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EMT
EQUIPMENT AND
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WSRC-TR-92-425
TASK 91-101-1

Process Water System
Acceptance Criteria
Heat Exchanger
Stainless Steel
Fracture
PTERM
CAMEL

Retention - Permanent

**HEAT EXCHANGER
HEAD AND SHELL ACCEPTANCE CRITERIA (U)**
Task Number: 91-101-1

By

P. S. Lam *P. S. Lam*
R. L. Sindelar *R. L. Sindelar*

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W. L. Daugherty

W. L. Daugherty

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Date

SRTC SAVANNAH RIVER TECHNOLOGY CENTER, AIKEN, SC 29808
Westinghouse Savannah River Company
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Approvals

N. K. Gupta Date: 10-23-92
N. K. Gupta, Technical Reviewer
SRTC - Materials Technology

James A. Begley Date: 10/9/92
J. A. Begley, Consultant
Packer Engineering, Inc.

James M. Morrison Date: 10-29-92
J. M. Morrison, Customer Review
RD - Technical Programs, Manager

N. G. Awadalla Date: 11-13-92
N. G. Awadalla, Manager
SRTC - Materials Technology

T. L. Capeletti Date: 11/13/92
T. L. Capeletti, Manager
SRTC - Materials Technology

EXECUTIVE SUMMARY

Monitoring the reactor process water system heat exchangers through in-service inspection is essential to demonstrate the structural integrity throughout service. An acceptance criteria methodology with technical bases is provided to disposition flaws detected during examination of the external pressure boundary components of the heat exchanger, namely, the head, shell, and cooling water nozzles. The methodology ensures that the defined safety margins¹ against failure are maintained throughout the heat exchanger service in full consideration of service-induced degradation.²

Acceptance criteria are established defining the acceptable flaw (length and depth) configurations for the heat exchanger head, shell, and cooling water nozzles. A large break or an equivalently double-ended guillotine break at the heat exchanger head could not occur with proper restraints supplied by the staybolts.³ For flaw stability evaluation, the heat exchanger head is divided into three general regions: Region I - a flat region near and including the head flange; Region II - a highly-curved region; and Region III - a pipe region near the 12-inch process water nozzle. The acceptable flaw lengths for these three regions are, respectively, 26", 16", and 23". The acceptable flaw lengths for the heat exchanger shell are, respectively, 6" and 30" in longitudinal (axial) and circumferential directions. The acceptable circumferential flaw length is 39" for the cooling water nozzles.

The acceptable flaw lengths are established based on the flaw stability solutions for throughwall cracks. To limit the flaw depths for leakage prevention, the acceptable depths for the flaws are set to 75% of the plate thicknesses for flaws with lengths up to the acceptable flaw length. This 75% depth criterion is consistent with the maximum acceptable depth provided by the ASME Boiler and Pressure Vessel Code.⁴

The stresses for evaluating the stability of postulated flaws are obtained by simple analytical and finite element solutions. Postulated flaws in the heat exchanger shell, where internal pressures are dominant, are analyzed by J-integral and limit load methods. Postulated flaws in the cooling water inlet nozzle and in the heat exchanger head pipe region near the process water inlet nozzle, where bending due to the seismic response is significant, are analyzed by the J-integral based limit load method. Flaws in the flat and the curved regions of heat exchanger head are analyzed by the J-integral method.

¹ The defined safety margins are adopted from Section XI of the American Society of Mechanical Engineers Boiler and Pressure Vessel Code, Article IWB-3642 for austenitic steel piping and are a factor of 3 on the normal operation (or design) loads and a factor of 1.5 on the loads under emergency/faulted (off-normal) conditions including transients. The normal loads include pressure, deadweight, and/or thermal loads. The faulted loads include normal operation plus seismic forces or water hammer pressures.

² The configuration and materials of construction of the K-reactor heat exchangers to be installed for post-cooling tower tie-in operation are not susceptible to service-induced degradation involving embrittlement, corrosion, or mechanical mechanisms. Monitoring the components provides information on the actual service condition.

³ "Heat Exchanger Staybolt Acceptance Criteria (U)," WSRC-TR-92-11, February 1992.

⁴ Section XI of the American Society of Mechanical Engineers Boiler and Pressure Vessel Code, Article IWB-3641, for austenitic steel piping.

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1.0 INTRODUCTION

1.1 Background

Analyses of the structural integrity of the Savannah River Site (SRS) reactor Process Water System (PWS) heat exchangers during service have been previously performed. The following conclusions are drawn from the summary of these analyses:

- a large break failure of a heat exchanger, equivalent to a double-ended guillotine break (DEGB) in the connected piping, is highly unlikely (failure frequency of 1×10^{-7} per reactor-year) [1];
- a large break failure of the heat exchanger head and shell must be preceded by detectable leakage [1, 2];
- and the structural integrity of the head against a large break failure is maintained through the redundant capacity of the staybolts, the primary restraint member; an acceptance criteria methodology was developed for in-service inspections (ISI) of the staybolts [3].

Instability of postulated flaws in the head component of the heat exchanger could not produce a large break, equivalent to a DEGB in the PWS piping, due to the configuration of the head and restraint provided by the staybolts [3]. Rather, leakage from throughwall flaws in the head would increase with flaw length with finite leakage areas that are bounded by a post-instability flaw configuration. Postulated flaws at instability in the shell of the heat exchanger or in the cooling water nozzles could produce a large break in the Cooling Water System (CWS) pressure boundary. An initial analysis of flaw stability for postulated flaws in the heat exchanger head was performed in January 1992 [4]. This present report updates that analysis and, additionally, provides acceptable flaw configurations to maintain defined structural or safety margins against flaw instability of the external pressure boundary components of the heat exchanger, namely the head, shell, and cooling water nozzles. Structural and flaw stability analyses of the heat exchanger tubes, the internal pressure boundary of the heat exchangers or interface boundary between the PWS and CWS, were previously completed in February 1992 [5] as part of the heat exchanger restart evaluation [6] and are not covered in this report.

The code-of-record for the SRS reactor PWS replacement heat exchangers (installed in K) is Section VIII, Division 1 (no stamp) of the ASME Boiler and Pressure Vessel Code [7]; in-service inspections to monitor the heat exchangers are not required per the construction code. The service history of the heat exchanger shows that service-induced degradation has led to several instances of cracking and leakage failures in the heads and shells [8]. Improvements in the design and materials of construction of the heat exchangers have been made throughout the reactor operating history to reduce or eliminate the conditions that have led to these failures [8]. The configuration and materials of construction for the heat exchangers in K reactor for operation following cooling tower tie-in should not lead to service-induced degradation of head and shells. In-service inspections of these external pressure boundary components provide a monitoring function of the heat exchangers and can show that service-induced degradation that led to cracking has been eliminated or that the degree of degradation is acceptable. The acceptance criteria for in-service inspections of the external pressure boundary components of the heat exchanger are provided in this report.

1.2 Acceptance Criteria for Flaw Disposition

Maintaining structural or safety margins throughout service is part of the structural integrity demonstration for components of the reactor PWS [1]. Acceptable flaw configurations or acceptance criteria for the reactor PWS heat exchanger heads, shells, and cooling water nozzles have been developed in this report to disposition flaws, both circumferential and axial, identified and characterized in the in-service inspections. The acceptance of flaws is in accordance with the ASME Boiler and Pressure Vessel (BPV) Code, Section XI [9], Article IWB-3640 to maintain ASME code-based factors of safety. Specifically, the factors of safety on loading specified by IWB-3642 for acceptance-by-analysis in austenitic stainless steel piping are adopted to define the safety margins to be maintained for the external pressure boundary components of the heat exchanger with flaws.

