3.9 MECHANICAL SYSTEMS AND COMPONENTS

The system and component design criteria and analyses in Section 3.9 are referenced to LGS operation at the originally licensed rated reactor power of 3293 MWt. The effects of increased power, pressure, and flow rates for power rerate conditions on the reactor vessel and internals, and main steam and recirculation piping were evaluated. The results demonstrate LGS compliance with appropriate design and licensing criteria at rerate conditions. These are documented in Ref. 3.9-23 and 3.9-24. Additional analyses were also performed which considered the use of GE13 and GE14 fuel; these analyses are documented in Reference 3.9-28 and Reference 3.9-34, respectively. The GE14 fuel is demonstrated to be bounded by the GE13 results. The reactor vessel components were evaluated for the effects of MUR power uprate in Reference 3.9-31. The results show that the reactor vessel components continue to comply with the structural requirements. The RPV internals were evaluated for loads associated with the MUR power uprate in Reference 3.9-32. The RPV internal components are demonstrated to be structurally qualified for operation in the MUR conditions. Reference 3.9-33 identifies new design basis values for Fuel Lift Margin and Control Rod GuideTube Lift Forces under MUR conditions. Subsequently, analyses were performed which considered the use of GNF2 fuel; these analyses aere documented in Reference 3.9-35. The GNF2 fuel is demonstrated to be bounded by the analyses in Reference 3.9-33.

3.9.1 SPECIAL TOPICS FOR MECHANICAL COMPONENTS

3.9.1.1 Design Transients

This section discusses the transients used in the design and fatigue analysis of the ASME Code Class 1 components, component supports, and reactor internals. The number of cycles or events for each transient based on available and projected plant operating data at the time of design is included. The design transients shown in this section are included in the design specifications for the components. Transients or combinations of transients are classified with respect to the component operating condition categories, identified as "normal," "upset," "emergency," "faulted," or "testing" in the ASME B&PV Code, if applicable. The first four operating conditions correspond to service levels A, B, C and D, respectively.

3.9.1.1.1 Control Rod Drive Transients

The normal and test service load cycles used for design purposes in the 40 year life of the CRD are as follows:

	Transient	Category	Cycles
a.	Reactor startup and shutdown	Normal/Upset[1]	120
b.	Vessel pressure tests	Normal/Upset	130
c.	Vessel overpressure	Normal/Upset	10
d.	Scram test plus startup scrams	Normal/Upset	300
e.	Operational scrams	Normal/Upset	300[2]
f.	Jog cycles	Normal/Upset	30,000[3]
g.	Shim/drive cycles	Normal/Upset	1,000[3]

(1) In Section 3.9.1.1, whenever a transient is categorized with two classifications, i.e., normal/upset, the most limiting of the two is considered in the design.
(2) 180 scram cycles constitute the design basis; however, 300 cycles are applied to the CRD for conservatism.
(3) 30,000 jog cycles and 1000 shim/drive cycles are applied because they impose mechanical loads on the CRD while contributing negligible thermal loads.

In addition to the above cycles, the following have been considered in the design of the CRD:

	Transient	Category	Cycles
h.	Scram with inoperative buffer	Normal/Upset	10
i.	Scram with stuck control blade	Normal/Upset	1
j.	OBE[3]	Upset	10
k.	SSE	Faulted	1

All ASME Class 1 components of the CRD have been analyzed according to ASME Section III. The capability of the CRD to withstand other emergency and faulted conditions is verified by test rather than analysis.

3.9.1.1.2 CRD Housing and Incore Housing Transients

Transients, classifications, and number of cycles considered in the design and fatigue analysis of the CRD housing and incore housing are as follows:

	Transient	Category	Cycles
a.	Normal startup and shutdown	Normal/Upset	120
b.	Vessel pressure tests	Normal/Upset	130
c.	Vessel overpressure tests	Normal/Upset	10
d.	Interruption of feedwater flow	Normal/Upset	80[1]
e.	Scram	Normal/Upset	200[2]
f.	OBE	Upset	10
g.	SSE	Faulted	1

(1) The interruption of feedwater flow imposes thermal loads on the CRD housing while contributing negligible mechanical loads.

(2) 180 scram cycles constitute the design basis; however, 200 cycles are applied to the CRD housing for conservatism.

(3) The frequency of occurrence of this transient indicates the emergency category. However, for conservatism, the OBE was analyzed as an upset condition. Ten peak stress cycles are postulated.

CRD HOUSING ONLY

h.	Stuck rod scram	Normal/Upset	1
i.	Scram with no buffer	Normal/Upset	1

3.9.1.1.3 Hydraulic Control Unit Transients

	Transient	Category	Cycles
a.	Reactor startup and shutdown	Normal/Upset	120
b.	Scram tests	Normal/Upset	300
c.	Operational scrams	Normal/Upset	300
d.	Jog cycles	Normal/Upset	30,000
e.	Scram with stuck scram discharge valve	Emergency	1
f.	OBE	Upset	10
g.	SSE	Faulted	1

3.9.1.1.4 Core Support and Reactor Internals Transients

Cycles considered in the reactor internals design and fatigue analysis are listed in Table 3.9-2.

3.9.1.1.5 Main Steam System Transients

Transients considered in the main steam piping stress analysis are as follows:

	Transient	Category	Cycles
a.	Startup	Normal	120[(1)]
b.	Loss of feedwater pumps, isolation valves closed	Upset	10
c.	Scram	Upset	180
d.	Shutdown	Normal	111[(1)]
e.	Hydrostatic test	Test	3

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(1) In the design of NSSS piping systems, there are 9 transients not counted for the shutdown; 8 are due to SRV blowdown and 1 is due to automatic depressurization.

f.	Design hydrotest	Test	130

g.	OBE	Upset	50
h.	Turbine stop valve closure	Upset	120
i.	Relief valve lift (at 3 cycles per actuation)	Upset	34,200

3.9.1.1.6 Recirculation System Transients

Transients considered in the recirculation piping stress analysis are as follows:

	Transient	Category	Cycles
a.	Startup	Normal	120[(1)]
b.	Turbine roll and increase to power	Normal	120
c.	Loss of feedwater heater	Upset	10
d.	Partial feedwater heater bypass	Upset	70
e.	Scrams	Upset	180
f.	Shutdown	Normal	111[(1)]
g.	Loss of feedwater pumps, isolation valves closed	Upset	10
h.	Single SRV blowdown	Upset	8
i.	Design hydrotest	Test	130
j.	OBE	Upset	50

3.9.1.1.7 Reactor Assembly Transients

The reactor assembly includes the RPV, support skirt, shroud support, and shroud plate. The cycles listed in Table 3.9-2 are as specified in the reactor assembly design and fatigue analysis.

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(1) In the design of NSSS piping systems, there are 9 transients not counted for the shutdown; 8 are due to SRV blowdown and 1 is due to automatic depressurization.

3.9.1.1.8 Main Steam Isolation Valve Transients

The MSIVs are designed for the following service conditions and thermal cycles:

	Transient	Category	Cycles
a.	Preop @100°F/hr	Normal/Upset	150
b.	Startup (heating @100°F/hr)	Normal/Upset	120
c.	Shutdown		
	1. Cooling cycles @100°F/hr, 540°F to 375°F	Normal/Upset	120
	2. Cooling cycles @270°F/hr, 375°F to 330°F	Normal/Upset	120
	3. Cooling cycles @100°F/hr, 330°F to 100°F	Normal/Upset	120
d.	Scram cooling cycles @100°F/hr	Normal/Upset	180
e.	Emergency and faulted transients		
	1. 546°F to 281°F in 15 seconds	Emergency/Faulted	1
	2. 546°F to 375°F in 3.3 minutes	Emergency/Faulted	1
	375°F to 281°F @300°F/hr	Emergency/Faulted	1
	3. 546°F to 375°F in 10 minutes	Emergency/Faulted	8
	375°F to 281°F @100°F/hr	Emergency/Faulted	8
	4. 546°F to 583°F in 2 seconds	Emergency/Faulted	1
	583°F to 538°F in 30 seconds	Emergency/Faulted	1
	538°F to 400°F @100°F/hr	Emergency/Faulted	1
	400°F to 546°F @100°F/hr	Emergency/Faulted	1
	5. 561°F to 500°F in 7 minutes	Emergency/Faulted	10
	500°F to 400°F @100°F/hr	Emergency/Faulted	10
	400°F to 546°F @100°F/hr	Emergency/Faulted	10

3.9.1.1.9 Main Steam Relief Valves Transients

The transients used in the analysis of the MSRVs are as follows:

	Transient	Category	Cycles
a.	Preop and inservice testing (100°F/hr)	Normal/Upset	150
b.	Startup (100°F/hr) and pressure increase (0 psig to 1000 psig)	Normal/Upset	120
c.	Shutdown (100°F/hr, pressure decrease to 0 psig, 270°F/hr between 375°F and 330°F)	Normal/Upset	120
d.	Scram	Normal/Upset	180
e.	System pressure and temperature decay is from 1000 psig and 546°F, to 35 psig and 281°F within 15 seconds.	Emergency/Faulted	1
f.	System temperature change is from 546°F to 375°F within 3.3 minutes, and from 375°F to 281°F at a rate of 300°F/hr. Pressure change is from 1000 psig to 35 psig.	Emergency/Faulted	1
g.	System temperature change is from 546°F to 375°F within 10 minutes, and from 375°F to 281°F, at a rate of 100°F/hr. Pressure change is from 1000 psig to 35 psig.	Emergency/Faulted	8
h.	System temperature change is from 546°F to 583°F within 2 seconds, from 583°F to 538°F within 30 seconds, and from 538°F to 400°F with return to 546°F at a rate of 100°F/hr. Pressure change is from 1000 psig to 1350 psig, thence to 240 psig, with return to 1000 psig.	Emergency/Faulted	1

i.	System temperature changes greater than 30°F, are from 561°F to 500°F within 7 minutes, and from 500°F to 400°F, with return-to-normal operating temperature of 546°F, at a rate of 100°F/hr. Pressure change is from 1000 psig to 1180 psig, to 240 psig, with return-to-normal operating pressure of 1000 psig.	Emergency/Faulted	10

In addition, for RPV, RPV internals and piping New Loads Adequacy Evaluation, at least 7700 SRV cycles are considered to account for the pool dynamic loads. These 7700 cycles are based on 1100 actuations (total for 40 years) of all SRVs times seven stress cycles per actuation. Further, 4700 actuations (total for 40 years) for the most frequently actuated SRV times three stress cycles per actuation (14,100 total cycles) are used in the analysis of the SRV downcomers and SRV discharge lines in the wetwell and in the main steam piping analysis. The 4700 actuations over the 40 year plant design life for the most frequently actuated SRV is based on the original two-stage Target Rock SRV design used for LGS. This SRV has a longer blowdown time, similar to the Crosby SRV design.

ASME Section III, Paragraph NB3552 excludes various transients, and provides means for combining those which are not excluded. Review and approval of the equipment supplier's certified calculation provides assurance of proper accounting for the specified transients.

3.9.1.1.10 Recirculation Flow Control Valve Transients

Not applicable; LGS has no flow control valve.

3.9.1.1.11 Recirculation Pump Transients

The following transients are listed in the design specification as a requirement for design considerations. The vendor was required to submit a certification of compliance with certified design calculations which considered only a pressure transient (no thermal-stresses were required to be considered). Nozzle piping loads were considered in accordance with the following paragraph from the design specification:

> "The pump case shall be designed to withstand secondary stresses due to piping reactions in accordance with Paragraph 452.4b of the ASME Standard Code for Pumps and Valves for Nuclear Power (1968 Draft)."

	Transient	Category	Cycles
a.	Heatup and cooldown at 100°F/hr	Normal/Upset	30
b.	±29°F temperature changes	Normal/Upset	600
c.	±50°F temperature changes	Normal/Upset	200
d.	RPV pressure transients to 110% design pressure	Normal/Upset	30
e.	SRV blowdowns	Emergency	1
f.	Improper pump startup, 100°F to 546°F in 15 seconds	Emergency	1
g.	Cooling transient, 552°F to 281°F in 15 seconds	Faulted	2
h.	Hydrotest to 1300 psig	Test	130
i.	Hydrotest to 1670 psig	Test	3

3.9.1.1.12 Recirculation Gate Valve Transients

The following transients are considered in the design of the recirculation gate valves.

	Transient	Cycles
a.	50°F-575°F-50°F at 100°F/hr	300
b.	±29°F between limits of 50°F and 575°F, instantaneous	600
c.	±50°F between limits of 50°F and 546°F, instantaneous	200
d.	546°F to 375°F, over a 10 minute period	30
e.	546°F to 281°F, over a 15 second period	2
f.	130°F to 546°F, over a 15 second period	1
g.	110% design pressure at 575°F	1
h.	1300 psi at 100°F installed hydrostatic test	130
i.	1670 psi at 100°F installed hydrostatic test	3

3.9.1.2 Computer Programs Used in Analysis

Sections 3.9.1.2.1 through 3.9.1.2.5 discuss computer programs used in the analysis of specific NSSS components (computer programs were not used in the analysis of all components, thus not all components are listed). Non-NSSS programs are discussed in Section 3.9.1.2.6.

GE Programs

The verification of the following GE programs has been performed in accordance with the requirements of 10CFR50, Appendix B. Evidence of the verification of input, output, and methodology is documented.

a. PIPST01
b. MASS
c. SNAP (MULTISHELL)
d. GASP
e. NOHEAT
f. FINITE
g. DYSEA
h. SHELL 5
i. HEATER
j. FAP 71
k. CREEP PLAST
l. ANSYS
m. SAP4
n. ANSI-7
o. NOZAR
p. TSFOR
q. PISYS
r. PDA
s. EZPYP
t. LION4
u. SIMOK
v. DISPL
w. WTNOZ
x. SPECA04
y. GEAPL01
z. POSUM
aa. FTFLG

Vendor Programs

The verification of the following two groups of vendor programs is assured by contractual requirements between GE and the vendors. In accordance with the requirements, the quality assurance procedure of these proprietary programs used in the design of N-stamped equipment and non-ASME code items is in full compliance with 10CFR50, Appendix B.

Pump Motor Vendor Programs

a. RTRMEC

Chicago Bridge and Iron Programs

a. 7-11-GENOZZ
b. 9-48-NAPALM
c. 1027
d. 846
e. 781-KALNINS
f. 979-ASFAST
g. 766-TEMAPR
h. 767-PRINCESS
i. 928-TGRV
j. 962-E09262A
k. 984
l. 992-GAS
m. 1037-DUNHAM'S
n. 1335
o. 1606&1657-HAP
p. 1634N

3.9.1.2.1 Reactor Vessel and Internals

3.9.1.2.1.1 Reactor Vessel

The computer programs used in the preparation of the reactor vessel stress report are identified, and their use summarized in the following paragraphs.

3.9.1.2.1.1.1 Chicago Bridge and Iron Program 7-11 - GENOZZ

The GENOZZ computer program is used to proportion barrel and double taper-type nozzles to comply with the specifications of ASME Section III, and contract documents. The program either designs such a configuration or analyzes the configuration input to comply to code. If the input configuration does not comply with the specifications, the program modifies the design and redesigns it to yield an acceptable result.

3.9.1.2.1.1.2 Chicago Bridge and Iron Program 9-48 - NAPALM

The basis for the Nozzle Analysis Program - All Loads Mechanical (NAPALM) is to analyze nozzles for mechanical loads and find the maximum stress intensity and location. The program provides analyses at each mechanical load point of application. The maximum stress intensity is calculated for both the inside and outside surfaces at each reference location. The program measures the maximum stress intensity for both the inside and outside surfaces of the nozzle, as well as their angular locations as measured from the 0° reference location. The principle stresses are also listed.

Stresses resulting from each component of loading (bending, axial, shear, and torsion) are listed, as well as the loadings which cause these stresses.

3.9.1.2.1.1.3 Chicago Bridge and Iron Program 1027

This program is a computerized version of the analysis method contained in Reference 3.9-1.

Part of this program provides for the determination of the shell stress intensities (S) around the perimeter of a loaded attachment on a cylindrical or spherical vessel. Eight (S) values are calculated, one at each of four cardinal points, for both the upper and lower shell plate surfaces (ordinarily considered outside and inside surfaces). With the determination of each (S), the components of that (S) (two normal stresses, δ_x and δ_y, and shear stress τ) are also determined. This program provides the same information as the manual calculation, and the input data is essentially the geometry of the vessel and attachment.

3.9.1.2.1.1.4 Chicago Bridge and Iron Program 846

This program computes the required thickness of a hemispherical head with a large number of circular parallel penetrations, by means of the area replacement method, in accordance with ASME Code, Section III.

In cases where the penetration has a counterbore, the thickness is determined so that the counterbore does not penetrate the outside surface of the head.

3.9.1.2.1.1.5 Chicago Bridge and Iron Program 781 - KALNINS

The KALNINS thin-shell program is used to establish the shell influence coefficient, and to perform the detailed stress analysis of the vessel. The stresses and the deformations of the vessel can be computed for any combination of the following axisymmetric loading:

a. Preload condition

b. Internal pressure

c. Thermal load

This program is a thin elastic shell program for shells of revolution developed by Dr. A. Kalnins of Lehigh University. Extensive revisions and improvements have been made by Dr. J. Endicott, to yield the Chicago Bridge and Iron version of this program.

The basic method of analysis was published by Professor Kalnins (Reference 3.9-2).

3.9.1.2.1.1.6 Chicago Bridge and Iron Program 979 - ASFAST

The ASFAST program performs the stress analysis of axisymmetric, bolted closure flanges between the head and cylindrical shell.

3.9.1.2.1.1.6.1 ANSYS Engineering Analysis System, Revision 5.6, ANSYS, Inc.

ANSYS Program performed the stress analysis of the reactor vessel head, flange & upper shell for reduced pass tensioning/detensioning of RPV studs connecting RPV head to RPV shell.

3.9.1.2.1.1.7 Chicago Bridge and Iron Program 766 - TEMAPR

This program reduces any arbitrary temperature gradient through the wall thickness to an equivalent linear gradient. The resulting equivalent gradient has the same average temperature, and the same temperature-moment as the given temperature gradient. The input consists of the wall thickness and actual temperature distribution. The output contains the average temperature and total gradient through the wall thickness. The program is written in FORTRAN IV.

3.9.1.2.1.1.8 Chicago Bridge and Iron Program 767 - PRINCESS

The PRINCESS program calculates the maximum alternating stress amplitudes from a series of stress values, by the method in ASME Section III.

3.9.1.2.1.1.9 Chicago Bridge and Iron Program 928 - TGRV

The TGRV program is used to calculate temperature distributions in structures or vessels. Although it is primarily a program for solving the heat conduction equations, some provisions have been made for including radiation and convection effects at the surfaces of the vessel.

The TGRV program is a highly modified version of the TIGER heat transfer program, written about 1958 at Knolls Atomic Power Laboratory, by A.P. Bray.

The program utilizes an electrical network analogy to obtain the temperature distribution of any given system as a function of time. The finite-difference representation of the three-dimensional equations of heat transfer are repeatedly solved for small time increments, and continually summed. Linear mathematics is used to solve the mesh network for every time interval. Three basic forms of heat transfer (conduction, radiation, and convection), as well as internal heat generation, are included in the analysis.

TGRV calculates and outputs the steady-state or transient temperature distributions in a given structure, as a function of time. The program inputs are any odd-shaped structure which can be represented by a three-dimensional field, its geometry and physical properties, boundary conditions, and internal heat generation rates.

3.9.1.2.1.1.10 Chicago Bridge and Iron Program 962 - E0962A

Program E0962A is one of a group of programs (E0953A, E1606A, E0962A, E0992N, E1037N, and E0984N) which are used together to determine the temperature distribution and stresses in pressure vessel components, using the finite-element method.

Program E0962A is primarily a plotting program. Using the nodal temperatures calculated by program E1606A or Program E0928A, and the node and element cards for the finite-element model, it calculates and plots lines of constant temperature (isotherms). These isotherm plots are used as part of the stress report to present the results of the thermal analysis. They are also useful in determining at which points in time the thermal-stresses should be determined.

In addition to its plotting capability, the program can also determine the temperatures of some of the nodal points by interpolation. This feature of the program is intended primarily for use with the compatible TGRV and finite-element models that are generated by program E0953A.

3.9.1.2.1.1.11 Chicago Bridge and Iron Program 984

Program 984 is used to calculate the stress intensity of stress differences, on a component level, between two different stress conditions. The calculation of the stress intensity of stress component differences (the range of stress intensity) is required by ASME Section III.

3.9.1.2.1.1.12 Chicago Bridge and Iron Program 992 GASP

The GASP program, originated by Professor E.L. Wilson of the University of California at Berkeley, uses the finite-element method to determine the stresses and displacements of plane or axisymmetric structures of arbitrary geometry, and is written in FORTRAN IV. See Reference 3.9-3, for a detailed account. GASP structures may have arbitrary geometry, and have linear or nonlinear material properties. The loadings may be thermal, mechanical, accelerational, or a combination of these.

A structure to be analyzed is broken up into a finite number of discrete elements or "finite-elements", which are interconnected at a finite number of "nodal points" or "nodes." The actual loads on the structure are simulated by statically equivalent loads acting at the appropriate nodes. The basic input to the program consists of the geometry of the stress model and the boundary conditions. The program then gives the stress components at the center of each element and the displacements at the nodes, consistent with the prescribed boundary conditions.

3.9.1.2.1.1.13 Chicago Bridge and Iron Program 1037 - DUNHAM'S

DUNHAM'S program is a finite ring element stress analysis program. It determines the stresses and displacements of axisymmetric structures of arbitrary geometry subjected to either axisymmetric loads, or nonaxisymmetric loads represented by a Fourier series.

This program is similar to the GASP program (Chicago Bridge and Iron Program 992). The major differences are that DUNHAM'S can handle nonaxisymmetric loads (which requires that each node have three degrees of freedom), while the material properties for DUNHAM'S must be constant. As in GASP, the loadings may be thermal, mechanical, and accelerational.

3.9.1.2.1.1.14 Chicago Bridge and Iron Program 1335

To obtain stresses in the shroud support, the baffle plate must be made a continuous circular plate. The program makes this modification and allows the baffle plate to be included in Chicago Bridge and Iron program 781 (KALNINS) as two isotropic parts, with an orthotropic portion at the middle (where the diffuser holes are located).

3.9.1.2.1.1.15 Chicago Bridge and Iron Programs 1606 and 1657 - HAP

The HAP program is an axisymmetric nonlinear heat analysis program. It is a finite-element program, used to determine nodal temperatures in a two-dimensional or axisymmetric body subject to transient disturbances. Programs 1606 and 1657 are identical, except that 1606 has a larger storage area allocated, and can thus be used to solve larger problems. The model for program 1606 is compatible with Chicago Bridge and Iron stress programs 992 (GASP) and 1037 (DUNHAM'S).

3.9.1.2.1.1.16 Chicago Bridge and Iron Program 1634N

This program is used to analyze thin cylindrical shells subjected to local loading beyond the range where Bijlaard's curves are directly applicable, i.e., R/t >300.

This program computes stress and displacements in thin-walled elastic cylindrical shells subjected to mechanical loading such as radial loads, longitudinal and circumferential moments.

3.9.1.2.1.2 Reactor Internals

3.9.1.2.1.2.1 Core Plate Beam Buckling - PIPST01

PIPST01 is a computer program that calculates approximate core plate beam buckling capability. It uses the Rayleigh-Ritz energy method to determine the applied moment needed to begin yielding and then to buckle a given tee beam. The tee beam models a segment of a BWR/2-5 core plate with a stiffener beam. The pressure differential across the plate that would have created this moment is calculated for a given length of beam or size of core plate.

Generic dimension and material properties are all input by the user.

3.9.1.2.1.2.2 Other Programs

Other computer codes used for the analysis of the internal components are:

a. MASS
b. SNAP (MULTISHELL)
c. GASP
d. NOHEAT
e. FINITE
f. DYSEA
g. SHELL 5
h. HEATER
i. FAP 71
j. CREEP PLAST
k. ANSYS

Descriptions of these programs are given in Section 4.1.4.1.

3.9.1.2.2 Piping

3.9.1.2.2.1 Structural Analysis Program - SAP4

SAP4 is a general Structural Analysis Program for static and dynamic analysis of linear elastic complex structures. The finite-element displacement method is used to solve the displacements, and to compute the stresses of each element of the structure. The structure can be composed of unlimited numbers of three-dimensional truss, beam, plate, shell, solid, plate strain-plane stress, brick, thick shell, spring, or axisymmetric elements. The program can treat thermal and various forms of mechanical loading, as well as internal element loading. The dynamic analysis includes mode- superposition, time history, and response spectrum analyses. Earthquake loading, as well as time-varying pressure, can be treated. The program is very versatile and efficient in solving large and complex structural systems. The output contains displacements of each nodal point, as well as stresses at the surface of each element.

3.9.1.2.2.2 Component Analysis - ANSI-7

The ANSI-7 Computer Program determines stress and accumulative usage factors in accordance with NB-3600 of ASME Section III. The program performs stress analyses in accordance with the ASME sample problem, and has been verified by reproducing the results of the sample problem analysis.

3.9.1.2.2.3 Area Reinforcement - NOZAR

The computer program Nozzle Area Reinforcement Program (NOZAR) performs an analysis of the required reinforcement area for openings. The calculations performed by NOZAR are in accordance with the rules of the 1974 edition of ASME Section III.

3.9.1.2.2.4 Turbine Stop Valve Closure - TSFOR

The TSFOR program computes the time history forcing function in the main steam piping due to turbine stop valve closure. The program utilizes the method of characteristics to compute fluid momentum and pressure loads at each change in pipe section or direction.

3.9.1.2.2.5 Piping Analysis Program/PISYS

PISYS is a computer code for analyzing piping systems subjected to both static and dynamic piping loads. Stiffness matrices representing standard piping components are assembled by the program to form a finite-element model of a piping system. The piping elements are connected to each other via nodes called pipe joints. It is through these joints that the model interacts with the environment, and loading of the piping system becomes possible. PISYS is based on the linear-elastic analysis in which the resultant deformations, forces, moments, and accelerations at each joint are proportional to the loading and the superposition of loading is valid.

PISYS has a full range of static and dynamic load analysis options. Static analysis includes dead weight, uniformly distributed weight, thermal expansion, externally applied forces, moments, imposed displacements, and differential support movement (pseudostatic load case). Dynamic analysis includes mode shape extraction, response spectrum analysis, and time history analysis by modal combination or direct integration. In the response spectrum analysis, i.e., Uniform Support Motion Response Spectrum Analysis (USMA) or Independent Support Motion Response Spectrum Analysis (ISMA), the user may request modal response combination in accordance with Regulatory Guide 1.92. In the ground motion (uniform motion) or independent support time history analysis, the normal mode solution procedure is selected. In analysis involving time-varying nodal loads, the step-by-step direct integration method is used.

The PISYS program has been benchmarked against NRC piping models. The results are documented in a report to the Commission, "PISYS Analysis of NRC Benchmark Problems," NEDO-24210, August 1979, for mode shapes and USMA options. The ISMA option has been validated against NUREG/CR-1677, "Piping Benchmark Problems Dynamic Analysis Independent Support Motion Response Spectrum Method," published in August 1985.

3.9.1.2.2.6 Piping Dynamic Analysis Program - PDA

The pipe whip analysis was performed using the PDA computer program. PDA is used to determine the response of a pipe subjected to the thrust-force occurring after a pipe break. The program treats the situation in terms of generic pipe break configuration, which involves a straight, uniform pipe fixed at one end, subjected to a time-dependent thrust-force at the other end. A typical restraint used to reduce the resulting deformation is also included at a location between the two ends. Nonlinear and time-independent stress-strain relations are used to model the pipe and the restraint. Similar to the popular elastic-hinge concept, bending of the pipe is assumed to occur at the fixed end, and at the location supported by the restraint, only.

Shear deformation is also neglected. The pipe bending moment-deflection (or rotation) relation used for these locations is obtained from a static nonlinear cantilever beam analysis. Using moment-rotation relations, nonlinear equations of motion are formulated using energy considerations, and the equations are numerically integrated in small time steps to yield the time history of the pipe motion. Additional discussion of PDA is provided in Section 3.6.2.2.2.

3.9.1.2.2.7 Piping Analysis Program - EZPYP

EZPYP links the ANSI-7 and SAP programs together. The EZPYP program can be used to run several SAP cases by making user- specified changes to a basic SAP pipe model. By controlling files and SAP runs, the EZPYP program makes it possible to perform a complete piping analysis in one computer run.

3.9.1.2.2.8 Thermal Transient Program - LION4

The LION4 program is used to compute radial axialthermal gradients in piping. The program calculates a time history of ΔT_1, ΔT_2, T_a, and T_b (defined in ASME Section III, Class 1 piping analysis) for uniform and tapered pipe wall thickness.

3.9.1.2.2.9 Synthetic Time History Program - SIMOK

The SIMOK program provides a time history that is equivalent to an input response spectrum. The synthetic time history is used to generate a new spectrum that is plotted with the input spectrum, to verify that the time history and spectrum are equivalent. Synthetic time histories are used in a multiple input analysis of the piping.

3.9.1.2.2.10 Differential Displacement Program - DISPL

The DISPL program provides differential movements at each piping attachment point, based on building modal displacements.

3.9.1.2.2.11 WTNOZ Computer Program

WTNOZ is a time-share program for piping weight calculations.

3.9.1.2.3 Pumps and Motors

3.9.1.2.3.1 Recirculation Pumps

No computer programs were used in the design of the recirculation pumps.

3.9.1.2.3.2 Core Spray Pumps and Motors

The RTRMEC computer program is used in the analysis of a motor design representative of (or very similar in mechanical construction to) the core spray pump motor.

RTRMEC calculates and displays the results of a mechanical analysis of a motor rotor assembly acted upon by external forces at any point along the shaft (rotating parts only). The shaft deflection analysis, including magnetic and centrifugal forces, was conducted. The calculation for the seismic condition assumes that the motor is operating, and that the seismic, magnetic, and centrifugal forces all act simultaneously and in phase on the rotor-shaft assembly. Note that the distributed motor assembly weight is lumped at the various stations. The shaft weight at a station is the sum of one-half the weight of the incremental shaft length just before the station, plus one-half the weight of the adjacent incremental shaft length just after the station. Bending and shear effects are accounted for in the calculations.

The FTFLG computer program was used to analyze the flange joints connecting the pump bowl castings. The description of this program is provided in Section 3.9.1.2.5.3.

3.9.1.2.4 Dynamic Loads Analysis

3.9.1.2.4.1 Acceleration Response Spectrum Program/SPECA04

The SPECA04 computer program generates acceleration response spectrum, consistent with Regulatory Guide 1.122 for an arbitrary input of time history of piecewise-linear accelerations, i.e., to compute maximum acceleration responses for a series of single-degree-of-freedom systems subjected to the same input. It can accept acceleration time histories from a random file.
It also has the capability of generating the broadened/enveloping spectra in conformance with Regulatory Guide 1.122 when the spectral points are generated equally spaced in a logarithmic scale axis of period/frequency. This program is also used in seismic and SRV transient analysis.

3.9.1.2.4.2 Forces and Moment Time Histories Program/GEAPL01

The GEAPL01 computer program converts distributed asymmetric pressure time histories over a given area into equivalent time- varying nodal forces and moments for use as input to perform dynamic analysis of a system. The overall resultant forces and moment time histories at specified points of resolution can also be obtained from GEAPL01.

3.9.1.2.5 Residual Heat Removal Heat Exchangers

3.9.1.2.5.1 Structural Analysis Program - SAP4

SAP4 is used to analyze the structural and functional integrity of the RHR heat exchangers. The description of this program is provided in Section 3.9.1.2.2.1.

3.9.1.2.5.2 Beam Element Data Processing Program/POSUM

POSUM is used to process SAP4 generated data. POSUM is a computer code designed to process SAP4 generated beam element data for pump or heat exchanger models. The purpose is to determine the load combination that would produce the maximum stress in a selected beam element. It is intended for use on RHR heat exchangers with four nozzles or core spray pumps with two nozzles.

3.9.1.2.5.3 Effects of Flange Joint Connections/FTFLG

The flange joints connecting the pump bowl castings are analyzed using FTFLG program. This program uses the local forces and moments determined by SAP4 to perform flat flange calculations in accordance with the rules set forth in ASME Appendix II and ASME Section III.

3.9.1.2.6 Seismic Category I Items Other than NSSS

A list of computer programs used in the non-NSSS system components is provided in Table 3.9-3. This list consists of computer programs developed and/or owned by Bechtel, and of computer programs that are recognized and widely used in industry.

The Bechtel developed and/or owned computer programs are documented, verified, and maintained by Bechtel, and meet the requirements of 10CFR50, Appendix B. A brief description of each of these Bechtel programs is provided below.

3.9.1.2.6.1 ME101, Linear-Elastic Analysis

ME101 is a finite-element computer program that performs linear-elastic analysis of piping systems using standard beam theory techniques. The input data format is specifically designed for pipe stress engineering. ME101 performs a thorough check of the input prior to analysis. In addition, the program automatically modifies the geometry to improve the finite-element model.

The output may be used directly for piping design, for conformation to Code, and for other regulatory requirements. Both ASME Section III and ANSI B31.1 piping code editions are incorporated in ME101 to the extent of computing flexibility factors, stress intensification factors, and stresses.

ME101 performs static and dynamic load analysis of piping systems, effective weight calculations, and ASME Section III, Class 2 and 3, and ANSI B31.1 Code stress checks.

Static analysis considers one or more of the following: thermal expansion, dead weight, uniformly distributed loads, and externally applied forces, moments, imposed displacements and rotations, individual force loads, static seismic (uniform directional acceleration) loads, or seismic anchor movement analysis.

Dynamic analysis is based on the standard normal superposition techniques. The input excitation may be in the form of seismic response spectra or time-dependent loading functions. In the single or multiple response spectrum analysis, the user may request modal synthesis by SRSS method or by Regulatory Guide 1.92 closely spaced mode 10% (equation 4) method. ME101 can consider further differential damping for large and small pipe according to Regulatory Guide 1.61. Various methods of eigenvalue solution are available. Determinant search or subspace iteration considers all data points as mass points. In the time history analysis, the excitation may be in the form of arbitrary nodal forces, support displacements, rotations, or support accelerations that are not necessarily in phase.

ME101 checks stresses from design loads versus allowable stresses according to ASME/ANSI Code equations. The user may request design load checks for sustained loads, occasional loads, multimode thermal expansion and pipe break, except for time history load cases.

The ME101 restraint load summary report prints the support load results from several load cases together in the same report, except for time history load cases.

The general loading combinations capability for ME101 can combine the results of several load cases together, according to certain algebraic rules, to form a new load case. The new load case resulting from this may be used in stress comparisons or restraint load summaries, except for time history load cases. ME101 has the capability of saving load case results on a tape and using these results in late runs for stress checks, restraint load summary reports, and general loading combinations, except for time history load cases.

For piping configurations with optional node numbering, ME101 generates isometric plots. The user may obtain plots on ZETA or CALCOMP plotters on a Tektronix 4014 graphics terminal, or on a RMS-600 printer/plotter.

ME101 uses out-of-core techniques for both static and response spectra analysis and has no practical limitations to the number of equations or band width. However, the use of very large systems may become prohibitive due to cost of computation. The maximum number of mode shapes allowable for response spectra analysis is currently 125.

This program considers the zero period acceleration effect in seismic response analysis. It accepts coordinate and key-word data in English or metric units.

The ASME Benchmark Problem 1 demonstrates the solution for natural frequencies of a three-dimensional structure, as described in Reference 3.9-4.

Natural frequencies, in Hertz, from ME101 and Reference 3.9-4, are as follows:

Mode	Reference 3.9-4	ME101
1	110	112
2	117	116
3	134	138

A total of 26 test problems were used for the verification of the ME101 results. These verification problems have been compared against one of the following:

a. ME632, Computer Program, "Seismic Analysis of Piping Systems", VERB MODB, 1976 Bechtel International Corporation, San Francisco, CA.

b. "Pressure Vessel and Piping 1972 Computer Programs Verification", ASME.

c. Hand Calculations

d. EDS Superpipe, EDS Nuclear, San Francisco, CA.

e. NUPIPE-IIM, Nuclear Services Corporation Piping Analysis Program, Campbell, CA.

f. TPIPE, A Computer Program for Analysis of Piping Systems, PMB Systems Engineering, San Francisco, CA.

g. ADINA, A Computer Program, Massachusetts Institute of Technology, Boston, MA.

h. MSC/NASTRAN Program, McNeal Schwendler Corporation, Los Angeles, CA.

i. EASE2 Program, Engineering/Analysis Corporation, San Francisco, CA.

j. ANSYS, Swanson Analysis System, Inc., 1975, Elizabeth, PA.

The J1 version of ME101 also includes seven NRC benchmarked problems, as referenced in NUREG/CR-1677, dated August 1980.

ME-101 was also validated with the four benchmarked problems (using the multiple response spectrum/independent support motion method) that were transmitted from M. Hartzman (NRC) to G. Wang (Bechtel) on August 10, 1983. The results and comparisons of the problems using ME-101 were transmitted to M. Hartzman (NRC) from T.J. McDonald (Bechtel) on December 15, 1983.

The ME-101 MRS/ISM methodology was again validated with four multiple response spectrum problems, as referenced in NUREG/ CR-1677, Volume II, dated August 1985.

3.9.1.2.6.2 ME632, Piping System Analysis

Program Description

ME632 performs stress analyses of three-dimensional piping systems. The effects of thermal expansion, uniform load of the pipe, pipe contents and insulation, concentrated loads, movements of the piping system supports, and other external loads, such as wind and snow, may be considered. The input data format is specifically designed for pipe stress engineering, and the English system of units is used. A thorough checking of the input has been coordinated in the program.

The output may be used directly for piping design, and for conformation to code and other regulatory requirements. Piping codes, the ASME B&PV code, the B31.1 code, and the B31.3 code have been incorporated into the program to the extent of computing flexibility factors, stress intensification factors, and stresses.

A response spectrum analysis may be performed to analyze the effect of earthquake forces on the piping system; transient effects of water hammer, steam hammer, or other impulsive types of dynamic loading are also handled by the program. A plot of piping geometry and/or response spectrum curves may be obtained to verify the accuracy of the model.

Program Version and Computer

The current UNIVAC version of ME632 is being used by Bechtel.

Extent of Application

ME632 is a piping program developed by Bechtel. Its development began in 1970, and it is being continuously supported by Bechtel. It has been used by various Bechtel projects.

Test Problems

The ASME Benchmark Problem No. 1 demonstrates the solution for natural frequencies of a three-dimensional structure, as described in Reference 3.9-4.

The following table lists the natural frequencies from ME632 and Reference 3.9-4:

Natural Frequency Comparison, Hz

Mode No.	Reference 3.9-4	ME 632
1	110	111
2	117	116
3	134	137

3.9.1.2.6.3 ME912, Thermal-Stress

Program Description

Finite-difference representation of the heat diffusion equation is used for the pipe or component wall section in contact with fluid of specified temperature and flow rate time histories. The program is quasi-two-dimensional, so that reduction of severity of a given transient with distance from inlet is accounted for.

Thermal properties of water, and stainless and carbon steel are built in the program. Film transfer coefficients for water are computed by the program for each time step and pipe section. For other fluids such as steam, the program is used on a one-dimensional basis with user-supplied film coefficients. Sequential computations are done for pipe lengths of different diameters or wall thicknesses. Fluid outlet temperature data from one pipe length are stored for use as the inlet to the next pipe length downstream. Average temperature differences, T_a - T_b, are thus calculated for structural discontinuity.

Program Version and Computer

The ME912 program has been used by Bechtel on various Bechtel projects. A Univac 1110 computer is used to run the ME912 program.

Extent of Application

The ME912 program was developed from References 3.9-7, 3.9-8, and 3.9-9 by Bechtel. The ME912 program has been extensively used since 1975 for nuclear Class 1 component design on the Fast Flux Test Facility project.

Test Problem

For local gradients, the program has been compared with analytical flat plate data of Reference 3.9-8, and numerical results by in-house program ME643. The results are acceptable. Table 3.9-4 shows the comparison of ME912 with ME643 and analytical results from Reference 3.9-8. For axial variations of fluid and wall temperatures, the program agrees closely with the analytical solution of Reference 3.9-9.

The ME643 program was developed from References 3.9-11 and 3.9-12 by Bechtel.

The results of ME643 transient temperature responses on both inside and outside surfaces of a sample pipe are compared with Chart 36 of Reference 3.9-13, and plotted in Figure 3.9-1.

3.9.1.2.6.4 ME913, Nuclear Class 1 Piping Stress Analysis

Program Description

ME913 can determine stress intensity levels for Class 1 nuclear power piping components, equations 9 through 14 of subarticle NB-3650, "Analysis of Piping Components", ASME Section III.

Prior to using this program, the following information external to the program is required:

a. Piping configuration

b. Piping and piping component properties

c. Moment reactions due to:

   1. Thermal expansion loads

   2. Weight loads

   3. Dynamic loads

d. The thermal response of the piping system due to the specified transients is:

   ΔT_1, ΔT_2 and the (T_a-T_b) values for the key points during system life.

Program Version and Computer

The current ME913 version is being used by Bechtel. A Univac 1100 computer is used to run the ME913 program.

Extent of Application

ME913 is the revised and expanded version of the LOTEMP program, originally developed by Bechtel, and made available for use through the CDC 6600 computer. The LOTEMP program has been extensively used by the Bechtel Fast Flux Test Facility Systems Analysis Group since 1972, in the preliminary design of Fast Flux Test Facility Class 1 piping. The ME913 program has been used to analyze nuclear Class 1 piping for Bechtel nuclear power plant projects.

Test Problems

The Grand Gulf Project feedwater line was selected as a test problem. Hand calculations of a selected component in the piping system were performed in accordance with the sample problem (Reference 3.9-14). The results were compared with the computer output for code equations 9 through 14 in ME913. Table 3.9-5 shows the comparison between the ASME sample problem (Reference 3.9-14) and ME913 results.

3.9.1.2.6.5 NE452, Submerged Steam Line Reflood Analysis

Program Description

NE452 is used to analyze reflood transient in steam relief valve discharge lines that discharge to a water pool and are equipped with a vacuum relief valve. The code models the effect of the vacuum relief valve (considering both the air flow and the valve dynamics) on the line reflood phenomenon by calculating the dynamics of the slug motion of the reflooding water. The slug motion is affected by the steam condensation on the surface of the reflooding water and by the presence of noncondensable air admitted by the vacuum relief valve.

The output (predicted maximum reflood) can be used in relief line clearing transient analysis codes (such as NE805, discussed in Section 3.9.1.2.6.6) to determine pipe run forces. NE452, together with NE805, can be used for vacuum relief valve sizing.

Program Version and Computer

The current NE452 version is being used by Bechtel. A UNIVAC 1100 computer is used to run the NE452 program.

Extent of Application

Development of the NE452 program began in 1977 and is being continuously supported by Bechtel. It has been used for various Bechtel nuclear power plant projects.

Test Problems

The NE452 program was verified against PBAPS and Monticello ramshead and Mark I T-quencher test data. It has also been verified against Karlstein Mark II T-quencher and Caorso X-quencher test data. The predicted results agreed reasonably well with test data.

3.9.1.2.6.6 NE805, Relief Valve Clearing Analysis

Program Description

NE805 is a computer code that analyzes transients in discharge lines of changing cross-sectional area following a relief valve opening. The code predicts the time-dependent forces on the various pipe segments of the relief valve discharge line. It models the steam flow through the relief valve and the steam/air flows in the line. It also models the water flow in the submerged part of the line. The options for the exit device are a straight pipe, a ramshead, or a quencher model in the reservoir. The quencher model considers sequential uncovering of the quencher holes during air/water clearing. NE805 uses the method of characteristics and allows for heat transfer through the pipe wall. It calculates flow parameters, pressure, velocity, and density as functions of time and the distance along the discharge line. Using these calculated values, the code computes the dynamic forcing functions induced on various pipe segments of the relief valve discharge line.

The force output can be used directly for piping stress analysis in codes such as ME101, described in Section 3.9.1.2.6.1.

NE805 generates plots of flow parameter time histories and/or force time histories, an option specified by the user. The plots are obtained by CALCOMP 1036 plotter.

Program Version and Computer

The current UNIVAC version of NE805 is being used by Bechtel. A UNIVAC 1100 computer is used to run the NE805 program.

Extent of Application

NE805 was developed by Bechtel. Its development began in 1975 and is being continuously supported by Bechtel. It has been used by various Bechtel projects.

Test Problems

The NE805 program has been verified against Monticello Mark I T-quencher test, Karlstein Mark II T-quencher test, and Caorso X-quencher test. Comparison with test data was found to be reasonable.

3.9.1.2.6.7 ANSYS

Refer to Section 3.8.7.10

3.9.1.2.6.8 ME210 - Local Stress in Cylindrical Shells Due to External Loads

This standard presents a method of analyzing and determining local stresses in cylindrical shells due to external moments and forces acting on rigid attachments of circular or rectangular shape. This program is based on a paper "Local Stresses in Spherical and Cylindrical Shells Due to External Loadings" by Wichman, Hopper, and Mershon, published in Welding Research Council Bulletin No. 107, August 1965 and March 1979 Revision. Values from Bijlaard curves are obtained by interpolation procedures.

This program also calculates piping stress intensity due to internal pressures and moments in accordance with the pressure and moment stress calculations specified in equation 9 and equation 10 of ASME Section III, NB-3650. The local stress intensity and piping stress intensity are summed and printed out if the required information for piping stress calculation is specified in the input. If no information for piping stress calculation is given, only the local stresses including primary plus secondary stress intensity and primary membrane stress intensities are printed out.

3.9.1.2.6.9 ME602 - Spectra Merging and Simplified Seismic Analysis

ME602 performs the seismic analysis of small diameter piping systems (2 inch and under) using the modified response spectrum method described in BP-TOP-1, Revision 3. The program generates a set of tables of seismic spans, support reactions, and stresses for various pipe sizes.

This program performs response spectrum curve merging along with the calculation of the seismic span. The program can also be used independently for the sole purpose of merging spectrum curves and storing the combined spectrum data for ME101 analysis. A neutral plot file of the "RAW" or "COMBINED" spectrum curves can be generated for plotting on RMS, TEKTRONIX, CALCOMP, or any neutral file compatible plotter.

3.9.1.2.6.10 ME351 - Pipe Rupture Analysis Program

This program performs nonlinear elastic-plastic analysis of three-dimensional piping systems subjected to concentrated static or dynamic time history forcing functions. These forces may result from fluid jet thrust at the location of a postulated rupture of high energy piping. PIPERUP is an adaptation of the finite-element method to the specific requirements of pipe rupture analysis. Straight and curved beam (elbow) elements are used to mathematically represent the piping, and axial and rotational springs are used to represent restraints. The stiffness characteristics of piping and restraints can reflect elastic/linear strain hardening material properties, and gaps between piping and restraints can be modeled.

3.9.1.2.7 Computer Programs Used for Component Supports

A list of computer programs used in pipe support analysis is provided in Table 3.9-3. The list consists of computer programs developed and/or owned by Bechtel, and computer programs that are recognized and widely used in the industry.

The Bechtel developed and/or owned computer programs are documented, verified, and maintained by Bechtel and meet the requirements of 10CFR50, Appendix B. A brief description of each of these programs is provided below.

3.9.1.2.7.1 CE050 - BOLTS

BOLTS is an interactive program for determining the loads on concrete expansion anchors used for baseplates with symmetrical bolt patterns. The program incorporates the effects of plate flexibility, bolt stiffness, and attachment size. BOLTS is particularly well suited for the evaluation of baseplate expansion anchors commonly used in pipe, conduit, HVAC, cable tray, and such other small equipment supports.

3.9.1.2.7.2 ME150 - FAPPS - Frame Analysis for Pipe Supports

ME150 is an interactive program for the analysis and design of pipe support frames. It has built-in standard frames and the capability to design any other nonstandard ones. It optimizes member sizes, welds, and embedments based on various user-specified design criteria.

3.9.1.2.7.3 ME225 - Anchor Plate

ME225 presents a method of designing plate-type piping anchors. It determines thickness of anchor plate, thickness of guide plate, weld joining the plate and process pipe, weld joining the supporting structure, and take out dimension of anchor plate and guide plate.

3.9.1.2.7.4 ME035 - BASEPLATE

ME035 analyzes baseplate-type structural supports. It assumes a flexible plate resting on a nonlinear foundation. It gives concrete stresses, bolt factors of safety, and weld forces. It can analyze baseplates with variable thickness with multiple columnlike attachments.

3.9.1.2.7.5 ME226 - PICLAMP

PICLAMP will design the components of six special cases of pipe support clamps. It computes the minimum required thickness at two critical sections of the clamp. It also calculates the stress in the clamp studs. It computes the stresses in the stanchion and baseplate, when applicable, and the minimum weld size based on the stress. It also computes certain clamp dimensions and the total weight of the clamp and its associated hardware.

3.9.1.2.7.6 ME425 - STAND

STAND will design and evaluate pipe support base plates with concrete anchor bolt assemblies. Plates can be of arbitrary geometry anchored with bolts that can be located in a random pattern.

3.9.1.2.7.7 ME120 - WELD

The program presents a method of determining fillet weld sizes for connecting structural members. It accepts five different types of structural shapes and analyzes for 2-weld to 16-weld configurations. It is based on the approach described in "Design of Welded Structures" by O.W. Blodgett and "Solutions to Design of Weldments" by O.W. Blodgett.

3.9.1.2.7.8 ME152 - SMAPPS

SMAPPS analyzes and designs commonly used standard frames for pipe support including associated welds and baseplates with anchors for AISC and ASME Section III, Subsection NF requirements, as well as project deflection/stiffness requirements.

SMAPPS provides the benefits of a structural frame analysis program and the simplicity of pre-engineering standards. SMAPPS provides margin factors for frame, welds, and baseplate with anchors that minimize the need for re-evaluation of pipe support due to load changes and as-built reconciliation.

3.9.1.2.7.9 CE-901 - ICES STRUDL II

STRUDL is a broad, extensive, and general program for solving problems in structural engineering.

3.9.1.2.7.10 ME153 - MAPPS

MAPPS is a miscellaneous application program for pipe supports which enables the user to access any or all of the following pipe support analysis computer programs within the same run:

- Uniform weld
- Nonuniform welded
- Beta angle
- Clip angle
- Bolt spacing
- Anchor plate
- Local effects
- Clamp (PI clamp)

3.9.1.3 Experimental Stress Analysis

When experimental stress analysis is used in lieu of analytical methods for seismic Category I ASME Code items, the applicable ASME Code requirements for experimental testing of the specific component are applied. If testing is required for seismic Category I non-ASME code items, consideration is given to size effects, dimensional tolerances, and material properties of the tested part to ensure that the test results are a conservative representation of the load-carrying capability of the item installed at LGS.

3.9.1.3.1 Experimental Stress Analysis of NSSS Seismic Category I Items

No experimental stress analysis methods are used.

3.9.1.3.2 Seismic Category I Items Other Than NSSS

No experimental stress analysis methods are used.

3.9.1.4 Considerations for the Evaluation of Faulted Conditions

All seismic Category I equipment is evaluated for the faulted loading conditions. However, emergency stress limits rather than faulted stress limits are used in many cases. The following paragraphs in this section show examples of the treatment of faulted conditions for the major components on a component-by- component basis. Additional discussion of faulted analysis can be found in Sections 3.9.3 and 3.9.5, and Table 3.9-6. These analyses are based on the power rerate analysis, and do not reflect the use of GE13 and GE14 fuel. The impact of GE13 fuel is documented in Reference 3.9-28. The impact of GE14 fuel is documented to be bounded by GE13 fuel in Reference 3.9-34. The impact of the MUR power uprate on GE14 fuel is evaluated in Reference 3.9-31 and Reference 3.9-32. Reference 3.9-33 identifies new design basis values for Fuel Lift Margin and Control Rod Guide Tube Lift Forces under MUR conditions. Additional analyses which consider the use of GNF2 fuel are documented in Reference 3.9-35. The GNF2 fuel is demonstrated to be bounded by the analyses in Reference 3.9-33.

Sections 3.9.2.2 and 3.7 discuss the treatment of dynamic loads resulting from the postulated seismic and hydrodynamic events. Section 3.9.2.5 discusses the dynamic analysis of loads on the reactor internals under faulted conditions including additional blowdown forces. Deformations under faulted conditions have been evaluated in critical areas, and no cases have been identified where design limits, such as clearance limits, are violated.

Elastic-plastic analysis has not been used in evaluating LGS seismic Category I systems and components for compliance with service Level D Limits. The stress levels of these components are below the ASME allowable stress.

3.9.1.4.1 Control Rod Drive System Components

3.9.1.4.1.1 Control Rod Drives

The ASME Section III Code components of the CRD have been analyzed for faulted conditions shown in Section 3.9.1.1.1. The loading criteria, calculated, and allowable stresses for various operating conditions are summarized in Table 3.9-6(u).

The design adequacy of non-ASME code components of the CRD has been verified by analysis and extensive testing programs on component parts, specially instrumented prototype drives, and production drives. The testing included postulated abnormal events, as well as the service life cycle listed in Section 3.9.1.1.1. The following three types of tests were performed:

a. A test was conducted with the lower CRD flange oscillating with a 2 inch peak-to-peak displacement. No adverse effects were observed during the normal continuous drive-in or jog operation.

b. To simulate the seismic interaction, the core plate and top guide structures of the test vessel were displaced relative to the CRD housing centerline. The results showed no effect in CRD performance.

c. The test vessel fuel channels were deflected to simulate the seismic interactions. The test was performed with fuel channel deflections up to 1.5 inches, which are greater than the expected deflection values. Because the CRD and control rod were not permanently deformed, the drive operability was maintained.

