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LOFT TECHNICAL REPORT LTR 1119-33

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MASTER

PRELIMINARY ANALYSIS OF LOFT REACTOR VESSEL CLOSURE

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EG&G Idaho, Inc.

IDAHO NATIONAL ENGINEERING LABORATORY

DEPARTMENT OF ENERGY

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LOFT TECHNICAL REPORT LOFT PROGRAM

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SUMMARY

This LTR contains analysis determining if the two-piece flat head closure considered by INC for the LOFT reactor vessel meets ASME Code, Section III requirements. These are considered general analyses, however, not intended to replace detailed thermal, stress, and fatigue calculations required for final vessel configuration.

This document was originally Appendix A-3 to CDD 1119F.

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APPENDIX A-3

PRELIMINARY ANALYSIS OF LOFT REACTOR VESSEL CLOSURE

MPR ASSOCIATES, INC., TECHNICAL REPORT M-239 ADDENDUM I, VOLUME I (NOVEMBER 1970)[a]

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PRELIMINARY ANALYSIS OF
LOFT REACTOR VESSEL CLOSURE

MPR-239

ADDENDUM I
VOLUME I

Prepared for:

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MPR ASSOCIATES, INC.

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I. INTRODUCTION

The purpose of this analysis is to determine if the two piece flat head closure concept presently being considered by INC for the LOFT reactor vessel will meet the structural and fatigue requirements of Article 4 of Section III, "Nuclear Vessels," of the ASME Boiler and Pressure Vessel Code, for the anticipated LOFT operational requirements. As previously noted in the analysis of the hemispherical head, report MPR-239, dated July 1970, these analyses are intended to be approximate in nature, and do not take the place of the detailed thermal, stress and fatigue calculations which the vessel manufacturer is required to perform for the final vessel configuration.

The head assumed for the analysis is shown on Figure 2-3 of CDD 1.1.1.9A (the reactor vessel component design description). The vessel and bolting are identical to those assumed for the previous analysis except that the bolts are longer to accommodate the thicker outer head.

The analyses performed include stress analyses based on classical stress analysis methods and proven computer programs, fatigue analyses based on the fatigue criteria of Section III of the Code, and thermal analyses based on multi-dimensional computer programs. These analyses are limited to the thicker, outer head which mates with the existing reactor vessel flange on the basis that (1) less design flexi-

bility is available in designing the outer head to fit the existing vessel, and (2) the smaller, inner head represents a less severe stress problem than the thick outer head, particularly for thermal transient conditions. Thus, if the outer head will meet Section III stress requirements, it is reasonable to expect that the smaller inner head can also be designed to meet Section III requirements.

Loading conditions considered in these analyses include internal pressure, initial bolt preload, gasket axial force, gasket shear force, bolt shear force, and bolt bending moments. The analysis did not include full redundancy between the inner and outer heads although the approximate bolting and gasket loads between the inner and outer heads were included. The effects of thermal deflections and gasket friction were investigated. As in the previous analysis, loadings from vibration of machinery attached to the vessel, and seismic effects have not been considered.

Thermal, stress, and fatigue analyses were performed for the following conditions:

- Preload only.
- End of 100°F/hr heat-up transient.
- Hot, steady-state operation at design conditions.
- End of 100°F/hr cooldown transient.
- Worst times in the blowdown and emergency core cooling system (ECCS) injection transient, considering the case in which re-flooding of the closure area of the vessel occurs.

In addition to the stress and fatigue analyses of the vessel closure discussed above, the deflections of the head and vessel flange in the seal area were checked to determine if metallic O-ring seals would perform satisfactorily under the imposed conditions.

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II. SUMMARY

The results of the thermal, stress and fatigue analyses of the vessel closure and the evaluation of metallic O-ring seal adequacy are summarized below:

A. Stress and Fatigue Analysis of Vessel Closure

The results of the stress and fatigue analyses are presented in Table II-1. Figure II-1 shows the areas for which stresses are tabulated. In the table, maximum calculated stress intensities and fatigue usage factors, where appropriate, are presented for each area analyzed. The corresponding allowable stress intensities and the allowable cumulative fatigue usage factors are also included.

Based on the information in Table II-1, it is concluded that the existing vessel shell and flange will meet the structural and fatigue requirements for the following conditions:

- Design pressure - 2500 psi
- Design temperature - 650° F
- Bolt preload - 12×10^6
- Heat-up/cooldown - 1500 cycles at 100° F/hr
- Closure opening and closing operations - 360 cycles
- Blowdown and ECCS operations - 360 cycles

The bolts and outer head meet the structural but not the fatigue requirements for the cyclic conditions listed above. The head and bolts will meet the fatigue requirements of Section III of the ASME Code for the following conditions:

	<u>Allowable No. of Cycles*</u>	
	<u>Bolts</u>	<u>Head</u>
° Heat-up/cooldown	450	530
° Closure opening and closing operations	110	125
° Blowdown-ECCS operations	110	125

*Allowable number of cycles for the head is based on the analysis method of Code Case 1441 as the range of primary plus secondary stress difference exceeds $3S_m$.

It is recognized that it may be possible to increase these allowable numbers of cycles somewhat during the detailed design. The following modifications would be expected to improve the fatigue life.

1. Decrease the heat-up/cooldown rate from its present 100° F/hr.
2. Optimize the shape of the head in the areas of high stress.
3. Provide a method for protecting the inner surface of the head from the relatively cold (120°F) ECCS injection water.
4. Use a bolt design with a thread stress concentration factor less than 4.0.

B. Adequacy of Metallic O-Ring Seals

The analyses presented in Appendix G of this report indicate that metallic O-rings should provide acceptable service for the head and

vessel design considered. Several conclusions were reached as a result of the calculations and discussions with personnel at the main supplier of metallic O-rings (United Aircraft Products, Inc.).

These were:

1. If possible, the seals for this vessel should be 1/2" metallic O-rings.
2. Inconel 718 seals should provide adequate service without the need to taper the flange surfaces to minimize flange separation.
3. Due to relaxation under load, stainless steel O-rings will probably require tapered flange seal surfaces in order to prevent leakage from the inner seal. This taper would have to be designed such that the seal surfaces of the vessel head and shell flange are parallel at the hot steady-state operating condition.

On the basis of this information, it is recommended that 1/2" OD Inconel 718 O-rings be used.

