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SUMMARY

Twelve process water heat exchangers are designed to remove heat generated in the reactor tank. Each heat exchanger has approximately 9000, 1/2" diameter x 0.049" thick tubes. Minimum structural tubing requirements and the leak rate through postulated tubing defects are developed in this report. A comparison of the structural requirements and the defect size calculated to produce leak rates of 0.5 lbs./day demonstrate adequate structural margins against gross tube rupture.

Commercial nuclear experience with pressurized water reactor (PWR) steam generator plugging criteria are used for guidance in performing this analysis. It is important to note that the SRS reactors are low energy systems with normal operating pressures of 203 psig at 130°F while the PWR is a high energy system with operating pressures near 2200 psig at 600°F. Clearly the PWR steam generator has loadings which are more severe than the SRS heat exchangers.

The commercial basis for plugging PWR steam generator tubing is defined by Regulatory Guide 1.121 which is explicitly limited to PWRs. However, where applicable, this guideline is applied in the current evaluation of the SRS heat exchanger tubing.

Consistent with the Regulatory Guide 1.121 criteria both wastage (wall thinning) and cracking are addressed. Structural limits on wall thinning and crack size are developed to preclude gross rupture. Gross wall thinning criteria is based on ASME Section VIII, which is the design code for the heat exchanger. Local wall thinning evaluation is based on ASME Code Case N-480. ASME Section XI criteria, with the factors of safety recommended by Regulatory Guide 1.121 are used to develop the allowable crack size criteria. Normal operating conditions (pressure, dead weight, and hydraulic drag) are considered with seismic and water hammer accident conditions. Both the wall thinning and crack size criteria are developed for the end-of-evaluation period. Allowances for corrosion, wear, or crack growth have not been included in this analysis.

Structurally, the tubing is over designed and can tolerate large defects with adequate margins against gross rupture. The structural margins of heat exchanger tubing are evident by contrasting the tubing's structural capacity, per the ASME Code, with it's operating conditions/configuration:

- 3600 psig capacity versus 200 psig actual normal operating pressure;
- 0.0037" minimum wall thickness versus 0.049" actual wall thickness;
- localized wall thinning below 0.0037" is allowable;
- large crack lengths (greater than one-half of the circumference) are allowable.

1. INTRODUCTION AND CONCLUSIONS

Twelve process water heat exchangers are designed to remove heat generated in the reactor tank. Each heat exchanger has approximately 9000, $\frac{1}{2}$ " diameter $\times$ 0.049" thick tubes. Minimum structural tubing requirements and the leak rate through postulated tubing defects are developed in this report. A comparison of the structural requirements and the defect size calculated to produce leak rates of 0.5 lbs./day demonstrate adequate structural margins against gross tube rupture.

1.1. Structural Design Capacity

Structurally, the heat exchanger tubing is over designed. The allowable normal operating pressure for the tubing, using ASME Section VIII, Division 2 and minimum material properties, is 3600 psig. Normal operating pressure is approximately 200 psig and the design pressure is 300 psig. For a water hammer transient, the allowable pressure is 4300 psig. The maximum analyzed water hammer transient has a peak pressure near the operating pressure (200 psig).

1.2. Structural Requirements for Minimum Thickness

The minimum wall thickness for the design pressure, using ASME Section VIII, Division 2 and minimum material properties, is 0.0037". Water hammer transients on the tube side require an additional 0.001" of wall thickness for each 100 psig of pressure over 300 psig. The maximum analyzed water hammer transient has a peak pressure near the operating pressure (200 psig). Water hammer will not burst the heat exchanger tubing.

ASME Code Case N-480 is used to determine the allowable wall thickness for a pit or wear scar. The minimum wall thickness for a wear scar or pit is 0.002", with a $\frac{1}{32}$ " width less than 0.0037" thick. Wider wear scars require thicker material.

1.3. Structural Requirements for Maximum Flaw Size

The ASME Section XI limit load criteria in Appendix C-3320 and C-3340 are used with Regulatory Guide 1.121 factors of safety (3 for normal operation and 1.5 for accident conditions) to determine the allowable sizes of postulated circumferential and axial flaws. Allowable flaw sizes are calculated for 1) the nominal wall thickness, 0.049 inches; and 2) a region previously thinned by wastage, with a postulated wall thickness of 0.025 inches. The non-mechanistic combination of wastage and cracking is used to determine the sensitivity of the allowable flaw length to variations in wall thickness. Both throughwall and part throughwall flaws are analyzed.

The allowable size of circumferential throughwall flaws is greater than 50% of the tubing circumference (≈ 0.8 "), for tubing 0.025" and thicker. The allowable depth for 360° circumferential part throughwall flaws is 90% of the remaining tubing thickness, for tubing 0.025" and thicker. The allowable size of circumferential flaws is relatively insensitive to variations in the wall thicknesses between 0.049" and 0.025".

The allowable length of axial flaws is sensitive to variations in the wall thickness. The allowable length of a throughwall axial flaw is 2" for 0.049" thick tubing and 0.74" for 0.025" thick tubing. Interpolation can be used to determine the allowable lengths for intermediate tubing thicknesses. The allowable depth of 2" long part throughwall axial flaws is 85% for tubing between 0.049" and 0.025".

1.4. Reference Leakage Size Flaw and Structural Margins

For the 0.049" thick tubing, the crack lengths required to obtain 0.5 lbs./day leakage are between 0.4" and 0.6" for both circumferential and axial flaws. The structural margins on stress are greater than 3 while the margin on crack length is 1.9. These margins are adequate and indicate that the tubing will leak without breaking. This calculation is supported by the past 37 years of reactor operating history, in which 63 leaks were detected in the heat exchanger tubing without a double ended guillotine break.

1.5. Inspection Acceptance Criteria

The structural and operation requirements are synthesized into the ASME Section XI compatible end-of-evaluation period inspection acceptance criteria in Table 11. This criteria covers general wall thinning, local wall thinning, and cracking. Allowances for further wall thinning and/or crack growth are not included in this criteria. This acceptance criteria is intended to 1) prevent gross rupture, and 2) prevent leakage, based on the configuration at the end of an evaluation period.

2. COMPONENT DESCRIPTION

The process water heat exchangers are designed to remove heat generated in the reactor tank. The once-through, counter flow type heat exchangers are supported horizontally on the -20' level, Figure 1. Pressures and temperatures are given in Table 1 for both normal operation and design conditions.

Each heat exchanger consists of a 7'-3 1/2" diameter cylindrical shell attached at each end to either a double or single tube sheet. The overall length of the shell section is 29' 2-1/2". Heads at each end of the shell are bolted to the process water piping and are attached to the tube sheet by 84 staybolts. Cooling water flows into and out of the shell through nozzles near each end of the shell. Inside the shell, the cooling water flows around 6 flow reversing baffles and 6 or 12 intermediate baffles. A diffuser plate and strainer are located at the cooling water inlet.

Inside the shell section are approximately 9000, 1/2" diameter $\times$ 0.049" thick (18 gauge) tubes which span axially between baffles [18]. Each of the approximately 9000 tubes are attached to the tube sheet by expansion (hydraulic or rolled) and seal welding. The bottom and top rows of tubes are plugged. The nominal spacing between adjacent baffle plates varies from 16.6 to 24.9 inches depending on the type of heat exchanger. Inside most of the tubes are 0.296" diameter concentrically supported core rods.

The original design of the shells and heads, Type "A", was by Foster Wheeler and utilized double tube sheets at each end of the shell, six intermediate baffles, and a solid diffuser plate without a CWS inlet strainer. Type 304 stainless steel was used for the tubes, shell, head, and tube sheets in the Type A design. Both seamless and welded Type 304 stainless steel tubes may have been used. The Type A heat exchangers were constructed to the 1949 ASME Section VIII design code.

