ATTACHMENT 1

GENE-771-39-0794, REVISION 1
HATCH UNIT 1
SHROUD REPAIR HARDWARE STRESS ANALYSIS

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Shroud Mechanical Repair Program

Hatch Unit 1

Shroud Repair Hardware Stress Analysis

September, 1994

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ABSTRACT

This document provides the results of the stress analysis of Hatch Unit I Shroud Repair Hardware during seismic, LOCA, and other loading. The objective of the analysis is to demonstrate the structural integrity of the shroud and repair hardware under critical loading conditions.

The results show that the shroud and repair hardware meet the requirements of the Design Specification 25A5572, Rev. 2.

Executive Summary

This report provides the results of the design and stress analysis of Hatch Unit I Shroud and Repair Hardware during seismic and other loading. The shroud repair hardware is to be installed based on assumed cracking of the horizontal shroud welds. This design has the following features:

- It consists of a lower spring and an upper spring which support the shroud under seismic, LOCA and other loading. It limits the displacements and stresses of the shroud to optimal values which are within the requirements of the design specification 25A5572, Reference 8.1. The hardware is acceptable for the shroud with and without cracks in the horizontal welds.

- A 3-D linear static Finite Element Analysis (FEA) of the overall shroud and spring assembly (a 180° or one-half model), a 2-D/3-D linear/non-linear static FEA of the lower spring, upper spring, and upper support bracket and hand calculations of the other components (except for the tie rod) have been performed. The FEA was done using FEA software - COSMOS/M, 1.70 version, Reference 8.2, which has been validated for this application using test cases. The tie rod mode frequency analysis was done using ANSYS, Reference 8.6. All the FEA results have been independently verified by using handbook analysis methods. As a result of these analyses, the designs of the repair hardware components have been optimized in order to meet stiffness and stress intensity requirements.

- A design specification, Reference 8.1 has been developed to establish the load combinations and criteria for allowable stresses for structural analysis. Several different load cases were analyzed. The most severe case for the shroud is the faulted or 1/2 SME plus steam line LOCA plus dead weight. For the other components, the critical cases vary and are discussed in the report.

- Based on FEA and hand calculation results, it is concluded that all the repair hardware components and the shroud meet the requirements of the design specification.

- The forces applied to the ASME Code Section III reactor pressure vessel are analyzed and shown to be acceptable in Reference 8.5.

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1.0 INTRODUCTION

Cracks have been found during both visual and ultrasonic examination (UT) in the shroud weld joints in several Boiling Water Reactors (BWR's). As a result, for Hatch Unit 1 Shroud, SNC is taking a pre-emptive measure by implementing a corrective action using the design modification developed by GENE without performing inspections. The report deals with an analysis of the GENE design modification.

2.0 SHROUD REPAIR HARDWARE DESIGN DESCRIPTION

The repair hardware and shroud are shown in Figure 1.

Four sets of radially acting stabilizers 90 degrees apart are used to maintain the alignment of the core shroud to the reactor pressure vessel (RPV) during seismic loading. The set of stabilizers replace the structural functions of the shroud horizontal welds that are postulated to contain cracks. Each stabilizer assembly consists of a tie rod, an upper spring, a lower spring, an upper bracket, and other minor parts. The tie rod provides the vertical load carrying ability from the upper bracket to the RPV gusset attachment as well as support for the springs. The upper spring provides radial load carrying ability from the shroud, at the top guide elevation, to the RPV. The lower spring provides radial load carrying ability from the shroud, at the core support plate elevation, to the RPV. The upper bracket provides an attachment feature to the top of the shroud as well as restraint of the upper shroud weld. A middle support is also provided for the tie rod so that its natural frequency will be higher than that of forcing frequency due to flow induced vibrations. Wedges between the core support plate and shroud are also provided at each stabilizer location to prevent relative motion of the core plate and the shroud.

There are 9 horizontal welds in the Hatch Unit 1 shroud, (Figure 2). These welds are called H1-H8, with H1 being the uppermost and H8 being the attachment of the shroud support cylinder to the shroud support plate. Each cylindrical section of the shroud is prevented from unacceptable motion by the stabilizers. The motion of the sections above H1, between H1 and H2, and between H2 and H3 are restrained by the upper bracket. The upper bracket contacts the shroud and is radially supported by the upper spring which contacts the RPV. There is a feature on the upper bracket which prevents unacceptable motion of the section between H3 and H4. The section between H4 and H5 is prevented from unacceptable motions by the middle support for the tie rod. The lower spring contacts the shroud such that it prevents unacceptable motion of the sections between H5 and H6A, as well as H6A and H6B and H7. The section between H7 and H8 is prevented from unacceptable motion by the gussets.

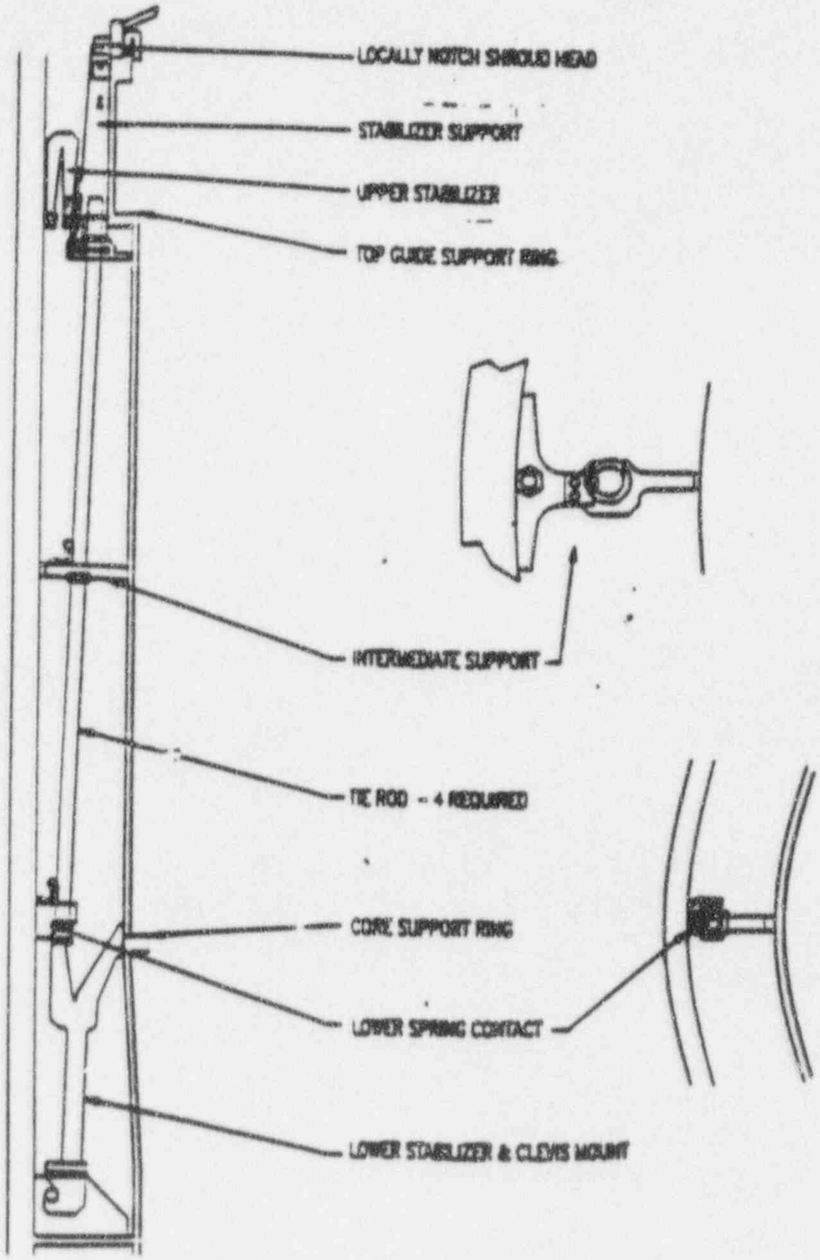

Figure 1. Pictorial View of Hatch Unit 1 Shroud Repair Hardware.

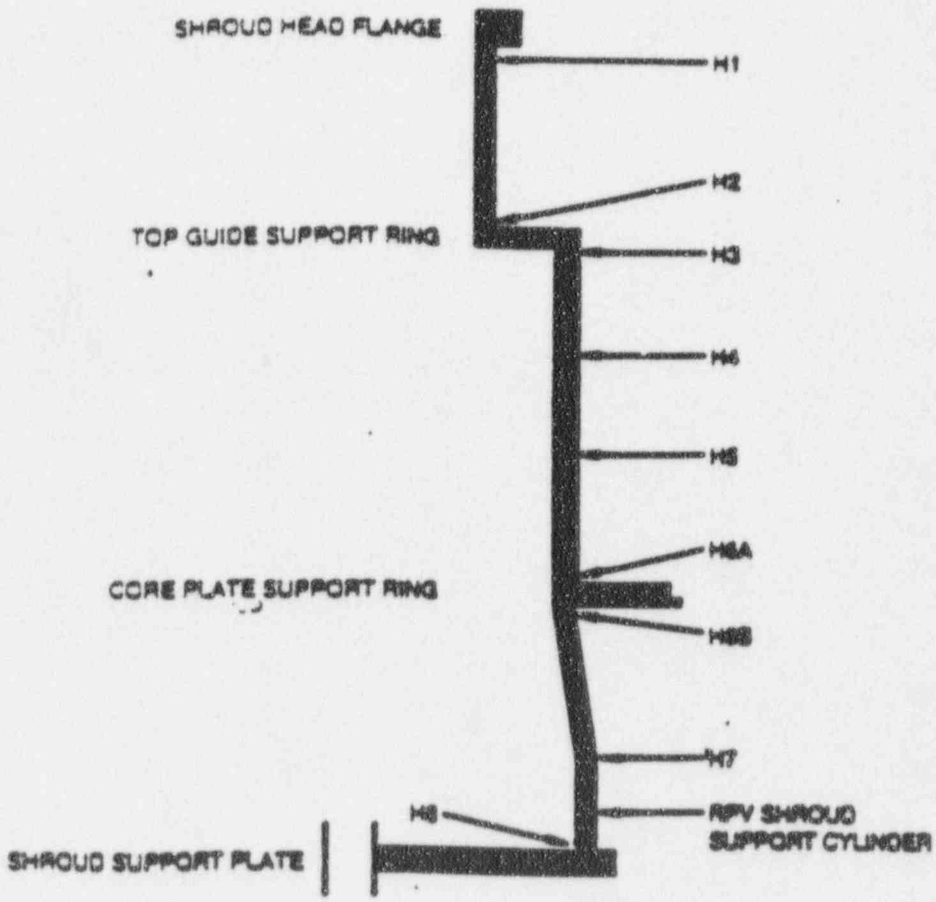

Figure 2. Shroud Weld Joint Designation.

3.0 MATERIAL PROPERTIES

3.1 The following material properties for the shroud, 304SST, 730E854 are obtained from ASME Code, Reference 8.3

<u>At 550°F</u>

S_m = 16900 psi

Modulus of Elasticity = E = 25.8×10^6 PSI

Coefficient of Expansion = $\alpha = 9.46 \times 10^{-6}$ in/in/°F

3.2 The tie rod, 316SST, 112D6312, has the following properties which are obtained from References 8.1 (design specification) and 8.3.

<u>At 525°F Operating Temperature</u>)

Sm = 22,200 PSI
$E = 25.8 \times 10^6$ in/in
$\alpha = 9.45 \times 10^{-6}$ in/in/°F

3.3 Lower spring, (Ni Cr Fe - X750), 112D6314, and upper spring, 112D6315, and upper spring bracket, 112D6316 have the following properties which are obtained from References 8.1 and 8.3.