The development of the acceptable flaw configurations includes a stress analysis of the components with the IWB-3642 [9] factor of safety of 3 on normal and 1.5 on emergency/faulted (off-normal) loads; residual stresses are also included in the fracture mechanics analysis. Fracture mechanics analysis is performed to determine stability of postulated throughwall flaws at various orientations and locations in the component. A limit on part throughwall surface flaws to 75% of the thickness of the component, in accordance with IWB-3641 [9], is imposed to preclude inadvertent leakage. This depth criterion, together with the most limiting length for throughwall flaws under normal and off-normal conditions, define the acceptable flaw configuration for each external pressure boundary component. (Acceptance criteria based on leakage for throughwall flaws in the heat exchanger head were developed previously [10].) The leakage-based acceptance criteria are less restrictive than the present criteria limiting the throughwall depth to 75% of the thickness of the component. The leakage-based criteria provide monitoring sufficient to demonstrate the structural integrity of the head with service-induced flaws since these flaws can not lead to a large break failure. The structural integrity demonstration against a large break failure of the head is based upon the integrity of the staybolts, the primary head restraint member [3]. Furthermore, the leakage-based criteria [10] are adequate to disposition flaws confined to the weldment region of the half-pipe seal since this seal is not treated as a load-carrying pressure boundary component.

Section 2 of this report provides the materials of construction of the K-reactor heat exchangers for post-cooling tower tie-in operation, a summary discussion of the potential for service-induced degradation, and the mechanical properties of the heat exchanger materials applied in the structural and fracture mechanics analyses. Section 3 contains the details of the flaw stability analysis for the head, shell, and cooling water nozzles including the stress analyses (loading conditions and application of safety factors) and the fracture mechanics analyses. Section 4 contains the acceptance criteria for in-service inspection and Section 5 contains the acceptance criteria methodology for demonstrating the structural integrity of the external pressure boundary components of the heat exchangers throughout service. In-service inspection requirements such as flaw characterization (e.g., detection/reporting requirements, rules for combination of adjacent indications, etc.), reexamination standards and examination frequencies, are governed by the in-service inspection plan for the SRS reactors and are outside the scope of this report.

2.0 HEAT EXCHANGER MATERIALS

2.1 Materials of Construction

Three major, separate designs of heat exchangers, Type A, B, and C, have been in service in the SRS reactor PWS [8]; the materials of the pressure boundary components for each of these designs are listed below [the cooling water nozzles are the same material as the shell]. Design changes were made throughout the reactor operating history to reduce or eliminate leakage failures of the heat exchangers [8]. Reference 11 provides a summary of the heat exchanger designs and materials of construction during the service history of the heat exchangers.

Type "A" - Original Design (1952) [Vendors: Foster Wheeler; American Locomotive]

- Type 304 Tubes, Shell and Heads
- Double Tubesheet

Type "B" - 1978 Design [Vendors: Mitsui; Hitachi]

- Type 316L Tubes
- Type 304L Shell and Heads
- Single Tubesheet

Type "C" - 1984 Design [Vendor: Nooter]

- Sea Cure Tubes
- Type 304L Shell (from Nooter) and Heads (from Hitachi)
- Single Tubesheet

The materials of construction of the components of an individual heat exchanger depend upon the date of procurement and the specific configuration of the heat exchanger upon installation. The information contained in this section has been updated from the information for K-reactor restart in Reference 11 to include the planned configuration (design type and equipment number) and materials of construction of the K-reactor heat exchangers for post-cooling tower tie-in operation.

The Types B and C design will be installed for post-cooling tower tie-in operation in K reactor. The Type B heat exchangers to be installed are manufactured by Mitsui. The Type C heat exchangers are Nooter shells with Hitachi heads [11]. Table 1 is a summary list of the various materials of construction for the components of the heat exchangers to be installed in K reactor. The specific materials of construction and the corresponding serial numbers of the individual heat exchangers to be installed in K reactor are listed in Table 2 (updated from the K reactor restart configuration reported in Table 3 of Reference 11). Schematics of the details of the heat exchanger components and the locations of the components within the heat exchanger are provided in Figures 1-6 of Reference 11.

TABLE 1. Summary of Materials of Construction for K-reactor Heat Exchanger Components (updated from Reference 11)

Component Description	Materials of Construction
Head and Head Flange	Type 304L
Shell	Type 304L
Tubesheet	Type 316L
Tubes	Type 316L or Sea Cure*
Staybolt	Type 303
Omega Seal	Type 304L **
Bolt Hole Plugs	1-1/4" diameter 304-L bar stock
12" Flange	Type 304L
Core Rod	Type 304L
Boss Closure Assembly	Type 304L

* Sea-Cure is a ferritic stainless steel manufactured by the Trent Tube Division of Crucible Steel (identified as UNS S44660 under ASTM A-268).

** The seal membrane was fabricated from 1/2" Sch. 40 pipe (0.840" O.D., 0.109" wall thickness), Type 304L, 1/8" fillet weld.

TABLE 2. Materials of Construction for the Individual Heat Exchangers for Post-Cooling Tower Tie-in Operation of K Reactor (updated from Reference 11)

Location	Heat Exchanger Equipment No.	Design		Inlet Head	Outlet Head	Shell	Tubes	Tubesheet
		Type	Tubesheet					
1A	105-999-6	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
1B	105-999-9	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
2A	105-999-4	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
2B	105-150-4	B	Single	Type 304L	Type 304L	Type 304L	Type 316L	Type 316L
3A	105-150-3*	B	Single	Type 304L	Type 304L	Type 304L	Type 316L	Type 316L
3B	105-999-7	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
4A	105-999-8	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
4B	105-999-5	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
5A	105-999-10*	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
5B	1315-120-5*	B	Single	Type 304L	Type 304L	Type 304L	Type 316L	Type 316L
6A	105-999-3	C	Single	Type 304L	Type 304L	Type 304L	Sea Cure	Type 316L
6B	1315-120-6	B	Single	Type 304L	Type 304L	Type 304L	Type 316L	Type 316L

* Heat exchanger planned to be installed for post-cooling tower tie-in (as of 8-31-92)

2.2 Service History and In-Service Degradation Summary

A detailed service history of the PWS heat exchangers has been compiled by the Reactor Engineering Department and is provided in Reference 8. Vibration, wear, and corrosion-induced degradation of the heat exchanger tubes were responsible for most of the leakage failures of the heat exchangers. Modifications made to the original design (Type A) heat exchangers include plugging tubes subject to vibration or replacing tubes with rods, installation of cooling water inlet distributor and strainer, and improvement in the corrosion-resistance of the tubing materials.

The structural capacity and flaw tolerance of the heat exchanger tubes has been analyzed previously [5]. Additional information regarding the fitness of the heat exchangers to minimize tube leakage during service is provided in Reference 6.

Service-induced degradation of the heads and shells is discussed in the following sections as summarized from References 8 and 12.

2.2.1 Degradation of the Heat Exchanger Head

Corrosion-induced degradation has led to leakage failures in the heads [8]. Chloride-induced stress corrosion cracking at the staybolt bosses and head flange was caused by contact with elastomeric seals at these locations. All gaskets and O-rings exposed to moderator (at the head staybolt bosses and the head-to-tubesheet closures) were eliminated and a welded boss closure of the staybolt and a half-pipe seal (Omega Seal) between the heat exchanger head and the tubesheet made from a 1/2" 40S half-pipe with a 1/8" fillet weld were constructed. In addition, the head vent and drain lines were relocated and the bolt holes in the head flange were filled by a seal weld and welding insert plugs. The present configuration for closure of the heads (inlet and outlet) to the shell section is by a seal weld and 72 C-clamps around the periphery of the head and tubesheet and by a series of 84 staybolts between the head and the tubesheet. References 8 and 11 provide additional details and figures of this configuration. The design changes to remove the elastomeric seals eliminated chloride-induced stress corrosion cracking as a potential degradation mechanism in the head.

Intergranular stress corrosion cracking (IGSCC) has also occurred in the heads made of Type 304 stainless steel and sensitized during fabrication [12]. In-service inspection of heads was initiated in 1978 to monitor the leakage due to head flange cracking. A summary of the inspections is in Reference 12. All heads now installed in K Reactor made of Type 304L stainless steel, with carbon content less than 0.03 wt%, would not typically sensitize during joining and, therefore, would not be susceptible to IGSCC. Weld metal with low carbon and controlled ferrite (such as 308L with 8% minimum ferrite) is resistant to sensitization and IGSCC.

2.2.2 *Degradation of the Heat Exchanger Shell*

Corrosion-induced failures by cracking and leakage have occurred in the shells at the circumferential girth welds [8]. The cracking was attributed to transgranular stress corrosion cracking due to chlorides leaching from slag inclusions in the weldments formed during the fabrication of the shell with manual metal-arc welding with coated electrodes. This type of corrosion degradation is eliminated in the newer (Type B and C) heat exchangers with a change in the welding procedures.