Table II-1

Summary of Stress and Fatigue Analyses for Vessel Closure

Component and Location ⁽²⁾	Material	Primary Membrane Stress Intensity (26,700 psi allowable)	Primary Membrane Plus Bending Stress Intensity (40,500 psi allowable)	Maximum Range of Primary plus Secondary Stress Intensity (80,100 psi allowable)	Cumulative Fatigue Usage Factor ⁽¹⁾ (1.0 allowable)
Head at A	SA-508	1534	38,440	103,230 ⁽⁴⁾	0.51
Head at B	SA-508		33,590	132,090 ⁽⁴⁾	2.83 ⁽³⁾
Vessel at D-D	SA-336 (Code Case 1332-1)	14,489	14,489	70,500	< 0.104
Bolts	SA-540, Gr. 23	44,423 (64,200 psi allowable)	75,695 (96,300 psi allowable)		3.31 ⁽³⁾

(1) Cyclic conditions considered are listed on page II-1.

(2) Locations shown on Figure II-1.

(3) Head and bolting fatigue life limited to cyclic conditions discussed on page II-2.

(4) Since the maximum range of primary plus secondary stress intensity (including effects of radial thermal gradients) exceeds $3S_m$ allowable, Code Case 1441 was used for the fatigue analysis. This Code Case covers the condition where the range of primary plus secondary stress intensity exceeds $3S_m$.

GUIDE TO CROSS SECTIONS FOR RESULTS OF STRESS ANALYSIS

FIGURE II-1

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III. DESCRIPTION OF CALCULATIONAL METHODS

The analysis of the flat outer head and closure was similar to that of the hemispherical head analyzed in MPR-239. In general, this method consists of determining the transient temperature distributions and resultant free body motions of the head, vessel and bolting for heat-up, cooldown and blowdown conditions; calculating redundant loads between these parts required to maintain compatibility of the structure; and determining the stresses which result from the combination of thermal conditions, internal pressure and redundant loads. Conditions analyzed include steady-state operating pressure and bolt preload in addition to the thermal transients discussed above. The specific differences in the analysis methods utilized herein in the analysis of the flat head and the analyses presented in MPR-239, are described below:

1. The transient thermal response of the outer head for the heat-up, cooldown and blowdown-ECCS transients was obtained using a multi-dimensional thermal analysis computer program instead of the approximate method used for the hemispherical head. This was required as no thermal response data were available on similar flat head designs.

2. Influence coefficients were obtained using the SEAL SHELL-2 computer program rather than the finite element program, FEPSAP-1. This method was used to obtain the gross motions of the head for the different unit loads and thermal distributions.
3. The redundant loads were obtained using the LOADS program described in MPR-239, and the influence coefficients previously determined for the reactor vessel shell and flange along with the new head and bolt influence coefficients.
4. Stresses in the flat head were calculated for each of the expected operating conditions using the redundant loads calculated in step 3 above in combination with the appropriate internal pressure and thermal conditions. Stresses in the bolts were calculated manually using these same redundant loads. Stresses at the critical section in the vessel shell for the two worst case loading conditions, i. e., heat-up and blowdown-ECCS, were determined by superposition using the stresses for unit loads calculated in MPR-239, and the redundant loads for step 3 above.

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V. APPENDICES

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DESCRIPTION OF COMPUTER PROGRAMS

Computer programs that were utilized for the flat head analyses which were not used for the previous vessel analyses are described below:

1. Description of Computer Programs

a. SEAL SHELL-2

SEAL SHELL-2 is a program which determines stresses, loads, and deflections in a thick or thin shell of revolution due to pressure, temperature, and distributed loads. The shell is linear-elastic with tensile, bending, and shear strains. The shell is divided into as many as 100 segments, each of which may have different material properties, geometry, loading (radial, axial, moment and pressure), and temperature distribution (radial and axial). An arbitrary set of 11 deformation shapes is applied to each segment. The resulting strain energy is determined with the restrictions that plane cross sections remain plane and the shear stress varies parabolically through the thickness. By application of the principle of virtual work to the strain energy of each segment, the forces on each segment are determined in terms of the deflections. The simultaneous equations are then solved to give the influence coefficients for deflections in terms

of forces. The middle surface slopes and deflections at all joints of adjacent segments are equated in order to calculate the forces connecting all the segments. Once the forces and deflections are known for each segment, the local stresses and strains are determined from the strain energy relationships.

The input consists of boundary conditions, geometry, material properties, loads, pressures, and temperatures. Linear interpolation is used between temperature values which can be input for each segment at the inner, outer, middle, and quarter thicknesses of the shell.

The output includes radial and axial centerline deflections, centerline rotations, mechanical and thermal stresses (principal, shear, and stress intensity at inner, outer, and mid-thicknesses), forces between elements, and flexibility influence coefficients for applied external loads.

b. TRANSIENT-2

TRANSIENT-2 is a digital computer program which can be used to determine transient or steady-state temperature distributions for complex multi-dimensional heat transfer problems.

TRANSIENT-2 solves the transient heat conduction equations for a multi-dimensional field using the "first forward difference" method. Since the method of solution is based on a nodal repre-

sentation of the geometry, almost any configuration can be handled through the proper specification of the nodal heat capacity and the evaluation of the "equivalent resistances" of nodal connectors.

In addition to conduction within the material medium, heat transfer from the surface may be by means of forced convection, free convection, radiation, nucleate boiling, or any combination of these. Heat transfer coefficients, boundary node temperatures, coolant flow rates, and internal heat generation rates may all be independently varied with time. Material conductivities and specific heats may be varied to correspond to the local temperature of the material. The input consists of individual nodal volumes, individual "equivalent resistances", physical properties, boundary conditions, internal heat generation rates, if any, and specification of the nodes which are to be temperature averaged.

The output consists of the complete nodal temperature distribution, the surface heat fluxes, and the temperature means and moments. These output quantities are printed at desired times in the solution of a problem.

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HEAD DESIGN ASSUMED FOR ANALYSIS

The vessel head design considered in this addendum consists of two concentric flat heads. The outer head contains 8 instrumentation ports and has a central opening sufficiently large to permit removal of the central fuel assembly without requiring the instrumentation leads to the outer fuel assemblies to be disconnected. A smaller head is used to close this central opening. As previously noted, this analysis covers only the outer head and the redundant loading effects on the vessel shell and flange, and the bolting.

The head thickness picked for the analysis was 16" based on preliminary sizing calculations performed using the SEAL SHELL-2 computer program. Preliminary sizing calculations for the inner head indicate that it will be approximately 7" thick.

Figure B-1 shows the dimensions of the vessel head analyzed in this report.

DIMENSIONS USED FOR LOFT OUTER HEAD ANALYSIS

FIGURE B-1

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THERMAL ANALYSES

Thermal analyses of the outer flat head were performed in order to determine the temperature distributions for the worst case times in the heat-up, cooldown and blowdown-ECCS transients. The thermal transient performance for the vessel shell and flange was determined in MPR-239 and since the boundary conditions have not changed, these analyses are still valid.

