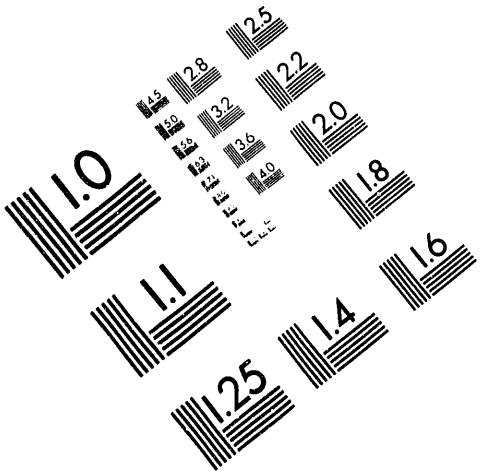

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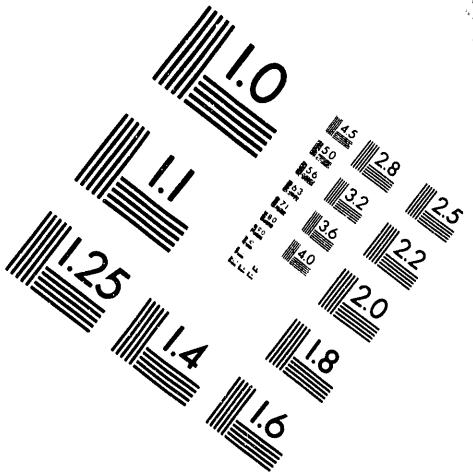

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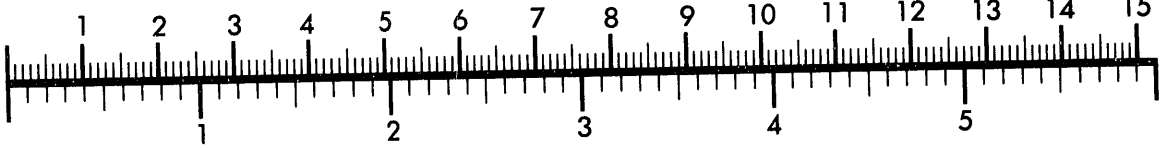

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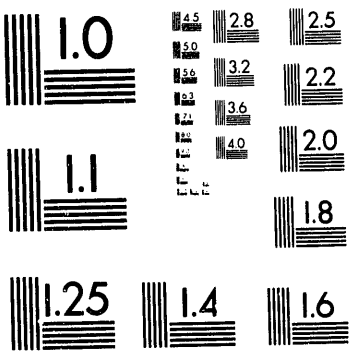

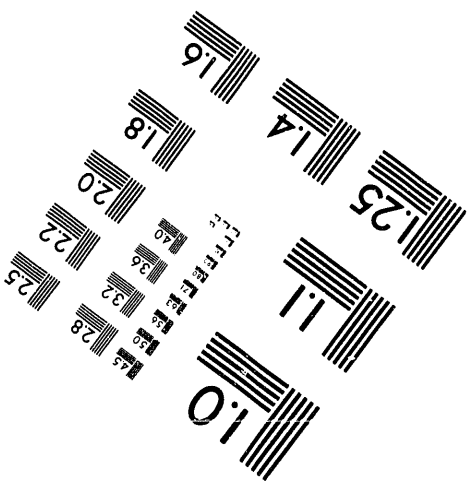

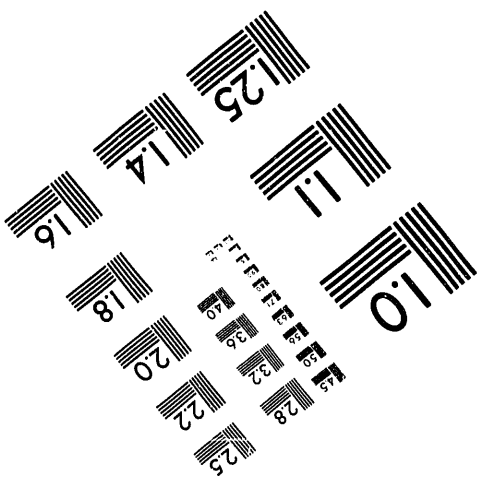

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ACCEPTANCE CRITERIA FOR REACTOR COOLANT PUMPS AND VALVES (U)

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ACCEPTANCE CRITERIA FOR REACTOR COOLANT PUMPS AND VALVES

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ABSTRACT

Each of the six primary coolant loop systems of the Savannah River Site (SRS) production reactors contains one reactor coolant pump, one pump suction side motor operated valve, and other smaller valves. The pumps are double suction, double volute, and radially split type pumps. The valves are different size shutoff and control valves rated from ANSI B16.5 construction class 150 to class 300. The reactor coolant system components, also known as the process water system (PWS), are classified as nuclear Safety Class 1 components. These components were constructed in the 1950's in accordance with the then prevailing industry practices. No uniform construction codes were used for design and analysis of these components. However, no pressure boundary failures or bolting failures have ever been recorded throughout their operating history.

Over the years, the in-service inspection (ISI) was limited to visual inspection of the pressure boundaries, and surface and volumetric examination of the pressure retaining bolts. Efforts are now underway to implement ISI requirements similar to the ASME Section XI requirements for pumps and valves.

The new ISI requirements call for volumetric examination of the pump casing and valve body welds. However, the volumetric examination of the such welds is difficult due to the limited access and high radiation exposure difficulties. Because of the uniqueness of SRS reactor design of low temperature and low pressure, fracture evaluations based on ASME code case N-481, which provides a method for relaxing these requirements, are performed. Conservative throughwall initial flaws are postulated and CEGB R6 type analyses are performed to establish flaw stability. The analyses show that detectable size throughwall flaws will remain stable at the operating conditions and volumetric examinations can be relaxed.