Cracking or leakage failures due to intergranular stress corrosion have not occurred in the heat exchanger shell materials [8]. The Type 304L shell weldments should not be susceptible to IGSCC in the cooling water since the material would not typically sensitize during joining.

2.3 Mechanical Properties

2.3.1 *Tensile Properties for Type 304L Stainless Steel*

Tensile properties for Type 304L stainless steel are from Section II of the ASME Boiler and Pressure Vessel Code [13] and from site-specific data [14, 15]. The flow stress, σ_f , applied in limit load analyses of the piping [14], is taken as $3S_m$ where S_m is the stress intensity value from Section II of the code [13]. [The material design parameters in Section II were originally part of Appendix I to Section III of the ASME BPV code prior to the 1992 edition of the code.] The stress intensity value for Type 304L plate (SA-240), forgings (SA-182), and welded or seamless pipe (SA-312) is 16.7 ksi for temperatures from -20 to 300°F. Therefore, the flow stress is calculated to be 50.1 ksi. The modulus of elasticity (Young's modulus), E , at a nominal temperature of 200°F is given [13] as 27.6×10^3 ksi. No interpolation of the ASME values for E at temperatures between the minimum and maximum operating temperatures is made; the value of E at 200°F is conservative in predicting the material tolerance of flaws at temperatures lower than 200°F. A value of 0.3 is taken as Poisson's ratio.

Ramberg-Osgood parameters [14] for the tensile response are applied in the elastic-plastic fracture analysis of Type 304L stainless steel material (see Section 3.2). Tensile specimens of Type 304L and archival Type 304 stainless steel were tested as part of the SRTC Reactor Materials Program; the results of tests of Type 304L stainless steel specimens at 125°C (257°F) are shown in Reference 15 and the Type 304 stainless steel specimens are shown in Figure II-1a in Reference 14. The tensile curves of the 304L specimens (F50-113, -101, -86, and -108) are within the envelope of curves of the Type 304 specimens; the average yield/tensile strengths and total elongations for the Type 304L specimens (29 ksi/69 ksi, 52%) are nearly identical to those for the Type 304 specimens (29ksi/70ksi, 61%) listed in Table 5-1 of Reference 14. Therefore, the Ramberg-Osgood parameters with values of $\alpha=2$ and $n=5$ from the data set of the archival Type 304 stainless steel materials (Figure II-1b of Reference 14) are applied in the calculation of the J-integral (see Section 3.2).

2.3.2 Elastic-Plastic Fracture Toughness of Type 304L Stainless Steel

Elastic-plastic fracture toughness properties for Type 304L stainless steel material are from data on plate material of Type 304L measured as part of the mechanical testing program for the SRTC Reactor Materials Program. Two compact tension specimens (F50-76 and F50-77) of Type 304L plate material with a 0.4T planform design [14] were tested at 257°F [15]. The average of the parameters for the power law fit to the J-R curve results give the following for the material J-R curve (deformation J):

$$J \left[\frac{\text{lb}}{\text{in}} \right] = 9370(\Delta a)^{0.3847}$$

with Δa , the crack extension, in inches. The limit on Δa for J-controlled crack extension is suggested to be 0.1181 inches [14]. The corresponding limit of J at Δa of 0.1181 inches is 4120 lb/in.

The tearing modulus, T, is defined by:

$$T = \frac{E}{s_f^2} \frac{dJ}{da}$$

where s_f is defined as the average of the yield and tensile (engineering) strengths [14]. With the average flow stress data at 257°F from tensile specimens F50-113, -101, -86, and -108, the material tearing modulus can be formulated as:

$$T = \frac{42.21}{\Delta a^{0.6153}}$$

with Δa in inches. The power law formulation for J(Δa) along with the tearing modulus, T(Δa), give the material J-T curve shown in Figure 1.

3.0 FRACTURE ANALYSIS

3.1 Overview

Fracture analysis was performed for postulated throughwall flaws in the external pressure boundary components of the heat exchanger: the shell, the cooling water inlet nozzle, and the head. The loadings include normal operation (design conditions were also applied to bound normal operation) and separate cases of off-normal conditions. A safety factor of 3 (for normal and design conditions) or 1.5 (for off-normal or emergency/faulted conditions) was applied to the calculated stresses at the corresponding loading conditions for input to the fracture analysis to determine instability flaw lengths. The acceptable flaw length for each component is the shortest of the instability lengths or the most limiting length from the flaw stability results of the different fracture analysis methods and loading cases presented in this section.

Limit load and J-integral methods for pipes or cylinders were applied to postulated axial and circumferential cracks in the shell part of the heat exchanger. The J-integral based limit load analysis (JiLLA), developed for SRS piping [16], was applied to analyze postulated flaws at the circumferential weld of the cooling water inlet nozzle and in the pipe region (PWS inlet) of the heat exchanger head. The postulated flaws in the remaining part of the head were analyzed by the J-integral method.

A finite element model for the heat exchanger head was built for the development of acceptance criteria for SRS heat exchanger head staybolts [3]. This model was used to calculate the stress distribution in the head including seismic loads [17] in the off-normal case. A double curvature correction was developed for the J-integral solution in the highly-curved region of the head.

The acceptable flaw lengths in the heat exchanger shell are 6" in the axial direction (based on limit load analysis), and 30" in the circumferential direction (based on J-integral analysis). The acceptable flaw length for the cooling water inlet nozzle is 19". For the heat exchanger head, the most limiting acceptable flaw length is 16" which occurs in the highly-curved area (Region II).

3.2 Heat Exchanger Shell

The heat exchanger shell is a cylinder with mean radius (R) of 44" and thickness (t) of 0.5". The length of the shell is 350.5" [18]. About 9000 tubes pass through the shell and are supported by baffles and the tubesheets which are attached at two ends to the heat exchanger inlet and outlet heads (Section 3.4) by 84 staybolts. The cooling water enters through an inlet nozzle (Section 3.3) in the bottom of the shell at the PWS outlet end of the heat exchanger and exits through an outlet nozzle in the top of the shell at the PWS inlet end.

As described in Section 2, the material of construction is Type 304L stainless steel, the flow stress ($3S_m$) is 50.1 ksi, and the Young's modulus (E) is 27.6×10^6 psi. The material J-T curve shown in Figure 1 is truncated at a crack extension limit of 3 mm (or 0.1181"). The corresponding J-integral limit is 4120 lb/in with a tearing modulus, T, of 157.

Safety factors of 3 or 1.5 are applied in the analysis based on the types of loading conditions described in Section 3.2.1. In the evaluation to determine the acceptable flaw lengths in axial (longitudinal) and circumferential (hoop) directions, analysis of postulated

flaws is performed with both the limit load and the elastic-plastic J-integral (J-T) methods. The most limiting results between these two methods together with the most limiting results for flaw stability under normal and off-normal loading conditions are chosen for the acceptance criteria for disposition of postulated shell flaws (see Section 4).

3.2.1 Loading Conditions and Safety Factors

The following loading conditions are considered in the calculation of throughwall flaw instability lengths for the heat exchanger shell:

- (1) Design pressure: 150 psi (normal operating pressure is 50 psi at cooling water inlet) [3].
- (2) Bending moment due to the weight of heat exchanger (251,000 lbs, [18]): 4,820,000 in-lb. The moment is estimated by assuming that the weight is uniformly distributed across the span of heat exchanger and is supported by two seismic bracings.
- (3) Water hammer pressure at partial check valve closure: 182 psia (168 psig) for January 1989 event in 190-K [19].
- (4) Water hammer pressure at full check valve closure: 256 psia (242 psig) for January 1989 event in 190-K [19].
- (5) Seismic response: The longitudinal force is 17,900 lb and the bending moment is 1,790,000 in-lb [17]. The cooling water pressure is assumed at normal condition (50 psi).
- (6) Heat exchanger tube rupture: Shell may be exposed to process water design pressure of 300 psi. [Note that the normal operating pressure at the heat exchanger inlet head is 218 psig or, at startup, is 271 psig [3].]

The hoop and longitudinal stresses (σ_H and σ_L , respectively) due to pressure loading (P) are estimated according to:

$\sigma_H = \frac{PR}{t}$ and $\sigma_L = \frac{PR}{2t}$. The maximum bending stress (σ_b) is $\frac{MR_o}{I}$, where R_o is the outside radius of the shell (44.25"), M is the applied bending moment, I is the moment of inertia of the shell (133,811 in⁴) and t is the thickness of the shell (0.5").

