

ATTACHMENT 2

**CONSUMERS ENERGY COMPANY
PALISADES PLANT
DOCKET 50-255**

**INSERVICE INSPECTION PROGRAM - SUBMITTAL OF RELIEF REQUEST NO. 14 FOR
NRC APPROVAL**

Structural Integrity Associates Report SIR-99-032, Rev 0

**Flaw Tolerance and Leakage Evaluation
Spent Fuel Pool Heat Exchanger #E-53B Nozzle
Palisades Nuclear Plant**

35 Pages

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Report No.: SIR-99-032
Revision No.: 0
Project No.: CPC-10Q
File No.: CPC-10Q-401
March 1999

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Palisades Nuclear Plant**

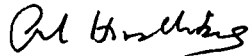
Prepared for:

Consumers Energy
(Contract No. C0030239)

Prepared by:

Structural Integrity Associates, Inc.
San Jose, California

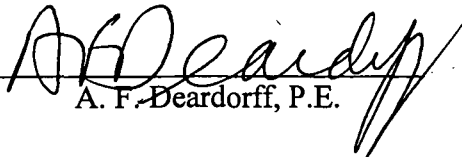
Prepared by:



P. Hirschberg, P.E.

Date: 3-25-99

Reviewed by:



A. F. Deardorff, P.E.

Date: 3/25/99

Approved by:



N. G. Cofie, Ph.D.

Date: 3/25/99



REVISION CONTROL SHEET

Document Number: SIR-99-032, Rev. 0

Title: Flaw Tolerance and Leakage Evaluation, Spent Fuel Pool Heat Exchanger E-53B
Nozzle Palisades Nuclear Plant

Client: Consumers Energy

SI Project Number: CPC-10Q

Section	Pages	Revision	Date	Comments
-	-	0	3/29/99	Initial Issue



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1.0 INTRODUCTION

At Consumers Energy (CE) Palisades Nuclear plant, evidence of leakage has been observed at the reinforcing pad of the spent fuel pool cooling water heat exchanger E-53B inlet nozzles. At the lower of the two vent holes (6 o'clock position), there is a residue of light-colored materials such as would exist after evaporation of water with mineral content. There is currently no evidence of any moisture at the vent hole. Based on this, it is concluded that there may have been some leakage from a small flaw in the weld in the past, but that it has now stopped leaking, becoming plugged due to foreign matter collecting in the leakage path.

Structural Integrity Associates (SI) was contracted to perform an assessment of the nozzle to justify continued operation without a repair. Since the operating pressure is very low and the nozzle moments are very low, it was expected that the nozzle should be very flaw tolerant. The nozzle also has a welded reinforcing pad that is significantly thicker than the heat exchanger shell, adding to the overall structural strength of the junction. The geometry, materials and loadings are described in Section 2.0

Thus, several evaluations have been conducted as described in this report:

- A detailed finite element analysis of the shell-to-nozzle intersection has been conducted. Pressure, dead weight, and seismic loadings were considered in the analysis. Stresses were predicted for the uncracked nozzle which were used as input to subsequent calculations. (See Section 3.0.)

- Based on the evidence of past leakage, the maximum possible leakage from the vent holes was calculated based on conservative assumptions of temperature and natural convection. The size of the nozzle-to-shell weld cracks that would result in the calculated leakage rates were determined. (See Section 4.0.)

- The critical size of through-wall cracks in the nozzle-to-shell weld was determined using ASME Section XI, Appendix C net section collapse methods, modified to account for a shear failure collapse mode. (See Section 5.0.)
- Based on the maximum loadings, fatigue crack growth analysis was performed to demonstrate that sub-critical cracks would not grow due to application of seismic and pressure cycling. In addition the potential for IGSCC crack growth was evaluated. (See Section 6.0.)
- An additional finite element analysis was done assuming that the through-wall flaw had grown completely around the circumference of the pipe, to evaluate the capability of the reinforcing pad to maintain the structural integrity of the nozzle to shell junction. (See Section 7.0.)
- Using the results above, a comparison of the leakage size cracks and critical size cracks was made to demonstrate that there is ample margin between the two. (See Section 8.0.)

Conclusions are provided in Section 9.0, showing that there is more than a factor of two between the crack size that meets ASME Section XI structural acceptance criteria and the sizes that would be necessary to pass the conservatively-estimated leakage rates. This includes the safety margins inherent in Section XI for normal/upset conditions even when the SSE seismic loadings are applied.

2.0 DESCRIPTION OF HEAT EXCHANGER 53B GEOMETRY, MATERIALS AND LOADS

The spent fuel pool heat exchanger is a shell and tube heat exchanger [1]. The tube-side water communicates with the spent fuel pool. At each end of the heat exchanger, there are heads with nozzles connecting to the spent fuel pool cooling piping.

The heads are made of SA-240, Type 304 stainless steel [1]. The cylindrical head has an inside diameter of 25 inches and a wall thickness of $\frac{3}{16}$ inches. The horizontal nozzle, from the side of the cylindrical head, is SA-312, Type 304 made from Schedule 10S pipe (12.75-inch OD and 0.18-inch thick wall). The reinforcing pad around the nozzle is SA-240, Type 304. The reinforcing pad has a thickness of 0.25 inches and extends 2.5 inches radially beyond the outer radius of the nozzle, welded to the nozzle and shell with a $\frac{1}{4}$ -inch structural fillet weld [2]. The nozzle, with a SA-182 F304 flange, extends 7.5 inches beyond the ID of the cylindrical head. Figure 2-1 shows the local geometry at the nozzle.

The reinforcing pad has two $\frac{3}{8}$ -inch vent holes located at 6 and 12 o'clock on the nozzle [2, 3]. The evidence of leakage from the vent hole is evident only at the bottom vent hole.

The heat exchanger has a design pressure of 125 psig and design temperature of 150°F. The maximum expected inlet/outlet temperatures of the spent fuel cooling water are 120-130°F inlet and 110°F outlet [4]. The nozzle loadings [5], are shown in Table 2-1. Thermal expansion moments were considered to be insignificant in the piping analysis.