INTRODUCTION

The structural integrity demonstration of the SRS reactor coolant piping system components includes evaluating the structural capacity of each component against a large break or equivalent Double Ended Guillotine Break (DEGB). The essential components of the reactor coolant system include reactor vessel, reactor vessel inlet and outlet nozzles, piping, reactor coolant pumps and valves, and heat exchangers. The reactor coolant pumps and valves control the flow of primary coolant (D_2O) to the reactor vessel and its components.

The structural adequacy of pumps and valves is demonstrated by performing stress analysis and fracture mechanics analysis. Three dimensional finite element (FE) models are developed for the pumps and valves. Design pressure, maximum operating temperature, and sustained and occasional mechanical loads are applied to the models. The critical flaw lengths are calculated using Flaw Assessment Diagram (FAD) methodology. The instability lengths are calculated by applying a safety factor of three on maximum stresses.

The purpose of this work is to develop acceptance criteria for reactor coolant pump casings, pump suction cover bolts, and valve bodies should flaws be detected in the UT inspections. The analysis follows the methodologies to meet the ASME Section III (1992) requirements. The evaluation also demonstrates that engineering type fracture analysis methods could be used to determine stable flaw sizes.

Pumps

The reactor coolant pumps were manufactured by the Bingham Pump Company and were installed in the 1950's. The pumps are double volute, double suction, and radially split type pumps. The two halves of the casing are joined by a full penetration weld. These types of pumps are classified as Type D pumps in the ASME Section III Code (1992). The access to the internals of the pump is made possible by removing the suction cover which is attached to the pump casing by 24, 1.75" high strength bolts. The pumps are anchored to the pedestal by 1" thick plates welded to the pump casing and base plates. Failure of a number of suction cover bolts could result in a breach in the reactor coolant pressure boundary or a crack in the HAZ of the casing weld could also cause a large break.

This report provides an acceptance criteria methodology with the technical bases to disposition flaws reported in the inspections of the suction cover bolts and the pump casing. The fracture evaluation is based on ASME Code Case N-481

(1990) which provides exemption from UT examination of the pump casing welds.

Valves

The valves are manufactured by different vendors and include gate valves and globe valves of different designs. Fracture of bolts holding the yoke in place, or a large crack in the valve body could cause a large break.

This report provides an acceptance criteria methodology with the technical bases to disposition flaws reported in the inspections of the valve bodies. Since the material of construction for valve bodies is the same as the pump casing, guidance from Code Case N-481 (1990) is used in the fracture evaluation of postulated cracks in the valve bodies.

MATERIALS OF CONSTRUCTION

Tensile Properties

The pump casings and valve bodies are constructed of CF-8, the cast equivalent of Type 304 stainless steel. Tensile properties for Type 304 and CF-8 stainless steel in Table 1 are provided by Section II of the ASME Boiler and Pressure Vessel Code (1992). The tensile strength of material A-193 Gr B7 at 196°F is not available in the literature (ASME, 1992). Therefore, the tensile strength was calculated approximately on the basis of material yield strength values at 100°F and 196°F, and the given tensile strength at 100°F, i.e. (125/105)*98.28 = 117 ksi.

TABLE 1. MATERIAL PROPERTIES OF CF-8 AND A-193 GR B7

Material	Temp. (°F)	Design Stress Intensity S_m (ksi)	Yield Strength S_y (ksi)
CF8	70 to 100 196	20 20	30 25
A-193 Gr B7	70 to 100 196	35 32.7	105 98.28

TABLE 1 ..... continued

Material	Tensile Strength S_u (ksi)	Young's Modulus E (psi)	Flow Stress (ksi)
CF8	70 66.35	28.3×10^6 27.61×10^6	60 (†) 60
A-193 Gr B7	125 117	29.7×10^6 29.0×10^6	115 108(††)

(ASME (1992), Tables 2A, 4, Y1, U and TM1)

(†) The flow stress is given as 3 Sm .

(††) The flow stress is taken as $(S_y + S_u)/2$. However, a conservative value of 105 ksi is used in the analysis.

Fracture Toughness of CF-8

Elastic-plastic fracture toughness properties for CF-8 cast stainless steel material are given by Hiser (1988). The CF-8 material in the SRS process water system (PWS) would not be embrittled by thermal aging at the low temperature service conditions of the PWS and, therefore, the properties for unaged cast material are applied. Upper and lower bound elastic-plastic toughness (J-R curve, Modified J theory) data from unaged CF-8 material at 77°F is shown together with upper and lower bound toughness at 77°F from the archival PWS piping, Type 304 stainless steel (Stoner et al., 1990), in Figure 1. The J-R curves of the unaged CF-8 are similar to the archival Type 304 stainless steel. The database of the archival piping in the above SRS report provides a large sample of materials and provides elastic-plastic results with Deformation J theory in addition to the Modified J theory. The J deformation theory is selected to determine K_{JC} for flaw stability in pumps and valves. For flaw analysis with linear elastic fracture analysis methodology, K_{JC} is calculated from J_{IC} (Deformation J theory) using the average results from the lower bound base material component of 338 kJ/m^2 (C-L orientation at 257°F). With plane stress conditions, $J_{IC} = K_{JC}^2/E$, and in English units with $E=28 \times 10^6$ psi, K_{JC} is 230 ksi$\sqrt{in}$. In the following sections, K_{JC} is used as K_{IC}.