In the evaluation of flaw stability, a safety factor of 3 is applied to the design condition. For faulted conditions such as water hammer, seismic and tube rupture events, the safety factor 1.5 is applied to the resultant stresses.

3.2.2 Limit Load Analysis

The limit load analysis was carried out separately for axial flaws and for circumferential flaws. The contribution from residual stress is not applied in the limit load analysis since the method is based on net section plastic collapse whereby the residual stresses would have been relieved. The flow stress (σ_f) is $3S_m$ [14] (Section 2.3.1).

3.2.2.1 Axial Cracks

The axial crack extension criterion for high-toughness vessels was proposed by Hahn et al. [20] as

$$\sigma_f = M \sigma_H.$$

The factor $M = \sqrt{1 + 1.61 \lambda^2}$ was used by Eiber [21], where $\lambda = \frac{a}{\sqrt{Rt}}$, and a is the half crack length. The acceptable flaw length can be obtained by solving for a in these equations under various loading conditions described in Section 3.2.1. Only the hoop stress due to internal pressure was considered as the applied load for longitudinal cracks.

Alternatively, the factor M used by Zahoor [22] is

$M = \sqrt{1 + 1.2987 \lambda^2 - 0.026905 \lambda^4 + 5.3549 \times 10^{-4} \lambda^6}$. The expression of M reported by Eiber gives slightly shorter crack length results compared to those with the M factor of Zahoor. The M factor of Eiber is applied in the analysis for conservatism.

3.2.2.2 Circumferential Cracks

The applied loads include the longitudinal stress (σ_L) due to internal pressure of the shell, and the bending stress (σ_b) due to the weight of heat exchanger and the seismic loads. However, the equation for a circumferential crack under combined tension and bending provided by Zahoor [23] poses a limitation on the range of applicability, that is, the applied stress in tension (in this case, σ_L) cannot exceed 25% of the flow stress (σ_f). For the shell material with a flow stress of 50.1 ksi, the limit for the internal pressure of the shell is, equivalently, 285 psi which is relatively low when the safety factors are applied. Therefore, the acceptable flaw length is calculated based on the solution for a circumferential crack under tension [23]:

$$\frac{\sigma}{\sigma_f} = \frac{1}{\pi} \left[2 \cos^{-1} \left(\frac{1}{2} \sin \theta \right) - \theta \right],$$

where θ is the half crack angle ($\theta = a/R$).

In the above equation, the bounding applied stress (σ) is the sum of the longitudinal stress and the maximum bending stress (i.e., $\sigma = \sigma_L + \sigma_b$). The crack length calculated by the above equation is more limiting than that by the combined tension and bending formulation.

3.2.3 Elastic-plastic J-integral Analysis (J-T Method)

The J-integrals were calculated for cracks in axial and circumferential directions. The crack growth ($J \geq J_{IC}$) is stable if $T < T_R$, where T_R is the tearing modulus of the material defined by replacing J by the J-resistance of the material, J_R , in the expression of tearing modulus (Section 2.3.2). In practice, the intersection point of the applied J-T curve and the material J-T curve will determine the crack growth stability limit [24]. It was observed in previous analyses [26] of SRS reactors and pipings that these curves do not intersect, probably because the applied loads were low and J values were dominated by its elastic portion. The

same condition is observed in this analysis. Under these circumstances, cut-off J values at the crack growth limit obtained in material testing (Section 2.3.2) are used to determine the instability crack length.

The elastic-plastic solutions for J-integral in a pipe or cylinder are not available for the size of heat exchanger shell ($R/t=88$). A solution of J-integral for a center-cracked panel (CCP) under plane stress condition for Ramberg-Osgood materials was obtained by Shih and Hutchinson (Section 3.2.3.1 [25]). This analytical solution provides a basis to construct an approximate solution for the geometry of the heat exchanger shell by a curvature correction factor (Section 3.2.3.2). The validity of this correction was evaluated by comparison to a finite element solution for the case of circumferential cracks as discussed in the appendix.

The procedure of applying the J-T methodology to postulated flaws in the heat exchanger shell is summarized in the following. [This procedure is slightly modified from the procedure in References 24 and 26.]

- (1) For a given applied remote stress, calculate the CCP solution of Shih and Hutchinson for various crack lengths. The J-integral (J^{ccp}) is composed of an elastic portion (J_{el}^{ccp}) and a plastic portion (J_{pl}^{ccp}), that is,

$$J^{ccp} = J_{el}^{ccp} + J_{pl}^{ccp}.$$

- (2) The plastic zone size correction (or small scale yielding correction) is applied to J_{el}^{ccp} .
- (3) For a given crack length, calculate the ratio (Y) of the stress intensity factor for a cylinder with the same R/t to $\sigma\sqrt{\pi a}$, the stress intensity factor for an infinite flat plate.
- (4) Analogous to a compounding method for approximating stress intensity factors, and noting that J is proportional to the square of the stress intensity factor in linear elastic fracture mechanics, the corrected values of elastic and plastic components of J for the heat exchanger shell (J_{el}^{cur} and J_{pl}^{cur}) are

$$J_{el}^{cur} = Y^2 J_{el}^{ccp} \text{ and}$$

$$J_{pl}^{cur} = Y^2 J_{pl}^{ccp}, \text{ respectively.}$$

- (5) The contributions of fracture parameters from other sources, such as thermal stress or residual stress, can be combined in the sense of linear elastic fracture mechanics. The elastic portion of J-integral (J_{el}^{cur}) in (4) above is first converted to K_I^{appl} , the Mode I stress intensity factor due to applied loads.

$$K_I^{appl} = \sqrt{E J_{el}^{cur}}, \text{ for the plane stress condition.}$$

- (6) In the present analysis, only residual stress is considered. The stress intensity factor due to the residual stress (K_I^{res}) was estimated by General Electric [24] and used in several analyses [26, 27] as an approximation:

$$K_I^{res} = \frac{27.067\sigma_r}{45}(1 - e^{-0.997a})$$

where K_I^{res} is in $\text{ksi}\sqrt{\text{in}}$, σ_r is in ksi and is the maximum amplitude of the residual stress distribution across the thickness of the plate. The residual stress distribution is a self-equilibrium, symmetric pattern with maximum tension ($+\sigma_r$) on the edges and maximum compression ($-\sigma_r$) in the mid-section of the plate. The through-thickness variation from tension-compression-tension is linear. The above equation for K_I^{res} is consistent with the results of recent analysis by Green and Knowles ($K_{I\max}^{res} = 0.43\sigma_r\sqrt{\pi t}$, [28]) when the plate thickness t is 0.5". Note that the K_I^{res} is saturated to a maximum value when the crack is extended in length only a fraction of the plate thickness. Therefore, the residual stress of this type is not subject to curvature correction.

- (7) The total elastic portion of J is calculated as

$$J_{el} = \frac{1}{E}(K_I^{cpl} + K_I^{res})^2$$

The plastic portion of J remains unchanged, that is, $J_{pl} = J_{pl}^{cur}$.

- (8) Finally, the total J -integral of the crack is

$$J = J_{el} + J_{pl}$$

The expression for J in CCP is described in Section 3.2.3.1 and the curvature correction factor Y^2 is calculated in Section 3.2.3.2. The J versus the half crack length a can be plotted. The slope of this curve can be obtained numerically to construct a J -T curve. The intersection point of the applied J -T curve and the material J -T curve determines the flaw size at the stability limit. In the case of heat exchanger shell (also in the reactor tank analysis [26]), no intersection points can be found within the range of test results for the material J -T curve. Therefore, the limiting value of J ($J = 4120 \text{ lb/in}$) is applied as a cut-off in the material J -T curve to define the acceptable flaw size (Section 3.2).

3.2.3.1 Center-Cracked Panel (CCP) Solution - Finite Plate

The J -integral solution for CCP [25] is used to construct approximate J -integrals for the postulated cracks in the heat exchanger shell. Both the axial and the circumferential cracks can be treated in a similar manner prior to curvature correction. The correspondence between the width of CCP and the shell or cylinder geometry can be seen in Figures 2 and 3. This section describes the procedure for calculating J -integral in a CCP with a finite size.