Table 2-1
Equipment Nozzle Load Summary

Equipment: Heat Exchanger E-53B

Analysis Node Pt.: 523

Load Case	Load or Load Combination	Forces (lbs.)			Moments (ft.-lbs.)		
		Fx	Fy	Fz	Mx	My	Mz
10	Dead Weight (DW)	-23	-314	-54	-336	143	-26
35	OBE SAM (OSAM)	0	0	0	0	0	0
30	Seismic OBE (OBE)	103	171	61	378	255	208
45	SSE SAM (SSAM)	0	0	0	0	0	0
40	Seismic SSE (SSE)	205	343	122	756	509	415

X = Horizontal (transverse to nozzle)
 Y = Vertical
 Z = Axial to nozzle

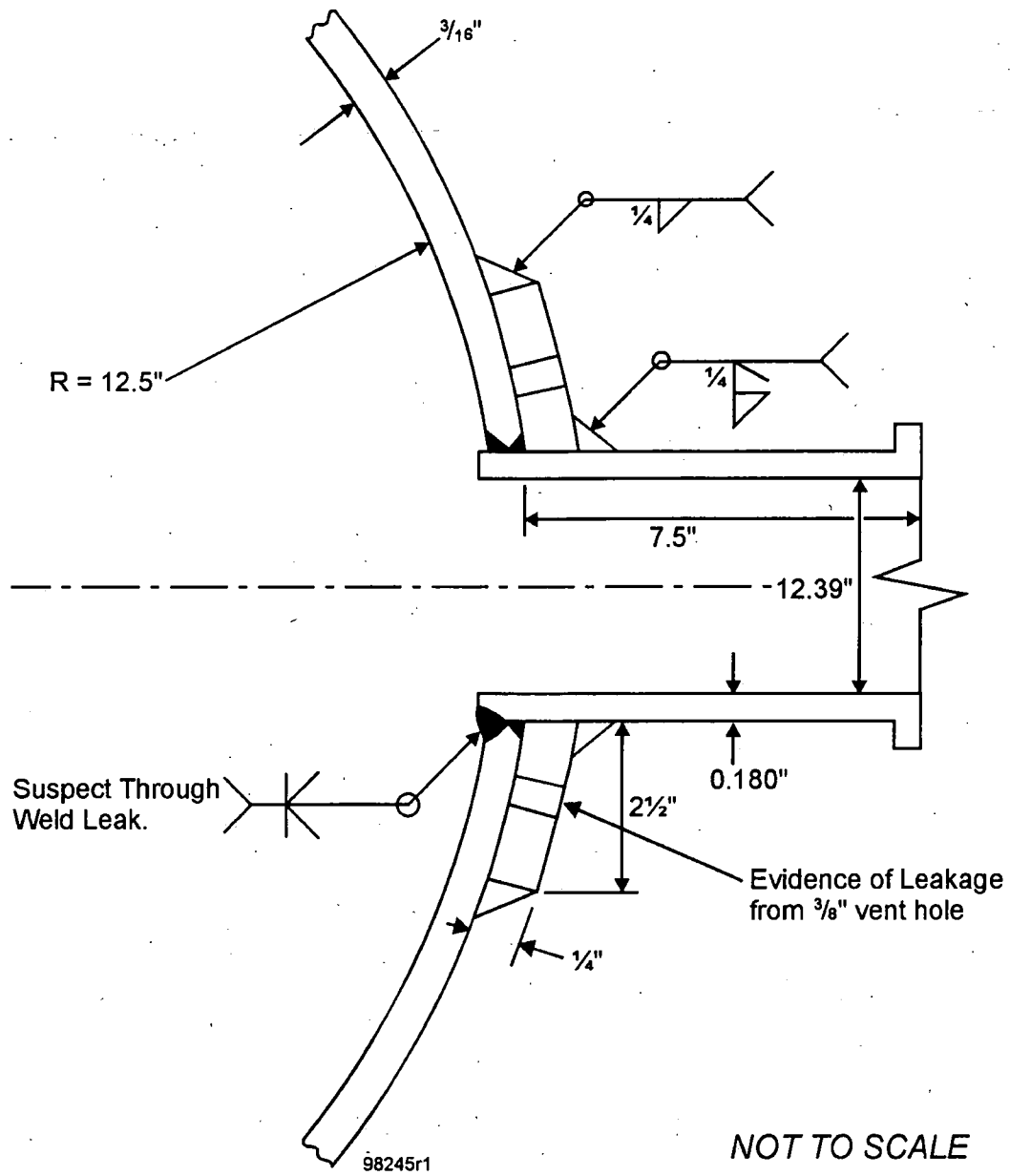


Figure 2-1. Geometry at Heat Exchanger 53B Nozzle

3.0 STRESS ANALYSIS OF HEAT EXCHANGER NOZZLE

To evaluate the stresses in the vicinity of the nozzle, a three dimensional finite element stress analysis was conducted using the ANSYS computer program [6]. The 3-D model was used so that the non-axisymmetric loading effects of the cylindrical nozzle-to-shell joint could be evaluated and the effects of the nozzle moments could be assessed. Figure 3-1 shows the model. The stress analysis was conducted using ANSYS version 5.3, SOLID-45 type 3-D structural solid elements, having 8 nodes and 3 translational degrees of freedom at each node. 3-D elastic beam (BEAM4) elements were used to transfer axial, shear, and moment loads to the face of the nozzle.

The finite element model included the heat exchanger shell, end cap, nozzle neck, nozzle reinforcing pad, and the two structural fillet welds connecting the reinforcing pad to the nozzle, neck and shell. The model assumed a fully intact double-v weld joining the nozzle neck and shell.

The loads applied to the heat exchanger vessel and nozzle were taken from [5] and [7]. The design pressure of 125 psi was applied, which represents normal operating conditions. The mechanical and seismic loads from the connected piping system shown in Table 2-1 were used. Material properties were taken from the ASME Code Appendices [8], evaluated at 150°F.

An analysis was run to evaluate the stresses in the unflawed component. The purpose of this evaluation was to determine the stresses in the vicinity of the shell-to-nozzle weld location. These stresses are important from the standpoint of opening potential cracks that would result in leakage during normal operation. Table 3-1 shows the computed membrane stresses due to pressure for a quarter section at the weld location. The stresses in the circumferential direction at the double-v weld are an order of magnitude higher than the radial direction due to the geometry of the combined nozzle/pad/shell model. Figure 3-2 shows the overall stress contour results for all loads.

The details of the finite element analysis are included in Reference [21].



Table 3-1

Membrane Stresses in Nozzle-to-Shell Weld

Angle on Nozzle (from horizontal)	Stress in Nozzle-to-Shell Weld, ksi (across crack)	
	Crack Parallel to Weld	Crack Perpendicular to Weld
0	0.896	18.42
15	1.084	17.33
30	1.393	14.42
45	1.367	10.48
60	0.908	6.38
75	0.420	3.11
90	0.217	2.02