The analyses in this appendix are as follows:

1. Temperature Distribution for Heat-Up Transient
2. Temperature Distribution for Cooldown Transient
3. Temperature Distribution for Blowdown-ECCS Transient

1. Temperature Distribution for Heat-Up Transient

The thermal analysis for the heat-up transient was performed using the TRANSIENT-2 computer program. The computer model assumed for the analysis is shown in Figure C-1. The fluid temperature profile and heat transfer coefficients are the same as those given in Figure C-1 of MPR-239.

The temperature profile in the head at the worst point in the transient (end of heat-up) is shown in Figure C-2. The complete computer data is given in Addendum 1, Volume II, Section 1.

2. Temperature Distribution for Cooldown Transient

The same method as described in part 2, Appendix C of MPR-239, was used to obtain cooldown temperature distributions.

3. Temperature Distribution for Blowdown-ECCS Transient

The same method was used to obtain thermal distributions for the blowdown-ECCS transient that was used for the plant heat-up analysis. The boundary conditions were the same as those in Figure C-13 of MPR-239. The detailed computer output data for these calculations can be found in Addendum 1, Volume II, Section 2.

In order to select the worst points in the transient to use for the stress analysis, two graphs were prepared. The first, Figure C-3, shows the axial and radial thermal gradients through the head as a function of time; the second, Figure C-4 shows the thermal mis-

OUTER HEAD

INNER HEAD

NOTES:

1. NUMBERS REPRESENT THE ELEMENT AND SURFACE NODE POINTS USED FOR THE OUTER HEAD THERMAL ANALYSIS.
2. CLADDING THICKNESS IS ACCOUNTED FOR IN THE THERMAL PROPERTIES OF THE SURFACE AND ADJACENT NODES.
3. COMPLETE DATA IS PRESENTED IN ADDENDUM I VOL. II SECTION I

CENTERLINE OF VESSEL

TRANSIENT- 2 MODEL FOR OUTER HEAD THERMAL ANALYSIS
FIGURE C-1

ISOTHERMS IN OUTER HEAD AT END OF PLANT
HEAT UP TRANSIENT
FIGURE C-2

AVERAGE LINEAR THERMAL GRADIENT FOR BLOWDOWN TRANSIENT

FIGURE C-3

DIFFERENCE BETWEEN AVERAGE HEAD AND VESSEL FLANGE TEMPERATURES

FIGURE C-4

match between the average head and average vessel flange temperatures.

Two points in the transient were selected for detailed analysis, the first, 15 minutes into the transient, is the point at which the axial and radial thermal gradients in the head are a maximum; the second, 100 minutes into the transient, is the point at which the greatest mismatch between average head and vessel flange temperatures occur. It is expected that the time of 15 minutes will produce the highest stresses in the head, while the time of 100 minutes will produce the highest stresses in the bolting and vessel flange.

Figures C-5 and C-6 show the thermal distributions at the two times noted above.

ISOTHERMS IN OUTER HEAD 15 MINUTES INTO BLOWDOWN -
ECCS TRANSIENT
FIGURE C-5

ISOTHERMS IN OUTER HEAD 100 MINUTES INTO BLOWDOWN -
ECCS TRANSIENT

FIGURE C-6

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LOADING ANALYSES

The redundant loads between the various vessel closure elements were obtained using the same type of compatibility analyses as performed in MPR-239. As in the previous report, several loading analyses were performed to investigate the effects of the coefficient of friction at the gasket surface. The preload was selected as 12×10^6 lb and the pressure as 2500 psi as per the previous analysis.

Based on the parametric studies of the gasket friction, a coefficient of friction of zero was selected for the stress analysis. This coefficient, which indicates that the flanges are free to slip, produces the highest mechanical loads on the flanges and bolts.

This appendix is divided into the following main sections:

1. Calculation of Influence Coefficients
2. Solution for Redundant Loads
3. Loads Used for Stress and Fatigue Analyses

1. Calculation of Influence Coefficients

The values of the head influence coefficients were determined using the Seal Shell-2 computer program, those for the vessel flange were obtained from MPR-239, and those for the bolts were calculated by hand. The Seal Shell-2 model used to calculate influence coefficients and stresses is shown on Figure D-1.

Calculation of influence coefficients using Seal Shell-2 was not as straightforward as for the finite element analysis method as deflections for the unit loads are not given at surface points. Accordingly, the motions of the vessel and bolt interface areas are obtained from the centerline displacements and rotations with manual corrections included for axial thermal growth. The following examples show the method used for the calculations.

◦ Influence Coefficients J1, J2, J3 & J4

From the Seal Shell-2 output for a unit gasket loading (see MPR-239, Addendum I, Volume II, Section 3), the following centerline motions are known:

Radius (in) =	<u>31.0</u>	<u>32.0</u>	<u>36.0</u>	<u>37.0</u>
rotation ($\times 10^{-3}$ rad)	-0.05763	-0.05879	-0.06087	-0.06052
radial deflection ($\times 10^{-3}$ in)	0.000	0.000	0.000	0.000
axial deflection ($\times 10^{-3}$ in)	0.8511	0.9184	1.269	1.350

SEAL SHELL-2 MODEL USED FOR OUTER HEAD ANALYSIS

FIGURE D-1

Points 1 and 2 on the head are located at Radii 31.7 and 36.4, respectively. Interpolating the Seal Shell-2 output yields:

Radius (in) =	<u>31.7</u>	<u>36.4</u>
rotation ($\times 10^{-3}$ rad)	-0.05833	-0.06073
radial deflection ($\times 10^{-3}$ in)	0.000	0.000
axial deflection ($\times 10^{-3}$ in)	0.8982	1.3012

To obtain the surface deflections from the centerline deflections listed above, the deflections caused by segment rotation must be added to the centerline deflections.

From the figure above, the changes in the radial and axial deflections are equal to

$$\Delta R = l \sin \phi \text{ and } \Delta Z = x \sin \frac{\phi}{2} = 2 l \sin \frac{\phi}{2} \sin \frac{\phi}{2}$$

However, as $\frac{\phi}{2} \rightarrow 0$, $\sin \frac{\phi}{2} \rightarrow \frac{\phi}{2}$. Therefore, $\Delta R = l \phi$ and

$\Delta Z = \ell \frac{\phi^2}{2}$. Noting that Seal Shell-2 deflections are positive outward (for radial) and downward (for axial) these deflections have the following signs:

$$\begin{array}{lll} \text{On bottom surface -} & \Delta R = +\ell\phi & \Delta Z = -\ell\frac{\phi^2}{2} \\ \text{On top surface -} & \Delta R = -\ell\phi & \Delta Z = +\ell\frac{\phi^2}{2} \end{array}$$

Adding these values to the centerline motions gives the surface deflections due to mechanical loads. Using these relations, the deflections at Point 1 (at radius 36.4 on the upper surface) and Point 2 (at radius 31.7 on the lower surface) are found to be:

	<u>Point 1</u>	<u>Point 2</u>
rotation ($\times 10^{-3}$ rad)	-0.06073	-0.05833
radial deflection ($\times 10^{-3}$ in)	0.4858	-0.4667
axial deflection ($\times 10^{-3}$ in)	1.301	0.8982

From Table D-1 of MPR-239, influence coefficient J1 is the deflection of point 1 on the head in the minus Z direction relative to point 2 for a unit axial gasket load of 1×10^6 lb. Therefore, $J1 = -[(1.301) - (0.8982)] = -0.403$.