The load criteria, calculated, and allowable stresses for various operating conditions is summarized in Table 3.9-6(v) including the results of the New Loads Adequacy Evaluation program.

3.9.1.4.1.2 Hydraulic Control Unit

The HCU was analyzed for the faulted condition. The seismic and hydrodynamic loads adequacy was demonstrated by test and analysis. The results show peak dynamic loads of 6.2 g (vertical) at the natural frequency of 8-12 Hz and 6 g (horizontal) at a natural frequency of 3-4 Hz. For comparison, the capability of the HCU to withstand loading is 15 g (vertical) at 8-12 Hz, 7 g (horizontal) at 3-4 Hz, and 9 g (horizontal z displacement) at 3-4 Hz.

In addition, a detailed analysis was performed to confirm that the test-mounting configuration is applicable to the LGS unique field installation.

The analysis of the HCU under faulted condition loads establishes the structural integrity of the system.

3.9.1.4.1.3 CRD Housing

The SSE is classified as a faulted condition; however, in the CRD housing analysis, the SSE event is treated as an emergency condition. The calculated and allowable stresses at various loading conditions are given in Table 3.9-6(v).

3.9.1.4.2 Standard Reactor Internal Components

3.9.1.4.2.1 Control Rod Guide Tube

The maximum calculated stress on the control rod guide tube occurs in the base during a faulted condition. In accordance with ASME Section III, the faulted limit is 2.4 S_m, where S_m = 16,000 psi at 575°F. The analysis and limiting stresses are summarized in Table 3.9-6(aa).

3.9.1.4.2.2 Incore Housing

The maximum calculated stress on the incore housing occurs at the outer surface of the vessel penetration during a faulted condition. The maximum allowable stress for the elastic analysis used is 2.4 S_m (39,948 psi), which bounds the calculated stress as given in Table 3.9-6(ab).

3.9.1.4.2.3 Jet Pump

The elastic analysis for the jet pump faulted conditions shows that the maximum stress is due to diffuser impulse loading during a pipe rupture and blowdown. The maximum allowable for this condition, in accordance with ASME Section III, is 3.6 S_m (60,840 psi). Table 3.9-6(w) summarizes the results of the analysis.

3.9.1.4.2.4 Low Pressure Coolant Injection Coupling

The maximum stress during a faulted condition on the LPCI coupling occurs at the "Bellows" (a purchased component designed to GE requirements for 120 normal operating condition cycles and 10 SSE cycles). Table 3.9-6(y) shows that calculated stresses are within allowable limits.

3.9.1.4.2.5 Orificed Fuel Support

Due to its complex configuration, a series of vertical and horizontal load tests were performed on the orificed fuel support to verify the design. The results show that the seismic and hydrodynamic loading is below the allowable limit with an average safety margin of 1.21 for normal and upset and 1.26 for faulted conditions.

3.9.1.4.3 Reactor Pressure Vessel Assembly

The RPV assembly was evaluated using elastic analysis methods for faulted conditions. Table 3.9-6(f) lists the calculated and allowable stresses for the various loading combinations.

3.9.1.4.4 Core Support Structure

The evaluations for faulted conditions for the core support structure are discussed in Section 3.9.5. The calculated and allowable stresses are summarized in Table 3.9-6(b).

3.9.1.4.5 Main Steam Isolation, Recirculation Gate, and MSRVs

Tables 3.9-6(g), 3.9-6(h), and 3.9-6(j) provide a summary of the analysis for the MSRVs, MSIVs, and recirculation gate valves, respectively.

Standard design rules, as defined in applied codes, are utilized in analyzing pressure boundary components of Class 1 active valves. Conventional, elastic stress analysis is used to evaluate components not defined in the code. The code allowable stresses are applied to determine acceptability of structure under applicable loading conditions, including faulted condition.

3.9.1.4.6 Main Steam and Recirculation Piping

For main steam and recirculation system piping, elastic analysis methods are used to evaluate faulted loading conditions. The allowable stresses using elastic techniques are obtained from the ASME Section III, Appendix F, "Rules for Evaluation of Faulted Conditions" (these are above elastic limits). Additional information for the main steam, recirculation piping, and pipe-mounted equipment is contained in Tables 3.9-6(d) and 3.9-6(e), respectively.

3.9.1.4.7 NSSS Pumps, Heat Exchangers, and Turbines

The recirculation, ECCS, RCIC, and SLC pumps, RHR heat exchangers and RCIC turbine are analyzed for the faulted loading conditions identified in Section 3.9.1.1. In all cases, stresses are within the elastic limits. The analytical methods, stress limits, and allowable stresses are discussed in Sections 3.9.2.2a and 3.9.3.1.

3.9.1.4.8 Control Rod Drive Housing Supports

Design adequacy of the CRD housing supports is shown by comparing the static and dynamic loads to the original design loads. The comparison, summarized in Table 3.9-6(z), shows that the hydrodynamic loads combined with other loads are less than the design load capability of the CRD housing supports.

3.9.1.4.9 Fuel Storage Racks

All analyses related to the Fuel Storage Racks are provided in UFSAR Section 9.1.

3.9.1.4.10 Fuel Assembly (Including Channel)

GE BWR fuel assembly (including channel) design bases, analytical methods, and evaluation results, including those applicable to the faulted conditions are contained in References 3.9-16 and 3.9-17. Evaluations specific to the LGS fuel assemblies have been performed in accordance with the methodology presented in Reference 3.9-17. The resulting acceleration profiles and fuel lift gap are summarized in Table 3.9-6(x).

3.9.1.4.11 Refueling Equipment

Refueling and servicing equipment which is important to safety is classified under essential components, per the requirements of 10CFR50, Appendix A. This equipment and other equipment whose failure would degrade an essential component is defined in Section 9.1 and is classified as seismic Category I. These components are subjected to an elastic dynamic finite-element analysis to generate loadings. This analysis utilizes appropriate seismic floor response spectra, and combines loads at frequencies up to 33 Hz for seismic and up to 100 Hz for hydrodynamic loads, in three directions. Imposed stresses are generated and combined for normal, upset, and faulted conditions. Stresses are compared, depending on the specific safety class of the equipment, to allowables of Industrial Codes, ASME, ANSI or Industrial standards, or AISC. Loading conditions, acceptance criteria, calculated and allowable stresses are shown in Table 3.9-6(s).

3.9.1.4.12 Seismic Category I Items Other than NSSS

The stress allowables for statically applied loads, of ASME Section III, Appendix F, Winter 1972, are used for code components. For noncode components, allowables are based on tests or accepted standards consistent with those in Appendix F of the code.

Dynamic loads for components loaded in the elastic range are calculated using dynamic load factors, time history analysis, or any other method that assumes the elastic behavior of the component.

The limits of the elastic range are defined in paragraph 1323 of Appendix F for the code components. The local yielding due to stress concentration is assumed not to affect the validity of the assumptions of elastic behavior. The stress allowables of Appendix F for elastically analyzed components are used for code components. For noncode components, allowables are based on tests or accepted material standards consistent with those in Appendix F for elastically analyzed components.

The methods used in evaluating the pipe break effects are discussed in Section 3.6.

3.9.2 DYNAMIC TESTING AND ANALYSIS

3.9.2.1a Piping Vibration, Thermal Expansion, and Dynamic Effects Testing for NSSS Piping

The test program is divided into three phases: piping vibration, thermal expansion, and dynamic effects.

3.9.2.1a.1 Piping Vibration

3.9.2.1a.1.1 Preoperational Vibration Testing of Recirculation Piping

The purpose of the preoperational vibration test phase is to verify that operating vibrations in the recirculation piping are within acceptable limits. This phase of the test uses visual observation and manual measurements by hand-held vibrograph to supplement remote measurements. If, during steady-state operation, visual observation indicates that vibration is significant, measurements are made with a hand-held vibrograph. Visual observations, and manual and remote measurements are made during the following steady-state conditions:

a. Recirculation pumps minimum flow

b. Recirculation pumps at 50% of rated flow

c. Recirculation pumps at 75% of rated flow

d. Recirculation pumps at 100% of rated flow

3.9.2.1a.1.2 Preoperational Vibration Testing of Small Attached Piping

During visual observation of each of the above test conditions (a. through d.), special attention is given to small attached piping and instrument connections to ensure that they are not in resonance with the recirculation pump motors or flow-induced vibrations. If the operating vibration acceptance criteria are not met, corrective action, such as modification of supports, is taken.

3.9.2.1a.1.3 Startup Vibration Testing of Main Steam Recirculation and RCIC Piping

The purpose of this phase of the program is to verify that the main steam, recirculation, and RCIC piping are within acceptable limits. The main steam and recirculation piping are instrumented with transducers to measure temperature, thermal movement, and vibration deflections. Because of limited access due to high radiation levels, no visual observation is required during this phase of the test. Remote measurements are made during the following steady-state conditions:

a. Main steam flow at 25% of rated

b. Main steam flow at 50% of rated

c. Main steam flow at 75% of rated

d. Main steam flow at 100% of rated

3.9.2.1a.1.4 Operating Transient Loads on Main Steam and Recirculation Piping

The purpose of the operating transient test phase is to verify that pipe stresses are within code limits. The amplitude of displacements and the number of cycles per transient of the main steam and recirculation piping are measured, and the displacements are compared with the acceptance criteria. The deflections are correlated with stresses to verify that pipe stresses remain within code limits. Remote vibration and deflection measurements are taken during the following transients:

a. Recirculation pump starts

b. Recirculation pump trip at 100% of rated flow

c. Turbine stop valve closure at 100% power

d. Manual discharge of each SRV at 1000 psig, and at planned transient tests that result in SRV discharge

3.9.2.1a.2 Thermal Expansion Testing of Main Steam and Recirculation Piping

The thermal expansion preoperational and startup testing program verifies that normal thermal movement occurs in the piping systems, and is performed through the use of potentiometer sensors. The main purpose of this program is to ensure the following:

a. The piping system is free to expand and move without unplanned obstruction or restraint in the x, y, and z directions, during system heatup and cooldown.

b. The piping system does "shake down" after a few thermal expansion cycles.

c. The piping system is working in a manner consistent with the assumption of the NSSS stress analysis

d. There is adequate agreement between calculated and measured displacements.

e. Thermal displacements are consistent and repeatable during heatup and cooldown of the NSSS systems.

Thermal expansion displacement limits are established prior to the start of piping testing. These are compared with the actual measured displacements to determine the acceptability of the actual motion. If the measured displacement does not vary by more than the specified tolerance from the acceptance limit, the piping system is responding in a manner consistent with predictions, and is therefore acceptable. Two levels of displacement limits are established to check the systems, as discussed in Section 3.9.2.1a.4.

3.9.2.1a.3 Dynamic Effects Testing of Main Steam and Recirculation Piping

To verify that snubbers adequately perform their intended function during plant operation, a dynamic testing program is planned, as part of the normal startup operation testing. The main purpose of this program is to ensure the following:

a. The vibration levels from the various dynamic loadings during transient and steady-state conditions are below the predetermined acceptable limits.

b. Due to underestimating the dynamic effects caused by cyclic loading during plant transient operations, long-term fatigue failure does not occur.

This dynamic testing is to account for the acoustic wave due to the SRV lifts (RV1), SRV load resulting from air clearing (RV2), and turbine stop valve closure load. The maximum stresses developed in the piping by the RV1, RV2, and turbine stop valve closure transients analysis are used as a basis for establishing criteria which assures proper functioning of the snubbers. If field measurements exceed criteria limits, this may indicate that the snubbers are not operating properly. If field measurements are within criteria limits, it is assumed that the snubbers are functioning properly. Sample production snubbers of each size (i.e. 10 kips, 20 kips, 50 kips, etc.) will also be qualified and tested for design and faulted condition loadings, prior to shipment to field. Snubbers will be tested to allow free piping movements at low velocity. During plant startup, the snubbers will be checked for improper settings and checked for any evidence of oil leak.

The criteria for vibration displacements is based on the assumed linear relationship between displacements, snubber loads and magnitudes of applied loads, for any function and response of the system. Thus the magnitudes of limits of displacements, snubber loads, and nozzle loads are all proportional. Maximum displacements (Level 1 limits) are established to prevent the maximum stress in the piping systems from exceeding the normal and upset primary stress limits, and/or the maximum snubber load from exceeding the maximum load to which the snubber has been tested.

Based on the above criteria, Level 1 displacement limits are established for all instrumented points in the piping system. These limits will be compared with the field measured piping displacements. Method of acceptance is as explained in Section 3.9.2.1a.4.

3.9.2.1a.4 Test Evaluation and Acceptance Criteria for Main Steam and Recirculation Piping

The piping response to test conditions is considered acceptable if the organization responsible for the stress report reviews the test results, and determines that the tests verify that the piping responded in a manner consistent with the predictions of the stress report, and/or that the tests verify that piping stresses are within code limits (ASME Section III, NB-3600). Acceptable deflection limits are determined after the completion of the piping systems stress analysis and are provided in the startup test specifications.

To ensure test data integrity and test safety, criteria have been established to facilitate assessment of the test while it is in progress. These criteria, designated Level 1 and Level 2, are described in the following paragraphs.

3.9.2.1a.4.1 Level 1 Criteria

Level 1 establishes the maximum limits for the level of pipe motion which, if exceeded, makes a test hold or termination mandatory.

If the Level 1 limit is exceeded, the plant will be placed in a satisfactory hold condition, and the responsible piping design engineer will be advised. Following resolution, applicable tests must be repeated to verify that the requirements of the Level 1 limits are satisfied.

3.9.2.1a.4.2 Level 2 Criteria

If the Level 2 criteria are satisfied for both steady-state and operating transient vibrations, there will be no fatigue damage to the piping system due to steady-state vibration, and all operating transient vibrations are bounded by the values in the stress report.

Exceeding the Level 2 specified pipe motion requires that the responsible piping design engineer be advised. Plant operating and startup testing plans would not necessarily be altered. Investigations of the measurements, criteria, and calculations used to generate the pipe motion limits would be initiated. An acceptable resolution must be reached by all appropriate and involved parties, including the responsible piping design engineer.

Detailed evaluation is needed to develop corrective action or to show that the measurements are acceptable. Depending on the nature of such resolution, the applicable tests may or may not be repeated.

3.9.2.1a.4.3 Acceptance Limits

For steady-state vibration, the piping break stress due to vibration only (neglecting pressure) will not exceed 10,000 psi for Level 1 criteria and 5,000 psi for Level 2 criteria. These limits are below the piping material fatigue endurance limits as defined in Design Fatigue Curves in appendix I of ASME code for 10^6 cycles.

For operating transient vibration, the piping bending stress (zero to peak) due to operating transient only will not exceed 1.2 S_m or pipe support loads will not exceed the Service Level D ratings for Level 1 criteria. The 1.2 S_m limit ensures that the total primary stress including pressure and dead weight will not exceed 1.8 S_m, the new Code Service Level B limit. Level 2 criteria are based on pipe stresses and support loads not to exceed design basis predictions. Design basis criteria require that operating transients stresses and loads not to exceed any of the Service Level B limits including primary stress limits, fatigue usage factor limits, and allowable loads on snubbers.

3.9.2.1a.5 Corrective Actions for Main Steam and Recirculation Piping

During the course of the tests, the remote measurements are regularly checked to determine compliance with Level 1 criteria. If trends indicate that Level 1 criteria may be violated, the measurements are monitored at more frequent intervals. The test is held or terminated as soon as Level 1 criteria are violated. As soon as possible after the test hold or termination, the following corrective actions are taken:

a. Installation Inspection: A walkdown of the piping and suspension is made to identify any obstruction or improperly operating suspension components. Snubbers are located close to the midpoint of the total travel range at the operating temperature. Hangers are in their operating range between the hot and cold settings. If vibration exceeds the criteria, the source of the excitation must be identified to determine if it is related to equipment failure. Action is taken to correct any discrepancies before repeating the test.

b. Instrumentation Inspection: The instrumentation installation and calibration is checked, and any discrepancies are corrected. Additional instrumentation is added, if necessary.

c. Repeat Test: If actions a. or b. identify discrepancies that could account for failure to meet Level 1 criteria, the test is repeated.

d. Resolution of Findings: If the Level 1 criteria are violated on the repeat test, or no relevant discrepancies are identified in a. or b., the organization responsible for the stress report reviews the test results and the criteria to determine if the test can be safely continued.

If the test measurements indicate failure to meet Level 2 criteria, the following corrective actions are taken after completion of the test:

a. Installation Inspection: A walkdown of the piping and suspension is made to identify any obstruction or improperly operating suspension components. Snubbers are located close to the midpoint of the total travel range at the operating temperature. Hangers are in their operating range between the hot and cold settings. If the vibration exceeds limits, the source of the vibration must be identified. Actions, such as suspension adjustment, are taken to correct any discrepancies.

b. Instrumentation Inspection: The instrumentation installation and calibration are checked, and any discrepancies are corrected.

c. Repeat Test: If a. or b. above identify a malfunction or discrepancy that could account for failure to comply with Level 2 criteria, and appropriate corrective action is taken, the test may be repeated.

d. Documentation of Discrepancies: If the test is not repeated, the discrepancies found under actions a. or b. above are documented in the test evaluation report and correlated with the test condition. The test is not considered complete until the test results are reconciled with the acceptance criteria.

3.9.2.1a.6 Measurement Locations for Main Steam and Recirculation Piping

Remote vibration measurements during initial startup will be made for each of the main steam lines and recirculation lines. The locations of the measurements will be described in the startup test specification.

During preoperational testing of recirculation piping, visual observation and manual measurements by hand-held vibrograph are made to supplement the remote measurements.

3.9.2.1b Piping Vibration, Thermal Expansion, and Dynamic Effects Testing of Non-NSSS Piping

The dynamic effects on all safety-related ASME Class 1, 2, and 3 piping systems, including their supports and restraints, are considered as required by NB-3622.3, NC-3622, and ND-3622 of ASME Section III. The structural and functional integrity of each piping system under postulated seismic events is verified by dynamic analysis only. Piping systems having significant anticipated transient loads, caused by main stop valve closure or relief valve discharge for example, are analyzed for time-dependent forces. In addition, piping steady-state vibration and dynamic transient tests will be performed as summarized below, to ensure that:

a. Excessive steady-state vibration is not present in the piping that would result in piping stresses and restraint loads above the allowables

b. The piping is adequately restrained to withstand the dynamic transient loads.

The power ascension tests performed for each non-NSSS piping system are provided in Table 3.9-7. The table also describes the systems as to ASME Code class, high energy or moderate energy piping designation, and seismic category.

The startup test program specifications describe in detail the piping that is instrumented for remote monitoring of vibrations and thermal expansion and the piping that is accessible for preoperational or startup walkdown testing by test personnel. The test criteria limit the permissible pipe vibratory stress to the allowable limits prescribed in the industry standard for startup testing of nuclear power systems, ANSI/ASME OM3.

The LGS startup testing program requires that the following conditions be demonstrated in accordance with Regulatory Guide 1.70:

a. Thermal expansion is free from significant and unacceptable restraint not accounted for in the design.

b. Piping vibration is within acceptable limits for long-term vibratory stress.

c. Dynamic transient response of the piping is compared to the design analysis expected values. If those values are exceeded, the test results are compared with the maximum that would be allowed under the ASME Code stress limits.

Cognizant design personnel familiar with the systems to be tested participate in the development of test procedures, and in the evaluation of the test results. The data acquired from the tests are compared with the anticipated results to determine the acceptability of the total system response. Refer to STP-17, STP-33 and STP-36 in Table 14.2-3 for these startup test program procedures.

Test specifications governing the scope of startup testing of BOP piping were prepared and were intended to be the repository for all primary information relating to the scope, objectives, methods, instrumentation, measurements, and criteria for evaluation of the test results. The BOP piping systems are categorized in terms of the following:

a. Test environment (hot deflection, steady-state vibration or dynamic transient response).

b. Test measurements (remotely monitored, visual or none required due to small expected response to test environment).

c. The appropriate testing phase (preoperational or power ascension).

Piping thermal expansion tests are performed for the safety-related piping systems with normal operating temperatures exceeding 300°F. Safety-related piping systems with normal operating temperatures less than 300°F do not have significant thermal expansion to warrant these tests. Engineering review of all seismic Category I piping systems, including their supports, and restraints or snubbers, was performed after completion of construction and prior to fuel load. This ensures that no restraint of normal thermal movement occurs due to interferences and obstructions, and that the support and restraints are in accordance with the design intent. For those systems receiving thermal expansion tests, pipe movements are monitored to ensure that no restraint of normal thermal movement occurs at locations other than at the designed restraint locations.

By monitoring the thermal movement, the thermal expansion test program verifies that the free thermal expansion of piping systems takes place at the snubbers. Performance of the snubbers designed for transient loads, such as those resulting from main stop valve closure or MSRV discharge, are verified by measuring the load in the snubber during the dynamic transient tests. The snubbers are qualified by dynamic testing for cyclic loading as described in Section 3.9.3.5.2.

The acceptance criterion for thermal expansion tests and dynamic transient tests is that the measured pipe displacements, accelerations, dynamic pressures, or restraint loads are below the calculated or design values.

3.9.2.1b.1 Piping Dynamic Transient Tests

During the power ascension, the piping dynamic transient tests identified in Table 3.9-7 are performed. The following modes are considered:

a. Main steam piping outside the containment for main steam turbine stop valve trip at 25%, 75% (Unit 1 only), and 100% power. Main Steam Turbine trip test for Unit 2 at 100% power will be performed during commercial operation of that Unit.

b. Main steam bypass piping for the turbine stop valve closure

c. MSRV discharge piping for the MSRV opening

d. HPCI turbine steam supply piping for HPCI turbine trip.

e. Feedwater piping for reactor feed pump trip/coastdown

f. Feedwater heater drain piping for dump and drain valve actuation (Unit 1 only)

g. Moisture separator drain piping for flashing during normal operation and moisture separator depressurization (Unit 1 only).

From past experience, the dynamic transients in other piping systems are not significant.

Dynamic transient analysis of the subject lines is performed to determine the response of the piping system and the restraint loads. During the test the accelerations of the pipe, loads in the snubbers and restraints and the pressure at representative locations will be measured as required.

The acceptance criterion for this test is that the measured response of the piping system, the snubber, and restraint loads are below the design values. When the measured response exceeds the calculated response, or restraint loads, detailed evaluation of the design will be made to determine the acceptability.3.9.2.1b.2 Piping Steady-State Vibration Testing

The piping systems identified in Table 3.9-7 are tested for steady-state vibration during preoperational test programs or during power ascension. The following operating modes are considered:

a. RHR pump operation

b. HPCI pump and turbine operation

c. RCIC pump and turbine operation

d. Core spray pump operation

e. Main steam

f. Feedwater

g. RWCU

In addition, during the system walkdown, upon initial startup or power escalation, any abnormal vibrations of other systems observed are reviewed and instrumented, if necessary, to determine the acceptability of such vibration.

All safety-related process piping systems and safety-related instrument lines are included in the vibration monitoring program (Table 3.9-7). A vibration monitoring test specification is prepared to categorize the requirements for the testing program. Safety-related systems are categorized as follows:

a. Systems or portions of systems having no flow within a significant portion of their lines; for those system, no testing is required.

b. Systems with flow:

- Accessible lines (including attached instrument lines) monitored visually or with hand-held instruments.

- Inaccessible lines other than instrumentation lines will be monitored by remote instrumentation.

- Inaccessible instrument lines.

The inside containment instrumentation lines listed below that are inaccessible for visual inspection during power ascension testing will be included in the vibration monitoring program:

- RPV level indicator instrumentation lines

- Main steam instrumentation lines used for monitoring main steam flow

- RCIC steam supply instrument lines used for monitoring steam flow

- CRD lines

- Main steam sample lines.

The vibration monitoring program for the above lines consists of walkdown by stress analysts experienced in vibration assessment prior to power ascension testing. The review is to include proper restraining of the lines near the vibration source, at elbows and bends, and at the concentrated masses. Prior to walkdown, the stress engineer will review the mode shapes and piping analytical responses to normal and upset conditions. Any lines found to be inadequately restrained, resulting in the potential for excessive vibration, will be evaluated on a case-by-case basis to assess the impact on plant safety that could result from operation of the lines. The evaluation will identify the mode(s) of operation of such lines that cause the excessive vibration. Corrective action to prevent excessive vibration will be taken prior to operating any lines in the operating mode(s) that cause excessive vibration.

Steady-state vibration is primarily induced by the flow in the pipe and the equipment motion. In general, the nature of the steady-state vibration is not known a priori. Therefore, design engineers with stress analysis experience and with familiarity with the subject piping system observe the lines during significant modes of system operation as shown in Items a. through g. above, and classify each line as acceptable, if the vibration is not significant, or as questionable, if the vibration is significant. The lines with questionable steady-state vibration are monitored as applicable by suitable instrumentation to determine the system response.

The type of any necessary instrumentation is determined by the design engineer, so that the maximum amplitude and frequency response of the piping system can be determined. The instrumentation does not screen out the significant frequencies.

The acceptance criterion for the steady-state vibration tests is that the maximum measured amplitude of the piping vibration does not induce more stress in the pipe than the endurance limit of the material. By limiting the maximum stress in the pipe due to steady-state vibration below the allowable limits prescribed in the industry standard for startup testing of nuclear power systems (ANSI/ASME OM3), the steady-state vibration induced stress does not contribute to reducing piping fatigue life. These limits are based on the piping design fatigue curves of up to 10^6 cycles of vibration given in ASME Section III, Appendix I. To account for fatigue with higher cycles, the design fatigue strength of carbon steels will be reduced by applying a factor of 0.8 and further employing a safety factor of 1.3. Austenitic pipe steels design fatigue strength reduction factor will be 0.6, and is further reduced by employing a safety factor of 1.3. Piping stress indices (K_2C_2) and intensification factors (2i) as applicable to each particular system are also applied in accordance with the standard.

When required, additional restraints are provided to reduce the steady-state vibration, and to keep the stresses below the acceptance criteria levels. Table 3.9-7 provides a reference to the appropriate test descriptions in Chapter 14.

3.9.2.2 Dynamic Qualification of Safety-Related Mechanical Equipment

The qualification discussion in the following sections is generally divided into two types of equipment: NSSS equipment and non-NSSS equipment. The design criteria and qualification procedures for NSSS equipment includes the effects of both seismic and hydrodynamic loads. The non-NSSS equipment qualification discussions include only seismic design criteria and qualification procedures. Refer to Appendix 3A.6.8 and 3A.7.1.7 for discussion of non-NSSS equipment subject to hydrodynamic loads. Appendix 3A.6.7 and 3A.7.1.5 contain further discussion of NSSS equipment qualification.

Safety-related NSSS and Non-NSSS mechanical equipment were reviewed to SRP 3.10 Seismic Qualification Review Team (SQRT) requirements including IEEE 344-1975 and Reg. Guides 1.100 and 1.92. The SQRT re-assessment concluded that the seismic and dynamic qualification program meets the intent of IEEE 344-1975 and Reg. Guides 1.100 and 1.92. Refer to Section 3.10.2.

3.9.2.2a Dynamic Qualification of NSSS Safety-Related Mechanical Equipment

This section describes the criteria for dynamic qualification of safety-related mechanical equipment, and the qualification testing and/or analysis applicable to this plant for all the major components, on a component-by-component basis. In some cases, a module or assembly consisting of mechanical and electrical equipment is qualified as a unit, for example, ECCS pumps. These modules are generally discussed in this section, rather than in Sections 3.10 and 3.11. Electrical supporting equipment, such as control consoles, cabinets, and panels, which are part of the NSSS, are discussed in Section 3.10. Dynamic qualification of NSSS pumps and valves is discussed in Section 3.9.3.2a.

3.9.2.2a.1 Tests and Analysis Criteria and Methods

The ability of equipment to perform its safety function during and after application of dynamic loads is demonstrated by tests and/or analysis. Selection of testing, analysis, or a combination of the two is determined by the type, size, shape, and complexity of the equipment being considered. When practical, equipment operability is demonstrated by testing. Otherwise, operability is demonstrated by analysis.

Analysis is also used to show there are no natural frequencies below 33 Hz for seismic loads and 100 Hz for hydrodynamic loads. If a natural frequency lower than these is discovered, dynamic tests may be conducted, and in conjunction with mathematical analysis, used to verify operability and structural integrity at the required dynamic input conditions.

When the equipment is qualified by dynamic testing, the response spectrum or time history of the attachment point is used in determining input motion.

Natural frequency may be determined by running a continuous sweep frequency search, using a sinusoidal steady-state input of low magnitude. Dynamic conditions are simulated by tests using random vibration input or single-frequency input (within equipment capability) at frequencies of interest. Whichever method is used, the input motion during testing envelopes the input amplitude expected during dynamic loading conditions.

Equipment having an extended structure, such as a valve operator, is analyzed by applying static-equivalent dynamic loads at the extended structure's center of gravity. In cases where the equipment's structural complexity makes mathematical analysis impractical, a test is used to determine operational capability at maximum equivalent dynamic load conditions. Pipe-mounted equipment are represented in the model used for the piping system dynamic analysis.

3.9.2.2a.1.1 Random Vibration Input

When random vibration input is used, the actual input motion envelopes the appropriate floor input motion at the individual modes. However, single-frequency input, such as sine beats, can be used, provided one of the following conditions are met:

a. The characteristics of the required input motion are dominated by one frequency.

b. The anticipated response of the equipment is adequately represented by no resonances, one resonance, or widely spaced resonances.

c. The input has sufficient intensity and duration to excite all modes to the required magnitude, so that the testing response spectra envelope the corresponding response spectra of the individual modes.

3.9.2.2a.1.2 Application of Input Motion

When dynamic tests are performed, the input motion is applied to one vertical and one horizontal axis simultaneously. However, if the equipment response along the vertical direction is not sensitive to the vibratory motion along the horizontal direction, and vice-versa, then the input motion is applied to one direction at a time. In the case of single-frequency input, the time phasing of the inputs in the vertical and horizontal directions is such that a purely rectilinear resultant input is avoided.

3.9.2.2a.1.3 Fixture Design

The fixture design simulates the actual service mounting, and causes no dynamic coupling to the equipment.

3.9.2.2a.1.4 Prototype Testing

Equipment testing is conducted on prototypes of the equipment installed in this plant.

3.9.2.2a.2 Dynamic Qualification of Specific NSSS Mechanical Components

The following sections discuss the testing or analytical qualification of NSSS equipment. Seismic and hydrodynamic qualification is also described in Sections 3.9.1.4, 3.9.3.1, and 3.9.3.2 and in Appendix 3A.6.7 and 3A.7.1.6.

3.9.2.2a.2.1 Jet Pumps

A dynamic analysis of the jet pumps was performed. The stresses resulting from the analysis are below the design allowables.

3.9.2.2a.2.2 CRD and CRD Housing

A dynamic analysis of the CRD housing (with enclosed CRD) was performed. The results of the analyses established the structural integrity of these components. Preliminary dynamic tests verified the operability of the CRD subjected to displacements resulting from a dynamic event. A simulation test, imposing a static bow in the fuel channels, was performed with the CRD functioning satisfactorily.

3.9.2.2a.2.3 Core Support (Fuel Support and Control Rod Guide Tube)

A detailed analysis imposing dynamic effects due to seismic and hydrodynamic events shows that the maximum stresses developed during these events are much lower than the maximum allowed for the component material.

3.9.2.2a.2.4 Hydraulic Control Unit

The HCU was evaluated by comparing the RRS with the TRS at mounting points through test and analysis.

3.9.2.2a.2.5 Fuel Assembly Including Channels

GE BWR fuel channel design bases, analytical methods, and evaluation results, including seismic and hydrodynamic considerations, are contained in References 3.9-16 and 3.9-17.

3.9.2.2a.2.6 Recirculation Pump and Motor Assembly

Calculations were made to assure that the recirculation pump and motor assembly is designed to withstand the specific static-equivalent seismic and hydrodynamic forces. The flooded assembly was analyzed as a free body supported by constant support hangers from the brackets on the motor mounting member, with snubbers attached to brackets located on the pump case and the top of the motor frame.

Primary stresses due to horizontal and vertical dynamic forces were considered to act simultaneously, and as a conservative measure, they were added directly. Horizontal and vertical seismic forces were applied at mass centers, and equilibrium reactions were determined for motor and pump brackets.

3.9.2.2a.2.7 ECCS Pump and Motor Assembly

The dynamic qualification of each ECCS pump and motor assembly (as a unit), while operating under faulted conditions, was met in the form of a response spectrum analysis. The maximum specified vertical and horizontal accelerations were applied, simultaneously, in the worst case combination. The results of the analysis indicate that the pumps are capable of sustaining the applicable loadings without overstressing the pump components.

A similarly designed motor was dynamically qualified by a combination of static analysis and dynamic testing. The complete motor assembly was seismically qualified by dynamic testing, in accordance with IEEE 344 (1975). The qualification test program included demonstration of startup and shutdown capabilities, as well as no-load operability during dynamic loading conditions. Refer to Section 3.10.

3.9.2.2a.2.8 RCIC Pump Assembly

The RCIC pump assembly was analytically qualified by static analysis for seismic and hydrodynamic loadings as well as the design operating loads of pressure, temperature, and external piping loads. The results of this analysis confirm that the stresses are less than the allowable.

3.9.2.2a.2.9 RCIC Turbine Assembly

The RCIC turbine was qualified for seismic and hydrodynamic loads by a combination of static analysis and dynamic testing.

The turbine assembly and its components were considered to be supported as designed, and horizontal/vertical accelerations were applied to the mass centers of gravity. The static analysis indicated that the turbine assembly is capable of sustaining the design accelerations and loadings without overstressing any components.

The complete turbine assembly was qualified by dynamic testing, in accordance with IEEE 344 (1975). The qualification test program demonstrated startup, steady-state operability, and shutdown capabilities. Refer to Section 3.10.

3.9.2.2a.2.10 SLCS Pump and Motor Assembly

The SLCS positive displacement pump and motor assembly is mounted on a common baseplate and qualified by a combination of static analysis and single-frequency testing.

The SLCS pump and motor assembly was analytically qualified by static analysis for dynamic loading, as well as for the design operating loads of pressure, temperature, and external piping loads. The results of this analysis confirm that the stresses are less than the allowable.

3.9.2.2a.2.11 RHR Heat Exchangers

A three-dimensional finite-element model of the RHR heat exchanger and its support was developed and analyzed using the response spectrum method to verify that the heat exchanger can withstand seismic and hydrodynamic loads. The same model was statically analyzed to evaluate the effect of the external piping loads and dead weight to ensure that the nozzle load criteria and stress limits were met. Critical location stresses were evaluated and found to be lower than the corresponding allowable values.

3.9.2.2a.2.12 SLCS Tank

The SLCS storage tank is a cylindrical tank, 9 feet in diameter and 12 feet high, bolted to the concrete floor. The SLCS tank was qualified by analysis for:

a. Stresses in the tank bearing plate

b. Bolt stresses

c. Sloshing loads imposed at the natural frequency of sloshing (0.58 Hz). The natural frequency of the tank is 58.8 Hz.

d. Minimum wall thickness

e. Buckling

The results confirm that the stresses at the investigated locations are below the allowable.

3.9.2.2a.2.13 Main Steam Isolation Valves

The MSIVs are qualified by dynamic testing and analysis. The dynamic characteristics of the MSIVs were modeled in the main steam piping analysis. The resulting moments and stresses from the piping interaction were below the test proven valve capability. This assured the structural integrity. The operability of the valve is demonstrated by the closure test while it was subjected to the faulted dynamic loads.

3.9.2.2a.2.14 Main Steam Relief Valves

Due to the complexity of this structure and the performance requirements of the valve, the total assembly of the MSRV (including electrical and pneumatic devices) was dynamically tested at accelerations equal to or greater than the combined SSE and hydrodynamic loading of this plant.

3.9.2.2a.2.15 HPCI Turbine

The HPCI turbine was dynamically qualified by a combination of static analysis and dynamic testing.

The turbine assembly and its components were considered to be supported as designed, and horizontal/vertical accelerations applied to the mass centers of gravity. The results of the analysis indicate that the turbine assembly is capable of sustaining the design accelerations and loadings without overstressing any components.

The complete turbine assembly was qualified by dynamic testing, in accordance with IEEE 344 (1975). The qualification test program demonstrated startup, steady-state operability, and shutdown capabilities. Refer to Section 3.10.

3.9.2.2a.2.16 HPCI Pumps

The HPCI booster pump and main pump assembly is a split body type, mounted on one common baseplate. The pump assembly was analytically qualified by three-dimensional dynamic analysis using the response spectrum modal analysis technique. Results are obtained by using acceleration forces acting simultaneously in two directions, one vertical and one horizontal.

3.9.2.2b Dynamic Qualification of Non-NSSS Safety-Related Mechanical Equipment

All non-NSSS seismic Category I equipment is designed to withstand the simultaneous horizontal and vertical accelerations caused by the OBE and the SSE, as defined herein, in conjunction with other applicable loads. Seismic Category I non-NSSS mechanical equipment and supports located in the containment, reactor enclosure, and control structure are also subjected to dynamic loads due to LOCA and MSRV discharge hydrodynamic phenomena. Appendix 3A.5.6, 3A.6.8 and 3A.7.1.7 provide a summary of the load combinations, capability assessment criteria, and methodology used to qualify the equipment to withstand these additional hydrodynamic loads in combination with seismic and all other applicable loads. The functions of instrumentation and controls, or other parts necessary for the functional requirements of the equipment, are not impaired when the equipment is subjected to these loads.

The criteria for the seismic qualifications of non-NSSS mechanical and electrical equipment, with the exception of valves and valve operators (other than relief valves), are contained in a project specification. IEEE 344, "Seismic Qualification of Class 1E Equipment for Nuclear Power Generating Stations," is used as a supplement to the project seismic specification in some of the individual equipment specifications. Both Standard IEEE 344 and the project specification address the requirements of the demonstration of the seismic adequacy of equipment by analysis and/or tests.

Seismic qualification of non-NSSS motor and air operated valves is addressed in Section 3.9.3.2b.2, and qualification of control valves is addressed in Section 3.10.

3.9.2.2b.1 Dynamic Qualification

Seismic Category I non-NSSS mechanical equipment is qualified by any one of the following methods:

a. Dynamic analysis

b. Testing

c. Combination of analysis and testing

d. Similarity to previously tested equipment.

A list of dynamic qualification package numbers for non-NSSS safety-related mechanical equipment is provided in Table 3.9-9. The qualification packages will be maintained by the licensee in a centrally located, readably auditable permanent file.

3.9.2.2b.2 Criteria

The criteria for dynamic qualification of the equipment are given below (Refer to Appendix 3A.6.8 for the criteria for the equipment subjected to hydrodynamic loads in addition to seismic loads).

3.9.2.2b.2.1 Response Spectrum Curves

The appropriate response spectrum curves for the equipment in question are issued with the material requisition or the equipment specification, for both OBE and SSE. Response spectrum curves are based upon the seismic analysis of the supporting structure, and represent a plot of the maximum dynamic response to a family of a single-degree-of-freedom damped oscillators at a particular location within the structure. Response spectrum curves, plotted in terms of acceleration versus frequency, correspond to various locations within the buildings, and are identified with respect to the points noted on the mathematical model for each direction of vibration to be considered. This may include the vertical, as well as both the north-south and the east-west horizontal directions. In addition, each response spectrum curve corresponds to a particular damping ratio, i.e., the ratio of damping of the single-degree-of-freedom system to critical damping. See Section 3.7 for the appropriate response spectrum curves.

3.9.2.2b.2.2 Load Combinations and Allowable Stress Limits

Seismic Category I equipment is designed to withstand the more severe of the following load combinations:

a. OBE Conditions

This includes gravity loads and operating loads (or DBA loads, if applicable), including associated temperatures and pressures combined by absolute sums, with the dynamic seismic loading of the OBE.

Allowable stresses in the structural steel portions may be increased to 125% of the allowable working stress limits as set forth in ASME Section III, or other applicable industrial codes. The customary increase in normal allowable working stress due to an earthquake is used if, according to the appropriate code, it is less than 25%. Resulting deflections, misalignment or binding of parts, or effects on electrical performance (for example, contact bounce) do not prevent operation of the mechanical equipment during or after the seismic disturbance.

b. SSE Conditions

This includes gravity loads and operating loads (or DBA loads, if applicable), including associated temperatures and pressures combined by absolute sums with the dynamic seismic loading of the SSE. Allowable stresses in the structural portions may be increased to 150% of allowable working stress limits (the appropriate codes listed in a.) but may not exceed 0.9 F_y in bending, 0.85 F_y for axial tension, and 0.5 F_y in shear, where (F_y) equals the material minimum yield stress at the design temperature. For equipment designed by the maximum shear stress theory, the difference between the maximum and minimum principal stresses does not exceed 0.9 F_y. The resulting deflections, misalignment, or binding of parts, or effects on electrical performance (for example, contact bounce) do not prevent operation of the mechanical equipment during or after the seismic disturbance.

3.9.2.2b.2.3 Prevention of Overturning and Sliding

Stationary equipment is designed to prevent overturning or sliding, by using anchor bolts or other suitable mechanical anchoring devices. The effect of friction on the ability to resist sliding is neglected. The effect of upward vertical seismic loads on reducing overturning resistance is considered. Anchoring devices are designed in accordance with the requirements of a. and b. above, and the AISC Manual of Steel Construction. The proposed anchoring system is shown on the Seller's drawings, so that the Buyer can provide the proper foundation.

3.9.2.2b.2.4 Dynamic Testing

For equipment qualified by testing, seismic adequacy was established by dynamically testing the equipment in accordance with the project specifications. The equipment was tested with its mountings simulating the actual installation conditions.

3.9.2.2b.2.5 Combined Analysis and Test

Some equipment is qualified by a combination of analysis and testing procedures.

An analysis is conducted on the overall assembly to determine its stress level and the transmissibility of motion from the base of the equipment to the critical components. The critical components are removed from the assembly and subjected to a simulation of the environment on a test table.

Testing methods are also used to aid the formulation of the mathematical model for any piece of equipment. Mode shapes and frequencies are determined experimentally and incorporated into a mathematical model of the equipment.

3.9.2.2b.2.6 Criteria for the Diesel Fuel Oil Storage Tanks

These tanks are buried below-grade under a cover of 9 feet of earth.

Tanks and tank supports are designed to withstand Cooper E80 loading, applied above 9 feet of saturated overburden. Tank walls and ends do not deflect more than 3% maximum under the most unfavorable loading conditions.

The diesel fuel oil storage tanks conform to the requirements of the UL 58 Code.

Tanks and their supports are designated seismic Category I, and are designed to resist the increased earth pressure from the OBE and the SSE.

Tanks are designed to withstand external pressure resulting from being buried in the ground when the tanks are empty. Uplift forces on buried tanks are resisted by the weight of the empty tank and the foundation mat, plus 9 feet of overburden.

The tanks were analyzed using a mathematical model considering soil/structure interaction.

3.9.2.3 Dynamic Response of Reactor Internals Under Operational Flow Transients and Steady-State Conditions

The major reactor internal components are subjected to extensive testing, coupled with dynamic system analyses, to properly describe the flow-induced vibration phenomena resulting from normal reactor operation, and from anticipated operational transients.

In general, the vibration forcing functions for operational flow transients and steady-state conditions are not predetermined by detailed analysis. Special analyses of the response signals, measured from reactor internals of similar designs, are performed to obtain the parameters that determine the amplitude and modal contributions in the vibration responses. These studies are useful for extrapolating the results from tests of internals and components of similar designs, are also performed. This vibration prediction method is appropriate where standard hydrodynamic theory cannot be applied due to the complexity of the structure and flow conditions. Elements of the vibration prediction method are outlined as follows:

a. Dynamic analysis of major components and subassemblies is performed to identify natural vibration modes and frequencies. The analysis models used for seismic Category I structures are similar to those outlined in Section 3.7.2.

b. Data from previous plant vibration measurements are assembled and examined to identify predominant vibration response modes of major components. In general, response modes are similar, but response amplitudes vary among BWRs of differing size and design.

c. Parameters are identified which are expected to influence vibration response amplitudes among the several reference plants. These include hydraulic parameters such as velocity and steam flow rates, and structural parameters such as natural frequency and significant dimensions.

d. Correlation functions of the variable parameters are developed which, when multiplied by response amplitudes, tend to minimize the statistical variability between plants. A correlation function is obtained for each major component and response mode.

e. Predicted vibration amplitudes for components of the prototype plant are obtained from these correlation functions, based on applicable values of the parameters for the prototype plant. The predicted amplitude for each dominant response mode is stated in terms of a range, taking into account the degree of statistical variability in each of the correlations. The predicted mode and frequency are obtained from the dynamic analysis of a. above.

The dynamic modal analysis also forms the basis for interpreting prototype plant preoperational and initial startup test results (Section 3.9.2.4 below). Modal stresses are calculated, and relationships are obtained between sensor response amplitudes and peak component stresses, for each of the lower normal modes. The allowable amplitude in each mode is that which produces a peak stress amplitude of ±10,000 psi.

3.9.2.4 <u>Confirmatory Flow-Induced Vibration Testing of Reactor Internals</u>

The LGS reactor internals are tested in accordance with the provisions of Regulatory Guide 1.20 (Rev 2) for nonprototype Category I plants. The test procedure requires operating the recirculation system at the rated flow, with internals installed (less fuel), followed by inspection for evidence of vibration, wear, or loose parts. The test duration is sufficient to subject critical components to at least 10^6 cycles of vibration during two-loop and single-loop operation of the recirculation system. At the completion of the flow test, the vessel head and shroud head are removed. The vessel is drained, and the major components are inspected on a selected basis. The inspection covers all components which were examined on the prototype design, including the shroud, shroud head, core support structures, the jet pumps, and the peripheral CRD and incore guide tubes. Access is provided to the reactor lower plenum. The prototype reactor for LGS is the Browns Ferry-3 design docketed on July 31, 1968 (Reference 3.9-18).

Reactor internals for LGS are substantially the same as the internal design configurations which have been tested in prototype BWR/4 plants. Results of the prototype tests are presented in a Licensing Topical Report, Reference 3.9-18. This report also contains additional information on the confirmatory inspection program.

3.9.2.5 <u>Dynamic System Analysis of the Reactor Internals Under Faulted Conditions</u>

In order to assure that no significant dynamic amplification of load occurs as a result of the oscillatory nature of the blowdown forces (Figure 3.9-2), a comparison is made between the periods of the applied forces and the natural periods of the core support structures being acted upon by the applied forces. These periods are determined from a 12 node vertical dynamic model of the RPV and internals. In addition to the masses of the RPV and core support structures, allowance is made for the water inside the RPV.

The time-varying pressures are applied to the dynamic model of the reactor internals described above. Except for the nature and locations of the forcing functions and the dynamic model, the dynamic analysis method is identical to that described for seismic analysis, and is detailed in Section 3.7.2.1. These dynamic components are combined with other dynamic loads (including hydrodynamic and seismic) by the SRSS method. The resultant force is then combined with other steady-state and static loads on an ABS basis to determine the design load. The results of the dynamic analysis of the reactor internals are summarized in Table 3.9-6(b). These results are based on the power rerate analysis, and do not reflect the use of GE13 or GE14 fuel. The impact of GE13 fuel is documented in reference 3.9-28. The impact of GE14 fuel is documented to be bounded by GE13 fuel in Reference 3.9-24. The impact of the MUR power uprate is evaluated in Reference 3.9-31 and Reference 3.9-32. Reference 3.9-33 identifies new design basis values for Fuel Lift Margin and Control Rod Guide Tube Lift Forces under MUR conditions. Additional analyses which considered the use o GNF2 fuel are documented in Reference 3.9-35. The GNF2 fuel is demonstrated to be bonded by the analyses in Reference 3.9-33.

3.9.2.6 Correlations of Reactor Internals Vibration Tests with the Analytical Results

Prior to initiating the instrumented vibration test program for the prototype plant, extensive dynamic analyses of the reactor and internals are performed. The results of these analyses are used to generate the allowable vibration levels during the vibration test. The vibration data obtained during the test are always analyzed in detail. The results of the data analysis, vibration amplitudes, natural frequencies, and mode shapes are then compared to those obtained from the theoretical analysis.

Such comparisons provide insight into the dynamic behavior of the reactor internals. The additional knowledge gained is utilized in the generation of the dynamic models for seismic and LOCA analyses for this plant. The models used for this plant are the same as those used for the vibration analysis of the prototype plant.

The vibration test data are supplemented by data from forced oscillation tests of reactor internal components, thereby providing the analysts with additional information concerning the dynamic behavior of the reactor internals.

3.9.3 ASME CODE CLASS 1, 2, AND 3 COMPONENTS, COMPONENT SUPPORTS, AND CORE SUPPORT STRUCTURES

3.9.3.1 Loading Combinations, Design Transients, and Stress Limits

This section delineates the criteria for selecting and defining design limits and loading combinations associated with normal operation, postulated accidents, and specified seismic and hydrodynamic events for the design of safety-related ASME Code components (except containment components, which are discussed in Section 3.8).

This section also lists the major ASME Class 1, 2, and 3 associated equipment and pressure-retaining parts and on a component-by-component basis, and identifies the applicable loadings, calculation methods, calculated stresses, and allowable stresses. Design transients for ASME Class 2 equipment are not addressed in the section, they are covered in Section 3.9.1.1; seismic and hydrodynamic related loads are discussed in Section 3.9.2.2.

Table 3.9-6 is the major part of this section; it presents the loading combination analytical methods (by reference or example) and also the calculated stress or other design values for the most critical areas of all ASME Class 1, 2, and 3 components supports and core support structures. These values (which are based on the power rerate analysis and do not reflect the use of GE13 or GE14 fuel) are also compared to applicable code allowables. The impact of GE13 fuel is documented in Reference 3.9-28. The impact of GE14 fuel is documented to be bounded by GE13 fuel in Reference 3.9-34. The impact of the MUR power uprate is evaluated in Reference 3.9-31 and Reference 3.9-32. Reference 3.9-33 identifies new design basis values for Fuel Lift Margin and Control Rod Guide Tube Lift Forces under MUR conditions. Additional analyses which consider the use of GNF2 fuel are documented in Reference 3.9-35. The GNF2 fuel is demonstrated to be bounded by the analyses in Reference 3.9-33.

3.9.3.1.1 Plant Conditions

All events that the plant might credibly experience during a reactor year are evaluated to establish a design basis for plant equipment. These events are divided into four plant conditions.

The plant conditions described in the following paragraphs are based on event probability (i.e., frequency of occurrence) and correlated design conditions as defined in the ASME Section III.

3.9.3.1.1.1 Normal Condition

Normal conditions are any conditions in the course of system startup; operation in the design power range; normal hot standby (with condenser available); and system shutdown other than upset, emergency, faulted, or testing.

3.9.3.1.1.2 Upset Condition

Upset conditions are any deviations from normal conditions which are anticipated to occur often enough that the design should include a capability to withstand the conditions without operational impairment. The upset conditions include those transients which result from any single operator error or control malfunction, transients caused by a fault in a system component requiring its isolation from the system, and transients due to loss of load or power. Vibratory motions due to an OBE are conservatively treated as upset. Hot standby with the main condenser isolated is an upset condition.

3.9.3.1.1.3 Emergency Condition

Emergency conditions are those deviations from normal conditions which require shutdown to correct the conditions or to repair damage in the RCPB. These conditions have a low probability of occurrence, but are included to provide assurance that no gross loss of structural integrity results as a concomitant effect of any damage developed in the system. Emergency condition events include, but are not limited to, transients caused by one or more of the following: a multiple valve blowdown of the reactor vessel; loss of reactor coolant from a small break or crack which does not depressurize the reactor system, nor result in leakage beyond normal makeup system capacity, but which does require the safety functions of containment isolation and reactor shutdown; improper assembly of the core during refueling; or vibratory motions of an OBE in combination with associated system transients.

3.9.3.1.1.4 Faulted Condition

Faulted conditions are those combinations of conditions associated with extremely unlikely postulated events, with consequences such that the integrity and operability of the system may be so impaired that considerations of public health and safety are involved. Faulted conditions encompass events that are postulated because their consequences include the potential for releasing significant amounts of radioactive material. These postulated events are the most drastic that must be designed against, and thus represent limiting design bases. Faulted condition events include, but are not limited to, one or more of the following: a control rod-drop accident; a fuel handling accident; a main steam line break; a recirculation loop break; the combination of small break/large break accident, dynamic motion associated with an SSE and hydrodynamic, and a LOOP; or the SSE.

3.9.3.1.1.5 Correlation of Plant Conditions with Event Probability

The probability per reactor year, (P), of an event associated with the plant conditions is listed below. This correlation can be used to identify the appropriate plant condition for any hypothesized event or sequence of events.

PLANT CONDITIONS	EVENT ENCOUNTERED (Probability per reactor year)
Normal (planned)	1.0
Upset (moderate probability)	$1.0>P>10^{-2}$
Emergency (low probability)	$10^{-2}>P>10^{-4}$
Faulted (extremely low probability)	$10^{-4}>P>10^{-6}$

3.9.3.1.1.6 Regulatory Guide 1.48

Regulatory Guide 1.48 was issued after the design of this plant was established, and is therefore not used as a design basis requirement. However, the GE design basis was representative of good industry practices at the time of design, procurement, and manufacture, and is shown to be in general agreement with the guidelines of Regulatory Guide 1.48 through the use of the alternate approaches. For a comparison of NSSS compliance with Regulatory Guide 1.48, refer to Table 3.9-10. This comparison reflects general GE practice on BWR/4s and BWR/5s, and therefore is applicable to LGS.