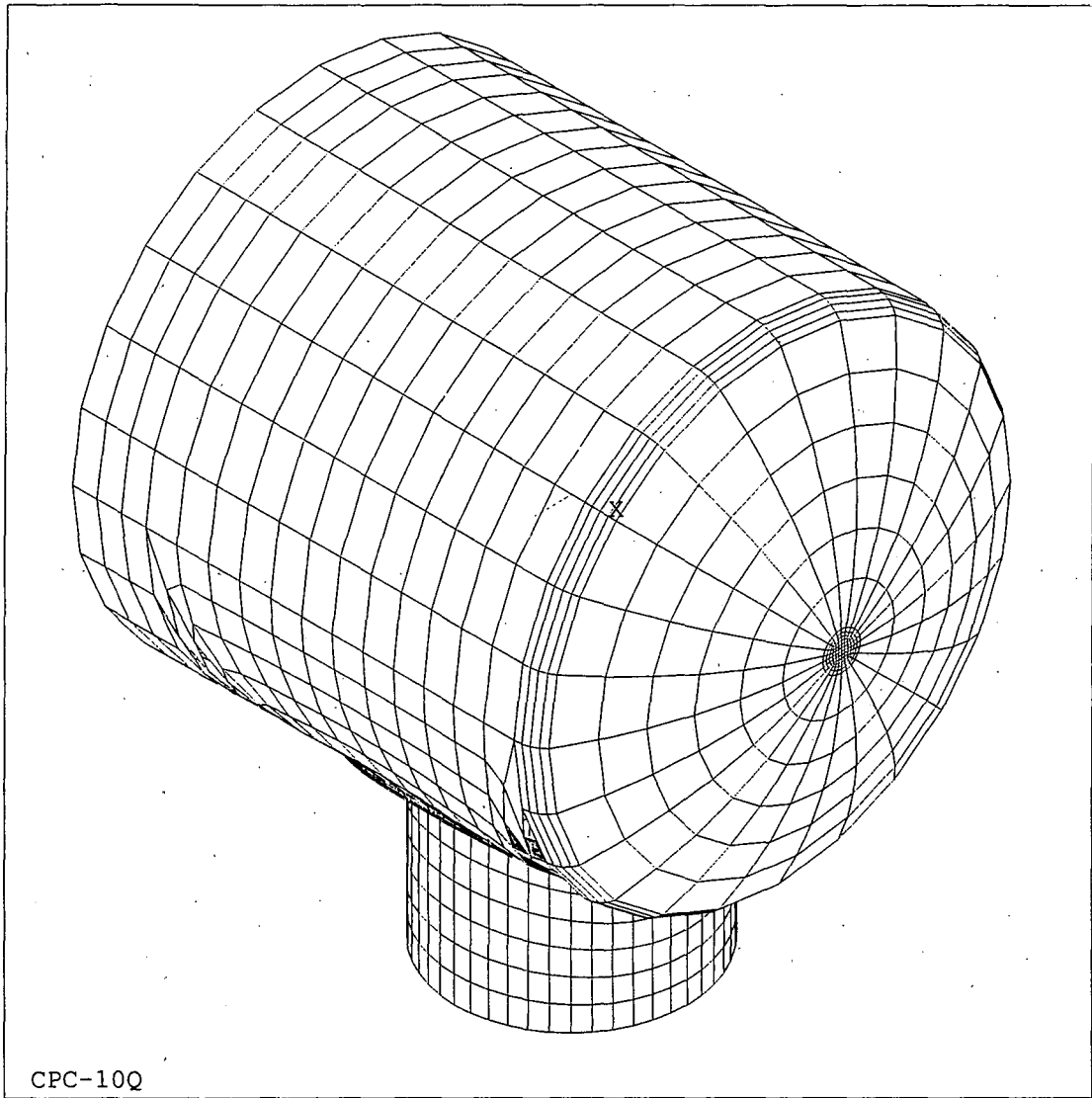


Figure 3-1. 3-D Finite Element Model of Heat Exchanger E-53B Head and Nozzle

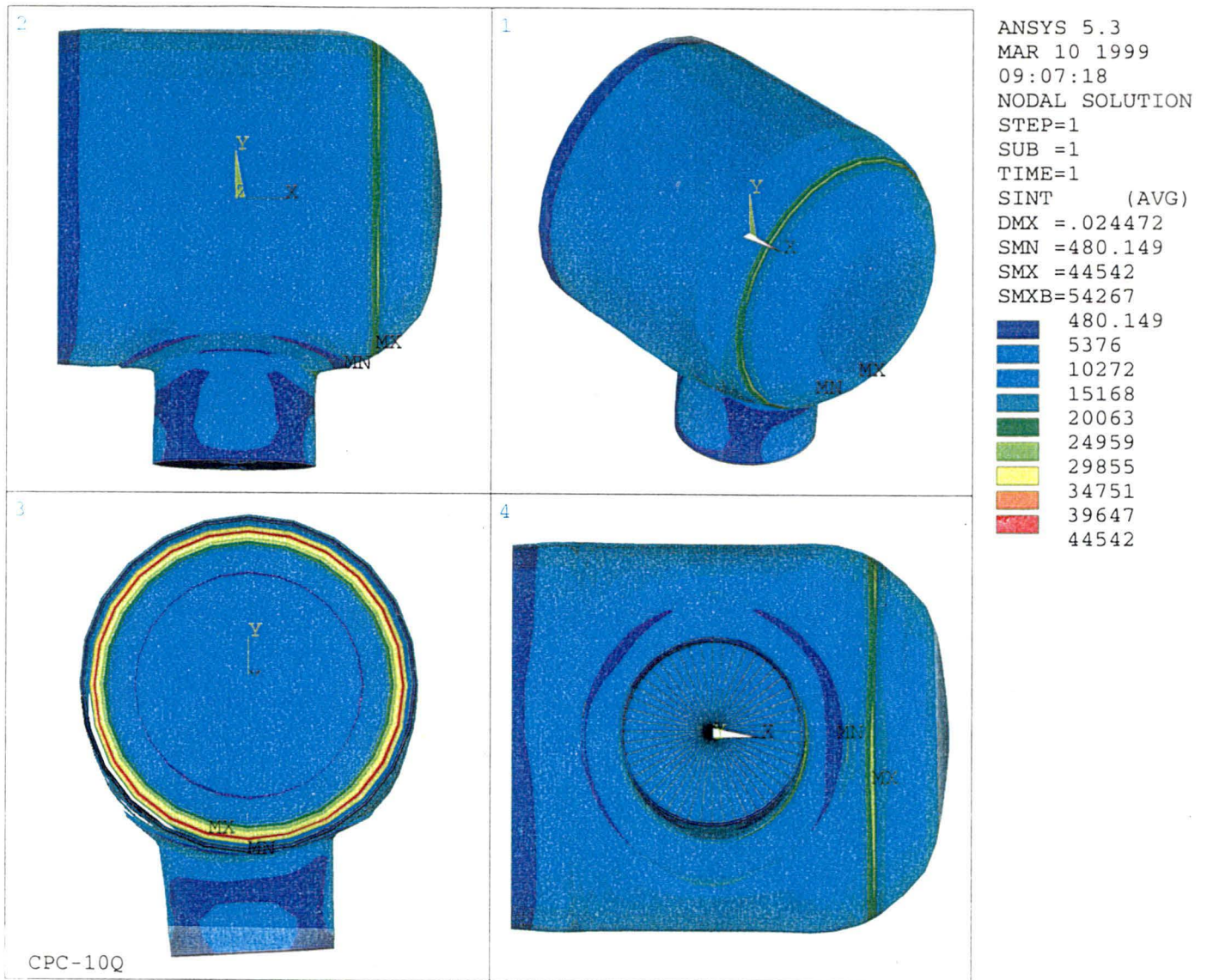


Figure 3-2. Nozzle-to-Shell Region Overall Stress Intensities

4.0 ESTIMATION OF LEAKAGE AND SIZE OF LEAKAGE CRACKS

To determine the potential size of the crack that produced the observed leakage at the vent hole, an estimate of the leakage was developed. Then using weld stresses developed from the finite element analysis, crack opening areas and the associated leakage rates were developed.