From Table D-1 of MPR-239, influence coefficient J2 is the deflection of point 1 on the head in the R direction for a unit axial gasket load of 1×10^6 lb. Therefore,

$$J2 = 0.4858$$

From Table D-1 of MPR-239, influence coefficient J3 is defined to be zero and influence coefficient J4 is the deflection of point 2 on the head in the R direction for a unit axial gasket load of 1×10^6 lb. Therefore,

$$J3 = 0.0$$

$$J4 = -0.4667$$

Influence coefficients for the other five unit loadings were calculated in a similar manner and are summarized in Table D-1.

◦ Thermal Motions N1, N2, N3 and N4

For cases involving temperature changes, Seal Shell-2 does not give the axial thermal expansion. Therefore, in order to obtain the surface deflections due to both radial and axial temperature distributions, the axial thermal expansion ($\ell \alpha \Delta T$) must be included in the calculation of axial deflections. The final relations for surface deflections are:

Table D-1
Influence Coefficients and Free Thermal Motions

	Influence Coefficients				Free Thermal Motions		
	Pressure I 2500 psi	Axial Gasket Load J 1×10^6 lb	Bolt Shear Force K 1×10^5 lb/rad	Gasket Shear Force M 1×10^5 lb/rad	Heat-Up Transient N	Blowdown Transient	
						15 Min. N'	100 Min. N''
$1(10^{-3} \text{ in})$	-13.140	-.4031	+.3504	-.3499	+47.33	37.57	25.649
$2(10^{-3} \text{ in})$	+20.753	+.4858	-.8247	+.3898	+68.44	154.78	79.363
$3(10^{-3} \text{ in})$	0.0	0.0	0.0	0.0	0.0	0.0	0.0
$4(10^{-3} \text{ in})$	-19.686	-.4667	+.3820	-.8023	+94.18	78.98	34.853
$5(10^{-3} \text{ in})$	+4.738*	+.9114*	-1.611*	-.7069*	-3.40*	+8.65*	0.0
$6(10^{-3} \text{ in})$	+1.305*	-.9556*	+3.891*	+1.957*	+121.3*	+89.81*	25.988
$7(10^{-3} \text{ in})$	0.0*	0.0*	0.0*	0.0*	0.0*	0.0*	0.0
$8(10^{-3} \text{ in})$	+1.206*	-1.145*	+3.760*	+2.340*	+107.9*	+73.21*	22.632
$9(10^{-3} \text{ in})$	-.674*	-.1125*	+.7159*	+.1249*	+.0286*	-1.188*	0.0

Axial Free Thermal Motions of the Bolts
Heat-Up Transient 29.9×10^{-3} in.
Blowdown Transient (15 min.) = 49.1×10^{-3} in.
Blowdown Transient (100 min.) = 27.7×10^{-3} in.

Bolt Influence Coefficients:

$$\phi_1 = 0.013549 \frac{\text{in.}}{2500 \text{ psi}}$$

$$\phi_2 = 1.6848 \times 10^{-3} \frac{\text{in.}}{1 \times 10^6 \text{ lb.}}$$

$$\phi_3 = 0.128025 \frac{\text{in.}}{1 \times 10^5 \text{ lb/rad}}$$

*From Table D-14 of MPR-239, Vol. I

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	<u>Upper Surface</u>	<u>Lower Surface</u>
radial deflection =	$R_o - l\phi$	$R_o + l\phi$
axial deflection =	$Z_o + l\frac{\phi^2}{2}$ $- l\alpha\Delta T$	$Z_o - l\frac{\phi^2}{2}$ $+ l\alpha\Delta T$

Where:

- R_o = centerline radial deflection
- Z_o = centerline axial deflection
- ϕ = segment rotation
- l = distance from centerline to surface
- ΔT = $T_f - T_i$
- T_f = average temperature between centerline and surface
- T_i = 70°F

◦ Thermal Motions N''5, N''6, N''7, N''8, and N''9 (Special Cases)

In order to obtain shell thermal motions at 100 min. into the blowdown transient, it was assumed that the shell moved radially with no rotation. (At 100 min. into the blowdown transient, the vessel thermal gradients causing rotation are relatively small.) In the absence of rotation, N''5 (the deflection of point 3 in the -Z direction with respect to point 4) and N''9 (the rotation of point 3 on the shell) are zero, and N''6 and N''8 (the radial deflections of points 3 and 4, respectively) are equal to $R_o\Delta T$.

Therefore,

$$N''5 = 0.0$$

$$N''6 = 25.99 \times 10^{-3}$$

$$N''7 = 0.0 \quad \text{by definition}$$

$$N''8 = 22.63 \times 10^{-3}$$

$$N''9 = 0.0$$

◦ Bolts

Influence coefficients for the bolts are calculated by hand using classical methods.

$$\begin{aligned} \delta_{5(-Z)} &= \frac{F_z L}{AE (38)} \\ &= \frac{16.28 \text{ in} (4) \text{ in}^2}{\pi (3.4188 \text{ in})^2 (27.7 \times 10^6 \text{ lb}) (38)} F_z \\ &= .1685 \times 10^{-8} F_z \text{ in.} \end{aligned}$$

The axial load on the bolts F_B for the closure is composed of two parts: one, which is a result of the gasket load, and the other, which is required to balance the pressure. This results in an influence coefficient for both the gasket load, ϕ_2 , and for the internal pressure, ϕ_1 .

$$\phi_1 = .1685 \times 10^{-8} (8.042 \times 10^6 \text{ lb}) = .01355 \text{ in/2500 psi}$$

$$\phi_2 = .1685 \times 10^{-8} (1 \times 10^6 \text{ lb}) = .001655 \text{ in/1} \times 10^6 \text{ lb}$$

$$\begin{aligned} \delta_R &= \frac{F_r L^3}{3 E I} \times \frac{2\pi}{38} \\ &= \frac{(16.28 \text{ in})^3 (2) (\pi) \text{ in}^2 64}{3 (27.7 \times 10^6 \text{ lb}) \pi (3.4188 \text{ in})^4 (38)} F_r \\ &= 12.80 \times 10^{-7} F_r \text{ in} \end{aligned}$$

Using the definitions of the influence coefficient from Table D-1.

$$\phi_3 = 12.80 \times 10^{-7} \text{ in } (1 \times 10^5 \text{ lb/rad}) = \frac{.1280 \text{ in}}{1 \times 10^5 \text{ lb/radian}}$$