The design limits and loading combinations for non-NSSS seismic Category I fluid systems components have been evaluated as being in conformance with Regulatory Guide 1.48, although in some cases the guidelines were not specifically a design basis.

Pump and valve operability assurance during and after design loading events, as discussed in footnotes 6 and 11 in section C of the regulatory guide, is discussed in Section 3.9.3.2b.

3.9.3.1.2 Reactor Pressure Vessel Assembly

The reactor vessel assembly consists of the RPV, RPV support skirt, shroud support, and shroud plate.

The RPV, RPV support skirt, and shroud support are constructed in accordance with ASME Section III. The shroud support consists of the shroud support plate, and the shroud support cylinder and its legs. The RPV is an ASME Class 1 component. Complete stress reports on these components have been prepared in accordance with ASME requirements. Table 3.9-6(f) provides a summary of stress criteria load combinations, calculated stress, and available stresses. The stress analyses performed for the reactor vessel assembly, including the faulted condition, were completed using elastic methods. The stress load combinations and stress analyses for core support structures and other reactor internals are discussed in Section 3.9.5.

3.9.3.1.3 Main Steam Piping

The main steam piping discussed in this paragraph includes that piping extending from the reactor pressure vessel to the outboard MSIV. This piping is designed in accordance with the ASME Section III, Class 1, Subsection NB-3650. The load combinations and stress criteria for the main steam piping and pipe-mounted equipment are shown in Table 3.9-6(d).

The rules contained in ASME Section III, Appendix F are used in evaluating faulted loading conditions, independently of all other design and operating conditions. Stresses calculated on an elastic basis are evaluated in accordance with F-1360.

3.9.3.1.4 Recirculation Loop Piping

The recirculation system piping which is bounded by the RPV nozzles is designed in accordance with the ASME Section III, Class 1, Subsection NB-3650. The load combinations and allowables for the recirculation piping and pipe-mounted equipment are shown in Table 3.9-6(e). The rules contained in ASME Section III, Appendix F are used in evaluating faulted loading conditions, independently of all other design and operating conditions. Stresses calculated on an elastic basis are evaluated in accordance with F-1360.

3.9.3.1.5 Recirculation System Valves

The recirculation system suction and discharge gate valves are designed in accordance with the ASME Section III, Class 1, Subsection NB-3600. Loading combinations and other stress analysis data are presented in Table 3.9-6(j).

3.9.3.1.6 Recirculation Pump

In the design of the recirculation pumps, the ASME Section VIII, Division 1, 1971 Edition (with latest addenda) is used as a guide in calculations made to determine the thickness of pressure-retaining parts and to size the pressure-retaining bolting. At the time the LGS recirculation pump was procured, ASME Section III design rules for pumps were under development. The New Loads Adequacy Evaluation has demonstrated that the design meets ASME Section III requirements.

The pump vendor's calculations for the design of the pressure- containing components include the determination of minimum wall thickness and allowable stress and pressures. Loading conditions and stress information of those calculations are shown in Table 3.9-6(i).

Load, shear, and moment diagrams were constructed to scale, using live loads, dead loads, and calculated snubber reactions. Combined bending, tension and shear stresses were determined for each major component of the assembly, including the pump driver mount, motor flange bolting, and pump case.

The maximum combined tensile stress in the cover bolting was calculated using tensile stress from design pressure.

Combined primary stresses did not exceed 150% of the Code allowable stress shown in ASME Section VIII, 1971 Edition.

These methods and calculations demonstrate that the pump will maintain pressure integrity.

3.9.3.1.7 SLCS Tank

The ASME Code allowable stress limits for the faulted category are 1.2 S_m for general membrane, and 1.8 S_m for bending plus local membrane.

The SLCS tank is designed in accordance with ASME Section III. A summary of the design calculations and stress criteria used is shown in Table 3.9-6(m).

3.9.3.1.8 RHR Heat Exchangers

The RHR heat exchangers are designed in accordance with ASME Section III. The loading combinations and stress analyses for the RHR heat exchangers are given in Table 3.9-6(o).

3.9.3.1.9 RCIC Turbine

Although not under the jurisdiction of the ASME Code, the RCIC turbine is designed and fabricated following the basic guidelines for an ASME Section III, Class 2 component.

Operating conditions for the RCIC turbine include:

a. Surveillance Testing - Periodic operation with the reactor pressure at 1000 psia, nominal, and saturated temperature; and turbine exhaust pressure at 25 psia, peak, and saturated temperature.

b. Auto-Startup - 30 cycles per year with the reactor pressure at 1150 psia, nominal, and saturated temperature; and turbine exhaust pressure at 25 psia, peak, and saturated temperature.

Design conditions for the RCIC turbine include:

a. Turbine Inlet - 1250 psig at saturated temperature

b. Turbine Exhaust - 165 psig at saturated temperature

Table 3.9-6(q) contains a summary of the loading conditions, stress criteria, calculated stresses and allowable stresses of the RCIC turbine.

3.9.3.1.10 RCIC Pump

The RCIC pump has been designed and fabricated to the requirements for an ASME Section III, Class 2 component.

The RCIC pump is surveillance tested in conjunction with the RCIC turbine. An operational test is performed, in which the RCIC pump takes condensate from the CST, and discharges back to the CST at the design flow through a closed test loop.

Design conditions for the RCIC pump include:

a. Required NPSH - 20 feet

b.	Total head -	High speed	2800 feet
		Low speed	525 feet

c. Pump capacity - 625 gpm

d. Normal ambient operating temperature - 60°F to 100°F

e. Normal plus upset conditions which control the pump design include:

1. Design pressure	1500 psig
2. Design temperature	40°F - 140°F
3. OBE[(1)]	½ of SSE

Table 3.9-6(r) contains a summary of the loading conditions, stress criteria, calculated and allowable stresses for the RCIC pump components.

3.9.3.1.11 ECCS Pumps

The RHR and core spray pumps are designed in accordance with ASME Section III. The stress criteria and calculated and allowable stresses are summarized in Table 3.9-6(n).

Design conditions for the RHR and core spray pumps are as follows:

	RHR	Core Spray
a. Design pressure		
1. Suction	220 psig	125 psig
2. Discharge	500 psig	500 psig
b. Design Temperature	40-360°F	40-212°F

3.9.3.1.12 SLCS Pump

The SLCS pumps are designed and fabricated following the requirements for an ASME Section III, Class 2 component.

The SLCS pumps and motors are functionally tested by pumping demineralized water through a closed test loop.

Design conditions for each SLCS pump include:

a.	Flow rate	43 gpm
b.	Available NPSH	12.9 psi @ 110°F
c.	Maximum operating discharge pressure	1220 psig

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(1) Combine with hydrodynamic loads of Table 3.9-6.

d. Ambient conditions:

1.	Temperature	65°F - 106°F
2.	Relative Humidity	20% - 95%

e. Normal plus upset conditions which control the pump design include:

1.	Design pressure	1400 psig
2.	Design temperature	150°F
3.	OBE	½ of SSE

f. Faulted or emergency conditions include:

1.	Design pressure	1400 psig
2.	Design temperature	150°F
3.	SSE(1)	
	horizontal	1.5 g
	vertical	0.14 g

A summary of loading combinations, stress criteria, and calculated and allowable stresses for the SLCS pump is contained in Table 3.9-6(l).

3.9.3.1.13 <u>MSIVs and MSRVs</u>

The MSIVs and MSRVs are designed in accordance with ASME Section III, NB-3500 for safety Class 1 components.

Load combination, analytical methods, calculated stresses, and allowable limits are shown for the MSRVs and MSIVs in Tables 3.9-6(g) and 3.9-6(h) respectively.

3.9.3.1.14 <u>HPCI Turbine</u>

Although not under the jurisdiction of the ASME Code, the HPCI turbine is designed and fabricated following the basic guidelines for an ASME Section III, Class 2 component.

Operating conditions for the HPCI turbine include:

a. Surveillance Testing - Periodic operation with the reactor pressure at 1000 psia, nominal, and saturated temperature; and the turbine exhaust pressure at 65 psia, peak, and saturated temperature.

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(1) Combine with hydrodynamic loads from Table 3.9-6.

b. Auto-Startup - 30 cycles per year with the reactor pressure at 1150 psia, nominal, and saturated temperature; and turbine exhaust pressure at 65 psia, peak, and saturated temperature.

Design conditions for the HPCI turbine include:

a. Turbine Inlet - 1250 psig at 575°F

b. Turbine Exhaust - 200 psig at 382°F

A summary of loading conditions, stress criteria, and calculated and allowable stresses for the HPCI turbine components is shown in Table 3.9-6(k).

3.9.3.1.15 HPCI Pump

The HPCI pump is designed and fabricated following the requirements for an ASME Section III, Class 2 component.

The HPCI pump is surveillance tested together with the HPCI turbine (Section 3.9.3.1.14). An operational test is performed in which the HPCI pump takes condensate from the CST, and discharges at the design flow rate back to the CST through a closed test loop.

Design conditions for the HPCI pump include:

			REACTOR CONDITION[2,3]		
			1	2	3
a.	Required NPSH (feet)		21	21	21
b.	Total head -	High speed (feet)	3030	850	700
		Low speed (feet)	525	525	525
c.	Constant flow rate (gpm)		5600	5600	3750

d. Normal ambient operating temperature - 60°F to 100°F

e. Normal plus upset conditions which control the pump design include:

1. Design pressure 1500 psig

2. Design temperature 40°F - 140°F

3. OBE[(1)] ½ of SSE

A summary of calculated stresses, allowable stresses, criteria and loading conditions for critical components is provided in Table 3.9-6(t).

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(1) Combine with hydrodynamic loads from Table 3.9-6.

(2) Reactor Conditions 1 and 2 refer to the range of reactor pressures within which the rated performance of the HPCI pump is guaranteed. This range is bounded by 1156 psia (saturation pressure, Condition 1) and 215 psia. Condition 3 refers to the 100 psig pump interlock point, and is the corresponding pump performance obtained from manufacturer's input.

(3) The HPCI design basis is that it must be capable of injecting design rated flow into the reactor vessel at a maximum reactor pressure equal to the lowest SRV nominal setpoint, plus the allowable setpoint tolerance. References 3.9-26 and 3.9-27 re-evaluated HPCI system operation at the maximum reactor pressure of 1205 psig (lowest SRV setpoint, 1170 psig, plus 3% setpoint tolerance). It was concluded that maintaining the currently rated turbine speed of 4190 rpm at 1205 psig would result in a maximum rated flow rate of 5400 gpm. Reference 3.9-27 performed a conservative analysis that demonstrated a reduced flow rate of 5000 gpm between reactor pressures of 1182 and 1205 psig is adequate for all design basis requirements of the HPCI system.

3.9.3.1.16 RWCU System

The requirements of ASME Section III, Class 3 components are used as guidelines in evaluating the RWCU system pump and heat exchangers. The loading combinations, stress criteria, calculated and allowable stresses are summarized in Tables 3.9-6(p) and 3.9-6(c), respectively.

3.9.3.1.17 Non-NSSS ASME Code Constructed Items

The design loading combinations, categorized by plant operating conditions, identified as normal, upset, emergency, or faulted for Non-NSSS ASME code constructed items, are presented in Table 3.9-11.

The design criteria and stress limits associated with each of the plant operating conditions for each type of ASME code constructed items are presented in Tables 3.9-12 through 3.9-18.

The component operating condition is the same as the plant operating condition, except that the emergency or faulted plant condition is considered normal for a pump or valve whose function must be ensured during an emergency or faulted plant condition.

3.9.3.2a NSSS Pump And Valve Operability Assurance

The NSSS active pumps and valves are listed in Table 3.9-19.

Active mechanical equipment classified as seismic Category I is designed to function during the life of the plant, under postulated plant conditions. Equipment with faulted condition functional requirements includes "active" pumps and valves in fluid systems such as the ECCS. ("active" equipment performs a mechanical motion to accomplish a safety function.)

Operability is ensured by satisfying the requirements of the following programs. Safety-related valves are qualified by prototype testing and/or analysis; and safety-related active pumps are qualified by analysis and/or testing with suitable stress limits and nozzle loads. The content of these programs is detailed below.

3.9.3.2a.1 ECCS Pumps

All active pumps are qualified for operability by first being subjected to rigid tests, before and after installation in the plant. The in-shop tests include hydrostatic tests of pressure-retaining parts at 125% of the design pressure; seal leakage tests; and performance tests, while the pump is operated with flow, to determine total developed head, minimum and maximum head, and NPSH requirements. Bearing temperatures (except water-cooled bearings) and vibration levels are also monitored during these operating tests. Both are below specified limits. After the pump is installed in the plant, it undergoes cold hydrotests, functional tests, and the required periodic inservice inspection and operation. These tests demonstrate reliability of the pump for the design life of the plant.

In addition to these tests, the safety-related active pumps are analyzed for operability during a faulted condition to ensure that the pump will not be damaged during the seismic and hydrodynamic event, and that the pump will continue operating despite the faulted loads.

3.9.3.2a.1.1 Analysis of Loading, Stress, and Acceleration Conditions

In order to avoid damage during the faulted plant condition, the stresses caused by the combination of normal operating loads, SSE, and dynamic system loads are limited to the material elastic limit, as indicated in Section 3.9.3.1 and Table 3.9-6. A three-dimensional finite-element model of the pump/motor and supports was developed using the response spectrum method of dynamic analysis. The average membrane stress (δ_m) for the faulted condition loads is maintained at 1.2 S_m, or approximately 0.75 δ_y (δ_y = yield stress); the maximum stress in local fibers (δ_m plus bending stress (δ_b)) is limited to 1.8 S_m, or approximately 1.1 δ_y.

The maximum dynamic nozzle loads were also considered in an analysis of the pump supports to ensure that a system misalignment will not occur.

A dynamic analysis was performed to determine the magnitude of the seismic loading from the applicable floor response spectra. An analysis was made to ensure that faulted condition nozzle loads and dynamic accelerations would not impair the operability of the pumps during or following the faulted conditions. Pump components having a natural frequency above 33 Hz are considered rigid. This frequency is considered sufficiently high to avoid problems with amplification between the component and structure for all seismic areas.

If the natural frequency is below 33 Hz, an analysis is performed to determine the amplified input accelerations necessary to perform the static analysis. The adjusted accelerations are determined using conservatisms contained in the horizontal and the vertical accelerations used for rigid structures. The static analysis is performed using the adjusted accelerations and must meet the same stress limit criterion stated in Table 3.9-6.

These analyses, with the conservative loads stated and with the restrictive stress limits of Table 3.9-6 as allowables, assure that critical parts of the pump are not damaged during the faulted condition, and that, therefore, the reliability of the pump for postfaulted condition operation will not be impaired by a hydrodynamic or seismic event.

3.9.3.2a.1.2 Pump Operation During and Following Faulted Condition Loading

Active pump/motor rotor combinations are designed to rotate at a constant speed under all conditions. Motors are designed to withstand short periods of severe overload. The high rotary inertia in the operating pump rotor and the nature of the random shot-duration loading characteristics of the seismic and hydrodynamic events prevents the rotor from becoming seized. The dynamic loadings cause a slight increase, if any, in the torque (i.e., motor current) necessary to drive the pump at constant design speed. Therefore the pump does not shut down during the faulted condition but operates at the design speed despite the faulted loads.

The functional ability of the active pumps after a faulted condition is assured, since only normal operating loads and steady-state nozzle loads exist at this time. For the active pumps, the faulted condition is greater than the normal condition due to seismic and hydrodynamic loads on the equipment and the increase in nozzle loads from the SSE on the connecting pipe. The SSE event is infrequent and of short duration compared to the design life of the equipment. Because it is demonstrated that the pumps would not be damaged during the faulted condition, the postfaulted condition operating loads are less than the normal plant operating limits. This is assured by requiring that the imposed nozzle loads (steady-state loads) for normal conditions and postfaulted conditions are limited by the normal condition nozzle loads. The postfaulted condition ability of the pumps to function under these applied loads is proven during the normal operating plant conditions.

3.9.3.2a.2 SLCS Pump and Motor Assemblies and RCIC Pump Assembly

These equipment assemblies are small, compact, rigid assemblies, with natural frequencies greater than 100 Hz. With this fact verified, each equipment assembly is dynamically qualified by static analysis only. This static qualification verifies operability under seismic and hydrodynamic conditions, and ensures that structural loading stresses are within code limitations.

3.9.3.2a.3 RCIC Turbine

Refer to Section 3.9.2.2a.2.9 and Table 3.9-6(q).

3.9.3.2a.4 ECCS Motors

Qualification of the Class 1E motors used for the ECCS motors is in compliance with IEEE 323 (1971). The qualification of all motor sizes is based on completion of a type test, followed with review and comparison of design and material details and seismic analysis of production units, ranging from 500 Bhp to 3500 Bhp, with the motor used in the type test. All manufacturing, inspection, and routine tests by motor manufacturer on production units are performed on the test motor.

The type test was performed on a 1250 hp vertical motor, in accordance with IEEE 323 (1971). First normal operation during the design life was simulated; then the motor was subjected to a number of seismic events; and then subjected to the abnormal environmental conditions possible during and after a LOCA. The type test plan was as follows:

a. Thermal aging of the motor electrical insulation system (which is a part of the stator only) was based on extrapolation, in accordance with the temperature life of the characteristic curve from IEEE 275 (1966) for the insulation type used on the ECCS motors. The amount of aging equals the total estimated operation days of maximum insulation surface temperature.

b. Radiation aging of the motor electrical insulation equals the maximum estimated integrated gamma dose during normal and abnormal conditions.

c. The normal induced current vibration effect on the insulation system was simulated by a 1.5 g horizontal vibration acceleration for one hour at current frequency.

d. Motor bearings were selected and their operating life established on the basis of bearing manufacturer's tests and operating data using the loads calculated to act on the bearings.

e. The dynamic load-deflection analysis on the rotor-shaft, performed to ensure adequate rotation clearance, is verified by static loading and deflection of the rotor for the type test motor.

f. Dynamic loading aging and testing were performed on a biaxial test table in accordance with IEEE 344 (1971). During this type test, the shake table was activated, simulating the maximum design limit of the SSE and hydrodynamic loads with motor starts and operation combination, as may possibly occur during the life of the plant. Refer to Section 3.10.

g. An environmental test simulating a LOCA condition was performed for 100 days with the test motor fully loaded, simulating pump operation. The test consists of startup and six hours of operation at 212°F ambient temperature, and a 100% steam environment. After 1 hour standstill in the same environment, another startup and operation of the test motor was followed by sufficient operation at high humidity and temperature. This was based on extrapolation in accordance with the temperature life characteristic curve from IEEE 275 (1966) for the insulation type used on the ECCS motors.

3.9.3.2a.5 HPCI Pump

Operability of the HPCI pump assembly is demonstrated by a combination of analytical stress calculations, pump manufacturer's operating experience, and testing. The stress definitions and the allowable stress criteria are based on the ASME Section III. The code is directly applicable to the stamped pressure boundary components of the pump.

The witnessed hydrostatic and performance tests, as performed at the pump manufacturer's plant, demonstrate that the pump as designed meets the ASME Code requirements and the parameters of the design specification.

A dynamic analysis was performed by the pump manufacturer. A static analysis was performed for dead weight and nozzle loads. Dynamic analysis was used for the SSE loading condition.

Rotating parts are verified by analysis to ensure no contact with the stationary parts, except at engineered wear points, therefore, continued operation during and after an SSE is assured because the calculated stresses and deformations are less than the prescribed limits.

3.9.3.2a.6 NSSS Valves

3.9.3.2a.6.1 Class 1 Active Valves

The Class 1 active valves are the MSIVs and the MSRVs and SLCS valves. Each of these valves is designed to perform its mechanical motion in conjunction with a design basis accident. Dynamic qualification for operability is unique for each valve type; therefore, each method of qualification is detailed individually below.

3.9.3.2a.6.1.1 Main Steam Isolation Valve

The MSIVs were evaluated for operability during a dynamic loading event by analysis and test as follows:

a. The valve body was designed in accordance with ASME Section III, Class 1, which limits deformations in the operating area of the valve body to those within the elastic limit of the material by limiting pressure and pipe reaction input loads (including seismic and hydrodynamic), thereby ensuring no interference with valve operability.

b. The complete valve topworks, including the bonnet and a simulated stem of a similar design MSIV, was dynamically tested for upset loads and faulted load. The test sample was subjected to 30 seconds of bi-directional random frequency input for each test event. There were a total of 10 upset test events and 2 faulted test events. Valve closure was simulated half way through the testing duration during each event. After the complete test program, there was no significant change to the valve closure rate.

To ensure that design limits were not exceeded for both piping input loads and actuator dynamic loads, the MSIV was mathematically modeled in the main steam line system analysis. The valve dynamic characteristics were modeled in the overall steam line analysis. Pipe anchors and restraints are applied as required to limit pipe system resonance frequencies and amplified accelerations to within acceptable limits for the MSIVs. Details concerning the analysis of these valves is given in Table 3.9-6(h).

The MSIV operability during LOCA conditions is demonstrated as defined in report APED-5750 (March 1969). The test specimen is a 20 inch valve representative in design to the MSIVs. Operability during seismic acceleration is addressed in Section 3.9.2.2a.

3.9.3.2a.6.1.2 Main Steam Relief Valves

The MSRVs were evaluated for operability during seismic and hydrodynamic events. Structural integrity of the configuration during a seismic event was demonstrated by both code analysis and dynamic testing.

a. Each valve was designed for maximum moments which may be imposed on it when installed in service. These moments result from the dead weight, hydrodynamic, and seismic loading of both the valve and the connecting pipe, thermal expansion of the connecting pipe, and reaction forces from valve discharge.

b. Dynamic tests were performed on the LGS MSRV configuration. The tests validated the design static analysis and determined that the equipment remains functional during application of the specified "g" loads.

A mathematical model of this valve was included in the overall main steam line system analysis, along with the MSIVs, to ensure that design limits were not exceeded.

The MSRV analytical qualification results are shown in Table 3.9-6(g).

3.9.3.2a.6.1.3 Explosive Valves

The SLCS explosive valves are qualified to IEEE 344 (1975). The generic qualification test demonstrated the absence of natural frequencies below 35 Hz and the ability to remain operable after the application of horizontal dynamic loading equivalent to 6.5 g and a vertical dynamic loading equivalent to 4.5 g at 33 Hz. In addition, analysis shows that there are no natural frequencies below 100 Hz.

3.9.3.2a.6.2 Class 2 and 3 Active Valves

3.9.3.2a.6.2.1 Gate/Globe Valves

Class 2 active gate/globe valves provided by GE include five RHR valves, one HPCI valve, one RCIC valve, and three core spray valves; all valves are motor-operated. There are no Class 3 active valves in the GE scope of supply. The gate/globe valves are generically qualified by testing valves that are typical of the valves supplied by GE. Operability is ensured by testing under the maximum capability static load which envelopes the static design basis load. These tests ensure operability during and after the design basis load. The actuators are qualified to IEEE 382 (1972), up to levels that exceed the design loadings.

3.9.3.2a.6.2.2 Check Valves

GE provides one swing-check valve and one stop-check valve which are Class 2 active in each of the HPCI and RCIC systems. In addition, GE provides six Class 2 check valves for the RHR system and two Class 2 check valves for the core spray system. Operability of check valves is assured by performing design calculations and by providing sufficient structural margins so that movement of the disc/hinge pin is not impaired under any loading conditions. There are no Class 3 active valves in the GE scope of supply.

3.9.3.2b Non-NSSS Pump and Valve Operability Assurance

3.9.3.2b.1 Pumps

The following pumps are active pumps:

a. Diesel oil transfer pumps

b. RHRSW pumps

c. ESW pumps

d. Control room chilled water pumps

e. Safeguard piping fill pumps.

The safeguard piping fill pumps are Class 2. All the remaining pumps are Class 3.

Safety-related active pumps are subjected to stringent tests both prior to and after installation in the plant. The in-shop tests include: hydrostatic tests of pressure-retaining parts to 150% of the design pressure; seal leakage tests at the same pressure used in the hydrostatic tests; and performance tests which are conducted while the pump is operated with flow to determine total developed head, minimum and maximum head, NPSH requirements, and other pump/motor properties. Bearing temperatures and vibration levels are also monitored during these operating tests. Both are shown to be within the limits specified by the manufacturer. After the pump is installed at the plant, it undergoes startup tests and required inservice inspection and operation.

In addition to these tests, the active pumps are qualified for operation during and after a faulted condition. That is, safety-related active pumps are qualified for operability during an SSE condition by assuring that the pump will not be damaged during the seismic event, and that the pump will continue operating despite the SSE loads. (Refer to 3A.7.1.7 for pumps subjected to hydrodynamic loads.)

In order to meet the first criterion, that the pump will not be damaged during the seismic event, the pump manufacturer is required to determine by test or dynamic analysis whether the lowest natural frequency of the pump is greater than 33 Hz. The pump, when having a natural frequency above 33 Hz, is considered essentially rigid. This frequency is considered sufficiently high to avoid problems with amplification between the component and structure for all seismic areas. A static shaft deflection analysis of the rotor is performed with the conservative SSE accelerations of 1.5 times the applicable floor acceleration. The deflections determined from the static shaft analysis are compared to the allowable rotor clearances.

If the lowest natural frequency is found to be below 33 hertz, the equipment is considered flexible. If flexible, the equipment is analyzed using the response spectrum modal analysis technique. The frequencies and mode shapes are determined in the vertical and horizontal directions. The loads due to the excitation of each mode and the loads due to the accelerations in the three orthogonal directions are added using the square root of the sum of the squares method. Coupling effects are included in the mathematical model.

In order to avoid damage to the pumps during the faulted plant condition, the stresses caused by the combination of normal operating loads, SSE, and dynamic system loads are limited. The maximum seismic nozzle loads are also considered in an analysis of the pump supports to assure that a system misalignment cannot occur. Performance of these analyses, based upon conservative loads and restrictive stress limits, assures that critical parts of the pump will not be damaged during the faulted condition and, therefore, that the reliability of the pump for postfaulted condition operation will not be impaired by the seismic events.

The second criterion necessary to assure operability is that the pump will function throughout the SSE. The pump/motor combination is designed to rotate at a constant speed under all conditions unless the rotor becomes completely seized, i.e., with no rotation. Typically, the rotor can be seized five full seconds before a circuit breaker, used to prevent damage to the motor, opens to stop the pump. However, the high rotary inertia in the operating pump rotor and the nature of the random, short duration loading characteristics of the seismic event prevent the rotor from becoming seized. Actually, the seismic loadings cause only a slight increase, if any, in the torque, i.e., motor current, necessary to drive the pump at the constant design speed. Therefore, the pump will not shut down during the SSE and will operate at the design speed despite the SSE loads.

To complete the seismic qualification procedures, the pump motor and all appurtenances vital to the operation of the pump are independently qualified for operation during the maximum seismic event as discussed in Section 3.10.

From this regimen, it is concluded that the safety-related pump/motor assemblies will not be damaged and will continue to operate under SSE loadings and, therefore, will perform their intended functions. These proposed requirements take into account the complex characteristics of the pump and are sufficient to demonstrate and assure the seismic operability of the active pumps.

The functional ability of active pumps after a faulted condition is assured since only operating loads and steady-state nozzle loads exist. Since it is demonstrated that the pumps would not be damaged during the faulted condition, the postfaulted operating loads will be limited to the normal plant operating loads. This is assured by requiring that the imposed nozzle loads (steady-state loads) for normal conditions and postfaulted conditions are limited by the magnitudes of the normal condition nozzle loads. The postfaulted ability of the pumps to function under these applied loads is proved during the normal operating plant conditions for active pumps.

3.9.3.2b.2 Valves

The active valves are tabulated in Reference 3.9-25. Those active valves which are supplied by the NSSS vendor are identified. See Section 3.9.3.2a for a discussion of operability assurance of these valves.

Safety-related active valves are subjected to a series of stringent tests prior to service, and during the plant life. Before installation, the following tests are performed: the shell hydrostatic test, in accordance with ASME Section III requirements; back-seat and main-seat leakage tests; the disc hydrostatic test; functional tests which verify that the valve opens and closes within the specified time limits; and the operability qualification of motor operators for the environmental conditions over the installed life (i.e., aging, radiation, accident environment simulation, etc.), in accordance with IEEE 382 (1972). After installation, cold hydrostatic tests, functional tests (in accordance with the requirements of Chapter 14), and periodic inservice operation (in accordance with the requirements of Chapter 16) are performed to verify and ensure the functional ability of the valve.

The valves are designed using either stress analyses, or the pressure-containing minimum wall thickness requirements. For all active valves with extended topworks, an analysis is also performed for static-equivalent SSE loads applied at the extended structure's center of gravity. The maximum stress limits allowed in the analyses demonstrate structural integrity, and are equal to the limits recommended by the ASME for the particular ASME class of valve analyzed. Limits for each of the loading combinations are presented in Tables 3.9-13 and 3.9-18.

In addition to these tests and analyses, a representative valve of each design type is tested to verify operability during a simulated seismic event by demonstrating operational capabilities within the specified limits. The qualification testing procedures are described below.

The valve is mounted in a manner that conservatively represents typical valve installations. The valve unit includes the actuator and all appurtenances normally attached to the valve in service. The operability of the valve during an SSE is demonstrated by satisfying the following criteria. (Refer to Appendix 3A.7.1.7 for valves subjected to hydrodynamic loads.)

a. All the active valves with topworks are basically designed to have a first natural frequency greater than 33 Hz. This may be shown by suitable testing or analysis. Regardless of value, the first natural frequency of the topworks is modeled into the piping analysis for determination of maximum accelerations.

b. While in the shop and installed in a suitable test rig, the extended topworks of the valve are subjected to a statically applied equivalent seismic load. The load (the specified g-force times the weight of the topworks) is applied at the center of gravity of the topworks, in the direction of the weakest axis of the yoke. The design pressure of the valve is simultaneously applied to the valve during the static load tests.

c. The valve is then operated at the minimum specified actuation supply voltage or pressure, with the equivalent seismic static load applied. The valve must perform its safety-related function within the specified operating time limits.

d. Motor operators are qualified as operable during and after the SSE prior to their installation on the valve, as discussed in Section 3.10.2.2.

The equivalent seismic static load, which is used for the static valve qualification, is the maximum load which the valve is designed to withstand. The piping designer must maintain the motor operator accelerations to these levels.

The valve is leak tested following the test described above, to show that the valve has not been damaged. The leak rates must not exceed the original allowable leakage rate specified for the valve.

The above factory testing program applies only to valves with overhanging structures, e.g., the motor operator or air actuator assembly. The testing is conducted on a representative number of valves. According to the size and pressure rating, valves from each of the primary safety-related design types, e.g., motor- operated gate valves, are tested. Valves that cover the range of sizes in service are qualified by tests, and the results are used to qualify all valves within the intermediate range of sizes, as shown in Table 3.9-20. Stress analyses are used to support the interpolation. Because of their simple characteristics, check and other compact valves are not adversely affected by seismic acceleration. Check valves have no extended structures to distort the valves and cause malfunctions. Check valve discs are designed to allow sufficient clearance around the disc to prevent binding or interference due to distortions from nozzle or other imposed loads. They are qualified by a combination of the following factory tests and analysis:

a. Stress analysis of critical areas and parts for SSE loads in accordance with the allowable specified in Tables 3.9-13 and 3.9-18

b. In-shop hydrostatic test

c. In-shop seat leakage test

d. Periodic valve exercise and inspection to ensure the functional ability of the valve in accordance with the requirements of Chapter 16

A study was also performed that considered a feedwater pipe rupture outside containment to assure that the feedwater isolation check valves can perform their function following a postulated pipe break of the feedwater line outside containment. Analysis of a full circumferential break showed that the check valve inside containment (one of 3 feedwater PCIV's that are check valves) would close rapidly because of flow reversal in the pipe between the reactor and break location just outside containment. This results in a pressure surge of 2780 psia between the valve and reactor. It was determined that the piping and valves could adequately sustain this pressure.

To determine the capability of the check valve seat to withstand the initial impact caused by rapid closure and to sustain the pressure surge that follows, an estimate of check valve seat energy absorption capability was made for a postulated feedwater pipe break accident occurring upstream of the containment isolation valves. The basis for the seat stress calculation was for a valve disk closing velocity of 100 rad/sec and a pressure break in the pipe of 2780 psia.

The valve seat was assumed to consist of an assembly of six discrete, bilinear, elastic-plastic elements. The analysis assumed that the disc kinetic energy at impact equals work done in terms of seat under load, or area under the seat load-displacement curve. Valve seat yield strength was based on its being stressed to 50% of yield at a design pressure of 2132 psi. The load-displacement curve was constructed using Roark's stiffness equations for an annular plate loaded at the inner radius and fixed at the outer radius. Failure was assumed to occur at a ductility ratio of 30.

The analysis indicated that all of the seat elements reach yield, but none reach ultimate strain or fail. Effects not included that are believed to make the analysis conservative are:

a. No credit was taken for disc deformation and energy absorption

b. No credit was taken for hinge deformation and energy absorption

c. No credit was taken for valve body deformation and energy absorption

d. Hinge friction was omitted

e. No strain hardening or rate of strain effects in the seat were included.

The water hammer effects on the closed seat were determined. It was shown that the natural frequency of the combined valve seat stiffness and disc mass was much larger than the frequency of the pressure pulse, so that the effective pressure that the seat must withstand is only the peak pressure in the water hammer surge. The seat is able to withstand this maximum pressure.

Results of this analysis show that, although the conditions of a hypothetical pipe rupture on the feedwater check valves are severe, the valves should remain together at impact and are capable of withstanding 2800 psi. The valve seat should yield at disc impact but not fail, and the water hammer pressure pulse following closure will not cause failure.

An additional confirmatory analysis was also conducted which used a simplified model of single-phase, liquid flow from the reactor vessel to the pipe break, with the check valve disk being closed by drag forces.

Results of this simplified analysis indicate that the check valve disk closes in approximately 70 msec, with a closing velocity at impact of approximately 65 rad/sec. The peak pressure at the closed disk is estimated to be 2157 psi for a finite valve closure time of 70 msec. The precise pressure at valve closure cannot be predicted rigorously by the simplified method used in this study.

The results of this study confirmed the validity of the original design analysis and are consistent with the results of the analysis done for SSES and in particular for the Atwood Morrill check valves.

Operability testing is also not performed for relief valves. Due to the particular, simple characteristics of these SRVs, they are qualified by a combination of the following tests and analyses:

a. Stress analysis, including seismic loads where applicable

b. In-shop hydrostatic test

c. In-shop seat leakage test

d. Performance tests

e. Periodic in situ valve inspection as applicable and periodic valve removal, refurbishment, performance testing and reinstallation

The above testing and analysis is sufficient to ensure the functional capability of the valve.

During a seismic event, it is anticipated that the seismic acceleration imposed upon the valve may cause it to open momentarily and discharge under system conditions that otherwise would not result in valve opening, but this is considered to be of no real safety or other consequence.

Using the methods described, all the safety-related active valves in the systems are qualified for operability during a seismic event. These methods conservatively simulate the seismic event and ensure that the active valves perform their safety-related functions when necessary.

3.9.3.3 Design and Installation of Pressure Relief Devices

3.9.3.3.1 Main Steam Relief Valves (NSSS)

Lifting of an MSRV results in a transient that produces momentary unbalanced forces acting on the discharge piping system from the time the MSRV opens, until a steady discharge flow from the RPV to the suppression pool is established. This period includes clearing the water slug from the end of the discharge piping submerged in the suppression pool. Pressure waves traveling through the discharge piping following the relatively rapid opening of the MSRV cause the MSRV discharge piping to vibrate. This in turn produces forces that act on the main steam piping.

The analysis of the relief valve discharge transient consists of a step-wise time history solution of the fluid flow equation, to generate a time history of the fluid properties at numerous locations along the pipe. The fluid transient properties are calculated based on the maximum set pressure specified in the steam system specification, and the value of ASME flow rating increased by a factor to account for the conservative method of establishing the rating. Simultaneous discharge of all valves is assumed in the analysis, because simultaneous discharge is considered to induce maximum stress in the piping. Reaction loads on the pipe are determined at each location corresponding to the position of an elbow. These loads are composed of pressure times area, momentum change, and fluid friction terms. Figure 3.9-3 shows a set of fluid property and pipe section load transients typical of those produced by relief valve discharge.

The method of analysis applied to determine piping system response to MSRV operation is time history integration. The forces are applied at locations on the piping system where the fluid flow changes direction, thus causing momentary reactions. The resulting loads on the MSRV, the main steam line, and the discharge piping are combined with loads due to other effects, as specified in Section 3.9.3.1. The code stress limits corresponding to load combination classifications of normal, upset, emergency, and faulted, are applied to the main steam lines and MSRV discharge piping.

In addition, a series of water discharge tests of SRVs used in BWRs was conducted to demonstrate the operational adequacy of the SRV and SRV discharge piping integrity for expected operating conditions for transients and accidents. The tests were performed to satisfy requirements of NUREG-0737, Item II.D.1.

The tests were run at the Wyle Laboratories test facility in Huntsville, Alabama. The facility included a steam and water supply system, the test SRV mounted on a representative steam line, and a representative SRV discharge line routed to a pool of water. Opening and closing of the SRVs were monitored. Fluid conditions and flows were measured, as were strains, accelerations, temperatures, and pressures in the SRV and associated piping.

The water discharge test conditions simulated the alternate shutdown cooling condition, which is an operating condition which is considered in the design evaluation of many BWR plants. The results show that all of the tested SRVs opened and closed on command for all water tests. The measured SRV discharge line loads for water discharge were significantly less than those for the high pressure steam discharge condition for which the piping is designed. LGS specific analyses have been performed (Reference 3.9-22) to establish RPV temperature and pressure conditions where initiation of alternate shutdown cooling will result in acceptable SRV discharge line loads. Alternate shutdown cooling is manually initiated only.

The tests and analyses as described in NEDE-24988-P (Reference 3.9-15) verify the adequacy of SRV operation and the integrity of the SRV discharge piping under expected liquid discharge conditions, and satisfy requirements of NUREG-0737, Item II.D.1.

A basis for concluding that the test results presented in NEDE-24988-P on SRV testing are applicable to LGS is described in the following paragraphs:

a. The SRV discharge piping configuration at LGS uses a T-quencher at the discharge pipe exit. The average length of the 14 SRV discharge line is 132 feet of 12 inch diameter pipe, and the submergence length in the suppression pool is approximately 18'-6". The SRV test program used a ramshead at the discharge pipe exit, a pipe length of 112 feet, a diameter of 10 inches, and a submergence length of approximately 13 feet. Loads on valve internals in the LGS configuration are within acceptable limits for the following reasons:

1. No dynamic mechanical load originating at the T-quencher is transmitted to the valve in the LGS configuration because there is at least one anchor point between the valve and the T-quencher.

2. The length of the first segment of piping downstream of the SRV in the test facility was selected to result in the test program having a bounding dynamic mechanical load on the valve. The LGS SRV discharge line piping configuration differs from the test facility in that the first line segment does not terminate in a 90° elbow, and the pipe size increases in the first segment. An assessment of the LGS configuration has confirmed that the mechanical loads imposed on the LGS valves by the low pressure water flow are enveloped by the high pressure steam loads.

3. Dynamic hydraulic loads (back pressure) are experienced by the valve internals in the LGS configuration. The back pressure loads may be either transient back pressures occurring during valve actuation or steady-state back pressures occurring during steady-state flow following valve actuation.

   The key parameters affecting the transient back pressures are the fluid pressure upstream of the valve, the valve opening time, the fluid inertia in the submerged SRV discharge line 0, and the SRV discharge line air volume. Transient back pressures increase with higher upstream pressure, shorter valve opening times, greater line submergence, and smaller SRV discharge line air volume. An evaluation of these differences has confirmed that the test facility and the LGS configuration have comparable back pressures. The maximum transient back pressure occurs with high pressure steam flow conditions. The transient back pressure for the alternate shutdown cooling mode of operation is always much less than the design for steam flow conditions because of the lower upstream pressure and the longer valve opening time.

   The steady-state back pressure in the test program was maximized by using an orifice plate in the SRV discharge line above the water level and before the ramshead. The orifice was sized to produce a back pressure greater than that calculated for any of the LGS SRV discharge lines.

An additional consideration in the selection of the ramshead for the test facility was to allow more direct measurement of the thrust load in the final pipe segment. The use of a T-quencher in the test program would have required quencher supports that would unnecessarily obscure accurate measurement of the pipe thrust loads.

The differences in the line configuration between the LGS plant and the test program as discussed above result in loads on the LGS valve internals that are within acceptable limits.

b. The LGS SRV discharge lines are supported by a combination of snubbers, rigid supports, and spring hangers. These supports were designed to accommodate combinations of loads resulting from piping, dead weight, thermal conditions, seismic and suppression pool hydrodynamic events, and a high pressure steam discharge transient. Each SRV discharge line at LGS has 2 to 5 spring hangers, all of which are located in the drywell. The test facility configuration utilized no spring hangers.

The dynamic load effects on the piping and supports of the test facility due to the water discharge events (the alternate shutdown cooling mode) were found to be significantly lower than corresponding loads resulting from the high pressure steam discharge event. As stated in Reference 3.9-15, this finding is considered generic to all BWRs because the test facility was designed to be prototypical of the features pertinent to this issue. Furthermore, assessment of a typical LGS SRV discharge line configuration has confirmed the applicability of the generic statement to LGS. LGS specific analysis for acceptable alternate shutdown cooling initiating conditions has been performed (Reference 3.9-22).

During the water discharge transient, there will be significantly lower dynamic loads acting on the snubbers and rigid supports than during the steam discharge transient. This will more than offset the small increase in the dead load on these supports due to the weight of the water during the alternate shutdown cooling mode of operation. Therefore, design adequacy of the snubbers and rigid supports is assured because they are designed for the larger steam discharge transient loads.

The design adequacy of the spring hangers with respect to the increased dead load due to the weight of the water during the liquid discharge transient has been addressed. As was discussed with respect to snubbers and rigid supports, the dynamic loads resulting from liquid discharge during the alternate shutdown cooling mode of operation are significantly lower than those from the high pressure steam discharge. The spring hangers have been reviewed for the deflections resulting from the steam discharge dynamic event and were found to be acceptable. In addition, the spring hangers have been evaluated for the increased dead load due to a water-filled condition. Both the spring hangers and piping stresses were acceptable. Furthermore, the effects of the water dead weight load does not affect the ability of SRVs to open to establish the alternate shutdown cooling path because the loads occur in the SRV discharge line only after valve opening.

c. The purpose of the SRV test program was to demonstrate that the SRV will open and reclose under all expected flow conditions. The expected valve operating conditions were determined through the use of analyses of accidents and anticipated operational occurrences referenced in Regulatory Guide 1.70 (Rev 2). Single failures were applied to these analyses so that the dynamic forces on the SRVs would be maximized. Test pressures were the highest predicted by conventional safety analysis procedures. The BWROG, in their enclosure to the September 17, 1980 letter from D.B. Waters (BWROG) to R.H. Vollmer (NRC), identified 13 events that may result in liquid or two-phase SRV inlet flow that would maximize the dynamic forces on the SRV. These events were identified by evaluating the initial events described in Regulatory Guide 1.70 (Rev 2), with and without the additional conservatism of a single active component failure or operator error postulated in the event sequence. It was concluded from this evaluation that the alternate shutdown cooling mode is the only expected event that will result in liquid at the valve inlet. Consequently, this was the event simulated in the SRV test program. This conclusion and the test results applicable to LGS are discussed below.

The 13 events and the plant specific features that mitigate these events are summarized in Table 3.9-32. Of these 13 events, only nine are applicable to LGS because of its design and specific plant configuration. Four events (5, 6, 10 and 13) are not applicable to LGS for the reasons listed below:

1. Events 5 and 10 are not applicable because LGS does not have a high pressure core spray system.

2. Event 6 is not applicable because LGS does not have RCIC head sprays.

3. Event 13 is not applicable because large breaks will not be isolated at LGS.

For the nine remaining events, the LGS specific features, such as trip logic, power supplies, instrument line configuration, alarms and operator actions, have been compared to the base case analysis presented in the BWROG submittal of September 17, 1980. The comparison has demonstrated that, in each case, the base case analysis is applicable to LGS because the base case analysis does not include any plant features that are not already present in the LGS design. For these events, Table 3.9-32 demonstrates that the LGS specific features are included in the base case analysis presented in the BWROG submittal of September 17, 1980. It is seen from Table 3.9-32 that all plant features assumed in the event evaluation are also existing features in the LGS plant. All features included in this base case analysis are similar to plant features in the LGS design. Furthermore, the time available for operator action is expected to be longer in the LGS plant than in the base case analysis for each case where operator action is required due to the conservative nature of the base case analysis.

Event 7, the alternate shutdown cooling mode of operation, is the only expected event that will result in liquid or two-phase fluid at the SRV inlet. Consequently, this event was simulated in the BWR SRV test program. In LGS, the event involves flow of subcooled water (approximately 31°F subcooled) at a pressure of approximately 156 psig. The SRV inlet fluid conditions tested in the BWROG SRV test program, as documented in Reference 3.9-15, are 15°F to 50°F subcooled liquid at 20 psig to 250 psig. These fluid conditions envelope the conditions expected to occur at LGS in the alternate shutdown cooling mode of operation.

As discussed above, the BWROG evaluated transients including single active failures that would maximize the dynamic forces on the SRVs. As a result of this evaluation, the alternate shutdown cooling mode is the only expected event involving liquid or two-phase flow. Consequently, this event was tested in the BWR SRV test program. The fluid conditions and flow conditions tested in the BWROG test program conservatively envelope the LGS plant specific fluid conditions expected for the alternate shutdown cooling mode of operation (Reference 3.9-22).

d. The flow coefficient, C_v, for the Target Rock SRV used in LGS was determined in the generic SRV test program (Reference 3.9-15). The average flow coefficient calculated from the test results for the Target Rock valve is reported in table 5.2-1 of Reference 3.9-15. This test value has been used by the licensee to confirm that the liquid discharge flow capacity of the LGS SRVs will be sufficient to remove core decay heat when injected into the RPV in the alternate shutdown cooling mode. The C_v of the valve determined in the SRV test demonstrates that the LGS SRVs are capable of returning sufficient flow to the suppression pool to accommodate injection by the RHR or core spray pump.

If it were necessary to place the LGS plant in the alternate shutdown cooling mode, the operator would ensure that adequate core cooling was being provided by monitoring the following parameters: RHR or core spray flow rate, reactor vessel pressure, and reactor vessel temperature.

The flow coefficient for the Target Rock valve reported in Reference 3.9-15 was determined from the SRV flow rate when the valve inlet was pressurized to approximately 250 psig. The valve flow rate was measured with the supply line flow venturi upstream of the steam chest. The C_v for the valve was calculated using the nominal measured pressure differential between the valve inlet (steam chest) and 3 feet downstream of the valve and the corresponding measured flow rate. Furthermore, the test conditions and test configuration were representative of LGS plant conditions for the alternate shutdown cooling mode, e.g., pressure upstream of the valve, fluid temperature, friction losses, and liquid flow rate. Therefore, the reported C_v values are appropriate for application to the LGS plant.

3.9.3.3.2 Design and Installation Details for Mounting of Pressure Relief Devices in ASME Class 1, 2, and 3 Systems (Non-NSSS)

The design of the pressure-relieving devices can be grouped into two categories: open discharge and closed discharge.

a. Open Discharge

There are no open discharge pressure-relieving devices with limited runs of discharge piping mounted on ASME Code Class 1, 2, and 3 systems.

b. Closed Discharge

A closed discharge system is characterized by piping between the valve and a tank, or some other terminal end. Under steady-state conditions, there are no net unbalanced forces. The initial transient response and resulting stresses are determined by using either a time history computer solution, or a conservative equivalent static solution. In calculating initial transient forces, pressure and momentum terms are included. Water slug effects are also considered.

Time history dynamic analysis is performed for the discharge piping and its supports. The effect of the loading on the header is also considered. The design load combinations for a given transient are shown in Table 3.9-11, and the design criteria and stress limits are shown in Tables 3.9-12 and 3.9-16.

3.9.3.4 Component Supports Furnished with the NSSS

3.9.3.4.1 Piping

Hangers are designed in accordance with ANSI B31.7. In general, the load combinations for the various operating conditions correspond to those used to design the supported pipe. Design transient cyclic data are not applicable to hangers because no fatigue evaluation is necessary to meet the code requirements. All hangers are designed, fabricated, and assembled so that they cannot become disengaged by the movement of the supported pipe or equipment after they are installed. The design load on hangers is the load caused by dead weight. The hangers are calibrated to ensure that they support the design load at both their hot and cold load settings. Hangers provide a specified down travel and up travel in excess of the specified thermal movement. Visual inspection and acceptance of pipe supports are performed in accordance with NCIG-01 requirements (Reference 3.9-10).

For pipe supports, reactions produced by primary and secondary pipe loads are categorized as primary. The primary and secondary loads are summed and compared to the load rating to ensure that the rating is not exceeded. Because no distinction is made between primary and secondary loads, and load rated components are designed to primary limits or qualified by testing, the supports meet primary stress criteria for primary and secondary loads combined.

Required load capacity and snubber location for NSSS piping systems are determined by GE as a part of the NSSS piping system design and analysis scope. However, design, installation and inspection of snubbers are included in the non-NSSS scope (Section 3.9.3.5).

The entire piping system, including valves and the suspension system between anchor points, is mathematically modeled for complete structural analysis. In the mathematical model, the snubbers are modeled as springs with a given stiffness depending on the snubber size. The analysis determines the forces and moments acting on each component and the forces acting on the snubbers due to all dynamic loading conditions defined in the piping design specification. The design load on snubbers includes those loads caused by seismic forces (OBE and SSE), system anchor movements, and reaction forces caused by relief valve discharge, turbine stop valve closure, and other hydrodynamic forces (SRV, LOCA, annulus pressurization).

The assessment of all affected piping including their supports and structural modifications necessitated by reconciliation of the suppression pool hydrodynamic loads have been completed.

The snubber location and loading direction are decided by estimation so that the stresses in the piping system have acceptable values. The snubber locations and direction are refined by performing the computer analysis on the piping system as described above.

The spring constant required by the suspension design specification for a given load capacity snubber is compared against the spring constant used in the piping system model. If the spring constants are not in agreement, they are brought into agreement, and the system analysis is redone to confirm the snubber loads.

If the stiffness of the backup structure for the snubber is not large compared to that of the snubbers, the reduced effective snubber stiffness (spring constant) is used in the analysis to account for backup structure flexibility.

Snubber design is discussed in Section 3.9.3.5.2.

3.9.3.4.2 NSSS Floor-Mounted Equipment (Pumps, Heat Exchangers, and RCIC and HPCI Turbines)

The ECCS pumps, RCIC and SLCS pumps, RHR heat exchanger, and RCIC and HPCI turbines are analyzed to verify the adequacy of their support structure under various plant operating conditions. In all cases, the stress loads in the critical support areas are within ASME Code allowables. The assessment of all affected equipment including their supports and structural modifications necessitated by reconciliation of the suppression pool hydrodynamic loads have been completed. The loading conditions, stress criteria, and allowable and calculated stresses in the critical support areas are summarized in Tables 3.9-6(k), 3.9-6(l), 3.9-6(m), 3.9-6(n), 3.9-6(o), 3.9-6(q), 3.9-6(r), and 3.9-6(t).

3.9.3.4.3 Supports for ASME Code Class 1, 2, and 3 Active Components

ASME Code Class 1, 2, and 3 active components are either pumps or valves. Because valves are supported by piping and are not tied to building structures, pipe design criteria govern.

Seismic Category I active pump supports are qualified for seismic and hydrodynamic loads by testing when the pump supports along with the pumps are fulfilling the following conditions:

a. Simulate actual mounting conditions

b. Simulate all static and dynamic loadings on the pump

c. Monitor pump operability during testing

d. Normal operation of the pump during and after the test indicates that the supports are adequate; any deflection or deformation of the pump supports that precludes the operability of the pump is not accepted.

e. Supports are inspected for structural integrity after the test; any cracking or permanent deformation is not accepted.

Seismic and hydrodynamic qualification of component supports by analysis is generally accomplished as follows:

a. Stresses at all support elements and parts such as pump holddown, baseplate holddown bolts, pump support pads, pump pedestal, and foundation are checked to be within the allowable limits as specified in ASME Subsection NF.

b. For normal and upset plant conditions, the deflections and deformations of the supports are assured to be within the elastic limits and not exceed the values permitted by the designer based on design verification tests to ensure the operability of the pumps.

c. For emergency and faulted plant conditions, the deformations must not exceed the values permitted by the designer to ensure operability of the pumps.

3.9.3.4.4 RPV Support Skirt

The permissible compressive load on the reactor vessel support skirt cylinder (modeled as plate and shell type component support) is limited by the design specification to 90% of the load which produces yield stress, divided by the safety factor for the condition being evaluated. The effects of fabrication and operational eccentricity are included. The safety factor for faulted conditions is 1.125.