4.1 Leakage Rate Determination

The evidence of leakage at the reinforcing pad vent hole is from the presence of a white residue left from the evaporation of water sometime in the past. There is currently no evidence of moisture in the vent hole. There is no evidence of leakage from the upper vent hole.

To estimate the amount of leakage, a mass transfer calculation was conducted assuming that the bottom and sides of the vent hole were wet (100 percent humidity at the surfaces) and that the air surrounding the heat exchanger was dry. To maximize the predicted evaporation rate, it was conservatively assumed that the heat exchanger shell was at 130°F. This high temperature maximizes the vapor pressure and mass concentration of water at the wet surfaces. The diffusion coefficient for water vapor in air was taken at this same temperature and was 1.1162 ft²/hr, based on correcting basic data for 32°F [9]. Performing a pure diffusion mass transfer analysis (analogous to a heat transfer conduction evaluation), a leakage rate of 13.88 lb/year was determined.

However, the effects of local air currents and density differences can affect the rates of heat transfer and mass transfer at a surface. To determine this effect, the natural convection heat transfer coefficient at the heat exchanger was calculated [10]:

$$h = 0.28 (\Delta T/L)^{0.25} = 0.7446 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

where

- ΔT = surface to ambient temperature difference = 50°F
- L = characteristic length = 1 foot (assumed)

The Nusselt number is an estimate of the natural convection effect compared to a pure conduction effect. Based on the evaluation above,

$$\text{Nu} = hL/k = 46.5$$

where

$$k = \text{air thermal conductivity} = 0.016 \text{ Btu/hr-ft-}^\circ\text{F at } 100^\circ\text{F}$$

Assuming the increase in mass transfer could be analogous to the increase in heat transfer, as indicated by the Nusselt number, the diffusion mass transfer determined above was conservatively increased by a factor of 50 to arrive as an upper bound estimate of the leakage rate of 694 lb/year (0.000165 gpm).

This is believed to be a very conservative upper bound because 1) the one foot dimension used in the Nusselt Number correlation is large compared to the size of the vent hole, 2) the ΔT used is conservative, 3) the local geometry of the vent hole would tend to mitigate the natural convection effects, and 4) it is assumed that there is no humidity in the room. In reality, it would be expected that leakage of this amount would lead to some evidence of wetness or drop formation around the vent hole.

Considering the amount of liquid leaked in one minute, the hemispherical drop size for this leakage rate was estimated to be approximately 0.5 inches. Thus, this is a very conservative estimate of the leakage rate. If this amount of water were leaking, there would definitely be clear evidence of water leakage from the vent hole, since the water could not evaporate into the air this rapidly. However, use of this leakage rate is conservative for estimating the size of potential flaws:

4.2 Determination of Potential Through-wall Crack Length

To calculate the crack size in the heat exchanger shell-to-nozzle weld, the SI program **pc-LEAK** [11] was used. This program is based on the concepts of linear elastic fracture mechanics for calculating crack opening area [12, 13, 14]. Fundamental fluid mechanics methods are used to calculate leakage of water through the crack based upon the crack opening displacement, surface roughness and discontinuity losses (with more complex methods being available for calculation of two-phase flow). This program has been qualified under SI's Quality Assurance Program.

The fluid conditions were taken as 120°F (from the assumptions above) and 125 psig. For computing leakage, the membrane stresses across the crack were used consistent with the assumption in the fracture mechanics models. The distribution of the membrane stresses around the shell-to-nozzle weld is shown in Table 3-1.

The calculation was conducted for leakage from a pipe the size of the heat exchanger shell. In the analysis, there was no correction for the plastic zone at the crack tip so that minimum leakage would be calculated, to maximize the computed crack size. A surface roughness of 0.02 inches was assumed, but this had no effect on the resulting calculations, since the flow was in the laminar flow regime for the very small leakage rates. An entrance loss coefficient K of 2.7 was assumed, simulating a sharp-edged entrance to an orifice ($C = 0.6$ and $K = 1/C^2$). This also had very little effect on the results since the pressure drop was mainly due to friction in very tight cracks.

Two analyses were conducted to evaluate the leakage as described in the following subsections.

4.2.1 Leakage from a Crack Parallel to the Weld

For this analysis, the crack was assumed to be through-wall and extend along the weld. To simulate the additional stiffness of the nozzle side of the crack, the modulus of elasticity was doubled, having the effect of only one side of the crack opening. Cases were run for a crack length up to 8 inches assuming a range of stresses.



The calculated maximum crack size (2a) to cause a leak of 0.000165 gpm was approximately 4.5 inches ($2\theta \approx 45^\circ$) assuming that the crack was over the most lowly stressed region. For the case of a crack located at the most highly stressed region, the crack length was reduced to about 2 inches.

4.2.2 *Leakage from a Crack Perpendicular to the Weld.*

For this analysis, the crack was assumed to be transverse to the weld. This type of crack is assumed because of the relatively large membrane stresses in the weld region in the hoop direction of the pipe penetration. The leakage calculation was conducted using a model for a longitudinal crack in a pipe.

The predicted crack size to produce 0.000165 gpm ranges was about 0.18 inches (2a), approximately equal to the shell thickness, for the most highly stressed location. Lower stresses were present at the other locations, which would require larger cracks for the same amount of leakage. Larger cracks were not evaluated since pre-existing cracks in the base material were not considered to be credible.

Details of these calculations are given in Reference [22].

5.0 DETERMINATION OF CRITICAL THROUGH-WALL CRACK LENGTH

An evaluation was done to determine the length of a through-wall crack circumferentially around the nozzle that would still be able to withstand the applied loads without becoming unstable. The calculation used the limit load approach based on net section plastic collapse. The ASME Boiler and Pressure Vessel Code, Section XI, Appendix C [15], provides rules for evaluating circumferential flaws in austenitic piping. In Subsection C-3300, equations are given for determining the maximum allowable flaw length and depth for a given load, or vice versa. The equations determine the maximum load carrying capability of the remaining ligament, which occurs when the cross section of concern reaches fully plastic action limit load. Using the design conditions and external piping forces and moments provided [5, 7], the maximum allowable flaw size for the applied loads was determined.

The following assumptions were used in this calculation, based on [3]:

- Thermal expansion loads were considered to be negligible, per Palisades Specification M-195 [24], for systems at temperatures of 150°F or less. A review of the piping isometrics showed that there is sufficient flexibility in the piping routing and support system for this assumption to be valid.
- The seismic loads on the nozzle of the attached piping system were considered, but the self-inertia loads of the heat exchanger were not. The heat exchanger is classified as Class III equipment, for which seismic qualification is not required. Of interest here was whether the nozzle-to-shell weld joint would maintain its integrity if the attached piping were to be subjected to seismic excitation.
- The Safe Shutdown Earthquake (SSE) loads were used with the normal operating condition safety factor. This is conservative because the SSE loads are higher than the normal (upset) condition (OBE) loads, with a lower probability of occurrence.