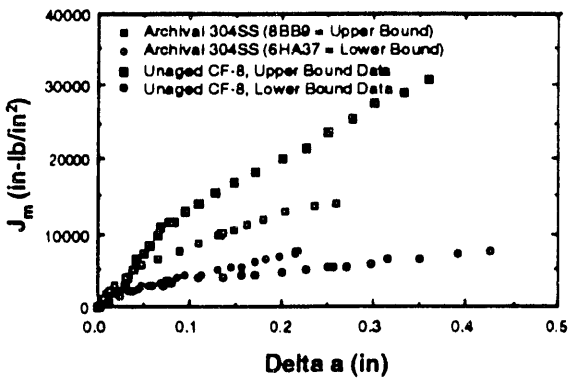

FIGURE 1. J_m FOR ARCHIVAL SS304 PIPING VS. UNAGED CF-8 CAST MATERIALS

Fracture Toughness of A-193 Gr B7 Bolts

Material A-193 Gr B7 is used for the high-strength bolting to form a leaktight joint between the suction cover and the pump casing. This material is a low alloy steel and is equivalent to AISI 1040, 1042, or 1045 steels. Fracture toughness K_{IC} is derived from Charpy V-Notch (CVN) test data at room temperature. A CVN value of 46 ft-lbs is obtained at room temperature. Since the operating temperature is higher, this is a conservative value. The following empirical relation (Eq. 4.7, Atkins & Mai, 1988) is used to estimate K_{IC}.

$$\frac{K_{IC}^2}{E} = 0.22 \times (CVN)^{1.5} \qquad (1)$$

The CVN units in the above equation are Joules and the left hand side units are kPa-m (kilo Pascals-meter). Using the relationship $K_{IC}^2/E = J_{IC}$ for Linear Elastic Fracture Mechanics (LEFM) regime in plane stress and converting to the ksi$\sqrt{in}$ units, we get K_{IC} = 134 ksi$\sqrt{in}$ and J_{IC} = 108 kJ/m^2. These values are summarized in the Table 2 below.

TABLE 2. FRACTURE TOUGHNESS OF A-193 GR B7 BOLTS

Material	Test Temperature (°F)	J_{IC}- Deformation (kJ/m^2)	K_{IC}- Deformation ($ksi\sqrt{in}$)
A-193 Gr B7	68	108	134

COMPONENT SERVICE HISTORY

Pump Service History

Each reactor coolant pump has operated over 35 years without any cracks in the pump casing. Leakage has occurred occasionally through the pump mechanical seals. Visual inspections of the pump internals have revealed wear scars, pits on wear rings, and scratches on the inside surface of the pump casings caused by tiny pieces of debris trapped between the mating surfaces. Some pits have been found on the suction cover O-ring surface.

Valve Service History

Similar to the reactor coolant pumps, the oldest valves in the system have been in service for over 35 years. No large break or leakage failures by cracking of the valve bodies or loss of bolting materials by cracking has occurred during this period. Radiographic examination upon installation was performed followed by periodic visual inspection.

LOADING CONDITIONS

Pumps

The reactor coolant pumps operate at 196°F. Suction and discharge pressures are 16 and 225 psig respectively. A design pressure of 300 psig is used in the analysis. The pumps are subjected to gravity, pressure, seismic, and nozzle loading from the suction and discharge piping. Fluid transient pressure is included in the design pressure. Thermal loads from the expansion of the pump casing itself are insignificant due to sliding provisions in the pump footings to accommodate thermal expansion of the casing.

Valves

The loading conditions considered are internal pressure, gravity, thermal, and seismic loads. For each case, the boundary conditions are fixed at one flange and free at the other. The pipe reaction loads were applied at the free flange end.

The applied internal pressure is 300 psi at 196°F, the design pressure for the valve. This value includes the potential effects of transient loading conditions. The largest applied force is the axial pressure thrust load. The applied pipe loads are relatively small due to the close proximity of the valve to an expansion joint.

FINITE ELEMENT (FE) MODEL FOR PUMP

Pumps Casings

A three dimensional finite element model of the pump casing is constructed to provide an accurate stress distribution for the fracture analysis. The model was constructed with the PATRAN 2.5 mesh generator (PDA, 1992). PATRAN outputs the node and element information in ABAQUS input format. ABAQUS is a general purpose finite element code that was used throughout this analysis (HKS, 1992). The pump FE model has 1528 shell elements (S4R5) to represent the model. There are 1536 nodes each with 5 degrees of freedom.

The pump casing is a complex cast geometry with internal flow passages and flow splitters. The internal details are not included to simplify the model. Since the flow passages and splitters act as stiffeners, the resulting stress distribution is expected to be conservative. Figure 2 shows the finite element model.

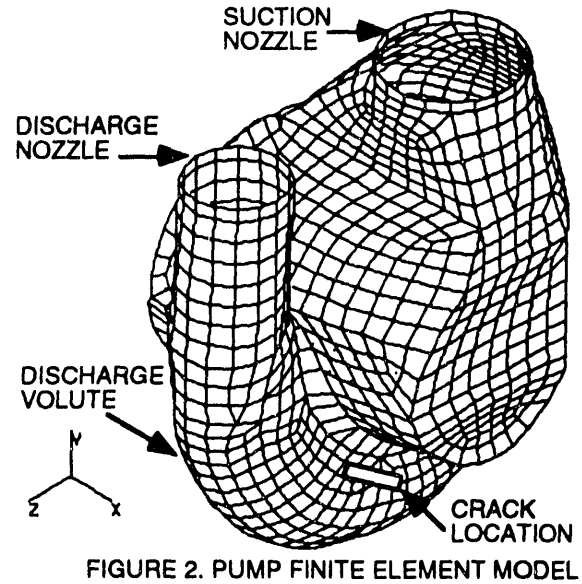

FIGURE 2. PUMP FINITE ELEMENT MODEL

Suction Cover

The FE model has 1444 shell elements (S4R5) and 24 beam elements (B31). The shell elements represent the pressure bearing surfaces and the beam elements represent the 24 bolts. There are 1511 nodes in the model.