<u>At 525°F</u> (Operating Temperature)
Sm = 47,500 psi
$E = 28.4 \times 10^6$ psi
$\alpha = 7.50 \times 10^{-6}$ in/in/°F

4.0 LOADING CONDITIONS

The applied loads which the repair hardware and the adjunct shroud must be capable of withstanding are defined in Reference 3.1. These loads are defined for the following conditions:

Condition	Description
Upset	Operating basis earthquake (OBE) + normal pressure + dead weight
Thermal	Normal and Upset
Emergency	Design basis earthquake (DBE) + normal pressure + dead weight
Emergency	LOCA + dead weight
Emergency	1/2 SME + Normal Pressure + Dead Weight
Faulted	DBE + LOCA + dead weight
Faulted	1/2 SME + MSL LOCA + Dead Weight

The seismic loads for this analysis were obtained from Reference 8.1. The thermal conditions for the thermal stress analysis are also defined in Reference 8.1.

5.0 ANALYSIS CRITERIA

The structural analysis criteria are defined in Reference 8.1. Basically, the primary and secondary stresses (Pm, Pm + Pb and Pm + Pb + Q) are compared with the allowable and the stress margins are calculated for the various loading conditions.

6.0 OVERALL SYSTEM MODEL

6.1 Model Description

A detailed 3-D FEA model, Figure 3 of Hatch 1 shroud was developed for stress analysis purposes to fully evaluate all of the loading conditions specified in Reference 8.1. The model consists of a 180 degree shroud segment composed of 3-D brick, gap, 3-D beam, and spring elements. A 180 degree segment was required in order to evaluate non-symmetric seismic loads specified in Reference 8.1. The model was created using the Geostar Module of COSMOS/M FEA software, 1.70 version. Dimensions of the shroud were obtained from 730E854 and those of the repair hardware were obtained from 112D6309 and component drawings.

The top, middle, and bottom sections of the shroud are modeled as 3-D brick (solid) elements The core support ring and the top guide ring were also modeled as 3-D brick elements. The core plate was modeled as 3-D beam elements connected like the spokes of a wheel.

At any time during a seismic event, only 1 out of a set of 4 springs is active. These 2 springs (upper and lower) are modeled as spring elements and are connected to the brick elements on the shroud by means of 3-D Beam elements with low moments of inertia and high areas. This

feature permits a pin connection (zero moment transfer) between the springs and the beam elements.

Cracks in the shroud are simulated using gap elements which are shown in Figure 4. The nodes at the ends of the gap elements are physically separated by 0.001 inch. Thus, a crack is simulated 360 degrees on the inner edge and the only connectivity between the two sections of the shroud is at the outer edge. This method of modeling the cracks in the shroud permits relative displacement between the nodes at the ends of the gap elements in the horizontal plane. It also permits a slight rolling of the rings adjacent to the gap elements. This modeling method is also conservative for the shroud stresses, since it allows some moment transfer, unlike a hinged joint in a cracked shroud.

As shown in Figure 4, only two sets of gap elements are used: one set at the core support ring and another set at the top guide ring. This scheme assumes that welds H2 and H6A have cracked 360 degrees through the wall thickness, except at the outer edges. The entire shroud analysis in this report is based on this assumption and is the worst case for the shroud. The reason for this conclusion is that if there are cracks at any other locations, the shroud sections would move radially and would not impose any stresses on the shroud.

The load from the fuel passes to the top guide and core plate and is then transferred through H2 and H6A weld joints. This is the main reason why the springs are located at these two elevations only.

6.2 BOUNDARY CONDITIONS

Symmetry boundary conditions were applied to all nodes on the x-y plane, i.e. UZ = RX = RY = 0 (UZ = Z - displacement, RX, RY are rotations). The anchor points for the spring elements which represent the reactor pressure vessel wall are fixed in all directions, i.e. UX = UY =UZ = RX =RY = RZ = 0. The nodes on the bottom surface of the shroud are fixed in the vertical direction only, i.e. UY = 0.

6.3 APPLIED LOADS

One load case was analyzed in the computer model, i.e. DBE per Reference 8.1. The other load cases are scaled by the ratio of the lower spring forces. Using iterative techniques, the loads were adjusted such that the spring forces from the COSMOS/M model were close to the following spring forces which are obtained from Reference 8.1, for DBE Case:

Upper spring force	=	32,096 lbs.
Lower spring force	=	83,257 lbs.

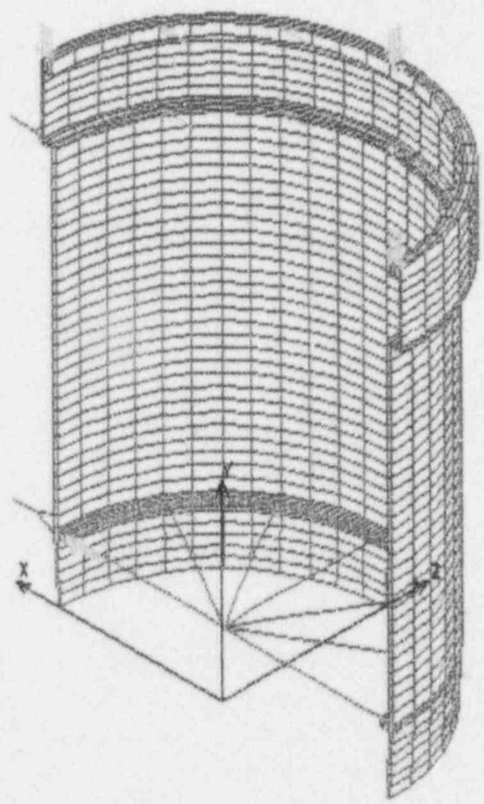

Figure 3. Shroud FEA Model (element plot)

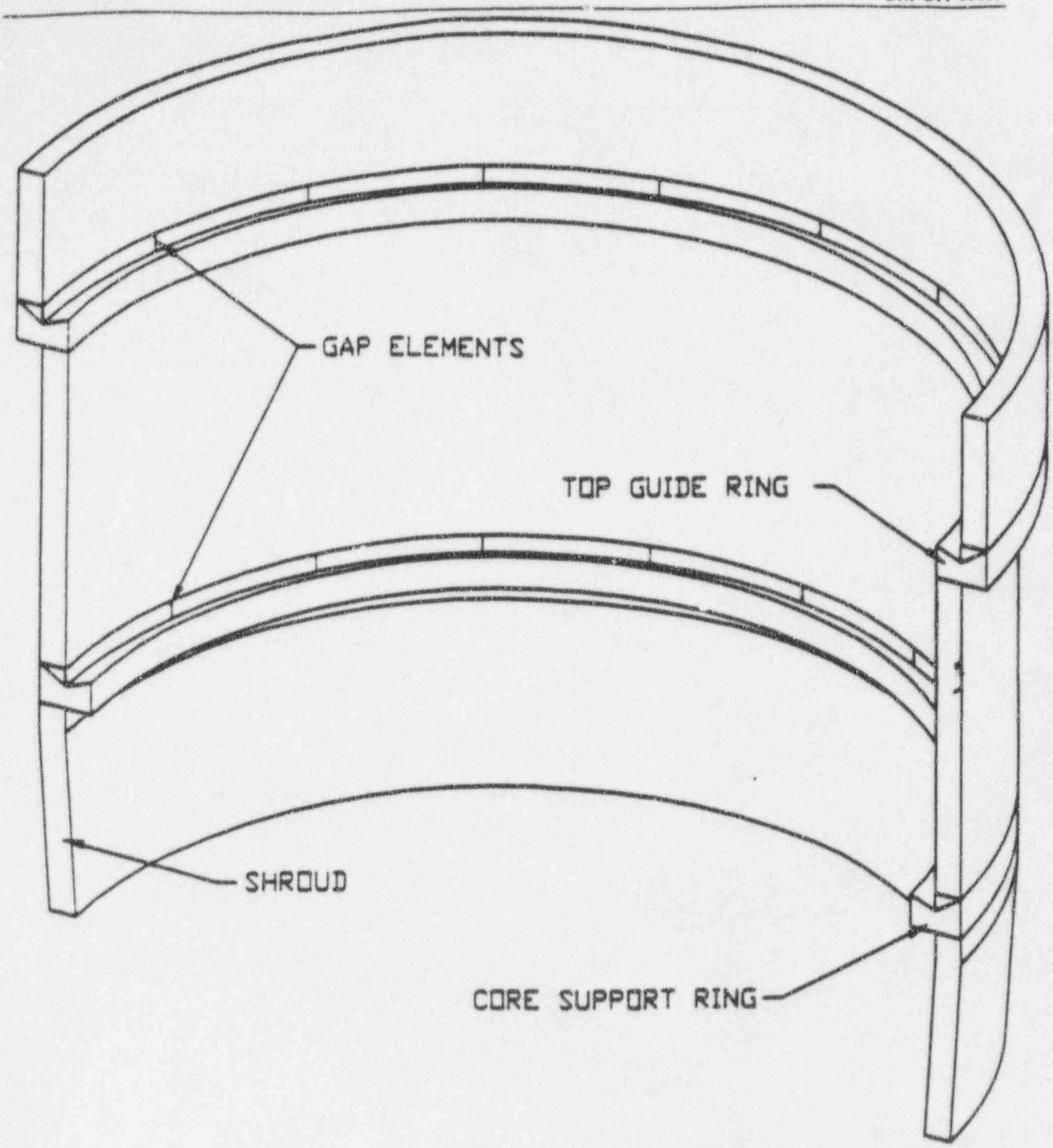

Figure 4. Gap Elements in Shroud Model to Depict Cracks.

The other case when the springs are at 45 degrees is not as severe for the shroud and therefore is not analyzed. It is, however, considered in the lower spring analysis.

A cosine distribution was assumed for the seismic loads Fig. 5 & 6 and applied along the core support location. Because of the way the core support ring is connected to the shroud, the load is maximum at 0 degrees and 0 at 90 degrees location. The same loads are applied for the nodes from 90 degrees to 180 degrees location.

Because of the way the top guide ring is connected to the shroud, the load is maximum at 90 degrees location and drops to zero at 0 degree location. This is true for shroud head loads also. For the top guide and shroud head, the loads are applied only for the quadrant from 0 to 90 degrees.

Tie rod tensile load for various cases are calculated as follows:

From Reference 8.1, the net uplift pressures are:

	Core Plate ΔPcp psi	Shroud Head ΔPsh psi
LOCA	28	30
Normal	23.8	8.4

The tie rods are in tension for the case of a 360 degree through wall shroud crack

FOR DBE + LOCA

Acp = Area of the core plate, in^2

Π = $(\Pi/4)(166^2 - 137 \times (\Pi/4)(10.875^2)$, 761E462

= 8917 in^2

Ash = Area of Shroud Head, in^2

= $\Pi/4\,(183.5^2)$, 730E854

= 26,446 in^2

Net upward force = F_{LOCA}

= ΔPcp Acp + ΔPsh Ash - Wsh

Wsh = Weight of Shroud (730E854) + weight of shroud head (732E109) + weight of top guide (730E852) + weight of core plate (732E109)