The J -integral solution for a Ramberg-Osgood material was derived in Reference 25 as

$$\frac{J}{\sigma_o \epsilon_o a (1-a/b)} = \psi \left(\frac{P}{P_o} \right)^2 g_1 \left(\frac{a_{eff}}{b}, n=1 \right) + \alpha \left(\frac{P}{P_o} \right)^{n+1} g_1 \left(\frac{a}{b}, n \right).$$

It can be expressed in terms of the notations used in Section 3.2.3, that is,

$$J_{el}^{cp} = \psi \sigma_o \epsilon_o a (1-a/b) \left(\frac{P}{P_o} \right)^2 g_1 \left(\frac{a_{eff}}{b}, n=1 \right)$$

and

$$J_{pl}^{cp} = \alpha \sigma_o \epsilon_o a (1-a/b) \left(\frac{P}{P_o} \right)^{n+1} g_1 \left(\frac{a}{b}, n \right),$$

where a is the half crack length, b is the half specimen width (Fig.2 or 3), σ_o is set to $3S_m$ or 50.1 ksi for the material of construction (304L stainless steel),

$$\epsilon_o = \sigma_o / E,$$

$$a_{eff} = a + \phi r_y, P \leq P_o \quad \dagger$$

$$a_{eff} = (a_{eff})_{P=P_o}, P > P_o,$$

$$\phi = \frac{1}{1 + (P/P_o)^2},$$

$P_o = 2(b-a)\sigma_o$ is the lower bound limit load,

$P = 2b\sigma^\infty$ is the applied load corresponding to a remote stress σ^∞ ,

$$r_y = \frac{1}{2\pi} \left(\frac{n-1}{n+1} \right) \left(\frac{K_I}{\sigma_o} \right)^2 = \frac{a}{2\pi} \left(\frac{n-1}{n+1} \right) \left(1 - \frac{a}{b} \right) \left(\frac{P}{P_o} \right)^2 g_1 \left(\frac{a}{b}, 1 \right) \text{ for plane stress, and}$$

$$\psi = \frac{a_{eff}}{a} \left(\frac{b-a}{b-a_{eff}} \right).$$

The function g_1 is calculated according to

$$G(a/b, n) = g_1(a/b, n) \left[\frac{1 + \zeta(n-1)a/b}{c\sqrt{n} (1 - 1/n) + g_1(a/b, 1) / n} \right]$$

which is given in Reference 25. The best choice of the constants was found by Shih and Hutchinson, that is, $\zeta = 1.40$ and $c = 3.85$. The function $g_1(a/b, 1)$ is known from the elastic solution:

[†] This equation is a modification of the one that originally appeared in Reference 25 by introducing a parameter ϕ which was proposed in "Fully Plastic Crack Solutions, Estimation Scheme, and Stability Analyses for the Compact Specimen," by V. Kumar and C. F. Shih, in Fracture Mechanics: Twelfth Conference, ASTM STP 700, American Society for Testing and Materials, 1980, pp. 406-438.

$$g_1(a/b, 1) = \pi [1 - 0.5a/b - 0.370(a/b)^2 - 0.044(a/b)^3]^2.$$

For 304L stainless steel, Ramberg-Osgood exponent n is 5. The values of the function $G(a/b, n)$ are tabulated in Reference 25 and reproduced here.

a/b	$G(a/b, n=5)$
0	1.005
1/8	1.031
1/4	1.037
1/2	0.939
3/4	0.860
1	0.800

Interpolation was used to calculate $G(a/b, n=5)$ for values of a/b from 0 to 1.

3.2.3.2 Curvature Corrections

The size of the heat exchanger shell prohibits direct use of existing J-integral solutions found in the literature. A curvature correction is applied to estimate the approximate J-integral values of the shell flaws. In linear elastic fracture mechanics, the stress intensity factor (K) for a crack can usually be expressed in a general form of $K = Y \sigma \sqrt{\pi a}$, where σ is the applied stress and Y ($Y = Y_1$ for axial crack and $Y = Y_2$ for circumferential crack) is a function of crack size and specimen dimensions. The curvature correction factor, used for estimating J-integral of a crack in a curved structure based on a corresponding flat plate solution with the same crack length, is therefore Y^2 . The degree of validity of this procedure is demonstrated in the appendix for circumferential throughwall cracks.

The curvature correction factors for the heat exchanger shell or a pipe can be obtained with handbook solutions, such as Reference 29 by Erdogan, and Reference 30 by Rooke and Cartwright. The solutions provided by Tada et al. [31] are simpler to use and the solutions are better behaved in some instances.

(i) Axial (Longitudinal) Cracks

From Reference 31, the stress intensity factor of an axial crack with length $2a$ subjected to a hoop stress σ_H in a cylinder with mean radius R and thickness t is

$$K_I = \sigma_H \sqrt{\pi a} Y_1(\lambda),$$

where $\lambda = \frac{a}{\sqrt{Rt}}$ has been defined in Section 3.2.2.1,

$$Y_1(\lambda) = \sqrt{1 + 1.25 \lambda^2} \text{ for } 0 < \lambda \leq 1,$$

and

$$Y_1(\lambda) = 0.6 + 0.9 \lambda, \text{ for } 1 \leq \lambda \leq 5.$$

In this case the curvature correction factor for J-integral for an axial or a longitudinal crack is Y_1^2 .

(ii) Circumferential Cracks

For a circumferential crack with length $2a$ or angle 2θ subjected to a longitudinal stress σ_L , the stress intensity factors, based on Reference 31 are

$$K_I = \sigma_L \sqrt{\pi a} Y_2(\lambda \text{ or } \theta),$$

where

$$Y_2(\lambda) = \sqrt{1 + 0.3225 \lambda^2} \text{ for } 0 < \lambda \leq 1,$$

$$Y_2(\lambda) = 0.9 + 0.25 \lambda, \text{ for } 1 \leq \lambda \leq 5,$$

and

$$Y_2(\theta) = \sqrt{\frac{2}{\eta \pi \theta}} f(\theta), \text{ for } \lambda > 5.$$

In the last equation,

$$\eta^2 = \frac{t/R}{\sqrt{12(1-\nu^2)}}$$

and

$$f(\theta) = \theta + \frac{1 - \theta \cot \theta}{2 \cot \theta + \sqrt{2} \cot \left(\frac{\pi - \theta}{\sqrt{2}} \right)}.$$

Therefore, the curvature correction factor for J-integral for a circumferential crack is Y_2^2 .

3.3 Cooling Water Inlet Nozzle

The cooling water nozzle is a 3/8" thick pipe circumferentially welded to the heat exchanger shell. The nozzle loads are described in Section 3.3.1. A J-integral based limit load analysis (JiLLA, Section 3.3.2), originally developed for leak-before-break of reactor process water piping system [16, 32], is adopted to evaluate the acceptable flaw length for the cooling water inlet nozzle. Section 3.3.2 describes this procedure.

3.3.1 Loading Conditions and Safety Factors

The normal operating pressure for cooling water inlet is 50 psi and the design pressure is 150 psi [3]. The nozzle loads reported in Reference 17 are 10,100 lb in its axial direction and a resultant bending moment of 575,000 in-lb. These loads include seismic, thermal, and gravity conditions. Adding a longitudinal stress due to the normal operating pressure (50 psi), the total applied membrane stress in the axial direction becomes 1126 psi. The maximum bending stress is calculated to be 3552 psi. These applied stresses with a safety factor of 1.5 are used in the fracture analysis (JiLLA) in Section 3.3.2.

A direct limit load analysis involving only the design pressure with safety factor of 3 was also carried out. It was found that the acceptable flaw length obtained by JiLLA with the seismic loads and safety factor of 1.5 is more limiting.

3.3.2 J-Integral Based Limit Load Analysis (JiLLA)

The results in the reactor process water piping leak-before-break analysis [16, 33] showed that the theoretical limit load result for a circumferential crack under bending could accurately predict the maximum loads in experiments, especially for small diameter pipes (e.g., 4" diameter piping). However, the limit load solutions (such as that reported in Section 3.2.2.2) do not contain explicit information of pipe diameter and wall thickness; the acceptable flaw size based on a given load may be over-estimated for large pipe sizes. On the other hand, the J-T approach with General Electric or Electric Power Research Institute (EPRI) estimation scheme does contain the pipe size information, but the measured maximum loads in experiments are underpredicted.