The suction cover is also a cast component with internal flow passages to direct the flow to the inlet of the impeller. The main concern in the structural integrity evaluation of the suction cover is the effect of the bolt failures on the stability of the bolted joint between the suction cover and the rest of the pump casing. Therefore, the FE modeling was simplified and only those surfaces where high and low pressures act to transfer the load to the bolts, are modeled. Figure 3 shows the finite element model.

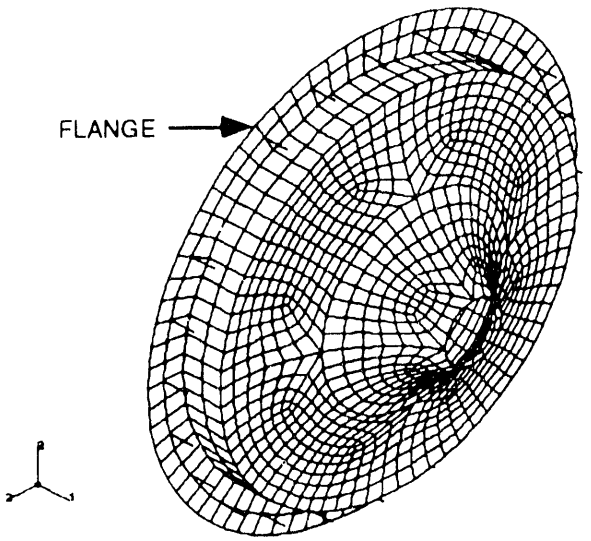

FIGURE 3. SUCTION COVER FE MODEL

FRACTURE ANALYSIS FOR PUMP

Pumps Casings

Fracture analysis is performed using Failure Assessment Diagram (FAD) methodology (Milne et al., 1988). FAD methodology covers the entire range from brittle failures to limit load failures. FAD methodology includes three categories of analyses. Category 1 analysis is based on linear elastic, plain strain fracture toughness K_{IC}. Category 2 and 3 analyses take advantage of increased material toughness due to ductile tearing for ductile materials. In general, Category 1 type of analysis gives conservative crack lengths.

The Category 1 FAD curve (Kr vs Lr for cast CF-8 SS) is based on the following relation.

$$K_{r,ss} = (1 - 0.14 \times L_r^2)[0.3 + 0.7\exp(-0.65 \times L_r^6)] \quad \text{for } L_r \le L_r^{max} \qquad (2)$$

$$K_{r,ss} = 0 \quad \text{for } L_r \ge L_r^{max} \qquad (3)$$

A value of $L_r^{max} = 1.8$ is suggested in Milne et al. (1988) for stainless steel.

K_r and L_r for the applied loading are defined as:

$$K_r = \frac{K_I}{K_{IC}} \quad \text{and} \quad L_r = \frac{\text{Applied Load}}{\text{Limit Load}} = \frac{P}{P_o}$$

Calculation of K_I. A review of maximum principal stresses in the pump casing shows that the pressure is the dominant loading. ASME Code Case N-481 (1990) recommends that a part throughwall crack should be postulated for fracture analysis. However, due to the highly localized nature of the maximum stress, a surface crack will propagate through the thickness before propagating axially. Therefore, an axial throughwall crack is postulated for the fracture analysis.

The location of high stress lies in the doubly curved region of the discharge volute and appears to lie on the crown of the region (Fig. 2). A crack is assumed to lie at this location. A review of the stresses at the chosen location of the crack shows that the stresses vary through the thickness of the casing. Such a stress distribution will drive a surface crack part through the wall until the stresses are such that the crack cannot propagate any further. However, since a throughwall crack is postulated for the analysis, only membrane stresses are used to analyze the throughwall crack stability. Table 3 below gives the maximum tensile principal stresses that are used to calculate the maximum membrane stress.

TABLE 3. MAXIMUM PRINCIPAL STRESSES

Maximum Principal Stress (outside) (ksi)	Maximum Principal Stress (inside) (ksi)	Membrane Stress (ksi)
18.132	0.074	9.103

The effect of seismic loading on the highest stress is negligible and, therefore, a safety factor of 3 for normal operating conditions is used on this stress to calculate K_r and L_r. The use of a safety factor of 3 is consistent with Subarticle IWB-3642 of the ASME Section XI Code (1992).

Figure 4 depicts the geometry of a 90° long radius elbow assumed to simulate the doubly curved region of the location of the crack. This is a reasonable configuration since the chosen location of the crack lies on the discharge volute.

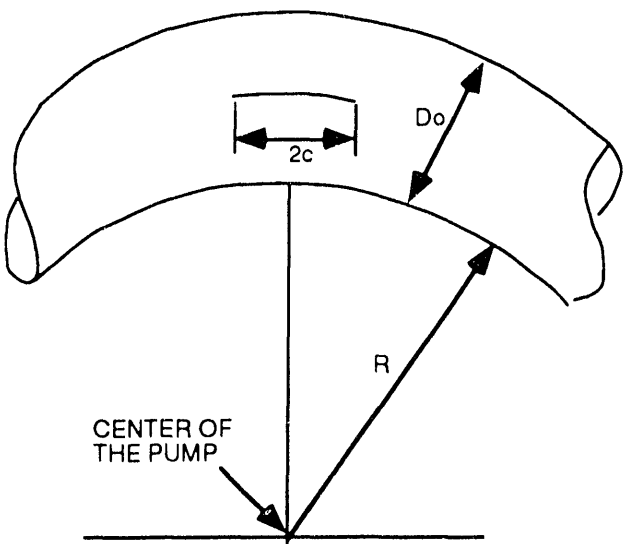

FIGURE 4. PUMP CASING CRACK MODEL

The stress intensity factor is calculated using the formula given in the EPRI Handbook (Ch. 9, Zahoor, 1991) The formula is given below:

$$K_I = \sigma \times (\pi Do)^{0.5} \times F \qquad (4)$$

where

$$F = 1.2198 \times \frac{c}{Do} \times \left(\frac{Do}{t}\right)^{0.36}$$

Do = OD of the elbow = 16.75"
t = wall thickness =1.25"
c = half length of the crack