= 76,500 + 102,200 + 11,300 + 17,500

= 207,500 lbs, neglecting weight of peripheral fuel

F_{LOCA} = 28 x 8,917 + 30 x 26,446 - 207,300

= 835,556 lbs

or per rod, F_{LOCA} = 835,556/4

= 209,000 lbs

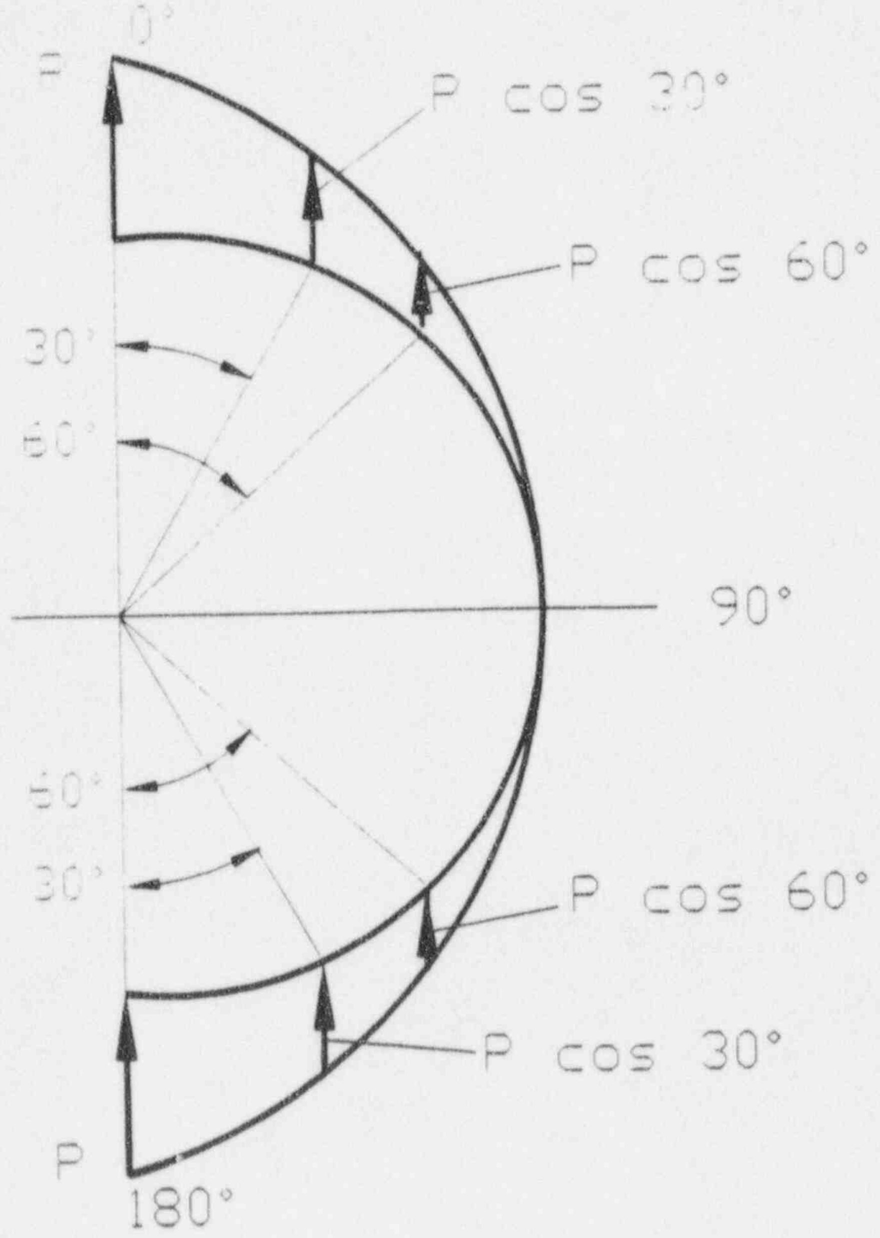

Figure 5. Cosine Loading at Core Plate

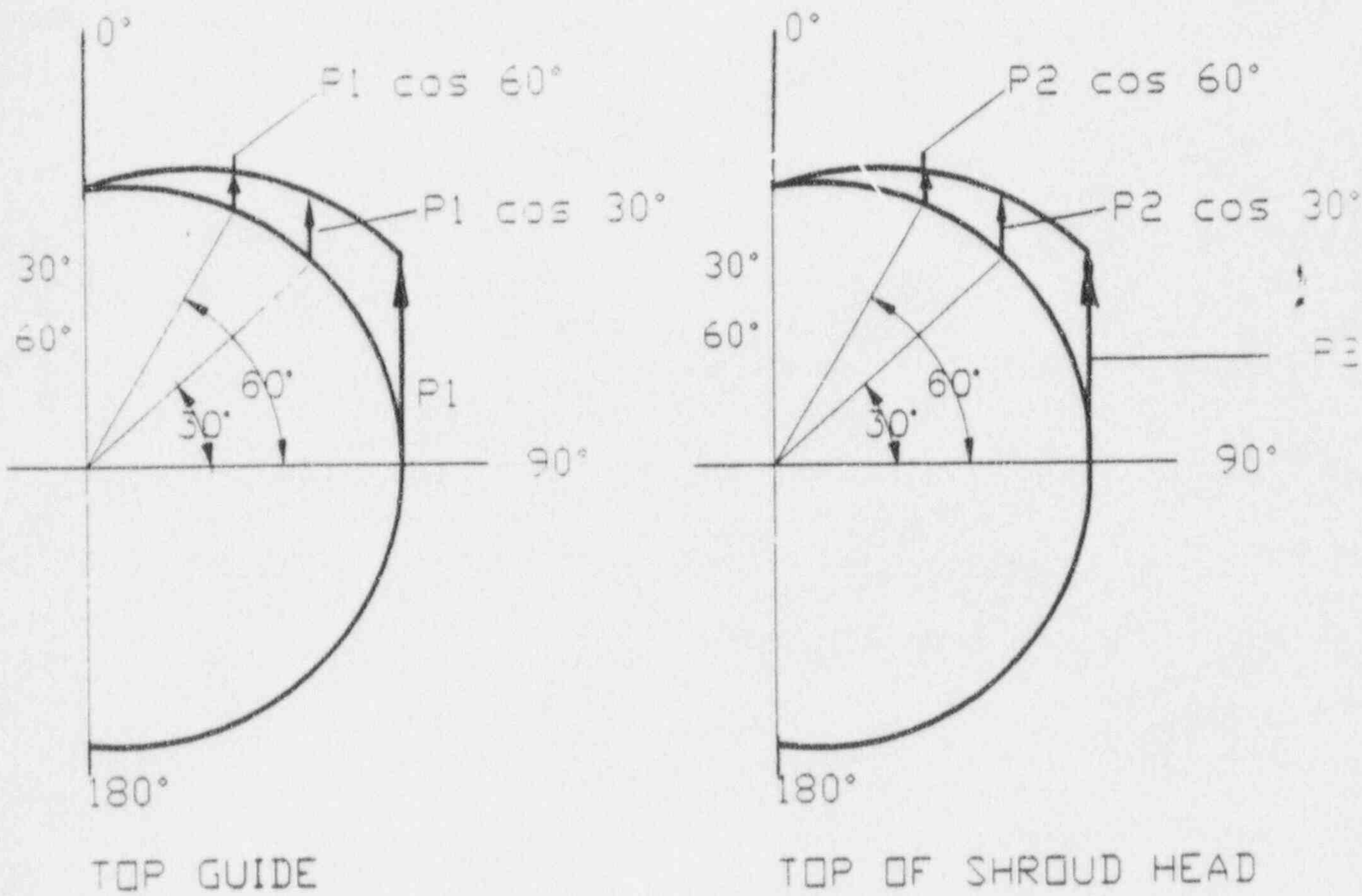

Figure 6. Cosine Loading at the Top Guide & Shroud Head

From Reference 8.1, for DBE, Tie rod moment = M = 19.87 E6 in-lbs

Assume that the moment produces Forces F in the tie rods at the mean radius R between the gusset pin and the shroud outside diameter.
Reactor pressure vessel (RPV) O.D. = 218 inches. Shroud head O.D. = 189.5 inches.

R = (189.5/2 + (218/2-4))/2 = 99.9 inches
F = M/2R = 19.87E6 x 106/(2x99.9) = 99,450 lbs

Vertical Seismic = $\pm$ (2/3) (Ground Horizontal Seismic)
= (2/3)(0.15) = $\pm$ .1g
= 0.1 x Wsh = 0.1 x 207,300
= 20,730 lbs
Per rod FVDBE = 20,730/4 = 5,200 lbs
= 99,450 + 5,200 = 104,650
For DBE + LOCA, maximum tensile load Ft
= 209,000 + 99,450 + 5,200
= 313,650 lbs

FOR DBE + NORMAL

F normal = 23.8 x 8,917 + 8.4 x 26,446 - 207,300
= 227,071 lbs
or per rod F normal = 227,071/4 = 56,768 lbs
Ft = 99,450 + 5,200 + 56,768 = 161,500 lbs

FOR OBE + NORMAL

M = 13.97 E6 in lbs
F = M/2R = 13.97 x 10^6/(2x99.9) = 66,920 lbs
FV_{OBE} = (2/3)x0.08 x 207,300 = 11,056 lbs/4 = 2,764 or 2,800 lbs
FT = 66,920 + 2,800 + 56,768 = 126,500 lbs

1/2 SME + NORMAL + W

M = 34.3 E6 in lbs
F = 34.3 E6/(3x99.9) = 114,500 lbs, The factor 3 is used per Reference 8.9
F 1/2 SME = 114,500 + 56,768 + 5,200 = 176,500 lbs

1/2 SME + MSL LOCA + W

M = 22.7 E6 in lbs
F = 22.7 E6/(2x99.9) = 113,614 lbs
F (1/2 SME + LOCA) = 113,614 + 209,000 + 5,200 = 327,900 lbs

VERTICAL LOADS IN SHROUD MODEL   DBE + LOCA

At 0 degrees	=	1/2 x 313,650 =156,825 lbs
90 degrees	=	F_{LOCA} = 209,000 lbs
180 degrees	=	(F_{LOCA} - F - FVDBE)/2
	=	(209,000 - 99,450 - 5200)/2
	=	52,175 lbs

Each of these loads is distributed over 4 nodes at the top of the shroud flange.
Since the objective of this analysis is only to evaluate the effect of the repair hardware, only the above loads are included in the model. Therefore, the pressure loads need not be considered in evaluating the shroud in this analysis. The vertical pressure force is included in the tie rod loading. The hoop stresses due to internal pressure are included in the SI values.

6.4 FEA MODEL RESULTS , SHROUD

6.4.1 Faulted Case (DBE + LOCA)

The nodal stress intensity (SI) (Pm + Pb) plot from COSMOS/M is shown in Figure 7. The maximum SI = 27,400 psi at 0 degree location on the core support ring. This is a peak value of SI and is calculated for a lower spring force of 92,480 lbs. versus required value of 83,257 lbs.
Ratio = 83,257/92,480 = 0.9
Hoop stress = pr/t = (29x177. 5/2)/1.5 =1657 psi
SI Max = 21,828 psi, after linearizing the nodal stresses across the thickness.

Total SI Max = 0.9 x 21,828 + 1,657 = 21,302 psi

Allowable SI	=	3 Sm
	=	3 x 16,900 = 50,700 psi

or, SI Max < SI allow

Hence the shroud stresses are acceptable.

The displacement of the shroud in X - direction = UX = 0.9x0.62 = 0.56 inch at the lower spring location.
This is < 1.49 inches, UX allow, Reference 8.1, Hence O.K.
The spring forces are :

Upper Spring	=	22,876 lb.
Lower Spring	=	92,480 lb.

Since this case is used only for shroud stresses, these spring forces are within an acceptable range for all practical purposes.

It can be seen that the calculated upper spring force of 22,886 lbs from COSMOS model is less than 32,096 lbs (required) at the top guide ring. This is acceptable since the stresses in the top guide location are much smaller than those at the core plate ring. Thus, the stresses in the core plate ring are governing.

Pm = 9,163 x 0.9 = 8,247 psi for the shroud, after linearizing the stresses.
Sm = 16,900 psi
SI allow = 2 x 16,900 = 33,800 psi
Pm < 2 Sm, Hence OK

Also, at the final iteration, the following horizontal shroud loads were used:

In Figure 6, P = 2,642 lbs

Thus, at the core plate ring, total load applied for the full model = 2x2x(P+Pcos60° + Pcos 30°)
= 2 (2(2,642 + 2,288 + 1,321)) = 25,000 lbs.