2. Solution for Redundant Loads

The computer program LOADS was used to solve for the redundant loads on the vessel closure components using the influence coefficients and free thermal motions given in Table D-1 of this appendix. Several analyses were performed using LOADS to investigate the

Table D-2
Summary of Loads Used for Stress Analysis

Condition*	REDUNDANT LOADS					
	Bolt Axial Load F_B (lb)	Bolt Shear Load F_2 (lb/rad)	Bolt Bending Moment M (lb-in/bolt)	Gasket Axial Load F_G (lb)	Gasket Shear Force F_3 (lb/rad)	Relative Motion Between Flanges
Preload (Unpressurized)	12.0×10^6	21.4×10^3	58.3×10^3	12.0×10^6	0	Yes
End of Heat-Up Transient (Pressurized)	15.5×10^6	$.806 \times 10^3$	2.20×10^3	7.35×10^6	0	Yes
Hot Steady-State Operation (Pressurized)	9.42×10^6	24.0×10^3	65.4×10^3	1.32×10^6	0	Yes
End of Cooldown Transient (Unpressurized)	5.78×10^6	50.0×10^3	132×10^3	5.78×10^6	0	Yes
15 Minutes into Blowdown Transient (Unpressurized)	3.26×10^6	73.6×10^3	194×10^3	3.26×10^6	0	Yes
100 Minutes into Blowdown Transient	1.14×10^7	62.7×10^3	165×10^3	1.14×10^7	0	Yes

*Boundary Conditions

Preload = 12×10^6 lb

Pressure = 2500 psi

Friction Coefficient = 0.0

Temperature Distributions

--Heat-up transient (see Figure C-2)

--Cooldown transient (see Figure C-2)

--Blowdown transient (see Figures C-5 and C-6)

effect of coefficient of friction on the redundant loads. The following cases were run:

<u>Case</u>	<u>Pressure (psi)</u>	<u>Preload (lb)</u>	<u>Gasket Friction</u>
1	2500	12×10^6	0
2	2500	12×10^6	1.5
3	2500	12×10^6	3.0

These calculations cover the entire range of gasket conditions, from "free to slip" to "no slip." The output for LOADS (See MPR-239, Addendum I, Volume II, Section 3) showed that the worst case conditions for the head, bolts and vessel flange exist when the head is free to slip relative to the vessel. Accordingly, for analysis purposes, a coefficient of friction of 0 was selected.

3. Loads Used for Stress and Fatigue Analysis

The loads used for the stress and fatigue analyses are given in Table D-2. These loads are based on the following boundary conditions:

Internal Pressure	2500 psi (vessel design pressure)
Bolt Preload	12×10^6 lb
Gasket Coefficient of Friction	0.0

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STRESS ANALYSES

Stress analyses were performed of the outer vessel head using the SEAL SHELL-2 computer program, the redundant loads given in Table D-2, and the temperature distributions calculated in Appendix C. Primary plus secondary and peak stresses at the worst point on the vessel shell, cross section D-D on Figure II-1, were obtained by superposition using the redundant loads in Table D-2 and the stresses for unit loadings calculated in the computer analysis in MPR-239. Bolt stress calculations were performed by hand using the loads in Table D-2.

The stresses calculated for the vessel outer head and shell, and the bolts are compared to the allowable stresses given in Section III of the ASME Boiler and Pressure Vessel Code.

Fatigue analyses, using the peak stresses calculated for the outer vessel head, vessel shell, and the bolts, are presented in Appendix F.

This appendix is divided into the following main sections:

1. Allowable Stresses
2. Calculation of Stresses in Vessel Head and Shell
3. Calculation of Stresses in Bolts
4. Surface Stress Distributions

GUIDE TO CROSS SECTIONS FOR RESULTS OF STRESS ANALYSIS

FIGURE E-1

1. Allowable Stresses

The allowable stresses given by Section III of the ASME Boiler and Pressure Vessel Code are as follows:

Vessel Head and Shell

Material - Head - SA-508 C1 2 or 3
Shell - SA-336 Code Case 1332-1

Temperature - 650° F (Design)

Allowable Stresses -

<u>Type of Stress</u>	<u>Allowable</u>
Primary Membrane S_m	26,700 psi
Primary Membrane plus Primary Bending $1.5S_m$	40,050 psi
Primary plus Secondary $3S_m$	80,100 psi
Peak Stress	(Peak stresses are used only for the fatigue analysis - See Appendix F)

Bolts

Material - SA-540 Gr. 23

Temperature - 650° F (Design)

Allowable Stresses -

<u>Type of Stress</u>	<u>Allowable</u>
Maximum service stress averaged across bolt cross section $2S_m$	64,200 psi

<u>Type of Stress (cont'd)</u>	<u>Allowable (cont'd)</u>
Maximum service stress at periphery of bolt cross section $3S_m$	93,300 psi
Peak stress	(Peak stresses are used only for the fatigue analysis- See Appendix F)

2. Calculation of Stresses in Vessel Head and Shell

Stresses in the vessel head and shell were calculated in the following manner:

a. Primary Membrane Stresses

Seal Shell-2 does not give primary membrane stresses directly as it calculates the stresses at the surfaces and at the center of each element. Primary membrane stresses can be approximated by determining the average stress which exists at a cross section from the computer data given for each element, or by hand calculations using classical methods. The latter method is used for the analysis of the outer head as it is relatively straightforward. Calculations of primary membrane stresses for the vessel shell are included in report MPR-239.

The primary membrane stresses in the head and vessel shell are as follows:

Head

$$R_i = 12.75 \text{ in}, R_o = 40.0 \text{ in}$$

$$\sigma_t = \frac{p R_i^2}{R_o^2 - R_i^2}$$

$$\sigma_t = \frac{2500 (12.75)^2}{40.0^2 - 12.75^2}$$

$$= 284 \text{ psi}$$

$$\sigma_r = \frac{-p}{2}$$

$$\sigma_r = \frac{-2500}{2}$$

$$= -1250 \text{ psi}$$

Vessel

$$\sigma_t = 13,239 \text{ psi}$$

$$\sigma_l = 6048 \text{ psi}$$

$$\sigma_r = -1250 \text{ psi}$$

from report MPR-239

The primary membrane stress intensities for both components are less than the Code allowable of 26,700 psi.

b. Primary Membrane Plus Primary Bending Stresses

The primary membrane plus primary bending stresses for the outer head can be obtained directly from the Seal Shell-2 computer output data by adding the stresses due to internal pressure and mechanical loads. These stresses are presented in Table E-1 for each of the operating conditions considered. This table indicates that the maximum primary membrane plus primary bending stress intensity is 38,440 psi, and occurs at point A on the vessel head. This stress is less than the allowable of 40,050 psi. There are no primary bending stresses in the vessel at cross section D-D.

c. Primary Plus Secondary Stresses

(1) Stresses in Head

Seal Shell-2 gives the primary plus secondary stresses directly if all thermal gradients are to be considered. In this analysis it would be difficult to separate the effects of the radial and axial thermal gradients. Therefore, the primary plus secondary stresses reported include the effects of both gradients. Table E-2 gives the raw stress data and Table E-3 gives the stress differences which were used to find the maximum range of primary plus secondary stress difference. The two points A and B on the head were selected by inspection as producing the worst case stresses.