σ = stress in ksi

The above formula for K_I is applicable for the following conditions.

1. 90-degree long-radius elbow, R/Do = 1.5
2. $0.02 < c/Do \leq 0.5$ and $15 \leq Do/t \leq 100$

Half crack length c will vary from zero to the stable crack length obtained by using FAD methodology described above. Code Case N-481 (ASME, 1990) requires that a minimum c = 6*t/4*1/2 = 0.9375" be postulated. For this case,

$$\frac{c}{Do} = \frac{0.9375}{16.75} = 0.056 \qquad \text{Within the range}$$

However,

$$\frac{Do}{t} = \frac{16.75}{1.25} = 13.4 < 15 \qquad \text{Beyond the range}$$

The Do/t value does not meet the applicability limits of the solution. However, the form factor F is proportional to Do/t and a value of Do/t = 15 will be conservative.

Calculation of L_r. A pipe configuration was used for limit load analysis since the postulated flaw location is similar to a pipe. The limit load solution is readily available for the pipe with an axial throughwall flaw (Ch. 6, EPRI, 1991). In addition, at limit load conditions the curvature effects are not significant. Furthermore, the stresses where the flaw is postulated are highly localized and, therefore, the double curvature effect will not significantly alter the conclusions. The following equations are used to calculate the limit load with an axial crack.

$$P_o = \sigma_y \frac{(t/R)}{M} \qquad (5)$$

where

P_o is the limit load for a pipe with crack size 2c

σ_y = 25 ksi is the yield stress of the material[1]

$M = (1+1.2987\lambda^2-0.026905\lambda^4+5.3549\times10^{-4}\lambda^6)^{0.5}$

$\lambda = \frac{c}{(Rt)^{0.5}}$

R = 8.375" (radius of the assumed pipe)

t = 1.25" (thickness of the assumed pipe)

The applied load P is calculated based on the stress σ (the membrane stress including a safety factor of 3), and is given as $P = \sigma \frac{t}{R}$. L_r can now be calculated from $L_r = P/P_o$.

The results of the FAD analysis for half crack sizes, c, from 0" to 3" long are shown in Figure 5. The limiting crack size is determined by the intersection point of the Category 1 FAD curve and the applied K_r and L_r curve. This occurs at K_r = 0.28. The stable flaw size can now be calculated as follows:

$$K_r = \frac{K_I}{K_{IC}} = 0.28$$

$$K_I = 0.28 * 230 = 64.4 \text{ ksi}\sqrt{\text{in}}$$

[1] Zahoor (1991) recommends using the flow stress of the material to calculate the limit load. However, the FAD formulation is based on the yield stress of the material.

Using Eq. 4 for K_I we can calculate the limiting flaw size. This gives a flaw size, 2c, equal to 3.5". This is a conservative estimate and a higher critical flaw size will be obtained if FAD Category 2 or 3 (Milne et al., 1988) analyses, which take into consideration increased fracture toughness due to ductile tearing, are used.

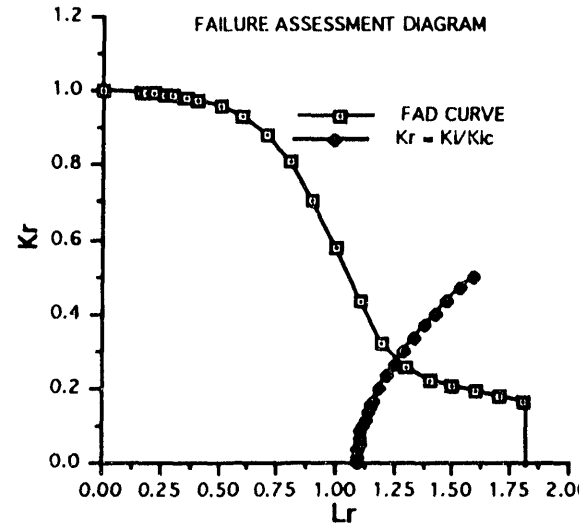

FIGURE 5. PUMP CASING FRACTURE ANALYSIS

Suction Cover Bolts

There are 24, 1-3/4" high strength bolts to hold the suction cover and pump casing together. The main purpose of the suction cover analysis was to assess the integrity of these bolts. The scenario examined here is the cracking of one or more bolts. The following analyses were performed to evaluate the bolt integrity.

1. With all bolts intact, maximum bolt deflections were obtained for high pressures ranging from 300 psig to 900 psig.

2. With all bolts cracked with a crack 25% deep through the minor diameter, maximum deflections were obtained for high pressures ranging from 300 psig to 900 psig. This analysis will show that even if all the bolts have flaws, the bolt stresses are well below the yield stress and, therefore, will not fail for the design pressure.