An analysis of RPV support skirt buckling for faulted conditions shows that the support skirt has the capability to meet ASME Section III, paragraph F-1370(c), faulted condition limits of 0.67 times the critical buckling strength of the support at temperature. The faulted condition analyzed included the compressive loads due to the design basis maximum earthquake, the overturning moments and shears due to the jet reaction load resulting from a severed pipe, and the compressive effects on the support skirt due to the thermal and pressure expansion of the reactor vessel.

Subsequently, based on currently defined faulted condition loads, the maximum compressive stress in the support skirt including axial and bending loads in less than the faulted condition allowable of appendix F (paragraph F-1325) determined by the methods of ASME Section III, NB 3133.6. (Axial loads include weight, fuel interaction, seismic SSE, and the maximum of condensation oscillation, chugging and vent clearing due to a LOCA. Bending loads include seismic SSE and jet reaction, jet impingement, and annulus pressurization due to a LOCA.) The loading criteria, stress criteria, calculated and allowable stresses are summarized in Table 3.9-6(f).

3.9.3.4.5 Bolting Stress Limits (NSSS)

3.9.3.4.5.1 Component Support Bolting

The support bolting of the RWCU pump that is not essential to safety is designed for the effects of pipe load and SSE load to the requirements of ASME Section III, Appendix XVII. The stress limits of 0.41 S_y for tension and 0.15 S_y for shear are used.

For RCIC/SLCS pumps and RCIC turbine, the equipment-to-base plate bolting satisfies the following design criteria: For normal and upset conditions, 1.0S is used for primary membrane and 1.5S for primary membrane plus bending, where (S) is the allowable stress limit from ASME Section III, Appendix I, table I-7.3. For emergency and faulted conditions, stresses shall be less than 1.2 times the allowable limits for "normal and upset" given above.

There are no flange-type connections in pipe mounted component supports.

3.9.3.4.5.2 Piping Supports and Pipe-Mounted Equipment (Valves and Pump) Supports

The hanger type supports (including clamps) and their bolting are designed in accordance with the requirements of ANSI B31.7. The allowable stress limit for the bolting is equal to or less than the yield strength of the bolt material at temperature.

3.9.3.5 Component Supports Not Furnished with the NSSS

3.9.3.5.1 Design Basis

ASME Section III, Subsection NF, is used for the design and installation of the CRD piping supports and TIP piping supports. For the remainder of the non-NSSS portion of the LGS design and installation, Subsection NF is not used. The codes used instead are ANSI B31.7 for nuclear class piping and ANSI B31.1 for non-nuclear class piping. Visual inspection and acceptance of non-Subsection NF pipe supports are performed in accordance with NCIG-01 requirements (Reference 3.9-10). For a graphical definition of jurisdictional boundaries between pipe supports and supporting structures, refer to Figures 3.9-9 and 3.9-10.

The design loading combinations for supports for ASME Class 1, 2 and 3 components, categorized with respect to plant operating conditions identified as normal, upset, emergency, and faulted are given in Table 3.9-21. This table also provides the stress limits for each plant operating condition.

The loads imposed on the ASME Class 1, 2 and 3 active valves and pumps are limited to values below the code allowable loads to ensure operability of the active components by the design of the supports. The supports are designed to remain elastic under the maximum loads. The minor local deformations associated with the elastic deformation of the support will not impair operability of the active components.

3.9.3.5.2 Snubbers

Snubbers are used in seismic Category I systems. The load ratings of the snubbers are appropriate for the design conditions and load combinations.

3.9.3.5.2.1 Analytical Methods

The methodology used for the stress analysis of seismic Category I, 2½ inches and larger piping systems is as follows:

a. For systems designed to seismic and hydrodynamic loads, the flexibility of the pipe supports are considered in the piping stress analysis. A stiffness tolerance criteria is used to facilitate support design and installation.

b. For systems designed to seismic loads, only the supports are considered as rigid members in the piping stress analysis model and are designed such that their fundamental frequencies in the direction of the applied load is within the rigid range of the seismic response spectra.

3.9.3.5.2.2 Snubber Design Specification

Snubbers for LGS are used to arrest shock due to seismic and other dynamic transient events. Under such applications, the snubbers will be subjected to a limited number of load cycles. Snubbers are not designed for vibration control. Therefore, no fatigue evaluation has been performed.

The purchase specification of new shock suppressors (snubbers) covers the following criteria for supplier's performance qualification tests and load tests. End clearance and lost motion are not considered in the piping stress analysis. Instead, linear average snubber stiffness is used in combination with that of the snubber support structure.

Mechanical Snubbers

a. The friction resistance of the suppressor to normal pipe movement shall be a maximum of 1% of the service level A rated load of the unit or 5 lb, whichever is greater.

b. The suppressor shall limit the acceleration of the pipe to a maximum of 0.02 g when subjected to any load up to the normal rated load.

   Based on experience from other plants, the activation threshold has been shown to be above the thermal growth rate of the piping systems.

c. The total lost movement at the suppressor shall not exceed ±0.040 inches due to any applied dynamic cycle load from 3-33 cps up to the rated load at the unit.

d. The suppressor shall be designed for an exposure to a temperature of room temperature (65° to 70°F) prior to initial startup and 200°F during continuous operations and to a radiation dose of 6.4×10^7 rads during the life of the plant. Functioning of the shock suppressor under 340°F temperature for a short duration under rated load shall be demonstrated.

Hydraulic Snubbers

a. The friction resistance of the snubbers rated at 20 kips or greater shall have a friction resistance to normal movement of less than or equal to one percent of the service level A rated load. Snubbers rated at less than 20 kips shall have a friction resistance to normal movement of less than or equal to two percent of the rated load.

b. Activation of the snubber shall be defined as the velocity at which the snubber begins to support load, and restricts movement to the maximum bleed rate of the snubber. The activation velocity may be referred to as the lock-up velocity or activation velocity interchangeably. These requirements shall be satisfied under both tension and compression. The activation velocity of new snubbers at room temperature (65 to 75°F) shall be greater than or equal to 4.72 IPM and less than or equal to 14.17 IPM for all sizes.

c. The bleed rate shall be defined as that velocity at which the snubber will move under constant rated loads after the activation velocity has been reached and the control valves have closed. These requirements shall be satisfied under both tension and compression. The bleed velocity for the new snubbers at room temperature (65 to 75°F) shall be greater than or equal to 0.47 IPM and less than or equal to 4.72 IPM for all sizes.

d. The snubber shall be designed to withstand the normal environmental conditions inside the drywell of -0.5 to 2.0 psig, 160°F (based on Drywell Air Cooling System Design Bases described in Section 9.4.5.2), 40 to 90% relative humidity, and a radiation dose of 6.4×10^7 rads during the life of the plant. This bounding radiaton dose may be reduced by component or model specific evaluations of normal and design basis accident radiation environments.

3.9.3.5.2.3 Snubber Performance Test

A production test is required to be performed on each unit.

a. Check unit to confirm that it operates freely over the total stroke.

b. Measure and record the force required to initiate motion over the stroke in tension and compression.

c. On units which allow movement after the initial suppression of load, determine that the maximum allowable acceleration (mechanical snubbers) or velocity (hydraulic snubbers) is not exceeded. This requirement must be met in both tension and compression at room temperature.

d. Measure and record lost motion of the snubber mechanism (mechanical snubbers only).

Qualification tests are to be performed on randomly selected production models. These tests are used to demonstrate the required load performance (load rating) and specified displacement when subjected to dynamic load cycling. Also included in these tests are low temperature, high temperature, humidity, radiation and faulted load conditions.

Preinstallation, installation and postinstallation inspections of snubbers are performed before a preoperational test. Additional inspections are required if more than 6 months have elapsed between the last inspection and initial system power operation (Section 3.9.3.5.2.4).

Steady state vibration conditions will be identified during the preoperational test program. Snubbers have not been used to control steady state vibration. If the snubbers are used to correct such conditions, they will be evaluated for acceptability under those conditions.

In addition, the snubber inservice inspection program ensures that any potential malfunction due to fatigue-type failure will be detected.

3.9.3.5.2.4 Snubber Preservice Examination

Preservice examination of snubbers should be performed after installation, but not more than six months prior to initial system preoperational testing.

The mechanical snubber examination is described as follows. The objective is to verify adequate preservice examination to mechanical snubbers on all safety-related systems. The prerequisites of all preinstallation, installation, and postinstallation inspections have been performed on mechanical snubbers by designated inspection organizations. Verify through document review that all inspection activities have been completed, verified, and signed. Reviews will be made by systems and additional visual inspections will be made if original inspections are performed more than 6 months prior to initial power operation of the system.

The preservice examination should as a minimum verify the following:

a. There are no visible signs of damage or impaired operability as a result of storage, handling or installation.

b. The snubber location, orientation, position setting, and configuration (attachments, extensions, etc.) are according to design drawings and specifications.

c. Snubbers are not seized, frozen, or jammed.

d. Adequate swing clearance is provided to allow snubber movement.

e. Structural connections such as pins, fasteners, and other connecting hardware such as lock nuts, tabs, wire, and cotter pins are installed correctly.

If the period between the initial preservice examination and initial power operation exceeds 6 months, re-examination of items a. and d. shall be performed. Snubbers which are installed incorrectly or otherwise fail to meet the above requirements must be repaired or replaced and re-examined in accordance with the above criteria.

3.9.3.5.3 Struts

The design load on struts includes those loads caused by dead weight, thermal expansion, primary dynamic forces, (i.e., OBE and SSE), system anchor displacements, and reaction forces caused by relief valve discharge, turbine stop valve closure, etc.

For pipe supports, reactions produced by primary and secondary pipe loads are categorized as primary. The primary and secondary loads are summed and compared to the load rating to ensure that the rating is not exceeded. Because no distinction is made between primary and secondary loads, and load rated components are designed to primary limits or qualified by testing, the supports meet primary stress criteria for primary and secondary loads combined.

3.9.3.5.4 Bolting Stress Limits (Non-NSSS)

The bolting used in pipe support components is designed to an allowable stress equal to or less than the yield strength of the bolt material at temperature.

For flanged connections, the bolt allowables used in the piping are ASME Section III, 1979 Summer Addenda, Sections NB, NC and ND for Class 1, 2, and 3, respectively.

3.9.4 CONTROL ROD DRIVE SYSTEM

The discussion in this Section includes the CRDM, the HCU, the condensate supply system, and the scram discharge volume, and extends to the coupling interface with the control rods.

3.9.4.1 Descriptive Information on CRD System

Descriptive information on the CRD system is contained in Section 4.6.

3.9.4.2 Applicable CRD System Design Specifications

The CRD system is designed to meet the functional design criteria as outlined in Section 4.6, and consists of the following:

a. Locking piston CRD

b. HCU

c. Hydraulic power supply (pumps)

d. Interconnecting piping

e. Flow and pressure and isolation valves

f. Instrumentation and electrical controls

Those components of the CRD forming part of the primary pressure boundary are designed according to ASME Section III.

The quality group classification of the CRD hydraulic system is outlined in Table 3.2-1; and the components are designed according to the codes and standards governing the individual quality groups.

Pertinent aspects of the design and qualification of the CRD components are discussed in the following locations: transients in Section 3.9.1.1, faulted conditions in Section 3.9.1.4, and dynamic testing in Section 3.9.2.2a.

3.9.4.3 Design Loads, Stress Limits, and Allowable Deformation

The ASME Code components of the CRD system are evaluated analytically, and the design loading conditions, stress criteria, calculated stresses, and allowable stresses are summarized in Tables 3.9-6(u) and 3.9-6(v). For the noncode components, experimental testing is used to determine the CRD performance under all possible conditions, as described in Section 3.9.4.4.

Deformation is not a limiting factor in the analysis of the CRD components based on the results of the numerous tests performed on the drive.

3.9.4.3.1 CRD Housing Supports

The CRD housing support system functions are described in Section 4.6.1.3.

The AISC Manual of Steel Construction, "Specification for the Design, Fabrication, and Erection of Structural Steel for Buildings," was used in designing the CRD housing support system. However, to provide a structure that absorbs as much energy as practical without yielding, the allowable tension and bending stresses used were 90% of yield and the shear stress used was 60% of yield. The design stresses are 1.5 times the AISC allowable stresses (60% and 40% of yield, respectively).

The CRD housing supports are designed as seismic Category I equipment. Loading conditions and examples of stress analysis results and limits are given in Table 3.9-6(z).

3.9.4.4 CRD Performance Assurance Program

The CRD test program consists of the following tests:

a. Development tests

b. Factory quality control tests

c. 5 year maintenance life tests

d. 1.5x design life tests

e. Operational tests

f. Acceptance tests

g. Surveillance tests

All of the above tests except c. and d. are discussed in Section 4.6.3. Tests c. and d. are discussed below:

Test c. - <u>5 Year Maintenance Life Tests</u>

Four CRDs are normally picked at random from the production stock each year, and subjected to various tests under simulated reactor conditions and 1/6 of the cycles specified in Section 3.9.1.1.

Upon completion of the test program, CRDs must meet, or surpass, the minimum specified requirements.

Test d. - <u>1.5x Design Life Tests</u>

When a significant design change is made to the components of the drive, the drive is subjected to a series of tests equivalent to 1.5 times the life test cycles specified in Section 3.9.1.1.

Two CRDs underwent such testing in 1976. Upon completion of the test program, these CRDs met or surpassed the minimum specified requirements.

3.9.5 REACTOR PRESSURE VESSEL INTERNALS

This section identifies and discusses the structural and functional integrity of the major RPV internals.

3.9.5.1 <u>Design Arrangements</u>

The core support structures and RPV internals (exclusive of fuel, control rods, CRDs, and incore nuclear instrumentation) are identified below:

<u>Core Support Structures</u>

Shroud

Shroud support

Core support plate and holddown bolts

Top guide (including bolts and keepers)

CRD housing

Fuel supports

Control rod guide tubes

<u>Reactor Internals</u>

*Jet pump assemblies and instrumentation

*Feedwater spargers

Vessel head spray nozzle (Removed-Unit 1. Never installed-Unit 2)

Differential pressure and liquid control lines

Incore flux monitor guide tubes

*Initial startup neutron sources

*Surveillance sample holders

Core spray lines and spargers

*Incore instrument housings

LPCI coupling

*Steam dryer

*Shroud head and steam separator assembly

*Guide rods

CRD thermal sleeves

* = Nonsafety class component

A general assembly drawing of the important reactor components is shown in Figure 3.9-4.

The floodable inner volume of the RPV can be seen in Figure 3.9-5. This is the volume inside the core shroud up to the level of the jet pump suction inlet.

The design arrangement of the reactor internals, such as the jet pumps, steam separators and guide tubes, is such that one end is unrestricted, and thus free to expand.

The LPCI couplings incorporate sleeves to allow free thermal expansion.

3.9.5.1.1 Core Support Structures

The core support structures consist of those items listed in Section 3.9.5.1. These structures form partitions within the reactor vessel, to sustain pressure differentials across the partitions, to direct the flow of the coolant water, and to laterally locate and support the fuel assemblies. Figure 3.9-5 shows the reactor vessel internal flow paths.

3.9.5.1.1.1 Shroud

The shroud support and shroud make up a stainless steel cylindrical assembly that provides a partition to separate the upward flow of coolant through the core, from the downward recirculation flow. This partition separates the core region from the downcomer annulus, thus providing a floodable region following a recirculation line break. The volume enclosed by this assembly is characterized by three regions. The upper portion surrounds the core discharge plenum, which is bounded by the shroud head on top and the top guide's grid plate below. The central portion of the shroud surrounds the active fuel, and forms the longest section of the assembly. This section is bounded at the top by the grid plate and at the bottom by the core plate. The lower portion, surrounding part of the lower plenum, is welded to the RPV shroud support.

3.9.5.1.1.2 Shroud Support

The shroud support is designed to support the shroud, and to support and locate the jet pumps. The shroud support provides an annular baffle between the RPV and the shroud. The jet pump discharge diffusers penetrate the shroud support to introduce the coolant to the inlet plenum below the core.

3.9.5.1.1.3 Shroud Head and Steam Separator Assembly

This component is not a core support structure. It is discussed here to describe coolant flow paths in the RPV. The shroud head and steam separator assembly is bolted to the top of the shroud, forming the top of the core discharge plenum. This plenum provides a mixing chamber for the steam-water mixture before it enters the steam separators. Individual stainless steel axial flow steam separators are attached to the top of standpipes that are welded into the shroud head. The steam separators have no moving parts. In each separator, the steam-water mixture rising through the standpipe passes vanes that impart a spin that establishes a vortex, separating the water from the steam. The separated water flows from the lower portion of the steam separator into the downcomer annulus.

3.9.5.1.1.4 Core Support Plate

The core support plate is a circular stainless steel plate with bored holes, which is stiffened with a rim and beam structure. The plate provides lateral support and guidance for the control rod guide tubes, incore flux monitor guide tubes, peripheral fuel supports, and startup neutron sources. The last two items are supported vertically by the core support plate.

The entire assembly is bolted to a support ledge on the lower portions of the shroud.

3.9.5.1.1.5 Top Guide

The top guide is formed by a series of stainless steel beams joined at right angles to form square openings, and fastened to a peripheral rim. Each opening provides lateral support and guidance for 4 fuel assemblies, or in the case of peripheral fuel, for less than 4 fuel assemblies. Sockets are provided in the bottom of the beam intersections to anchor the incore flux monitors and startup neutron sources. The rim of the top guide rests on a ledge between the upper and central portions of the shroud. The top guide has alignment pins that engage and bear against slots in the shroud which are used to correctly position the assembly before it is secured. Lateral restraint is provided by wedge blocks between the top guide and the shroud wall.

3.9.5.1.1.6 Fuel Supports

The fuel supports, shown in Figure 3.9-6 are of two basic types; peripheral supports, and four-lobed orificed fuel supports. The peripheral fuel support is located at the outer edge of the active core, and is not adjacent to control rods. Each peripheral fuel support holds one fuel assembly, and contains a single orifice assembly designed to ensure proper coolant flow to the peripheral fuel assembly. Each four-lobed orificed fuel support holds four fuel assemblies, and is provided with four orifice plates to ensure proper coolant flow distribution to each rod-controlled fuel assembly. The four-lobed orificed fuel supports rest in the top of the control rod guide tubes, which are supported laterally by the core plate. The control rods pass through slots in the center of the four-lobed orificed fuel support. A control rod and the four adjacent fuel assemblies represent a core cell (Section 4.1.2).

3.9.5.1.1.7 Control Rod Guide Tubes

The control rod guide tubes, located inside the vessel, extend from the top of the CRD housings, and up through holes in the core plate. Each tube is designed as the guide for a control rod, and as the vertical support for a four-lobed orificed fuel support piece and the four fuel assemblies surrounding the control rod. The bottom of the guide tube is supported by the CRD housing, which in turn transmits the weight of the guide tube, fuel support, and fuel assemblies to the reactor vessel bottom head. A thermal sleeve is inserted into the CRD housing from below, and is rotated to lock the control rod guide tube in place. A key is inserted into a locking slot in the bottom of the CRD housing to hold the thermal sleeve in position.

3.9.5.1.1.8 Jet Pump Assemblies

The jet pump assemblies are not core support structures, but are discussed here to describe coolant flow paths in the vessel. The jet pump assemblies are located in two semicircular groups in the downcomer annulus, between the core shroud and the reactor vessel wall. The design and performance of the jet pumps are covered in detail in References 3.9-19 and 3.9-20. Each stainless steel jet pump consists of driving nozzles, a suction inlet, a throat or mixing section, and a diffuser (Figure 3.9-7). The driving nozzle, suction inlet, and throat are joined together as a removable unit, and the diffuser is permanently installed. High pressure water from the recirculation pumps is supplied to each pair of jet pumps through a riser pipe welded to the recirculation inlet nozzle thermal sleeve. A riser brace consists of cantilever beams welded to a riser pipe and to pads on the reactor vessel wall.

The nozzle entry section is connected to the riser by a metal-to-metal, spherical-to-conical seal joint. Firm contact is maintained by a holddown clamp. The throat section is supported laterally by a bracket attached to the riser. There is a slip-fit joint between the throat and diffuser. Some jet pumps have been equipped with a clamp on this slip-fit joint to dampen vibration forces. Some jet pumps have been equipped with an anti-vibration solution between the restrainer bracket and inlet mixer to dampen vibration forces. The diffuser is a gradual conical section, changing to a straight cylindrical section at the lower end.

The licensee will reduce the preload on the beams from 30 kips to 25 kips in accordance with GE recommendations. This increases the expected life of the beams to 19-40 years. Inservice inspection of the jet pump holddown beam will be performed to detect cracking. Inspection frequencies will be based on a lead-plant experience and GE testing, and will be such that any crack initiation will be detected prior to beam failure.

3.9.5.1.1.9 Steam Dryers

The steam dryer assembly is not a core support structure. It is discussed here to describe coolant flow paths in the vessel. The steam dryers remove moisture from the wet steam leaving the steam separators. The extracted moisture flows down the dryer vanes to the collecting troughs, then flows through tubes and into the downcomer annulus. A skirt extends from the bottom of the dryer vane housing to the steam separator standpipe, below the water level. This skirt forms a seal between the wet steam plenum and the dry steam flowing from the top of the dryers to the steam outlet nozzles.

The steam dryer and shroud head are positioned in the vessel during installation with the aid of vertical guide rods. The dryer assembly rests on steam dryer support brackets attached to the reactor vessel wall. Upward movement of the dryer assembly, which may occur under accident conditions, is restricted by steam dryer holddown brackets attached to the reactor vessel top head.

3.9.5.1.1.10 Feedwater Spargers

The feedwater nozzle and sparger design follows the resolution presented in Reference 3.9-6. These components are not core support structures. They are discussed here to describe flow paths in the vessel. The feedwater spargers are stainless steel headers located in the mixing plenum above the downcomer annulus. A separate sparger is fitted to each feedwater nozzle, and is shaped to conform to the curvature of the vessel wall. Sparger end brackets are pinned to vessel brackets to support the spargers. Feedwater flow enters the center of the spargers, and is discharged radially inward, mixing the cooler feedwater with the downcomer flow from the steam separators and steam dryer, before it contacts the vessel wall.

The feedwater also serves to condense the steam in the region above the downcomer annulus, and to subcool the water flowing to the jet pumps and recirculation pumps.

3.9.5.1.1.11 Core Spray Lines

This component is not a core support structure. It is discussed here because the core spray lines are the means for directing flow to the core spray nozzles, which distribute coolant during accident conditions.

Two core spray lines enter the reactor vessel through the two core spray nozzles. The lines divide immediately inside the reactor vessel. The two halves are routed to opposite sides of the reactor vessel, and are supported by clamps attached to the vessel wall. The lines are then routed downward into the downcomer annulus, passing through the upper shroud immediately below the flange. The flow divides again as it enters the center of the semicircular sparger, which is routed halfway around the inside of the upper shroud. The two spargers are supported by brackets designed to accommodate thermal expansion. The line routing and supports are designed to accommodate differential movement between the shroud and vessel. The other core spray line is identical, except that it enters the opposite side of the vessel, and the spargers are at a slightly different elevation inside the shroud. The correct spray distribution pattern is provided by a combination of distribution nozzles pointed radially inward and downward from the spargers (Section 6.3).

3.9.5.1.1.12 Head Cooling Spray Nozzle (Removed)

This component is not a core support structure.

The head cooling spray nozzle (component B11-D072) was mounted on a short length of pipe and a flange, which was bolted to a mating flange (RPV Nozzle N6A) on the reactor vessel head. The piping supplying coolant to the nozzle has been disconnected, partially removed and the remainder abandoned in place in Unit 1 and was never installed in Unit 2. The head cooling spray nozzle in

Unit 1 was removed with the discontinued piping and never installed in Unit 2. RPV Nozzle N6A is empty and blind flanged in both units.

3.9.5.1.1.13 Differential Pressure Line

This component is not a core support structure. It is discussed here to describe the coolant flow paths in the reactor vessel. The differential pressure line senses the differential pressure across the core support plate (Section 7.7). This line enters the reactor vessel at a point below the core shroud, as two concentric pipes. In the lower plenum, the two pipes separate. The inner pipe terminates inside the lower shroud support, with a capped, perforated length below the core support plate. This section of pipe was formerly utilized as the liquid control sprager but now is only used to sense the pressure below the core support plate. The outer pipe terminates open-ended immediately above the core support plate, and senses the differential pressure across the core support plate and the fuel support assemblies.

3.9.5.1.1.14 Incore Flux Monitor Guide Tubes

This component is not a core support structure, but is discussed here to describe the coolant flow paths in the reactor vessel. Incore flux monitor guide tubes provide a means of positioning fixed detectors in the core, as well as providing a path for calibration monitors (TIP system).

The incore flux monitor guide tubes extend from the top of the incore flux monitor housing (Section 5.3) in the lower plenum, to the top of the core support plate. The power range detectors for the PRNM System units, and the dry tubes for the SRM and IRM detectors are inserted through the guide tubes. A latticework of clamps, tie bars, and spacers give lateral support and rigidity to the guide tubes. The bolts and clamps are welded, after assembly, to prevent loosening during reactor operation.

3.9.5.1.1.15 Surveillance Sample Holders

This component is not a core support structure. It is discussed here to describe the coolant flow paths in the reactor vessel. The surveillance sample holders are welded baskets containing impact and tensile specimen capsules (Section 5.3). The baskets hang from the brackets that are attached to the inside wall of the reactor vessel, and extend to mid-height of the active core. The radial positions are chosen to expose the specimens to the same environment and maximum neutron fluxes experienced by the reactor vessel itself, while avoiding jet pump removal interference or damage.

3.9.5.1.1.16 Low Pressure Coolant Injection Lines

This component is not a core support structure, but is discussed here to describe the coolant flow paths in the reactor vessel. Four LPCI lines penetrate the core shroud through separate LPCI nozzles. Coolant is discharged inside the core shroud.

3.9.5.1.1.17 Startup Neutron Sources

The startup neutron sources are held in place by spring pressure between the top of the core support and the bottom of the top guide. For Unit 1, each source consists of two irradiated antimony rods within a single beryllium cylinder. Both the antimony and the beryllium are encased in stainless steel tubes. For Unit 2, californium is used; it is also encased in stainless steel tubes. The design provides for a sufficient source of neutrons present in the core to ensure that the core neutron flux is continuously detectable by installed neutron monitors and to ensure that significant changes in core reactivity are readily detectable by installed neutron flux instrumentation.

3.9.5.2 Design Loading Conditions

3.9.5.2.1 Events to be Evaluated

Examination of the spectrum of conditions that the safety design basis must satisfy by core support structure and ESF components reveals four significant faulted events:

a. Recirculation line break: a break in a recirculation line between the reactor vessel and the recirculation pump suction

b. Steam line break accident: a break in one main steam line between the reactor vessel and the flow restrictor. This accident results in significant pressure differentials across some of the structures within the reactor.

c. Earthquake: subjects the core support structures and reactor internals to significant forces as a result of ground motion.

d. SSE/relief valve discharge: SRV discharge in combination with SSE.

Analysis of other conditions existing during normal operation, abnormal operational transients, and accidents shows that the loads affecting the core support structures and other ESF reactor internals are less severe than these three postulated events. The faulted conditions for the RPV internals are discussed in Section 3.9.1.4. Loading combination and analysis for the RPV internals are discussed in Section 3.9.3.1, and Tables 3.9-2 and 3.9-6. These results are based on the power rerate analysis and do not reflect the use of GE13 or GE14 fuel; the impact of GE13 fuel is documented in Reference 3.9-28. The impact of GE14 fuel is documented to be bounded by GE13 fuel in Reference 3.9-34. The impact of the MUR power uprate is evaluated in Reference 3.9-31 and Reference 3.9-32. Reference 3.9-33 identifies new design basis values for Fuel Lift Margin and Control Rod Guide Tube Lift Forces under MUR conditions. Additional analyses which consider the use of GNF2 fuel are documented in Reference 3.9-35. The GNF2 fuel is demonstrated to be bounded by the analyses in Reference 3.9-33.

3.9.5.2.2 Pressure Differential During Rapid Depressurization

A digital computer code is used to analyze the transient conditions within the reactor vessel following the recirculation line break accident and the steam line break accident. The analytical model of the vessel consists of nine nodes, connected to the necessary adjoining nodes by flow paths having the required resistance and inertial characteristics. The program solves the energy and mass conservation equations for each node, giving the depressurization rates and pressure in the various regions of the reactor. Figure 3.9-8 shows the nine reactor nodes. The computer code used is the GE Short-Term Thermal-Hydraulic Model, described in Reference 3.9-21. This model is approved for use in ECCS conformance evaluation under 10CFR50, Appendix K. In order to adequately describe the blowdown pressure effect on the individual assembly components, three features are included in the model that are not applicable to the ECCS analysis and are, therefore, not described in Reference 3.9-21. These additional features are discussed below:

a. The liquid level in the steam separator region, and in the annulus between the dryer skirt and the pressure vessel, is tracked to more accurately determine the flow and mixture quality in the steam dryer and in the steam line.

b. The flow path between the bypass region and the shroud head is more accurately modeled, since the fuel assembly pressure differential is influenced by flashing in the guide tubes and in the bypass region for a steam line break. In the ECCS analysis, the momentum equation is solved in this flow path; but its irreversible loss coefficient is conservatively set at an arbitrary low value.

c. The enthalpies in the guide tubes and the bypass region are calculated separately, since the fuel assembly ΔP is influenced by flashing in these regions. In the ECCS analysis, these regions are lumped.

3.9.5.2.3 Recirculation Line and Steam Line Break

3.9.5.2.3.1 Accident Definition

Both a recirculation line break (the largest liquid break) and an inside steam line break (the largest steam break) are considered in determining the design basis accident for the ESF reactor internals. The recirculation line break is the same as the design basis LOCA described in Section 6.3. A sudden, complete circumferential break is assumed to occur in one recirculation loop. The resulting pressure differentials on the reactor internals and core support structures are in all cases less than for the main steam line break.

The analysis of the steam line break assumes a sudden, complete circumferential break of one main steam line, between the reactor vessel and the main steam line restrictor. A steam line break upstream of the flow restrictors produces a larger blowdown area, and thus a faster depressurization rate, than a break downstream of the restrictors. A larger blowdown area results in greater pressure differentials across the reactor internal structures.

The steam line break accident produces significantly higher pressure differentials across the reactor internal structures than does the recirculation line break. This results from the higher reactor depressurization rate associated with the steam line break. Therefore, the steam line break is the DBA for internal pressure differentials.

3.9.5.2.3.2 Effects of Initial Reactor Power and Core Flow

The maximum internal pressure loads can be considered to be composed of two parts: steady-state and transient pressure differentials. For a given plant, the core flow and the core power are the two major factors which influence the reactor internal pressure differentials. The core flow essentially affects only the steady-state part. For a fixed power, the greater the core flow, the larger the steady-state pressure differentials. The core power affects both the steady-state and the transient parts. As the power is decreased, there is less voiding in the core, and consequently the steady-state core pressure differential is less. However, less voiding in the core also means that less steam is generated in the RPV, thus increasing the depressurization rate and the transient part of the maximum pressure load. As a result, the total loads on some components are higher at low power.

To ensure that the calculated pressure differences bound those which are expected if a steam line break should occur, an analysis is conducted at a low power, high recirculation flow condition, in addition to the standard safety analysis condition at high power-rated recirculation flow. The power chosen for analysis is the minimum value permitted by the recirculation system controls at rated recirculation drive flow (that is, the drive flow necessary to achieve rated core flow at rated power).

This condition maximizes those loads which are inversely proportional to power. It must be noted that this condition, while possible, is unlikely, because the reactor generally operates at or near full power; and because high core flow is neither required, nor desirable at such a reduced power condition.

3.9.5.2.4 Seismic and Hydrodynamic Loads

The seismic and hydrodynamic loads acting on the structures within the reactor vessel are based on a dynamic analysis, as described in Section 3.7. Seismic analysis is performed by coupling the lumped-mass model of the reactor vessel and internals (Section 3.7), with the building model to determine the acceleration, force, and moment time histories in the reactor vessel and internals. This is accomplished by using the modal superposition method. Acceleration response spectra are also produced for subsystem analysis of selected components.

3.9.5.3 Design Bases

3.9.5.3.1 Safety Design Bases

The reactor core support structures and internals meet the following safety design bases:

a. Core support structures are arranged to provide a floodable volume, in which the core can be adequately cooled in the event of a breach in the nuclear system process barrier, external to the reactor vessel.

b. Deformation is limited to ensure that the control rods and the core standby cooling systems can perform their safety functions.

c. Mechanical design of applicable structures ensures that safety design bases a. and b., above, are satisfied so that the safe shutdown of the plant and removal of decay heat are not impaired.

3.9.5.3.2 Power Generation Design Bases

The reactor core support structures and internals are designed to the following power generation design bases:

a. They provide the proper coolant distribution during all anticipated normal operating conditions up to full power operation of the core without fuel damage.

b. They are arranged to facilitate refueling operations.

c. They are designed to facilitate inspection.

3.9.5.3.3 Design Loading Categories

Loading combinations for the core support structures are shown in Table 3.9-26. The basis for determining faulted loads on the reactor internals is shown for seismic and hydrodynamic loads in

Section 3.7, and for pipe rupture loads in Sections 3.9.5.2.3 and 3.9.5.3.4. Table 3.9-6(b) gives analytical methods and allowable and calculated stresses for typical core support structures and reactor internal components.

Stress intensity and other design limits are discussed in Section 3.9.5.3.5. The core support structures which are fabricated as part of the RPV assembly are discussed in Section 3.9.1.3.

LGS reactor internals were designed and procured prior to the issuance of ASME Section III, Subsection NG. However, an earlier draft of the ASME Code was used as a guide in the design of the reactor internals. These criteria are presented in this section and were used in lieu of Subsection NG. Subsequent to the issuance of Subsection NG, comparisons were made to ensure that the pre-NG design meets the equivalent level of safety as presented by Subsection NG.

The design requirements for equipment classified as "other internals," e.g., steam dryers and shroud heads, are specified by the designer with appropriate consideration of the intended service of the equipment and expected plant and environmental conditions under which it operates. Where possible, design requirements are based on applicable industry codes and standards. If these are not available, the designer relies on accepted industry or engineering practices.

3.9.5.3.3.1 Reactor Core Support and Internals Structural Margin Evaluations

The analyses of conditions found during inspections of reactor internal and core support structures use approved industry codes and standards as described in the LGS Inservice Inspection Program, references 5.2-10 and 5.2-11. These analyses are performed as described in the above referenced Sections except that limit load, linear elastic (LEFM), and elastic-plastic (EFPM) fracture mechanic methods may be used as discussed in Tables 3.9-23, 3.9-24, 3.9-28, and 3.9-30.

Neutron fluence is evaluated when the irradiation induced changes in the material fracture toughness properties are judged to be significant. These material properties include yield and ultimate tensile strengths, uniform elongation and upper-shelf Charpy energy. The trends in these properties as a function of fluence level are reviewed to determine a fluence value above which the use of LEFM or EPFM techniques would be necessary and to determine the appropriate flaw growth rate to be used in the structural margin analyses. The fluence calculations use the methodology discussed in Section 4.3.2.8.

a. The design loads for the LGS Unit 1 and Unit 2 core shroud horizontal welds H1 through H8 have been calculated and are documented in reference 3.9-29. The loads and their combinations are based on power rerate and new loads design adequacy evaluations as discussed in reference 3.9-23. The effects of additional loads from GE13 fuel design (reference 3.9-28), fuel lift loads, and increased core flow (3% noise) beyond power rerate and the new loads programs are included in the updated core shroud loads and analysis. The impact of GE14 fuel is documented to be bounded by GE13 fuel in Reference 3.9-34. Reference 3.9-33 identifies new design basis values for Fuel Lift Margin and Control Rod Guide Tube Lift Forces under MUR conditions. Additional analyses which consider the use of GNF2 fuel are documented in Reference 3.9-35. The GNF2 fuel is demonstrated to be bounded by the analyses in Reference 3.9-33.

Reference 3.9-30 was prepared in response to Generic Letter (GL) 94-03, Intergranular Stress Corrosion Cracking of Core Shrouds in Boiling Water Reactors, which required a plant specific safety assessment supporting continued operation of core shrouds. The report provides accident loading information applicable to LGS Unit 1 and Unit 2, as well as the safety assessment for postulated through-wall flaws at core shroud horizontal welds H1 through H7. The analysis considers both normal plant operations and limiting abnormal operational occurrences. It provides information on the plant response i.e., control rod insertion and ECCS injection, to postulated Main Steam Line Break (MSLB) and Recirculation Line Break (RLB), including a coincident Safe Shutdown Earthquake (SSE). The overall conclusion concerning plant safety is that the core shroud is an extremely flaw tolerant core support structure such that the probability is a very small that flawed welds would result in shroud separation under any desgn basis operational event.

The design requirements for equipment classified as "other internals," e.g., steam dryers and shroud heads, are specified by the designer with appropriate consideration of the intended service of the equipment and expected plant and environmental conditions under which it operates. Where possible, design requirements are based on applicable industry codes and standards. If these are not available, the designer relies on accepted industry or engineering practices.

3.9.5.3.4 Response of Internals Due to Inside Steam Break Accident

The maximum pressure loads acting on the reactor internal components result from an inside steam line break; on some components the loads are maximum when operating at the minimum power associated with the maximum core flow. This is substantiated by the analytical comparison of liquid versus steam breaks, and by the investigation of the effects of core power and core flow.

It has also been pointed out that it is possible but not probable that the reactor is operating at the rather abnormal condition of minimum power and maximum core flow. More realistically, the reactor is at or near a full power condition, and thus the maximum pressure loads acting on the internal components would be less.

3.9.5.3.5 Stress, Deformation, and Fatigue Limits for ESF Reactor Internals (Except Core Support Structure)

Elastic displacement is considered in the design of reactor internal components in which deflection can affect control rod insertability. Plastic deformation will not occur in any permanent core support structure component of the reactor vessel. Radiation-induced deformation can occur in the fuel channel over the core life. These effects are considered in control rod insertability tests. No fatigue analysis is required under the faulted conditions due to the low encounter frequency of faulted events and the low number of cycles. The forcing functions applicable to the reactor internals are discussed in Section 3.9.2.5. The stress, deformation, and fatigue limits are given in Table 3.9-6(b).

3.9.5.3.6 Stress, Deformation, and Fatigue Limits for Core Support Structures

The stress, deformation, and fatigue limits are given in Table 3.9-6(f).

3.9.6 Inservice Testing Of Pumps And Valves

Inservice testing of pumps and valves is accomplished in accordance with the requirements of 10CFR50.55a, using the date of commercial operation for determining test intervals.

The Preservice Testing Program included provisions for design and access to enable the operational readiness testing of pumps and valves, and was required to comply, as a minimum, with the 1971 Edition of Section XI of the ASME B&PV Code, including the winter of 1972 Addenda (this being in effect 6 months prior to the LGS construction permit date of June 1974). That publication did not include requirements for preservice testing of pumps and valves to ensure operational readiness. The requirements for inservice testing of pumps and valves were added as Subsections IWV and IWP to the ASME B&PV Code, Section XI, Summer 1973 Addenda, effective December 30, 1973. The Preservice Testing Program for assessing operational readiness of pumps and valves was conducted, however, to the extent practical within design limitations, to comply with the 1980 Edition of ASME Section XI, with addenda through Winter 1981.

Inservice testing of pumps and valves to ensure operational readiness for the first 120-month interval was performed in accordance with the requirements of 10CFR50.55a and met, to the extent practical within design limitations, the requirements of the Code in effect 12 months prior to the date of commercial operation. The first 120-month interval for Unit 2 was required to comply with the 1986 Edition of ASME Section XI. As permitted by 10CFR50.55a, the Unit 1 IST Program was updated to comply with the 1986 Edition of ASME Section XI. Subsequently, NRC authorized a one-time extension of the first 120-month interval for Unit 1, resulting in both Units being on concurrent intervals.

During the second and successive 120-month intervals, inservice testing of pumps and valves to ensure operational readiness shall be performed in accordance with the requirements and limitations specified in 10CFR50.55a. Detailed information regarding current Code requirements, component selection, testing requirements, Code Class, and deviations from referenced Code requirements is provided in Reference 3.9-25.

There is a Risk Informed Categorization and Treatment Program at Limerick which is based on 10 CFR 50.69. This regulation provides an alternative approach for establishing requirements for treatment of SSCs using a risk-informed method of categorizing SSCs according to their safety significance. Specifically, for SSCs categorized as low safety significant, alternate treatment requirements may be implemented rather than treatments chosen by the inservice testing of pumps and valves program. Refer to Section 13.5.5 for further information.

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3.9-26 "Limerick Generating Station, Units 1 and 2, SRV Setpoint Tolerance Relaxation Licensing Report," NEDC-32645P (December 1998).

3.9-27 "Limerick Generation Station, Units 1 and 2, Analysis of Reduced HPCI Flow," GENE-L12-00860-00-01, DRF L12-00860-00 (December 1998).

3.9-28 "Assessment of Reactor Internals due to GE13 Fuel Change," GENE-B13-02005-01, Revision 0, March 1999).

3.9-29 General Electric Nuclear Energy Report, "Core Shroud Loads at Horizontal Welds H1 through H8", B13-02066-00-20 Lt1, dated September 19, 2000, is issued as controlled document DWG, B11-D001-S-005.

3.9-30 General Electric Nuclear Energy Report, "Fermi 2 Hope Creek Unit 1, Limerick Units 1 and 2, Peach Bottom Units 2 and 3, and Susquehanna Units 1 and 2 Shroud Safety Assessment," GENE-523-A094-0995, Dated July 1999, shall be issued as Controlled Document B11-D001-S-011.

3.9-31 Design Analysis LEAM-MUR-0061, "Exelon Nuclear Limerick Units 1 and 2 Nuclear Stations Thermal Power Optimization (TPO) (3517 MWt) Task T0302: Reactor Vessel Integrity Stress and Fatigue Evaluation," GE Hitachi Nuclear Energy, 0000-0097-3401-R0, Revision 0, Janauary 2010.

3.9-32 Design Analysis LEAM-MUR-0050, "Exelon Nuclear Limerick Units 1 and 2 Thermal Power Optimization Task T0303: RPV Internals Mechanical Evaluation," GE Hitachi Nuclear Energy, 0000-0107-2835-R0, Revision 0, December 2009.

3.9-33 Design Analysis LEAM-MUR-0040, "Exelon Nuclear Limerick Units 1 and 2 Nuclear Stations Thermal Power Optimization (TPO) (3517 MWt) Task T0304: Reactor Internal Pressure Differences and Fuel Lift Evaluation," GE Hitachi Nuclear Energy, 0000-0101-4297-R0, Revision 0, July 2009.

3.9-34 "GE14 Fuel Design Cycle-Independent Analyses for Limerick Generating Station Unit 1 and 2," GE-NE-L12-00884-00-01P, March 2001 (G-080-VC-00146).

3.9-35 "GNF2 Fuel Design Cycle-Independent Analyses for Exelon Limerick Generating Station Unit 1 and 2," NEDC-33627P, Rev. 1, March 2001 (G-080-VC-00446).

Table 3.9-1

APPLICABLE THERMAL TRANSIENTS

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)(15)	ΔTEMPERATURE (°F)
A. 130 CYCLES, CONDITION - TEST (PRE-STARTUP LEAK TEST)(1)					
Main Steam Line	70	100	30 min	60	30
Recirculation Suction	70	100	30 min	60	30
Recirculation Discharge	70	100	30 min	60	30
Bottom Drain	70	100	30 min	60	30
SLCS	70	100	30 min	60	30
	100	50	Step	10 min	50
	50	100	Step	duration	50
Core Spray	70	100	30 min	60	30
Feedwater	70	100	30 min	60	30
B. 120 CYCLES, CONDITION - NORMAL (STARTUP)(2)					
Main Steam Line	100	546	-	100	446
Recirculation Suction	100	546	-	100	446
	546	538	Step	-	8
	538	522	Step	-	16
Recirculation Discharge	538	522	Step	-	16
Bottom Drain	538	522	Step	-	16
SLCS	538	522	Step	-	16
Core Spray	100	406	-	100	306
only 10 cycles	406	50	Step	-	356
	50	406	Step	-	356
	406	546	-	100	140
Feedwater	100	546	-	100	446
	546	90	Step	-	456
	90	420	30 min	660	330

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)[15]	ΔTEMPERATURE (°F)
C. 10,400 CYCLES, CONDITION - NORMAL (DAILY POWER REDUCTION AND ROD PATTERN CHANGE)[3]					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	522	-	-	-
Recirculation Discharge	522	522	-	-	-
Bottom Drain	522	522	-	-	-
SLCS	522	522	-	-	-
Core Spray	522	522	-	-	-
Feedwater	420	354	15 min	264	66
	354	420	15 min	264	66
Cleanup Return	435	435	-	-	-
D. 2000 CYCLES, CONDITION - NORMAL (WEEKLY POWER REDUCTION)[3]					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	522	-	-	-
Recirculation Discharge	522	522	-	-	-
Bottom Drain	522	522	-	-	-
SLCS	522	522	-	-	-
Core Spray	522	522	-	-	-
Feedwater	420	324	30 min	192	96
	324	420	30 min	192	96
Cleanup Return	435	435	-	-	-
E. 70 CYCLES ,CONDITION - UPSET (PARTIAL FEEDWATER HEATER BYPASS)[3]					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	512	2 min	300	10
	512	522	4 min	150	10
Recirculation Discharge	512	522	4 min	150	10
Bottom Drain	512	522	4 min	150	10
SLCS	512	522	4 min	150	10
Core Spray	512	522	4 min	150	10
Feedwater	420	265	1.5 min	6200	155
	265	420	3 min	3100	155

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)[15]	ΔTEMPERATURE (°F)
F. 10 CYCLES, CONDITION - UPSET (TURBINE TRIP 100 PERCENT BYPASS)[3]					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	490	1.5 min	1,280	32
	490	522	4 min	480	32
Recirculation Discharge	490	522	4 min	480	32
Bottom Drain	490	522	4 min	480	32
SLCS	490	522	4 min	480	32
Core Spray	490	522	4 min	480	32
Feedwater	420	100	1.5 min	12,800	320
	100	420	4 min	4,800	320
G. 40 CYCLES, CONDITION - UPSET (SCRAM - T/G TRIP FEEDWATER ON - MSIV OPEN)[4]					
Main Steam Line	546	565	10 sec7000		19
	565	538	15 sec6500		27
	538	400	-	100	138
	400	546	-	100	146
Recirculation Suction	522	400	-	100	122
	400	546	-	100	146
	546	538	Step	-	18
	538	522	Step	-	16
Recirculation Discharge	538	522	Step	-	16
Bottom Drain	538	522	Step	-	16
SLCS	538	522	Step	-	16
Core Spray	538	522	Step	-	16
Feedwater	420	275	1 min	8700	145
	275	100	15 min	700	175
	100	250	Step	-	150
	250	420	30 min	340	170

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)[15]	ΔTEMPERATURE (°F)
H. 140 CYCLES, CONDITION - UPSET (ALL OTHER SCRAMS)[5]					
Main steam line	546	538	15 sec	1920	8
	538	400	-	100	138
	400	546	-	100	146
Recirculation Suction	522	400	-	100	122
	400	546	-	100	146
	546	538	Step	-	18
	538	522	Step	-	16
Recirculation Discharge	538	522	Step	-	16
Bottom Drain	538	522	Step	-	16
SLCS	538	522	Step	-	16
Core Spray	538	522	Step	-	16
Feedwater	420	275	1 min	8700	145
	275	100	15 min	700	175
	100	250	Step	-	150
	250	420	30 min	340	170
I. CYCLES LISTED BELOW, CONDITION - NORMAL (RATED POWER)[3]					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	546	-	-	-
Recirculation Discharge	522	522	-	-	-
Bottom Drain	522	150	1 hr	372	372
240 Cycles	150	522	Step	NA	372
SLCS	522	60	Step	NA	462
10 Cycles	60	522	60 min	462	462
Core Spray	522	522	-	-	-
Feedwater	420	420	-	-	-
Cleanup Return	435	435	-	-	-

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)[15]	ΔTEMPERATURE (°F)
J. 111 CYCLES, CONDITION - NORMAL (SHUTDOWN)[6]					
Main Steam Line	546	375	-	100	171
	375	330	10 min	270	45
	330	100	-	100	230
RHR Return	375	50	Step	15 sec	325
	50	300	Step	duration	250
	300	100	-	100	200
Recirculation Suction	522	546	Step	-	24
	546	375	-	100	171
	375	330	10 min	270	45
	330	100	-	100	230
Bottom Drain	330	100	-	100	230
SLCS	330	100	-	100	230
Core Spray	330	100	-	100	230
Recirculation Discharge	522	546	Step	-	24
	546	375	-	100	171
	375	300	Step	-	75
	300	260	10 min	240	40
	260	100	-	100	160
Feedwater	420	265	30 min	310	155
	265	420	Step	-	155
	420	546	-	100	126
	546[7]	100	-	100	446
K. 10 CYCLES, CONDITION - EMERGENCY (LOSS OF FEEDWATER PUMPS - MSIV CLOSE)[8]					
Main Steam Line	546	573	3 sec	32,400	27
	573	561	10 sec	4300	12
	561	538	3 min	560	23
	538	561	73 min	19	23
	561	500	7 min	520	61
	500	400	-	100	100
	400	546	-	100	146
Recirculation Suction	522	300	30 min	444	222
	300	546	-	100	246
	546	538	Step	-	8
	538	522	Step	-	16

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)(15)	ΔTEMPERATURE (°F)
Recirculation Discharge	538	522	Step	-	16
Bottom Drain	522	300	3.7 min	3600	222
	300	500	23 min	523	200
	500	300	7 min	1720	200
	300	546	-	100	246
	546	538	Step	-	8
	538	522	Step	-	16
SLCS	538	522	Step	-	16
Core Spray	538	522	Step	-	16
Feedwater	420	546	Step	-	126
	546	40	Step	-	506
	40	546	23 min	1300	506
	546	40	Step	-	506
	40	546	51 min	600	506
	546	40	Step	-	506
	40	300	5 min	3120	260
	300	546	-	100	246
	546	100	Step	-	446
	100	250	Step	-	150
	250	420	30 min	340	170

L. 1 CYCLE, CONDITION - EMERGENCY (REACTOR OVERPRESSURE DELAYED SCRAM)(9)

Main Steam Line	546	583	2 sec	66,600	37
	583	538	30 sec	5,400	45
	538	400	-	100	138
Recirculation Suction	522	562	11 sec	13,100	40
	562	400	-	100	162
Recirculation Discharge	562	400	-	100	162
Bottom Drain	562	400	-	100	162
SLCS	562	400	-	100	162
Core Spray	562	400	-	100	162
Feedwater	420	276	1 min	8,640	144
	276	100	15 min	704	176

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)[15]	ΔTEMPERATURE (°F)
M. 8 CYCLES, CONDITION - EMERGENCY (SINGLE SRV BLOWDOWN)[10]					
Main Steam Line	546	375	10 min	1026	171
	375	100	-	100	275
Recirculation Suction	522	375	10 min	882	147
	375	100	-	100	275
Recirculation Discharge	375	100	-	100	275
Bottom Drain	375	100	-	100	275
SLCS	375	100	-	100	275
Core Spray	375	100	-	100	275
Feedwater	420	276	1 min	8640	144
	276	100	15 min	704	176
N. 1 CYCLE, CONDITION - EMERGENCY (AUTOMATIC DEPRESSURIZATION)[11]					
Main Steam Line	546	375	3.3 min	3100	171
	522	375	3.3 min	2700	147
	375	281	-	300	94
Recirculation Discharge	375	281	-	300	94
Bottom Drain	375	281	-	300	94
SLCS	375	281	-	300	94
Core Spray	375	281	-	300	94
Feedwater	420	276	1 min	8640	144
	276	100	15 min	784	176
O. 1 CYCLE, CONDITION - EMERGENCY (IMPROPER START OF COLD RECIRCULATION LOOP)[3]					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	130	Step	26 sec	392
	130	522	Step	duration	392
Recirculation Discharge	522	130	Step	34 sec	392
	130	522	Step	duration	392
Bottom Drain	522	522	-	-	-
SLCS	522	522	-	-	-
Core Spray	522	268	Step	34 sec	254
	268	522	Step	duration	254
Feedwater	420	420	-	-	-

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)(15)	ΔTEMPERATURE (°F)
P. 1 CYCLE, CONDITION - EMERGENCY (SUDDEN PUMP START IN COLD LOOP)(3)					
Main Steam Line	546	546	-	-	-
Recirculation Suction	522	522	-	-	-
Recirculation Discharge	522	130	Step	34 second	392
	130	522	Step	duration	392
Bottom Drain	522	350	Step	34 second	172
	350	522	Step	duration	172
SLCS	350	522	Step	34 second duration	172
Core Spray	522	522	-	-	-
Feedwater	420	420	-	-	-
Q. 1 CYCLE, CONDITION - EMERGENCY (IMPROPER START WITH RECIRCULATION PUMPS OFF)(12)					
Main Steam Line	100	546	-	100	446
Recirculation Suction	100	546	-	100	446
Recirculation Discharge	100	546	-	100	446
	546	130	Step	34 sec	416
	130	546	Step	duration	416
Bottom Drain	100	546	5 min	5352	446
SLCS	100	546	5 min	5352	446
Core Spray	100	546	-	100	446
Feedwater	90	546	-	100	456
	546	90	Step	-	456
	90	420	30 min	660	330
R. 1 CYCLE, CONDITION - FAULTED (PIPE RUPTURE AND BLOWDOWN)(13)					
Main Steam Line	546	281	15 sec	63,500	265
Recirculation Suction	522	281	15 sec	57,000	241
Recirculation Discharge	522	281	15 sec	57,000	241
	281	223	35 sec		58
	223	50	Step	90 sec	173
	50	130	Step	duration	80

Table 3.9-1 (Cont'd)

PIPELINE	INITIAL TEMPERATURE (°F)	FINAL TEMPERATURE (°F)	TIME	TEMPERATURE RATE (°F/hr)(15)	ΔTEMPERATURE (°F)
Bottom Drain	522	281	15 sec	57,000	241
	281	273	35 sec	822	8
	273	50	Step	90 sec	223
	50	130	Step	duration	80
SLCS	50	130	Step	90 sec duration	80
Core Spray	522	406	10 sec	41,700	116
	406	50	Step	90 sec	356
	50	130	Step	duration	80
Feedwater	420	281	15 sec	33,400	139

S. BECHTEL CRITERIA FOR BOP PIPING

1. ½ SSE Cycles (OBE) — Condition - Upset

Expected number of equivalent ½ SSE in life of pipe system	5
Average duration of strong motion vibration ½ SSE	15 sec
Average number of maximum seismic load cycles of pipe system for each ½ SSE	10
Total lifetime number of maximum seismic load cycles of piping system	50

2. SSE Cycles (Design Basis Earthquake) — Condition - Faulted

Expected number of equivalent SSE in life of pipe system	1
Average duration of strong motion vibration SSE	15 sec

Table 3.9-1 (Cont'd)

T. GENERAL ELECTRIC CRITERIA FOR NSSS PIPING

1.	½ SSE Cycles (OBE)	Condition - Upset
	Expected number of equivalent ½ SSE in life of pipe system	1
	Average duration of strong motion vibration ½ SSE	30 sec
	Average number of maximum seismic load cycles of pipe system for each ½ SSE	10
	Total lifetime number of maximum seismic load cycles of piping system	10
2.	SSE Cycles (Design Basis Earthquake)	Condition - Faulted
	Expected number of equivalent SSE in life of pipe system	1
	Average duration of strong motion vibration SSE	30 sec
	Average number of maximum seismic load cycles of pipe system for each SSE	1
	Total lifetime number of maximum seismic load cycles of piping system	1
3.	Turbine Stop Valve(14) Closure	Condition - Upset 120 cycles
4.	Relief Valve Lift Cycles(14) (at 3 cycles per actuation)	Condition - Upset 34,200 cycles

Table 3.9-1 (Cont'd)

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(1) After temperature is raised to 100°F, reactor pressure is increased to 1250 psig and then decreased to 0 psig.
(2) Reactor pressure increases from 0 to 1000 psig at rate of temperature increase.
(3) Reactor pressure remains at 1000 psig.
(4) Reactor pressure increases to 1125 psig all relief valves open. Pressure decreases to 240 psig and then increases to 1000 psig.
(5) Reactor pressure decreases to 240 psig and then increases to 1000 psig.
(6) Reactor pressure decreases from 1000 psig to 0 psig.
(7) 5 step changes to 100°F and back during cooldown.
(8) Reactor pressure increases to 1180 psig. All relief valves open. Pressure decreases to 1125 psig and relief valves close. RCIC initiates and pressure decreases to 875 psig. RCIC trips off on high level and pressure increases to 1125. One relief valve opens and then closes as pressure decreases at rate of 100°F/hr.
(9) Reactor pressure increases to 1350 psig. All relief valves and safety valves open. Pressure decreases to 240 psig.
(10) Reactor pressure decreases to 0 psig with one relief valve or safety valve open.
(11) Reactor pressure decreases with auto-blowdown relief valves open to 35 psig.
(12) Reactor pressure increases to 1000 psig as temperature increases.
(13) Reactor pressure decreases from 1000 psig to 35 psig in 15 seconds.
(14) Not applicable to recirculation piping due to negligible effect.
(15) Temperature rates are informational only and approximated.