In Figure 6, top guide, P1 = 9,547 lbs
or Load for full model = 2(P1 + P1 Cos 60° + P1 cos 30°)
= 2(9,547 + 8,268 + 4,774)
= 45,178 lbs

In Figure 5, shroud head, P2 = 9,547 lbs
or Load at shroud head = 45,178 lbs
= 115,356 lbs

This is equal to Sum of spring forces,
= (22,876 + 92,480) lbs

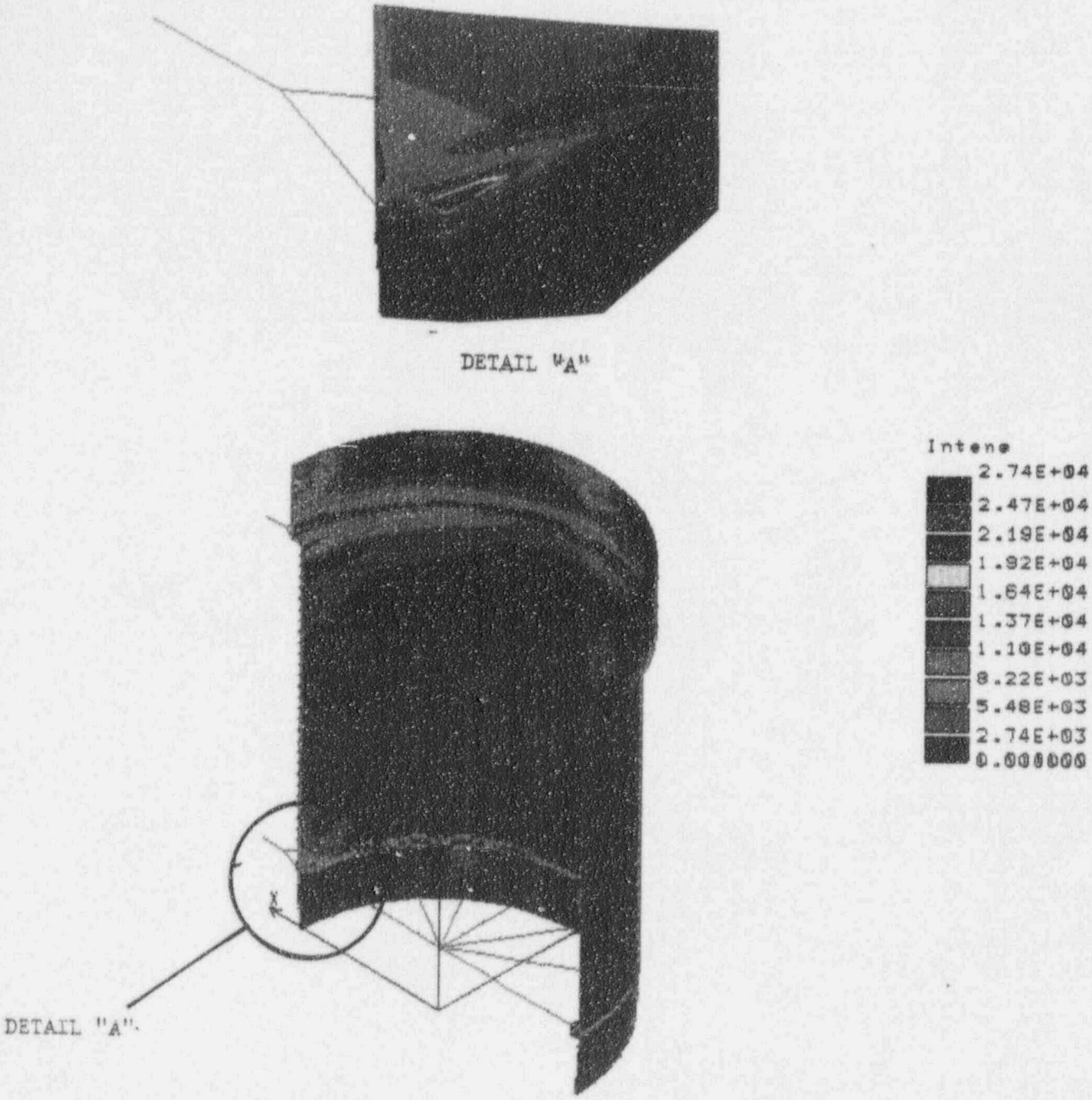

Figure 7. Shroud Stress Intensity Plot (Faulted, DBE + LOCA)

6.4.2 Upset Case (OBE + Normal Pressure)

Since the stresses at the core plate ring are governing, the stresses can be scaled down for this case.

Ratio	=	39,437/92,480 =0.43
Pm	=	9,163 x 0.43 = 3,940 psi < Sm or 16,900 psi, Hence OK
Pm + Pb	=	0.43 x 21,828 = 9,386 psi from previous section.
Hoop stress	=	pr/t = (23.8 x 177.5)/(2x1.5) = 1,408 psi
Total Pm + Pb	=	9,386 + 1,408 = 10,794 psi < 1.5 Sm or 25,350 psi , Hence OK
SI MAX	=	Pm + Pb + Q = 10,794 + 9,126 (thermal stress, from Appendix A-1)
	=	19,920 psi < 3 Sm or 50,700 psi, Hence OK

The same comment as in the previous section is valid for the top guide ring stresses.

UX = 0.27	=	< 0.75 UX allow, Reference 8.1

6.4.3 Emergency (DBE and Normal Pressure)

This Emergency case is a DBE with normal operating pressure differences. The same FEA results for the shroud can be used as for the faulted event of DBE + LOCA

Maximum Pm + Pb	=	SI Max = 21,828 psi, from Section 6.4.1
SI Allow	=	2.25 Sm (Reference 8.1)
	=	38,000 psi

SI Max < SI Allow

Pm = 8,247 psi and less than 1.5 Sm.

Hence the shroud meets the Emergency requirements.

UX = 0.56 inches, UX allow = 1.49 inches

Hence UX < UX allow OK

6.4.4 Emergency (LOCA Only)

This Emergency case is a LOCA without any seismic.

LOCA Load for tie rod = F_{LOCA} = 209,000 lbs., from Section 6.3.

This load acts on the two plates of the upper support bracket and is then transferred to the shroud flange.

Bearing Area	=	2.00 x 2 x 5.25 from 730E854 and 112D6316
	=	21 in^2
Bearing Stress SB =		209,000/21 = 9,952 psi

S Allow = SY = 17,800 psi, Reference 8.3

SB < S Allow

Hence ok.

Using the method of appendix A-1 for conservatism.

SI Max = (Pm +Pb) = (9,126 x 209,000)/57,571 = 33, 130 psi for F = 209,000 x 4= 836,000 lbs.

SI allow = 2.25 Sm = 38,000 psi

Hence SI Max <SI allow. Hence ok.

Pm is very small and <1.5 Sm.

UX is very small for LOCA and < 1.12 inches Hence OK

Thus, the design is acceptable.

7.0 COMPONENT FEA MODELS AND HAND CALCULATIONS

7.1 Lower spring, (112D6314) FEA

The element plot is shown in Figure 7. The lower spring is modeled as shell 4T (thick shell) elements with varying thicknesses. Three different load cases were analyzed.

7.1.1 Combined Radial and Axial Loads (faulted case, DBE + LOCA)

As axial load of 313,650 lb. is applied at the junction between the tie rod and the spring. At the same time, a radial load of 83,257 lb. (Ref. 8.1) is applied at the point of contact between the shroud and the spring.

For boundary conditions, the node at the block (near the tie rod) is fixed in Y direction, i.e. UY = 0. Also, for one node at the pin at the lower end, UX = 0 and for another node at the pin, UY = UZ = RX = RY = RZ = 0. The nodal stress intensity (Pm + Pb) plot is shown in Figure 8.

SI Max	=	100,843 psi, after linearizing the stresses.
Sm for spring material	=	47,500 psi at 525°F, Reference 8.1
Allowable SI	=	3 x Sm, faulted conditions
	=	142,500 psi.

Hence, SI Max < SI allow.
Pm = 36,553 psi, from COSMOS Model, after linearizing the stresses
SI allow = 2.0 Sm = 95,000 psi
Hence, Pm <2.0 Sm

Hence, the lower spring meets the design specification requirements.

The X - displacement at the node where the axial load is applied = UX = 0.3448 inches from COSMOS/M model.
Load = 313,650 lb.
Axial stiffness = 313,650/0.3448 = 909,724 lb./in

This is used in Section 7.5.1 to calculate the overall axial stiffness of the tie rod assembly.

7.1.2 Radial Load + Normal Axial Load (Emergency, DBE + Normal)

A radial load of 83,257 lb. and a normal axial load of 161,500 lbs (Section 6.3) are applied at the shroud contact point.

Pm + Pb	=	90,821 psi, after linearizing the stresses from COSMOS model
Allowable SI	=	2.25 Sm, emergency case
SI	=	2.25 x 47,500
	=	106,875 psi

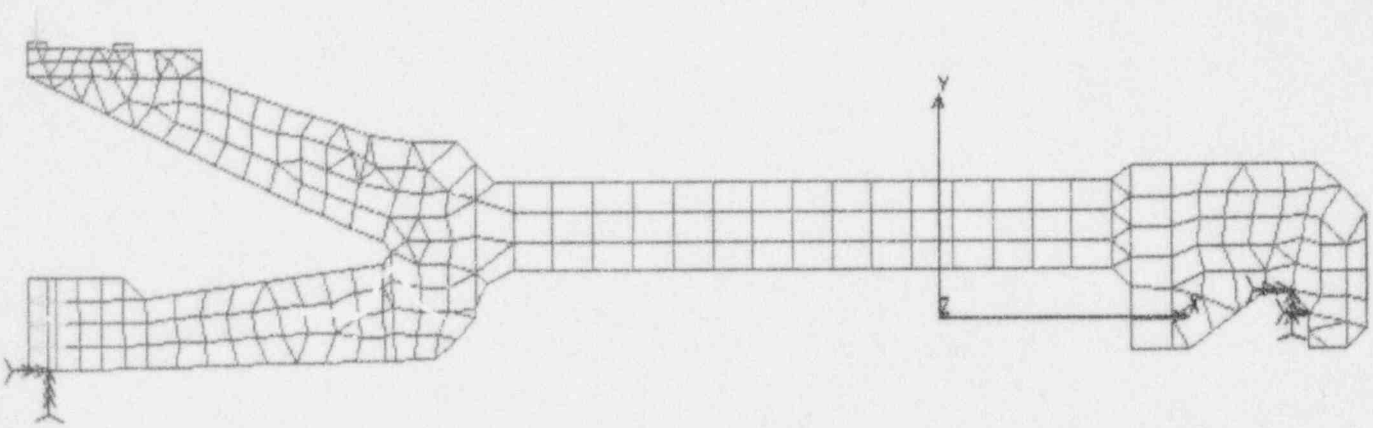

Figure 8. Lower Spring Element

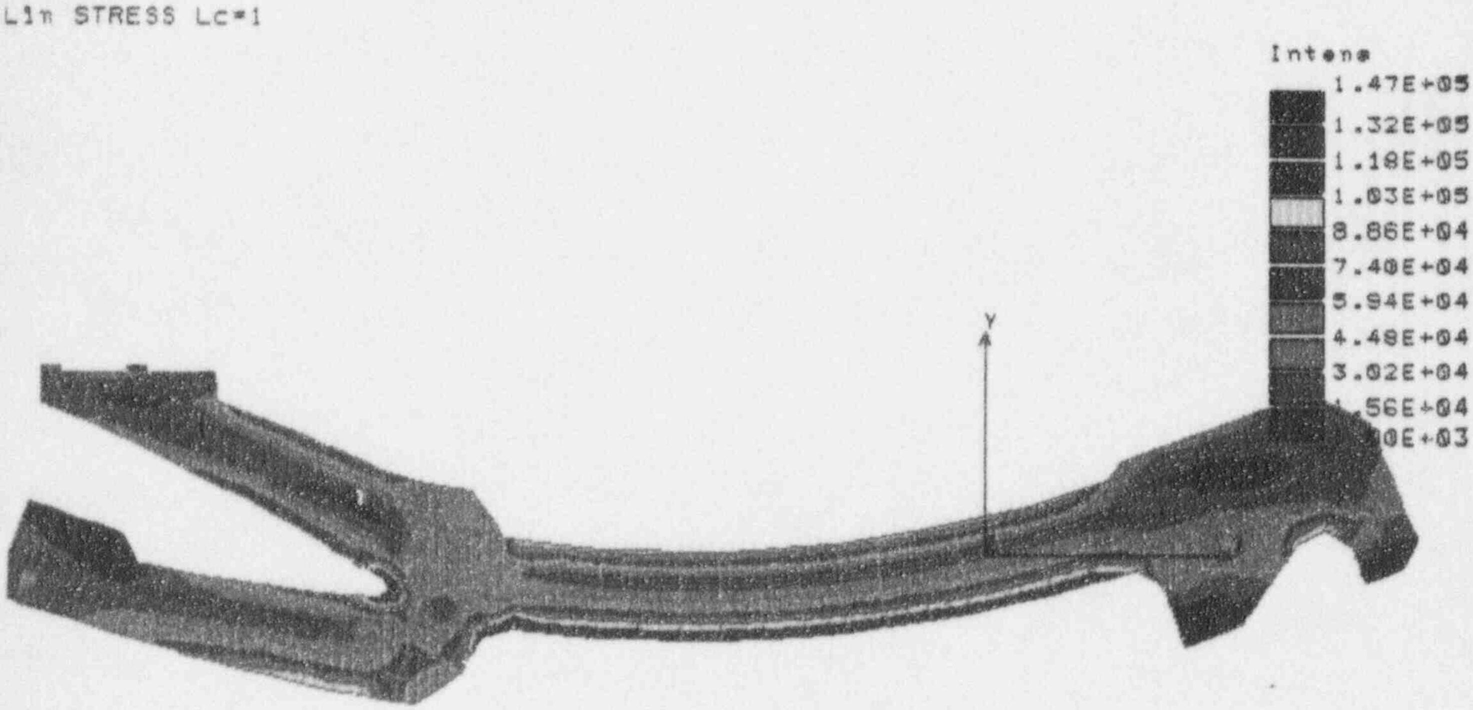

Figure 9. Stress Intensity Plot (lower spring).