Several modifications were made to the original design (Type "A") heat exchangers, prior to 1966, to resolve operational problems. The heat exchangers were completely overhauled and rotated, tubes were plugged in critical positions, a perforated cooling water inlet diffuser plate and strainer were installed, and an inhibitor system was installed between the double tube sheets. All gaskets and O-rings exposed to moderator (head-to-tube sheet and head to staybolt) were eliminated by using a welded boss closure around the staybolt and a

Table 1. Heat Exchanger Operating and Design Conditions

		PW Inlet	PW Outlet	CW Inlet	CW Outlet
Temperature (° F)	Operating	131*	79*	55*	109*
	Design	250	250	200	200
Pressure (psig)	Operating	203**	163	50	20
	Design	300	300	150	150
Heat Exchanger End		Hot	Cold	Cold	Hot

* Actual operating data obtained from the Reactor Control Computer RMS (Reactor Monitoring System) 1308 MW, RMS, 3/11/88, K13.2.

**Maximum occurring during start-up of the pump's AC motor before opening the rotovalves is 267 psig for approximately 6 seconds $\pm$ 3 sec. [3]

welded omega seal to join the head to the tube sheet. The head vent and drain lines were relocated. Head flange bolt holes were filled with plugs and seal welded.

Replacement heat exchangers purchased in the early 1980's from Mitsui and Hitachi are designated as Type B. These heat exchangers are based on the original Type A design, incorporating the changes made to the Type A design during the 1960 heat exchanger overhaul program. The double tube sheet on the Type A design was consolidated into a single tube sheet on the Type B design. Twelve intermediate baffles are used on the Type B design versus six on the Type A design. Additionally, the tubing and tube sheets in the Type B design are made of Type 316L stainless steel, while the shell, nozzles, distributor plates, and baffles are made of Type 304L stainless steel. Only seamless Type 316L stainless steel tubes are used in the Type B design.

Replacement heat exchangers purchased in the mid 1980's from Nooter use Hitachi heads and are designated as Type C. Geometrically, the Type B and C heat exchangers are equivalent. The Type C heat exchangers have seamless Sea-Cure¹ tubing, while the Type A and B designs used Type 304 and 316L stainless steels, respectively. The Types B and C replacement heat exchangers were built in accordance with the 1980 ASME Section VIII and 1968 TEMA design codes.

Table 2 lists the various materials of construction for the different types of heat exchangers. Table 3 provides the heat exchanger purchase information including date received, serial numbers, vendor and BPF number. The materials of construction for the components of the individual heat exchangers installed in SRS K reactor, as well as the corresponding serial numbers, are listed in Table 4.

¹ Sea-Cure is a ferritic stainless steel manufactured by the Trent Tube Division of Crucible Steel. Trent Sea-Cure stainless steel is approved for ASME Boiler and Pressure Vessel Code and is included in Section VIII Division 1, Table UHA-23 and in Code Cases 1900 and 1922. Sea-Cure tubing is a ferritic stainless steel produced by either argon-oxygen decarburization (AOD) refining or vacuum-oxygen decarburization (VOD), to achieve a low carbon and nitrogen interstitial content. The steel is also stabilized with titanium and/or niobium to provide optimum weld and annealing sensitization resistance. Alloying additions of chromium, molybdenum, and nickel provide an optimum combination of corrosion resistance and improved mechanical properties. Nickel additions reduce the ductile-brittle transition temperature to below -20C (-10F). Sea-Cure stainless is highly resistant to localized pitting and crevice corrosion and highly resistant to stress corrosion.

Table 2. Heat Exchanger Materials of Construction

Description	Heat Exchanger Materials of Construction		
	Type A	Type B	Type C
Head and Head Flange	Type 304	Type 304 or 304-L*	Type 304-L
Shell	Type 304	Type 304-L	Type 304-L
Tube Sheet(s)	Type 304	Type 316-L	Type 316-L
Tube	Type 304	Type 316-L	Sea Cure
Staybolt	Type 303	Type 303	Type 303
Omega Seal	Type 304-L	Type 304-L	Type 304-L
Bolt Hole Plugs	Type 304-L	Type 304-L	Type 304-L
12" Flange	Type 304-L	Type 304-L	Type 304-L
Core Rod	Type 304-L	Type 304-L	Type 304-L
Boss Closure Assembly	Type 304-L	Type 304-L	Type 304-L

* Eighteen Type B heads were salvaged from Type A heat exchangers, and are Type 304 stainless steel. Heads fabricated for the Type B heat exchangers are Type 304-L

Table 3. Heat Exchanger Purchase Information

Date Rec'd	Type	Serial No's	Vendor	BPF Number
1953-54	A	OH-7249 to OH-7260	Foster Wheeler	120020
1954-56	A	OH-7811 to OH-7848	Foster Wheeler	120221
1954-56	A	44344 to 44355	American Locomotive	120236
1962	A	58-9478 to 58-9479	Foster Wheeler	211069
1963	A	58-9509 to 58-9512	Foster Wheeler	211259
1983	B	1315-120-1 to 1315-120-12	Mitsui	213314
1983	B	105-999-1 to 105-999-2	Hitachi	213501
1983	B	105-150-1 to 105-150-4	Mitsui	213225
1987	C	105-999-3 to 105-999-10	Nooter-Shells Hitachi-Heads	216254 216386

**Table 4. Materials of Construction for the K Reactor
Heat Exchangers In Service -- 2/1/92**

Location	I.D.	Type	Head	Shell	Tubes	Tube Sheet	
1 A	OH 7831	A	304	304	304	304	Double
1 B	105-999-9	C	304-L	304-L	Sea Cure	316-L	Single
2 A	OH 7839	A	304	304	304	304	Double
2 B	OH 7825	A	304	304	304	304	Double
3 A	OH 7847	A	304	304	304	304	Double
3 B *	105-999-7	C	304-L	304-L	Sea Cure	316-L	Single
4 A *	105-999-8	C	304-L	304-L	Sea Cure	316-L	Single
4 B *	105-999-5	C	304-L	304-L	Sea Cure	316-L	Single
5 A	44352	A	304	304	304	304	Double
5 B	44344	A	304	304	304	304	Double
6 A	105-999-3	C	304-L	304-L	Sea Cure	316-L	Single
6 B	1315-120-11	B	304	304-L	316-L	316-L	Single

* Heat Exchangers installed in January 1992

Figure 1 Process Water System Heat Exchanger

3. OPERATING HISTORY

The operating history of the Savannah River Site Process Water System (PWS) heat exchanger tubes was reviewed to determine critical or suspect areas of the tube bundle where degradation is most likely to occur [1]. During the past 37 years of reactor operation, 63 heat exchangers were replaced because of tube leaks. Thirty seven of the tube leaks were due to mechanical degradation, twenty three were due to corrosion, and the leak mechanism is unknown in three instances¹. No rupture equivalent to a complete circumferential severance has occurred in the operating history of the SRS reactor PWS heat exchangers. A hypothetical complete severance of the tube (neglecting the core rods) is an accident scenario considered in the SRS K Reactor Safety Analysis Report.

3.1. Mechanical Degradation

Twenty four heat exchangers leaked because of tube wear against baffle plates prior to 1965. This wear occurred primarily in the bottom row tubes, and a few at corner tubes adjacent to the shell, where the inlet cooling water velocity was high. Nine heat exchangers leaked due to wear from a foreign body (plastic bristles, wire etc.). Two instances of vibration fatigue, one instance of core rod vibration, and one instance of tie rod spacer wear have also been observed. Except for the core rod vibration, all of the mechanical degradation leaks have occurred before 1965.