Table 3.9-2

PLANT EVENTS

EVENT NO.		NO. OF CYCLES
	NORMAL, UPSET, AND TESTING CONDITIONS	
1.	Bolt-up(1)	123
2.	Design hydrostatic test	130
3.	Startup (100° F/hr heatup rate)(2)	120
4.	Daily reduction to 75% power(1)	10,000
5.	Weekly reduction to 50% power(1)	2,000
6.	Control rod pattern change(1)	400
7.	Loss of feedwater heaters	80
8.	OBE event at rated operating conditions	10/50(3)
9.	Scram:	
	a. Turbine-generator trip, feedwater on, isolation valves stay open	40
	b. Other scrams	140
10.	Reduction to 0% power, hot standby, shutdown (100° F/hr cooldown rate)(2)	111
11.	Unbolt	123
12.	Pre-op blowdown	10
13.	Loss of RWCU	240
14.	MSRV actuations	7700
15.	SLCS Operation	10

Table 3.9-2 (Cont'd)

EVENT NO.			NO. OF CYCLES
	EMERGENCY CONDITIONS		
16.	Scram:		
	a.	Reactor overpressure with delayed scram, feedwater stays on, isolation valves stay open	1[(4)]
	b.	Loss of feedwater pumps, isolation valves closed	5
	c.	Automatic Blowdown	1[(4)]
	d.	Single safety or relief valve blowdown	8
17.	Improper start of cold recirculation loop		1[(4)]
18.	Sudden start of pump in cold recirculation loop		1[(4)]
19.	Improper startup with reactor drain shut off		1[(4)]
	FAULTED CONDITION		
20.	SSE at rated operating conditions		1[(4)]
21.	Pipe rupture and blowdown		1[(4)]

(1) Applies to RPV only.

(2) Bulk average vessel coolant temperature change in any 1-hour period.

(3) An environmental fatigue calculation provides the basis for reduced OBE cycle limits for the following piping systems: RHR Return and Supply piping – 20 cycles; Recirculation Drain piping – 20 cycles; Core Spray piping – 40 cycles (Unit 1), 30 cycles (Unit 2); and Reactor Recirculation piping – 30 cycles. All other piping has a limit of 50 OBE cycles. These administrative transient cycle limits are imposed to meet License Renewal Commitment T04740.

(4) The annual encounter probability of the one cycle events is $<10^{-2}$ for emergency and $<10^{-4}$ for faulted events.

Table 3.9-3

LIST OF COMPUTER PROGRAMS USED FOR NON-NSSS MECHANICAL SYSTEMS, COMPONENTS, AND COMPONENT SUPPORTS

COMPUTER PROGRAM No.	NAME	DOCUMENT TRACEABILITY	SYSTEM USED
ME101	Linear Elastic Analysis of Piping	Bechtel	UNIVAC 1100 series, Unix Workstation
ME632	Piping System Analysis	Bechtel	Honeywell 6000, UNIVAC 1100 series
ME912	Thermal Stress Program	Bechtel	UNIVAC 1100 series
ME913	Nuclear Class 1 Piping Stress Analysis	Bechtel	UNIVAC 1100 series
CE798	ANSYS	Swanson Analysis System, Inc. Elizabeth, Penn.	UNIVAC 1100 series
NE452	Reflood Analysis	Bechtel	UNIVAC 1100 series
NE805	Relief Valve	Bechtel Clearing Analysis	UNIVAC 1100 series
ME210	Local Stresses in Cylindrical Shells Due to External Loadings	Bechtel	UNIVAC 1100 series
ME602	Spectra Merging and Simplified Seismic Analysis	Bechtel	UNIVAC 1100 series
ME351	Pipe Rupture Analysis Program	Control Data Corporation	CDC CYBER
ME150	Frame Analysis Program for Pipe Support	Bechtel	UNIVAC 1100 series, VAX/VMS, UNIX Workstation
ME152	Standard Frame Analysis Program for Pipe Support	Bechtel	VAX/VMS, UNIX Workstation

Table 3.9-3 (Cont'd)

COMPUTER PROGRAM No.	NAME	DOCUMENT TRACEABILITY	SYSTEM USED
ME035	Base-Plate Analysis Program	Bechtel	UNIVAC 1100 series, UNIX Workstation
CE050	Concrete Expansion Anchor Bolt Program	Bechtel	UNIVAC 1100 series
CE901	Frame Analysis Program	Bechtel	UNIVAC 1100 series
ME225	Anchor Plate Program	Bechtel	UNIVAC 1100 series
ME226	Pipe Clamp Program	Bechtel	UNIVAC 1100 series
ME120	Weld Program	Bechtel	UNIVAC 1100 series
ME425	Strength Design of Pipe Support Anchor Bolt	Bechtel	UNIVAC 1100 series
ME153	Miscellaneous Application Program for Pipe Supports	Bechtel	UNIVAC 1100 series, VAX/VMS, UNIX Workstation
NUPIPE-SWPC	Linear Elastic Analysis of Piping	Stone and Webster	PC Workstation

Table 3.9-4

COMPARISON OF ME912 WITH ME643 AND ANALYTICAL RESULTS

CASE	PROGRAM	TEMPERATURE GRADIENTS[1] ΔT_1	ΔT_2	$T_a - T_b$[1]
450° F to 553° F Step	ME643	79.0	38.0	24.0
3 Inch Schedule 160, Stainless	ME912	79.7	40.6	24.3
Thicknesses 1.50:1	Reference 3.9-8	82.0	41.0	-
408° F to 100° F Step	ME643	136.2	40.1	83.0
12 Inch Schedule 80, Carbon	ME912	134.4	41.9	81.6
Steel Thicknesses 1.69:1	Reference 3.9-8	139.0	43.0	-

(1) Defined in the ASME B&PV Code, Section III, Subsection NB-3650.

Table 3.9-5

COMPARISON BETWEEN SAMPLE PROBLEM AND
COMPUTER PROGRAM ME913 RESULTS[(1)]

	ME 913	Sample Problem[(2)]
Equation 9	20,810 psi	20,825 psi
Equation 10	65,567 psi	65,596 psi
Equation 11	128,950 psi	128,920 psi
Equation 12	39,536 psi	39,564 psi
Equation 13	23,152 psi	23,155 psi
Total Usage Factor	0.3439	0.3699

(1) Comparison made for Butt-Welding Tee, Location 10.
(2) See Reference 3.9-14.

Table 3.9-6

LOADING COMBINATIONS, STRESS LIMITS, AND ALLOWABLE STRESSES

The following is a list of the tables that give the design loading combinations, allowable stresses, and calculated stresses for the major mechanical safety-related components in the plant and referenced in Section 3.9.

3.9-6(a)	Load Combinations and Acceptance Criteria for ASME Class 1, 2, and 3 NSSS Piping, Equipment, and Supports
3.9-6(b)	Reactor Internals and Associated Equipment
3.9-6(c)	RWCU Heat Exchangers
3.9-6(d)	Class 1 Main Steam Piping and Pipe-mounted Equipment
3.9-6(e)	Class 1 Recirculation Loop Piping and Pipe-mounted Equipment
3.9-6(f)	RPV and Shroud Support Assembly
3.9-6(g)	Main Steam Relief Valves
3.9-6(h)	Main Steam Isolation Valve
3.9-6(i)	Recirculation Pump
3.9-6(j)	Reactor Recirculation System Gate Valves
3.9-6(k)	HPCI Turbine
3.9-6(l)	SLCS Pump
3.9-6(m)	SLCS Tank
3.9-6(n)	ECCS Pumps
3.9-6(o)	RHR Heat Exchanger
3.9-6(p)	RWCU Pump
3.9-6(q)	RCIC Turbine
3.9-6(r)	RCIC Pump

Table 3.9-6 (Cont'd)

3.9-6(s)	Reactor Refueling and Servicing Equipment
3.9-6(t)	HPCI Pump
3.9-6(u)	Control Rod Drive
3.9-6(v)	CRD Housing
3.9-6(w)	Jet Pumps
3.9-6(x)	Fuel Assembly (Including Channel)
3.9-6(y)	LPCI Coupling
3.9-6(z)	RPV Support Equipment; CRD Housing Support
3.9-6(aa)	Control Rod Guide Tube
3.9-6(ab)	Incore Housing

Table 3.9-6(a)

LOAD COMBINATION AND ACCEPTANCE CRITERIA FOR
ASME CLASS 1, 2, AND 3 NSSS PIPING, EQUIPMENT, AND SUPPORTS

LOAD COMBINATION	DESIGN BASIS	EVALUATION BASIS	SERVICE LEVEL
N + $SRV_{(ALL)}$	Upset	Upset	(B)
N + OBE	Upset	Upset	(B)
N + OBE + $SRV_{(ALL)}$	Emergency	Upset	(B)
N + SSE + $SRV_{(ALL)}$	Faulted	Faulted	(D)[1]
N + SBA + SRV	Emergency	Emergency	(C)[1]
N + SBA + $SRV_{(ADS)}$	Emergency	Emergency	(C)[1]
N + SBA/IBA + OBE + $SRV_{(ADS)}$	Faulted	Faulted	(D)[1]
N + SBA/IBA + SSE + $SRV_{(ADS)}$	Faulted	Faulted	(D)[1]
N + LOCA[2] + SSE	Faulted	Faulted	(D)[1]

LOAD DEFINITION LEGEND

N	-	Normal loads (e.g., weight, pressure, temperature, etc)
OBE	-	Operational basis earthquake loads
SSE	-	Safe shutdown earthquake loads
SRV	-	Safety/relief valve discharge induced loads from two adjacent valves (one valve actuated when adjacent valve is cycling)
SRV_{ALL}	-	Loads induced by actuation of all 14 safety/relief valves that activate within milliseconds of each other (e.g., turbine trip operational transient)

Table 3.9-6(a) (Cont'd)

SRV_{ADS}	-	Loads induced by the actuation of all 5 safety/relief valves associated with automatic depressurization system that actuate within milliseconds of each other during the postulated small or intermediate size pipe rupture.
LOCA	-	Loss-of-coolant accident associated with the postulated pipe rupture of large pipes (e.g., main steam, feedwater, recirculation piping)
$LOCA_1$	-	Pool-swell drag/fallback loads on piping and components located between the main vent discharge outlet and the suppression pool water upper surface
$LOCA_2$	-	Pool-swell impact loads on piping and components located above the suppression pool water upper surface
$LOCA_3$	-	Oscillating pressure induced loads on structures and equipment during condensation oscillation
$LOCA_4$	-	Oscillating pressure induced loads on structures and equipment during chugging
$LOCA_5$	-	Building motion induced loads from main vent air clearing
$LOCA_6$	-	Vertical and horizontal loads on main vent piping
$LOCA_7$	-	Annulus pressurization loads
SBA	-	Abnormal transients associated with a small break accident
IBA	-	Abnormal transients associated with an intermediate break accident.

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(1) All ASME Class 1, 2 and 3 piping that are required to function for safe shutdown under the postulated events are designed to meet the requirements described in NEDO-21985.

(2) The most limiting case of load combinations among $LOCA_1$ through $LOCA_7$.

Table 3.9-6(b)

REACTOR INTERNALS AND ASSOCIATED EQUIPMENT

ASME SECTION III, SUBSECTION NG PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	ALLOWABLE STRESS (psi)	MAXIMUM CALCULATED STRESS[3] (psi)
<u>TOP GUIDE - HIGHEST STRESSED BEAM</u>				
MATERIAL 304 S.S.				
A. NORMAL AND UPSET CONDITION:				
$P_m \leq S_m$ $S_m = 16{,}900$ @ $550\,^{o}F$	Normal and Upset Condition Loads: 1. Normal loads 2. Upset pressure 3. OBE 4. SRV	Primary membrane	16,900	1,889
$P_L + P_b \leq 1.5\ S_m$ $1.5\ S_m = 25{,}350$ @ $550\,^{o}F$		Primary membrane plus bending	25,350	17,735
B. EMERGENCY CONDITION:				
$P_m \leq 1.5\ S_m$ $1.5\ S_m = 25{,}350$ @ $550\,^{o}F$	Emergency Condition Loads: 1. Normal loads 2. Upset pressure 3. Chugging 4. SRV_1	Primary membrane	25,350	326
$P_L + P_b\ 2.25\ S_m$ $2.25\ S_m = 38{,}025$ @ $550\,^{o}F$[2]		Primary membrane plus bending	38,025	12,192

Table 3.9-6(b) (Cont'd)

ASME SECTION III, SUBSECTION NG PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[3] (psi)
C. FAULTED CONDITION:				
$P_m \leq 2.4\ S_m$ $2.4\ S_m = 40{,}560$ @ 550 ^{o} F	Faulted Condition Loads: 1. Normal loads 2. Accident pressure 3. SSE 4. Jet reaction 5. Delta P	Primary membrane	40,560	3,383
$P_L + P_b \leq 3.0\ S_m$ $3.0\ S_m = 50{,}700$ @ 550 ^{o} F[2]	1. Normal loads 2. Accident pressure 3. SSE 4. SRV_1 5. Chugging	Primary membrane plus bending	50,700	34,412
D. MAXIMUM CUMULATIVE USAGE FACTOR: 0.901 at beam slot location				

Table 3.9-6(b) (Cont'd)

ASME SECTION III, SUBSECTION NG PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[3] (psi)
CORE PLATE (LIGAMENT IN TOP PLATE)				
MATERIAL: 304 S.S.				
A. NORMAL AND UPSET CONDITION				
$P_m \leq S_m$ $S_m = 16,900$ @ 550^o F	Normal and Upset Condition Loads: 1. Normal loads 2. Upset pressure 3. OBE 4. SRV	Primary membrane	16,900	8,580
$P_L + P_b \leq 1.5\ S_m$ $1.5\ S_m = 25,350$ @ 550°F		Primary membrane plus bending	25,350	15,270
B. EMERGENCY CONDITION:				
$P_m \leq 1.5\ S_m$ $1.5\ S_m = 25,350$ @ 550°F	Emergency Condition Loads: 1. Normal loads 2. Upset pressure 3. Chugging 4. SRV_{ADS}	Primary membrane	25,350	14,300
$P_L + P_b \leq 2.25\ S_m$ $2.25\ S_m = 38,030$ @ 550°F[2]		Primary membrane plus bending	38,030	7,050

Table 3.9-6(b) (Cont'd)

ASME SECTION III, SUBSECTION NG PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER(1)	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS(3) (psi)
C. FAULTED CONDITION:				
$P_m \leq 2.4\ S_m$	Faulted Condition Loads:	Primary membrane	40,560	15,650
$2.4\ S_m = 40{,}560$ @ 550^o F	1. Normal loads 2. Accident pressure 3. Jet reaction 4. SSE 5. Delta P			
$P_L + P_b \leq 3\ S_m$		Primary membrane plus bending	50,700	25,650
$3\ S_m = 50{,}700$ @ 550^o F(2)				

D. MAXIMUM CUMULATIVE USAGE FACTOR: 0.257 at Core plate stud

Table 3.9-6(b) (Cont'd)

ASME SECTION III, SUBSECTION NG PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[3] (psi)
<u>DIFFERENTIAL PRESSURE AND LIQUID CONTROL LINES</u>				
MATERIAL: 304 S.S.				
A. NORMAL AND UPSET CONDITION:				
$P_m \leq S_m$	Normal and Upset Condition Loads:			
S_m = 16,950 @ 550°F	1. OBE 2. SRV			
$P_L + P_b \leq 3\ S_m$		Primary membrane plus bending plus secondary membrane	49,200	8,654
$3\ S_m$ = 50,850 @ 550°F				
B. EMERGENCY CONDITION:				
$P_m \leq S_m$	Emergency Condition Loads:			
S_m = 16,950 @ 550°F	1. OBE 2. SRV			
$P_L + P_b \leq 2.25\ S_m$		Primary membrane plus bending	36,900	8,654
$2.25\ S_m$ = 37,120 @ 550°F[2]				
C. FAULTED CONDITION:				
$P_m \leq S_m$	Faulted Condition Loads:			
S_m = 16,950 @ 550°F	1. Annulus pressurization 2. SSE			
$P_L + P_b \leq 3.6\ S_m$		Primary membrane plus bending plus secondary membrane	59,040	15,106
$3.6\ S_m$ = 61,020 @ 550°F[2]				

(1) Load cases are defined in Table 3.9-6.
(2) Value of S_m or S_y is shown depending on the controlling criteria (e.g., 1.8 S_m or 1.5 S_y for B)
(3) The loads listed here are associated with the operating level of 3458 MWt. Per Reference 3.9-32, the loads on the differential pressure and liquid control lines are bounding at MUR power uprate condition; the loads on the top guide and core plate are increased but within the allowable limits. Increased loads due to the revised Fuel Lift Margin and CRGT lift forces under MUR conditions are addressed in Reference 3.9-33.

Table 3.9-6(c)

RWCU HEAT EXCHANGERS

REGENERATIVE RWCU HX

Part	Thickness Required (in)	Allowable Stress(psi)	Actual Thickness (in)
Shell	0.779	15900	0.875
Shell head	0.745	15900	0.760
Channel shell	0.780	15900	2.6875
Tube sheet	3.087	15900	3.25
Tubes	0.0427	11950	0.0524
Piping	0.195	15900	0.337
Channel cover	3.09	17500	3.25

NONREGENERATIVE RWCU HX

Part	Thickness Required (in)	Allowable Stress(psi)	Actual Thickness (in)
Shell	0.1171	15000	0.375
Shell head	0.117	17500	0.375
Channel shell	0.7814	15900	2.6815
Channel cover	3.09	17500	3.25
Tube sheet	3.087	13900	3.25
Tubes	0.0561	11950	0.0585
Channel piping	0.185	15900	0.337
Shell piping	0.0608	15900	0.280

Table 3.9-6(d)

ASME CODE CLASS 1 MAIN STEAM PIPING AND PIPE-MOUNTED EQUIPMENT - HIGHEST STRESS SUMMARY

(UNIT 1)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Allowable Limits	Ratio Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers
ASME Section III, NB-3600						
Design Condition: Eq. 9 ≤ 1.5 S_m	Primary	15,765	26,550	0.594	1. Pressure 2. Weight 3. OBE	Steam Line D Riser lug (009)
Service Levels A & B (Normal & Upset) Condition:						
Eq. 12 ≤ 3.0 S_m	Secondary	32,318	54,600	0.59	1. Thermal expansion	Steam Line C Sweepolet (059)
Service Levels A & B (Normal & Upset) Condition: Eq. 13 ≤ 3.0 S_m	Primary Plus Secondary (Except Thermal Expansion)	54,509	54,600	0.99	1. Pressure 2. Weight 3. OBE 4. Temperature discontinuity	Steam Line A Sweepolet (063)
Service Levels A & B (Normal & Upset) Conditions:						
Cumulative Usage Factor	N.A.	0.457	1.0		N/A	Steam Line A Sweepolet (063)
Service Level B (Upset) Condition: Eq. 9 ≤ 1.8 S_m & 1.5 S_y	Primary	22,453	32,760	0.69	1. Pressure 2. Weight 3. OBE 4. SRV (Acoustic wave)	Steam Line D Sweepolet (060)

Table 3.9-6(d) (Cont'd)

(Unit 1)

Acceptance Criria	Limiting Stress Type	Calculated Stress or Usage Factor	Allowable Limits	Ratio Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers
Service Level C (Emergency) Condition:					1. Pressure 2. Weight 3. OBE 4. Chugging	Steam Line B Sweepolet (500)
Eq. 9 < 2.25 S_m & 1.8 S_y	Primary	22,050	40,950	0.54		
Service Level D (Faulted) Condition:					1. Pressure 2. Weight 3. SSE 4. Annulus pressurization	Steam Line B Sweepolet (500)
Eq. 9 < 3.0 S_m	Primary	48,775	54,600	0.89		

Table 3.9-6(d) (Cont'd)

(Unit 1)

Component/ Load Type	Highest Calculated Load	Allowable Load	Ratio Calculated/ Allowable	Loading	Identification of Equipment with Highest Loads
Snubber					
Service Level B	44,988	50,000	0.900	OBE + SRV	Steam Line A Snubber - SA9
Service Level C	25,555	66,500	0.384	Chugging + SRV (acoustic wave)	Steam Line B Snubber - SB2
Service Level D	72,874	75,000	0.972	Annulus Pressurization + SSE	Steam Line A Snubber - SA9
Accelerations					
Horizontal Level D	6.314g	6.5g	0.9714	SSE + Condensation + SRV (acoustic wave)	Steam Line D SRV Inlet (086)
Vertical Level D	4.20g	6.0g	0.7	SSE + Condensation + SRV (acoustic wave)	Steam Line B SRV Inlet (075)

Table 3.9-6(d) (Cont'd)

ASME CODE CLASS 1 MAIN STEAM PIPING AND PIPE-MOUNTED EQUIPMENT - HIGHEST STRESS SUMMARY

(UNIT 2)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Allowable Limits	Ratio Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers
ASME Section III, NB-3650						
Design Condition: Eq. 9 $\leq$ 1.5 S_m	Primary	21,333	26,550	0.78	1. Pressure 2. Weight 3. OBE	Steam Line A Riser lug (063)
Service Levels A & B (Normal & Upset) Condition:						
Eq. 12 $\leq$ 3.0 S_m	Secondary	32,318	54,600	0.59	1. Thermal expansion	Steam Line C Sweepolet (059)
Service Levels A & B (Normal & Upset) Condition: Eq. 13 $\leq$ 3.0 S_m	Primary Plus Secondary (Except Thermal Expansion)	54,509	54,600	0.99	1. Pressure 2. Weight 3. OBE 4. Temperature discontinuity	Steam Line A Sweepolet (063)
Service Levels A & B (Normal & Upset) Conditions:						
Cumulative Usage Factor	N.A.	0.457	1.0		N/A	Steam Line A Sweepolet (063)

Table 3.9-6(d) (Cont'd)

(Unit 2)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Allowable Limits	Ratio Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers
Service Level B (Upset) Condition:					1. Pressure 2. Weight 3. OBE	Steam Line D Sweepolet (060)
Eq. 9 $\leq$ 1.8 S_m & 1.5 S_y	Primary	22,453	32,760	0.69	4. SRV (Acoustic wave)	
Service Level C (Emergency) Condition:					1. Pressure 2. Weight 3. OBE	Steam Line B Sweepolet (500)
Eq. 9 < 2.25 S_m & 1.8 S_y	Primary	22,050	40,950	0.54	4. Chugging	
Service Level D (Faulted) Condition:					1. Pressure 2. Weight 3. SSE	Steam Line B Sweepolet (500)
Eq. 9 < 3.0 S_m	Primary	48,715	54,600	0.89	4. Annulus pressurization	

Table 3.9-6(d) (Cont'd)

(Unit 2)

Component/ Load Type	Highest Calculated Load	Allowable Load	Ratio Calculated/ Allowable	Loading	Identification of Equipment with Highest Loads
Snubber					
Service Level B	44,988	50,000	0.900	OBE + SRV	Steam Line A Snubber - SA9
Service Level C	25,555	66,500	0.384	Chugging + SRV (acoustic wave)	Steam Line B Snubber - SB2
Service Level D	72,874	75,000	0.972	Annulus Pressurization + SSE	Steam Line A Snubber - SA9
Accelerations					
Horizontal Level D	6.314g	6.5g	0.9714	SSE + Condensation + SRV (acoustic wave)	Steam Line D SRV Inlet (086)
Vertical Level D	4.20g	6.0g	0.7	SSE + Condensation + SRV (acoustic wave)	Steam Line B SRV Inlet (075)

Table 3.9-6(e)

ASME CODE CLASS 1 RECIRCULATION PIPING AND PIPE-MOUNTED EQUIPMENT - HIGHEST STRESS SUMMARY

(UNIT 1)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Ratio Allowable Limits	Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers[1]
ASME Section III, NB-3650						
Design Condition: Eq. 9 $\leq$ 1.5 S_m	Primary	17,248	20,588	0.84	1. Pressure 2. Weight 3. OBE	Node (028) Recirculation Loop B (Hanger Lugs)
Service Levels A & B (Normal & Upset) Condition: Eq. 12 $\leq$ 3.0 S_m	Secondary	17,745	51,750	0.34	1. Pressure 2. Weight 3. Thermal expansion 4. OBE 5. SRV (structural feedback)	Node (500) Recirculation Loop A (Elbow)
Service Levels A & B (Normal & Upset) Condition: Eq. 13 $\leq$ 3.0 S_m	Primary plus secondary (except thermal expansion)	41,677	51,750	0.81	1. Pressure 2. Weight 3. OBE 4. SRV (structural feedback)	Node (200) Recirculation Loop B (Sweepolet)
Service Levels A & B (Normal and Upset) Condition:						
Cumulative Usage Factor	N.A.	0.25	1.0	0.25		

Table 3.9-6(e) (Cont'd)

(Unit 1)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Ratio Allowable Limits	Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers(1)
Service Level B (Upset) Condition: Eq. 9 ≤ 1.8 S_m & 1.5 S_y	Primary	19,227	23,472	0.82	1. Pressure 2. Weight 3. OBE 4. SRV (structural feedback)	Node (016) Recirculation Loop B (Small Tee)
Service Level C (Emergency) Condition: Eq. 9 ≤ 2.25 S_m & 1.8 S_y	Primary	18,886	28,166	0.67	1. Pressure 2. Weight 3. Chugging 4. SRV (structural feedback)	Node (016) Recirculation Loop B (Small Tee)
Service Level D (Faulted) Condition: Eq. 9 ≤ 3.0 S_m	Primary	24,895	31,296	0.80	1. Pressure 2. Weight 3. SSE 4. Annulus pressurization	Node (016) Recirculation Loop B (Small Tee)

Table 3.9-6(e) (Cont'd)

(Unit 1)

Component/ Load Type	Highest Calculated Load	Allowable Load	Ratio Calculated/ Allowable	Loading	Identification of Equipment with Highest Loads
Snubber Level B	104,265 lb	120,000 lb	0.87	OBE + SRV	Recirculation Loop A (SA 2)
Snubber Level C	20,022 lb	66,500 lb	0.301	Chugging + SRV	Recirculation Loop B (SB 9)
Snubber Level D	71,583 lb	75,000 lb	0.954	Annulus Pressurization + SSE	Recirculation Loop B (SB 9)
Suction Valve Level B	428,263 in-lb	1,747,285 in-lb	0.25	1. Weight 2. Thermal expansion 3. OBE 4. SRV	Recirculation Loop A (suction valve)
Discharge Valve Level C	284,213 in-lb	1,747,285 in-lb	0.16	1. Weight 2. Thermal expansion 3. OBE 4. SRV	Recirculation Loop B (discharge valve)
Discharge Valve Level D	1,438,256 in-lb	1,747,285 in-lb	0.82	1. Weight 2. Thermal expansion 3. Annulus pressurization 4. SSE	Recirculation Loop B (discharge valve)

Table 3.9-6(e) (Cont'd)

ASME CODE CLASS 1 RECIRCULATION PIPING AND PIPE-MOUNTED EQUIPMENT - HIGHEST STRESS SUMMARY

(UNIT 2)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Allowable Limits	Ratio Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers[1]
ASME Section III, NB-3650						
Design Condition: Eq. 9 $\leq$ 1.5 S_m	Primary	18,549	25,875	0.72	1. Pressure 2. Weight 3. OBE	Node (028) Recirculation Loop A (Buttweld)
Service Levels A & B (Normal & Upset) Condition: Eq. 12 $\leq$ 3.0 S_m	Secondary	40,919	51,750	0.79	1. Pressure 2. Weight 3. Thermal expansion 4. OBE 5. SRV (structural feedback)	Node (256) Recirculation Loop A (Transition)
Service Levels A & B (Normal & Upset) Condition: Eq. 13 $\leq$ 3.0 S_m	Primary plus secondary (except thermal expansion)	41,667	51,750	0.81	1. Pressure 2. Weight 3. OBE 4. SRV (structural feedback)	Node (200) Recirculation Loop B (Sweepolet)
Service Levels A & B (Normal and Upset) Condition:						
Cumulative Usage Factor	N.A.	0.098	1.0	0.098		

Table 3.9-6(e) (Cont'd)

(Unit 2)

Acceptance Criteria	Limiting Stress Type	Calculated Stress or Usage Factor	Ratio Allowable Limits	Actual/ Allowable	Loading	Identification of Locations of Highest Stress Points - NODG Point Numbers(1)
Service Level B (Upset) Condition:					1. Pressure 2. Weight 3. OBE	Node (016) Recirculation Loop B
Eq. 9 ≤ 1.8 S_m & 1.5 S_y	Primary	19,227	23,472	0.82	4. SRV (structural feedback)	Small Tee)
Service Level C (Emergency) Condition:					1. Pressure 2. Weight 3. Chugging	Node (016) Recirculation Loop B
Eq. 9 ≤ 2.25 S_m & 1.8 S_y	Primary	18,886	28,166	0.67	4. SRV (structural feedback)	(Small Tee)
Service Level D (Faulted) Condition:					1. Pressure 2. Weight 3. SSE	Node (016) Recirculation Loop B
Eq. 9 ≤ 3.0 S_m	Primary	24,895	31,296	0.80	4. Annulus pressurization	(Small Tee)

Table 3.9-6(e) (Cont'd)

(Unit 2)

Component/ Load Type	Highest Calculated Load	Allowable Load	Ratio Calculated/ Allowable	Loading	Identification of Equipment with Highest Loads
Snubber Level B	52,023 lb	120,000 lb	0.433	OBE + SRV	Recirculation Loop B (SB 2)
Snubber Level C	24,339 lb	159,600 lb	0.153	Chugging + SRV	Recirculation Loop B (SB 2)
Snubber Level D	86,272 lb	180,000 lb	0.479	Annulus Pressurization + SSE	Recirculation Loop B (SB 2)
Discharge Valve Level B	265,030 in-lb	1,747,285 in-lb	0.152	1. Weight 2. Thermal expansion 3. OBE 4. SRV	Recirculation Loop B (discharge valve)
Suction Valve Level C	250,422 in-lb	1,747,285 in-lb	0.143	1. Weight 2. Thermal expansion 3. OBE 4. SRV	Recirculation Loop B (suction valve)
Suction Valve Level D	535,499 in-lb	1,747,285 in-lb	0.307	1. Weight 2. Thermal expansion 3. Annulus pressurization 4. SSE	Recirculation Loop B (suction valve)

(1) Refer to Figure 3.6-4 for the identification of node point numbers.

Table 3.9-6(f)

RPV AND SHROUD SUPPORT ASSEMBLY

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[4] (psi)
<u>VESSEL SUPPORT SKIRT</u>				
MATERIAL: SA516, Grade 70				
A. NORMAL AND UPSET CONDITION:				
$P_m \le S_m$ S_m = 19,150 @ 575°F	Normal and Upset Condition Loads: 1. Normal loads 2. Upset pressure 3. OBE 4. SRV	Primary membrane	19,150	14,723
$P_L + P_b \le 1.5\ S_m$ 1.5 S_m = 28,725 @ 575°F		Primary membrane plus bending	28,725	20,640
B. EMERGENCY CONDITION:				
$P_m \le S_y$ S_y = 28,425 @ 546°F	Emergency Condition Loads: 1. Normal loads 2. Upset pressure 3. OBE 4. Chugging 5. SRV_{ADS}	Primary membrane	29,425	20,565
$P_L + P_b \le 1.5\ S_y$ 1.5 S_y = 44,137 @ 546°F[2]		Primary membrane plus bending	44,150	29,377

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER(1)	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS(4) (psi)
C. FAULTED CONDITION:				
$P_m \leq 2.4\ S_m$ $2.4\ S_m = 45{,}960$ @ 575°F	Faulted Condition Loads: 1. Normal loads 2. Accident pressure 3. Jet reaction 4. VC 5. SSE	Primary membrane	45,960	30,436
$P_m + P_b \leq 3.6\ S_m$ $3.6\ S_m = 68{,}940$ @ 575°F(2)	1. Normal loads 2. Accident pressure 3. Chugging 4. SRV_{ADS} 5. SSE	Primary membrane plus bending	68,940	45,044
D. MAXIMUM CUMULATIVE USAGE FACTOR: 0.83 at Skirt base junction				

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[4] (psi)
<u>SHROUD SUPPORT</u>				
MATERIAL: SB168 Inconel				
A. NORMAL AND UPSET CONDITION:				
$P_m \leq S_m$ S_m = 23,300 @ 575°F	Normal and Upset Condition Loads: 1. Dead weight 2. OBE	Primary membrane	23,300	21,700
$P_L + P_b \leq 1.5\ S_m$ 1.5 S_m = 34,950 @ 575°F		Primary membrane plus bending	34,950	33,640
B. EMERGENCY CONDITION:				
$P_m \leq S_y$ S_y = 28,120 @ 575°F	Emergency Condition Loads: 1. Dead weight 2. SSE 3. Jet reaction	Primary membrane	28,120	25,440
$P_L + P_b \leq 1.5\ S_y$ 1.5 S_y = 42,180 @ 575°F[2]		Primary membrane plus bending	42,180	40,360

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[(1)]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[(4)] (psi)
C. FAULTED CONDITION:				
$P_m \leq S_y$	Faulted Condition Loads:	Primary membrane	28,120	25,440
S_y = 28,120 @ 575°F	1. Dead weight 2. SSE 3. Jet reaction			
$P_L + P_b \leq 1.5\ S_y$		Primary membrane plus bending	42,180	40,360
1.5 S_y = 42,180 @ 575?F[(2)]				

D. MAXIMUM CUMULATIVE USAGE FACTOR: 0.373 at point 24 in shroud support plate

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[4] (psi)
RPV FEEDWATER NOZZLE				
MATERIAL: SA508 Class 1 Safe-end				
A. NORMAL AND UPSET CONDITION				
$P_m \leq S_m$ S_m = 17,700 @ 575°F	Normal and Upset Condition Loads: 1. Normal loads 2. Upset pressure 3. OBE 4. SRV	Primary membrane	17,700	16,220
$P_L + P_y \leq 1.5\ S_m$ 1.5 S_m = 26,550 @ 575°F		Primary membrane plus bending	26,550	22,930
B. EMERGENCY CONDITION				
$P_m \leq S_y$ S_y = 25,900 @ 594°F	Emergency Condition Loads: 1. Normal loads 2. Upset pressure 3. Chugging 4. SRV_{ADS}	Primary membrane	25,900	21,420
$P_L + P_b \leq 1.5\ S_y$ 1.5 S_y = 38,900 @ 594°F[2]		Primary membrane plus bending	38,850	22,400

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[4] (psi)
C. FAULTED CONDITION				
$P_m \leq 2.4\ S_m$ $2.4\ S_m$ = 42,480 @ 575°F	Faulted Condition Loads: 1. Normal loads 2. Accident pressure 3. Chugging 4. SSE 5. SRV_{ADS}	Primary membrane	42,480	23,210
$P_L + P_b \leq 1.5\ S_y$ $1.5\ S_y$ = 38,900 @ 594°F[2]		Primary membrane plus bending	38,850	33,740
D. MAXIMUM CUMULATIVE USAGE FACTOR: 0.9957 at Safe-end				

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[1]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS[4] (psi)
<u>CRD PENETRATION (Stub Tube)</u>				
MATERIAL: SB167 - Inconel				
A. NORMAL AND UPSET CONDITION:				
$P_m \leq S_m$ S = 20,000 @ 575°F	Normal and Upset Condition Loads: 1. Normal leads 2. Upset pressure 3. OBE 4. SRV	Primary membrane	20,000	5,005
$P_L + P_b \leq 1.5\ S_m$ $1.5\ S_m$ = 30,000 @ 575°F		Primary membrane plus bending	30,000	28,200
B. EMERGENCY CONDITION:				
$P_m \leq S_y$ S_y = 24,100 @ 575°F	Emergency Condition Loads: 1. Normal loads 2. Upset pressure 3. Chugging 4. SRV_{ADS}	Primary membrane	24,100	6,755
$P_L + P_b \leq 1.5\ S_y$ $1.5\ S_y$ = 36,150 @ 575°F[2]		Primary membrane plus bending	36,150	30,260

Table 3.9-6(f) (Cont'd)

ASME SECTION III, SUBSECTION NB PRIMARY STRESS LIMIT CRITERIA	LOAD CASE NUMBER[(1)]	PRIMARY STRESS TYPE	MAXIMUM ALLOWABLE STRESS (psi)	CALCULATED STRESS (psi)
C. FAULTED CONDITION:				
$P \leq 2.4\ S_m$ $2.4\ S_m$ = 48,000 @ 575°F	Faulted Condition Loads: 1. Normal loads 2. Accident pressure 3. Jet reaction 4. Scram 5. SSE	Primary membrane	48,000	7,287
$P_L + P_b \leq 3.6\ S_m$ $3.6\ S_m$ = 72,000 @ 575°F[(2)]		Primary membrane plus bending	72,000	30,260
D. MAXIMUM CUMULATIVE USAGE FACTOR: 0.153 at Stub tube				

(1) Load cases are defined in Table 3.9-6.
(2) Value of S_m or S_y is shown depending on the controlling criteria (e.g., 1.8 S_m or 1.5 S_y for B).
(3) Maximum calculated values are based on design loads because they are greater than new loads.
(4) The loads here correspond to the operating power level of 3458 MWt. Per Reference 3.9-31, the loads for the vessel support skirt, feedwater nozzle and the CRD penetration are not changed as a result of MUR power uprate. Per Reference 3.9-32, the loads on the shroud support are bounding at MUR power uprate condition.

Table 3.9-6(g)

MAIN STEAM SAFETY/RELIEF VALVES (PILOT-OPERATED)
(ASME SECTION III, 1968, Including Addenda through Summer 1970)

Topic	Method of Analysis	Target Rock 9867F Analysis	Allowable Value	Calculated
1. Body inlet and outlet flange stresses	$S_H = \frac{fM_o}{Lg^2B} + \frac{PB}{4g_o} < 1.5\ S_m$	P_D (Target Rock) = P (codes)	$1.5\ S_m$ = 29,100 psi	Inlet: S_H = 27,656 psi
Note, Topics 1 and 2: Design Pressures: P_d = 1250 psig (inlet) P_b = 500 psig (outlet)	$S_R = \frac{(4te/3+1)M_o}{Lt^2B} < 1.5\ S_m$ $S_T = \frac{YM_o}{t^2B} - Z\ S_R < 1.5\ S_m$			S_R = 15,295 psi S_T = 14,257 psi Outlet:
Analyses include applied moments of M = 409,000 in-lb (inlet) and M = 372,000 in-lb (outlet)	where: S_H = Longitudinal "hub" wall stress, psi			S_H = 15,591 psi S_R = 23,614 psi
Actual tested capability (including accelerations and moments) is as described in Topic No. 11.	S_R = Radial "flange" stress, psi S_T = Tangential "flange" stress, psi Body Material: A105 Grade II			S_T = 557 psi
The analyses also include consideration of seismic, operational, and flow reaction forces. Allowable vs. tested capabilities are provided in Topic No. 12	S_m = 19,400 psi (500°F, equivalent inlet and outlet temperature)			

Table 3.9-6(g) (Cont'd)

Topic	Method of Analysis	Target Rock 9867F Analysis	Allowable Value	Calculated
2. Inlet and outlet stud area requirements	Total cross-sectional area excees the greater of: $Am_1 = \frac{Wm_1}{S_b}$, or $Am_2 = \frac{Wm_2}{S_a}$, where: Am_1 = total required bolt (stud) area for operating condition Am_2 = total required bolt (stud) area for gasket seating	$Am_1 = \frac{Wm_1}{S_b}$ $Am_2 = \frac{Wm_2}{S_a}$ * Bolting Material: SA193 Grade B7 *Where AM (required minimum) is the greater of Am_1 and Am_2; and A_b (actual bolt area) must exceed Am.	Inlet: Am_1 (>Am_2) = 8.15 in^2 Outlet: Am = 5.39 in^2	Inlet: A_b (actual area) = 13.85in^2 Outlet: A_b = 9.68 in^2
3. Body Wall thickness	1. Valve Wall Thickness Criterion: $t_{min.} < t_A$ where: $t_{min.}$ = minimum calculated thickness requirement, including corrosion allowance. t_A = Actual wall thickness (NOTE: This $t_{min.}$ is t_m per notation of the codes.)	Section at inlet: $t_{RQD} < t_{ACT}$ Section at middle of body $t_{RQD} < t_{ACT_c}$ Section at outlet: $t_{RQD} < t_{ACT}$ Section at neck: $t_{RQD} < t_{ACT}$	t_{RQD} = 0.7 in Actual thickness greater than t_m at the section under consideration.	t_{ACT} = 1.098 in. t_{ACT_c} = 0.805 in. t_{ACT} = 1.359 in. t_{ACT} = 0.953 in.

Table 3.9-6(g) (Cont'd)

Topic	Method of Analysis	Target Rock 9867F Analysis	Allowable Value	Calculated
3. (Cont'd)	Cycle Rating: Thermal $I_t = \Sigma \frac{N_{ri}}{N_i}$	$I_t = \Sigma \frac{N_{ri}}{N_i}$ (i = 1, 2 & 3)	I_t (max.) ≤ 1.0	$I_t = 0.518$
	Fatigue Na > 2,000 cycles, as based on S_a, where S_a is defined as the larger of $S_{p_1} = (2/3)Q_p + \frac{P_{eb}}{2} + Q_{T_2} + 1.3Q_{T_1}$ or $S_{P_2} = 0.4\ Q_p + \frac{K}{2}(P_{eb}+2Q_{T_2})$ where: S_{p_1} = Fatigue stress intensity at inside surface of crotch psi S_{p_2} = Fatigue stress intensity outside surface of crotch psi	Na ≥ 2,000 cycles as based on $S_A = S_{p_2}$ ($>S_{p_3}$), where S_A (Target Rock) = S_a (codes) Uses same notation as codes	Na ≥ 2,000 cycles	Na (based on S_{p_2}) = $1.5x10^5$ cycles ∴ satisfies criterion

Table 3.9-6(g) (Cont'd)

Topic	Method of Analysis	Target Rock 7567F Analysis	Allowable Value	Calculated
4. Body to pilot base Flange Stresses (body side)	$S_{H_1} = \frac{PB_1}{4g_1} \pm \frac{6M_H}{\pi B_1 g^2_1}$ (longitude hub stress adjacent to flange) $S_{H_2} = \left(\frac{Q}{\pi B_1 t} + P\right)(Z + Y) + \frac{Et\theta_B}{B_1} + \frac{0.075\ PB_1}{g_1} \pm \frac{1.8\ M_H}{\pi B_1 g^2_1}$ (circumferential stress in hub adjacent to flange) $S_R = \frac{6(M_p + M_s)}{t^2(\pi C\text{-}nD)}$ (@ Bolt circle) $S_R = \left(\frac{Q}{\pi B_1 t} + P\right) \pm \frac{6M_s}{\pi B_1 t^2}$ (adjacent to hub) $ST = \left(\frac{Q}{\pi B_1 t} + P\right)Z \pm \left(\frac{Et\theta_B}{B_1} + \frac{1.8M_S}{\pi B_1 t^2}\right)$	$S_H < 1.5\ S_m$ $S_R < 1.5\ S_m$ $S_T < 1.5\ S_m$ P_{FD} (Target Rock) = P (codes) Material: SA-105 Grade II S_m = 19,400 psi (@ 500°F)	1.5 S_m = 29,100 psi	S_H = 19,590 psi S_R = 11,851 psi S_T = 6,364 psi
Bonnet Flange Stresses (Bonnet side)	Base side flange stresses less than body side flange stresses (Both sides see identical loads. The base side is thicker than the body side at all points.)	Same as Topic 4 Analysis	1.5 S_m = 29,000 psi	S_H 19,590 psi S_R 11,851 psi S_T 6,364 psi

Table 3.9-6(g) (Cont'd)

Topic	Method of Analysis	Target Rock 9867F Analysis	Allowable Value	Calculated
6. Base of pilot body stud area requirements	Total cross-sectional area shall exceed: $Am_1 = \frac{Wm_1}{S_b}$ where: Am_1 = Total required bolt (stud) area.	$Am_1 = \frac{Wm_1}{S_b}$ Bolting Material: SA193 Grade B16	$Am_1 = 7.10$ in^2	A_b (actual) = 9.04 in^2
7. Pilot body wall thickness	Body Wall Per ASME Section VIIII $T_m = \frac{P R_i}{S_m - 0.5 P}$	$t_m < t_a$ Material: SB-166 Grade 600 Sm = 23,300 psi (@500°F)	Bonnet Wall t_m = .085 in	Bonnet Wall t_a = 3.22 in.
10. Main disc stress	Using Roark's formulas for stress and strain, 4th edition, page 250 $S_{max} = \frac{\beta Wa^2}{t_o^2}$ where: $\beta = 1.63$ w = applied load a = radius of disc t_o = thickness at center	$S_{max} < S_m$ Material: SA182 S_m = 13,000 psi (@ 500°F)	S_m = 13,600 psi 1.55m = 19,500 psi	S_{max} = 18,180 psi
11. Seismic Capability: Stress analysis uses $F_{vertical}$ = (mass of value) • (2.0 g) and $F_{horizontal}$ = (mass of value) • (3.0 g), with concurrent 409,000 in-lb and 372,000 in-lb applied at the inlet and outlet, respectively. Valve operability has been verified by test, with applied moments of 800,000 in-lb and 600,000 in-lb at the inlet and outlet, respectively, and at equivalent acceleration levels of 6.5 g horizontal and 6.0 g vertical. Tests were per IEEE 344 (1975).				
12. Valve Loads: For comparison of calculated loadings vs. seismic capability see Table 3.9-6(e).				

