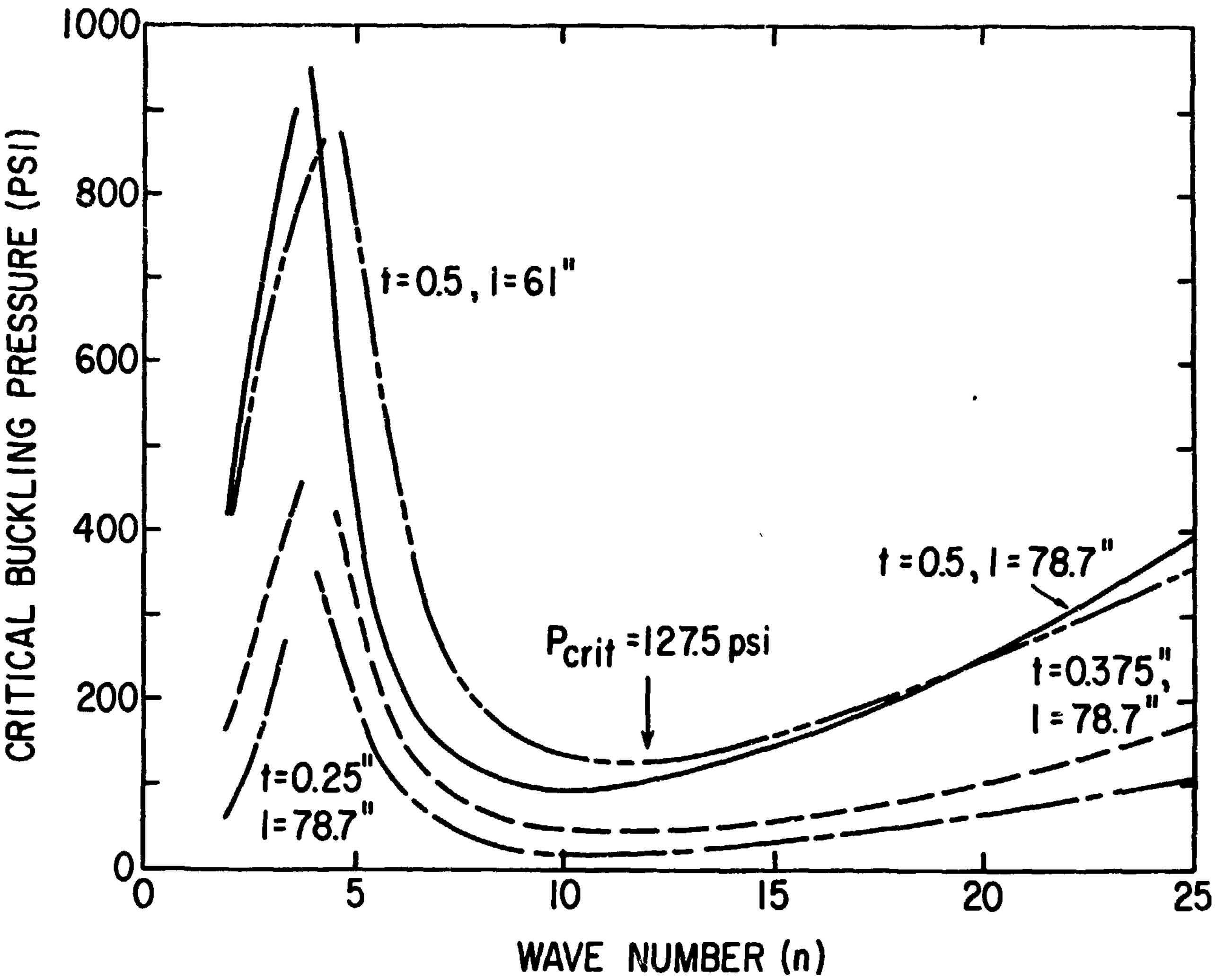
1000
800
600
400
200
0
CRITICAL BUCKLING PRESSURE (PSI)
0
5
10
15
20
25
WAVE NUMBER (n)
t=0.5, l=61"
t=0.5, l=78.7"
P_{crit} =127.5 psi
t=0.375",
l=78.7"
t=0.25"
l=78.7"