Hence, SI Max < SI allow, thus OK.
Pm = 32,077 psi from COSMOS Model, after stress linearization
S allow = 1.5 Sm = 71,250 psi
Pm < S allow   Thus ok

The Y displacement at the load = UY = 0.56 inches.
Or radial stiffness = (83,257/0.56) = 148,526 lb./in.

This stiffness is close to the required value of 150,000 lbs/in, and hence is acceptable.
UY allow = 1.12 inches , Reference 8.1
Hence, UY < UY allow

7.1.3 Lower spring at 45 degree location

In this case, the bottom spring will slide and this case is very similar to Section 7.1.2, thus the spring stiffness would be the same as if the force acts on spring axis. Hence this case is acceptable.

7.1.4 Emergency (LOCA only)

Axial load on spring = 209,000 lbs. Use 211,000 lbs for conservatism.
Pm = 45,504 psi from COSMOS Model
SI allow = 1.5 Sm = 71,250 psi
Pm < SI allow
Hence, ok
Pm + Pb = 77,500 from COSMOS Model
SI allow = 2.25 Sm = 106,875 psi
Pm + Pb < Si allow Hence OK

7.1.5 Upset (OBE + Normal Pressure)

Axial Load = 126,500 lbs, Section 6.3
Radial Load = 39,437 lbs, Reference 8.1

Pm + Pb = SI max = 67,500 psi from COSMOS model
SI allow = 1.5 Sm = 71,250 psi > SI max. Hence OK.
Pm = 35,500 psi from COSMOS model
Sm = 47,500 psi > Pm. Hence OK.
Hence, the design is acceptable for all the load cases.

7.2 UPPER SPRING FEA

The upper spring (one half model) is modeled as SHELL 4T elements. Symmetry boundary conditions were applied on the plane of symmetry. A unit load of 32,210 lb. is applied at the point which represents the reactor pressure vessel wall.

The element plot is shown in Figure 10 and stress intensity plot is shown in Figure 11.

7.2.1 Emergency & Faulted Cases

Load = 32,096 lbs, Reference 8.1, for faulted
SI Max (Pm + Pb) = (94,982 x 32096)/32,210 psi = 94,700 psi, after linearizing the stresses.
SI allow = 2.25 Sm = 106,875 psi, using Emergency allowables for conservatism
SI Max < SI allow
Pm = (39,693 x 32,096)/32,210 = 39,500 psi
SI allow = 1.5 Sm = 71,250 psi
Pm < SI allow, hence, ok

Hence, the design is acceptable.
Under the load, UX = 0.636 inches
Radial stiffness = 32,210/0.636 = 50,645/2 = 25,322 lb./in.
Stiffness for both halves of spring in series = 50,645/2 = 25,322 lb./in.

Radial Shroud stiffness = 136,406 lb./in from a separate model of the top section of the shroud.

Stiffness of the shroud and upper spring
= (25,322 x 136,406) / (25,322 + 136,406)
= 21,357 lb./in

This is very close to the required stiffness of 20,000 lb./in.
UX for top guide =32,210/21,357 = 1.51, for faulted case.
Hence the design meets the requirements of the design specification, Reference 8.1.

7.2.2 Upset Case (OBE + Normal Pressure)

For OBE case, load = 16,113 lbs., reference 8.1
Pm + Pb = SI Max = (16,113/32,210) x 94,982
= 46,600 psi
SI allow = 1.5 Sm = 71,250 psi
SI < SI allow Hence OK.
Pm = (16,113/32,210) x 39,693 = 19,900 psi
SI allow = 1.0 Sm = 47,500 psi
Pm < SI allow, hence ok

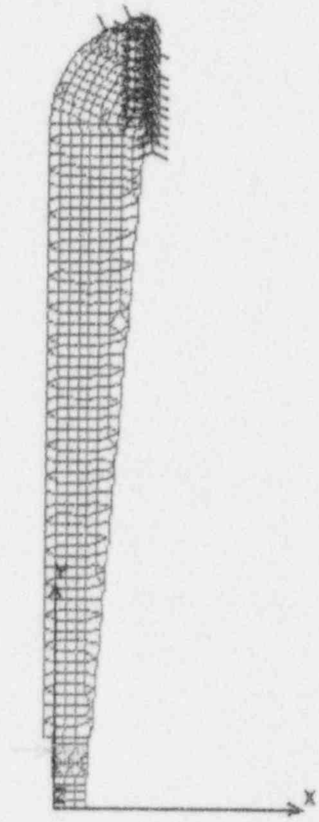

Figure 10. Element Plot (Upper Spring).

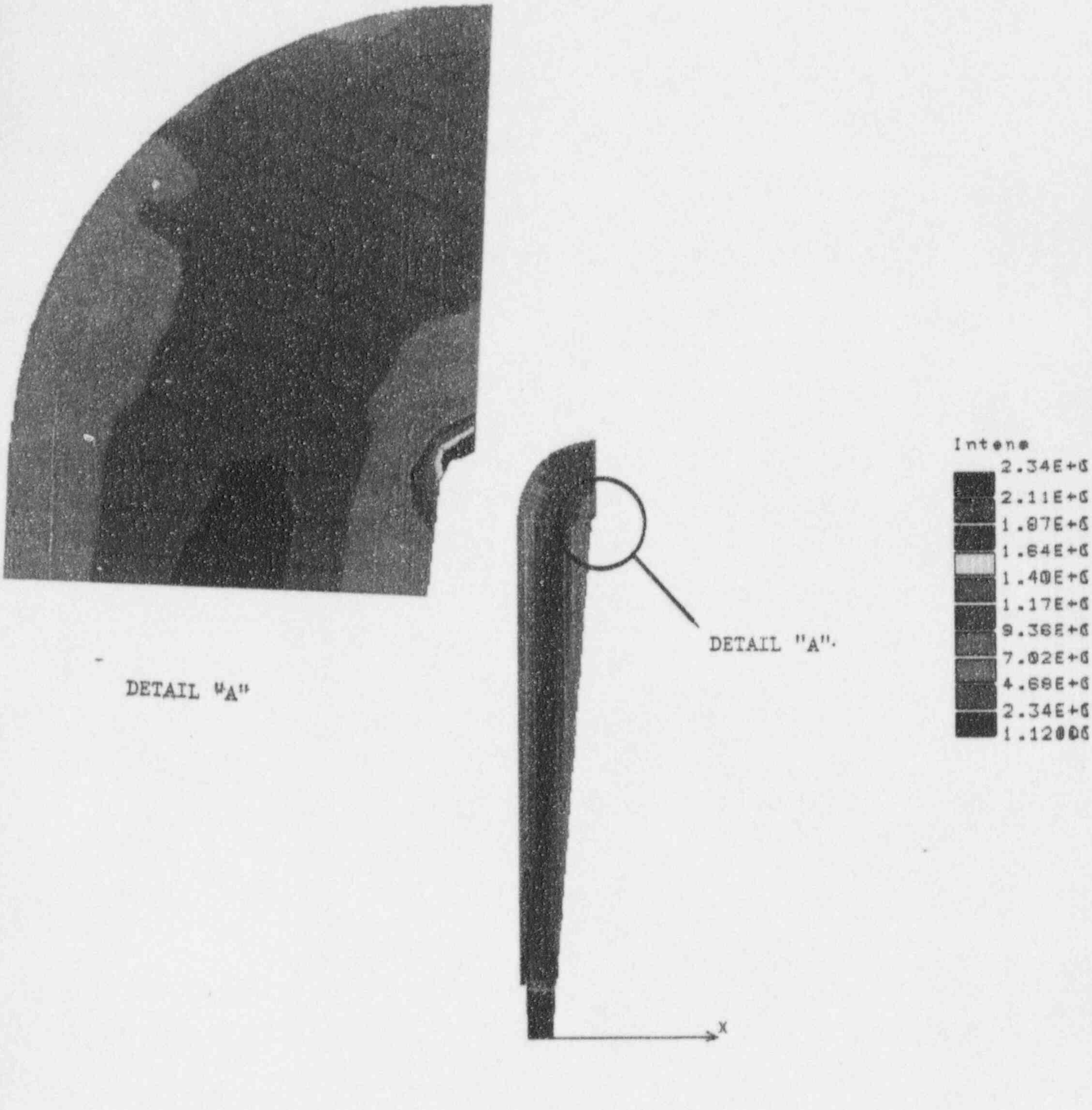

Figure 11. Stress Intensity Plot (Upper Spring).

7.3 Upper Support Bracket FEA

7.3.1 Faulted (LOCA + DBE)

The element plot is shown in Figure 12. The model consists of shell 4T elements. A downward unit load of 326,500/2 = 163,250 lb. is applied on the bottom plate and is distributed over 3 nodes. Because of geometry and loading symmetry, only one half of the upper support bracket is modeled. The calculated load for this load case for this load case = 313,650 lbs, Section 6.3

For boundary conditions, symmetry boundary conditions were applied on the plane of symmetry. For one node at the top where the bracket is reacted upon by the shroud flange, UX = UY = RX = RY = 0. For one node on the vertical leg near the inside diameter of the shroud flange, UX = 0. Also, UX = 0 for one node where the support bracket contacts the shroud on the outside diameter.

The stress intensity (Pm + Pb) plot is shown in Figure 13.

SI Max = (90,832 x 313,650)/326,500 = 87,257 psi, after linearizing the stresses.
Sm = 47,500 psi
SI Allow = 3 Sm (Faulted)
= 142,500 psi

SI Max < SI Allow

Pm = 35,436 psi SI allow = 2.0 Sm = 95,000 psi or Pm < SI allow, hence ok.

Thus, the design is acceptable.
Also, UY = 0.0883 inches under the load
Axial Stiffness = (326,500)/ 0.08829 = 3.692E6 lb./in.

7.3.2 Emergency 2 (LOCA)

F_{LOCA} = 209,000 lb.
SI Max = (209,000/326,500) x 90,832 = 58,144 psi
SI Allow = 2.25 Sm = 106,875 psi
Pm = 23,600 psi Pm <1.5 Sm or 71,250 psi
Hence, the design is acceptable.

7.3.3 Upset (OBE + Normal Pressure)

Load = 126,500 lbs, Section 6.3
SI = 35,200 psi; SI allow = 1.5 Sm = 71,250 psi
SI < SI allow, hence ok
Pm = 14,300 psi < 1.0 Sm of 47,500 psi

7.3.4 Emergency 1 (DBE + Normal Pressure)

SI max = 44,900 psi,
SI allow = 2.25 Sm = 106,875 psi > SI max Hence O.K.
Pm = 18,300 psi < 1.5 Sm or 71,250 psi Hence O.K.