Table 3.9-6(h)

MAIN STEAM ISOLATION VALVE

(UNIT 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Design of Pressure-Retaining Parts	All references are made to ASME Code for Pumps and Valves for Nuclear Power, dated November 1968. Reference the same code for explanation of the symbols used.		
Body Minimum Wall Thickness	Reference Article 452.1b(2), Nonstandard Pressure - Rated Value. Table NB 451.4 For design condition of 1,250 psig and 575ºF. The primary service rating = 655 based on a core diameter of 23 in t_m = 1.925 in (including a corrosion allowance of 0.12 in).	1.925 in	1.9375 in
Body Shape Rules	Reference Article 452.2, Body Shape Rules		
Radius of Crotch	Reference Article 452.2a(1), Radius of Crotch. Criterion: $r_2 \geq 0.3\ t_m$; r_2 = 1.0 in, t_m = 1.925 $(0.3 \cdot 1.925) = 0.578 < 1.0$; criterion satisfied.	0.578 in	1.0 in
Out of Roundness	Reference Article 452.2e. Since no ovality was built into the valve body, the requirements of this article are satisfied.	Not applicable	Not applicable
Flat wall Limitation	Reference Article 452.2g, Flat Wall Limitation. Since no flat sections were built into the valve body design, the requirements of this article are satisfied.	Not applicable	Not applicable
Primary Crotch Stress Due to Internal Pressure	Reference Article 452.3 Criterion: $P_m = (A_f/A_m + 0.5)\ P_s < S_m$		
	where A_f = 504 in^2, A_m = 58 in^2, P_s = 1,375 psig, P_m = 12,650 psi, S_m = 19,400 psi; since $S_m > P_m$ criterion satisfied.	19,400 psi	12,650 psi
Valve Body Secondary Stress	Reference Article 452.4		
Primary Plus Secondary Stress Due to Internal Pressure	Reference Article 452.4a $Q_p = C_p\ (r_i/t_e + 0.5)\ P_s\ C_a$		
	where C_p = 3, r_i = 11.625 in, P_s = 1,375 psi, t_e = 2.75 for wye-type valve C_a = 1.33 ---> Q_p = 25,965 psi		|

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Secondary Stress Due to Pipe Reaction	Reference Article 452.4b, figures 452.4b(3), 452.4b(4), 452.4b(5)		
Direct or Axial Load Effect	$P_{ed} = F_d S/G_d$ where S = 30,750, F_d = 30 in^2, G_d = 183 in^2	19,400 psi	5,040 psi
	---> P_{ed} = 5,040 psi		
Bending Load Effect	$P_{eb} = C_b F_b S/G_b$ where S = 30,750, F_b = 340 in^3,		
	i.d. = 23.25 in, r_i = 11.625, t_e = 2.75, $\bar{r}$ = 13.90 in as t_e/r = .197 > .19 ---> C_b = 1		
	$G_b = I/(r_i + t_e)$ where I = 15,028 in^4, r_i = 11.625 in,		
	t_e = 2.75 in ---> G_b = 1052 in^3		
	---> P_{eb} = 9,940 psi	19,940 psi	9,940 psi
Torsion Load Effect	Reference Article 452.4b		
	$P_{et} = 2 F_b S/G_t$ where F_b = 340 in^3, S = 30,750 psi		
	G_t = 2,162 in^3	19,400 psi	9,670 psi
	P_{et} = 9,670 psi		

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Thermal Secondary Stress at Crotch Region	Reference Article 452.4C, figures 452.4C(4), 452.4C(3), 452.4C(5)		
	$Q_T = Q_{T_1} + Q_{T_2}$		
	where Te_1 = 3 in, Q_{T_1} = 1,100 psi		
	$Q_{T_2} = C_6 C_2 \Delta T_2$ where C_2 = .21, C_6 = 220, and T_2 = 5.6		
	Q_{T_2} = 260 psi, Q_T = 1,360 psi		
	Criterion: $S_N = Q_{P_2} + P_e = 2Q_T \leq 3 S_m$		
	where Q_p = 25,965, P_e = 9,940, Q_T = 1360	58,200 psi	38,625 psi
	as 38,625 $\leq$ 58,200; criterion satisfied		
Normal Duty Valve Fatigue Requirements	Reference Article 452.5, figure 452.5(a) Criterion $N_a \geq$ 2,000 cycles		
	$S_{p_1} = 2/3\ Q_p + P_{eb_2}/2 + Q_{T_1} + 1.3\ Q_T$,		
	$S_{p_2} = 0.4\ Q_p + (K/2)_2 (P_{eb} + 2Q_T)$		
	where Q_p = 25,965, P_{eb} = 9,940 K-2,		
	Q_{T_1} = 1,160, Q_{T_2} = 260 psi		
	-->S_{p_1} = 23,970, S_{p_2} = 20,845, S_a equal to the larger of		
	S_{p_1} and S_{p_2} ---> S_a = 23,970 psi --->		
	N_a = 55,000 $\geq$ 2,000; criterion satisfied		

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Cyclic Loading Requirements at Valve Crotch	Reference Article 454 Thermal Transients Not Excluded by Code Criterion: ?(N_{ri}/N_i) < 1 Calculate the fatigue usage factor (I_t) as follows: S_n Max = $Q_p + P_{eb} + C_6 (C_3 + C_4) \Delta T_f$ max ---> S_n max = 105,810 psi for ΔT_{fi} = 90, N_{ri} = 120, N_i = 2,700 N_{ri}/N_i = 0.044 for ΔT_{fi} = 122, N_{ri} = 10, N_i = 1,600 N_{ri}/N_i = 0.006 for ΔT_{fi} = 342, N_{ri} = 8, N_i = 55 N_{ri}/N_i = 0.143 -->I_t = ?(N_{ri}/N_i) = 0.196 < 1; criterion satisfied		

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Disk Design Calculation	From Reference 3.9-5, Pages 198, 200, 201 Disk design conditions, P_S = 1,250 psi at 575ºF, S_m = 17,800 psi at 600ºF		
	Case No. 13: $S_t = \frac{3W}{4mt^2 (a^2\text{-}b^2)} [a^4 (3m + 1) + b^4 (m - 1) -$ $4m\ a^2b^2 - 4(m + 1)\ a^2b^2\ (1n(a/b))]$		
	where W = 1,250 psi, m = $\frac{10}{3}$, t = 5.625 in,		
	a = 10.75 in, b = 1.75 in, $S_{t_{13}}$ = 10,354 psi		
	Case No. 14: $S = \frac{3W}{2?\ mt^2} \left[\frac{2a^2 (m + 1)}{a^2 - b^2} 1n(a/b) + (m - 1)\right]$		
	where W = 59,044 lb_f, t = 5.625 in, m = $\frac{10}{3}$,		
	a = 10.75 in, b = 1.75 in, $S_{t_{14}}$ = 4,943 psi	17,800 psi	15,297 psi
	Case No. 21: $S_r = \frac{3W}{4t^2} \left[\frac{4a^4(m + 1)\ 1n(a/b)\ a^4(m + 3) + b^4(m - 1) + 4a^2b^2}{a^2(m + 1) + b^2(m - 1)}\right]$		
	where W = 1,250, m = 10/3, t = 3.125 in, a = 10.75 in, b = 7.25 in		
	--> $S_{r_{21}}$ = 5760 psi		
	Case No. 22: $S_r = \frac{3W}{2t^2} \left[\frac{2a^2(m + 1)\ 1n(a/b) + a^2(m - 1) - b^2(m - 1)}{a^2(m + 1) + b^2(m - 1)}\right]$		
	where W = 1,250, m = 10/3, t = 3,125, a = 10.75, b = 7.25		
	--> $S_{r_{22}}$ = 10,740 psi		
	Total stress = $S_{r_{21}} + S_{r_{22}}$ = 16,500 psi, allowable stress = 17,800 psi	17,800 psi	16,500 psi

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Tensile Stress at Thread Relief Valve Stem	Valve open $S_A = F/A_t$ where F = 31,586 lbs, A_t = 1,956 in^2, S_{max} = 16,148 psi		
	Valve Closed F = 46,342 lbs S_{max} = 23,692 psi	30,600 psi	23,692 psi
Bonnet Design Calculations Including Seismic Accelerations For SSE	Paragraph UG-34c(2) on ASME Code Section VIII		
Minimum Thickness	$P_{fd} = P + P_{eg}$, $P_{eg} = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2}$ where M = 1,292,000 in-lbs, F = 53,739 lbs, G = 24.75 in P_{eg} = 546 psi, P_{fd} = 1,796 psi $t = d\left(\frac{CR}{1.5S} + \frac{1.78\ W\ hg}{1.5\ Sd^3}\right)^{1/2}$ where C = 0.3, P_{fd} = 1,459 psi, S = 17,800 psi, hg = 2.625 in, W = 1,120,000 lbs, d = 24.75 in -->t = 4.503 in, t = 4.503 + 0.120 = 4.623 in (corrosion allowance is 0.120 in)	4.623 in	5.344 in
Reinforcement	Reference paragraph I-704.41(c) of USAS B31-7 to account for the opening for stem in the bonnet Required reinforcement d x t x 0.5 = $(d_3t_3 = d_4t_4)/2$ d_3 = 1.875, t_4 = 2.223, t_3 = 2.875, d_4 = 3 Reinforcement = 6.030 in^2 required = 6.6126 in^2 available	6.030 in^2	6.6126 in^2
Bonnet Studs Design Calculation	Reference Article E-1000 Bolt used 20 pieces of 2.652 in^2/bolts Total bolt area = 53.04 in^2		
Normal Operation	1. Pressure stress at Operating Condition $S_1 = W_{m1}/A_b$ = 21,116 psi where W_{m1} = 1,120,000 lbs and A_b = 53.04 in^2	27,700 psi	21,116 psi

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
	2. Gasket load at ambient condition with no internal pressure	35,000 psi	2,019 psi
	$S_2 = W_{m2}/A_b$ = 2,019 psi where W_{m2} = 107,065 lb_f and		
	A_b = 53.04 in^2		
	Maximum tensile stress = 21,116 psi		
	Thermal-stress is assumed negligible because the coefficients of thermal expansion of bonnet place and stud are the same.		
Body Flange Design Calculations	Reference paragraph I-704.5.1 of USAS B 317 Total flange moment under operating conditions		
	$M_O = M_D + M_G + M_T$		
	$M_D = H_D\ h_D$, $H_D = 0.785\ B^2P$, $h_D = R + 0.591$		
	where B = 21.75, P = 1,796 psi --> H_D = 667,290 lb_f, h_D = 2.813 in, M_D = 1,877,000 in-lbs		
	$M_G = H_G\ h_g$, $H_G = W - H$, $h_G = \frac{C-G}{2}$		
	where W is the higher of W_{m1} and W_{m2}		
	W_{m1} = 1,120,000 lbs		
	W_{m2} = 107,065 lbs		
	H_G = 256,392 lbs, h_G = 2.625 in -->M_G = 673,000 in-lbs		
	$M_T = H_T\ h_T$		
	H_T = 196,775 lbs, h_T = 3.375 in, M_T = 664,000 in-lb_f		
	M_o = 3,214,000 in-lbs		
	Total flange moment under gasket seating condition		
	$M_o = W(C-G)/2$, $W = S_a(A_m + A_b)/2$		
	where C = 30 in, A_b = 53.04 in^2, G = 24.75 in,		
	A_m = 32.857 in^2, s_a = 35,000 psi at 100ºF		
	--->W = 1,503,193 lbs ---> M_o = 3,945,895 in-lbs		

Table 3.9-6(h) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Longitudinal Hub Stress	$S_H = fM_o/((LS_1)^2B) + PB/(4g_o) = 22{,}317$ psi $< 1.5\ S_{fo} = 26{,}700$ psi	26,700 psi	22,317 psi
Radial Stress	Reference UA-51 (1), Equation (7) of Section VIII of ASME B&PV Code, 1971 Edition		
	$S_R = \frac{(1.33\ t_e+1)M_o}{Lt^2B} = 13{,}132$ psi $< 1.5\ S_{fo} = 26{,}700$ psi	26,700 psi	13,132 psi
Tangential	$S_T = \frac{(YM_o)}{t^2B} - ZS_R = 7{,}563$ psi $< 1.5\ S_{fo} = 26{,}700$		
	where Y = 4.5, t = 4.125 in, z = 2.4, B = 21.75 in	26,700 psi	7,563 psi

Table 3.9-6(h) (Cont'd)

MAIN STEAM ISOLATION VALVE

(UNIT 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Design of Pressure-Retaining Parts	All references are made to ASME Code for Pumps and Valves for Nuclear Power, dated November 1968. Reference the same code for explanation of the symbols used.		
Body Minimum Wall Thickness	Reference Article 452.1b(2), Nonstandard Pressure - Rated Value. Table NB 451.4 For design condition of 1,250 psig and 575ºF. The primary service rating = 655 based on a core diameter of 23 in t_m = 1.925 in (including a corrosion allowance of 0.12 in).	1.925 in	1.9375 in
Body Shape Rules	Reference Article 452.2, Body Shape Rules		
Radius of Crotch	Reference Article 452.2a(1), Radius of Crotch. Criterion: $r_2 \geq 0.3\ t_m$; r_2 = 1.0 in, t_m = 1.925 $(0.3 \cdot 1.925) = 0.578 < 1.0$; criterion satisfied.	0.578 in	1.0 in
Out of Roundness	Reference Article 452.2e. Since no ovality was built into the valve body, the requirements of this article are satisfied.	Not applicable	Not applicable
Flat wall Limitation	Reference Article 452.2g, Flat Wall Limitation. Since no flat sections were built into the valve body design, the requirements of this article are satisfied.	Not applicable	Not applicable
Primary Crotch Stress Due to Internal Pressure	Reference Article 452.3 Criterion: $P_m = (A_f/A_m + 0.5)\ P_s < S_m$		
	where A_f = 504 in^2, A_m = 58 in^2, P_s = 1,375 psig, P_m = 12,650 psi, S_m = 19,400 psi; since $S_m > P_m$, criterion satisfied.	19,400 psi	12,650 psi
Valve Body Secondary Stress	Reference Article 452.4		
Primary Plus Secondary Stress Due to Internal Pressure	Reference Article 452.4a $Q_p = C_p\ (r_i/t_e + 0.5)\ P_s\ C_a$		
	where C_p = 3, r_i = 11.625 in, P_s = 1,375 psi, t_e = 2.75 for wye-type valve C_a = 1.33 ---> Q_p = 25,965 psi		

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Secondary Stress Due to Pipe Reaction	Reference Article 452.4b, figures 452.4b(3), 452.4b(4), 452.4b(5)		
Direct or Axial Load Effect	$P_{ed} = F_dS/G_d$ where S = 30,750, F_d = 30 in^2, G_d = 183 in^2	19,400 psi	5,040 psi
	---> P_{ed} = 5,040 psi		
Bending Load Effect	$P_{eb} = C_b\ F_bS/G_b$ where S = 30,750, F_b = 340 in^3,		
	i.d. = 23.25 in, r_i = 11.625, t_e = 2.75, $\bar{r}$ = 13.90 in as t_e/r = .197 > .19 ---> C_b = 1		
	$G_b = I/(r_i + t_e)$ where I = 15,028 in^4, r_i = 11.625 in,		
	t_e = 2.75 in ---> G_b = 1052 in^3		
	---> P_{eb} = 9,940 psi	19,940 psi	9,940 psi
Torsion Load Effect	Reference Article 452.4b		
	$P_{et} = 2\ F_bS/G_t$ where F_b = 340 in^3, S = 30,750 psi		
	G_t = 2,162 in^3	19,400 psi	9,670 psi
	P_{et} = 9,670 psi		

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Thermal Secondary Stress at Crotch Region	Reference Article 452.4C, figures 452.4C(4), 452.4C(3), 452.4C(5)		
	$Q_T = Q_{T_1} + Q_{T_2}$		
	where $Te_1 = 3$ in, $Q_{T_1} = 1,100$ psi		
	$Q_{T_2} = C_6 C_2 \Delta\, T_2$ where $C_2 = .21$, $C_6 = 220$, and $T_2 = 5.6$		
	$Q_{T_2} = 260$ psi, $Q_T = 1,360$ psi		
	Criterion: $S_N = Q_{P_2} + P_e = 2Q_T \leq 3\, S_m$		
	where $Q_p = 25,965$, $P_e = 9,940$, $Q_T = 1360$	58,200 psi	38,625 psi
	as $38,625 \leq 58,200$; criterion satisfied		
Normal Duty Valve Fatigue Requirements	Reference Article 452.5, figure 452.5(a) Criterion $N_a \geq 2,000$ cycles		
	$S_{p_1} = 2/3\, Q_p + P_{eb_2}/2 + Q_{T_1} + 1.3\, Q_T$,		
	$S_{p_2} = 0.4\, Q_p + (K/2)_2 (P_{eb} + 2Q_T)$		
	where $Q_p = 25,965$, $P_{eb} = 9,940$ K-2,		
	$Q_{T_1} = 1,160$, $Q_{T_2} = 260$ psi		
	-->$S_{p_1} = 23,970$, $S_{p_2} = 20,845$, S_a equal to the larger of		
	S_{p_1} and S_{p_2} ---> $S_a = 23,970$ psi --->		
	$N_a = 55,000 \geq 2,000$; criterion satisfied		

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Cyclic Loading Requirements at Valve Crotch	Reference Article 454 Thermal Transients Not Excluded by Code Criterion: ?(N_{ri}/N_i) < 1 Calculate the fatigue usage factor (I_t) as follows: S_n Max = $Q_p + P_{eb} + C_6 (C_3 + C_4) \Delta T_f$ max ---> S_n max = 105,810 psi for ΔT_{fi} = 90, N_{ri} = 120, N_i = 2,700 N_{ri}/N_i = 0.044 for ΔT_{fi} = 122, N_{ri} = 10, N_i = 1,600 N_{ri}/N_i = 0.006 for ΔT_{fi} = 342, N_{ri} = 8, N_i = 55 N_{ri}/N_i = 0.143 -->I_t = ?(N_{ri}/N_i) = 0.196 < 1; criterion satisfied		

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Disk Design Calculation	From Reference 3.9-5, Pages 198, 200, 201 Disk design conditions, P_S = 1,250 psi at 575°F, S_m = 17,800 psi at 600°F		
	Case No. 13: $S_t = \frac{3W}{4mt^2(a^2-b^2)}[a^4(3m+1) + b^4(m-1) -$ $4m a^2b^2 - 4(m+1) a^2b^2 (1n(a/b))]$ where W = 1,250 psi, m = $\frac{10}{3}$, t = 5.625 in, a = 10.75 in, b = 1.75 in, $S_{t_{13}}$ = 10,354 psi		
	Case No. 14: $S = \frac{3W}{2?\, mt^2}\left[\frac{2a^2(m+1)}{a^2-b^2} 1n(a/b) + (m-1)\right]$ where W = 59,044 lb_f, t = 5.625 in, m = $\frac{10}{3}$, a = 10.75 in, b = 1.75 in, $S_{t_{14}}$ = 4,943 psi	17,800 psi	15,297 psi
	Case No. 21: $S_r = \frac{3W}{4t^2}\left[\frac{4a^4(m+1) 1n(a/b) a^4(m+3) + b^4(m-1) + 4a^2b^2}{a^2(m+1) + b^2(m-1)}\right]$ where W = 1,250, m = 10/3, t = 3.125 in, a = 10.75 in, b = 7.25 in --> $S_{r_{21}}$ = 5760 psi		
	Case No. 22: $S_r = \frac{3W}{2t^2}\left[\frac{2a^2(m+1) 1n(a/b) + a^2(m-1) - b^2(m-1)}{a^2(m+1) + b^2(m-1)}\right]$ where W = 1,250, m = 10/3, t = 3,125, a = 10.75, b = 7.25 --> $S_{r_{22}}$ = 10,740 psi		
	Total stress = $S_{r_{21}} + S_{r_{22}}$ = 16,500 psi, allowable stress = 17,800 psi	17,800 psi	16,500 psi

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Tensile Stress at Thread Relief Valve Stem	Valve open $S = F/A_t$ where F = 31,586 lbs, A_t = 1,956 in^2, S_{max} = 16,148 psi		
	Valve Closed F = 46,342 lbs S_{max} = 23,692 psi	30,600 psi	23,692 psi
Bonnet Design Calculations Including Seismic Accelerations For SSE	Paragraph UG-34c(2) on ASME Code Section VIII		
Minimum Thickness	$P_{fd} = P + P_{eg}$, $P_{eg} = \frac{16M}{\pi G^3} + \frac{4F}{\pi G^2}$ where M = 1,292,000 in-lbs, F = 53,739 lbs, G = 24.75 in P_{eg} = 546 psi, P_{fd} = 1,796 psi $t = d\left(\frac{CR}{1.5S} + \frac{1.78\ W\ hg}{1.5\ S d^3}\right)^{½}$ where C = 0.3, P_{fd} = 1,459 psi, S = 17,800 psi, hg = 2.625 in, W = 1,120,000 lbs, d = 24.75 in -->t = 4.503 in, t = 4.503 + 0.120 = 4.623 in (corrosion allowance is 0.120 in)	4.623 in	5.344 in
Reinforcement	Reference paragraph I-704.41(c) of USAS B31-7 to account for the opening for stem in the bonnet Required reinforcement d x t x 0.5 = $(d_3t_3 = d_4t_4)/2$ d_3 = 1.875, t_4 = 2.223, t_3 = 2.875, d_4 = 3 Reinforcement = 6.030 in^2 required = 6.6126 in^2 available	6.030 in^2	6.6126 in^2
Bonnet Studs Design Calculation	Reference Article E-1000 Bolt used 20 pieces of 2.652 in^2/bolts Total bolt area = 53.04 in^2		
Normal Operation	1. Pressure stress at Operating Condition $S_1 = W_{m1}/A_b$ = 21,936 psi where W_{m1} = 1,163,504 lbs and A_b = 53.04 in^2	27,700 psi	21,936 psi

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
	2. Gasket load at ambient condition with no internal pressure	35,000 psi	2,019 psi
	$S_2 = W_{m2}/A_b$ = 2,019 psi where W_{m2} = 107,065 lb_f and		
	A_b = 53.04 in^2		
	Maximum tensile stress = 21,116 psi		
	Thermal-stress is assumed negligible because the coefficients of thermal expansion of bonnet place and stud are the same.		
Body Flange Design Calculations	Reference paragraph I-704.5.1 of USAS B 317 Total flange moment under operating conditions		
	$M_O = M_D + M_G + M_T$		
	$M_D = H_D\ h_D$, $H_D = 0.785\ B^2P$, $h_D = R + 0.591$		
	where B = 21.75, P = 1,865 psi --> H_D = 692,927 lb_f, h_D = 2.813 in, M_D = 1,949,204 in-lbs		
	$M_G = H_G\ h_g$, $H_G = W - H$, $h_G = \frac{C-G}{2}$		
	where W is the higher of W_{m1} and W_{m2}		
	W_{m1} = 1,163,504 lbs		
	W_{m2} = 107,065 lbs		
	H_G = 266,242 lbs, h_G = 2.625 in -->M_G = 698,885 in-lbs		
	$M_T = H_T\ h_T$		
	H_T = 204,335 lbs, h_T = 3.375 in, M_T = 689,631 in-lb_f		
	M_o = 3,337,720 in-lbs		
	Total flange moment under gasket seating condition		
	$M_o = W(C-G)/2$, $W = S_a(A_m + A_b)/2$		
	where C = 30 in, A_b = 53.04 in^2, G = 24.75 in,		
	A_m = 32.857 in^2, s_a = 35,000 psi at 100ºF		
	--->W = 1,503,193 lbs ---> M_o = 3,945,895 in-lbs		

Table 3.9-6(h) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS, (psi) MINIMUM THICKNESS (in), OR MINIMUM AREA (in^2)	CALCULATED STRESS, (psi) ACTUAL THICKNESS (in), OR MINIMUM AREA (in^2)
Longitudinal Hub Stress	$S_H = fM_o/((LS_1)^2B) + PB/(4g_o)$ = 23,031 psi < 1.5 S_{fo} = 27,900 psi	27,900 psi	23,031 psi
Radial Stress	Reference UA-51 (1), Equation (7) of Section VIII of ASME B&PV Code, 1971 Edition		
	$S_R = \frac{(1.33\ t_e+1)M_o}{Lt^2B}$ = 13,637 psi < 1 S_{fo} = 18,600 psi	18,600 psi	13,637 psi
Tangential	$S_T = \frac{(YM_o)}{t^2B} - ZS_R$ = 7,855 psi < 1 S_f = 18,600		
	where Y = 4.5, t = 4.125 in, z = 2.4, B = 21.75 in	18,600 psi	7,855 psi

Table 3.9-6(i)

RECIRCULATION PUMP

STRESS OR CRITERIA	METHOD OF ANALYSIS	ANALYTICAL RESULTS	ALLOWABLE ACTUAL THICKNESS
1. Casing Minimum Wall Thickness	$t = \frac{PR}{SE - 0.6P} + C$		
A. Loads: Normal and upset condition Design pressure and temperature	where: t = minimum required thickness, in P = design pressure, psig R = maximum internal radius, in S = allowable working stress, psi E = joint efficiency C = corrosion allowance, in	t = 2.69 in	S_{allow} = 15,075 psi t_{act} = 3.00 in
B. Primary membrane stress limit Allowable working stress per ASME Section III, Class 1			
2. Casing Cover Minimum Thickness			
A. Loads: Normal and upset condition Design pressure and temperature	Bending Stress $S_B \text{ max.} = \frac{KQa^2}{h^2}$	S_B max = 5950 psi	$S_{B_{allow}}$ = 15,075 psi t_{act} = 7 in

Table 3.9-6(i) (Cont'd)

CRITERIA		METHOD OF ANALYSIS	ANALYTICAL RESULTS	ALLOWABLE STRESS OR ACTUAL THICKNESS
B.	Primary membrane stress limit Allowable working stress per Section III, Class 1	where: Q = design pressure a = outer radius b = inner radius $\frac{a}{b} = K = 0.711$ h = thickness		
		Shear Stress S_S max = F/A_S where: F = force A_S = area	S_S max = 3380 psi	$S_{S_{allow}}$ = 8750 psi t_{act} = 3.5 in
3.	Pump Discharge Nozzle Stress (Pressure Pending, Axial and Torsional)	Pressure $P_P = \frac{SEt}{R + 0.6t}$ where: S = allowable stress E = joint efficiency t = thickness R = inside radius P_P = pressure load	P_P = 1594 psi P_{eb} = 18,500 psi	1.5 S_m = 28,837 psi (S_m = 19,225 psi)
A.	Loads: Normal and upset condition Design pressure and temperature piping reaction during normal operation	Bending $P_{eb} = \frac{C_b F_b S}{G_b}$	P_{ed} = 7670 psi P_{et} = 16,000 psi	

Table 3.9-6(i) (Cont'd)

CRITERIA	METHOD OF ANALYSIS	ANALYTICAL RESULTS	ALLOWABLE STRESS OR ACTUAL THICKNESS
B. Combined Stress limit: 1.5 S_m per ASME Code for Pumps and Valves for Nuclear Power Class I	Axial $P_{ed} = \frac{F_d S}{G_d}$ Torsional $P_{et} = \frac{2F_b S}{G_t}$		
4. Cover and Seal Flange Bolt Areas A. Loads: Normal and upset condition Design pressure and temperature Design gasket load B. Bolting Stress Limit Allowable working stress per ASME Section III, Class C	Bolting loads, areas and stresses shall be calculated in accordance with "Rules for bolted Flange Connections" - ASME Section VIII, Paragraph UA-49.	Cover Flange Bolts S_{act} = 19,050 psi A_m = 90.2 in^2 Seal Flange Bolts S_{act} = 18,000 psi A_m = 9.85 in^2	S_{allow} = 20,000 psi A_{act} = 101.0 in^2 S_{allow} = 20,000 psi A_{act} = 11.1 in^2

Table 3.9-6(i) (Cont'd)

CRITERIA	METHOD OF ANALYSIS	ANALYTICAL RESULTS	ALLOWABLE STRESS OR ACTUAL THICKNESS
5. Cover Clamp Flange Thickness A. Loads: Normal and upset condition Design pressure and temperature Design gasket load Design bolting load B. Tangential Flange Stress Limit Allowable working stress per ASME Section III, Class C	Flange thickness and stress shall be calculated in accordance with "Rules for Bolted Flange Connections" - ASME Section VIII, Paragraph UA-49.	Flange Thickness and Stress t = 8.9 in	t_{act} = 9.25 in S_{allow} = 17,500 psi
6. Seal Compartment Wall Thickness A. Loads: Normal and upset condition Design pressure and temperature B. Primary membrane stress limit Allowable working stress per ASME Section III, Class C	$t = \frac{PR}{SE - 0.6P} + C$ where: t = minimum required thickness, in P = design pressure, psig R = maximum internal radius, in S = allowable working stress, psi E = joint efficiency C = corrosion allowance, in	t = 0.741	S_m = 15,075 psi t_{act} = 1.375 in

Table 3.9-6(i) (Cont'd)

CRITERIA	METHOD OF ANALYSIS	ANALYTICAL RESULTS	ALLOWABLE STRESS OR ACTUAL THICKNESS
7. Stresses Due to Seismic Loads A. Loads: Operation pressure and temperature SSE horizontal seismic force = 2.27 g SSE vertical seismic force = 1.39 g B. Combined Stress Limit: Yield stress per ASME Section VIII	The flooded pump-motor assembly shall be analyzed as a free body supported by constant support hangers from the pump brackets. Horizontal and vertical seismic forces shall be applied at mass center of assembly and equilibrium reactions shall be determined for the motor and pump brackets. Load, shear, and moment diagrams shall be constructed using live loads, dead loads, and calculated snubber reactions. Combined bending tension and shear stresses shall be determined for each major component of the assembly including motor support barrel, bolting and pump casing. The maximum combined tensile stress in the cover bolting shall be calculated using tensile stresses determined from loading diagram plus tensile stress from operating pressure.	Motor Bolt Tensile Stress S_{act} = 22,471 psi Pump Cover Bolt Tensile stress S_{act} = 19,417 psi Motor Support Barrel Combined Stress S_{act} = 3307 psi	 S_{allow} = 30,800 psi S_{allow} = 32,000 psi S_{allow} = 22,400 psi

Table 3.9-6(j)

REACTOR RECIRCULATION SYSTEM GATE VALVES, STRUCTURAL AND MECHANICAL LOADING CRITERIA

(UNIT 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	ALLOWABLE DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
I. 28" DISCHARGE VALVE			
Body and Bonnet			
Loads: Design pressure, design temperature, pipe reaction, thermal effects	System requirement not specified	1525 psi 575?F	1525 psi 575?F
Pressure rating, psi	Used ASME Section III Figure NB-3545.1-2	$p_r = 800$	$p_r = 800$
Minimum wall thickness, inches	Used ASME Section Para NB-3542	$t_m \geq 2.1164$ in	$t_m = 2.25$ minimum
Primary membrane stress, psi	Used ASME Section III Para NB-3545.1	$P_m \leq S_m$ (500°F) = 19,600 psi	$P_m = 11,068$ psi
Secondary stress due to pipe reaction	Used ASME Section III Para NB-3545.2 (b)(1) (S = 30,000 psi)	$P_e \leq 1.5\ S_m$ (500°F) 1.5 (16,800) = 25,200 psi	$P_{ed} = 5,580$ psi $P_{eb} = 12,702$ psi $P_{et} = 12,277$ psi $P_e = P_{eb} = 12,702$ psi
Primary plus secondary stress due to internal pressure	Used ASME Section III Para NB-3545.2(a)(1)	$S_n \leq 3\ S_m$ (500°F) = 58,800 psi	$Q_p = 24,284$ psi
Thermal secondary stress	Used ASME Section III Para NB-3545.2(c)	$S_n \leq 3\ S_m$ (500°F) = 58,800 psi	$Q_T = 5,409$ psi
Sum of primary plus secondary stress	Used ASME Section III Para NB-3545.2	$S_n \leq 3\ S_m$ (500°F) = 58,800 psi	$S_n = Q_p + P_e + 2Q_T$ = 47,804 psi

Table 3.9-6(j) (Cont'd)

(Unit 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE(1)	ALLOWABLE DESIGN VALUE(2)	ACTUAL DESIGN VALUE(2)
Fatigue requirements	Used ASME Section III Para NB-3545.3	$N_a \geq 2{,}000$ cycles	$N_a > 10^5$ cycles
Cyclic rating	Used ASME Section III Para NB-3550	$I_t \leq 1$	$I_t = 0.00387$
Body-to-Bonnet Bolting			
Loads: Design pressure and temperature, gasket loads, seismic, hydrodynamic	Used ASME Section III Para NB-3647.1		
Bolt area	Used ASME Section III Para NB-3647.1	$A_b \geq 44.41$ in^2 ($S_b \leq 27{,}975$ psi)	$A_b = 55.86$ in^2 $S_b = 23{,}437$ psi
Body flange stresses	Used ASME Section III Para NB-3647.1		
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (575°F) = 28,838 psi $S_R \leq\ S_m$ (575°F) = 19,225 psi $S_T \leq\ S_m$ (575°F) = 19,225 psi	$S_H = 27{,}854$ psi $S_R = 8{,}220$ psi $S_T = 9{,}270$ psi
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_R \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_T \leq 1.5\ S_m$ (100°F) = 30,000 psi	$S_H = 29{,}981$ psi $S_R = 11{,}671$ psi $S_T = 12{,}972$ psi
Bonnet flange			
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (575°F) = 28,838 psi $S_R \leq\ S_m$ (575°F) = 19,225 psi $S_T \leq\ S_m$ (575°F) = 19,225 psi	$S_H = 27{,}854$ psi $S_R = 8{,}220$ psi $S_T = 9{,}270$ psi
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_R \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_T \leq 1.5\ S_m$ (100°F) = 30,000 psi	$S_H = 29{,}981$ psi $S_R = 11{,}671$ psi $S_T = 12{,}972$ psi

Table 3.9-6(j) (Cont'd)

(Unit 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	ALLOWABLE DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Stresses in Stem			
Not applicable valve is passive			
Disc Analysis			
Loads: Maximum differential pressure			
Maximum stress in the disc	Calculate maximum stress according to table 10 of Reference 3.9-5	$S_{max} < 1.5\ S_m$ (500°F) = 28,500	S_{max} = 22,885 psi
Yoke and Yoke Connections			
Loads: Seismic and hydrodynamic	Calculate stresses in the yoke and yoke connections to acceptable structural analysis methods.		

Table 3.9-6(j) (Cont'd)

(Unit 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	ALLOWABLE DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Stress in yoke legs bolts		$S_{max} \leq S_m$ = 28,800 psi (500°F)	S_{max} = 9,767 psi
Stress of yoke legs		$S_{max} \leq S_m$ = 19,400 psi (500°F)	S_{max} = 13,949 psi
Stress of yoke-bonnet connection		$S_{max} \leq S_m$ = 19,225 psi (575°F)	S_{max} = 15,650 psi
II. 28" SUCTION VALVE			
<u>Body and Bonnet</u>			
Loads: Design pressure, design temperature, pipe reaction thermal effects	System requirement not specified	1,275 psi 575°F	1,275 psi 575?F
Pressure rating, psi	Used ASME Section III Figure NB-3545.1-2	p_r = 668	p_r = 668
Minimum wall thickness, inches	Used ASME Section III Para NB-3542	$t_m \geq$ 1.7724 in	t_m = 2.0 minimum
Primary membrane stress, psi	Used ASME Section III Para NB-3545.1	$P_m \leq S_m$ (500°F) = 19,600 psi	P_m = 9,275 psi
Secondary Stress due to pipe reaction	Used ASME Section III Para NB-3545.2(b) (1) (S = 30,000 psi)	P_e < 1.5 S_m (500°F) 1.5 (16,800) = 25,200 psi	P_{ed} = 5,318 psi P_{eb} = 11,980 psi P_{et} = 11,575 psi P_e = 11,980 psi
Primary plus secondary stress due to internal pressure	Used ASME Section III Para NB-3545.2(a)(1)	$S_n \leq$ 3 S_m (500°F) - 58,800 psi	Q_p = 20,580 psi
Thermal secondary stress	Used ASME Section III Para NB-3545.2	$S_n \leq$ 3 S_m (500°F) - 58,800 psi	Q_t = 5,489 psi

Table 3.9-6(j) (Cont'd)

(Unit 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE(1)	ALLOWABLE DESIGN VALUE(2)	ACTUAL DESIGN VALUE(2)
Sum of primary plus secondary stress	Used ASME Section III Para NB-3545.2	$S_n \leq 3\ S_m$ (500°F) = 58,800	$S_n = Q_p + P_e + 2Q_t$ = 43,538 psi
Fatigue Requirements	Used ASME Section III Para NB-3545.3	$N_a \geq$ 2,000 cycles	$N_a = >10^6$ cycles
Cyclic rating	Used ASME Section III Para NB-3550	$I_t \leq 1$	I_t = 0.00274
<u>Body-to-Bonnet Bolting</u>			
Loads: Design pressure and temperature, gasket loads, seismic, and hydrodynamic	Used ASME Section III Para NB-3647.1		
Bolt area	Used ASME Section III Para NB-3647.1	$A_b \geq 37.53\ in^2$ $S_b \leq$ 27,975 psi (575°F)	$A_b = 55.86\ in^2$ S_b = 19,470 psi
Body flange stresses	Used ASME Section III Para NB-3647.1		
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$(575°F) = 28,838 psi $S_R \leq\ S_m$(575°F) = 19,225 psi $S_T \leq\ S_m$(575°F) = 19,225 psi	S_H = 24,456 psi S_R = 6,539 psi S_T = 8,718 psi
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$(100°F) = 30,000 psi $S_R \leq 1.5\ S_m$(100°F) = 30,000 psi $S_T \leq 1.5\ S_m$(100°F) = 30,000 psi	S_H = 28,945 psi S_R = 10,253 psi S_T = 13,619 psi
Bonnet flange			
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$(575°F) = 28,838 psi $S_R \leq\ S_m$(575°F) = 19,225 psi $S_T \leq\ S_m$(575°F) = 19,225 psi	S_H = 24,456 psi S_R = 6,539 psi S_T = 8,718 psi

Table 3.9-6(j) (Cont'd)

(Unit 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	ALLOWABLE DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m(100°F) = 30{,}000$ psi $S_R \leq 1.5\ S_m(100°F) = 30{,}000$ psi $S_T \leq 1.5\ S_m(100°F) = 30{,}000$ psi	$S_H = 28{,}945$ psi $S_R = 10{,}253$ psi $S_T = 13{,}619$ psi
Stress in Stem			
Not applicable valve is passive			
Disc Analysis			
Loads: Maximum differential pressure			
Maximum stress in the disc	Calculate maximum stress according to table 10 of Reference 3.9-5	$S_{max} \leq 1.5\ S_m(500°F) = 28{,}500$ psi	$S_{max} = 19{,}418$ psi

Table 3.9-6(j) (Cont'd)

(Unit 1)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	ALLOWABLE DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Yoke and Yoke Connections			
Loads: Seismic and hydrodynamic	Calculate stresses in the yoke and yoke connections to acceptable structural analysis methods		
Stress in yoke legs bolts		$S_{max} \leq S_m$ (500°F) = 28,800 psi	S_{max} = 5,247 psi
Stress at yoke legs		$S_{max} \leq S_m$ (500°F) = 19,400 psi	S_{max} = 7,372 psi
Stress at yoke-bonnet connection		$S_{max} \leq S_m$ (575°F) = 19,225 psi	S_{max} = 8,248 psi

Table 3.9-6(j) (Cont'd)

REACTOR RECIRCULATION SYSTEM GATE VALVES, STRUCTURAL AND MECHANICAL LOADING CRITERIA

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	REQUIRED DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
III. 28" DISCHARGE VALVE			
Body and Bonnet			
Loads: Design pressure, design temperature, pipe reaction, thermal effects	System requirement not specified	1525 psi 575°F	1525 psi 575°F
Pressure rating, psi	Used ASME Section III Figure NB-3545.1-2	$p_r = 800$	$p_r = 800$
Minimum wall thickness, inches	Used ASME Section III Para NB-3542	$t_m \geq 2.1164$ in	$t_m = 2.25$ minimum
Primary membrane stress, psi	Used ASME Section III Para NB-3545.1	$P_m \leq S_m$ (500°F) = 19,600 psi	$P_m = 11,068$ psi
Secondary stress due to pipe reaction	Used ASME Section III Para NB-3545.2 (b)(1) (S = 30,000 psi)	$P_e \leq 1.5\ S_m$ (500°F) 1.5 (16,800) = 25,200 psi	$P_{ed} = 5,580$ psi $P_{eb} = 12,702$ psi $P_{et} = 12,277$ psi $P_e = P_{eb} = 12,702$ psi
Primary plus secondary stress due to internal pressure	Used ASME Section III Para NB-3545.2(a)(1)	$S_n \leq 3\ S_m$ (500°F) = 58,800 psi	$Q_p = 24,284$ psi
Thermal secondary stress	Used ASME Section III Para NB-3545.2(c)	$S_n \leq 3\ S_m$ (500°F) = 58,800 psi	$Q_T = 5,409$ psi
Sum of primary plus secondary stress	Used ASME Section III Para NB-3545.2	$S_n \leq 3\ S_m$ (500°F) = 58,800 psi	$S_n = Q_p + P_e + 2Q_T$ = 47,804 psi

Table 3.9-6(j) (Cont'd)

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	REQUIRED DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Fatigue requirements	Used ASME Section III Para NB-3545.3	$N_a \geq 2,000$ cycles	$N_a > 10^5$ cycles
Cyclic rating	Used ASME Section III Para NB-3550	$I_t \leq 1$	$I_t = 0.00387$
Body-to-Bonnet Bolting			
Loads: Design pressure and temperature gasket loads, stem operational load, seismic load (SSE)	Used ASME Section III Para NB-3647.1		
Bolt area	Used ASME Section III Para NB-3647.1	$A_b \geq 44.41$ in^2 ($S_b \leq 27,975$ psi (575°F)	$A_b = 55.86$ in^2 $S_b = 22,619$ psi
Body flange stresses	Used ASME Section III Para NB-3647.1		
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (500°F) = 28,838 psi $S_R \leq 1.5\ S_m$ (500°F) = 19,225 psi S_T ? 1.5 S_m (500°F) = 19,225 psi	$S_H = 27,049$ psi $S_R = 7,934$ psi $S_T = 8,946$ psi
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_R \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_T \leq 1.5\ S_m$ (100°F) = 30,000 psi	$S_H = 29,981$ psi $S_R = 11,671$ psi $S_T = 12,972$ psi
Bonnet flange			
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (500°F) = 28,838 psi $S_R \leq 1.5\ S_m$ (500°F) = 19,225 psi $S_T \leq 1.5\ S_m$ (500°F) = 19,225 psi	$S_H = 27,049$ psi $S_R = 7,934$ psi $S_T = 8,946$ psi
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_R \leq 1.5\ S_m$ (100°F) = 30,000 psi $S_T \leq 1.5\ S_m$ (100°F) = 30,000 psi	$S_H = 29,981$ psi $S_R = 11,671$ psi $S_T = 12,972$ psi

Table 3.9-6(j) (Cont'd)

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE	REQUIRED DESIGN VALUE[(2)]	ACTUAL DESIGN VALUE[(2)]
Stresses in Stem			
Not applicable, valve is passive			
Disc Analysis			
Loads: Maximum differential pressure			
Maximum stress in the disc	Calculate maximum stress according to table 10 of Reference 3.9-5	S_{max} < 1.5 S_m (500°F) = 28,500	S_{max} = 22,885 psi
Yoke and Yoke Connections			
Loads: Stem operational load	Calculate stresses in the yoke and yoke connections to acceptable structural analysis methods.		
Tensile stress in yoke legs bolts		$S_{max} \le S_m$ = 28,800 psi (500°F)	S_{max} = 3,716 psi
Stress of yoke legs		$S_{max} \le$ 1.5 S_m = 19,400 psi (500°F)	S_{max} = 5,145 psi
Stress of yoke-bonnet connection		$S_{max} \le S_m$ = 19,225 psi (575°F)	S_{max} = 5,741 psi

Table 3.9-6(j) (Cont'd)

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	REQUIRED DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
IV. 28" SUCTION VALVE			
Body and Bonnet			
Loads: Design pressure, design temperature pipe reaction thermal effects	System requirement not specified	1,275 psi 575°F	1,275 psi 575°F
Pressure rating, psi	Used ASME Section III Figure NB-3545.1-2	$p_r = 668$	$p_r = 668$
Minimum wall thickness, inches	Used ASME Section III Para NB-3542	$t_m \geq 1.7724$ in	$t_m = 2.0$ minimum
Primary membrane stress, psi	Used ASME Section III Para NB-3545.1	$P_m \leq S_m$ (500°F) = 19,600 psi	$P_m = 9,275$ psi
Secondary stress due to pipe reaction	Used ASME Section III Para NB-3545.2(b)(1) (S = 30,000 psi)	$P_e < 1.5\ S_m$ (500°F) 1.5 (16,800) = 25,200 psi	$P_{ed} = 5,318$ psi $P_{eb} = 11,980$ psi $P_{et} = 11,575$ psi $P_e = 11,980$ psi
Primary plus secondary stress due to internal pressure	Used ASME Section III Para NB-3545.2(a)(1)	$S_n \leq 3\ S_m$ (500°F) = 58,800	$Q_p = 20,580$ psi
Thermal secondary stress	Used ASME Section III Para NB-3545.2	$S_n \leq 3\ S_m$ (500°F) = 58,800	$Q_t = 5,489$ psi

Table 3.9-6(j) (Cont'd)

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	REQUIRED DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Sum of primary plus secondary stress	Used ASME Section III Para NB-3545.2	$S_n \leq 3\ S_m$ (500?F) = 58,800	$S_n = Q_p + P_e + 2Q_t$ = 43,538 psi
Fatigue Requirements	Used ASME Section III Para NB-3545.3	$N_a \geq$ 2,000 cycles	$N_a > 10^6$ cycles
Cyclic rating	Used ASME Section III Para NB-3550	$I_t \leq 1$	I_t = 0.00274
Body-to-Bonnet Bolting			
Loads: Design pressure and temperature, gasket loads, stem operational load, seismic load (SSE)	Used ASME Section III Para NB-3647.1		
Bolt area	Used ASME Section III Para NB-3647.1	$A_b \geq$ 37.53 in^2 $S_b \leq$ 27,975 psi (575°F)	A_b = 55.86 in^2 S_b = 19,283 psi
Body flange stresses	Used ASME Section III Para NB-3647.1		
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$(575°F) = 28,838 psi $S_R \leq 1.5\ S_m$(575°F) = 19,225 psi $S_T \leq 1.5\ S_m$(575°F) = 19,225 psi	S_H = 24,264 psi S_R = 6,476 psi S_T = 8,364 psi
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$(100°F) = 30,000 psi $S_R \leq 1.5\ S_m$(100°F) = 30,000 psi $S_T \leq 1.5\ S_m$(100°F) = 30,000 psi	S_H = 28,945 psi S_R = 10,253 psi S_T = 13,619 psi
Bonnet flange			
Operating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m$(575°F) = 28,838 psi $S_R \leq 1.5\ S_m$(575°F) = 19,225 psi $S_T \leq 1.5\ S_m$(575°F) = 19,225 psi	S_H = 24,264 psi S_R = 6,476 psi S_T = 8,634 psi

Table 3.9-6(j) (Cont'd)

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	REQUIRED DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Gasket seating condition	Used ASME Section III Para NB-3647.1	$S_H \leq 1.5\ S_m(100°F) = 30,000$ psi $S_R \leq 1.5\ S_m(100°F) = 30,000$ psi $S_T \leq 1.5\ S_m(100°F) = 30,000$ psi	$S_H = 28,945$ psi $S_R = 10,253$ psi $S_T = 13,619$ psi
Stress in Stem			
Not applicable, valve is passive			
Disc Analysis			
Loads: Maximum differential pressure			
Maximum stress in the disc	Calculate maximum stress according to table 10 of Reference 3.9-5	$S_{max} \leq 1.5\ S_m(500°F) = 28,500$ psi	$S_{max} = 19,418$ psi
Yoke and Yoke Connections			
Loads: Stem operational load	Calculate stresses in the yoke and yoke connections to acceptable structural analysis methods		

Table 3.9-6(j) (Cont'd)

(Unit 2)

COMPONENT/ LOADS/ DESIGN	DESIGN PROCEDURE[1]	REQUIRED DESIGN VALUE[2]	ACTUAL DESIGN VALUE[2]
Tensile stress in yoke legs bolts		$S_{max} \leq S_m$ (500°F) = 28,800 psi	S_{max} = 3,877 psi
Stress at yoke legs		$S_{max} \leq 1.5\ S_m$ (500°F) = 19,400 psi	S_{max} = 5,378 psi
Stress at yoke-bonnet connection		$S_{max} \leq S_m$ (575°F) = 19,225 psi	S_{max} = 6,004 psi

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(1) ASME Section III refers to 1968 Edition of ASME B&PV Code.

(2) Terms used are defined as follows:

A_b	=	actual total cross-sectional area of bolts at root of thread or section of least diameter under stress, in^2
I_t	=	thermal cyclic index for a particular valve application
N_a	=	permissible number of complete startup/shutdown cycles at 100°F/hour fluid temperature change rate
p_r	=	primary pressure rating, lb
P_e	=	largest value among P_{eb}, P_{ed}, P_{et}, psi
P_{eb}	=	secondary stress in crotch region of valve body caused by bending of connected standard pipe, psi
P_{ed}	=	Secondary stress in crotch region of valve body caused by direct or axial load imposed by connected standard piping, psi
P_{et}	=	secondary stress in crotch region of valve body caused by twisting of connected standard pipe, psi
P_m	=	general primary-membrane stress intensity at crotch region, psi
Q_p	=	sum of primary-plus-secondary stresses at crotch resulting from internal pressure, psi
Q_T	=	thermal-stress in crotch region resulting from 100°F/hour fluid temperature change rate, psi
S_b	=	allowable bolt stress at design temperature, psi
S_B	=	bending stress, psi
S_H	=	calculated longitudinal stress in hub, psi
S_m	=	design stress intensity, psi
S_{max}	=	maximum stress, psi
S_n	=	sum of primary-plus-secondary stress intensities at crotch region resulting from 100?F/hour temperature change rate, psi
S_R	=	calculated radial stress in flange, psi
S_s	=	shear stress due to operator torque, psi
S_t	=	stress due to operator thrust, psi
S_T	=	tangential stress in flange, psi
t_m	=	minimum body wall thickness, in

Table 3.9-6(k)

HPCI TURBINE

CRITERIA/COMPONENT	LOADING CONDITION	STRESS TYPE	ALLOWABLE STRESS (PSI)	CALCULATED STRESS (PSI)
PRESSURE BOUNDARY CASTINGS				
Allowable stresses based on ASME Section III. Pressure boundary castings for Type SA216-WCB				
	For normal condition: Design pressure Design temperature Inlet and exhaust nozzle loads			
1. Stop Valve (100% radiograph)		General membrane	8,975	17,500
2. Turbine inlet (high pressure)		General membrane	6,550	14,000
3. Turbine wheel case (low pressure)		General membrane	6,000	14,000
	For upset, Emergency, or faulted condition: Design pressure Design temperature Controlling combination of OBE, SSE, SRV, & LOCA Inlet and exhaust nozzle loads			
1. Stop valve (100% radiograph)		General membrane	17,700	19,250
2. Turbine inlet (high pressure)		General membrane	15,250	15,400
3. Turbine wheel case (low pressure)		General membrane	13,690	15,400
PRESSURE BOUNDARY BOLTING				
Allowable stresses are based on ASME Section III, for pressure boundary bolting for Type SA193-B7				
	For normal condition: Design pressure Design temperature Inlet and exhaust nozzle loads			

Table 3.9-6(k) (Cont'd)

CRITERIA/COMPONENT	LOADING CONDITION	STRESS TYPE	ALLOWABLE STRESS (PSI)	CALCULATED STRESS (PSI)
PRESSURE-CONTAINING BOLTS				
Stop valve		Tensile	17,600	25,000
Turbine flange		Tensile	18,290	25,000
	For upset emergency, or faulted condition: Design pressure Design temperature Controlling combination of OBE, SSE, SRV, & LOCA Inlet and exhaust nozzle loads			
PRESSURE-CONTAINING BOLTS				
Stop valve		Tensile	17,950	25,000
Turbine flange		Tensile	18,655	25,000
	For faulted condition: Design pressure Design temperature Combination (ABS) of SSE, SRV, & LOCA Inlet and exhaust nozzle loads			
Turbine shaft stress		Bending	12,850 psi	45,000 psi
Turbine shaft deflection	-		0.032 inch	0.125 inch
Thrust bearing load		Force	4,600 lb_f	5,600 lb_f
Journal bearing load		Force	5,020 lb_f	19,500 lb_f
Pedestal bolts, coupling end		Tension	28,740 lb	31,100 lb
Taper pins, coupling end		Shear	39,110 lb	42,050 lb
Guide block weld, governor end		Shear	43,550 lb	43,650 lb

Table 3.9-6(l)

SLCS PUMP

Criteria/Loading		Component	Limiting Stress Type	Allowable Stress (psi)	Calculated Stress (psi)
Based on ASME Section III.					
Pressure Boundary Parts:					
1. Fluid Cylinder - SA182-F304,	S_y = 30,000 psi				
2. Discharge valve stop, stuffing box and cylinder head extension SA479-304,	S_y = 30,000 psi				
3. Discharge valve cover, cylinder head and stuffing box flange plate, SA285, Grade C	S_y = 30,000 psi				
4. Stuffing box gland, ASTM A461, Grade 630	S_y = 90,000 psi				
5. Studs, SA193-B7,	S_y = 105,000 psi				
6. Dowel pins[(2)] alignment, SAE4140,	S_y = 117,000 psi				
7. Studs, cylinder tie, SA193-B7,	S_A = 25,000 psi				
8. Pump holddown bolts, SAE Grade 1	T_A = 15,000 psi Q_A = 12,000 psi				
9. Power frame, foot area, cast iron,	S_A = 15,000 psi				
10. Motor holddown bolts, SAE Grade 1	T_A = 15,000 psi Q_A = 12,000 psi				
11. Motor frame foot area, cast iron,	S_A = 15,000 psi				

Table 3.9-6(l) (Cont'd)

Criteria/Loading	Component	Limiting Stress Type	Allowable Stress (psi)	Calculated Stress (psi)
Normal and Upset Condition Loads:				
1. Design pressure	1. Fluid Cylinder	General membrane	17,800	
2. Design temperature	2. Discharge valve stop	General membrane	17,800	(3)
3. Operating basis earthquake	3. Cylinder head extension	General membrane	17,800	
4. Nozzle loads(1)	4. Discharge valve cover	General membrane	17,800	
5. Thermal expansion	5. Cylinder head	General membrane	17,800	
6. SRV	6. Stuffing box flange plate	General membrane	17,800	
7. Dead weight	7. Stuffing box gland	General membrane	35,000	
	8. Cylinder head studs	Tensile	25,000	
	9. Stuffing box studs	Tensile	25,000	
Emergency or Faulted Condition:				
1. Design pressure	1. Fluid cylinder	General membrane	21,360	4,450
2. Design temperature	2. Discharge valve stop	General membrane	21,360	13,600
3. Weight of structure	3. Cylinder head extension	General membrane	21,360	13,600
4. Thermal expansion	4. Discharge valve cover	General membrane	21,360	8,150
5. Safe shutdown earthquake	5. Cylinder head	General membrane	21,360	8,150
6. LOCA	6. Stuffing box flange plate	General membrane	21,360	10,390
7. Nozzle loads	7. Stuffing box gland	General membrane	42,000	11,420
	8. Cylinder head studs	Tensile	25,000	18,820
	9. Dowel pins(2)	Shear only(2)	23,400	19,400
	10. Studs, cylinder tie	Tensile(2)	25,000	24,750
	11. Pump holddown bolts	Shear	12,000	7,560
	12. Pump holddown bolts	Tensile	15,000	9,950
	13. Power frame-foot area	Shear	15,000	1,850
	14. Power frame-foot area	Tensile	15,000	11,390
	15. Motor holddown bolts	Shear	12,000	3,470
	16. Motor holddown bolts	Tensile	15,000	5,660
	17. Motor frame-foot area	Shear	15,000	2,550
	18. Motor frame-foot area	Tensile	15,000	4,125

Table 3.9-6(l) (Cont'd)

Criteria/Loading	Component	Allowable Loads [5] (lb, ft-lb)	Calculated Loads (lb,ft-lb)
Normal and Upset Condition Loads:			
1. Design pressure	Suction	F_o = 730	F_i = 571
2. Design temperature	Nozzle	M_o = 282	M_i = 218
3. Weight of structure			
4. Thermal expansion	Discharge	F_o = 350	F_i = 138
5. Operating basis earthquake	Nozzle	M_o = 69	M_i = 38
6. SRV			
Emergency or Faulted Condition Loads:			
1. Design pressure	Suction	F_o = 730	F_i = 620
2. Design temperature	Nozzle	M_o = 282	M_i = 228
3. Weight of structure			
4. Thermal expansion	Discharge	F_o = 350	F_i = 170
5. Safe shutdown earthquake	Nozzle	M_o = 67	M_i = 46
6. LOCA			

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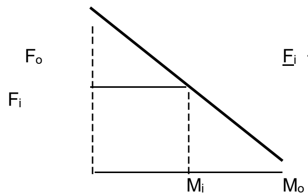
(1) Nozzle loads produce shear loads only.
(2) Dowel pins take all shear.
(3) Calculated stresses for emergency or faulted condition are less than the allowable stresses for the normal and upset condition loads, therefore the normal and upset condition is not evaluated.
(4) Operability: The sum of the plunger and rod assembly, pounds mass times 1.75, acceleration is much less than the thrust loads encountered during normal operating conditions. Therefore, the loads during the faulted condition have no significant effect on pump operability.