The heat exchangers were given a complete overhaul between June 1964 and November 1966. During the overhaul, a perforated cooling water inlet diffuser plate and strainer were added to the heat exchanger, which reduced the inlet cooling water velocity and prevented large items from impacting the tubes. Additionally, the bottom row of tubes and select tubes on the edge of the tube bundle were plugged. Reducing the cooling water velocity eliminated the vibration problems. The potential for wear from foreign bodies was minimized by reducing the screen size in the cooling water (186) basin to 0.25", installing the CWS inlet strainer and procedural changes.

The only tube leak due to mechanical degradation since 1966 occurred in 1972 when a core rod moved out of the tube and entered a septifoil line where flow induced vibration caused the rod to wear through the tube. In 1990, all core rods aligned with the septifoil piping underwent a pull test to ensure that there was adequate resistance to prevent movement of the core rod into the septifoil line.

3.2. Corrosion

Several instances of chloride stress corrosion were noted between the double tube sheets in the leak collection spaces prior to 1965. The heat exchangers were reversed to extend their life and an inhibitor system was installed to circulate a phosphate solution through the leak collection space to control the corrosion. The Type B and C heat exchangers have a single tube sheet and do not require the corrosion inhibitor system.

¹ Tube leaks with an unknown cause occurred in the L Reactor on 6/57 and in the K reactor on 5/63 and 12/91. The investigation of the 12/91 tube leak is in progress (2/14/92).

Beginning in the 1970's, several instances of stress corrosion cracking at tube to tube sheet junction were noted along with pitting corrosion. The tubing materials in the Types B and C heat exchangers were changed from Type 304 stainless steel to Type 316L stainless steel and Sea Cure to prevent corrosion. All of the tube leaks due to corrosion have occurred in Type A heat exchangers.

4. STRUCTURAL REQUIREMENTS

Limits on wall thinning and crack size are developed in this section to preclude gross rupture. Gross wall thinning criteria is based on ASME Section VIII Division 2 while local wall thinning is based on ASME Code Case N-480. ASME Section XI criteria, with the factors of safety recommended by Regulatory Guide 1.121 are used to develop the crack size criteria. Both the wall thinning and crack size criteria are developed for the end-of-evaluation period. Allowances for corrosion or crack growth have not been included in this analysis.

4.1. Applied Loadings

Self Weight

The weight of the 0.5" outer diameter by 0.049" thick tubing (Metal Area=0.0694 in²), 0.296" diameter core rod (Metal Area=0.0688 in²) and the weight of the D₂O (69.33 pcf), neglecting the buoyancy of river water on the shell side is

$$w = \frac{490 \text{ pcf}}{12^3 \text{ cu in/cf}} (0.0694+0.0688)\text{in}^2 + \frac{69.333 \text{ pcf}}{12^3 \text{ cu in/cf}} (0.1269-0.0688)\text{in}^2 = 0.0415\text{lbs/in}$$

The tubes span horizontally between baffles, with a maximum center to center spacing of 24.9". The maximum bending moment due to self weight is

$$M_{dl} = \frac{0.0415\text{lbs/in} (24.9 \text{ in})^2}{12} = 2.14 \text{ in lb}$$

Internal Pressure

The pressure resisted by the tubing is the difference between internal and external pressure. During normal operation, the internal pressure at the heat exchanger inlet is 203 psig and the corresponding external pressure is 20 psig. During pump start up, the internal pressure may be as high as 267 psig for a short period of time. This analysis conservatively neglects the shell side pressure and uses the design pressure of 300 psig as the normal operating pressure. This assumption bounds the hydraulic loadings specified in Regulatory Guide 1.121.

At the tube sheet and baffles the radial expansion of the tube due to internal pressure is restrained by the diameter of the hole¹ as shown in Figure 2. The unrestrained radial expansion, due to internal pressure, P, is

$$\Delta R = \frac{P r^2}{E t}$$

Solving for the shear, Q, and moment, M, in the wall of the tube due to the radial constraint imposed by the tube sheet, using ASME Section VIII, Division 2, Appendix 4, Article 4.231 for long tubes, yields

¹ The tube sheet hole is between 0.01" to 0.022" larger than the nominal tube diameter. Recognizing the tolerances in the tube diameter, tube roundness, tube straightness, hole diameter, and hole location; the gap between the tube and tube sheet is conservatively neglected.

$$w(x) = \frac{Q_0}{2\beta^3 D} f_1(\beta x) + \frac{M_0}{2\beta^2 D} f_2(\beta x)$$

$$\frac{\theta(x)}{\beta} = \frac{-Q_0}{2\beta^3 D} f_3(\beta x) - \frac{2M_0}{2\beta^2 D} f_1(\beta x)$$

Where $\beta = \sqrt[4]{\frac{3(1-\nu^2)}{t^2(R+t/2)^2}}$, $D = \frac{Et^3}{12(1-\nu^2)}$, E is the elastic modulus, 27,400,000 psi, ν is

Poissons ratio, 0.3, R is the mean radius, t is the thickness, M_0 and Q_0 are the bending moment and shear in the pipe wall at the baffle plate or tube sheet. At the tube sheet and baffles, $x=0$, $w(0)=-\Delta R$, $\theta(0)=0$ and $f_1(\beta x)=f_2(\beta x)=f_3(\beta x)=1$. Solving for the maximum moment M_0 yields

$$M_0 = \frac{P R^2}{E t} 2\beta^2 D$$

And the maximum bending stress in the wall is

$$\sigma = \pm \frac{6}{t^2} \frac{P R^2}{E t} 2\beta^2 D$$

Note that the bending stress is classified as a secondary stresses because it is deformation controlled. The shear stress is small compared to the bending stress and is neglected.

Hydraulic Drag

On the secondary side of the heat exchanger, 14,000 gpm (31.2 cfs) of cooling water flows through the tube bundle, creating a lateral loading (drag) on each of approximately 9000 tubes (number of tubes varies for Types A, B and C). The 0.5" OD tubes have a triangular pitch of 0.75" and a spacing between tubes of 0.25". The distance between flow reversing baffles is 49.8" and the width of the top and bottom row of tubes is 62.6". The cooling water velocity between tubes is

$$V = \frac{31.2 \text{ cfs} \times 144 \text{ in}^2/\text{ft}^2}{62.6" (.25/.75) 49.8"} = 4.3 \text{ fps} = 51.9 \text{ ips}$$

The hydraulic diameter of the space between tubes is [19]

$$D_e = \frac{2 X Y}{X + Y} = \frac{2 \times 0.25 \times 24.9}{0.25 + 24.9} = 0.50"$$

And the Reynolds number for flow between tubes is

$$N_{Re} = \frac{D_e V}{\text{kinematic viscosity}} = \frac{0.5" \ 4.3 \text{ fps}}{12"/\text{ft} \ 0.739 \times 10^{-5} \text{ ft}^2/\text{sec}} = 24000$$

The drag on a single tube in an infinite medium is expressed by

$$w = \frac{C_D \text{ Area Density } V^2}{2 g} = \frac{C_D (0.5/12)\text{ft } 62.4 \text{ pcf } (4.3 \text{ fps})^2}{2 \times 32.2 \text{ ft/sec}^2} = 0.746 C_D \text{ lbs/ft}$$