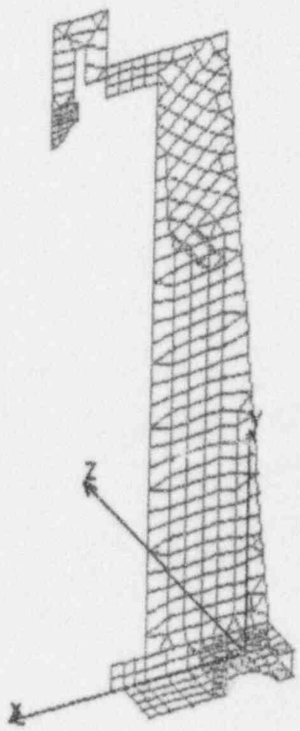

Figure 12. Element Plot (Upper Support Bracket)

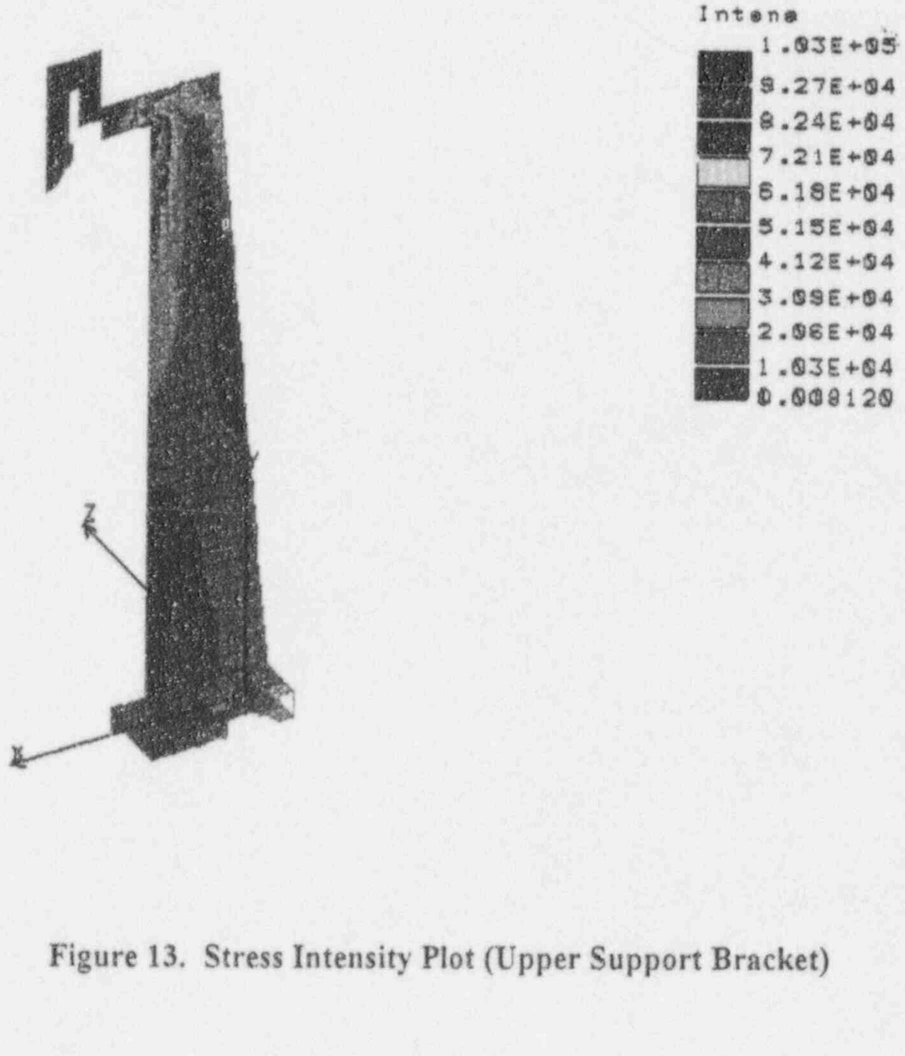

Figure 13. Stress Intensity Plot (Upper Support Bracket)

7.4 Tie Rod Analysis

7.4.1 Tie Rod Hand Calculations, Faulted (DBE + LOCA)

The maximum tensile axial load on tie rod = 313,650 lb. from Section 6.3

Thread Relief Area	=	$(\Pi/4) \times 3.33^2$ 112D6312
	=	8.709 in^2
Tensile Stress	=	Pm = (313,650)/8.709
	=	36,014 psi
SI Allow	=	2 Sm faulted case
	=	2 x 22,200, from Section 3.0
	=	45,600 PSI

Pm < SI Allow. Hence, OK.

7.4.2 For Emergency (LOCA only)

Axial Load	=	209,000 lb., Section 6.3
Pm	=	209,000/8.709
	=	24,000 psi
S Allow	=	1.5 Sm
	=	1.5 x 22,200
	=	34,200

Pm < S Allow. Hence, O.K.

7.4.3 For Upset Case (OBE + Normal Pressure)

Axial Load	=	126,500 lbs., Section 6.3
Pm	=	126,500/8.709
	=	14,600 psi
SI Allow	=	Sm
	=	22,800 psi

Pm < SI allow. Hence, O.K.

7.4.4 Emergency (DBE + Normal Pressure)

Axial Load	= 161,420 lbs, Section 6.3
Pm	= 18,600 psi < 1.5 Sm or 34,200 psi Hence OK.

7.4.2 Tie Rod FEA and Natural Frequency.

The tie rod FEA was done using FEA software ANSYS. It was modeled as 2-D beam elements (Stiff3) and the element plot is shown in Figure 14. Included in the model are the upper support bracket and lower spring both modeled as 2-D beam elements. A load of 57,571 lb. (Tie Rod Preload) was applied at the Upper support bracket.

For boundary conditions, for the node at the clevis pin,

UX = UY = UZ = ROTX = 0

For the nodes at the tie rod to lower spring junction, mid support and at the top of the upper support bracket, UY = UZ = ROT X = 0 (Rollers)

A mode frequency analysis was performed using ANSYS. The frequencies for the first 6 modes are shown in Table 3. The hydrodynamic mass was included in the model.

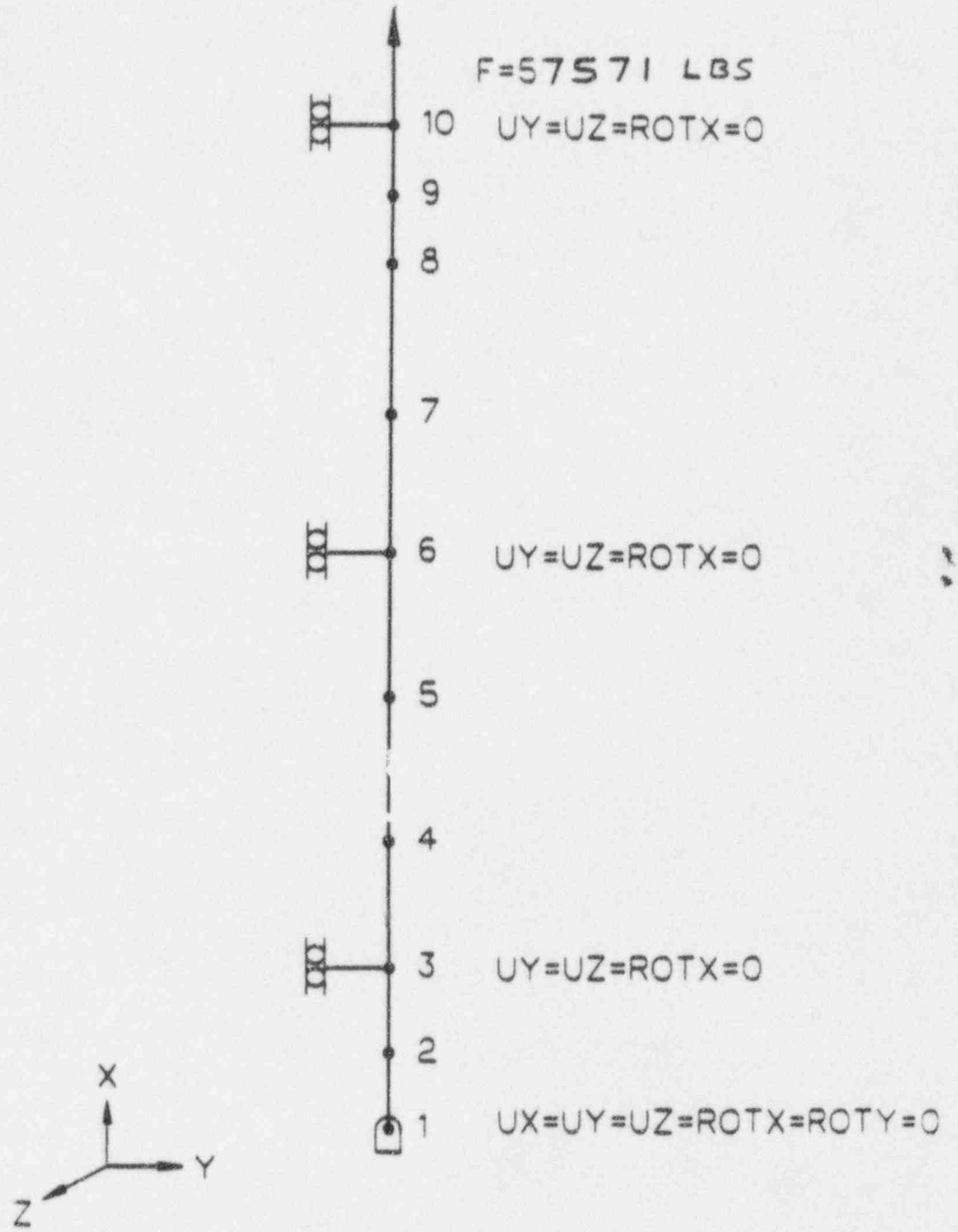

Figure 14. Tie Rod FEA Model (ANSYS).

Table 1. Tie Rod Mode Frequency List.

Mode	Frequency (Cycles/Time)
1	17.15
2	41.81
3	82.73
4	97.91
5	100.87
6	135.28

The natural frequency = 17.15 Herz. This is compared with the forcing frequency due to flow induced vibrations (FIV).

7.4.3 Effect of FIV on Tie Rod Assembly.

The tie rods are 3.5 inches diameter bars that are 172.65 inches long. They are threaded on both ends, have a tension preload due to differential thermal expansion of 57571 lbs. and there is a mid-span support.

The tie rods have the same diameter as the guide rods and have approximately one fourth the unsupported length compared to the guide rods which have not experienced FIV.

The calculated lowest natural frequency of the tie rod is 17 Hz.

The potential excitation forces would come from the shroud, has a lowest natural frequency well below that of the tie rod and the fluid flow.

The tie rods are installed at 45, 135, 225, and 315 degree locations. The flow in these regions is primarily parallel to the tie rods. At the limiting elevation, the bulk flow velocity is approximately 6.3 feet per second. Even if this was a cross flow velocity, the lowest tie rod frequency is well above the classic excitation

$$f = .22\ v/d$$

$$f = .22 \times 6.3\ \ 12/3.5 = 5\ \text{Hz}$$

The only remaining concern is the cross flow adjacent to the jet pump inlet. The tie rod will be approximately 5.5 inches from the nearest approach of the jet pump inlet. Figure 17 predicted the cross flow velocity near the jet pump inlet at the tie rod.

The resulting cross flow is approximately 3 feet per second, which yields a frequency of approximately 2 Hz. Note that this cross flow velocity only exists for approximately 1 foot of length. This frequency is less than 17 hertz, hence OK.
Therefore, the design is acceptable.