To use the limit load solution and to maintain the pipe size information, an empirical scaling factor (M') was developed to scale the limit load results for pipes under bending. The M' factor is the ratio of the instability stress (predicted by J-T approach) for a 4" diameter, Schedule 80 pipe and that for another pipe of larger size (e.g., 24" diameter, Schedule 20). In the case of heat exchanger cooling water nozzles, the value of M' is 1.6. Therefore, for a given circumferential crack, the modified instability stress would be the maximum stress predicted by a limit load analysis divided by M' .

The procedure of using JiLLA concept to estimate the acceptable circumferential flaw length for cooling water inlet nozzle weldment under significant bending load is as following:

- (1) Apply safety factor (SF) to the applied longitudinal membrane stress (σ_m^{appl}) and the bending stress (σ_b^{appl}).

$$\sigma_m = SF \times \sigma_m^{appl}$$

$$\sigma_b = SF \times \sigma_b^{appl}$$

- (2) Scale the maximum stress by factor M' with the membrane stress unchanged.

$$\sigma_m + \sigma_b' = M' (\sigma_m + \sigma_b)$$

- (3) Calculate the modified bending stress (σ_b') in terms of the applied stresses.

$$\sigma'_b = SF \left[M' \left(\sigma_m^{\text{appl}} + \sigma_b^{\text{appl}} \right) - \sigma_m^{\text{appl}} \right]$$

- (4) Solve for half crack angle $\left(\theta = \frac{a}{R} \right)$ from the limit load equation [32] for a throughwall circumferential crack:

$$\sigma'_b = \frac{2 \sigma_f}{\pi} (2 \sin \beta - \sin \theta)$$

where

$$\beta = \frac{\pi}{2} \left(1 - \frac{\theta}{\pi} - \frac{\sigma_m}{\sigma_f} \right) \text{ defines the location of the neutral axis}$$

and σ_f is the flow stress which is $3S_m$ throughout this analysis.

3.4 Heat Exchanger Head

The heat exchanger head provides a pressure boundary of the PWS. The primary restraint of the head pressure boundary is provided by 84 staybolts attached to the tubesheet. The 72 C-clamps and a half-pipe seal across the flanges of the head and the tubesheet are leak prevention devices designed to replace the flange bolts and gaskets, but were included as strength members in the structural analysis of the head in Reference 2. They were not included in the development of acceptance criteria for the head; only the staybolts were considered to provide load-carrying capability.

The head can be divided into three general regions regarding to its shape [4]: the flat region including the head flange (Region I), the highly-curved region (Region II), and the 12-inch pipe region (Region III) which connects the process water piping system. The diameter of the head including the flange is 96.25" and its height is 26.125" [34]. The minimum thickness of the head along the contour is 1.25" [35]. In Section 3.4.2 and in Reference 3, the thickness of 1.5" was used in stress analyses. The measured thickness was reported to be 1.6" to 1.65" [36].

The instability flaw lengths for heat exchanger head were estimated earlier by linear elastic fracture mechanics in Reference 4. A more detailed analysis is carried out here. A finite element calculation (Section 3.4.2) was performed to obtain the maximum principal stresses for input to the fracture analysis (Section 3.4.3) in Regions I and II. A double curvature correction procedure is developed in Section 3.4.4 for the highly-curved region. In the pipe region, a J-integral based limit load approach (JiLLA, Section 3.3.2) is applied with the piping reaction forces described in the next section.

3.4.1 Loading Conditions and Safety Factors

The normal operating pressure for the heat exchanger inlet head is 218 psi. A pressure surge up to 267 ± 4 psi was observed for approximately 6 ± 3 seconds during the pump startup. These pressures are bounded by the design pressure of 300 psi [3]. A set of loads due to seismic, thermal, and gravity conditions can occur at the pipe flange [17]. The loads include a force of 10,500 lb in the heat exchanger axial direction, forces of 4500 lb and 6800 lb in the lateral directions, a torsion of 182,000 in-lb along the axial direction, and moments of 420,000 in-lb and 168,000 in-lb acting about two remaining Cartesian coordinate axes. In addition, the reaction force due to the process water design pressure (300 psi) in the 12-inch pipe is also added to the axial force, resulting in 44,429 lb in the axial direction.

The complete set of piping reaction forces is applied directly in the finite element analysis (Section 3.4.2). The staybolts connect the tubesheet to the head defining the boundary subject to the 300 psi design pressure. Maximum principal stresses are obtained in the heat exchanger head. A safety factor of 1.5 for faulted condition could have been applied to the maximum principal stresses in the flat (Region I) and highly-curved (Region II) regions. However, it was noted in Reference 3 that the seismic reaction forces applied to the pipe flange of the finite element model are small compared to the resultant force due to the process water design pressure alone. Therefore, the pipe flange loading has insignificant impact on the stress distribution of the head, in the areas away from the pipe region, when all 84 staybolts are intact. It implies that the finite element results, obtained with faulted stress boundary conditions, are also valid for the design condition. A safety factor of 3 is applied to the maximum principal stresses in the flat region and in the highly-curved region for the flaw stability analysis.

For the pipe region of the head (Region III), the J-integral based limit load analysis (JiLLA, Section 3.3.2) is adopted. A safety factor of 1.5 is applied to the axial force (44,429 lb) and the resultant bending moment (452,400 in-lb) in accordance with the JiLLA approach.

3.4.2 Finite Element Analysis

The finite element model contains 1224 shell elements including 216 thick shell elements representing the pipe flange and the head flange and 300 beam elements for the 84 staybolts. The heat exchanger head is loaded with 300 psi design pressure and the piping reaction forces described in Section 3.4.1. The model was designed according to the engineering drawings as discussed under Section 3.4. More detailed description of the finite element model can be found in Reference 3.

In the analysis, all 84 staybolts are intact and are fixed in the tubesheet ends. The C-clamps and the half-pipe seals are not designed as primary load-carrying components and, therefore, are not included. The maximum principal stresses were obtained from the finite element analysis and plotted in Figure 4. The highest value for the maximum principal stress in the curved region is about 6 ksi; the highest value in the flat region is about 11 ksi. These stresses are used in the fracture analysis with J-T method in the next section to determine the acceptable flaw lengths in the highly-curved region (Region II) and in the flat region (Region I) of the heat exchanger head.

3.4.3 Elastic-Plastic J-integral Analysis (J-T Method)

The concept and procedure for J-T method applied to the head of heat exchanger is similar to those described in Section 3.2.3. Since the shape of the head is complicated, the infinite plate result of Shih and Hutchinson is used as a base solution (Section 3.4.3.1). In the highly-curved region, a double curvature correction empirical procedure is developed to account for the effects of meridional and tangential radii of curvature (Section 3.4.3.2). In the flat region, the base solution is directly employed without curvature corrections.

The residual stress distribution is assumed to have the same pattern as discussed in Section 3.2.3. To be consistent with the finite element analysis, the thickness of the head (t) is taken to be 1.5". The maximum, saturated stress intensity factor due to the residual stress is calculated according to

$$K_{\max}^{\text{res}} = 0.43\sigma_r\sqrt{\pi t},$$

which was proposed in Reference 28.

3.4.3.1 Center-Cracked Panel (CCP) Solution - Infinite Plate

In the case of an infinite plate, $a/b = 0$ and the equations presented in Section 3.2.3.1 are greatly simplified. The elastic and plastic portions of J-integral in an infinite CCP are denoted by $J_{\text{el}}^{\infty\text{ccp}}$ and $J_{\text{pl}}^{\infty\text{ccp}}$. Other symbols used in Section 3.2.3.1 remain the same.

(i) The elastic portion of J-integral is calculated according to

$$J_{\text{el}}^{\infty\text{ccp}} = \frac{1}{E} (K_I^{\infty})^2$$

where

$$K_I^{\infty} = \sigma_{\infty} \sqrt{\pi a_{\text{eff}}}$$

and

$$\frac{a_{\text{eff}}}{a} = 1 + \frac{1}{2} \frac{n-1}{n+1} \frac{(\sigma_{\infty} / \sigma_0)^2}{1 + (\sigma_{\infty} / \sigma_0)^2} \quad \text{for } \sigma_{\infty} \leq \sigma_0$$

or

$$\frac{a_{\text{eff}}}{a} = 1 + \frac{1}{4} \frac{n-1}{n+1} \quad \text{for } \sigma_{\infty} > \sigma_0,$$

Note that the above two equations vary slightly from those in Reference 25 due to the factor Φ in the plastic zone correction, as noted in Section 3.2.3.1.