A single tube in an infinite medium with a Reynolds number of 24000 has laminar flow with a coefficient of drag near 1. In the heat exchanger, the cooling water flows around an deflector plate, through a strainer before it flows through the tube bank. The first row of tubes in the tube bank are either solid bars or plugged tubes. For tubes inside the heat exchanger tube bank, the flow is trip turbulent and a drag coefficient corresponding to fully turbulent flow is appropriate. For turbulent flow of a long isolated tube the drag coefficient is $C_D=0.33$ [20] and the loading on the pipe is $w=0.021$ lbs/inch. The maximum moment due to hydraulic drag is

$$M_{\text{drag}} = \frac{0.021 \text{ lbs/in } (24.9 \text{ in})^2}{12} = 1.09 \text{ in lb}$$

Since the velocity of the cooling water in the heat exchanger is constant, the total drag force is related to a pressure drop across the heat exchanger. Summing the drag forces on 8800 tubes

$$F_{\text{drag}} = 0.021 \text{ lbs/in} \times 12 \text{ in/ft} \times 29' \times 8800 \text{ tubes} = 64,000 \text{ lbs}$$

The theoretical pressure drop across the heat exchanger resulting from the tube drag forces is

$$\Delta p = \frac{64000}{\pi 12^2} = 140 \text{ psig}$$

Since the total pressure drop across the heat exchangers is only 30 psig, which is far less than the theoretical pressure drop based on the calculated drag force, then the calculated drag force is conservative.

Flow Induced Vibration

Flow induced vibration was examined in Calc. M-CLC-K-00225 [6]. The natural frequency of a single heat exchanger tube and the surrounding fluid is 147 hertz, which is far from the vortex shedding frequency of 27 hertz. Thus, vortex shedding will not excite a tube harmonic and vibration amplitude due to flow induced vibration is a minimum. This is consistent with commercial experience which has shown that vortex shedding does not generally produce large amplitude tube vibrations in tube arrays with a tube spacing of 1.5 tube diameters or less because the proximity of the tubes causes regular vortex shedding to degenerate into broad band turbulence [5,11]. The tube spacing in the SRS heat exchangers is 1.5 tube diameters. Fluid elastic vibration of the SRS heat exchanger tubing was examined in Reference 23 which concluded that the heat exchanger tubes are stable.

Type A heat exchangers did experience vibration failures prior to the complete overhaul that was finished in 1966. Prior to the overhaul, the heat exchangers had a solid deflector plate without a strainer. At the edge of the solid deflector plate the water velocity was very high causing vibration problems. These vibration problems were solved with the introduction of the perforated cooling water inlet diffuser plate and strainer, which reduced the peak water velocity at the edge of the deflector plate. Types B and C heat exchangers also have a perforated cooling water inlet diffuser plate and strainer. Operating experience since 1966

has demonstrated that flow induced vibrations on the shell side are negligible. Thus, loads due to flow induced vibration, on the shell side, are not considered.

One instance of core rod vibration occurred on the tube side since the overhaul. A core rod moved out of the tube and entered a septifoil line where flow induced vibration caused the rod to wear through the tube. In 1990, all core rods aligned with the septifoil piping underwent a pull test to ensure there was adequate resistance to prevent movement of the core rod into the septifoil line. Thus, loads due to flow induced vibration, on the tube side, are not considered.

Tube Side Water Hammer

Water hammer in the PWS was examined in References 12 and 13. The maximum calculated pressure on the tube side of the heat exchanger is 195 psia and occurs from closing the rotovalve in 8.5 seconds¹ [12, 13]. This pressure is less than the 300 psig design pressure discussed previously and water hammer transient pressure on the inside of the heat exchanger tubes is nominal.

A non-mechanistic water hammer pressure of 450 psig is assumed in this analysis to determine the sensitivity of (1) the minimum wall thickness; and (2) the allowable flaw length; to water hammer transients. Pressure on the shell side of the tube is conservatively neglected for this non-mechanistic accident condition.

Shell Side Water Hammer

Water hammer in the CWS was examined in References 13, 14 and 15. The peak water hammer pressure on the shell side of the heat exchanger is 182 psia, which occurred during the 1989 incident in the 190K pumphouse. A typical pressure pulse in the heat exchanger has a pressure rise (Δp) of 110 psi in about 0.1 second, as shown in Figure B3 of Reference 15. Bending stresses in the tube are caused by differential pressure across the diameter of a tube. Pressure is transmitted at the speed of sound (≈ 2000 fps), thus the differential pressure acting on a 0.5" diameter tube is

$$\Delta p = \frac{110 \text{ psi}}{0.1 \text{ second}} \times \frac{0.5''}{2000 \times 12 \text{ ips}} = 0.023 \text{ psi}$$

and the moment due to water hammer, with a dynamic load amplification factor of 2 is

$$M_{\text{water hammer}} = \frac{0.023 \text{ lbs/in} \times 0.5 \text{ in} (24.9 \text{ in})^2}{12} \times 2 = 1.19 \text{ in lb}$$

This water hammer scenario occurred when the ECS cross tie header was pressurized without flow to the heat exchangers. Thus M_{drag} is not combined with $M_{\text{water hammer}}$.

Seismic Loadings

The seismic loading on a heat exchanger tube was evaluated in Calc. M-CLC-K-00225 [6]. The natural frequency of a single heat exchanger tube (0.5" OD, 0.049" thick) and the surrounding fluid is 147 hertz, while the natural frequency of the heat exchanger is 8.6 hertz, yielding a maximum tube moment of $M_{\text{seismic}} = 0.662 \text{ in lbs}$ [6]. Reducing the

¹ Recent K reactor valve test indicated that the rotovalve closure time is between 13.7 and 15.9 seconds [17].

tube's wall thickness will reduce the tubes natural frequency. At 25% of the original wall thickness, the tube natural frequency is reduced to 73 hertz, but the seismic response of the tube is essentially the same because the systems response is dominated by the heat exchangers natural frequency. Thus, a seismic moment of 0.662 in lbs is used regardless of the wall thickness.

Thermal Stress due to Differential Temperature Across the Wall Thickness

The temperature differential between the primary and secondary loops is approximately 25°F. [RMS DATA 1600MW at 3/11/88 K-13.2] Conservatively adding 5°F for local variations in the tube temperature, the secondary bending stress in the tube wall becomes [21]

$$\sigma_h = \sigma_l = 0.5 \Delta T \frac{\alpha E}{1-\nu} = \pm 5226 \text{ psi}$$

Where α is the coefficient of thermal expansion, $\alpha=8.9 \times 10^{-6} / ^\circ\text{F}$, E is the elastic modulus, $E=27,400,000$ psi, and ν is Poissons ratio, $\nu=0.3$. The stresses are in tension on the outer diameter and compression on the inner diameter. Calc. M-CLC-K-00225 [6] calculated thermal stresses less than one-half of the values calculated herein, indicating that the current values are conservative.

Differential Thermal Expansion of the Tube and Heat Exchanger Shell

The mean temperature of the tubes is approximately 15°F hotter than the heat exchanger shell. Thus, the tubes are in axial compression due to thermal expansion. Assuming that the heat exchanger shell is perfect rigid, then the secondary expansion stress is

$$\sigma = \alpha E \Delta T = -3660 \text{ psi}$$

Including the flexibility of the heat exchanger shell reduces the secondary expansion stress to

$$\sigma = \left(1 - \frac{A_{\text{tubes}}}{A_{\text{tubes}} + A_{\text{shell}}} \right) (-3660) = -730 \text{ psi}$$

Where the area of the tubes is 8800(0.0694 in²) and the area of the shell is 138 in². Since this thermal stress reduces the longitudinal pressure stress (PR/2t) it is conservatively neglected.