- Shielded Metal Arc Welding (SMAW) was assumed to be the welding process used. This results in more conservative Z factors (see below) than if the weld process is known to be Gas Tungsten Arc welding (GTAW).
- The external piping applied lateral forces were given at the flange joint, approximately 7.5 inches from the shell. In order to apply them at the shell interface, additional moments, equivalent to the force times the distance from the flange to the shell, were applied.
- The load carrying capability of the reinforcing pad was conservatively neglected in this evaluation.
- The flow stress in shear is assumed to be half the flow stress in tension.
- Piping moments were combined by SRSS per ANSI B31.1, as required by [24].

In the limit load approach, the applied forces and moments are compared with the plastic load-carrying capacity of the cross-section containing the flaw. The assumption is that the material will form a plastic hinge, i.e. the entire cross section will deform plastically before the flaw grows to an unstable size. This assumption is valid for austenitic materials and non-flux weld metal. The remaining unflawed ligament must be able to accommodate the applied primary membrane and bending stresses in order to be acceptable. Safety factors are applied on the combined membrane and bending stresses to arrive at allowable loads. Relationships are given in ASME Section XI Appendix C between allowable applied stress and unflawed ligament size.

In this calculation, the loads were known but the flaw size was not. The maximum allowable through-wall flaw length was determined for the applied loads. The equations given in Appendix C which relate applied load to unflawed ligament at plastic collapse are the following:



C-3320(a) Equation (3)

$$P'_b = \frac{6S_m}{\pi} \left(2 \sin \beta - \frac{a}{t} \sin \theta \right)$$

and

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

where:

S_m = material stress allowable

P_m = piping membrane stress

a = flaw depth

t = section thickness

β = angle between vertical and neutral axis (see Figure 5-1)

θ = half angle of circumferential flaw length

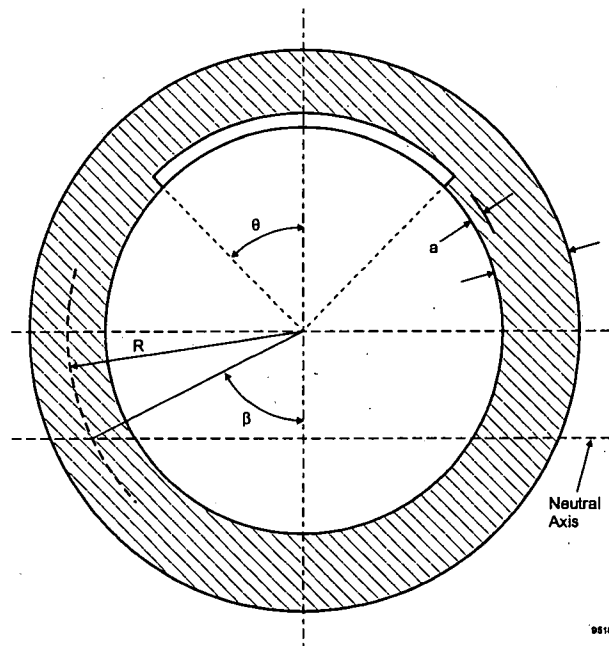


Figure 5-1. Appendix C Allowable Flaw Approach

Although the equations shown above were developed to determine the allowable stresses in piping with through-wall flaws, they are applicable to evaluating flaws in a weld between a pipe and a shell, if the allowable flow stress ($3S_m$) is reduced by a factor of two.

To determine the allowable flaw length, safety factors were applied. In addition, for flux welds (SMAW), an additional Z_1 factor was applied, to account for the fact that the weld metal is less ductile and may not reach fully plastic action before the flaw becomes unstable. Since the welding process was not known, SMAW was conservatively assumed. The Code equation is the following:

C-3320(c) Equation (6)

$$P'_b = Z_1(SF) \left(P_m + P_b + \frac{P_e}{SF} \right) - P_m$$

where:

SF = safety factor = 2.77 for normal operating conditions

P_b = piping bending stress

$Z_1 = 1.15 [1 + .013 (D_o - 4)]$

D_o = pipe outside diameter

P'_b = limit bending stress

P_e = piping expansion stress

The evaluation used the highest seismic loads, the SSE case. Using a Code safety factor of 2.77 was conservative because it is intended for normal operating conditions, while the SSE seismic loads are faulted conditions. A safety factor of 1.39 is allowed for faulted conditions.

It should be noted that the quantity $6S_m$ in the first equation represents twice the flow stress (and $3S_m$ in the second equation is the flow stress). In this calculation, the section with the flaw is being loaded in shear. The limiting stress of interest is the shear stress on the weld cross-section. An allowable flow stress of half the flow stress under tension, or $1.5 S_m$, was therefore used.

Thus, the quantity $6S_m$ in the first equation becomes $3S_m$; the quantity $3S_m$ in the second equation becomes $1.5 S_m$.

Results for ultrasonic testing were available which showed that the shell thickness was larger than nominal [16], but the nominal values were conservatively used. S_m values for the base metal were conservatively used for the weld metal.

The result of the evaluation was that angle $\theta = 73.9$ degrees. This represents the allowable half angle for a through-wall crack in the weld circumferentially around the pipe that meets normal Code allowable stresses. The total crack length is double this, about 148 degrees, or 16.4 inches.

Thus, the result of this calculation was that if the nozzle to shell double-v weld were to contain a through-wall circumferential flaw extending approximately 148 degrees around the circumference, it would still be able to withstand the applied loads. This result conservatively takes no credit for the reinforcing pad, and includes the Section XI factor of safety for normal/upset loads.

Another case was run applying a factor of $\sqrt{2}$ to the applied stresses on the flaw, consistent with the philosophy of the Leak Before Break methodology specified by the NRC [17]. The result was that even with a factor of safety of $\sqrt{2}$ on stress, a flaw length of about 119 degrees around the circumference, or 13.25 inches, would be acceptable.

Details of these calculations are given in Reference [23].

6.0 EVALUATION OF POTENTIAL CRACK GROWTH

An evaluation was done to determine how much the existing flaw could potentially grow over time under the applied loads. ASME Section XI, Appendix C, Subsection C-3200 [15] gives rules for determining flaw growth in austenitic materials. There are two mechanisms for growing flaws in austenitic piping – stress corrosion cracking, and fatigue cycling. Reference [18] indicates that IGSCC is not a concern in lines where the normal operating temperature is less than 200° F. As the spent fuel pool heat exchanger inlet lines operate at 150°F or less, IGSCC is not considered to have the potential to cause the flaw in the nozzle weld to grow.