(ii) The plastic portion of J-integral is

$$J_{pl}^{ccp} = \alpha \sigma_o \epsilon_o a \left(\frac{\sigma_{\infty}}{\sigma_o} \right)^{n+1} g_1(0,n)$$

where

$$g_1(0,n) \equiv 3.85 \sqrt{n} (1 - 1/n) + \pi/n$$

(iii) The total J-integral for the CCP is, therefore,

$$J = J_{el}^{ccp} + J_{pl}^{ccp}$$

for the case without contributions from residual stress, thermal stress, etc. If the residual stress is present, as is in the case presented here, the treatment follows that described in Section 3.2.3. The expression for the residual stress intensity factor K_{max}^{res} can be found in Section 3.4.3.

3.4.3.2 Double Curvature Correction

The shape of the heat exchanger head at any location is defined by a pair of local radii of curvature: one is on the meridional plane containing the heat exchanger axis (R_m), the other is on the plane perpendicular to the heat exchanger axis (R_t , the radius from the heat exchanger centerline to the tangent line at that point on the head surface). In the highly-curved region of the head, the curvature effects on the fracture parameters are significant. A procedure for double curvature correction is developed for this region. A single pair of averaged radii of curvature is used for the correction. The averaged meridional radius of curvature (R_m) and the averaged tangential radius of curvature (R_t) for the highly-curved region are estimated to be 18.2" and 13.4", respectively.

The most critical crack orientation in the curved region is in the meridional direction, due to the hoop stress as a result of process water pressure. For a crack in a structure with double curvature, such as in a bend pipe shown in Figure 5, this configuration can be achieved by first transforming a CCP to an axial crack in a cylinder through an operation Y_1^2 , then by another transformation Y_2^2 into a circumferential crack. Therefore, the final configuration is considered as a combined transformation, $Y_2^2 \otimes Y_1^2$, from the original configuration of a center-cracked panel. In an engineering approach for estimating J-integral for a crack in a structure with double curvature, the approximated solution is given as

$$J_{el/pl}^{cur} = Y_2^2 Y_1^2 J_{el/pl}^{ccp}$$

where Y_1 and Y_2 can be found in Section 3.2.3.2 and $J_{el/pl}$ represents the elastic or plastic portion of J-integrals.

4.0 HEAT EXCHANGER SHELL AND HEAD ACCEPTANCE CRITERIA

4.1 Overview

Postulated throughwall flaw configurations were analyzed in Section 3 for stability for the heat exchanger head, shell, and cooling water nozzles, the external pressure boundary components of the heat exchanger. The resultant flaw lengths maintain the ASME-based margins of safety in Section XI [9], Article IWB-3642 for austenitic stainless steel piping. Flaw depth up to 75% of the piping wall thickness is provided by Section XI, Article IWB-3641 [9]. The fracture analysis was performed for postulated throughwall flaws and is, therefore, more limiting than part-throughwall flaws at the same length; nevertheless, a depth of 75% of the component thickness for flaws open to the inside surface with lengths up to the acceptable flaw length is adopted as the depth criterion for the external pressure boundary of the heat exchanger to avoid the potential for inadvertent leakage during operation. Note that the 75% depth requirement was also adopted in ASME BPV Code, Code Case: Nuclear Components, N-494.

The following sections provide the acceptable flaw lengths for the head, shell, and cooling water nozzles by identifying the shortest instability lengths or most limiting lengths from the results obtained in Section 3. The acceptance criteria for flaws in the external pressure boundary of the heat exchanger produced by service-induced degradation are the acceptable flaw length and the acceptable depth of 75% of the thickness of the respective components. Section 5 provides an acceptance criteria methodology to ensure that the structural margins for the external pressure boundary components of the heat exchanger are maintained through in-service inspection.

4.2 Heat Exchanger Head

The seismic load applied to the pipe flange in the finite element model has minimal influence on the overall stress state of the head. Therefore, the design condition (300 psi) with a safety factor 3 will lead to the acceptable lengths for Region I (flat region including the head flange) and Region II (highly-curved region). The acceptable flaw length in Region III (pipe region) is obtained by the J-integral based limit load analysis with seismic piping reaction forces reported in Reference 17 and a safety factor of 1.5. The results are summarized in Table 3 and the acceptable flaw lengths are in bold characters.

TABLE 3
Acceptable Flaw Lengths in Heat Exchanger Head

Region	Maximum Principal Stress (ksi)	Safety Factor	Acceptable Flaw Length (2a) Inches
I. Flat (incl. Flange)	11	3	26
II. Highly-curved	6	3	16 (meridional)
III. Pipe	(not applicable)	1.5	23 (circumferential)

Note: The dominant flaw directions are identified next to the flaw length. In the flat region the analysis does not distinguish preferential directions. The flaw lengths for Regions I and II were determined with J-T method (Section 3.4.3). The flaw length for Regions III was determined with the J-integral based limit load analysis (JiLLA, Section 3.3.2).

4.3 Heat Exchanger Shell

A summary of the instability flaw length results in the analysis of postulated throughwall axial and circumferential flaws is given in Tables 4 and 5, respectively. The acceptable flaw lengths (in bold characters) are 6 inches for an axial flaw (Table 4) and 30 inches for a circumferential flaw (Table 5).

TABLE 4
Axial (Longitudinal) Cracks in Heat Exchanger Shell

Case	Limit Load Approach		J-T Approach	
	a (Half Length) inches	2a inches	a (Half Length) inches	2a inches
P=50 psi Normal Pressure Safety Factor = 3	13.5	27	14.5	29
P=150 psi Design Pressure Safety Factor = 3	2.9	6	3.8	7.6
P=168 psi Water Hammer ⁽¹⁾ Safety Factor = 1.5	7.5	15	9	18
P=242 psi Water Hammer ⁽²⁾ Safety Factor = 1.5	4.5	9	5.5	11
P=300 psi Tube Rupture Safety Factor = 1.5	2.9	6	3.8	7.6
P=50 psi Seismic Safety Factor = 1.5	Not Applicable to Axial Cracks			

Note. (1) corresponds to partial check valve closure (maximum 182 psia) [19].
(2) corresponds to full check valve closure (maximum 256 psia) [19].

TABLE 5
Circumferential Cracks in Heat Exchanger Shell

Case	Limit Load Approach			J-T Approach		
	a (Half Length) inches	2a inches	θ (Half Angle) Degrees	a (Half Length) inches	2a inches	θ (Half Angle) Degrees
P=50 psi Normal Pressure Safety Factor = 3	61.5	123	80	30	60	39
P=150 psi Design Pressure Safety Factor = 3	36.9	74	48	15	30	20
P=168 psi Water Hammer ⁽¹⁾ Safety Factor = 1.5	56.9	114	74	26	52	34
P=242 psi Water Hammer ⁽²⁾ Safety Factor = 1.5	47.4	95	62	22	44	29
P=300 psi Tube Rupture Safety Factor = 1.5	40.8	82	53	17	34	22
P=50 psi Seismic Safety Factor = 1.5	73.7	147	96	45	90	59
Note: (1) corresponds to partial check valve closure (maximum 182 psia) [19]. (2) corresponds to full check valve closure (maximum 256 psia) [19].						

4.4 Heat Exchanger Cooling Water Nozzle

The nozzle loads [17] and normal operating pressure 50 psi were used to determine the maximum tensile and bending stresses. The *J*-integral based limit load analysis for the cooling water pipe gives an acceptable flaw length of 39" at the circumferential weld joint. [If the design condition (150 psi with no bending moments) is used along with a safety factor of 3, the acceptable flaw length is 39.5".]