Differential Radial Thermal Expansion of the Tube and Baffle Plate

Since the baffle plates are in contact with the tubes, the baffle plates are assumed to have the same temperature as the mean tube temperature. Thus, secondary bending stresses in the wall of the tubing from baffle restraint are not considered

4.2. Minimum Wall Requirements for Gross Thinning

The tubing was originally designed to the ASME Section VIII, Division 1 code. The minimum required wall thickness is determined using the Division 2 rules, given in Appendix 4, Figure 4-130.1. The primary membrane stress intensities (P_M) for normal operating conditions are

$$\sigma_h = \frac{P R}{t} < S_m$$

$$\sigma_l = \frac{P R}{2t} + \frac{M_{dl} + M_{drag}}{Z} < S_m$$

Where R and t are the mean radius and thickness respectively. The section modulus, Z, is conservatively calculated by assuming that all of the wall thinning occurs on the outer fibers. S_m is the ASME Section VIII design stress intensity given in Table 5.

Primary membrane stress intensities (P_M) for accident conditions are

$$\sigma_{h \text{ seismic}} = \sigma_{h \text{ SSWH}} = \frac{P_{300} R}{t} < k S_m$$

$$\sigma_{h \text{ TSWH}} = \frac{P_{450} R}{t} < k S_m$$

$$\sigma_{l \text{ seismic}} = \frac{P_{300} R}{2t} + \frac{M_{dl} + M_{drag} + M_{seismic}}{Z} < k S_m$$

$$\sigma_{l \text{ SSWH}} = \frac{P_{300} R}{2t} + \frac{M_{dl} + M_{water \text{ hammer}}}{Z} < k S_m$$

$$\sigma_{l \text{ TSWH}} = \frac{P_{450} R}{2t} + \frac{M_{dl} + M_{drag}}{Z} < k S_m$$

Where k is equal to 1.2, $\sigma_{l \text{ seismic}}$ is the longitudinal stress due to seismic loading, $\sigma_{l \text{ SSWH}}$ is the longitudinal stress due to the shell side water hammer and $\sigma_{l \text{ TSWH}}$ is the longitudinal stress due to the non-mechanistic tube side water hammer. The pressure P or P₃₀₀ is the normal operating pressure, 300 psig. P₄₅₀ is the non-mechanistic tube side water hammer accident condition pressure of 450 psig.

Table 5 ASME Section VIII design Stress Intensity (psi) at 200°F [7]

Material Welded or Seamless	304 Welded	304 Seamless	316L Seamless	Sea-Cure Welded	Sea-Cure Seamless
S_m	17	20	16.7	24.1	28.3
1.5 S_m	25.5	30	25.05	36.15	42.45
3 S_m	51	60	50.1	72.3	84.9
k S_m	20.4	24	20.04	28.92	33.96
1.5 k S_m	30.6	36	30.06	43.38	50.94

k = 1.2

The local primary membrane plus bending stress intensities ($P_L + P_B$) for normal operating conditions are

$$\sigma_h = \frac{P R}{t} < 1.5 S_m$$

$$\sigma_l = \frac{P R}{2t} + \frac{M_{dl} + M_{drag}}{Z} < 1.5 S_m$$

For accident conditions the local primary membrane plus bending stress intensities ($P_L + P_B$) are

$$\sigma_{h \text{ seismic}} = \sigma_{h \text{ SSWH}} = \frac{P_{300} R}{t} < 1.5 k S_m$$

$$\sigma_{h \text{ TSWH}} = \frac{P_{450} R}{t} < 1.5 k S_m$$

$$\sigma_{l \text{ seismic}} = \frac{P_{300} R}{2t} + \frac{M_{dl} + M_{drag} + M_{seismic}}{Z} < 1.5 k S_m$$

$$\sigma_{l \text{ SSWH}} = \frac{P_{300} R}{2t} + \frac{M_{dl} + M_{water \text{ hammer}}}{Z} < 1.5 k S_m$$

$$\sigma_{l \text{ TSWH}} = \frac{P_{450} R}{2t} + \frac{M_{dl} + M_{drag}}{Z} < 1.5 k S_m$$

The local primary membrane plus bending plus secondary stress intensities ($P_L + P_B + Q$) for normal operating conditions are

$$\sigma_h = \frac{P R}{t} + 0.5 \Delta T \frac{\alpha E}{1-\nu} < 3 S_m$$

$$\sigma_l = \frac{P R}{2t} + \frac{M_{dl} + M_{drag}}{Z} + \frac{6}{t^2} \frac{P R^2}{E} 2\beta^2 D + 0.5 \Delta T \frac{\alpha E}{1-\nu} < 3 S_m$$

The compressive thermal force caused by the longitudinal differential expansion of the heat exchanger tubes and shell is conservatively neglected from σ_l . ASME Section VII does not consider secondary stresses during an accident condition.

The stresses are calculated in Table 6 with various wall thicknesses for normal operation, seismic and shell side water hammer accident conditions. The non-mechanistic water hammer accident condition on the tube side is discussed later. Allowable stresses were previously given in Table 5. The limiting case is hoop stress during normal operation which yields a minimum tube thickness of 0.0037" for 316L stainless steel. The primary membrane, bending and secondary stresses ($P_L + P_B + Q$) for normal operation are also close to the allowable stress. Thus, a reduction of the internal pressure from 300 psig would not change the minimum wall thickness significantly.

**Table 6 Calculated ASME Section VIII Stresses
for Various Tube Wall Thicknesses**

Section Properties						
t (in)	0.0490	0.0375	0.0250	0.0125	0.003688	
R (in)	0.2255	0.2198	0.2135	0.2073	0.2028	
Z (in ³)	.007144	.005280	.003394	.001639	.000472	
ΔR (in)	.000011	.000014	.000020	.000038	.000122	
β (in ⁻¹)	11.61	13.59	17.10	24.88	46.78	
D (in lb)	295.2	132.3	39.2056	4.9007	0.12586	
Normal Operation Hoop Stresses (psi)						
Pm	1,381	1,758	2,562	4,974	16,500	< Sm
Pl+Pb	1,381	1,758	2,562	4,974	16,500	< 1.5 Sm
Pl+Pb+Q	6,606	6,984	7,788	10,200	21,726	< 3 Sm
Normal Operation Longitudinal Stresses (psi)						
Pm	1,142	1,491	2,233	4,458	15,087	< Sm
Pl+Pb	1,142	1,491	2,233	4,458	15,087	< 1.5 Sm
Pl+Pb+Q	8,629	9,657	11,853	18,450	50,002	< 3 Sm
Accident Condition Longitudinal Stresses - Seismic (psi)						
Pm	1,235	1,616	2,427	4,861	16,484	< k Sm
Pl+Pb	1,235	1,616	2,427	4,861	16,484	< 1.5 k Sm
Accident Condition Longitudinal Stresses - Shell Side Water Hammer (psi)						
Pm	1,156	1,510	2,262	4,519	15,298	< k Sm
Pl+Pb	1,156	1,510	2,262	4,519	15,298	< 1.5 k Sm

Notes: k=1.2

Allowable stresses are given in Table 5.

Note that the stresses were developed for a 24.9" span whereas the heat exchangers with 316L stainless steel have a 16.6" span. Additionally, general wall thinning was considered, whereas local wall thinning may have smaller required thickness. This analysis, with the largest span, minimum material properties and general wall thinning is bounding. A separate location dependant analysis, that relaxes these conservatisms, may be performed on a case-by-case basis to disposition indications less than 0.0037" thick.