3. Next, the cracked bolts were removed in succession. This was done by using the CHANGE MODEL option in ABAQUS. First, two highly loaded bolts were removed. (The suction cover model is not symmetric all around since the outboard bearing is bolted to the lower half of the suction cover.) Then using the RESTART and CHANGE MODEL options in ABAQUS, the next two bolts were removed. A total of 8 bolts were removed in this manner to see if bolts will experience any yielding. The loss of 8 bolts (out of 24) will cause sufficient movement of the suction cover so as to cause a D_2O leak and the subsequent detection by the tritium monitoring equipment. However, the O-ring between the suction cover and pump casing may prevent any significant leakage.

4. A fracture analysis using the Crack Tip Opening Displacement (CTOD) criterion was performed for the bolts. This analysis determines the number of cracked bolts that could be missing at various pressures without any additional bolt failures.

Figure 6 shows the cracked bolt model for fracture analysis. Since the most likely location of a crack is the minor diameter of the bolt thread, a crack 25% deep through the minor diameter was assumed. In that case, the remaining cross sectional area A_r (Fig. 6) is given as ,

$$A_r = R^2(\pi - \cos^{-1}\frac{R-a}{R}) + (R-a)\sqrt{R^2-(R-a)^2} \qquad (6)$$

Where ,

R = minor radius of the 1.75" bolt (Mark, 1964) = 0.7523"

a = depth of the crack =0.25x2R = 0.37615"

Therefore,

$A_r = 1.43 \text{ in}^2$

The effective radius for the cracked bolt analysis is,

$$\sqrt{\frac{1.43}{\pi}} = 0.6748 \text{ in.}$$

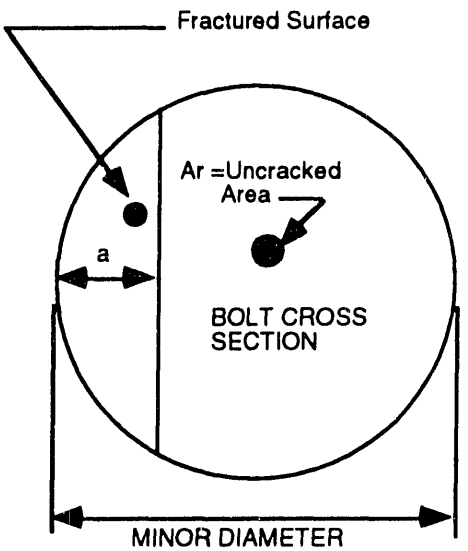

FIGURE 6. BOLT CRACK MODEL

Fracture analysis of the bolts is based on the CTOD criterion. A "strip yield model" or Dugdale crack model leads to $J=\sigma_f\delta_t$, where J is the fracture parameter J-integral, δ_t is the crack tip opening displacement, and σ_f is the flow stress used for the bolt material. The fracture condition $J = J_{IC}$ is met when $\delta_t = \delta_t^{critical}$ = CTOD. In the section on materials, J_{IC} is given as 108 kJ/m^2 or 618.0 lb/in. Therefore, the critical crack tip opening displacement (CTOD) is $\frac{J_{IC}}{\sigma_f} = 0.006"$ for σ_f = 105 ksi. Assuming that the bolts are rigid up to the point of plastic collapse, the critical bolt extension will be equal to CTOD.

Figure 7 gives the bolt deflections for pressures ranging from 0 psig to 900 psig. The deflections include the initial extension (0.00128") due to pretension. The bolt deflections are given for all the cases described in the beginning of this section. The inset in Figure 7 is an exploded view for small bolt deflections. From the pressure vs. bolt deflection curves in Figure 7, it can be seen that the critical pressure level corresponding to this critical displacement, CTOD, is approximately 270 psig for the bolt configuration with zero, two, four, and six inactive bolts. Conversely, when all bolts contain 25% deep flaws, the pump can be safely operated at a maximum operating pressure of 270 psig for the cover with 0, 2, 4, or 6 nonfunctional bolts (without fracturing additional bolts). However, a maximum stress check based on ASME Code (1992) limits the bolt failures to four.

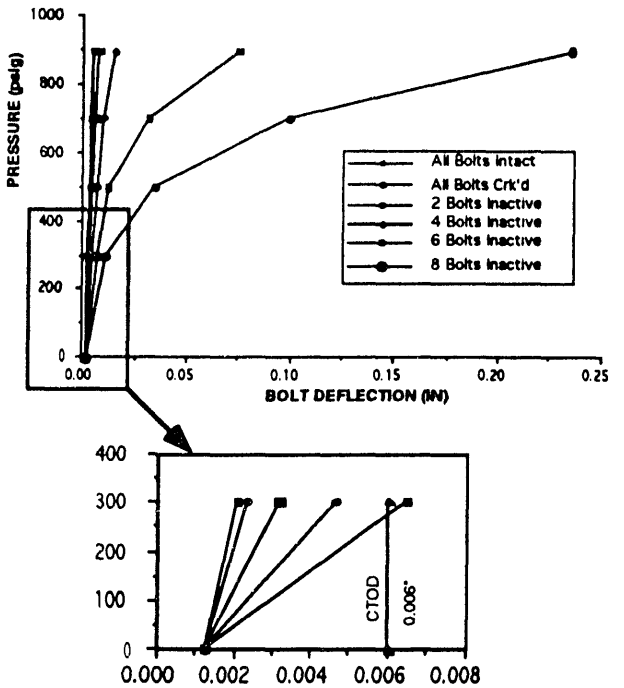

FIGURE 7. BOLT FRACTURE ANALYSIS

FINITE ELEMENT MODEL FOR VALVES

A detailed finite element analysis of the selected valve was performed to provide accurate stresses for the failure assessment.