5.0 ACCEPTANCE CRITERIA METHODOLOGY FOR THE HEAT EXCHANGER EXTERNAL PRESSURE BOUNDARY COMPONENTS

5.1 Basis for In-Service Inspection

Leakage failures have occurred in the service history of the heat exchangers (see Section 2 and Reference 8). No service-induced degradation is predicted in the head, shell, and cooling water nozzles for the post-cooling tower tie-in configuration (see Section 2). Examinations (with visual, surface, or volumetric methods), conducted under an in-service inspection program, provide information on the service condition of the component and may also provide the information showing that service-induced degradation has been mitigated or eliminated. An acceptance criteria methodology is a framework for periodically monitoring the service condition through in-service examinations and includes the disposition of degraded conditions while maintaining safety margins against failure.

An acceptance criteria methodology is proposed for the external pressure boundary components of the heat exchangers^{††} to meet the safety margins against failure stated in Sections 1.2, 3.1, and 4.1 and involves the following sequence in an in-service inspection program: 1) Baseline Examination; 2) Evaluation; 3) Acceptance; and 4) Successive Examinations.

Additional details on the acceptance criteria methodology are discussed in Section 5.2. The disposition of flaws is discussed in Section 5.3. The development of programmatic requirements as part of an in-service inspection plan for the in-service examinations of the heat exchangers is outside the scope of this report.

5.2 Acceptance Criteria Methodology

The acceptance criteria methodology for the external pressure boundary components of the heat exchangers is shown in a flowchart format in Figure 6. The analyses in this report provided the technical bases for the acceptance criteria for the heat exchanger components (Section 4). The acceptance criteria methodology includes monitoring of the heat exchangers in periodic examinations to demonstrate that the heat exchanger components meet the stated safety margins against failure.

Baseline examinations of the heat exchanger components would be performed in accordance with the code case for In-Service Inspection for Low Temperature Heavy Water Reactors, presently in draft [37]. Evaluation and acceptance of reported flaws would include comparison of the flaw size with the allowable sizes in the acceptance standards for the heat exchanger components provided in the code case. If the flaw size exceeds the allowable sizes in the acceptance standards, then acceptance-by-analysis and/or additional examinations would be required for acceptance of the condition (see Section 5.3).

Periodic monitoring through successive examinations would be continued with an increase in the frequency of examinations, if degradation is significant (see Section 5.3).

^{††} An acceptance criteria methodology has been proposed previously for in-service inspection of the PWS heat exchanger staybolts [3].

5.3 Disposition of Flaws

Acceptance standards for circumferential or axial flaws reported in the in-service inspection of the heat exchanger external pressure boundary components are provided in the code case for Low Temperature Heavy Water Reactors, presently in draft [37]. If a relevant indication does not exceed the allowable flaw size specified in the acceptance standards, no further evaluation would be necessary and examination of the component would be performed at the next interval in the inspection program. If a flaw exceeds the size in the acceptance standards, acceptance-by-analysis and/or additional examinations would be required per the code case for acceptance of the flaw. Evaluation of the service-induced degradation and the flaw growth rate would be performed, and, if a flaw is predicted to exceed the acceptance criteria (Section 4) prior to the next interval in the inspection program, then the flaw would be re-inspected before the next interval such that the acceptance criteria are met. The flaw growth rate evaluation would be updated following each successive examination or reexamination.

The acceptance criteria provided in Section 4 are the acceptable flaw configurations (length and depth) to maintain the defined safety margins for the heat exchanger head, shell, and cooling water nozzles serving as the limit to flaw configurations acceptable for service. No flaws in the external pressure boundary components of the heat exchanger shall exceed or be predicted to exceed these configurations. Flaws in the external pressure boundary of the heat exchanger that exceed or are predicted to grow to exceed the acceptable configurations while in service would require repair or replacement of the heat exchanger.

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Figure 1. Material J-T Curve for 304L Stainless Steel

Figure 2. Longitudinal Crack and Corresponding Configuration of a Center-Cracked Panel

Figure 3. Circumferential Crack and Corresponding Configuration of a Center-Cracked Panel

Figure 4. Maximum Principal Stress Distribution on the Heat Exchanger Head at Design Pressure 300 psi under Seismic Piping Reaction Forces

Figure 5. Transformation of a Center-Cracked Panel to a Crack with Double Curvature

Figure 6. Flowchart for Disposition of Flaws

APPENDIX J-INTEGRAL FOR CIRCUMFERENTIAL CRACKS (Finite Element Solution versus Curvature Correction)

In Section 3.2.3, a curvature correction method is used to construct an approximate elastic-plastic J-integral solution for Geometry 1 (e.g. a heat exchanger shell) based on a known solution for Geometry 2 (e.g. a center-cracked panel). The elastic stress intensity factors for both geometries must be known a priori and their ratio is denoted by Y (Section 3.2.3.2). The estimated J-integral for Geometry 1 is, therefore, the J-integral for Geometry 2 multiplied by Y^2 . In other words, the proportionality factor (Y^2) obtained from the elastic portion of J has been assumed to exist and to be applicable to the plastic portion. In an elastically dominant case, this treatment should provide an accurate approximation for the J-integral.

The degree of accuracy of curvature correction for the case of circumferential cracks under tension has been demonstrated by an elastic-plastic finite element analysis. Five finite element meshes^{†††} were obtained for the size of the heat exchanger shell, that is, the radius is 44" and the thickness is 0.5". The half crack angles in these models include $\pi/16$, $\pi/8$, $\pi/4$, 0.4π and $\pi/2$. Because of symmetry, only a quarter of a cylinder containing one half of the circumferential crack is modeled. Figures A-1 and A-2 show two typical finite element meshes. The lengths of all models are 623.33". With this distance between the crack and the applied loads, a uniform far field stress can be achieved. These models were first shown to be capable of reproducing the (elastic) stress intensity factors due to either remote tension or bending [31]. Multiple (nine) contours around the crack tip are used to ensure the path-independency of the J-integrals. The J-integral values were obtained as the average of results of all integration contours excluding the crack tip where inaccuracy is usually expected. The sizes of the finite element models are summarized below.

Half Crack Angle	Number of Elements (ABAQUS S9R5)	Number of Nodes
$\pi/16$	384	1629
$\pi/8$	384	1629
$\pi/4$	384	1629
0.4π	455	1921
$\pi/2$	440	1859

Figure A-3 shows the finite element calculated J-integral as a function of far field stress for various crack sizes. By changing parameters, the curves can be replotted as the J-integral versus the half crack angle for a series of far field stresses. The results of this procedure (isolated data points) are plotted in Figure A-4 to compare with the curvature correction results (solid curves). Note that the curvature correction curves contain discontinuities, because the stress intensity factor solutions used to derive the curvature correction factors change at those crack lengths (at $a = 5\sqrt{Rt} = 23.5"$, see Section 3.2.3.2).

^{†††} Courtesy of the South Carolina Universities Research and Education Foundation (SCUREF) Task No.68, "Advanced Fracture Mechanics to Assess Complicated Piping Flaws," by Mr. W. Gi, Dr. Y.-J. Chao, and Dr. M. A. Sutton, Department of Mechanical Engineering, University of South Carolina, Columbia, South Carolina, 1992.

Both the finite element analysis and the curvature correction procedure were performed for a cylinder (or pipe) with the same size as the heat exchanger shell ($R/t=88$), and for a Ramberg-Osgood material with $\alpha=2$ and $n=5$. Three representative internal pressures or associated longitudinal (axial) stresses were used to generate Figure A-4. In the heat exchanger case, the design pressure for the cooling water inlet or the shell is 150 psi, and the design pressure for the process water inlet to the head or for the tubes is 300 psi. Figure A-4 shows that the results with the curvature correction applied to the flat plate solution nearly matches the finite element results, especially under the stress level of interest and this class of circumferential cracks in this Ramberg-Osgood material ($\alpha = 2$ and $n = 5$).

Overall Finite Element Mesh

Near Crack Tip Mesh

Cracked Area

Figure A-1. Finite Element Mesh for a Circumferential Crack ($\theta = \pi/4$) in a Cylinder with $R = 44$ "

Overall Finite Element Mesh

Near Crack Tip Mesh

Cracked Area

Figure A-2. Finite Element Mesh for a Circumferential Crack ($\theta = \pi/2$) in a Cylinder with $R = 44$ "

Figure A-3. J-integral for Circumferential Cracks in a Cylinder ($R=44''$) under Tension
(Finite Element Solution)

Figure A-4. Comparison of J-integral Values from Direct Finite Element Calculations and from Curvature Correction based on a Flat Plate Solution

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