Table 3.9-6(l) (Cont'd)

(5) Allowable nozzle load criteria:

Units: Forces - lb
Moments - ft-lb

The allowable combinations of forces and moments are as follows:



$$\frac{F_i}{F_o} + \frac{M_i}{M_o} \leq 1$$

where:

F_i = Largest absolute value of the three actual external orthogonal forces (F_x, F_y, F_z) that may be imposed by the interface pipe.

M_i = Largest absolute value of the three actual external orthogonal moments (M_x, M_y, M_z) permitted from the interface pipe when they are combined simultaneously for a specific condition.

F_o = Allowable value of F_i when all moments are zero.

M_o = Allowable value of M_i when all forces are zero.

Table 3.9-6(m)

SLCS TANK

(UNIT 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MIN. THICKNESS REQ'D OR LOAD		ACTUAL STRESS OR MIN. THICKNESS REQ'D OR LOAD
1. Shell Thickness				
Loads: Normal and Upset	Brownell and Young			
Design Pressure and Temperature where:	"Process Equipment Design" $t = \frac{PR}{SE-0.6P}$	0.01542 in		0.1875 in
	t = min req'd. thickness, in P = design pressure, psig R = maximum internal radius, in S = allowable working stress, psi E = joint efficiency			
Stress Limit	Dynamic Analysis	25,800 psi		1200 psi
2. Nozzle Loads				
Loads: Normal and Upset Design pressure and temperature	The maximum moments due to pipe reaction and maximum forces shall not exceed the allowable limits.	Design Upset Emergency	25,800 psi 28,380 psi 30,960 psi	
Overflow Nozzle and Discharge Nozzle				Less than faulted
Loads: Faulted Dead weight, thermal expansion, and SSE	The maximum moments due to pipe reaction and maximum forces shall not exceed the allowable limits.	Faulted	41,280 psi	
Overflow nozzle Discharge nozzle				9556 psi 4962 psi

Table 3.9-6(m) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MIN. THICKNESS REQ'D OR LOAD	ACTUAL STRESS OR MIN. THICKNESS REQ'D OR LOAD
3. Anchor Bolts	API - 650 (ASME Section III)	18,750 psi (tensile) 15,000 psi (shear)	11,770 psi (tensile) 8,962 psi (shear)
4. Dynamic Loads	Equivalent static	Horizontal 1.5 g	0.578 g
SSE SRV LOCA		Vertical 0.14 g	0.898 g
Chugging	Dynamic Analysis	Load - See RRS curves	1,200 psi

Table 3.9-6(m) (Cont'd)

SLCS TANK

(UNIT 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MIN. THICKNESS REQ'D OR LOAD		ACTUAL STRESS OR MIN. THICKNESS REQ'D OR LOAD
1. Shell Thickness				
Loads: Normal and Upset	Brownell and Young			
Design Pressure and Temperature where:	"Process Equipment Design" $t = \frac{PR}{SE-0.6P}$	0.01542 in		0.1875 in
	t = min req'd. thickness, in P = design pressure, psig R = maximum internal radius, in S = allowable working stress, psi E = joint efficiency			
Stress Limit	Dynamic analysis	25,800 psi		1200 psi
2. Nozzle Loads				
Loads: Normal and Upset Design pressure and temperature	The maximum moments due to pipe reaction and maximum forces shall not exceed the allowable limits.	Allowables: Design Upset Emergency	 25,800 psi 28,380 psi 30,960 psi	
Overflow Nozzle and Discharge Nozzle				5556 psi(1) 6357 psi
Loads: Faulted Dead weight, thermal expansion, and SSE	The maximum moments due to pipe reaction and maximum forces shall not exceed the allowable limits			
Overflow nozzle		Faulted	41,280 psi	9556 psi(1)
Discharge nozzle		Faulted	41,280 psi	6357 psi

Table 3.9-6(m) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MIN. THICKNESS REQ'D OR LOAD	ACTUAL STRESS OR MIN. THICKNESS REQ'D OR LOAD
3. Anchor Bolts	ASME Section III	18,750 psi (tensile) 15,000 psi (shear)	11,770 psi tensile 8,962 psi shear
4. Dynamic Loads	Equivalent static	Horizontal 1.5 g	0.578 g
SSE SRV LOCA		Vertical 0.14 g	0.898 g
Chugging	Dynamic Analysis	28,500 psi	1200 psi

(1) Since the actual stresses resulting from the analysis using the faulted loads are so low, it is not necessary to reanalyze for actual stresses from upset and emergency conditions.

Table 3.9-6(n)

ECCS PUMPS

LOCATION	LOADING CONDITION	CRITERIA	CALCULATED STRESS (psi)	ALLOWABLE STRESS (psi)
RESIDUAL HEAT REMOVAL PUMP				
Stuffing Box Pipe	FAULTED CONDITION Design Pressure Static Loads Dynamic Loads	ASME Section III	11,718	26,250
Discharge Elbow	FAULTED CONDITION Design Pressure Static Loads Dynamic Loads	ASME Section III	16,172	20,000
Nozzle Shell Intersection	FAULTED CONDITION Design Pressure Static Loads Dynamic Loads	ASME Section III	15,661	28,875
Motor Stand	FAULTED CONDITION Static Loads Dynamic Loads	Bolting Loads and Stresses per ASME Section III, Subsection NF	15,374 (Tensile) 3,058 (Compressive)	22,800 19,326
Motor Bolting	FAULTED CONDITION Static Loads Dynamic Loads	Bolting Loads and Stresses per ASME Section III, Subsection NF	18,755 (Tensile) 4,623 (Shear)	62,500 25,833

Table 3.9-6(n) (Cont'd)

LOCATION	LOADING CONDITION	CRITERIA	CALCULATED STRESS (psi)	ALLOWABLE STRESS (psi)
CORE SPRAY PUMP				
Stuffing Box Pipe	FAULTED CONDITION Design Pressure Static Loads Dynamic Loads	ASME Section III	12,629	26,250
Discharge Elbow	FAULTED CONDITION Design Pressure Static Loads Dynamic Loads	ASME Section III	18,097	20,000
Nozzle Shell Intersection	FAULTED CONDITION Design Pressure Static Loads Dynamic Loads	ASME Section III	28,352	34,650
Motor Stand	FAULTED CONDITION Static Loads Dynamic Loads	Bolting Loads and Stresses per ASME Section III, Subsection NF	11,626 (Tensile) 2,031 (Compressive)	22,800 19,352
Motor Bolting	FAULTED CONDITION Static Loads Dynamic Loads	Bolting Loads and Stresses per ASME Section III, Subsection NF	7,203 (Tensile) 2,622 (Shear)	62,500 25,833

Table 3.9-6(o)

RHR HEAT EXCHANGER

(UNIT 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL STRESS OR MINIMUM THICKNESS REQUIRED
1. Closure Bolting	Bolting loads and stresses calculated per "Rules for Bolted Flange Connections" ASME Section VIII, Appendix II		
Loads: Normal and upset			
Design pressure and temperature			
Design gasket load			
Bolting Stress Limit	a. Shell to tube sheet bolts	39,375 psi	33,599 psi
Allowable working stress per ASME Section VIII	b. Channel cover bolts	39,375 psi	31,797 psi
2. Wall Thickness	Shell side ASME Section III Class 2 and TEMA Class C		
Loads: Normal and upset			
Design pressure and temperature	Tube side ASME Section VIII-Div. I and TEMA Class C		
Stress Limit			
ASME Section III	a. Shell	0.7915 in	0.8125 in
	b. Shell cover	1.0214 in	1.0625 in (minimum)
	c. Channel	1.1372 in	1.1825 in
	d. Tubes	0.047 in*	0.049 in
	e. Channel cover	6.856 in	6.875 in
	f. Tube sheet	5.7119 in	5.75 in
	g. Shell at nozzles	1.1372 in	1.1875 in

* Minimum wall thickness based on general corrosion.

Table 3.9-6(o) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL STRESS OR MINIMUM THICKNESS REQUIRED
3. Nozzle Loads		See (a) and (b) below	See (c) below
Design pressure and temperature			
Dead weight, thermal expansion and SSE	Primary local membrane stress less than 1.8 ASME Section VIII allowable.		

a) Maximum allowable piping loads for faulted conditions (including SSE) shall not exceed the following relationship for each nozzle:

$$\frac{F_{ix}}{F_{ox}} + \frac{F_{iy}}{F_{oy}} + \frac{F_{iz}}{F_{oz}} \leq 1 \quad \text{and} \quad \frac{M_{ix}}{M_{ox}} + \frac{M_{iy}}{M_{oy}} + \frac{M_{iz}}{M_{oz}} \leq 1$$

where F_i (lbs) is the maximum piping load imposed on each nozzle in the x, y and z direction and M_i (lb-ft) is the maximum moment imposed on each nozzle in the x, y and z directions.

b) Allowable design basis limits (forces F_{ox}, F_{oy}, F_{oz} and Moments M_{ox}, M_{oy}, M_{oz}) for nozzles N1, N2, N3, and N4:

	N1 (Channel inlet)	N2 (Channel outlet)	N3 (Shell inlet)	N4 (Shell outlet)
F_{ox}	11,395 lb	11,395 lb	44,143 lb	26,692 lb
F_{oy}	25,621 lb	25,621 lb	19,630 lb	26,692 lb
F_{oz}	25,621 lb	25,621 lb	44,143 lb	11,871 lb
M_{ox}	65,926 lb-ft	65,926 lb-ft	21,839 lb-ft	12,230 lb-ft
M_{oy}	8,537 lb-ft	8,537 lb-ft	113,596 lb-ft	12,230 lb-ft
M_{oz}	8,537 lb-ft	8,537 lb-ft	21,839 lb-ft	68,680 lb-ft

Note: The calculated loads in (c) below that exceed these allowable loads have been evaluated and are acceptable.

Table 3.9-6(o) (Cont'd)

(Unit 1)

CRITERIA		METHOD OF ANALYSIS		ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL[2] STRESS OR MINIMUM THICKNESS REQUIRED
c)[1]					
F_{ox}	6,367 lb	6,822 lb	6,638 lb	1,101 lb	
F_{oy}	4,442 lb	8,034 lb	10,370 lb	2,526 lb	
F_{oz}	4,836 lb	1,818 lb	11,327 lb	8,178 lb	
M_{ox}	6,141 lb-ft	9,669 lb-ft	37,996 lb-ft	7,900 lb-ft	
M_{oy}	28,863 lb-ft	1,168 lb-ft	16,782 lb-ft	4,195 lb-ft	
M_{oz}	10,670 lb-ft	52,242 lb-ft	16,966 lb-ft	1,863 lb-ft	
4. Support Brackets and Attachment Welds Loads: Faulted Design pressure and temperature, dead weight, nozzle loads, SSE.		Stress allowables per ASME Section III Subsection NT (Upset Condition).			
		a. Lower bracket welds			
		Bending stress		12,863 psi	10,907 psi
		Shear stress		7,074 psi	6,496 psi
		b. Upper Bracket welds			
		Bending stress		14,438 psi	6,069 psi
		Shear stress		7,074 psi	3,631 psi
5. Anchor Bolts Loads: Faulted Design pressure and temperature, dead weight, nozzle loads, SSE, SRV.		Stress allowable per ASME III, Subsection NF XVIII Lower support bolting			
		Interaction criteria		52,500 psi	33,048 psi

Table 3.9-6(o) (Cont'd)

(Unit 1)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL(2) STRESS OR MINIMUM THICKNESS REQUIRED
6. Shell Adjacent to Support Brackets	Shell stress allowables per ASME Section III Subsection NC (Upset Condition).		
Loads: Faulted Design pressure and temperature, dead weight, nozzle loads, SSE.	a. Maximum principal stress adjacent to upper support	28,875 psi	17,068 psi
	b. Maximum principal stress adjacent to lower support	28,875 psi	20,929 psi
7. Shell Away from Discontinuities	Stress allowable per ASME Section III Subsection NC (Upset Condition)		
Loads: Faulted Design pressure and temperature, dead weight, nozzle loads, SSE.	Principal stress	19,250 psi	15,662 psi

Table 3.9-6(o) (Cont'd)

RHR HEAT EXCHANGER

(UNIT 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL STRESS OR MINIMUM THICKNESS REQUIRED
1. Closure Bolting Loads: Normal and upset	Bolting loads and stresses calculated per "Rules for Bolted Flange Connections" ASME Section VIII, Appendix II		
Design pressure and temperature			
Design gasket load			
Bolting Stress Limit	a. Shell to tube sheet bolts	39,375 psi	31,973 psi
Allowable working stress per ASME Section VIII	b. Channel cover bolts	39,375 psi	30,721 psi
2. Wall Thickness Loads: Normal and upset	Shell side ASME Section III Class 2 and TEMA Class C		
Design pressure and temperature	Tube side ASME Section III Class 3 and TEMA Class C		
Stress Limit			
ASME Section III	a. Shell	0.7915 in	0.8125 in
	b. Shell cover	1.0214 in	1.0625 in (minimum)
	c. Channel	1.1372 in	1.1825 in
	d. Tubes	0.047 in	0.049 in
	e. Channel cover	6.856 in	6.875 in
	f. Tube sheet	5.7119 in	5.75 in

Table 3.9-6(o) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL STRESS OR MINIMUM THICKNESS REQUIRED
3. Nozzle Loads		See (a) and (b) below	See (c) below
Design pressure and temperature			
Dead weight, thermal expansion and SSE	Primary local membrane stress less than 1.8 ASME Section VIII allowable.		

a) Maximum allowable piping loads for faulted conditions (including SSE) shall not exceed the following relationship for each nozzle:

$$\frac{F_{ix}}{F_{ox}} + \frac{F_{iy}}{F_{oy}} + \frac{F_{iz}}{F_{oz}} \leq 1 \text{ and } \frac{M_{ix}}{M_{ox}} + \frac{M_{iy}}{M_{oy}} + \frac{M_{iz}}{M_{oz}} \leq 1$$

where F_i (lbs) is the maximum piping load imposed on each nozzle in the x, y and z direction and M_i (lb-ft) is the maximum moment imposed on each nozzle in the x, y and z directions.

b) Allowable design basis limits (forces F_{ox}, F_{oy}, F_{oz} and Moments M_{ox}, M_{oy}, M_{oz}) for nozzles N1, N2, N3, and N4:

	N1 (Channel inlet)	N2 (Channel outlet)	N3 (Shell inlet)	N4 (Shell outlet)
F_{ox}	11,395 lb	11,395 lb	44,143 lb	26,692 lb
F_{oy}	25,621 lb	25,621 lb	19,630 lb	26,692 lb
F_{oz}	25,621 lb	25,621 lb	44,143 lb	11,871 lb
M_{ox}	65,926 lb-ft	65,926 lb-ft	21,839 lb-ft	12,230 lb-ft
M_{oy}	8,537 lb-ft	8,537 lb-ft	113,596 lb-ft	12,230 lb-ft
M_{oz}	8,537 lb-ft	8,537 lb-ft	21,839 lb-ft	68,680 lb-ft

Note: The calculated loads in (c) below that exceed these allowable loads have been evaluated and are acceptable.

Table 3.9-6(o) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL STRESS OR MINIMUM THICKNESS REQUIRED

c)(1)

RHR HEAT EXCHANGER NO. 2AE205

COMP. (FAULTED)	NOZZLE NOS. N1 (Channel Inlet)	N2 (Channel Outlet)	N3 (Shell Inlet)	N4 (Shell Outlet)
F_x (lbs)	4,253	4,792	7,079	2,384
F_y (lbs)	2,626	3,452	7,940	6,250
F_z (lbs)	2,144	1,861	13,677	4,417
M_x (ft-lbs)	10,895	12,011	21,948	43,883
M_y (ft-lbs)	16,393	1,461	33,124	7,688
M_z (ft-lbs)	23,153	7,357	31,163	3,242

RHR HEAT EXCHANGER NO. 2BE205

COMP. (FAULTED)	NOZZLE NOS. N1 (Channel Inlet)	N2 (Channel Outlet)	N3 (Shell Inlet)	N4 (Shell Outlet)
F_x (lbs)	6,720	8,469	5,568	2,870
F_y (lbs)	3,254	7,950	9,440	3,369
F_z (lbs)	404	1,867	10,762	3,977
M_x (ft-lbs)	3,521	9,418	24,813	14,848
M_y (ft-lbs)	81	693	21,090	11,111
M_z (ft-lbs)	11,116	49,060	25,455	2,521

Table 3.9-6(o) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL(2) STRESS OR MINIMUM THICKNESS REQUIRED
4. Support Brackets and Attachment Welds	Stress allowables per ASME Section III Subsection NF (Upset Condition).		
Loads: Faulted	a. Lower bracket welds		
Design pressure and temperature, dead weight, nozzle loads, SSE.	Bending stress	14,149 psi	10,107 psi
	Shear stress	7,074 psi	6,200 psi
	b. Upper Bracket welds Bending stress	14,149 psi	3,766 psi
	Shear stress	7,074 psi	2,183 psi
5. Anchor Bolts	Stress allowable per ASME III, Subsection NF		
Loads: Faulted	Lower support bolting		
Design pressure and temperature, dead weight, nozzle loads, SSE, SRV.	Interaction criteria	52,500 psi	25,427 psi
6. Shell Adjacent to Support Brackets	Shell stress allowables per ASME Section III Subsection NC (Upset Condition).		
Loads: Faulted	a. Maximum principal stress adjacent to upper support	28,875 psi	18,083 psi
Design pressure and temperature, dead weight, nozzle loads, SSE.	b. Maximum principal stress adjacent to lower support	28,875 psi	19,311 psi

Table 3.9-6(o) (Cont'd)

(Unit 2)

CRITERIA	METHOD OF ANALYSIS	ALLOWABLE STRESS OR MINIMUM THICKNESS REQUIRED	ACTUAL[3] STRESS OR MINIMUM THICKNESS REQUIRED
7. Shell Away from Discontinuities	Stress allowable per ASME Section III Subsection NC (Upset Condition)		
Loads: Faulted	Principal stress	19,250 psi	12,080 psi
Design pressure and temperature, dead weight, nozzle loads, SSE.			

[1] The actual nozzle loads are provided by AE and are used to calculate the actual stresses.

[2] The "actual" stresses tabulated were calculated using the "maximum" nozzle loads that the RHR Heat Exchanger can withstand without exceeding the allowable stress.

[3] The "actual" stresses tabulated were calculated using the "actual" nozzle loads supplied by the AE for the Limerick 2 RHR Heat Exchanger.

Table 3.9-6(p)

RWCU PUMP

Following is a summary of the design calculations on the RWCU pump (B, C only):

Pump Part[(1)]	CALCULATED STRESS (psi)	ALLOWABLE STRESS (psi)
Casing wall	10,820	12,814
Cover bolting	20,000	25,000
Pedestal bolt (shear)	18,015	44,000
Motor Part[(2)]		
Motor foot bolts (shear)	174	60,000
Pump pedestal bolt (shear)	194	60,000
Foundation bolting	230	60,000

Following is a summary of the design calculations on the "A" RWCU pumps:

Part(1)	Calc. Stress (psi)	Allowable Stress (psi)
Pump Suction Nozzle	12,774 (U1) 1,582 (U2)	15,000
Pump Discharge Nozzle	12,824 (U1) 1,546 (U2)	17,500
Motor Case Outlet Nozzle	5,995 (U1) 5,995 (U2)	17,500
Motor Case Inlet Nozzle	5,997 (U1) 5,997 (U2)	17,500
Pump Support Flange Bolts (shear)	3,437 (U1) 3,325 (U2)	25,833 (U1) 11,800 (U2)
Pump Case/Motor Case Studs	23,229 (U1) 23,229 (U2)	25,000

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(1) ASME Code calculations.
(2) Non-ASME Code calculations.

Table 3.9-6(q)

RCIC TURBINE

(UNIT 1)

CRITERIA/LOADING	COMPONENT	LIMITING STRESS TYPE	ALLOWABLE STRESS (PSI)	CALCULATED STRESS (PSI)
The highest stressed sections of the various components of the RCIC turbine assembly are identified. Allowable stresses are based on ASME Section III, for:				
Pressure Boundary Castings SA216-WCB				
Pressure Boundary Boltings, SA193-B7				
Alignment Dowel Pins: AISI 4037, Rc 28-35				
Normal Condition Loads:				
1. Design pressure	Castings:			
2. Design Temperature	1) Stop valve[3]	General membrane	17,500	[1]
3. Inlet Nozzle Loads	2) Governor valve[3]	General membrane	17,500	
4. Exhaust Nozzle Loads	3) Turbine inlet	Local bending	21,000	
	4) Turbine case	Local bending	21,000	
	Pressure-containing bolts:	Tensile	25,000	
	Structure alignment pins:	Shear	61,100	
Upset, Emergency or Faulted Condition:				
1. Design Pressure	Castings:			
2. Design Temperature	1) Stop valve[3]	General membrane	19,250	14,160
3. Controlling Combinations of	2) Governor valve[3]	General membrane	19,250	15,300
SSE, SRV, and LOCA	3) Turbine inlet	Local bending	23,100	15,300
4. Inlet nozzle loads	4) Turbine case	Local bending	23,100	18,000
5. Exhaust Nozzle loads	Pressure-Containing Bolts	Tensile	25,000	20,100
	Structure Alignment pins	Shear	61,100	51,600

Table 3.9-6(q) (Cont'd)

(Unit 1)

CRITERIA/LOADING	COMPONENT	ALLOWABLE LOAD CRITERIA[4]	CALCULATED CRITERIA[4]
Nozzle Load Definition:			
Turbine vendor has defined allowable nozzle loads for the turbine assembly. The above calculated stresses assume these allowable nozzle loads have been satisfied.			
Normal Condition Loads:			
1. Design pressure 2. Design temperature 3. Weight of structure 4. Thermal expansion	Inlet Nozzle	$F = \frac{2620-M}{3}$	F = 421 lb M = 814 ft-lb
	Exhaust Nozzle	$F = \frac{6000-M}{3}$	F = 1,226 lb M = 2,043 ft-lb
Upset, Emergency, and Faulted Condition Loads:			
1. Design pressure 2. Design temperature 3. Weight of structure	Inlet Nozzle	$F = \frac{7500-M}{3.75}$	F = 1,156 lb M = 1,432 ft-lb
4. Thermal expansion 5. Controlling combination of SSE, SRV, and LOCA	Exhaust Nozzle	$F = \frac{8500-M}{0.34}$, but less than 7000 lb	F = 2,440 lb M = 6,253 ft-lb

Table 3.9-6(q) (Cont'd)

RCIC TURBINE

(UNIT 2)

CRITERIA/LOADING	COMPONENT	LIMITING STRESS TYPE	ALLOWABLE STRESS (PSI)	CALCULATED STRESS (PSI)
The highest stressed sections of the various components of the RCIC turbine assembly are identified. Allowable stresses are based on ASME Section III, for:				
Pressure Boundary Castings SA216-WCB				
Pressure Boundary Boltings, SA193-B7				
Alignment Dowel Pins: AISI 4037, Rc 28-35				
Normal Condition Loads:				
1. Design pressure	Castings:			
2. Design Temperature	1) Stop valve(3)	General membrane	17,500	(1)
3. Inlet Nozzle Loads	2) Governor valve(3)	General membrane	17,500	
4. Exhaust Nozzle Loads	3) Turbine inlet	Local bending	21,000	
	4) Turbine case	Local bending	21,000	
	Pressure-containing bolts:	Tensile	25,000	
	Structure alignment pins:	Shear	61,100	
Upset, Emergency or Faulted Condition:				
1. Design Pressure	Castings:			
2. Design Temperature	1) Stop valve(3)	General membrane	19,250	14,160
3. Controlling Combinations of SSE, SRV, and LOCA	2) Governor valve(3)	General membrane	19,250	15,300
	3) Turbine inlet	Local bending	23,100	15,300
4. Inlet nozzle loads	4) Turbine case	Local bending	23,100	18,000
5. Exhaust Nozzle loads	Pressure-Containing Bolts	Tensile	25,000	20,100
	Structure Alignment pins	Shear	61,100	51,600

Table 3.9-6(q) (Cont'd)

(Unit 2)

CRITERIA/LOADING	COMPONENT	ALLOWABLE LOAD CRITERIA[4]	CALCULATED CRITERIA[4]
Nozzle Load Definition:			
Turbine vendor and GE have defined allowable nozzle loads for the turbine assembly. The above calculated stresses assume these allowable nozzle loads have been satisfied.			
Normal Condition Loads:			
1. Design pressure 2. Design temperature 3. Weight of structure 4. Thermal expansion	Inlet Nozzle	$F = \frac{2620-M}{3}$	F = 563 lb M = 575 ft-lb
	Exhaust Nozzle	$F = \frac{6000-M}{3}$	F = 324 lb M = 1,748 ft-lb
Upset, Emergency, and Faulted Condition Loads:			
1. Design pressure 2. Design temperature 3. Weight of structure 4. Thermal expansion 5. Controlling combination of SSE, SRV, and LOCA	Inlet Nozzle	$F = \frac{7500-M}{3.75}$	F = 1,518 lb M = 1,137 ft-lb
	Exhaust Nozzle	$F = \frac{8500-M}{0.34}$, but less than 7000 lb	F = 1,016 lb M = 3,030 ft-lb

(1) Calculated stresses for the faulted condition are lower than the allowable stresses for the normal condition, therefore the normal condition is not evaluated.
(2) Operability: Analysis indicated that shaft deflection with faulted loads is 0.014 inch; which is fully acceptable; and maximum bearing load with faulted condition is 80% of allowable.
(3) 100% radiograph
(4) F = resultant force (lb); M = resultant moment (ft-lb).

Table 3.9-6(r)

RCIC PUMP

CRITERIA/LOADING		COMPONENT	LIMITING STRESS TYPE	ALLOWABLE STRESS (PSI)	CALCULATED STRESS (PSI)
Pressure boundary stress limits of the various components for the RCIC pump assembly are based on ASME Section III, for pressure boundary parts at 140ºF.					
1. Forged barrel, SA105 Grade II	S_y = 36,000 psi				
2. End cover plates, SA105 Grade II	S_y = 36,000 psi				
3. Nozzle connections, SA105 Grade II	S_y = 36,000 psi				
4. Aligning pin, SA105 Grade II	S_y = 36,000 psi				
5. Closure bolting, SA193-87	S_y = 105,000 psi				
6. Pump holddown bolting, SA449	S_y = 81,000 psi				
7. Taper pins, SA108 Grade B1112	S_y = 75,000 psi				
Normal and Upset Condition Loads:					
1. Design pressure		1. Forged barrel	General membrane[1]		
2. Design temperature		2. Nozzle reinforcement	Tensile shear		
3. Operating basis earthquake		3. Alignment pin	Tensile		
4. Suction nozzle loads		4. Taper pins			
5. Discharge nozzle loads		5. Pump holddown bolts			
6. Thermal expansion					
7. SRV					
8. Dead weight					
Emergency or Faulted Condition Loads:					
1. Design pressure		1. Forged barrel	General membrane	17,500	7,792
2. Design temperature		2. Nozzle reinforcement at barrel	General membrane	26,250	8,680
3. Safe shutdown earthquake					
4. Suction nozzle loads		3. Alignment pin	Shear	18,000	2,465
5. Discharge nozzle loads		4. Taper pins (bearing housing)		15,000	2,520
6. Thermal expansion					
7. LOCA		5. Pump holddown bolts	Tension	48,000	37,196
8. Dead weight					

Table 3.9-6(r) (Cont'd)

CRITERIA/LOADING	COMPONENT	ALLOWABLE LOADS[4] (lb, ft-lb)	CALCULATED LOADS (lb, ft-lb)
Normal and Upset Condition Loads:			
1. Design pressure	Suction	F_o = 1940	F_i = 1173
2. Design temperature	Nozzle	M_o = 2950[3]	M_i = 2555
3. Weight of structure			
4. Thermal expansion	Discharge	F_o = 3715	F_i = 913
5. Operating basis earthquake	Nozzle	M_o = 4330	M_i = 1193
6. SRV			
Emergency or Faulted Condition Loads:			
1. Design pressure	Suction	F_o = 2325	F_i = 1033
2. Design temperature	Nozzle	M_o = 2950	M_i = 2438
3. Weight of structure			
4. Thermal expansion	Discharge	F_o = 4450	F_i = 1061
5. Safe shutdown earthquake	Nozzle	M_o = 5200	M_i = 1751
6. LOCA			

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(1) Calculated stresses for emergency or faulted condition are less than the allowable for normal plus upset condition.

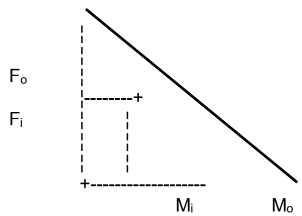
(2) Operability static analysis for emergency or faulted condition shows that the maximum shaft deflection is 0.0044 inch with a 0.0055 inch allowable, shaft stresses are 5602 psi with 32,000 psi allowable, drive end bearing loads are 45 lb with 7670 lb allowable, and thrust end bearing loads are 1462 lb with 17,200 lb allowable.

(3) This allowable moment was determined by analytical qualification which exceeds the Code-determined allowable.

Table 3.9-6(r) (Cont'd)

(4) Allowable nozzle load criteria:
Units: Forces - lb
Moments - ft-lb

The allowable combinations of forces and moments are as follows:



$$\frac{F_i}{F_o} + \frac{M_i}{M_o} \le 1$$

where:

F_i = Largest absolute value of the three (F_x, F_y, F_z) that may be imposed by actual external orthogonal forces the interface pipe.

M_i = Largest absolute value of the three actual external orthogonal moments (M_x, M_y, M_z) permitted from the interface pipe when they are combined simultaneously for a specific condition.

F_o = Allowable value of F_i when all moments are zero.

M_o = Allowable value of M_i when all forces are zero.

Table 3.9-6(s)

REACTOR REFUELING AND SERVICING EQUIPMENT

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Table 3.9-6(s) (Cont'd)

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Table 3.9-6(s) (Cont'd)

ACCEPTANCE CRITERIA	LOADING	PRIMARY STRESS TYPE	ALLOWABLE STRESS (psi)	CALCULATED STRESS (psi)
FUEL PREPARATION MACHINE				
The allowable axial plus bending loads stresses are based on manual of steel construction, 1980, 8th edition				
F_y = 35,000 psi				
F_u = 38,000 psi				
For normal condition:(2) S_{limit} = 0.66 F_y	For normal condition: 1. Static	Axial load plus bending	23,100	4,441
For emergency condition:(2) S_{limit} = 0.88 F_y	For emergency condition: 1. Static 2. OBE 3. LOCA 4. SRV	Axial load plus bending	30,800	27,619
For faulted condition:(2) S_{limit} = 0.88 F_y	For faulted condition: 1. Static 2. SSE 3. LOCA 4. SRV	Axial load plus bending	30,800	30,118

Table 3.9-6(s) (Cont'd)

ACCEPTANCE CRITERIA	LOADING	PRIMARY STRESS TYPE	ALLOWABLE STRESS (psi)	CALCULATED STRESS (psi)
REFUELING PLATFORMS				
The allowable axial plus bending loads stresses are based on ASME Section III, Subsection NA				
For type				
S_u = 58,000 psi				
S_y = 36,000 psi				
For normal condition: $S_{limit} = S_m = 0.66\ S_y$	For normal condition: 1. Static loads	Axial load plus bending	23,760	6,366 [(3)]
For upset condition: $S_{limit} = 0.9\ S_y$	For upset condition: 1. Static 2. OBE 3. LOCA 4. SRV	Axial load plus bending	31,680	31,552 [(3)]
For faulted condition: $S_{limit} = 0.7\ S_u$	For faulted condition: 1. Static 2. SSE 3. LOCA 4. SRV	Axial load plus bending	40,600	32,455 [(3)]

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(1) Pins in shear are limiting factor for horizontal loads applied to all rack castings. Allowable S_s assumed @ 50% allowable stress.
(2) The allowable stresses are shape dependent; therefore, these factors apply only at the location of the calculated stress.
(3) Calculated stresses shown on this table are based on conservative set of spectra, and pre-upgrade conditions. Calculation LS-0266 demonstrates on a comparative basis that these stresses bound the stresses in the upgraded configuration.

Table 3.9-6(t)

HPCI PUMP

LOCATION[1]	LOADING CONDITION	CRITERIA[2]	CALCULATED STRESS (psi)	ALLOWABLE STRESS (psi)
PRESSURE BOUNDARY PARTS				
Closure Bolting (Main)	Emergency/Faulted Condition: 1. Design pressure 2. Design temperature 3. Seismic loads 4. Nozzle loads 5. SRV 6. LOCA	The allowable stresses are based on normal and upset condition in accordance with ASME Section III for boundary parts at 140°F	19,950	25,000
Closure Bolting (Booster)			17,400	25,000
Casing Wall Thickness (Main)			12,050	14,000
Casing Wall Thickness (Booster)			3,650	14,000
NON-PRESSURE BOUNDARY PARTS				
Pump Bolts (Booster) (Tensile)	Emergency/Faulted Condition: 1. Design pressure 2. Design temperature 3. Seismic loads 4. Nozzle loads	F_o, M_o Actual	20,860 15,929	30,000 25,000
Pump Bolts (Main) (Tensile)		F_o, M_o Actual	29,042 14,813	30,000 25,000
Dowel Pins (Booster) (Shear)		F_o, M_o Actual	30,880 21,498	42,000 33,600
Dowel Pins (Main) (Shear)		F_o, M_o Actual	38,488 23,451	42,000 33,600

(1) Eight anchor bolts, each carries the stresses for both units mounted on a common base-plate.
(2) The allowable stress values for bolts are 0.5 S_u, and 0.4 S_u for pins where F_o, M_o criteria are used. For actual nozzle loads, the stress limits are based on ASME Section III at 140°F.

Table 3.9-6(u)

CONTROL ROD DRIVE (INDICATOR TUBE)

CRITERIA	LOADING	PRIMARY STRESS TYPE	ALLOWABLE STRESS (psi)	CALCULATED STRESS[2] (psi)
Allowable primary membrane stress plus bending stress plus bending is based on ASME Section III for type 316 stainless steel @ 250°F S_m = 20,000 psi				
For normal and upset condition: S_{allow} = 50,000 psi	For normal and upset condition: 1. Normal loads[1]	Primary membrane plus bending	50,000	45,500
For emergency condition: S_{allow} = 50,000 psi	For emergency condition: 1. Dynamic P 2. OBE 3. SRV	primary membrane plus bending	50,000	45,500
For faulted condition: S_{allow} = 59,900 psi	For faulted condition: 1. Dynamic P 2. SSE 3. LOCA 4. SRV	Primary membrane plus bending	59,900	46,600

[1] Normal loads include pressure, temperature, weight and mechanical loads.
[2] The loads listed here correspond to the operating power level of 3458 MWt. Per Reference 3.9-31, the loads are not changed for the MUR power uprate conditions.

Table 3.9-6(v)

CRD HOUSING[1]

CRITERIA	LOADING	PRIMARY STRESS TYPE	ALLOWABLE STRESS (psi)	CALCULATED STRESS[2] (psi)
Primary Stress Limit - The allowable primary membrane stress is based on ASME Section III, for Class 1 vessels, stainless steel.				
For normal and upset condition: $S_{limit} = 1.0\ S_m$ = 16,660 psi @ 575?F	Normal and upset condition loads: 1. Design pressure 2. Stuck rod scram loads 3. OBE, with housing lateral support installed 4. SRV 5. Hydraulic line loads	Maximum membrane stress intensity occurs at the tube-to-tube weld near the center of the housing for normal, upset, emergency, and faulted conditions.	16,660	11,900
For faulted conditions:	Faulted conditions loads: 1. Design pressure 2. Stuck rod scram loads 3. SSE, with housing lateral support installed 4. Annulus pressurization 5. LOCA 6. Hydraulic line loads		59,760	25,850

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(1) Analyzed to emergency conditions limits
(2) The loads listed here correspond to the operating power level of 3458 MWt. Per Reference 3.9-32, the loads are bounding for the MUR power uprate condition.

Table 3.9-6(w)

JET PUMPS

Criteria	Loading Combinations	Stress Type	Allowable Stress (psi)	Calculated Stress[1] (psi)
Primary membrane plus bending stress based on ASME Section III				
For service levels A and B (normal and upset) condition: For type 304SS @ 550°F S_m = 16,900 psi S_{limit} = 3.0 S_m psi	Normal and Upset Condition Loads: 1. Pressure 2. Weight 3. Clamping force 4. OBE 5. Vibration force 6. Thermal loads 7. SRV_{ALL}	Primary membrane plus bending plus secondary	50,700	18,185
For service level C (emergency) condition: For type 304SS @ 550°F S_m = 16,900 psi S_{limit} = 2.25 S_m psi	Emergency Condition Loads: 1. Pressure 2. Weight 3. Clamping force 4. Chugging 5. SRV_{ALL}	Primary membrane plus bending	38,025	10,702
For service level D (faulted) condition: For type 304SS @ 550°F S_m = 16,900 psi S_{limit} = 3.6 S_m psi	Faulted Condition Loads: 1. Pressure (internal) 2. Pressure (external) 3. Weight 4. Clamping force 5. Shock wave loads 6. SSE 7. Annulus pressurization 8. Jet reaction	Primary membrane plus bending	60,840	46,788

[1] The loads listed here correspond to the operating power level of 3458 MWt. Per Reference 3.9-32, the loads are bounding for the MUR power uprate condition.

Table 3.9-6(x)

FUEL ASSEMBLY (INCLUDING CHANNEL)[1][2][3]

Acceptance Criteria	Loading	Primary Load Type	Calculated Peak Acceleration	Evaluation Basis Acceleration
Acceleration Envelope	Horizontal Direction: Peak Pressure SSE Annulus Pressurization	Horizontal Acceleration	2.6 g	[1]
	Vertical Direction: Peak Pressure SSE Safety/Relief Valve Chugging	Vertical Accelerations	2.6 g[4]	[1]

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(1) Evaluation basis accelerations and evaluations are contained in NEDE-21175-3-P.

(2) The calculated maximum fuel assembly gap opening for the most limiting load combination is <0.01 inch based on the methodology contained in NEDE-21175-3-P.

(3) The fatigue analysis indicates that the fuel assembly has adequate fatigue capability to withstand the loadings resulting from multiple SRV actuations and the OBE + SRV event.

(4) This value is determined using the methodology contained in NEDE-21175-3-P.

Table 3.9-6(y)

HIGHEST STRESSED REGION ON THE LPCI COUPLING (ATTACHMENT RING)

Criteria	Loading Combinations	Stress Type	Allowable Stress (psi)	Calculated Stress[1] (psi)
Primary membrane plus bending stress based on ASME Section III NG-3000 for type CF3				
For service levels A and B (normal & upset) condition: S_{limit} = 1.5 S_m psi S_m = 16,900 psi @ 550°F	Normal and Upset Condition Loads: 1. Normal loads 2. Upset pressure 3. OBE 4. SRV	Primary membrane + bending	25,350	12,000
For service level C (emergency) condition: S_{limit} = 2.25 S_m S_m = 16,900 psi @ 550°F	Emergency Condition Loads: 1. Normal loads 2. Emergency pressure 3. Chugging 4. SRV	Primary membrane + bending	38,025	21,100
For service level D (faulted) condition: S_{limit} = 3.6 S_m S_m = 16,900 psi @ 550°F	Faulted Condition Loads: 1. Normal loads 2. Faulted pressure 3. Annulus pressurization 4. SSE	Primary membrane + bending	60,840	28,000

(1) The loads listed here correspond to the operating power level of 3458 MWt. Per Reference 3.9-32, the loads are bounding for the MUR power uprate condition.

Table 3.9-6(z)

REACTOR VESSEL SUPPORT EQUIPMENT:
CRD HOUSING SUPPORT

CRITERIA	LOADING	LOCATION	ALLOWABLE STRESS (psi)	CALCULATED STRESS (psi)
Primary Stress Limit				
AISC specification for the design, fabrication, and erection of structural steel for buildings	Faulted condition loads 1. Deadweight 2. Impact force from failure of a CRD housing	Beams (top chord)	33,000 33,000	f_a = 12,200 f_b = 16,500
		Beams (bottom chord)	33,000 33,000	f_a = 10,300 f_b = 11,700
For normal and upset condition: $f_a = 0.60\ f_y$ (tension) $f_b = 0.60\ f_y$ (bending) $f_v = 0.40\ f_y$ (shear) For faulted conditions: f_a limit = 1.5 f_a (tension) f_b limit = 1.5 f_b (bonding) f_v limit = 1.5 f_v (shear) f_y = material yield strength	(Deadweights and earthquake loads are very small compared to jet forces.)	Grid structure	41,500 27,500	f_b = 40,700 f_v = 11,100

Table 3.9-6(aa)

CONTROL ROD GUIDE TUBE

Criteria	Loading	Primary Stress Type	Allowable Stress (psi)	Calculated Stress[1] (psi)
Control Rod Guide Tube Primary stress limit - The allowable primary membrane stress plus bending stress is based on ASME Section III for type 304 stainless steel tubing.		Maximum stress occurs in the guide tube base		
For normal and upset conditions: 1.5 S_m = 1.5x16,000 psi = 24,000 psi	Upset condition loads 1. dead weight 2. external pressure 3. lateral flow impingement 4. OBE + SRV_{max} 5. scram	$P_m + P_b$	24,000	14,820
For emergency condition: 2.25 S_m = 36,000 psi	Emergency condition loads: 1. dead weight 2. external pressure 3. lateral flow impingement 4. SRV_{ADS} + CHUG	$P_m + P_b$	36,000	20,920
For faulted condition: 2.4 S_m = 38,400 psi	Faulted condition loads: 1. dead weight 2. external pressure 3. lateral flow impingement 4. SSE + SRV_{LSPA} + BCO	$P_m + P_b$	38,400	33,230

(1) The loads are calculated in Reference 3.9-32.

Table 3.9-6(ab)

INCORE HOUSING

Criteria	Loading	Primary Stress Type	Allowable Stress (psi)	Calculated Stress[1] (psi)
Primary Stress Limit - The allowable primary membrane stress is based on ASME Section III for Class 1 vessels for type 304 stainless steel.				
For normal, upset and emergency condition: $S_{limit} = 1.0\ S_m$ $= 16{,}660$ psi at 575°F	Service Level C (Emergency) condition loads 1. Design pressure 2. Design basis earthquake 3. SRV	Maximum membrane stress intensity occurs at the outer surface of the vessel penetration.	16,660	13,850
For faulted condition: $S_{limit} = 2.4\ S_m$	Faulted condition loads: 1. Faulted pressure 2. LOCA 3. SRV 4. SSE	Maximum membrane stress intensity occurs at the outer surface of the vessel penetration.	39,984	21,225

(1) The loads listed here correspond to the operating power level of 3458 MWt. Per Reference 3.9-32, the loads are bounding for the MUR power uprate condition.

Table 3.9-7

NON-NSSS PIPING SYSTEMS POWER ASCENSION TESTING

PIPING SYSTEM	CODE(S)/ SC/HE ME [(1)]	TEMP>200°F	THERMAL EXPANSION TEST [(2)]	DYNAMIC TRANSIENT TEST [(3)]	STEADY-STATE VIBRATION TEST [(4)]	REMARKS
Main steam and main steam relief	ASME III-2, B 31.1; SC I, SC II; HE	yes	yes	yes	yes	Main stop valve closure and SRV opening transients
Extraction steam	B 31.1; SC II; HE	yes	N/A[(5)]	N/A	N/A	
Condensate storage and transfer	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Feedwater	ASME III-1,2, B 31.1; SC I, SC II; HE	yes	yes	yes	yes	Power ascension test for safety-related piping portion only
Air removal and seal steam	B 31.1; SC II, HE	yes	N/A	N/A	N/A	
Service water	B 31.1; SC IIA; ME	no	N/A	N/A	N/A	A portion of the system has 200°F < T < 300°F.
Condensate	B 31.1; SC II; HE	yes	N/A	N/A	N/A	
Clarified water	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Fuel and diesel oil storage and transfer	B 31.1; SC I, SC II; ME	no	N/A	N/A	N/A	Emergency diesel exhaust has T>300°F and thermal expansion test performed.
RHRSW	ASME III-3, B 31.1; SC I, SC IIA, SC II; ME	no	N/A	N/A	N/A	

Table 3.9-7 (Cont'd)

PIPING SYSTEM	CODE(S)/ SC/HE ME (1)	TEMP>200°F	THERMAL EXPANSION TEST (2)	DYNAMIC TRANSIENT TEST (3)	STEADY-STATE VIBRATION TEST (4)	REMARKS
ESW	ASME III-3, B 31.1; SC I, SC IIA; ME	no	N/A	N/A	N/A	
Auxiliary steam	B 31.1; SC II; HE	yes	N/A	N/A	N/A	
Lube oil	B 31.1; SC II; ME	no				
Fire protection	SC II, SC IIA; ME	no	N/A	N/A	N/A	
Process sampling	ASME III-1,2,3, B 31.1; SC I, SC II; ME	no	N/A	N/A	N/A	
Chlorination	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Compressed air	ASME III-2,3, B 31.1; SC I, SC II; ME	no	N/A	N/A	N/A	
Instrument gas	ASME III-2,3, B 31.1; SC I, SC II; ME	no	N/A	N/A	N/A	
TECW	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Circulating water	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Demineralized water makeup	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Safeguard piping fill	ASME III-2, B 31.1; SC I, SC IIA; ME	no	N/A	N/A	N/A	
RECW	ASME III-2, 3, B 31.1; SC I, SC IIA; ME	no	N/A	N/A	N/A	
MSIV-LCS	ASME III-1,2, B 31.1; SC I, SC II; HE	yes	no	N/A	N/A	Abandoned

Table 3.9-7 (Cont'd)

PIPING SYSTEM	CODE(S)/ SC/HE ME (1)	TEMP>200°F	THERMAL EXPANSION TEST (2)	DYNAMIC TRANSIENT TEST (3)	STEADY-STATE VIBRATION TEST (4)	REMARKS
CSCWS	B 31.1; SC I; ME	no	N/A	N/A	N/A	
HPCI	ASME III-1,2, B 31.1; SC I, SC IIA, SC II; HE, ME	yes	yes	yes	yes	Steady-state vibration for steam supply and turbine exhaust Dynamic transient for turbine stop valve closure.
RCIC	ASME III-1,2, B 31.1; SC I; HE, ME	yes	yes	N/A	yes	Steady-state vibration for RCIC steam supply and turbine exhaust
Plant heating steam	B 31.1; SC II, SC IIA; HE	yes	N/A	N/A	N/A	
RWCU	ASME III-1,2,3; SC I, SC II, SC IIA; HE, ME	yes	yes	N/A	yes	Steady-state vibration for RWCU line inside containment
RHR	ASME III-1,2,3; SC I; HE, ME	yes	yes	N/A	yes	Majority of the system has normal operating temperature less than 300°F. Thermal expansion tests are done for SC I systems with T>300°F. Steady-state vibration for inside containment piping and RHR pump discharge

Table 3.9-7 (Cont'd)

PIPING SYSTEM	CODE(S)/ SC/HE ME (1)	TEMP>200?F	THERMAL EXPANSION TEST (2)	DYNAMIC TRANSIENT TEST (3)	STEADY-STATE VIBRATION TEST (4)	REMARKS
Condensate filter demineralizer	B 31.1; SC II; ME	no	N/A	N/A	N/A	
CRD hydraulic	ASME III-2, B 31.1; SC I, SC II, SC IIA; ME	no	N/A	N/A	N/A	
SLCS	ASME III-1,2, B 31.1; SC I, SC IIA; HE, ME	no (See remarks)	N/A (See remarks)	N/A	N/A	Only a small portion of the line near RPV has temperature >200°F.
Core spray	ASME III-1,2; SC I; HE, ME	yes	yes	N/A	yes	Steady-state vibration for core spray pump discharge
FPCC	ASME III-2,3, B 31.1; SC I, SC II, SC IIA; ME	no	N/A	N/A	N/A	
CAC	ASME III-2, B 31.1; SC I, SC IIA; ME	no	N/A	N/A	N/A	No safety-related piping with T>300°F
Solid radwaste	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Liquid radwaste	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Gaseous radwaste	B 31.1; SC II; ME	yes	N/A	N/A	N/A	
DCWS	B 31.1; SC II; ME	no	N/A	N/A	N/A	
Generator H_2 cooling and CO_2 purge	B 31.1; SC II; ME	no	N/A	N/A	N/A	

Table 3.9-7 (Cont'd)

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(1) Code(s): ASME III B&PV Code, -1, -2 or -3: Denotes nuclear Class 1, 2, or 3 piping;
B 31.1: Denotes ANSI B 31.1, Code for Pressure Piping;
SC I, II, or IIA: Denotes seismic Category I, II, or IIA;
HE: Denotes high energy piping system, i.e., pressure ≥275 psi or temperature ≥200°F during normal plant operation;
ME: Denotes moderate energy piping system.

(2) Thermal expansion test for the indicated systems corresponds to test description STP-17 in Table 14.2-3.

(3) Dynamic transient test for the indicated systems corresponds to test description STP-36 in Table 14.2-3. Main steam turbine trip test for Unit 2 (ref. Startup Test STP-36) at 100% power level will be performed during commercial operation of that Unit.

(4) Steady-state vibration tests for the indicated systems corresponds to test description STP-33 Table 14.2-3.

Instrument lines connected to process pipes on which steady-state vibration testing is performed are evaluated on the following basis;
a) for accessible lines; visually monitored
b) for inaccessible lines; instrument lines are monitored, inspected, and measured in accordance with startup test specification for BOP piping as committed in SSER-2.

(5) N/A: Denotes not applicable. Test is not performed for the following reason:
a) For thermal expansion tests: The system is not safety-related or the normal operating temperature < 300°F;
b) For dynamic transient test: The system is not safety-related or does not experience any significant transients;
c) For steady-state vibration tests: The system is not safety-related or no significant vibration is expected.