As for fatigue cycling, there are no significant cyclic loading conditions at the nozzle. The spent fuel pool pumps are judged to be too far away to cause vibration at the nozzle. Over the next several years, the number of pressure cycles from system starts will not be significant. The main cyclic loading would be the seismic loading from the attached piping, should a seismic event occur. It will be conservatively assumed that the seismic loads will produce the equivalent of 50 stress cycles.

The EPRI Ductile Fracture Handbook [19] gives an equation for calculating the stress intensity factor for a through-wall flaw in a cylinder under tensile loading:

$$K_I = \sigma_t (\pi R \theta)^{0.5} F_t$$

where:

σ_t = axial stress

$$F_t = 1 + A [5.3303(\theta/\pi)^{1.5} + 18.773(\theta/\pi)^{4.24}]$$

$$A = [0.4(R/t) - 3.0]^{0.25} \quad \text{for } 10 \leq R/t \leq 20$$

θ = defined in Figure 5-1

R = outside radius of cylinder

This model is intended for a flaw in the nozzle pipe; however, there is no model available that exactly matches the geometry being evaluated, and it is judged that this model provides a good

approximation. Although this formulation is not strictly applicable for $R/t = 33$ of this case, it is sufficiently accurate for the purposes of demonstrating that crack growth would be small. Since this equation assumes that the loading is tensile, the shear stresses calculated above were conservatively doubled. The pressure and dead load stresses were separated out from the P_m and P_b terms and treated as mean stresses, and the seismic (SSE) stress was considered the varying stress.

Figure C-3210-1 of ASME Section XI gives reference curves that determine the rate of crack growth per load cycle, da/dN , as a function of ΔK and K_{min}/K_{max} , where K is the crack tip stress intensity. This subparagraph also gives equations which describe the curves, including accounting for temperature effects. Since these curves are for air environments, the rates were multiplied by a factor of 2 to account for the difference between water and air environments [20]. These equations were used to calculate the amount of flaw growth that would take place in an SSE event.

From Section 4.2.1 above, the estimated maximum length of a circumferential crack to produce the observed leakage was determined to be 4.5 inches. This very conservative length was used in calculating the crack tip stress intensity factor. The result was that for the maximum possible crack length, da/dN is $1.03 \text{ E-}5$ in/cycle. For 50 seismic cycles, the crack would grow $5.15 \text{ E-}4$ inches, which is an insignificant amount.

A load case was also run assuming the crack would not be repaired. It was conservatively assumed that there would be 50 pressure startup cycles along with the 50 seismic cycles. For the case of pressure plus seismic cycling, da/dN was $1.43 \text{ E-}4$ in./cycle. For 50 cycles, the crack would grow .007 inches from an initial length of 4.50 inches, or about 0.1%, which is also considered to be insignificant. Thus, it was concluded that the length of the flaw will essentially not change at all.

Details of these calculations are given in Reference [23].

7.0 STRESS ANALYSIS OF THE CRACKED NOZZLE

As a very conservative way to evaluate the effect of a flaw on the stress distribution in the nozzle attachment, a finite element analysis model was developed assuming the through-wall flaw has grown completely around the pipe. It was desired to determine whether the reinforcing pad and its structural fillet welds would be able to maintain the structural integrity of the heat exchanger vessel and the attached piping.

To evaluate the acceptability of the stresses, the stress allowables given in ASME Section III, Subsection ND-3300 [8] were used. Table ND 3321-1 indicates that the allowable stresses for membrane and membrane plus bending are 2.0 S and 2.4 S, respectively, where S is the allowable stress value of the material at temperature. The Level D allowables were used for this case because we are only interested in demonstrating that the reinforcing pad welds will allow the system to maintain structural integrity, under the assumption that the nozzle to shell weld were to have a crack that extends all the way around the pipe. Although the vessel is designed to Section III, 1966, this edition does not provide allowables that are applicable for this case. As this is not a true design condition, the 1989 rules were used as a reasonable acceptance criteria.

For the nozzle neck, three through-wall stress sections were evaluated near the reinforcing pad. The first was taken at the location of maximum stress intensity in the nozzle reinforcing pad and is designated as Path A. This location corresponds to the toe of the weld on the nozzle neck. The stress allowables considered were membrane plus bending. The second was taken at the maximum stress intensity location of the nozzle neck and is designated as Path B. This is the through-wall path before the toe of the weld. The third through-wall path for the nozzle neck was taken from the root of the weld. It is designated as Path C.

Figure 7-1 shows the overall stress intensity values for the case of the through-wall, all around crack. The stress intensity results for the nozzle reinforcing pad and nozzle neck are shown in Figures 7-2 and 7-3, respectively.

The results are summarized in the table below:



Path	Membrane (ksi)		Mem + Bending (ksi)		Allowable Stress (ksi)	
	With Flaw	No Flaw	With Flaw	Without Flaw	Membrane	Mem + Bending
A	23.9	20.0	37.4	25.3	36.6	43.92
B	23.4	18.8	37.1	30.0	36.6	43.92
C	25.7	21.7	28.1	24.1	36.6	43.92

For the structural fillet weld attaching the reinforcing pad, one path through the wall section was taken along the base of the weld. The path begins at the toe of the weld on the nozzle neck and ends at the root of the weld. The stress allowable considered was membrane plus bending. This section is designated as Path D.

Path	Membrane (ksi)		Mem + Bending (ksi)		Allowable Stress (ksi)	
	With Flaw	No Flaw	With Flaw	Without Flaw	Membrane	Mem + Bending
D	23.3	19.6	28.9	24.1	36.6	43.92

For the reinforcing pad, one linearized stress path was taken through the thickness of the pad. The path begins at the toe of the weld on the pad and proceeds through the pad thickness. The stress allowable considered was the membrane plus bending. This section is designated as Path E.

Path	Membrane (ksi)		Mem + Bending (ksi)		Allowable Stress (ksi)	
	With Flaw	No Flaw	With Flaw	Without Flaw	Membrane	Mem + Bending
E	19.42	15.8	26.65	20.29	36.6	43.92

It was found that the membrane and membrane plus bending stress intensity results for the with-flaw case satisfy the Level D stress limit requirements of the ASME Code. Thus, even if the nozzle to heat exchanger shell weld were to completely rupture, the reinforcing pad and the welds attaching it to the nozzle would be able to maintain the structural integrity of the nozzle joint.

Details of these calculations are given in Reference [21].