Fig. 4

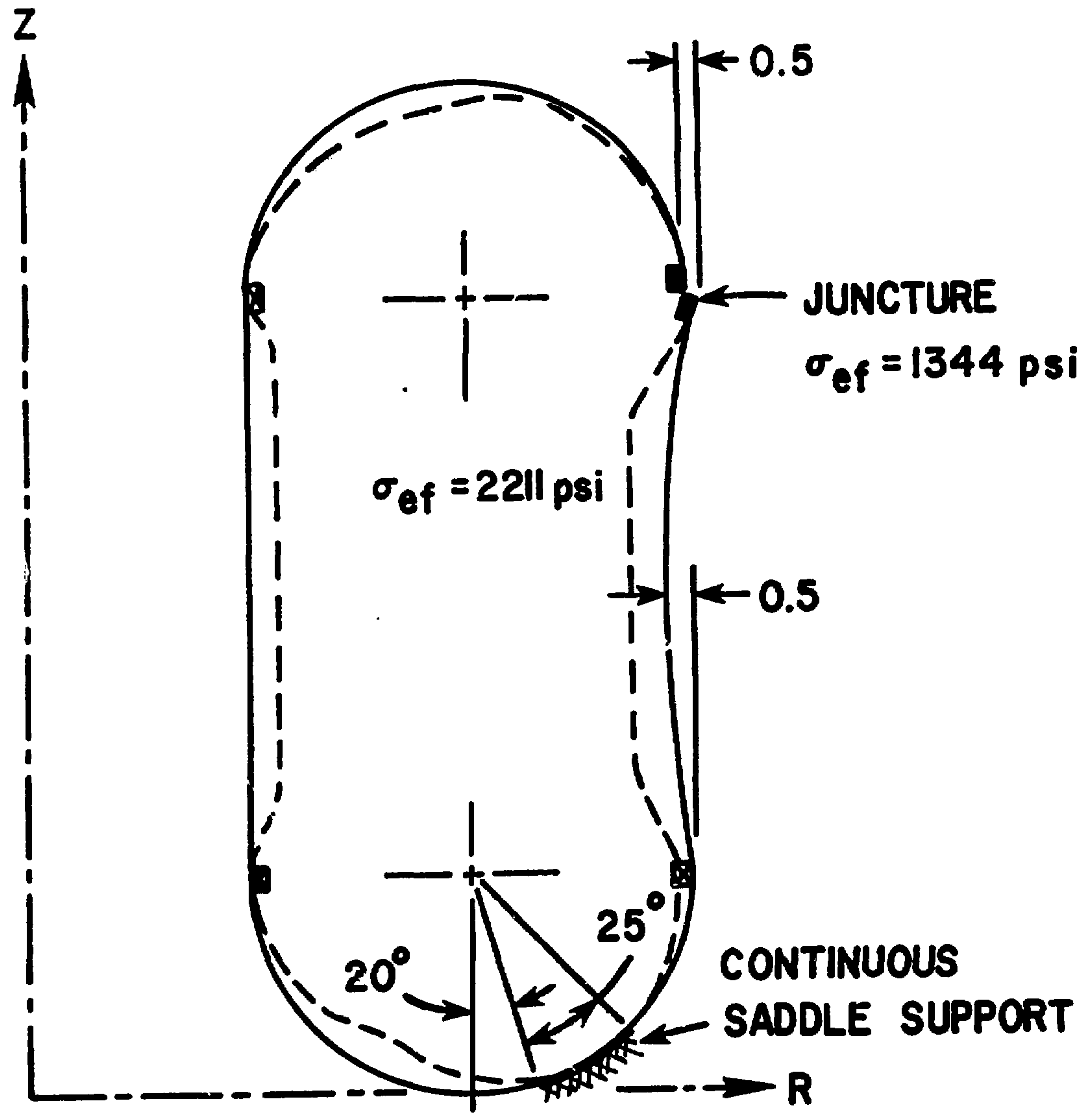
Z
0.5
JUNCTURE
σ_{ef} = 1344 psi
σ_{ef} = 2211 psi
0.5
25°
20°
CONTINUOUS
SADDLE SUPPORT
R