Table E-1
Calculation of Primary Membrane plus Primary Bending Stress Intensity

Location	Stress	Preload	End of Heat-Up Transient	Hot Steady State Operation	End of Cooldown Transient	15 Minutes into Blow-down-ECCS Transient	100 Minutes into Blow-down-ECCS Transient
Head at A	σ_l	40	-2400	-2420	10	0	20
	σ_t	10,890	35,940	30,540	2550	-330	8070
	σ_r	0	-2500	-2500	0	0	0
	S	10,890	38,440*	33,040	2550	-330	8070
Head at B	σ_l	-40	-2600	-2580	-10	0	-20
	σ_t	-12,830	30,990	25,600	-2910	-220	-8530
	σ_r	0	-2500	-2500	0	0	0
	S	12,830	33,590	28,180	2910	220	8530

*Maximum primary membrane plus bending stress intensity

Table E-2
Primary Plus Secondary Stresses

Location	Stress	Preload	End of Heat-Up Transient	Hot Steady State Operation	End of Cooldown Transient	15 Minutes into Blow-down-ECCS Transient	100 Minutes into Blow-down-ECCS Transient
Head at A	σ_l	40	-4350	-2410	1690	3940	1130
	σ_t	10,890	-18,060	30,540	50,950	87,670	62,820
	σ_r	0	-2500	-2500	0	0	0
Head at B	σ_l	-40	-4450	-2580	1580	6480	1530
	σ_t	-12,830	-69,880	-25,600	28,720	64,710	25,290
	σ_r	0	-2500	-2500	0	0	0

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Table E-3
Primary Plus Secondary Stress Differences

Location	Stress Difference (psi)	Preload	End of Heat-up Transient	Hot Steady State Operation	End of Cooldown Transient	15 Minutes into Blow-down-ECCS Transient	100 Minutes into Blow-down Transient	Maximum Range of Primary plus Secondary Stress Difference (psi)
Head at A	S_{r-l}	-40	+1850	-90	-1690	-3940	-1130	103,230
	S_{l-t}	-10,850	13,710	-32,950	-49,260	-83,730	-61,690	
	S_{t-r}	10,890	-15,560 *	33,040	50,950	87,670*	62,820	
Head at B	S_{r-l}	40	1950	80	-1580	-6480	-1530	132,090
	S_{l-t}	12,790	65,430	23,020	-27,140	-58,230	-23,760	
	S_{t-r}	-12,830	-67,380*	-23,100	28,720	64,710*	25,290	

*Stress components combined to obtain the maximum range of stress difference.

The maximum range of primary plus secondary stress difference, 132,090 psi, exceeds the ASME Code allowable of 80,100 psi. Accordingly, the fatigue analysis in Appendix F has been performed using Code Case 1441, which is applicable to cases where the range of primary plus secondary stress difference exceeds $3S_m$.

(2) Stresses in Shell

The primary plus secondary stresses in the vessel shell were evaluated at the highest stressed point for the worst combination of loading conditions based on the hemispherical head analysis presented in report MPR-239. The point investigated was the outer surface of the vessel at cross section D-D, and the loading conditions were the heat-up and blowdown transients.

The analysis method used was to superimpose the stresses due to thermal gradients, internal pressure, gasket load, bolt shear and gasket shear. The thermal and pressure stresses were taken directly from the unit loading calculation in MPR-239, Volume II, Part I, Sections 10, 14 and 15. The stresses for the gasket load, bolt shear, and gasket shear were scaled from the unit load stresses given in Sections 11, 12, and 13 of the above referenced volume using the redundant

loads presented in Table D-2 of this report. Calculations presented on page E-7 of MPR-239 indicate maximum range of primary plus secondary stress intensity is given by the equation:

$$S_{\text{max range}} = \sigma_{\text{heatup}} - \sigma_{\text{blowdown}}$$

Figure E-2 shows the peak and primary plus secondary components of these two stresses. The maximum range of primary plus secondary stress difference is therefore 70,500 psig which is less than the 80,100 psi allowable stress.

d. Peak Stresses

The SEAL SHELL-2 computer program used for the outer head analysis does not include the effects of all stress concentrations, however, the head geometry analyzed was not expected to include significant stress concentrations. As a check on this assumption, a finite element model was run for the case 15 minutes into the blowdown-ECCS transient. The results of this computer run confirmed that there are no significant stress concentrations in the head. (Note: Stress concentrations at bolt holes, etc., have been considered separately in Appendix F.)

The peak stresses for the vessel shell at cross section D-D are shown on Figure E-1.

PRIMARY PLUS SECONDARY AND PEAK STRESSES THROUGH VESSEL SHELL

FIGURE E-1

The computer data giving the complete results of the stress analysis can be found in Sections 11 to 16 of MPR-239, Addendum I, Volume II.

3. Calculation of Stresses in Bolts

Stresses in the bolts are determined using the redundant loads given in Table D-2. The following calculational technique is used:

- ° Maximum service stress averaged across bolt cross section, σ'_ℓ

$$F'_B = \text{Axial load for one bolt} = \frac{\text{total axial bolt load, } F_B}{\text{no. of bolts, 38}}$$

$$\sigma'_\ell = \frac{F'_B}{\text{Cross sectional area of bolt}}$$

$$= \frac{4 F'_B}{\pi (3.4188)^2 (38)}$$

$$= 2.866 \times 10^{-3} F'_B \text{ psi}$$

- ° Maximum service stress at periphery of bolt (excluding stress concentrations), σ_ℓ :

$$M_B = \text{bending moment for each bolt} = \text{radial load/bolt} \times \text{moment arm}$$

$$= \frac{F_2 (2\pi) (16.50 \text{ in})}{38}$$

$$\begin{aligned}\sigma_{\ell} &= \frac{M_B}{Z} + \sigma'_{\ell} = \frac{32 \cdot (2) (\pi) (16.50)}{\pi (3.4188)^3 (38)} F_2 + \sigma'_{\ell} \\ &= .6862 F_2 + 2.866 \times 10^{-3} F'_B\end{aligned}$$

The results of the calculations are presented in Table E-4.

4. Surface Stress Distributions

Figures E-2 to E-7 show the peak stresses in the longitudinal and circumferential direction on the surfaces of the outer head for each of the six operating conditions analyzed in this report.