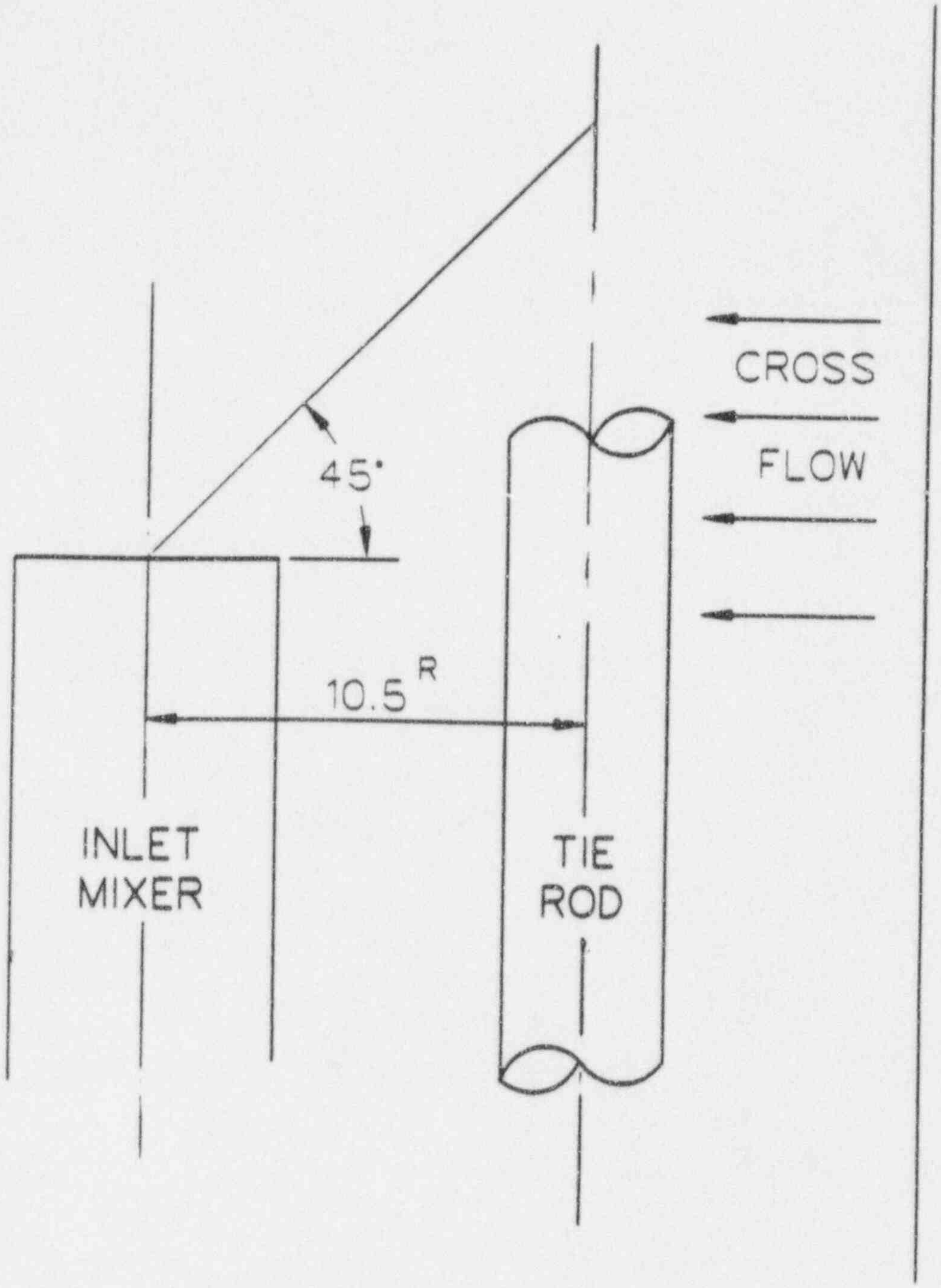

Figure 15. Tie Rod FIV Model.

7.5 Thermal Stress Analysis

7.5.1 Overall Axial Stiffness Of Tie Rod Assembly

Axial stiffness of a upper support bracket = kb = 3.692 E5 lb./in, from Section 7.3.

Axial Stiffness of Lower Spring = ks = 9.097 E5 lb./in, from Section 7.1.1.

For Tie Rod

$$A = (\Pi/4) \times 3.5^2, \text{ 112D6312}$$
$$= 9.62 \text{ in}^2$$
$$E = 25.8 \text{ E6 psi, from Section 3.2}$$
$$L = 177.65 - 5.00, \text{ 112D6312}$$
$$= 172.65 \text{ inches}$$

$$\text{Stiffness} = kr = (AE)/L = (9.62 \times 25.8E6)/172.65$$
$$= 1.44E6 \text{ lb./in}$$

k = Axial stiffness of Tie Rod assembly

$$(1/k) = (1/ks) + (1/kr) + (1/kb)$$

Substituting and simplifying

$$(1/k) = 1.0993E\text{-}6 + 0.6969E\text{-}6 + 2.7086E\text{-}7$$

or $k = 483{,}790$ lb./in

This stiffness is very close to the required value of 500,000 lb./in, Reference 8.4 and hence is acceptable.

7.5.2 Normal Conditions

This represents the case when the shroud is at 534°F and the tie rod assembly is at 522°F. Since the coefficient of thermal expansion for Inconel X-750, the material for the upper support bracket and the lower spring is less than that of the shroud material (304SS), the shroud grows more than the tie rod assembly. This produces differential thermal expansion and a tensile load on the tie rod assembly.

Upper Support Bracket Expansion:

$$L1 = 48.9 - 4.50, \text{ 112D6317 and 112D6318}$$
$$= 44.4 \text{ in.}$$
$$\alpha = 7.50 \text{ E - 6 in/in./}^\circ\text{F}$$
$$\Delta T = (522 - 70) = 452\ ^\circ\text{F}$$
$$\text{Hence, } \Delta L1 = 44.4 \times 7.50 \text{ E-6} \times 452$$
$$= 0.151 \text{ inches}$$

Tie Rod Expansion

$L2 = 172.65$ inches, from Section 7.5.1

$\alpha = 9.45$ E-6 in/in/°F

$\Delta T = 452°F$

$\Delta L2 = 172.65 \times 9.45$ E-6 $\times 452$

$= 0.737$ inches

Lower Spring Expansion:

$L3 = 65.50$ inches, 112D6314

$\alpha = 7.50$ E-6 in/in /°F

$\Delta T = 464°F$

$\Delta L3 = 65.50 \times 7.5$ E-6 $\times 452$

$= 0.222$ inches

For Total Tie Rod Assembly

$LTA = 44.4 + 172.65 + 65.50$

$= 282.55$ inches

$\Delta LTA = 0.151 + 0.737 + 0.222$

$= 1.11$ inches

Shroud and Inconel 600 Expansions:

Shroud Length Ls = 267.44 inches from 730E854

Length of Inconel 600 piece = 282.55 - 267.44

LI= 15.11 inches

α 600 = 7.85 E - 6 in/in/°F, Reference 8.3

α Shroud = 9.46E-6 in/in/°F, Section 3.1

$\Delta Ts = 534 - 70 = 464°F$

$\Delta Ls = 267.44 \times 9.46E - 6 \times 464$

$= 1.1739$ inches

$\Delta LI = 15.11 \times 7.85E - 6 \times 464$

$= 0.055$ inches

Total ΔLSA for shroud assembly = 1.229 inches

Net differential expansion = 1.229 - 1.11 = 0.119 inches

Stiffness of Tie Rod assembly:

= k = 489,570 lb./in., from Section 7.5.1

$\therefore$ Force in Tie Rod = 483,790 x 0.119

= 57,571 lb.

Tie Rod area at the thread relief = $(\Pi/4) \times 3.33^2 = 8.709$ in^2

Tensile Stress in Tie Rod = 57,571/8.709 = 6,610 psi

Tensile Stress in Tie Rod = 57,571/8.709 = 6,610 psi

or, Pm = 6610 psi

Sm = 22,200 psi, from Section 3.2 Pm < Sm, Hence, O.K.

7.5.3 Upset Conditions

This is the case when the shroud temperature is at 430°F and the tie rod assembly is at 300 °F. Using the method of Section 7.5.2.

Upper Support Bracket Expansion:

LI = 48.9 - 4.50, 112D6317 and 112D6318

= 44.4 in.

α = 7.20E - 6 in/in. /°F

ΔT = (300 - 70) = 230°F

Hence, $\Delta L1$ = 44.4 x 7.20 E-6 x 230

= 0.0735 inches

Tie Rod Expansion

L2 = 172.65 inches, from Section 7.5.1

α = 9.23 E-6 in/in/°F

ΔT = 230°F

$\Delta L2$ = 172.65 x 9.23 E-6 x 230

= 0.3665 inches

Lower Spring Expansion

L3 = 65.50 inches, 112D6314

α = 7.20 E-6 in/in/°F

ΔT = 467°F

$\Delta L3$ = 65.50 X 7.2 E-6 X 230

= 0.1085 inches

For Total Tie Rod Assembly:

LTA = 44.4 + 172.65 + 65.50

= 282.55 inches

ΔLTA = 0.0735 + 0.3665 + 0.1085

= 0.5485 inches

Shroud and Inconel 600 Expansions:

Shroud Length Ls = 267.44 inches from 730E859

Length of Inconel 600 piece = 282.55 - 267.44

LI = 15.11 inches

α 600 = 7.73 E-6 in/in/°F, Reference 8.3

α Shroud = 9.23 E-6 in/in/°F, Section 3.1

ΔTs = 400 - 70 = 330 °F

ΔLS = 267.44 x 9.23 E-6 x 330

ΔLS = 0.8146 inches

= 15.11 x 7.73 E-6 x 330

= 0.0385 inches

Total Δ LSA for shroud Assembly = 0.8531 inches

Net differential expansion = 0.8531 - 0.5485 = 0.3046 inches

Stiffness Of Tie Rod Assembly = k = 483,790 lb./in.

$\therefore$ Force in Tie Rod = 483,790 x 0.3046 = 147,362 lb.

Tensile Stress in Tie Rod =Pm= 147,362 / 8.709 = 16,921 psi.

Sm = 22,200 psi, Section 3.2

Pm < Sm , Hence, O.K.

Hence, the tie rod meets the design specification requirements for both normal and upset conditions.

7.6 Summary of Hand Calculations of Other Component

These calculations are filed in DRF B11-00604.

7.6.1 Mid Support Bracket, 112D6330

The load on this bracket = 25,000 lb., Reference 8.1 , Emergency Case

The maximum bearing stress = 10,246 psi

Sy = 17,800 psi, Reference 8.3, for 316SST

S allow = 1.5 Sy, faulted case

= 1.5 x 17,800 = 29,025 psi

Pm + Pb < S allow,

Hence, the design is acceptable.

7.6.2 Core Plate Spacer/Wedge, 112D6333, 112D6334.

Load on Spacer = 83,257 lbs, Reference 8.1

Bearing Stress = Sb

= 7,177 psi

Sy = 17,800 psi

S Allow = 2.0 Sy = 35,600 psi, Faulted Case

Pm + Pb < S allow.

Hence, the design is acceptable.

7.6.3 Shroud Head Modification (Cut Out)

Pm + Pb = SI = 26,439 psi during lifting. Sallow = 1.5 Sm = 30,000 psi at 70 degrees F

Hence, Pm < S allow

Hence, the shroud head meets the design specification requirements.

7.7 Fatigue Evaluation

For the shroud, since the number of applied load cycles for faulted and other load cases is very small, no formal fatigue analysis is required.

The only critical case is for the tie rod thermal upset case.

From Section 7.5.3, S_{alt} = (Pm/2) x k = 17123/2 = 8562 x 4 = 34,248 psi
From, Figure I-9.1 of Reference 8.3, N allow = 2 x 10^5 cycles.

N actual = 10 cycles, Reference 8.1 or N actual < N allow, Hence, O.K.

For the shroud, the same conclusion holds true. Thus, fatigue requirements are satisfied.

7.8 Gusset Pin Hole Bearing Stress and Pin Shear Stress

Maximum Load = Tie Rod Load = P
For faulted (DBE + LOCA)
P = 313,650 lb.
Bearing area = 2 x 3 = 6in ^{2}
Bearing Stress = Sb = 52,275 psi
S allow = 2 x 1.5 x Sy = 2 x 1.5 x 28,580 = 85,740 psi, Reference 8.3, alloy 600
Sb < S Allow
The other cases are not as severe.

The gusset pin is in double shear.
Shear area = (2x3.14)/4) x 9 = 14.1372 sq.in.
Shear Stress = Tau = 313,650/14.1372 = 22,200 psi
S allow = 2 x 0.5Sm = 2 x 0.5x47,500, (X-750, same as lower spring material)
= 47,500 psi
Tau < Sallow ... Hence, o.k.

8.0 References

8.1 25A5572, Revision 2, Shroud Repair Hardware Design Specification.