Fig. 5

F_1 = 1826 lbs

F_2 = 1851 lbs

F_3 = 1482 lbs

F_4 = 2405 lbs

M_1 = 22424 in-lbs

M_2 = 38698 in-lbs

M_3 = 23818 in-lbs

M_4 = 40637 in-lbs

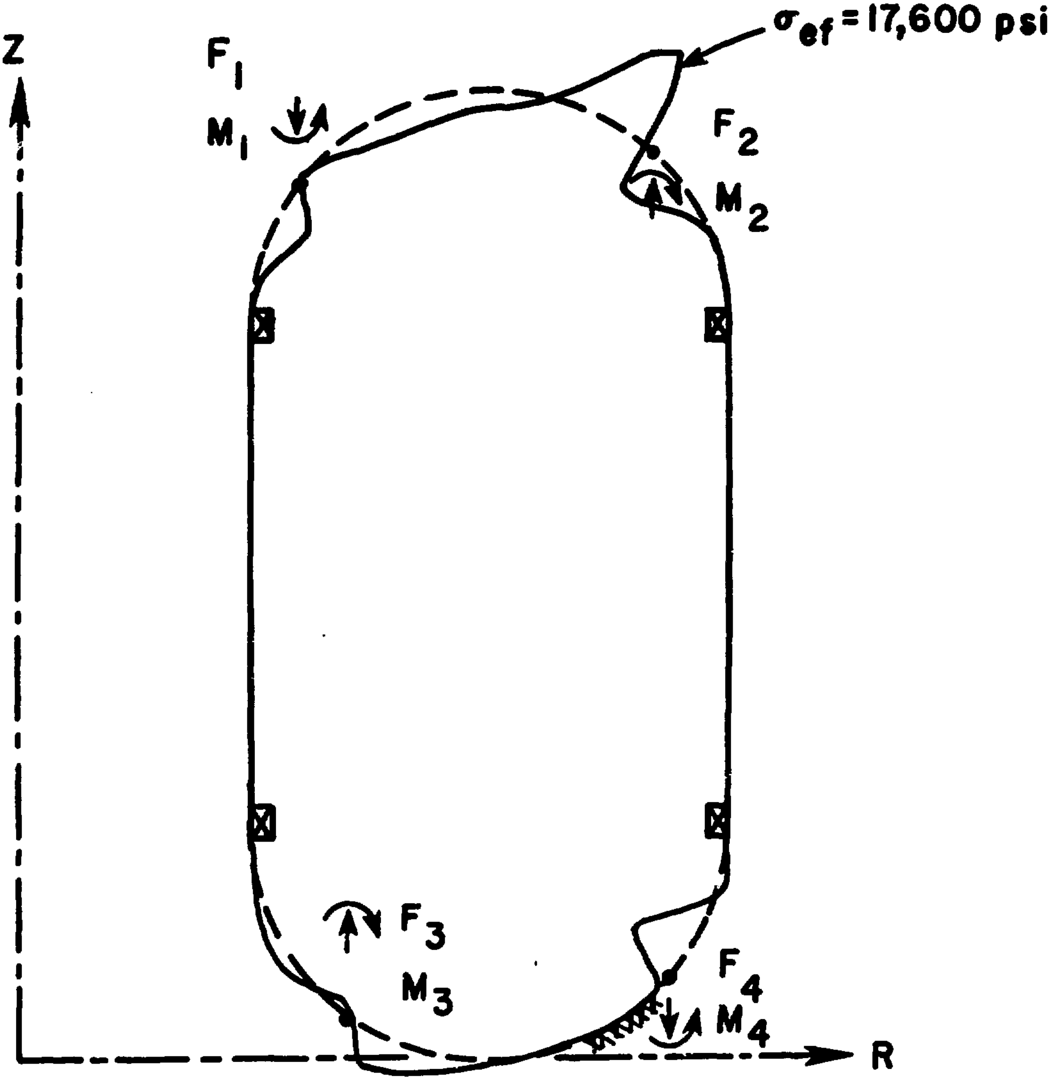

Fig. 6