The model is a full three dimensional structure composed of 2,424 second order, eight-noded shell elements (S8R5) that are recommended for use where the response is linear-elastic such as this case. There are 7,455 nodes, each with 5 degrees-of-freedom.

Only the pressure boundary portions of the valve bodies are modeled. The valve has considerable internal structure that serves to stiffen the body, thereby reducing the primary stress level for a given load. By neglecting the internal structure, higher stresses are predicted in the finite element analysis making this assumption conservative. Figure 8 depicts the valve finite element model.

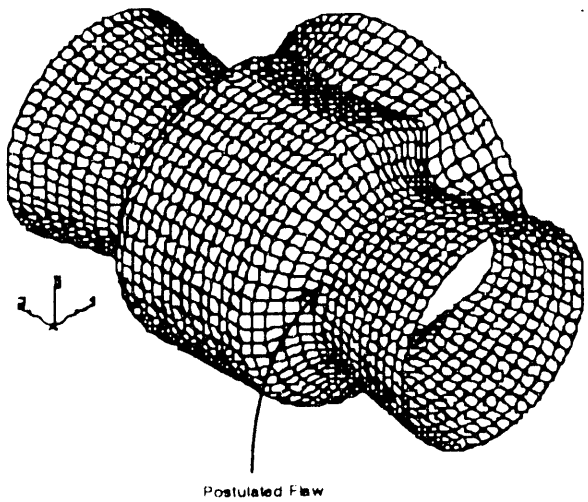

FIGURE 8. VALVE FINITE ELEMENT MODEL

Mode-I cracks are driven by tensile stresses. A maximum principle stress of 7.7 ksi is present in the fillet area near the pipe/valve flange. The results showed that the stresses due to pipe reactions - including seismic - are secondary compared to internal pressure. This is due mostly to the valves close proximity to an expansion joint that essentially does not allow the transmission of moments from the pipe. The maximum principal stress was chosen because it represents a bounding, tensile stress that could drive a Mode I crack.

FRACTURE ANALYSIS FOR VALVES

Calculation of K_r

The FAD requires that the applied stress intensity factor, K_I, be determined so that K_r can be calculated. Handbook solutions for K_I are not available for the complex geometry and loading of a valve body. Simplifying assumptions about the geometry and loading conditions must be made. The results of the stress analysis indicated that the worst-case flaw, based on the highest tensile stress level, is a circumferential crack near the intersection of the pipe-like portion of the valve and the major portion of the valve body as depicted in Figure 8. The stress was assumed to be either due solely to axial tension or to pure bending. Solutions to both cases were calculated to provide a worst case acceptable flaw length.

Assuming this is similar to a circumferential crack in a pipe as depicted in Figure 9, K_I solutions are available for different loading conditions. Equation 7 is the K_I solution for a circumferential, throughwall crack due to axial tension (EPRI, 1991).

The irregular valve geometry and loading conditions produced combined, unsymmetric stresses including throughwall bending. For determining K_r, the maximum principal tensile stress was assumed to be constant throughwall and all the way around the valve at the point where it occurred in the finite element analysis as if it were due to axial tension alone. This is a very conservative assumption.

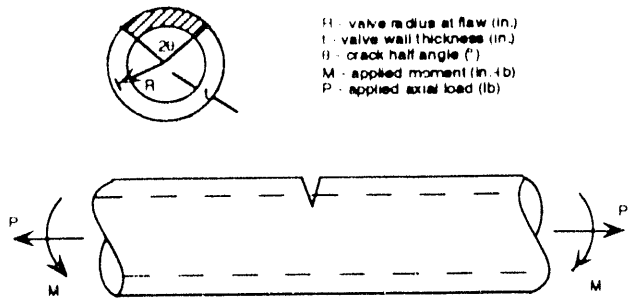

FIGURE 9. GEOMETRY ASSUMED FOR FAILURE ASSESSMENT

$$K_I = \sigma_t[\pi R\theta]^{0.5} \cdot F_t \qquad (7)$$

$$\text{Where:} \quad F_t = 1 + A\left[5.3303\left(\frac{\theta}{\pi}\right)^{1.5} + 18.773\left(\frac{\theta}{\pi}\right)^{4.24}\right]$$

$$A = \left[0.125\left(\frac{R}{t}\right) - 0.25\right]^{0.25} \qquad \text{for } 5 \le \frac{R}{t} \le 10$$

$$A = \left[0.4\left(\frac{R}{t}\right) - 3.0\right]^{0.25} \qquad \text{for } 10 \le \frac{R}{t} \le 20$$

K_I - stress intensity factor (ksi$\sqrt{\text{in}}$)
σ_t - applied maximum tensile stress (ksi)
R - pipe mean radius (in)
θ - crack half angle (radians)
t - pipe wall thickness (in)

$$\text{Applicability:} \quad 0 < \frac{\theta}{\pi} \le 0.55 \quad \text{and} \quad 5 \le \frac{R}{t} \le 20$$

Another solution is available for K_I due to bending moment. Equation (8) is the K_I solution for a circumferential, throughwall crack due to bending moment (EPRI, 1991).

$$K_I = \sigma_b[\pi R\theta]^{0.5} \cdot F_b \qquad (8)$$

$$\text{Where:} \quad F_b = 1 + A\left[4.5967\left(\frac{\theta}{\pi}\right)^{1.5} + 2.6422\left(\frac{\theta}{\pi}\right)^{4.24}\right]$$

$$A = \left[0.125\left(\frac{R}{t}\right) - 0.25\right]^{0.25} \qquad \text{for } 5 \le \frac{R}{t} \le 10$$

$$A = \left[0.4\left(\frac{R}{t}\right) - 3.0\right]^{0.25} \qquad \text{for } 10 \le \frac{R}{t} \le 20$$

K_I - stress intensity factor (ksi$\sqrt{\text{in}}$)
σ_b - maximum bending stress (ksi)
R - pipe mean radius (in)
θ - crack half angle (radians)
t - pipe wall thickness (in)

$$\text{Applicability:} \quad 0 < \frac{\theta}{\pi} \le 0.55 \quad \text{and} \quad 5 \le \frac{R}{t} \le 20$$

Both of these equations are valid for crack half angles of 100° or less. Equations (7) and (8) are solved for K_I, the

applied stress intensity factor. This value is divided by K_{IC} to determine K_r for use with the FAD. K_{IC} is equivalent to K_{JC}, the fracture toughness value for CF-8 material described above.