The minimum allowable wall thickness for a heat exchanger tube to resist a 450 psig non-mechanistic water hammer loading is 0.0046" as shown in Table 7. The allowable non-mechanistic water hammer pressure that the 0.0037" thick tube can resist is 365 psig. Comparing these two results, water hammer transients require approximately 0.001" of thickness for every 100 psig above the 300 psig design pressure. The maximum allowable internal pressure on a 0.5" diameter, 0.049" thick tube is 3600 psig during normal operation and 4300 psig during a water hammer transient, also shown in Table 7.

Table 7 Calculated ASME Section VIII Stresses for Various Pressures

Loading Condition	Non-Mechanistic Water Hammer	Non-Mechanistic Water Hammer	Normal Operation	Non-Mechanistic Water Hammer	
Internal Pressure (psig)	450	365	3600	4300	
k	1.2	1.2	1	1.2	
Section Properties					
t (in)	0.004565	0.003688	0.049000	0.049000	
R (in)	0.2033	0.2028	0.2255	0.2255	
Z (in ³)	.000586	.000472	.007144	.007144	
ΔR (in)	.000149	.000122	.000136	.000163	
β (in ⁻¹)	41.96	46.78	11.61	11.61	
D (in lb)	0.238652	0.12586	295.2	295.2	
Hoop Stresses (psi)					
Pm	20,040	20,075	16,567	19,789	< k Sm
Pl+Pb	20,040	20,075	16,567	19,789	< 1.5 k Sm
Pl+Pb+Q	n/a	n/a	21,793	n/a	< 3 Sm
Longitudinal Stresses (psi)					
Pm	15,531	16,874	8,736	10,347	< k Sm
Pl+Pb	15,531	16,874	8,736	10,347	< 1.5 k Sm
Pl+Pb+Q	n/a	n/a	41,094	n/a	< 3 Sm

Allowable stresses are given in Table 5.

4.3. Minimum Wall Requirements for Localized Thinning

ASME Code Case N-480 contains an evaluation procedure and acceptance criteria for local wall thinning of piping due to erosion corrosion. The N-480 evaluation procedures are adopted, in this analysis, for local wall thinning regardless of the wasting mechanism (erosion corrosion or wear).

If $L_m(t)$ is less than or equal to $\sqrt{R t_{min}}$, then the allowable local thickness, t_{aloc} is given by curve 1 of Figure 3. Where R is the tube outside radius and t_{min} is the minimum thickness developed in the previous section and $L_m(t)$ is the length of the region less than t_{min} in the transverse (hoop) direction, Figure 4.

If $L_m(t)$ is greater than $\sqrt{R t_{min}}$ and L_m (Figure 4) is less than or equal to $2.65\sqrt{R t_{min}}$, then the allowable local thickness, t_{aloc} is given by the larger of

$$\frac{t_{aloc}}{t_{min}} \geq \frac{1.5\sqrt{R t_{min}}}{L} \left[1 - \frac{t_{nom}}{t_{min}} \right] + 1$$

$$\frac{t_{aloc}}{t_{min}} \geq \frac{0.353L_m}{\sqrt{R t_{min}}}$$

Where L and L_m are defined in Figure 4, and t_{nom} is the nominal tube thickness, 0.049".

If the requirements of the previous two criteria are not met then the allowable local thickness is given by the maximum of curve 2 in Figure 3 and

$$\frac{t_{aloc}}{t_{min}} \geq \frac{0.5 + \frac{t_{nom}}{t_{min}} \frac{\text{Moment}}{Z \times S_m}}{1.5}$$

Where the section modulus Z is based on t_{nom} , and S_m is multiplied by $k=1.2$ for accident conditions.

The allowable local wall thinning for normal operation, seismic and water hammer are calculated using the minimum thicknesses, pressures and moments in Table 8. An envelop of the allowable local wall thinning for normal operation, seismic and water hammer accident conditions, is shown in Figure 5, and based on $L=2"$ and minimum material properties. As discussed previously, the minimum thickness for gross wall thinning is 0.0046" (including the non-mechanistic tube side water hammer transient). From Figure 5, the allowable thickness for a very small wear scar is 0.0011". The allowable thickness $t_{aloc}=0.0032"$ for a wear scar that is 0.02" transverse by 0.085" axially and less than 0.0046" thick.

Table 8 Load Summary for Local Wall Thinning Analysis

	t_{min} (inch)	Pressure (psig)	Moment (in-lbs)	
Normal Operation	0.003688	300	3.23	$M_{dl} + M_{drag}$
Cold End Normal Operation	0.002180	110	3.23	$M_{dl} + M_{drag}$
Accident				
Seismic	0.003038	300	3.89	$M_{dl} + M_{drag} + M_{seismic}$
Shell Side Water Hammer	0.003037	300	3.33	$M_{dl} + M_{water\ hammer}$
Tube Side Water Hammer ¹	0.004565	450	3.23	$M_{dl} + M_{drag}$

The allowable wear scar for normal operation of a tube at the cold end of the heat exchanger, with a differential pressure of 110 psig is shown in Figure 6. At this operating condition, without accident loadings, the minimum thickness for gross wall thinning is 0.00218". The allowable thickness for a very small wear scar is 0.000654". The allowable thickness $t_{aloc}=0.0017"$ for a wear scar that is 0.02" transverse by 0.085" axially and less than 0.00218" thick.

¹ This is a non-mechanistic loading.

4.4. Allowable Flaw Sizes

Instability and allowable flaw sizes are calculated with the limit load criteria based in ASME Section XI, Appendix C-3320 and C-3420. For circumferential flaws,

$$SF \times Pb \geq \frac{6 Sm}{\pi} \left(2 \sin \beta - \frac{a}{t} \sin \theta \right)$$

$$\text{where } \beta = \frac{1}{2} \left(\pi - \frac{a}{t} \theta - \pi \frac{SF P_m}{3 Sm} \right)$$

θ , a , t , R and β are shown in Figure 7,

$a \leq 0.75 t$,

SF is the factor of safety,

P_m is the membrane stress, $P_m = \frac{P R}{2t}$,

P_b is the bending stress, $P_b = \frac{M}{Z}$ and Z is the section modulus.

When $(\beta + \theta) > \pi$,

$$SF \times Pb \geq \frac{6 Sm}{\pi} \left(2 - \frac{a}{t} \right) \sin \beta$$

$$\text{where } \beta = \frac{\pi}{2 - \frac{a}{t}} \left(1 - \frac{a}{t} - \frac{SF P_m}{3 Sm} \right)$$

$a \leq 0.75 t$

For axial flaws,

$$\sigma_h = \frac{P \times R}{t} \leq \frac{3 Sm}{SF} \left[\frac{\frac{t}{a} - 1}{\frac{t}{a} - \frac{1}{M_2}} \right]$$

$$\text{where } M_2 = \sqrt{1 + \frac{1.61 \text{ length}^2}{4 R t}}$$

$a \leq 0.75 t$

the length is limited to l_{crit} where l_{crit} is defined by

$$\sigma_h = \frac{3 Sm}{M_2}, \text{ Solving } M_2 \text{ for the length yields}$$

$$\text{length} \leq l_{crit} = \sqrt{\left[\left(\frac{3 Sm}{\sigma_h} \right)^2 - 1 \right] \frac{4 R t}{1.61}}$$

if $\text{length} > l_{crit}$, ASME Section XI, IWB-3514-3, shall be used.

Allowable flaw sizes have a factor of safety of 3 on normal operating loads and a factor of safety of 1.5 on emergency or faulted conditions, which is consistent with Regulatory Guide 1.191. The flow stress of $3S_m$ is used where S_m is the minimum design stress intensity given in Table 5, or 16.7 ksi.