Table 3.9-8

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Table 3.9-9

DYNAMIC QUALIFICATION SUMMARY
NON-NSSS SAFETY-RELATED MECHANICAL EQUIPMENT

EQUIPMENT DESCRIPTION	ITEM NO.	SUPPLIER	DYNAMIC QUALIFICATION PACKAGE NO.
ESW pumps	8031-M-12	Byron Jackson	D-2
RHRSW pumps	8031-M-12	Byron Jackson	D-3
Diesel generators	8031-M-71	Colt Industries	D-7
Diesel oil transfer pumps	8031-M-79	Crane Dening	D-8
Diesel fuel oil storage tanks	8031-C-28	Buffalo Tank Div., Bethlehem Steel	D-209
Centrifugal fans	8031-M54A	Buffalo Forge	D-55
Drywell Sumps	8031-M-43A	Process Equip-ment Co.	D-5
Control room chilled water pumps	8031-M-58	Ingersoll-Rand	D-139
Control room chiller	8031-M-57A	Carrier Corp	D-138
Reactor enclosure crane	8031-M-16	Harnischfeger	D-50
Fuel pool skimmer surge tanks	8031-C-45	Pittsburgh-Des Moines Steel	D-172
RHR pump suction strainers	NE-265	ABB-Combustion Engineering	D-69
RCIC pump suction strainers	8031-M-162	Newark Wire Cloth	D-69

Table 3.9-9 (Cont'd)

EQUIPMENT DESCRIPTION	ITEM NO.	SUPPLIER	DYNAMIC QUALIFICATION PACKAGE NO.
HPCI pump suction strainers	8031-M-162	Newark Wire Cloth	D-69
Core spray pump suction strainers	NE-265	ABB-Combustion Engineering	D-69
Safeguard piping fill pumps	8031-M-164	Hayward Tyler Pump Co.	D-73
Primary containment vacuum breakers	8031-M-81	Anderson Greenwood Co.	D-135
Nuclear safety and relief valves	8031-M-204B	Crosby	D-74 D-154
		Lonergan	D-65 D-66
	8031-M-204C	Crosby	D-197
RHR HX vacuum relief valve	8031-M-204B	Crosby	D-67
RHR HX relief valve	8031-M-204B	Crosby	D-68
Pressure relief valves		Lonergan	D-31
CREFAS and RERS filter assembly	8031-M-72A	CIV	D-140
SGTS filters	8031-M-56	American Air Filter	D-137

Table 3.9-9 (Cont'd)

EQUIPMENT DESCRIPTION	ITEM NO.	SUPPLIER	DYNAMIC QUALIFICATION PACKAGE NO.
Containment hydrogen recombiners, recombiner power supply panels, recombiner control panels	8031-M-40	Atomics International	D-75
Rupture discs	8031-M-106	Continental Disc Corp.	D-97
MSIV and MSRV accumulator tanks	8031-M-170	Western piping and Engineering	D-157
Diesel oil transfer pump strainers	8031-M-36	Zurn Industries	D-4
Spray pond nozzles	8031-M-112	Spray Engineering Co.	D-9
Spray pond nozzle junction boxes	8031-M-112	Spray Engineering Co.	D-10
Drywell HVAC pressure relief valves	8031-M-123	American Air Filter	D-11
HPCI/RCIC Exhaust steam traps	8031-M-90A	Yarway	D-71
Nuclear wye strainers	8031-M-92AA	Western Piping and Engineering	D-76
MSRV Vacuum relief valves	8031-M-81	Anderson Greenwood Co.	D-136
HVAC Isolation valves	8031-M-70	Allis Chalmers	D-141 D-145
Back pressure steam isolation dampers	8031-M-113	American Warming and Ventilation Co.	D-142

Table 3.9-9 (Cont'd)

EQUIPMENT DESCRIPTION	ITEM NO.	SUPPLIER	DYNAMIC QUALIFICATION PACKAGE NO.
Volume control balancing dampers	8031-M-113	American Warming and Ventilation, Co.	D-148
Fire dampers	8031-M-113	American Warming and Ventilation, Co.	D-150
Slide gate dampers	8031-M-113	American Warming and Ventilation, Co.	D-151
SGTS heaters	8031-M-56	Industrial Engineering and Equipment Co.	D-184
Fuel oil and lube oil filter	8031-M-71	Colt Industries	D-218
Back flood check valves	8031-M-178	Zurn Industries	D-122
HVAC control panels	8031-M-66	Alison	D-162
HVAC control panels (Groups 1-10)	8031-M-66	MCC Powers	D-57 D-187 D-188 D-189 D-190 D-191 D-192 D-193 D-215 D-216
Drywell coolers	8031-M-123	American Air Filter	D-30
Spray pond pumphouse fan cabinets	8031-M-123	American Air Filter	D-48
Reactor enclosure fan cabinets	8031-M-123	American Air Filter	D-62

Table 3.9-9 (Cont'd)

EQUIPMENT DESCRIPTION	ITEM NO.	SUPPLIER	DYNAMIC QUALIFICATION PACKAGE NO.
Control enclosure fan cabinets	8031-M-123	American Air Filter	D-64
Diesel generator enclosure exhaust fans	8031-M-69C	Joy Mfg. Co.	D-6
Control enclosure recirculation fans	8031-M-69C	Joy Mfg. Co.	D-181
Reactor enclosure recirculation fans	8031-M-69C	Joy Mfg. Co.	D-63
Electro-hydraulic operated fan isolation and flow control dampers	8031-M-113	American Warming and Ventilation, Inc.	D-149
Pneumatic operated dampers	8031-M-113	American Warming and Ventilation, Inc.	D-152
Gravity back draft dampers	8031-M-113	American Warming and Ventilation, Inc.	D-153

Table 3.9-10

NSSS COMPARISON WITH REGULATORY GUIDE 1.48

COMPONENT	REGULATORY GUIDE 1.48[(1)] PLANT CONDITION[(2)]	REGULATORY GUIDE 1.48[(1)] LOADING COMBINATION 1/	REGULATORY GUIDE 1.48[(1)] DESIGN LIMIT	REGULATORY GUIDE PARAGRAPH	LGS[(1)] LOADING COMBINATION[(c)]	CODE ALLOWABLE STRESSES	ASME SECTION III REFERENCE	HOW LGS COMPARES WITH REGULATORY GUIDE 1.48
Class 1 Vessels	U	(NPC or UPC) + 0.5 SSE	NB-3223[2/]	1.a	(NPC or UPC), 0.5 SSE	3.0 S_m (includes) secondary stresses)	NB-3223	GE reflects industry position
	E	EPC	NB-3224[2/]	1.b	EPC, 0.5 SSE + transient	1.8$_m$	NB-3224 NB-3225	
	F	NPC + SSE +DSL	NB-3225[2/]	1.c	NPC + SSE + DSL	App. F - Section III		
Class 1 Piping	U	(NPC or UPC) + 0.5 SSE	NB-3654[2/]	1.a	(NPC or UPC), 0.5 SSE	3.0 S_m (includes secondary stresses	NB-3654	GE reflects industry position
	E	EPC	NB-3655[2/]	1.b	EPC, 0.5 SSE + transient	2.25 S_m	NB-3655	
	F	NPC + SSE + DSL	NB-3656[2/]	1.c	NPC + SSE + DSL	3.0 S_m	NB-3656	
Class 1 Pumps (inactive)	U	(NPC or UPC) + 0.5 SSE	NB-3223[5/1/]	2.a	(NPC or UPC), 0.5 SSE	1.63 S_m	NB-3223	GE reflects industry position
	E	EPC	NB-3224[1/]	2.b	EPC, 0.5 SSE + transient	1.8 S_m	NB-3224	
	F	NPC + SSE + DSL	NB-3225[1/]	2.c	NPC + SSE + DSL	App. F - Section III	NB-3225	
Class 1 Pumps (active)	U	(NPC or UPC) + 0.5 SSE	NB-3222[5/6/] 7 8	4.a	(NPC or UPC), 0.5 SSE	Not applicable	Not applicable	Not applicable
	E	EPC	NB-3222[5/6/] 7/8/	4.a	EPC			
	F 7/8/	NPC + SSE + DSL	NB-3222[5/6/]	4.a	NPC + SSE + DSL			
Class 1 Valves (inactive) Designed by analysis.	U	(NPC or UPC) + 0.5 SSE	NB-3223[5/4/]	2.a	(NPC or UPC), 0.5 SSE	Not applicable	Not applicable	Not applicable
	E	EPC	NB-3224[4/]	2.b	EPC			
	F	NPC + SSE + DSL	NB-3225[2/4/]	2.c	NPC + SSE + DSL			

Table 3.9-10 (Cont'd)

COMPONENT	PLANT CONDITION[2]	REGULATORY GUIDE 1.48[1]: LOADING COMBINATION 1/	REGULATORY GUIDE 1.48[1]: DESIGN LIMIT	REGULATORY GUIDE 1.48[1]: REGULATORY GUIDE PARAGRAPH	LGS[1]: LOADING COMBINATION[c]	LGS[1]: CODE ALLOWABLE STRESSES	LGS[1]: ASME SECTION III REFERENCE	HOW LGS COMPARES WITH REGULATORY GUIDE 1.48
Class 1 Valves (inactive)	U	(NPC or UPC) + 0.5 SSE	1.1 Pr	3.a	(NPC or UPC), 0.5 SSE	1.1 P_r	NB-3525	GE reflects industry position
Designated by either standard or alternative	E	EPC	1.2 Pr	3.b	EPC, 0.5 SSE + transient	1.2 P_r	NB-3526	
design rules.	F	NPC + SSE + DSL	1.5 Pr	3.c	NPC + SSE + DSL	1.5 P_r	NB-3527	
Class 1 Valves (active)	U	(NPC or UPC) + 0.5 SSE	NB-3222[5/6/7/8/]	4.a	(NPC or UPC), 0.5 SSE	Not applicable	Not applicable	Not applicable
Designed by analysis.	E	EPC	NB-3222[5/6/7/8/]	4.a	EPC			
	F	NPC + SSE + DSL	NB-3222[5/6/7/8/]	4.a	NPC + SSE + DSL			
Class 1 Valves (active)	U	(NPC or UPC) + 0.5 SSE	1.0 Pr[6/]	5.a	(NPC or UPC), 0.5 SSE	1.0 P_r[a]	NB-3525	GE reflects industry position
Designed by standard or alternative design	E	EPC	1.0 Pr[6/]	5.a	EPC, 0.5 SSE + transient	1.0 P_r[a]	NB-3626	
rules.	F	NPC + SSE + DSL	1.0 Pr[6/]	5.a	NPC + SSE + DSL	1.0 P_r[a]	NB-3527	
Class 2 & 3 Vessels (Division 1) of	U	(NPC or UPC) + 0.5 SSE	1.1 g[9/]	6.a	(NPC or UPC), 0.5 SSE	σ_m = 1.1 S[b]	Code Case 1607	Faulted condition: NRC more conservative. GE reflects industry position.
ASME Section VIII	E	EPC	1.1 g[9/]	6.a	EPC, 0.5 SSE + transient		NC/ NB 3221.1(b)	
	F	NPC + SSE + DSL	1.5 g[9/]	6.b	NPC + SSE + DSL	?$_m$ = 2.0 S[b]		
Class 2 Vessels (Division 2) of	U	(NPC or UPC) + 0.5 SSE	NB-3223[2/]	7.a	(NPC or UPC), 0.5 SSE	Not applicable	Not applicable	Not applicable
ASME Section VIII	E	EPC	NB-3224[2/]	7.b	EPC			
	F	NPC + SSE + DSL	NB-3225[2/]	7.c	NPC + SSE + DSL			

Table 3.9-10 (Cont'd)

	REGULATORY GUIDE 1.48[(1)]				LGS[(1)]			
COMPONENT	PLANT CONDITION[(2)]	LOADING COMBINATION 1/	DESIGN LIMIT	REGULATORY GUIDE PARAGRAPH	LOADING COMBINATION[(c)]	CODE ALLOWABLE STRESSES	ASME SECTION III REFERENCE	HOW LGS COMPARES WITH REGULATORY GUIDE 1.48
Class 2 & 3 Piping	U	(NPC or UPC) + 0.5 SSE (b)(1)	NC-3611.1[10/] (b)(4)(c)	8.a	(NPC or UPC), 0.5 SSE	1.2 S_h	NC/ND 3611.3(b)	NRC more conservative. GE reflects industry position.
	E	EPC (b)(1)	NC-3611.1[10/] (b)(4)(c)	8.a	EPC, 0.5 SSE + transient	1.8 S_h	NC/ND 3611.3(c) (4)(b)	
	F	NPC + SSE + DSL (b)(4)(c) (b)(2)	NC-3611.1[10/]	8.b	NPC + SSE + DSL	2.4 S_h	Code case 1606	
Class 2 & 3 Pumps (inactive)	U	(NPC or UPC) + 0.5 SSE	$\sigma_m \leq 1.1\ S \geq \frac{\sigma_m + \sigma_b}{1.5}$	9.a	(NPC or UPC), 0.5 SSE	Not applicable	Not applicable	Not applicable
	E	EPC	$\sigma_m \leq 1.1\ S \geq \frac{\sigma_m + \sigma_b}{1.5}$	9.a	EPC			
	F	NPC + SSE + DSL	$\sigma_m \leq 1.2\ S \geq \frac{\sigma_m + \sigma_b}{1.5}$	9.b	NPC + SSE + DSL			
Class 2 & 3 Pumps (active)	U	(NPC or UPC) + 0.5 SSE	$\sigma_m \leq 1.0\ S \geq$[11/] $\frac{\sigma_m + \sigma_b}{1.5}$	10.a	(NPC or UPC), 0.5 SSE	$\sigma_m = 1.1\ S$[(b)(d)]	Code case 1636	GE reflects industry position
	E	EPC	$\sigma_m \leq 1.0\ S \geq$[11/] $\frac{\sigma_m + \sigma_b}{1.5}$	10.a	EPC, 0.5 SSE + transient		NC/ND 3423	
	F	NPC + SSE + DSL	$\sigma_m \leq 1.0\ S \geq$[11/] $\frac{\sigma_m + \sigma_b}{1.5}$	10.a	NPC + SSE + DSL	$\sigma_m = 1.2\ S$[(b)(d)]		
Class 2 & 3 Valves (inactive)	U	NPC or UPC) + 0.5 SSE	1.1 P_r	11.a	(NPC or UPC), 0.5 SSE	$\sigma_m = 1.1\ S$[(b)]	Code case 1635	Equally conservative
	E	EPC	1.1 P_r	11.a	EPC, 0.5 SSE + transient			
	F	NPC + SSE + DSL	1.2 P_r	11.b	NPC + SSE + DSL	$\sigma_m = 1.2\ S$[(b)]	NC/ND 3621	

Table 3.9-10 (Cont'd)

	REGULATORY GUIDE 1.48[1]				LGS[1]			
COMPONENT	PLANT CONDITION[2]	LOADING COMBINATION 1/	DESIGN LIMIT	REGULATORY GUIDE PARAGRAPH	LOADING COMBINATION[c]	CODE ALLOWABLE STRESSES	ASME SECTION III REFERENCE	HOWLGSCOMPARES WITH REGULATORYGUIDE 1.48
Class 2 & 3 Valves (active)	U	(NPC or UPC) + 0.5 SS	1.0 P_r 11/	12.a	(NPC or UPC), 0.5 SSE	σ_m = 1.1 S[(a)(b)]	Code case 1635	Equally conservative. One valve, E41-F005, does not meet this LGS alternate position.
	E	EPC	1.0 P_r 11/	12.a	EPC, 0.5 SSE + transient			
	F	NPC + SSE + DSL	1.0 P_r 11/	12.a	NPC + SSE + DSL	σ_m = 1.2 S[(a)(b)]	NC/ND 362	Structural integrity under its peak transient conditions was justified under applicable provisions of B31.1 (1967). This valve fully meets the load combination and acceptance criteria of Table 3.9-6.

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(1) Numerical indicators (e.g. 1/) in the Regulatory Guide portion of the table correspond to footnotes of Regulatory Guide 1.48.
Alphabetical indicators in the LGS portion of table (or comparative column) correspond to the following:

(a) In addition to compliance with the design limits specified, assurance of operability under all design loading combinations shall be in accordance with Section 3.9.3.2.

(b) The design limit for local membrane stress intensity or primary membrane plus primary bending stress intensity is 150% of that allowed for general membrane (except as limited to 2.45 for inactive components under faulted condition).

(c) When selecting plant events for evaluation, the choice of the events to be included in each plant condition is selected based on the probability of occurrence of the particular load combination. The combination of loads are those identified in Table 3.9-2.

UPC = Upset Plant Conditions
NPC = Normal Plant Conditions
EPC = Emergency Plant Conditions
DSL = Dynamic System Loading
SSE = Safe Shutdown Earthquake

(d) Inactive limits may be used since operability will be demonstrated in accordance with Section 3.9.3.2.

(2) U = Upset
E = Emergency
F = Faulted

Table 3.9-11

DESIGN LOADING COMBINATIONS FOR ASME CODE
CLASS 1, 2, AND 3 NON-NSSS COMPONENTS(2)

CONDITION		DESIGN LOADING COMBINATIONS(1)
Design		PD
Normal		PD + DW
Upset	(a)	$PO + DW + (OBE^2 + SRV_x^2)^{1/2}$
	(b)	$PO + DW + (RVC^2 + OBE^2)^{1/2}$
	(c)	PO + DW + FV
	(d)	PO + DW + OBE + RVO
Emergency	(a)	$PO + DW + (OBE^2 + SRV^2_{ADS} + SBA^2)^{1/2}$
	(b)	$PO + DW + (FV^2 + OBE^2)^{1/2}$
Faulted	(a)	$PO + DW + (OBE^2 + SRV^2_{ADS} + IBA^2)^{1/2}$
	(b)	$PO + DW + (SSE^2 + SRV^2_{ADS} + IBA^2)^{1/2}$
	(c)	$PO + DW + (SSE^2 + DBA^2)^{1/2}$

where:

PD = design pressure
PO = operating pressure
DW = dead weight
OBE = operating basis earthquake (inertia portion)
SSE = safe shutdown earthquake (inertia portion)
SRV_x = loads due to SRV blow, axisymmetric or asymmetric
SRV_{ADS} = loads due to automatic depressurization SRV blow, axisymmetric
SBA = small break accident
IBA = intermediate break accident
DBA = design basis accident
FV = transient response of the piping system associated with fast valve closure (transients associated with valve closure times less than 5 seconds are considered)
RVC = transient response of the piping system associated with relief valve opening in a closed system
RVO = sustained load or response of the piping system associated with relief valve opening in an open system or last segment of the closed system with steady-state load

Table 3.9-11 (Cont'd)

SBA, IBA, and DBA include all event induced loads, as applicable, such as chugging, condensation oscillation, pool swell, drag loads, annulus pressurization, etc.

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(1) As required by the appropriate subsection (ie, NB, NC, or ND, of ASME Section III, Division I), other loads, such as thermal transient, thermal gradients, and anchor point displacement portion of the OBE or SRV, are considered in addition to the primary stress-producing loads listed.

(2) Table 3.9-6 lists the load combinations for ASME Code Class 1, 2, and 3 NSSS piping, equipment, and supports.

Table 3.9-12

DESIGN CRITERIA FOR ASME CLASS 1 NON-NSSS PIPING

CONDITION	SERVICE LEVEL	APPLICABLE CODE PARAGRAPH[(1)(2)]	PRIMARY STRESS LIMITS
Design	-	NB-3221 and NB-3652	1.5 S_m
Normal	A	NB-3222 and NB-3653	1.5 S_m
Upset	B	NB-3223 and NB-3654	1.8 S_m and 1.5 S_y
Emergency	C	NB-3224 and NB-3655	2.25 S_m and 1.85 S_y
Faulted	D	NB-3225 and NB-3656	3.0 S_m

(1) As specified by ASME Section III, 1977 Edition through Summer 1979 Addenda.

(2) Functional capability of essential piping is assured in accordance with NEDO-21985, September 1978.

Table 3.9-13

DESIGN CRITERIA FOR NON-NSSS ASME CODE CLASS 1 VALVES

CONDITION	STRESS LIMITS
Design	NB-3521[1]
Normal and upset	NB-3200 or NB-3500[1] (Standard Design Rules)
Emergency[2]	NB-3526[3]
Faulted[2]	NB-3527[3]

(1) As specified by ASME Section III, 1971 through Winter 1972 Addenda.
(2) Where valve function must be ensured (active valve) during the emergency or faulted conditions, the specified emergency or faulted condition for the plant shall be considered the normal condition for the valve.
(3) As specified by ASME Section III, 1971, through Winter 1973 Addenda.

Table 3.9-14

DESIGN CRITERIA FOR NON-NSSS ASME CODE CLASS 2 AND 3 VESSELS DESIGNED TO NC-3300 AND ND-3300

CONDITION	STRESS LIMITS[1]
Design and normal	The vessel shall conform to the requirements of NC-3300 and ND-3300.
Upset, emergency, and faulted	The vessel shall conform to the requirements of ASME Code Case 1607-1.

(1) As specified by ASME Section III, 1971 through Winter 1972 Addenda.

Table 3.9-15

DESIGN CRITERIA FOR NON-NSSS ASME CODE CLASS 2 VESSELS
DESIGNED TO ALTERNATE RULES OF NC-3200

CONDITION	STRESS LIMITS[1][2]
Design and normal	The vessel shall conform to the requirements of NC-3200.
Upset[3]	$P_e \leq 3\ S_m$
	$P_m \leq 1.1\ S_m$
	$(P_m \text{ or } P_L) + P_b \leq 1.65\ S_m$
Emergency	$P_m \leq$ greater of $1.2\ S_m$ or $1.0\ S_y$
	$(P_m \text{ or } P_L) + P_b \leq$ greater of
	$1.8\ S_m$ or $1.5\ S_y$
Faulted[4]	$P_m \leq 2.0\ S_m$
	$(P_m \text{ or } P_L) + P_b \leq 2.4\ S_m$

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(1) Definition of symbols:

P_m = General primary membrane stress intensity. This stress intensity is derived from the average value across the solid section under consideration. Excludes discontinuities and concentrations. Produced only by pressure and other mechanical loads.

P_L = Local primary membrane stress intensity. Same as P_m except that discontinuities are considered.

P_b = Primary bending stress intensity. Component of primary stress intensity proportional to distance from centroid of solid section. Excludes discontinuities and concentrations. Produced only by pressure and other mechanical loads.

P_e = Secondary stress intensity range. Developed by constraint of adjacent parts or by self-constraint of a structure. Considers discontinuities but not concentrations. Produced by mechanical loads and by thermal expansion.

Table 3.9-15 (Cont'd)

S_m = Design stress intensity value, ASME Section III, Appendix I, Table I-1.0.

S_y = Yield strength value, ASME Section III, Appendix I, Table I-2.0.

(2) These limits do not take into account either local or general buckling that might occur in thin-wall vessels. Such buckling shall be considered for upset conditions, but need not be considered for emergency or faulted conditions unless required by the design specification.

(3) Fatigue analysis requirements of NC-3219 and Appendix XIV are considered.

(4) As an alternative to satisfying these limits, the faulted condition stress limits of Appendix F may be applied provided that a complete analysis in accordance with NC-3211.1(c) is performed.

Table 3.9-16

DESIGN CRITERIA FOR NON-NSSS ASME CLASS 2 AND 3 PIPING

CONDITION	APPLICABLE CODE PARAGRAPH[1][2]	PRIMARY STRESS LIMITS
Design:		
Sustained Loads	NC, ND 3652.1	1.0 S_h
Occasional Loads	NC, ND 3652.2	1.2 S_h
Normal & Upset	NC, ND 3652.2 & 3611	1.2 S_h
Emergency	NC, ND 3611	1.8 S_h
Faulted	Code Case 1606	2.4 S_h

(1) As specified by ASME Code Section III, 1971 through Winter 1972 Addenda except the following:

Nuclear Class 2 and 3 flanges are analyzed in accordance with ASME Section III 1977 edition through 1979 Summer Addenda.

(2) Functional capability of essential piping is assured in accordance with NEDO-21985, September 1978.

Table 3.9-17

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Table 3.9-18

SEISMIC CATEGORY I SYSTEM SNUBBER DESIGN INFORMATION

DESIGN CRITERIA FOR NON-NSSS ASME CODE CLASS 2 AND 3 VALVES

CONDITION	STRESS LIMITS[2]
Design and normal	The valve shall conform to the requirements of Section III, Paragraphs NC-3500 and ND-3500
Upset, emergency[1], and faulted[1]	The valve shall conform to the requirements of ASME Code Case 1635-1

(1) Where valve function must be ensured (active valve) during the emergency or faulted condition, the specified emergency or faulted conditions for the plant shall be considered as the normal condition for the valve.

(2) As specified by ASME Section III, 1971 through Winter 1972 Addenda.

Table 3.9-19

SEISMIC CATEGORY I ACTIVE PUMPS AND VALVES
(GE SCOPE OF SUPPLY)

COMPONENT NAME	IDENTIFICATION NUMBER	GE MPL NUMBER
MSIVs	M41-F022(A,B,C,D) M41-F028(A,B,C,D)	B21-F022(A,B,C,D) B21-F028(A,B,C,D)
MSRVs	M41-F013(A,B,C,D E,F,G,H,J,K,L, M,N,S)	B21-F013(A,B,C,D, E,F,G,H,J,K,L, M,N,S)
CRD Vent and Drain Globe Valves	M47-F010 M47-F011 M47-F180 M47-F181	C11-F010 C11-F011 C11-F180 C11-F181
SLCS Pump	1AP208 1BP208 1CP208(1)	C41-C001A C41-C001B C41-C001C(1)
SLCS Relief Valves	M48-F029(A,B,C)	C41-F029(A,B,C)
SLCS Explosive Valves	M48-F004(A,B,C)	C41-F004(A,B,C)
RHR Pump	1AP202 1BP202 1CP202 1DP202	E11-C002A E11-C002B E11-C002C E11-C002D
Core Spray Pump	1AP206 1BP206 1CP206 1DP206	E21-C001A E21-C001B E21-C001C E21-C001D
RCIC Pump	10P203	E51-C001
RCIC Turbine	10S212	E51-C002
HPCI Pump	10P204	E41-C001
HPCI Turbine	10S211	E41-C002

Table 3.9-19 (Cont'd)

COMPONENT NAME	IDENTIFICATION NUMBER	GE MPL NUMBER
RHR System		
Globe Valves	M51-F015(A,B)	E11-F015(A,B)
Gate Valves	M51-F016(A,B)	E11-F016(A,B)
Gate Valves	M51-F017(A,B,C,D)	E11-F017(A,B,C,D)
Gate Valves	M51-F021(A,B)	E11-F021(A,B)
Globe Valves	M51-F027(A,B)	E11-F027(A,B)
Testable Check Valves	M51-F041(A,B,C,D)	E11-F041(A,B,C,D)
Testable Check Valves	M51-F050(A,B)	E11-F050(A,B)
Core Spray		
Gate Valves	M52-F001(A,B,C,D)	E21-F001(A,B,C,D)
Gate Valve	M52-F005	E21-F005
Testable Check Valves	M52-F006(A,B)	E21-F006(A,B)
Gate Valve	M52-F037	E21-F037
HPCI		
Swing-Check	M55-F005	E41-F005
Globe Valve	M55-F012	E41-F012
Stop-Check Valve	M55-F021	E41-F021
RCIC		
Globe Stop-Check	M49-F001	E51-F001
Swing-Check	M49-F014	E51-F014
Globe Valve	M49-F019	E51-F019

(1) This SLCS pump is not within the GE scope of supply.

Table 3.9-20

VALVE QUALIFICATION TEST RANGE
(NON-NSSS SCOPE OF SUPPLY)

QUALIFICATION VALID FOR OTHER VALVES (in)[(1)]

Valve Size Tested	½	1	1 ½	2	3	4	6	8	10	12	14	16	18	20	22	24	26	28	30	36
½	X	X																		
1	X	X	X																	
1½		X	X	X																
2			X	X	X															
3				X	X	X														
4					X	X	X													
6						X	X	X												
8							X	X	X	X										
10								X	X	X	X									
12								X	X	X	X									
14									X	X	X	X	X	X						
16										X	X	X	X	X	X					
18											X	X	X	X	X	X				
20											X	X	X	X	X	X	X	X		
22												X	X	X	X	X	X	X	X	
24													X	X	X	X	X	X	X	
26														X	X	X	X	X	X	X
28														X	X	X	X	X	X	X
30															X	X	X	X	X	X
36																	X	X	X	X

(1) Test data acquired for a qualified valve may be used to qualify valves of the same type that fall within the range of sizes permitted by this table, provided geometric similarity is maintained and supporting stress calculations are provided. If the qualified valve is larger than 36 inch nominal diameter, extrapolation may be made to valves whose nominal size does not vary more than 25% from that of the qualified valve.

Table 3.9-21

DESIGN LOADING COMBINATIONS FOR SUPPORTS FOR ASME CODE CLASS 1, 2, AND 3 COMPONENTS

CONDITION STRESS[(2)(3)]	DESIGN LOADING COMBINATIONS[(1)]	ALLOWABLE
Hydrostatic Test	a) HTDW	$0.8\ S_y$
Normal and Upset	a) $DW + TH + (OBE^2 + SRV_x^2)^{1/2}$	S_h
	b) $DW + TH + (RVC^2 + OBE^2)^{1/2}$	
	c) DW + TH + FV	
	d) DW + TH + OBE + RVO	
Emergency	a) $DW + TH + (OBE^2 + SRV^2_{ADS} + SBA^2)^{1/2}$	$1.8\ S_h$
	b) $DW + TH + (OBE^2 + FV^2)^{1/2}$	
Faulted	a) $DW + TH + (SSE^2 + SRV^2_{ADS} + IBA^2)^{1/2}$	$0.9\ S_y$
	b) $DW + TH + (OBE^2 + SRV^2_{ADS} + IBA^2)^{1/2}$	
	c) $DW + TH + (SSE^2 + DBA^2)^{1/2}$	

where:

HTDW = piping dead weight due to hydrostatic test

TH = reaction of the support due to thermal expansion of the pipe

S_y = yield stress

S_h = allowable stress per ANSI B31.1

See Table 3.9-11 for additional nomenclature.

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(1) Loads due to OBE, SSE, SRV_x, SRV_{ADS}, SBA, IBA, and DBA include both the inertia portion and the anchor motion portion when the response spectra method is used. The loads from the inertia portion and anchor motion are combined by the SRSS method.

(2) The allowable stress shall be limited to 2/3 of the critical buckling stress.

(3) Snubbers, compensating starts, and struts comply with all the requirements of ASME Section III, Subsection NF; they are not commercially available to meet the requirements of ANSI B31.1.

Table 3.9-22

FATIGUE LIMIT
(FOR SAFETY CLASS REACTOR INTERNAL STRUCTURES ONLY)

Summation of fatigue damage usage with design and operation loads following Miner hypotheses [1]

CUMULATIVE DAMAGE IN FATIGUE	LIMIT FOR SERVICE LEVELS A AND B (NORMAL AND UPSET) DESIGN CONDITIONS
Design fatigue cycle usage from analysis using the method of ASME Code	$\leq$1.0

(1) M.A. Miner, "Cumulative Damage in Fatigue," Journal of Applied Mechanics, 12, (67), pp. A159-164, (September 1945).

Table 3.9-23

CORE SUPPORT STRUCTURES
STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY FOR SERVICE LEVELS A AND B (NORMAL AND UPSET) CONDITIONS

STRESS CATEGORY	PRIMARY STRESSES: MEMBRANE P_m (4)(7)(8)	PRIMARY STRESSES: BENDING P_B (4)(7)(8)	SECONDARY STRESSES: MEMBRANE AND BENDING SECONDARY, Q (2)(4)(6)	PEAK STRESSES: PEAK, F (2)(4)(6)
	P_m	P_m+P_b	P_m+P_b+Q	P_m+P_b+Q+F
SERVICE LEVEL A AND B (NORMAL AND UPSET)	S_m ELASTIC ANALYSIS(6) OR $0.67\ L_L$ LIMIT ANALYSIS(10) OR $0.44\ L_u$ TEST(11)	$1.5\ S_m$ ELASTIC ANALYSIS(6) OR $0.67\ L_L$ LIMIT ANALYSIS(10) OR $0.44\ L_a$ TEST(11)	$3\ S_m$ ELASTIC ANALYSIS(1) OR S_L PLASTIC ANALYSIS(5)	S_a ELASTIC FATIGUE(3)(9)
			FOR CYCLES LESS THAN 1,000, USE PEAK (12)	P_m+P_b+Q+F — S_a ELASTIC PLASTIC FATIGUE(3)(9)(12)

Table 3.9-23 (Cont'd)

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(1) This limitation applies to the range of stress intensity. When the secondary stress is due to a temperature excursion at the point at which the stresses are being analyzed, the value of S_m shall be taken as the average of the S_m values tabulated in tables I-1.1, I-1.2, and I-1.3 of ASME Section III for the highest and the lowest temperature of the metal during the transient. When part of the secondary stress is due to mechanical load, the value of S_m shall be taken as the S_m value for the highest temperature of the metal during transient.

(2) The stresses in Category Q are those parts of the total stress which are produced by thermal gradients, structural discontinuities, etc, and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly and, when appropriate, this calculated value represents the total of $P_m + P_b + Q$ and not Q alone. Similarly, if the stress in Category F is produced by a stress concentration, the quantity F is the additional stress produced by the notch, over and above the nominal stress. For example, if a plate has a nominal stress intensity, $P_m = S$, $P_b = O$, $Q = O$ and a notch with a stress concentration K is introduced, then $F = P_m (K-1)$ and the peak stress intensity equals $P_m + P_m (K-1) = KP_m$.

(3) S_a is obtained from the fatigue curves, figures I-9.1 and I-9.2 of ASME Section III. The allowable stress intensity for the full range of fluctuation is 2 S_a.

(4) The symbols P_m, P_b, Q, and F do not represent single quantities, but rather sets of six quantities representing the six stress components σ_t, σ_1, σ_r, γ_{tl}, γ_{lr}, γ_{rt}.

(5) S_L denotes the structural action of shakedown load as defined in paragraph NB 3213.18 of ASME Section III calculated on a plastic basis as applied to a specific location on the structure.

(6) The triaxial stresses represent the algebraic sum of the three primary principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) for the combination of stress components. Where uniform tension loading is present, triaxial stresses are limited to 4 S_m.

(7) For configurations where compressive stresses occur, the stress limits shall be revised to take into account critical buckling stresses (see paragraph NB-3211(c) of ASME Section III). For external pressure, the permissible "equivalent static" external pressure shall be as specified by the rules of paragraph NB-3133 of ASME Section III. Where dynamic pressures are involved, the permissible external pressure shall be limited to 25% of the dynamic instability pressure.

(8) When loads are transiently applied, consideration should be given to the use of dynamic load amplification, and possible change in modulus of elasticity.

(9) In the fatigue data curves, where the number of operating cycles are less than 10, use the S_a value for 10 cycles; where the number of operating cycles are greater than 10^6, use the S_a value for 10^6 cycles.

Table 3.9-23 (Cont'd)

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(10) L_L is the lower bound limit load with yield point equal to 1.5 S_m (where S_m is the tabulated value of allowable stress at temperature as contained in ASME Section III). The "lower bound limit load" is here defined as that produced from the analysis of an ideally plastic (nonstrain hardening) material where deformations increase with no further increase in applied load. The lower bound load is one in which the material everywhere satisfies equilibrium and nowhere exceeds the defined material yield strength using either a shear theory or a strain energy of distortion theory to relate multiaxial yielding to the uniaxial case.

(11) For service levels A and B (normal and upset) conditions, the limits on primary membrane plus primary bending need not be satisfied in a component if it can be shown from the test of a prototype or model that the specified loads (dynamic or static-equivalent) do not exceed L_u, where L_u is the ultimate load or the maximum load to load combination used in the test. In using this method, account shall be taken of the size effect and dimensional tolerances which may exist between the actual part and the test part, or parts, as well as differences which may exist in the ultimate strength or other governing material properties of the actual part and the tested part to assure that the loads obtained from the test are a conservative representation of the load-carrying capability of the actual component under the postulated loading for service level A and B (normal and upset) conditions.

(12) The allowable value for the maximum range of this stress intensity is 3 S_m except for cyclic events which occur less than 1000 time during the design life of the plant. For this exception, in lieu of meeting the 3 S_m limit, an elastic-plastic fatigue analysis in accordance with ASME Section III may be performed to demonstrate that the cumulative fatigue usage attributable to the combination of these low events, plus all other cyclic events, does not exceed a fatigue usage value of 1.0.

Table 3.9-24

CORE SUPPORT STRUCTURES
STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY FOR SERVICE LEVEL C (EMERGENCY) CONDITIONS

STRESS CATEGORY	PRIMARY STRESSES: MEMBRANE P_m[1][2][10] (P_m)		PRIMARY STRESSES: BENDING P_B[1][2][10] (P_m+P_B)		SECONDARY STRESSES: MEMBRANE AND BENDING SECONDARY, Q	PEAK STRESSES: PEAK F
SERVICE LEVEL D (FAULT)[9]	$1.5\ S_m$	ELASTIC ANALYSIS[3]	$2.25\ S_m$	ELASTIC ANALYSIS[3]	EVALUATION NOT REQUIRED	EVALUATION NOT REQUIRED
	OR		OR			
	L_L	LIMIT ANALYSIS[4]	L_L	LIMIT ANALYSIS[4]		
	OR		OR			
	$1.5\ S_n$	PLASTIC ANALYSIS[6]	$2.25\ S_m$	PLASTIC ANALYSIS[5][6]		
	OR		OR			
	$0.6\ L_e$	TESTS[7]	$0.5\ S_u$	[5]		
	OR		OR			
	S_E	STRESS RATIO ANALYSIS[8]	$0.6\ L_e$	TESTS[7]		
			OR			
			KS_E	STRESS RATIO ANALYSIS[8]		

Table 3.9-24 (Cont'd)

(1) The symbols P_m, P_b, Q, and F do not represent single quantities, but rather sets of six quantities representing the six stress components σ_t, σ_l, σ_r, γ_{tl}, γ_{lr}, γ_{rt}.

(2) For configurations where compressive stresses occur, stress limits shall be revised to take into account critical buckling stresses. For external pressure, the permissible "equivalent static" external pressure shall be taken as 150% of that permitted by the rules of paragraph NB-3133 of ASME Section III. Where dynamic pressures are involved, the permissible external pressure shall satisfy the preceding requirements or be limited to 50% of the dynamic instability pressure.

(3) The triaxial stresses represent the algebraic sum of the three primary principal stresses ($\sigma_1 + \sigma_2 + \sigma_3$) for the combination of stress components. Where uniform tension loading is present, triaxial stresses should be limited to 6 S_m.

(4) L_L is the lower bound limit load with yield point equal to 1.5 S_m (where S_m is the tabulated value of allowable stress at temperature as contained in ASME Section III). The "lower bound limit load" is here defined as that produced from the analysis of an ideally plastic (nonstrain hardening) material where deformations increase with no further increase in applied load. The lower bound load is one in which the material everywhere satisfies equilibrium and nowhere exceeds the defined material yield strength using either a shear theory or a strain energy of distortion theory to relate multiaxial yielding to the uniaxial case.

(5) S_u is the ultimate strength at temperature. Multiaxial effects on ultimate strength shall be considered.

(6) This plastic analysis uses an elastic-plastic evaluated nominal primary stress. Strain hardening of the material may be used for the actual monotonic stress-strain curve at the temperature of loading or any approximation to the actual stress-strain curve which everywhere has a lower stress for the same strain as the actual monotonic curve may be used. Either the shear or strain energy of distortion flow rule shall be used to account for multiaxial effects.

(7) For service level C (emergency) conditions, the stress limits need not be satisfied if it can be shown from the test of a prototype or model that the specified loads (dynamic or static-equivalent) do not exceed 60% of L_e, where L_e is the ultimate load or the maximum load or load combination used in the test. In using this method, account shall be taken of the size effect and dimensional tolerances which may exist between the actual part and the tested part or parts as well as differences which may exist in the ultimate strength or other governing material properties of the actual part and the tested parts to assure that the loads obtained from the test are a conservative representation of the load-carrying capability of the actual component under postulated loading for service level C (emergency) conditions.

(8) Stress ratio is a method of plastic analysis which uses the stress ratio combinations (combination of stresses that consider the ratio of the actual stress to the allowable plastic or elastic stress) to compute the maximum load a strain hardening material can carry. K is defined as the section factor; $S_e \leq 2\ S_m$ for primary membrane loading.

(9) Where deformation is of concern in a component, the deformation shall be limited to two-thirds the value given for service level C (emergency) conditions in the Design Specification.

(10) When loads are transiently applied, consideration should be given to the use of dynamic load amplification and possible change in modulus of elasticity.

Table 3.9-25

MAXIMUM PRESSURE DIFFERENTIALS ACROSS REACTOR VESSEL INTERNALS DURING A STEAM LINE BREAK

Reactor Component	Pressure Differential(psid)[3] Case 1[1]	Case 2[2]
Core Plate and Guide Tube	24.5	24.5
Shroud Support Ring and Lower Shroud	48	49.0
Upper Shroud	26.5	28.5
Average Channel Wall (Bottom)	13.1	10.6
Top Guide	2.4	3.5

(1) Reactor initially at 1.02 of 110% original power, 110% recirculation flow

(2) Reactor initially at 20% rated steam flow, 110% recirculation flow

(3) Values taken from Reference 3.9-28

Table 3.9-26

CORE SUPPORT STRUCTURE DESIGN LOADING
CONDITIONS AND COMBINATIONS

OPERATING CONDITION AND STRESS LIMITS	SERVICE LEVEL	DESIGN LOADINGS CONDITIONS AND COMBINATIONS
Normal and Upset	A and B	$N + A_D$ and $N + U$
Emergency	C	N and R, or other conditions which have a 40 year encounter probability from 10^{-1} to 10^{-3}
Fault	D	N and A_m and $\overline{R}$ or other conditions which have a 40 year encounter probability from 10^{-3} to 10^{-6}

where:

N	=	service level A (normal) loads
U	=	service level B (upset) loads excluding earthquake
A_D	=	½SSE including any associated transients.
A_m	=	SSE
R	=	automatic blowdown or equivalent auxiliary pipe rupture loading (pipe rupture loadings are not directly considered on piping itself because this is handled by a failure mode analysis)
$\overline{R}$	=	primary loadings which result from rupture of a main steam line or a recirculation line

Table 3.9-27

DEFORMATION LIMIT
(FOR REACTOR INTERNAL STRUCTURES ONLY)

EITHER ONE OF (NOT BOTH)	GENERAL LIMIT
a. $\left[\dfrac{\text{Permissible deformation, DP}}{\text{Analyzed deformation causing loss of function, DL}}\right]$	$\leq \dfrac{0.9}{SF_{min}}$
b. $\left[\dfrac{\text{Permissible deformation, DP}}{\text{Experiment deformation causing loss of function, DE}}\right]^{(1)}$	$\leq \dfrac{1.0}{SF_{min}}$

where:

DP = permissible deformation under stated conditions of service levels A, B, C or D (normal, upset, emergency or faulted)

DL = analyzed deformation which could cause a system loss of function[2]

DE = experimentally determined deformation which could cause a system loss of function

SF_{min} = minimum safety factor

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(1) Equation b is not used unless supporting data are provided to the NRC by GE.

(2) "Loss of function" can only be defined quite generally until attention is focused on the component of interest. In cases of interest, where deformation limits can affect the function of equipment and components, they are specifically delineated. From a practical viewpoint, it is convenient to interchange some deformation condition at which function is assured with the loss of function condition if the required safety margins from the functioning conditions can be achieved. Therefore, it is often unnecessary to determine the actual loss of function condition because this interchange procedure produces conservative and safe designs. Examples where deformation limits apply are: CRD alignment and clearances for proper insertion, core support deformation causing fuel disarrangement or excess leakage of any component.

Table 3.9-28

PRIMARY STRESS LIMIT
(FOR SAFETY CLASS REACTOR INTERNAL STRUCTURES ONLY)

ANY ONE OF (NO MORE THAN ONE REQUIRED)		GENERAL LIMIT
a.	$\left[\dfrac{\text{Elastic evaluated primary stresses, PE}}{\text{Permissible primary stresses, PN}}\right]$	$\leq \dfrac{2.25}{SF_{min}}$
b.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Largest lower bound limit load, CL}}\right]$	$\leq \dfrac{1.5}{SF_{min}}$
c.	$\left[\dfrac{\text{Elastic evaluated primary stresses, PE}}{\text{Conventional ultimate strength at temperature, US}}\right]$	$\leq \dfrac{0.75}{SF_{min}}$
d.	$\left[\dfrac{\text{Elastic − plastic evaluated nominal primary stress, EP}}{\text{Conventional ultimate strength at temperature, US}}\right]$	$\leq \dfrac{0.9}{SF_{min}}$
e.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Plastic instability load, PL}}\right]^{(1)}$	$\leq \dfrac{0.9}{SF_{min}}$
f.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Ultimate load from fracture analysis, UF}}\right]^{(1)}$	$\leq \dfrac{0.9}{SF_{min}}$
g.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Ultimate load or loss of function load from test, LE}}\right]^{(1)}$	$\leq \dfrac{1.0}{SF_{min}}$

Table 3.9-28 (Cont'd)

where:

PE = primary stresses evaluated on an elastic basis. The effective membrane stresses are to be averaged through the load-carrying section of interest. The simplest average bending, shear or torsion stress distribution which will support the external loading will be added to the membrane stresses at the section of interest.

PN = permissible primary stress levels under service levels A or B (normal or upset) conditions under ASME Section III.

LP = permissible load under stated conditions of service levels A, B, C, or D (emergency or faulted).

CL = lower limit load with yield point equal to 1.5 S_m where S_m is the tabulated value of allowable stress at temperature of the ASME Section III Code or its equivalent. The "lower bound limit load" is here defined as that produced from the analysis of an ideally plastic nonstrain hardening material where deformations increase with no further increase in applied load. The lower bound load is one in which the material everywhere satisfies equilibrium and nowhere exceeds the defined material yield strength using either a shear theory or a strain energy of distortion theory to relate multiaxial yield to the uniaxial case.

US = conventional ultimate strength at temperature or loading which would cause a system malfunction, whichever is more limiting.

EP = elastic-plastic evaluated nominal primary stress. Strain hardening of the material may be used for the actual monotonic stress-strain curve at the temperature of loading or any approximation to the actual stress-strain curve which everywhere has a lower stress for the same strain as the actual monotonic curve may be used. Either the shear or strain energy of distortion flow rule may be used.

PL = plastic instability load. The "plastic instability load" is defined here as the load at which any load- bearing section begins to diminish its cross-sectional area at a faster rate than the strain hardening can accommodate the loss in area. This type analysis requires a true stress-true strain curve or a close approximation based on monotonic loading at the temperature of loading.

Table 3.9-28 (Cont'd)

UF = ultimate load from fracture analyses. For components which involve sharp discontinuities (local theoretical stress concentration <3) the use of a "fracture mechanics" analysis where applicable utilizing measurements of plane strain fracture toughness may be applied to compute fracture loads. Correction for finite plastic zones and thickness effects as well as gross yielding may be necessary. The methods of linear-elastic stress analysis may be used in the fracture analysis where its use is clearly conservative or supported by experimental evidence. Examples where "fracture mechanics" may be applied are for fillet welds or end of fatigue life crack propagation.

LE = ultimate load or loss of function load as determined from experiment. In using this method, account shall be taken of the dimensional tolerances which may exist between the actual part and the tested part or parts as well as differences which may exist in the ultimate tensile strength of the actual part and the tested parts. The guide to be used in each of these areas is that the experimentally determined load shall use adjusted values to account for material property and dimension variations, each of which has no greater probability than 0.1 of being exceeded in the actual part.

SF_{min} = minimum safety factor

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(1) Equations e., f., and g. are not used unless supporting data are provided to the NRC by GE.

Table 3.9-29

BUCKLING STABILITY LIMIT
(FOR SAFETY CLASS REACTOR INTERNAL STRUCTURES ONLY)

ANY ONE OF (NO MORE THAN ONE REQUIRED)		GENERAL LIMIT
a.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Service level A (normal) permissible load, PN}}\right]$	$\leq \dfrac{2.25}{SF_{min}}$
b.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Stability analysis load, SL}}\right]$	$\leq \dfrac{0.9}{SF_{min}}$
c.	$\left[\dfrac{\text{Permissible load, LP}}{\text{Ultimate buckling collapse load from test, SE}}\right]$ (1)	$\leq \dfrac{1.0}{SF_{min}}$

where:

LP = permissible load under stated conditions of service levels A, B, C or D (normal, upset emergency or faulted)

PN = applicable service level A (normal) permissible load

SL = stability analysis load. The ideal buckling analysis is often sensitive to otherwise minor deviations from ideal geometry and boundary conditions. These effects shall be accounted for in the analysis of the buckling stability loads. Examples of this are ovality in externally pressurized shells or eccentricity on column members.

SE = ultimate buckling collapse load as determined from experiment. In using this method, account shall be taken of the dimensional tolerances which may exist between the actual part and the tested part. The guide to be used in each of these areas is that the experimentally determined load shall be adjusted to account for material property and dimension variations, each of which has no greater probability than 0.1 of being exceeded in the actual part.

SF_{min} = minimum safety factor

(1) Equation c. is not used unless supporting data are provided to the NRC by GE.

Table 3.9-30

CORE SUPPORT STRUCTURES
STRESS CATEGORIES AND LIMITS OF STRESS INTENSITY FOR SERVICE LEVEL D (FAULT) CONDITIONS

STRESS CATEGORY	PRIMARY STRESSES: MEMBRANE P_m(1)(2)(10)		PRIMARY STRESSES: BENDING P_B(1)(2)(10)		SECONDARY STRESSES: MEMBRANE AND BENDING SECONDARY, Q	PEAK STRESSES: PEAK F
	P_m		P_m+P_B			
	$2.4\ S_m$	ELASTIC ANALYSIS(3)	$3.0\ S_m$	ELASTIC ANALYSIS(3)		
	OR		OR			
	$0.75\ S_u$	(5)(10)	$1.33\ L_L$	LIMIT ANALYSIS(4)		
	OR		OR			
	$1.33\ I_L$	LIMIT ANALYSIS(4)(11)	$0.75\ S_u$	PLASTIC ANALYSIS(5)(6)	EVALUATION NOT REQUIRED	EVALUATION NOT REQUIRED
	OR		OR			
	$0.67\ S_u$	PLASTIC ANALYSIS (5)(6)(11)	$0.8\ L_F$	TESTS(7)		
	OR		OR			
	$0.8\ L_F$	TESTS(7)(11)	KS_F	STRESS-RATIO ANALYSIS(8)		
	OR					
	S_F	STRESS-RATIO ANALYSIS(8)				

Table 3.9-30 (Cont'd)

(1) The symbols P_m, P_b, Q, and F do not represent single quantities, but rather sets of six quantities representing the six stress components δ_t, δ_l, δ_r, γ_{tl}, γ_{tr}, γ_{rl}.

(2) When loads are transiently applied, consideration should be given to the use of dynamic load amplification and possible changes in modulus of elasticity.

(3) For configurations where compressive stresses occur, stress limits take into account critical buckling stresses. For external pressure the permissible "equivalent static" external pressure shall be taken as 2.5 times that given by the rules of paragraph NB-3133 of ASME Section III. Where dynamic pressures are involved, the permissible external pressure satisfies the preceding requirements or be limited to 75% of the dynamic instability pressure.

(4) L_L is the lower bound limit load with yield point equal to 1.5 S_m (where S_m is the tabulated value of allowable stress at temperature as contained in ASME Section III). The "lower bound limit load" is defined as that produced from the analysis of an ideally plastic (nonstrain hardening) material where deformations increase with no further increase in applied load. The lower bound load is one in which the material everywhere satisfies equilibrium and nowhere exceeds the defined material yield strength using either a shear theory or a strain energy of distortion theory to relate multiaxial yielding to the uniaxial case.

(5) S_u is the ultimate strength at temperature. Multiaxial effects on ultimate strength are considered.

(6) This plastic analysis uses an elastic-plastic evaluated nominal primary stress. Strain hardening of the material may be used for the actual monotonic stress-strain curve at the temperature of loading or any approximation to the actual stress-strain curve which everywhere has a lower stress for the same strain as the actual monotonic curve may be used. Either the maximum stress or strain energy of distortion flow rule shall be used to account for multiaxial effects.

(7) For service level D (fault) conditions, the stress limits need not be satisfied if it can be shown from the test of a prototype or model that the specified loads (dynamic or static-equivalent) do not exceed 80% of L_F, where L_F is the ultimate load or load combination used in the test. In using this method, account is taken of the size effect and dimensional tolerances, as well as differences which may exist in the ultimate strength or other governing material properties of the actual part and the tested parts, to assure that the loads obtained from the test are a conservative representation of the load-carrying capability of the actual component under postulated loading for service level D (fault) condition.

(8) Stress ratio is a method of plastic analysis which uses the stress ratio combinations (combination of stresses that consider the ratio of the actual stress to the allowable plastic or elastic stress) to compute the maximum load a strain hardening material can carry. K is defined as the section factor; S_f is the lesser of 2.4 S_m or 0.75 S_u for primary membrane loading.

(9) Where deformation is of concern in a component, the deformation is limited to 80% of the value given for service level D (fault) conditions in the Design Specifications.

(10) 0.7 S_u per ASME Section III Appendix F.

(11) Same as ASME Section III Appendix F.

(12) 3.6 S_m per ASME Section III Appendix F.

Table 3.9-31

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Table 3.9-32

SRV TEST PROGRAM

	EVENTS [1][2]												
PLANT FEATURES	1	2	3	4	5	6	7	8	9	10	11	12	13
High Water Level 7 Alarm	X, S		X, S	X, S	X, NA				X, S	X, NA	X, S	X, S	X, NA
High Drywell Pressure Alarm													
FW Level 8 Trip	X, S	X, S											
RCIC Level 8 Trip			X, S	X, S	X, NA				X, S	X, NA	X, S		X, NA
HPCS Level 8 Trip				X, NA	X, NA				X, NA	X, NA			X, NA
HPCI Level 8 Trip			X, S	X, S					X, S		X, S		X, NA
HPCI/S and RCIS Initiation on Low Water Level	X, S	X, S	X, S	X, S	X, NA	X, NA		X, S	X, S				X, NA
HPCI/S Initiation of High Drywell Pressure			X, S	X, S					X, S	X, NA	X, S	X, S	X, NA
RCIC Initiation on High Drywell Pressure													X, NA
Low Pressure ECCS Initiation on High Drywell Pressure												X, S	X, NA
Low Pressure ECCS Initiation on Low Water Level													X, NA
FW Pumps Trip on Low Suction Pressure	X, S												
HPCI Trip on High Back pressure			X, S								X, S		
RCIC Trip on High Back pressure				X, S					X, S				
Turbine Trip on Vessel High Level	X, S	X, S											
MSIVs Closure on Low Turbine Inlet Pressure	X, S	X, S					X, S						
MSIVs Closure on High Steam Flow		X, S					X, S						

Table 3.9-32 (Cont'd)

PLANT FEATURES	EVENTS [1][2] 1	2	3	4	5	6	7	8	9	10	11	12	13
MSIVs Closure on High Steam Tunnel Temperature								X, S					
MSIV Closure on High Radiation								X, S					
Reactor Scram on Turbine Trip	X, S	X, S											
Reactor Scram on Nuetron Flux Monitor		X, S											
Reactor Scram on MSIVs Closure		X, S											
Reactor Scram on High Radiation								X, S					
Reactor Scram on High Drywell Pressure									X, S	X, NA	X, S	X, S	X, NA
Reactor Scram on Low Water Level													X, NA
Reactor Isolation on Low Water Level													X, NA

(1) Events

1 FW Cont. Failure, FW L8 Trip Failure
2 Pressure Regulator Failure
3 Transient HPCI, HPCI L8 Trip Failure
4 Transient RCIC, RCIC L8 Trip Failure
5 Transient HPCS, HPCS L8 Trip Failure
6 Transient RCIC Hd. Spr.
7 Alternate Shutdown Cooling, Shutdown Suction Unavailable
8 Main Steam Line Break - Outside Containment
9 SBA, RCIC, RCIC L8 Trip Failure
10 SBA, HPCS, HPCS L8 Trip Failure
11 SBA, HPCI, HPCI L8 Trip Failure
12 SBA, Depressurization & ECCS Overfill, Operator Error
13 LBA, ECCS Overfill Break Isolation

(2) X - Feature considered in Base Case Analysis
S - Feature in Plant Specific Design
NA - Not Applicable