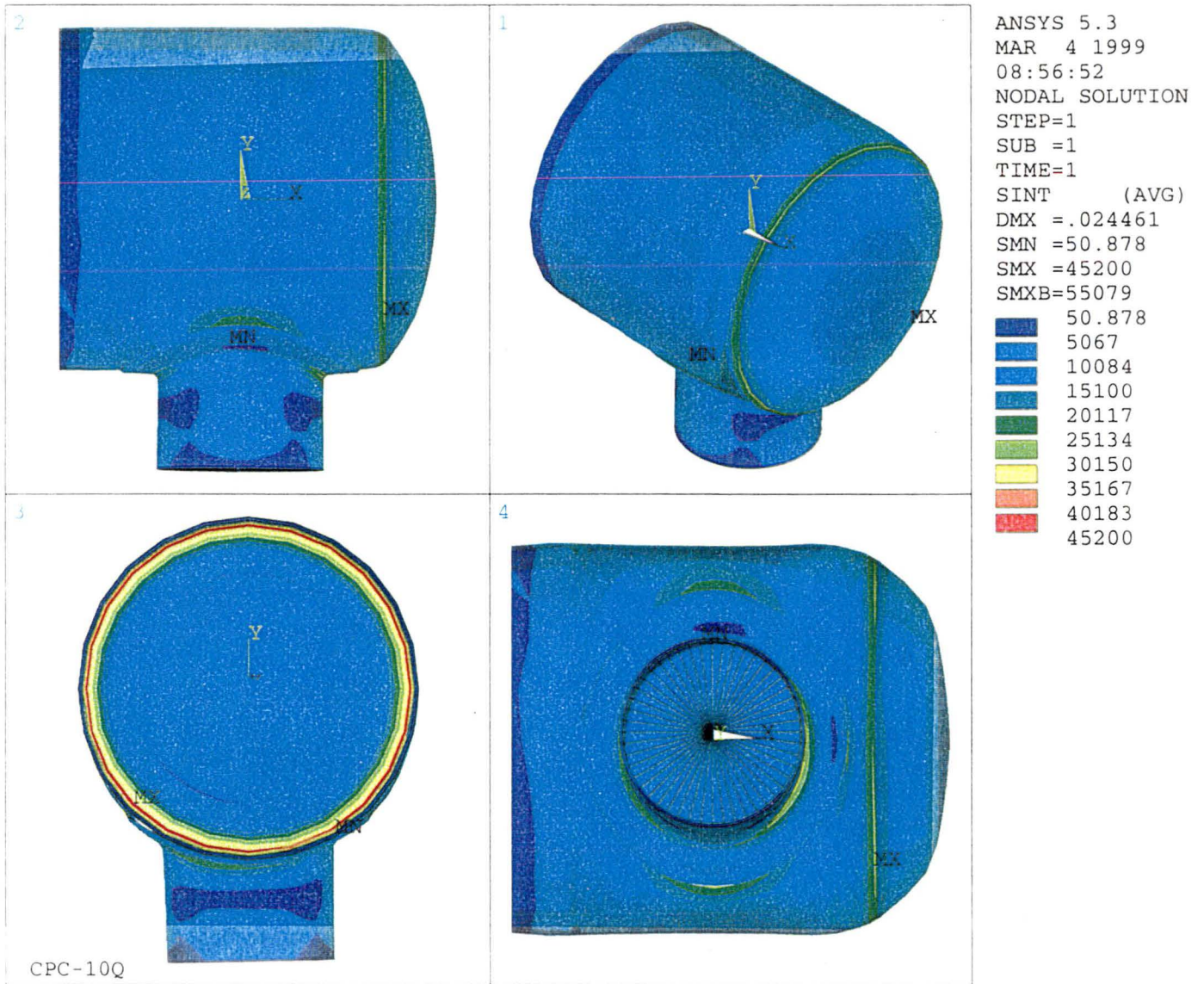


Figure 7-1. Overall Stress Intensities, All-Around Crack Model

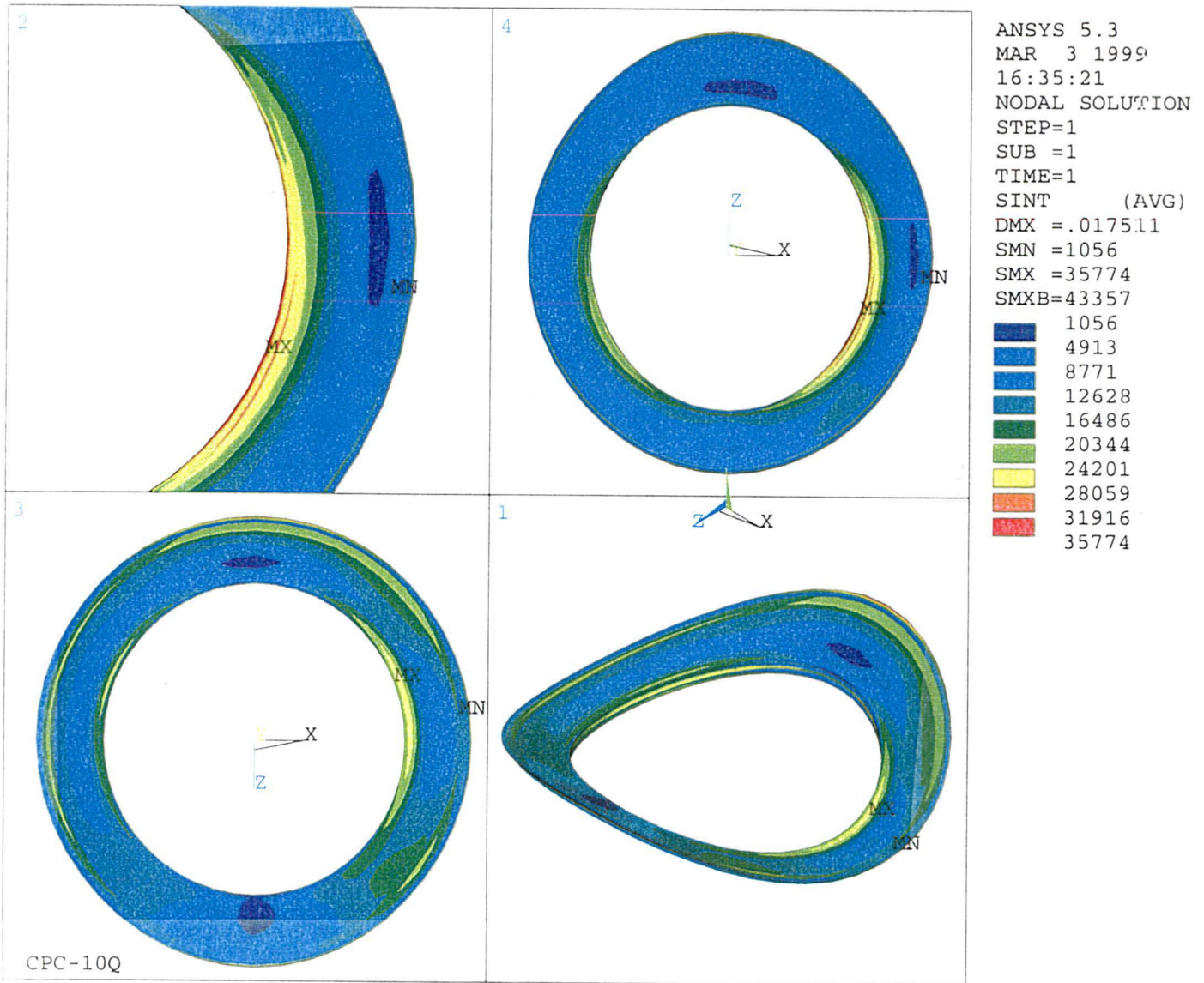


Figure 7-2. Nozzle Pad Stress Intensities

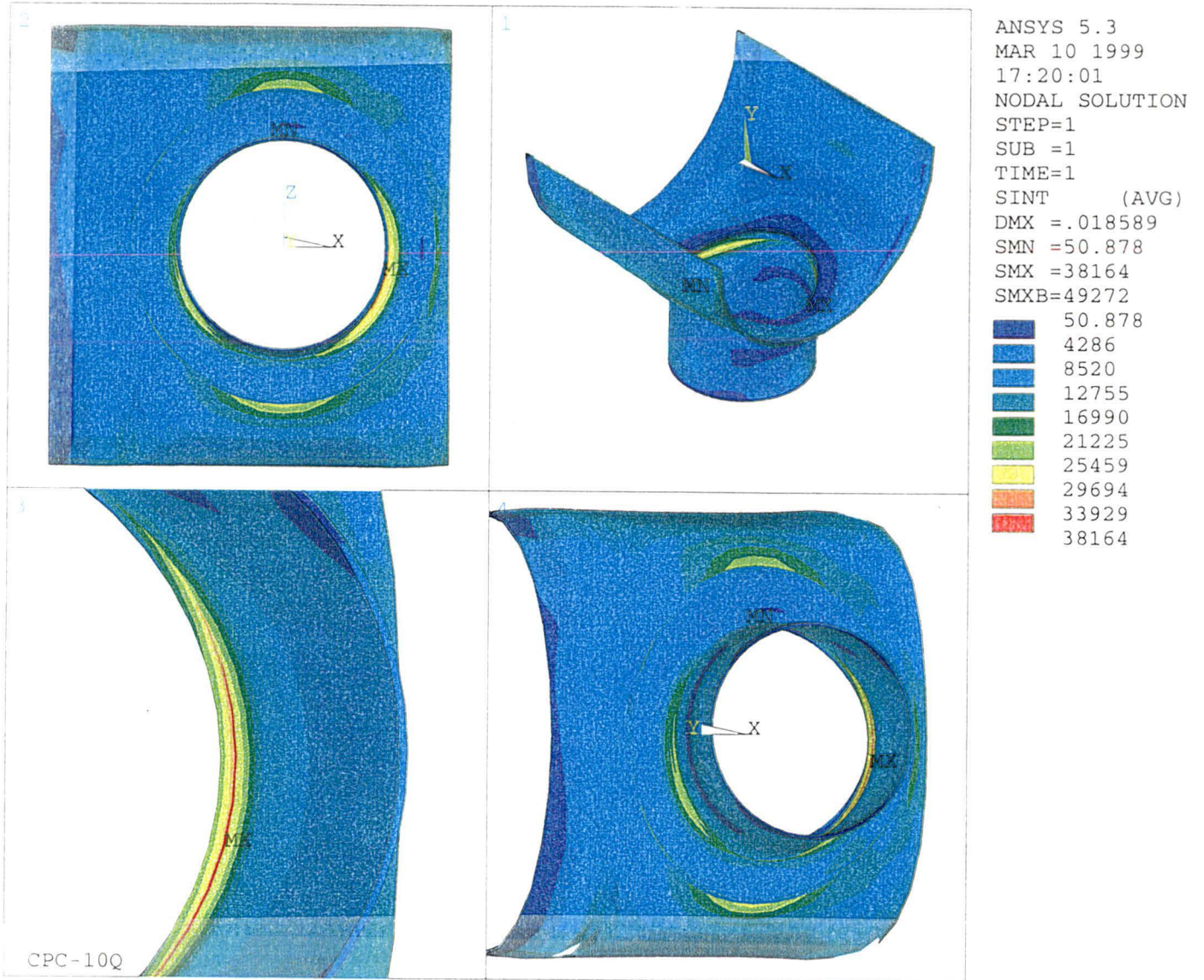


Figure 7-3. Nozzle Neck Stress Intensities

8.0 EVALUATION OF MARGINS

A comparison was made between the largest flaw that could be present, and the smallest flaw that would still meet Section XI structural criteria.

Using very conservative assumptions for the rate of evaporation, the maximum possible leak flow rate was determined. Applying this flow rate at the location with the least stress available to pull the crack open, a maximum possible crack length along the nozzle to shell weld of approximately 4.5 inches was determined.

Using a limit load approach, it was determined that a through-wall crack in the nozzle to shell double-v weld could extend 16.5 inches around the circumference, and still withstand the applied loads without rupture. This includes incorporation of the Section XI safety factors for normal / upset conditions, and does not take credit for the resistance offered by the reinforcing pad.

Applying an additional $\sqrt{2}$ safety factor on stress, the allowable circumferential flaw length, assuming the flaw is through-wall over its entirety, is reduced to approximately 13.25 inches. This is a factor of 3 larger than the maximum possible flaw size.

The calculated crack growth, assuming 50 cycles of both seismic SSE loads and pressure, was less than 1%.

The observed flow rate through the crack has been characterized as being sufficiently low such that any leakage evaporates before it can be detected. This leakage rate is probably characteristic of a much smaller leak than has been conservatively determined by analysis. There has been no observable change in the flow rate over the past several years.

Thus, it is concluded that there is significant margin between the probable through-wall crack size, and the crack size that just meets Section XI structural margins.

9.0 CONCLUSIONS