Normal operating loadings consist of 300 psig internal pressure combined with M_{dl} and M_{drag} as shown in Table 8. Seismic accident conditions consist of the normal operating loadings combined with seismic loading. The water hammer accident condition, on the shell side, consist of 300 psig internal pressure combined with M_{dl} and $M_{water\ hammer}$. A non-mechanistic water hammer accident is postulated on the tube side with 450 psig internal pressure combined with M_{dl} and M_{drag} . Secondary stresses due to (1) the radial restraint of the tubing; and (2) the ΔT through the thickness of the tube; are not included in the limit load flaw stability analysis. These secondary loads are relieved during the formation of the plastic hinge.

The allowable flaw sizes¹ for normal operating and the controlling accident loadings are given in Figures 8 and 9 for 0.049" and 0.025" tube wall thicknesses. Note that the flaw sizes given in Figures 8 and 9 do not contain allowances for further wall thinning or crack growth.

Normal operation controls the allowable flaw size for both cases, regardless of orientation or thickness. The allowable length of a long, part through wall axial flaws, l_{crit} , is limited by the assumed non-mechanistic water hammer pressure of 450 psig. Relaxing this artificial pressure, the allowable length is limited by normal operation. A summary of the allowable flaw sizes is given in Table 9.

Table 9 Summary of Allowable Flaw Sizes

	Tube Wall Thickness	
	0.049	0.025
Circumferential Flaws		
Throughwall	65% Circumference	56% Circumference
Part Throughwall	94% Depth	89% Depth
Axial Flaws		
Throughwall	2 inches	0.74 inches
Part Throughwall	91% Depth $\times$ 6" long	85% Depth $\times$ 2.2" long

Notes: $\% \text{ Depth} = \frac{\text{Crack Depth}}{\text{Tube Wall Thickness}} \times 100$

$\% \text{ Circumference} = \frac{\text{Crack Length}}{\text{Tube Circumference}} \times 100$

¹ Neglecting the $a \leq 0.75t$ limitation on the limit load equations.

The allowable size of a circumferential flaw is not sensitive to variations in wall thicknesses between 0.049" and 0.025". Whereas reducing the wall thickness by one-half reduces the allowable flaw length of a throughwall axial flaw by more than one-half. Similarly, reducing the wall thickness by one-half reduces the maximum length of a part throughwall flaw, L_{crit} , by more than one-half. L_{crit} for a part throughwall axial crack in 0.0375" thick tubing is 4". Thus, linear interpolation for wall thicknesses between 0.049" and 0.025" is permissible.

Figure 2 Tubing Restrain at Tube Sheet or Baffle

Figure 3 Allowable Depth and Length of Locally Thinned Areas [22]

Figure 4 Illustration of Local Wall Thinning [22]

Figure 5 Allowable Locally Thinned Areas for Normal Operating and Accident Conditions

Figure 6 Allowable Locally Thinned Areas for Normal Operating Conditions on the Cold End of the Heat Exchanger ($\Delta P=110$ psid)

Figure 7 Cross Section of Flawed Pipe

Figure 8. Allowable End of Life Flaw Size for Circumferential Flaws
(not including allowances for crack growth and wall thinning)

Notes:

$$\% \text{ Depth} = \frac{\text{Crack Depth}}{\text{Tube Wall Thickness}} \times 100$$

$$\% \text{ Circumference} = \frac{\text{Crack Length}}{\text{Tube Circumference}} \times 100$$

**Figure 9 Allowable End of Life Flaw Size for Axial Flaws
(not including allowances for crack growth and wall thinning)**

Notes:

$$\% \text{ Depth} = \frac{\text{Crack Depth}}{\text{Tube Wall Thickness}} \times 100$$

5. OPERABILITY REQUIREMENTS

An operational criteria for heat exchanger tube service is leakage. Historically, heat exchangers have been removed from service if total leakage from the heat exchanger increased from no leakage up to typically less than 1 lb./day. The present guideline for scheduled replacement of an individual heat exchanger due to leakage is 0.5 lbs./day [9]. This leakage level is monitored by the gamma monitors and grab sample analysis.

The IGSCC Leak Rate Model [10] is used to calculate the tube leakage with various assumed throughwall crack sizes. The IGSCC leak rate model was developed for leakage through intergranular stress corrosion cracks. This model has been benchmarked against Battelle Columbus experimental data with flows between 200 and 38,000 lbs./day. The IGSCC leak rate model's friction factor is greater than 100 for flows less than 1 lbs./day. This is judged to be conservative, even for highly branched transgranular cracks.

The crack opening calculation is based on linear elastic fracture mechanics with a correction for small scale yielding, using primary tubing stresses [16]. The change in crack opening area due to thermal stresses was neglected in this analysis.

The flow through an orifice is used to calculate the leakage through a pit

$$Q = C_d \rho \frac{\pi D^2}{4} \sqrt{\frac{2 g \Delta p}{\rho}} \quad (2078.5)$$

Where Q is the flow in lbs/day
D is the diameter of the pit (in)
C_d is the orifice coefficient ≈ 0.82
ρ is the density (lb./ft³)
g = 386.4 in/sec²
Δp is the pressure drop (psi)

At the hot end of the heat exchanger, the differential pressure between the tube and shell side fluids is 180 psi, Table 1, while at the cold end of the heat exchanger, the differential pressure is 110 psi.

Total heat exchanger leakage may be conservatively ascribed to a single tube in applying the results of the Leak Rate Model to calculate leakage flaw sizes. The leakage flaw sizes are compared to the flaw allowable lengths to provide a margin of safety on the leakage flaw size.

The results of the Leak Rate Model calculations for circumferential and axial tube cracks for tubes near the heat exchanger inlet and outlet (hot and cold ends) are shown in Figures 10 and 11. Similarly, the leak rate through a pit is shown in Figure 12. The flow rates (lbs./day) versus crack length are shown for 0.049" and 0.025" heat exchanger tube thicknesses. For the 0.049" thick tubing, the crack lengths required to obtain 0.5 lbs./day leakage (reference leakage size crack) range from approximately 0.4" on the hot end and 0.6" on the cold end. In the previous chapter, the allowable throughwall flaw length was determined to be 1" (65% of the circumference) for a circumferential flaw and 2" for an axial flaw with factors of safety (applied to stress) of 3 on normal operation and 1.5 on accident conditions. The allowable flaw lengths are recalculated with a uniform factor of safety of 3 on both normal operation and accident conditions in Table 10. Since the

allowable flaw size is larger than the leakage flaw size, the tubing has structural margins on stress which are greater than 3.

The structural margin on crack size is the ratio of the crack instability length to the 0.5 lb./day leak size crack (reference leakage size crack). The crack instability length is calculated with the limit load and a factor of safety (on stress) of one. The instability length, reference leakage size crack and structural margin on length are given in Table 10. As seen in Table 10, the minimum structural margin on length is 1.9, which is adequate. As the thickness decreases, the limiting structural margin increases because the reference leakage size crack decreases more than the instability length.

The structural integrity of deep pits is bounded by a throughwall crack. Pit sizes approaching the allowable throughwall crack size, Table 9, are not credible. Thus, the remaining ligament at the head of the pit is conservatively limited to the minimum wall thickness.

SRS heat exchanger tube structural requirements developed in the previous chapter define minimum thicknesses and allowable flaw lengths to avoid gross rupture of the tube. The structural margins on stress and crack length, Table 10, demonstrate that a degraded tube with a throughwall crack would leak enough moderator to allow detection, and without the possibility of failure by gross rupture. Furthermore, the Site Safety Analysis Report (SAR) explicitly considers a bounding scenario of full tube rupture.