TABLE E-14
SUMMARY OF BOLT STRESS CALCULATIONS

<u>Parameter</u>	<u>Preload</u>	<u>End of Heat-Up Transient</u>	<u>Hot Steady State Operation</u>	<u>End of Cool- Down Transient</u>	<u>15 Minutes into Blowdown Transient</u>	<u>100 Minutes into Blowdown Transient</u>
1. Axial bolt load F_B (lb)	12×10^6	15.5×10^6	9.42×10^6	5.78×10^6	3.26×10^6	11.4×10^6
2. Average axial stress σ'_l (psi)	34,392.	44,423.	26,998.	16,565.	9,343.	32,670.
3. Bolt shear force F_2 (lb/rad)	21,400.	806.	24,000.	50,000.	73,600.	62,700.
4. Bolt bending stress σ_b (psi)	$\pm 14,685.$	$\pm 553.$	$\pm 16,469.$	$\pm 34,310.$	$\pm 50,504.$	$\pm 43,025.$
5. Maximum stress at periphery $\sigma_{l \max}$ (psi)*	49,077.	44,976.	43,467.	50,875.	59,847.	75,695.
6. Minimum stress at periphery $\sigma_{l \min}$ (psi)*	19,707.	43,870.	10,529.	-17,745.	-41,161.	-10,355.

*Does not include a stress concentration factor.

SURFACE STRESSES ON OUTER HEAD FOR PRELOAD CONDITION

FIGURE E-2

SURFACE STRESSES ON OUTER HEAD FOR END OF HEAT UP TRANSIENT

FIGURE E-3

SURFACE STRESSES ON OUTER HEAD FOR HOT STEADY STATE OPERATION

FIGURE E-4

LEGEND:

- CIRCUMFERENTIAL STRESS
- - - MERIDIONAL STRESS

SURFACE STRESSES ON OUTER HEAD FOR END OF COOL DOWN TRANSIENT

FIGURE E-5

SURFACE STRESSES ON OUTER HEAD 15 MINUTES INTO BLOWDOWN-ECCS TRANSIENT

FIGURE E-6

SURFACE STRESSES ON OUTER HEAD 100 MINUTES INTO BLOWDOWN-ECCS TRANSIENT

FIGURE E-8

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FATIGUE ANALYSES

Fatigue analyses of the vessel head, shell and bolting were performed using Figures N-415(A) and N-416 in the 1968 edition of Section III of the ASME Code. The cyclic conditions listed in the system functional requirements are as follows:

- Closure opening and closure operations - 360 cycles
- Heat-up/cooldown - 1500 cycles
- Blowdown and ECCS operations - 360 cycles

For purposes of this analysis, it is assumed that the blowdown and ECCS operations include a heat-up cycle from cold preloaded condition. The fatigue calculations follow.

1. Fatigue Analysis for Outer Head at Points A and B

Since the range of primary plus secondary stress intensity calculated in Appendix F exceeded the $3S_m$ allowable, the fatigue analysis must be in accordance with Code Case 1441 which applies to such conditions. The fatigue life calculations using this Code Case are as follows:

Point A

- 360 Preload Cycles

Range of stress difference $0 \rightarrow 10,890 \rightarrow 0$

$$= 10,890 \text{ psi} < 3S_m = 80,100 \text{ psi}$$

$$S_{alt} = 5,445 \text{ psi}$$

$$\text{Allowable number of cycles} = > 10^6$$

$$\text{Usage factor} = 0$$

◦ 1500 Heat-up/Cooldown Cycles

$$\text{Range of stress difference } 10,890 \rightarrow -15,560 \rightarrow 33,040$$

$$\rightarrow 50,950 \rightarrow 10,890$$

$$= 66,510 \text{ psi} < 3S_m = 80,100 \text{ psi}$$

$$S_{alt} = 33,255 \text{ psi}$$

$$\text{Allowable number of cycles} = 10,200$$

$$\text{Usage factor} = .147$$

◦ 360 Blowdown-ECCS Cycles

$$\text{Range of stress difference } 10,890 \rightarrow -15,560 \rightarrow 33,040$$

$$\rightarrow 87,670 \rightarrow 10,890$$

$$= 103,230 \text{ psi} > 3S_m = 80,100 \text{ psi}$$

Following the method in Code Case 1441

$$K_e = 1.0 + \frac{(1-.2)}{.2(3-1)} \left[\frac{103,230}{80,100} - 1 \right] = 1.58$$

$$S_{alt} = 51,615 \text{ psi} \times 1.58 = 81,750 \text{ psi}$$

$$\text{Allowable number of cycles} = 1000$$

$$\text{Usage factor} = .36$$

- Cumulative Fatigue Usage Factor = .51

Point B:

- 360 Preload Cycles

Range of stress difference $0 \rightarrow -12,830 \rightarrow 0$

$$= 12,830 \text{ psi} < 3S_m = 80,100$$

$$S_{alt} = 6415 \text{ psi}$$

$$\text{Allowable number of cycles} = > 10^6$$

$$\text{Usage factor} = 0$$

- 1500 Heat-up/cooldown cycles $-12,830 \rightarrow -67,380 \rightarrow -23,100$

$$\rightarrow 28,720 \rightarrow 12,830$$

$$= 96,100 \text{ psi} > 3S_m = 80,100$$

Following the method in Code Case 1441

$$K_e = 1.0 + \frac{(1-.2)}{.2(3-1)} \left[\frac{96,100}{80,100} - 1 \right] = 1.4$$

$$S_{alt} = 48,050 \text{ psi} \times 1.4 = 67,500 \text{ psi}$$

$$\text{Allowable number of cycles} = 1800$$

$$\text{Fatigue usage factor} = .835$$

- 360 Blowdown-ECCS Cycles $-12,830 \rightarrow -67,380 \rightarrow -23,100$

$$\rightarrow 64,710 \rightarrow -12,830$$

$$= 132,090 > 3S_m = 80,100$$

Following the method in Code Case 1441

$$K_e = 1.0 + \frac{1 - .2}{.2 (3 - 1)} \left[\frac{132,090}{80,100} - 1 \right]$$

$$S_{alt} = 66,045 \text{ psi} \times 2.3 = 152,000 \text{ psi}$$

Allowable number of cycles = 180

Usage factor = 2.0

◦ Cumulative Fatigue Usage Factor = 2.835

2. Fatigue Analysis for Outer Head at Closure Plate Bolt Holes

The fatigue life will be checked at the closure plate bolt holes in the outer head as there is a relatively high stress concentration at this point. The stress conditions at the worst point of the bolt hole are as shown below. Peterson ⁽¹⁰⁾ gives a stress concentration

factor of 2.88 for a 2" circular hole in a 16" thick plate loaded in bending. The usage factor calculations are as follows:

◦ 360 Preload Cycles

Range of stress difference 460* → 9130 → 460

$$= 8670 \text{ psi} < 3S_m = 80,100 \text{ psi}$$

* 0 - 460 psi included in Blowdown - ECCS cycles as Section III requires that superposition be used to obtain the maximum stress range for all possible cycles.

$$S_{alt} = 4334 \text{ psi} \times 2.88 = 12,500 \text{ psi}$$

$$\text{Allowable number of cycles} = 700,000$$

$$\text{Usage factor} = \sim 0.0$$

◦ 1500 Heat-up-Cooldown Cycles

$$\text{Range of stress difference } 9130 \longrightarrow 460 \longrightarrow 25,160$$

$$\longrightarrow 30,360 \longrightarrow 9130$$

$$= 29,900 \text{ psi} < 3S_m = 80,100 \text{ psi}$$

$$S_{alt} = 14,950 \text{ psi} \times 2.88 = 43,000 \text{ psi}$$

$$\text{Allowable number of cycles} = 6,500$$

$$\text{Usage factor} = .231$$

◦ 360 Blowdown-ECCS Transients

$$\text{Range of stress difference } 9130 \longrightarrow 0 \longrightarrow 25,160$$

$$\longrightarrow 40,950 \longrightarrow 9130$$

$$= 40,950 < 3S_m = 80,100$$

$$S_{alt} = 20,475 \text{ psi} \times 2.88 = 59,000 \text{ psi}$$

$$\text{Allowable number of cycles} = 250$$

$$\text{Usage factor} = .697$$

◦ Cumulative Usage Factor = .928

3. Fatigue Analyses of Vessel at D-D

As previously calculated in Appendix E, the primary plus secondary and peak stresses for the vessel at cross section D-D are less than those calculated in MPR-239 for the hemispherical head design.

Since the fatigue usage factor for the hemispherical head design was 0.104 at Section D-D, the fatigue usage factor for the flat head will be less than 0.104 and therefore acceptable.

4. Fatigue Analysis of Bolts

As required by Section III, the stress concentration factor for the bolts will be taken as 4.0 for these calculations. It may be possible, however, to justify a lower stress concentration factor for the actual thread design used for the closure. The following are the peak stresses for each of the operating conditions:

<u>Condition</u>	<u>Nominal Stress</u>	<u>SCF</u>	<u>Peak Stress</u>
Preload	19,710	4.0	88,900
Heat-up	43,900	4.0	175,500
Steady-State	10,530	4.0	42,200
Cooldown	-17,750	4.0	-71,000
Blowdown-ECCS	-41,160	4.0	-165,000

The stress cycles are:

Opening and closing 0 → 88,900 → 0

Heat-up/cooldown 88,900 → 175,500 → 42,200 → 71,000 → 88,900

Blowdown-ECCS 88,900 → 175,500 → 42,200 → -165,000 → 88,900

The following cycles, which include the superposition of stresses are used for the stress analysis.

360 cycles of $-165,000 \rightarrow 175,500 \rightarrow -165,000$

1500 cycles of $-71,000 \rightarrow 175,500 \rightarrow -71,000$

360 cycles of $0 \rightarrow 88,900 \rightarrow 0$

The alternating stresses, allowable cycles and usage factors are:

<u>S_{alt}</u>	<u>Allowable No. of Cycles*</u>	<u>Required No. of Cycles</u>	<u>U</u>
170,250	360	360	1.0
123,250	670	1500	2.24
44,900	5000	360	.072

*Allowable number of cycles is based on the maximum nominal stress = $2.7 S_m$ curve of Figure N-416 of Section III, as the maximum nominal stress of 43,900 psi is $< 2.7 S_m$ where $S_m = 32,100$ psi.

The cumulative usage factor is therefore 3.31.

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SEPARATION AT SEAL SURFACES

The purpose of this appendix is to determine if the separation of the head and vessel in the seal area is compatible with the use of metallic O-rings. The figure below shows the seals and the mating surfaces of the head and vessel.

Table G-1 gives the rotations of the head and vessel flange for each of the expected operating conditions as well as the calculated separation at the inner and outer seal locations. The rotations of the head were obtained from the Seal Shell-2 results for each of the operating conditions. The rotations of the vessel flange were obtained from the

Table A-3
Rotations and Separations of Gasket Surfaces

Condition	Head Rotation (10^{-3} radians)	Vessel Flange Rotation (10^{-3} radians)	Total Rotation (10^{-3} radians)	Separation at Inner Seal (10^{-3} in)	Separation at Outer Seal (10^{-3} in)
Preload	-0.813	2.29	3.10	6.98	2.71
End of Heat-up Transient	-0.016	3.05	3.07	6.91	2.69
Hot Steady-State Operation	-1.57	2.29	3.86	8.69	3.38
End of Cooldown Transient	-2.89	1.66	4.55	10.1	3.94
Worst Case Time in Blowdown-ECCS Transient	-3.06	3.77	6.83	15.4	5.98

FEPSAP-1 results presented in MPR-239, Volume I. The redundant loads used in the analysis in MPR-239, Volume I, are slightly higher than those calculated for the flat head analyzed in this addenda. Therefore, the rotations presented for the vessel are a little bit higher than would result from using the flat head redundant loads.

Table G-2 presents a summary of the springback characteristics of metallic O-rings. This data was obtained from United Aircraft Products, Inc.

The following conclusions were based on a comparison of the data in Tables G-1 and G-2.

1. The seals for LOFT should be 1/2" diameter, if possible, as they have the best springback characteristics, and accordingly, will accommodate the greatest separations at the seals.
2. One-half inch Inconel 718 O-rings appear capable of providing a good seal at the inner and outer seal areas without the necessity of tapering the flange surfaces.
3. One-half inch stainless steel O-rings will probably require that the head and vessel flanges be tapered such that the head and vessel surfaces are parallel at the hot steady-state operating condition if the inner seal is not to leak.

Table G-2

Summary of Metallic O-Ring Springback Characteristics

O-Ring Material	Diameter of O-Ring (in)	Springback after Cold Preload (in)	Springback after Hot Operation (in)
Stainless Steel	3/8"	.008	.003
	1/2"	.012	.005 - .007
Inconel 718	3/8"	NA	NA
	1/2"	.015 - .017	.015

NA = Not Available

Note: United Aircraft Products indicated that effective sealing is provided only through one-third to two-thirds of the listed total springback.

In summary, there does not appear to be a seal problem which would rule out use of a flat head design. However, depending on the O-ring size and material selected by the manufacturer, it may be necessary to taper the seal surfaces in order to provide a continuous seal at the inner O-ring for all of the anticipated transients where pressure acts inside the vessel. It is expected that the inner seal could be permitted to leak during the blowdown-ECCS transient.