Limit Load Solutions

The corresponding limit load (plastic collapse) solutions for a pipe containing a circumferential, throughwall crack are described below (EPRI, 1991). Equation (9) defines the limit load stress for axial tension (see Figure 9).

$$\sigma_l = \frac{\sigma_f(2\alpha - \theta)}{\pi} \qquad (9)$$

Where: $\alpha = \cos^{-1}(0.5\sin\theta)$

σ_l - limit load stress (ksi)
σ_f - flow stress (ksi)
θ - crack half angle (radians)

Equation (10) represents the limit load for a thick-walled pipe containing a throughwall, circumferential crack subjected to a bending moment (Zahoor,1991).

$$M_l = 4\sigma_f R_o^2 t\left(1 - \zeta + \frac{\zeta^2}{3}\right)\left(\cos\alpha - 0.5\sin\theta\right) \qquad (10)$$

Where: $\zeta = \frac{t}{R_o}$

$$\alpha = 0.5\theta(1-\zeta)\frac{\left[1 + \frac{0.5\zeta}{(1-\zeta)}\right]}{(1 - 0.5\zeta)}$$

$$M_l = \frac{\sigma_l \pi (R_o^4 - (R_o - t)^4)}{4R_o}$$

R_o - pipe outer radius (in)
σ_f - flow stress (ksi)
t - pipe wall thickness (in)
θ - crack half angle (radians)
σ_l - limit load stress (ksi)

Applicability: $\frac{t}{R} > 0.1$

The limit load equations are solved for σ_l. The applied maximum membrane stress is divided by σ_l to determine L_r, the limit load ratio for use with the FAD. The membrane stress (mean stress through the thickness) is used rather than the maximum stress (acting on the outside fiber of the pipe) as with the K solution. The through-thickness bending stress is due to local geometry changes and is not included in the limit load calculation.

The appropriate factor of safety is applied to the stresses before using the stresses in the failure assessment. The normal operating condition stresses, when multiplied by the safety factor of 3.0, were higher and, therefore, more limiting than the seismic stresses multiplied by the safety factor of 1.5.

The physical interpretation of the FAD shows that failure by unstable crack propagation primarily will intercept the failure line on the upper left portion of the curve. Failure by plastic collapse will be indicated by the intersection occurring on the right portion of the diagram.

Figure 10 shows a FAD for a zero to ten-inch, circumferential, throughwall crack assuming the stress is due solely to axial tension. Another solution assuming the stress was due to pure bending yielded similar results. Ten inches is the limit of applicability of the K solution for this geometry. A ten-inch crack is shown to be stable for a 7.7 ksi load. The analysis includes a safety factor of three on stress.

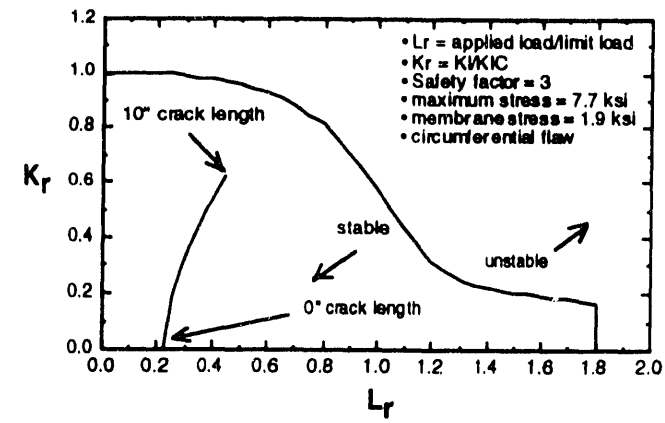

FIGURE 10. VALVE FAILURE ASSESSMENT DIAGRAM

ACCEPTANCE CRITERIA

Pump Casing

The fracture analysis shows that a throughwall 3.5" long crack will remain stable. However, to prevent any reactor coolant leakage, the flaw depth should be limited to 75% of the casing wall. This is consistent with subarticle IWB-3641 of the ASME Section XI Code (1992).

Pump Suction Cover Bolts

The bolt fracture analysis shows that cracks 25% deep through the minor diameter of all the bolts are acceptable.

Valve Bodies

The fracture analysis shows that a throughwall 10" long crack will remain stable. However, to prevent any reactor coolant leakage, the flaw depth should be limited to 75% of the casing wall. This is consistent with subarticle IWB-3641 of the ASME Section XI Code (1992).

CONCLUSIONS

The fracture analyses described above meet the requirements of ASME Code Case N-481 (1990). Therefore, the volumetric examination requirements for the pump casing welds can be relaxed. The analysis also shows that the structural integrity of the suction cover/pump casing bolted joint will be maintained even if four bolts are inactive and the remaining bolts have 25% deep cracks through the minor diameter of the bolts.

The fracture analyses for the valve bodies also meet the requirements of ASME Code Case N-481 (1990). Therefore, the volumetric examination requirements for the valve body welds can be relaxed.

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