Table 10 Structural Margins on length for Heat Exchanger Tubing

Flaw Orientation	Reference Leakage ^a Size Crack (in)	Allowable Length ^b (in)	Structural Margin on Stress	Instability Length ^c (in)	Structural Margin on Length
0.049" thickness					
Circum.	0.6	0.96	>3	1.13	1.9
Axial	0.6	1.3	>3	4.00	6.7
0.025" thickness					
Circum.	0.4	0.80	>3	1.03	2.6
Axial	0.4	0.48	>3	2.25	5.6
Pit	0.001	0.48	very large	1.08	very large

a) 0.5 lbs./day

b) Allowable length with a factor of safety of 3 on both normal operating and accident conditions.

c) Allowable length with a factor of safety of 1 on both normal operating and accident conditions.

Figure 10 Leak Rate for a Circumferential Heat Exchanger Tube Crack

Figure 11 Leak Rate for an Axial Heat Exchanger Tube Crack

Figure 12 Leak Rate for a Heat Exchanger Tube Pit

6. INSPECTION ACCEPTANCE CRITERIA

The structural and operation requirements developed in the previous two chapters are synthesized into an ASME Section XI compatible end-of-evaluation period acceptance criteria. Allowances for further wall thinning and/or crack growth are not included in this criteria. This acceptance criteria is intended to 1) prevent gross rupture, and 2) prevent leakage, based on the configuration at the end of an evaluation period.

To prevent leakage, ASME Section XI does not allow throughwall cracks and limits the crack depth of part throughwall flaws to 75% of the thickness. Thus, this structural acceptance criteria limits flaw depths to 75% of the remaining tube wall thickness.

ASME Section XI restricts the length of part throughwall axial flaws to l_{crit} . Part throughwall axial flaws longer than l_{crit} are governed by ASME Section XI, IWB-3514-3, which does not specifically address tubing. Long, part throughwall axial flaws, less than 10% of the wall thickness deep, are judged to be consistent with the intent of ASME Section XI, and are allowed in this acceptance criteria.

The local wall thinning criteria developed in Chapter 4 is the acceptance criteria for pits and wear scars. The minimum required remaining ligament is derived from Figure 5.

Combining these limitations, with the minimum wall thickness criteria yields the heat exchanger tubing allowable end-of-evaluation period acceptance criteria, shown in Table 11.

**Table 11a Heat Exchanger Tubing Allowable End-Of-Evaluation Period
Wall Thickness^c**

Remaining Wall Thickness	General Thinning	Local Thinning	Circumferential Cracks	Axial Cracks
0.049"	Permitted	Permitted	Table 11b	Table 11c
0.025"	Permitted	Permitted	Table 11b	Table 11c
0.005"	Permitted	Permitted	Not permitted without further evaluation	Not permitted without further evaluation
0.004"	Not permitted	OD ^e <1/16"	Not permitted	Not permitted
0.002	Not permitted	OD ^e <1/32"	Not permitted	Not permitted
0.0014	Not permitted	OD ^e <1/50"	Not permitted	Not permitted
<0.0014	Not permitted	Not permitted	Not permitted	Not permitted

**Table 11b Heat Exchanger Tubing Allowable End-Of-Evaluation Period
Flaw Depth To Thickness Ratios for Circumferential Flaws^{b,d}**

Thickness	Ratio of Flaw Length to Pipe Circumference							
	0	0.1	0.2	0.3	0.4	0.5	0.75	1.0
0.049"	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75
0.025"	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.75

**Table 11c Heat Exchanger Tubing Allowable End-Of-Evaluation Period
Flaw Depth To Thickness Ratios for Axial Flaws^{a,b,d}**

Thickness	Flaw Length (inches)							
	0	1	2	3	4	5	6	7
0.049"	0.75	0.75	0.75	0.75	0.75	0.75	0.75	0.10
0.043"	0.75	0.75	0.75	0.75	0.75	0.75	0.10	0.10
0.025"	0.75	0.75	0.75	0.10	0.10	0.10	0.10	0.10

- a) Interpolation between thicknesses is permitted.
- b) Flaws in tubing with wall thicknesses less than 0.025" are not permissible without further evaluation.
- c) Remaining wall thicknesses less than 0.005" are not permissible without further evaluation.
- d) Allowances for crack growth and wall thinning are not included.
- e) OD is the diameter of the pit or wear scar with a thickness less than 0.005"

7. REFERENCES

- 1 WSRC-TR-91-42-DRAFT, *Process Water Heat Exchanger Design and Operating History*, D. R. Ketcham and K. E. Kehr.
- 2 *Stainless Steel and High Alloy Tubing and Pipe*, Trent Tube Division of Crucible Materials Corp. Literature
- 3 NES-ETH-920039, *HX Inlet Pressure with Bingham Pump Dead-Headed*, J. Whitehouse, January 1992.
- 4 *Bases for Plugging Degraded PWR Steam Generator Tubes*, U.S. NRC Regulatory Guide 1.121, 1976.
- 5 *Inservice Inspection of Pressurized Water Reactor Steam Generator Tubes*, U.S. NRC Regulatory Guide 1.83, 1975.
- 6 Calc #M-CLC-K-00225, *PW HX Tube Stress Analysis*, H. Flanders, G. B. Rawls and B. J. Bryant, January 1992.
- 7 ASME Boiler and Pressure Vessel Code, Sections III (1989), VIII (1986) and XI (1989).
- 8 Formulas for Stress and Strain, R. J. Roark and W. C. Young, 1975.
- 9 DPSOL 105-2313, *Determining D2O Leakage from Heat Exchangers*.
- 10 WSRC-RP-90-93, *Reactor Materials Program Single Phase Leak Rates Through Intergranular Stress Corrosion Cracks*, S. A. Swamy, G. E. Mertz and K. J. Stoner., February 1990.
- 11 Flow-Induced Vibration, R.D. Blevins.
- 12 DPST-87-822, *PW System Water Hammer Stress Analysis - Phase I*, N. T. Hightower, 1987.
- 13 DPST-88-825, *CW and PW System Water Hammer Stress Analysis - Phase II*, N. T. Hightower, et al, 1988.
- 14 DPST-89-298, *Water Hammer Analysis of the January 22, 1989 Event in Building 190-K*, N. T. Hightower, W. F. Hillyard, 1989.
- 15 WSRC-RP-89-646, *Bechtel Water Hammer Analysis of the January 22, 1989 Event in Building 190-K*, N. T. Hightower, A. H. Arastu, 1989.
- 16 WSRC-RP-90-91, *Reactor Materials Program Leak-Before-Break Analysis of the Reactor Process Water Piping System*, R. L. Sindelar, G. E. Mertz and K. J. Stoner, March 1990.
- 17 RRD-RSE-910465, *Rotovale and Septifoil Valve Stroke Times*, D. G. Hoezee, D. M. Barnes, W. S. Large, April 1991.

- 18 SRL-MTS-921002, *Materials of Construction for Heat Exchangers - 105K*, D. M. Barnes and D. R. Ketchum, February 1992.
- 19 *Introduction to Fluid Mechanics*, R. W. Fox and A. T. McDonald, 1987.
- 20 *Standard Handbook for Mechanical Engineers*, T. Baumeister and L. S. Marks, Editors, 1967.
- 21 *Formulas For Stress and Strain*, R. J. Roark and W. C. Young, 1975.
- 22 ASME Code Case N-480 *Examination Requirements for Pipe Wall Thinning Due to Single Phase Erosion and Corrosion Section XI, Division I*, May 1990.
- 23 *Fluidelastic Vibration Assessment*, Westinghouse Science and Technology Center Inter-Office Memo from H. J. Connors to J. A. Begley, 2/14/92.

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