The flaw causing the leakage at the vent hole in the Palisades spent fuel pool heat exchanger E-53B inlet nozzle reinforcing pad was evaluated for acceptability for continued service. The evaluations included the following analyses:

- A finite element stress analysis of the nozzle and vessel junction region determined the stresses in the unflawed nozzle for use in flaw evaluation calculations.
- A very conservative analysis of the evaporation rate at the vent hole determined the maximum leakage flow rate that could exist without showing water accumulation.
- A calculation was done using the stresses determined in the finite element model to open a through-wall crack to obtain the calculated leakage flow rate.
- A net section plastic collapse analysis was done which determined the maximum allowable circumferential length of a through-wall crack under the applied loads.
- A flaw growth analysis was performed which showed that the crack would not grow significantly under both seismic and pressure cycling.
- A finite element stress analysis was run assuming the flaw had extended completely around the pipe. This evaluated the ability of the reinforcing pad and its welds to maintain the integrity of the nozzle to shell junction.

The result was that the maximum potential leakage-size crack that could be present in the nozzle weld was about one-third the allowable crack size, including the factor of safety for normal/upset loading conditions. This margin existed even when an additional $\sqrt{2}$ factor was applied to the stresses. The more probable existing flaw size is a pinhole sized crack that is plugged with residue from the leakage. The analysis of the nozzle weld that was assumed to be cracked completely around showed that the reinforcing pad and its structural fillet welds were capable of

maintaining the structural integrity of the joint, with all stresses meeting ASME Level D allowables.

Thus, since there is no current observation of leakage and there is significant margin between the allowable crack size and the potential size that might exist, it is concluded that continued service is acceptable.

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