8.2 COSMOS/M, Finite Element Structural Analysis Computer Code, Structural Research and Analysis Corporation, Low Angeles, California

8.3 ASME, Boiler & Pressure Vessel Code (B&PV), Section III, Appendices, 1989 Edition.

8.4 GENE-771-48-0894, Revision 1, Seismic Design Report For Hatch Unit 1, Nuclear Power Plant With Shroud Repair, GENE,

8.5 225A5594, Revision 1, Reactor Pressure Vessel Code Stress Report, GENE.

8.6 ANSYS, Finite Element Structural Analysis Code, Swanson Analysis Systems, Houston, Texas.

8.8 Formulas for stress and strain, R.J. Roark & W.C. Young, fourth & fifth Editions, McGraw-Hill Book Co. , New York, NY

8.9 GENE-771-43-0894 Rev. 0, Shroud Repair, 1/2 SME Time History Seismic Analysis, GENE.

Appendix A-1

Shroud Stresses due to differential thermal expansion

Normal Case

Bearing area = 21 sq. in from Section 6.4.4

Tie rod load = 57,571 lbs., Section 7.5.2

Bearing stress = Sbrg = 58,258/21 = 2741 psi

S allow = Sy = 17,800 psi, Reference 8.3

Sbrg < Sy Hence ok

The stress in the top guide ring, Figure A-1.0 with failed H2 & H3 welds is obtained from Reference 8.8, page 334

$Sb = (MRc)/I$ $M = 6F/2\Pi R$

$c = 6$ inches

R= 94.75 inches, 730E854 F = 57,571 x 4 = 230,284 lbs.

If the ring is disconnected by failure of welds H2 & H3, ring rotation will occur and the load will be reduced. From Reference 8.8, theta = $(MR^2)/EI$

and $Sb = (6F \times 6)/(2\Pi\, t^2 b)$

F = 230,284, R = 94.75, t = 3, b = 7.5, E = 25.8E6

theta = 0.0723 radians = 4.14°

UY = (7.5/2) tan 4.14° = 0.272 inches

Thus, the ring will rotate with a stiffness of k= F/(2 UY)

or k = 230,284/(2 x 0.272) = 423,726 lbs./in.

S_{bs} for shroud = (Sb x k)/(k + k tie rod)

Sb = $(6 \times 230{,}284 \times 6)/(2 \times \Pi \times 3^2 \times 7.5)$

= 19,546 psi

k tie rod = 489,570 lbs./in

S_{bs} = (483,790/(483,790 + 489,570)) x 19,546

= 9126 psi This is used in Section 6.4.2

Thermal Upset

Using the method of the previous section,

Sbs = 23,088 psi for F = 147,362 x 4 = 589,448 lbs.

or Pm + Pb + Q = 23,088 psi

S allow = 3Sm = 3 x 16,900= 50,700 psi

Pm + Pb + Q < S allow, hence ok

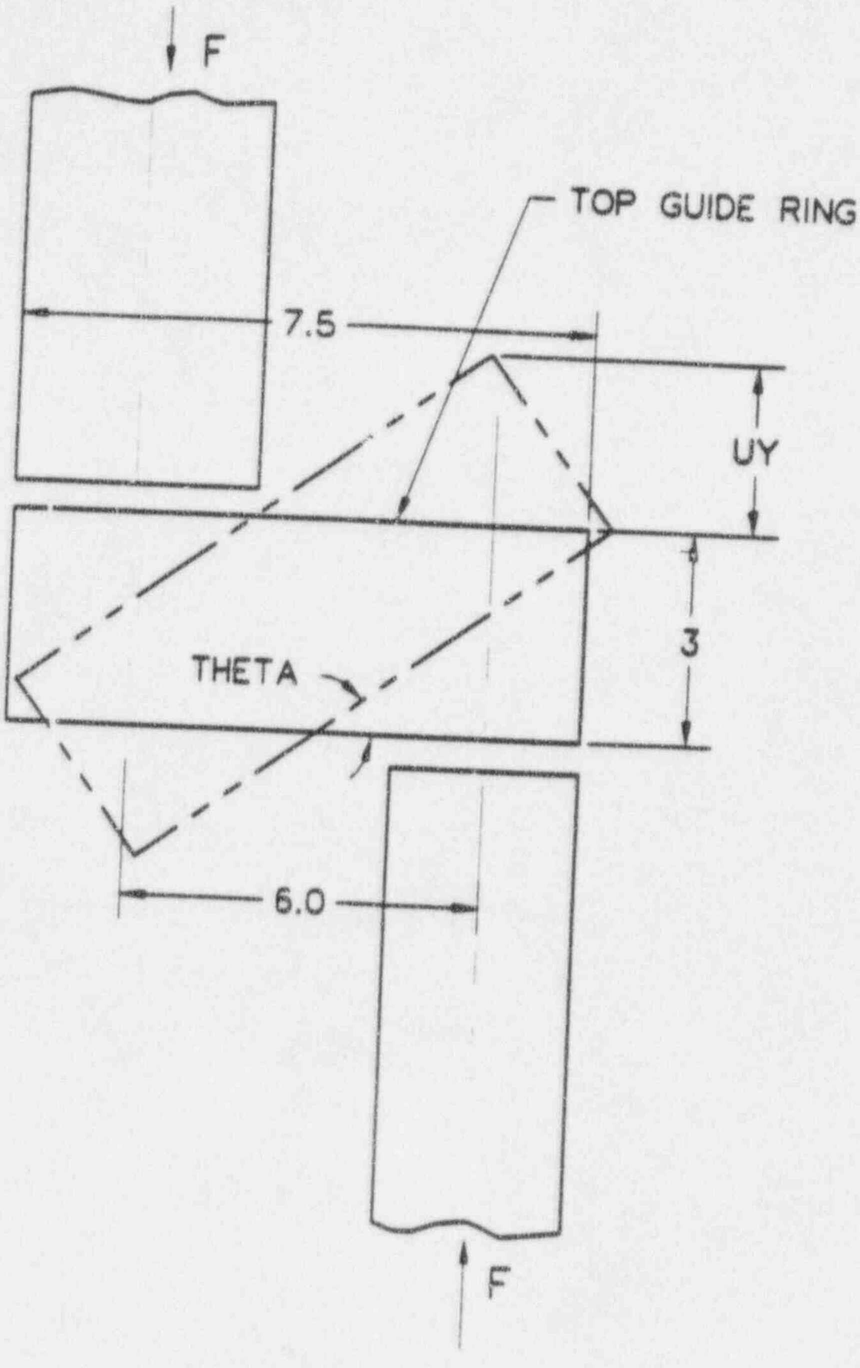

Figure A-1.0 Top Guide Ring Rotations

Appendix A-2

1/2 SME LOADING

A-2.1 SHROUD

1/2 SME + P + W

From shroud model used in Section 6.4, the lower spring force = 92,480 lbs. From Reference 8.1, required Lower Spring Force = 78,000 lbs

Ratio = 78,000/92,480 = 0.84
Pm = 9,163 x 0.84 = 7,730 psi <1.5Sm or 25,350
Pm + Pb = 21,828 x 0.84 + 1,657 = 20,000 psi < 2.25 Sm or 38,000 psi

1/2 SME + MSL LOCA + W

Using the method discussed in the previous paragraph,
Ratio = 189/92.4 = 2.04
Pm = 18,700 psi < 2Sm or 33,800 psi
Pm + Pb = 46,200 psi < 3Sm or 50,700 psi

A-2.2 UPPER BRACKET

Scaling the stresses by the ratio of axial forces the following results are obtained:

1/2 SME + P + W

Axial Load = 257,800 lbs Ratio = 257,800/326,500 = 0.79
Pm = 29,200 psi < 1.5Sm or 71,250 psi, Hence OK
Pm + Pb = 71,800 psi < 2.25 Sm or 106,875 psi, Hence OK

1/2 SME + MSL LOCA + W

Axial Load = 313,650 lbs
Pm = 33,500 psi < 2Sm or 95,000 psi
Pm + Pb = 94,000 psi < 3Sm or 142,500 psi

A-2.3 TIE ROD

1/2 SME + P + W
Pm = 114,500/8.709
Pm = 13,147 psi < 1.5Sm or 34,200 psi, Hence OK
Pm + Pb = 13,147 psi < 2.25 Sm or 51,300 psi, Hence OK

1/2 SME + MSL LOCA + W
Pm = 327,900/8.709
Pm = 37,650 psi < 2Sm or 45,600 psi
Pm + Pb = 37,650 psi < 3Sm or 68,400 psi

A-2.4 UPPER SPRING

To analyze the 1/2 SME cases, the design of the deflection limiter (stop) was considered. The two halves of the spring close at a displacement of 2.0 inches. Through linear analysis, it was found that Pm + Pb was greater than the yield strength (Sy) when the spring halves closed. Thus, a non-linear plasticity analysis was performed using COSMOS. The von-mises isotropic hardening option for material non-linearity was chosen for the shell 4T elements. Also, the bilinear plasticity method was used. For this option, the following material properties were used:

Sy = 92,300 psi from Reference 8.3
Tangent modulus is obtained as follows:
Su = 142,300 psi
Elongation at yield = 0.2%
Hence, slope of the stress strain curve beyond yield = (142,300 - 92,300)/(0.3 - 0.002)
= 1.6879E5
Both loading and unloading was modeled by specifying the load in dummy time steps as follows:

Time	Time Step	Load
0	0	0
1	10	Full value
2	20	0

By trial and error the following results were obtained:

Time Step	Time	Force lbs	Pm + Pb psi	Displacement inches (both halves)
10	1	52,700	111,000	2.54
9	0.9	47,430	109,000	2.04
8	0.8	42,160	108,000	1.74

Hence, it was concluded that when the spring closes at the stop or at 2.00 inch displacement, Pm + Pb = 109,000 psi (peak value) and Load = 47,000 lbs.

The peak stress occurs at the top corner. After linearization, the following results are obtained:

Pm = 45,300 psi < 1.5Sm or 71,250 psi for 1/2 SME + P + W
Pm + Pb = 106,300 psi < 2.25Sm or 106, 875 psi

Thus for both 1/2 SME cases, Load = 49,700 lbs > 47,000 lbs and hence the spring will bottom out.

1/2 SME + MSL LOCA + W

The faulted allowables are higher and Pm and Pm + Pb are the same as above. Also, from COSMOS model, at time step 20, corresponding to zero load (unloading), displacement = 0.22 inches, under the load. This is a permanent displacement and is < 1.4 inches allowable for 1/2 SME + P + W & <1.86, allowable for 1/2 SME -MSL LOCA + W.

Hence, the upper spring meets the requirements for both 1/2 SME cases.

A-2.5 LOWER SPRING

The lower spring also has a stop in the Y Section. This is considered in the (1/2 SME + MSL LOCA + W) case.

1/2 SME + NORMAL PRESSURE

Using the method of Section 7.1,
radial load = 78,000 lbs
axial load = 176,500
Pm = 30,800 psi after linearizing the stresses
< 1.5 Sm or 71,250 psi Hence OK
Pm + Pb = 87,000 psi < 2.25Sm or 106,875 psi,
Uy at the load = 0.51 inch < 1.12 Uy allow Hence OK
There is no permanent displacement for this load case

1/2 SME + MSL LOCA + W

For this case, axial load = 327,900 lbs
Radial load = 189,000 lbs
A linear analysis shows that the lower spring goes plastic under this load, but its deflection is limited by the stop. A plasticity analysis similar to that of the upper spring was performed. The following results were obtained by trial and error.

Time Step	Force lbs	Pm + Pb psi	Displacement inches
10	138,000	99,200	1.31
9	124,200	95,300	0.925
8	110,400	88,800	0.772

The stop limits the displacement to 0.95 inches. For this displacement, by approximate interpolation,

Pm + Pb = 96,000 psi < 3Sm or 142,500 psi

Pm = 39,600 psi < 2.0 Sm or 95,000 psi Hence OK

Also, during unloading when the force is reduced to zero, Uy = 0.4 inches under the load. This is a permanent displacement < 0.66 Uy allowable. Hence OK. Thus, the lower spring is acceptable for both 1/2 SME load cases.