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SUBJECT: Forwards info requested by NRC 950609 ltr to support NRC approval of util evaluation of flaw indications & subsequent ISI plans submitted by util 950302 ltr.

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FPL

P.O. Box 128, Ft. Pierce, FL 34954-0128

July 7, 1995

L-95-185
10 CFR 50.4
10 CFR 50.55a

U. S. Nuclear Regulatory Commission
Attn: Document Control Desk
Washington, D. C. 20555

RE: St. Lucie Unit 2
Docket No. 50-389
In-Service-Inspection Plan
Second Ten-Year Interval
Request for Additional Information
Fracture Mechanics Evaluation
Pressurizer Instrument Nozzles - Response

By letter dated June 9, 1995, NRC requested additional information (RAI) to support the Staff's approval of the Florida Power and Light Company (FPL) evaluation of flaw indications and subsequent in-service inspection (ISI) plans submitted by FPL letter L-95-74 on March 2, 1995. FPL requested the approval by July 1, 1995, to support planning for the Fall 1995 Unit 2 refueling outage. Your letter requested a response to the RAI within 30 days of receipt (June 15, 1995).

The initial flaw evaluation, prepared according to ASME Section XI, Appendix A, justified operation for at least one cycle. It was submitted to the NRC, the authority having jurisdiction, by FPL letter L-94-72 on March 28, 1994. NRC approval was documented in your letters dated April 1, 1994, and April 5, 1994.

The information you requested is attached. Please contact us should you need any additional clarifications.

Very truly yours,

C L Burton for
D. A. Sager
Vice President
St. Lucie Plant

DAS/GRM

cc: Stewart D. Ebneter, Regional Administrator, Region II, USNRC
Senior Resident Inspector, USNRC, St. Lucie Plant

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St. Lucie Unit 2
Docket No. 50-389
In-Service-Inspection Plan
Second Ten-Year Interval
Request for Additional Information
Fracture Mechanics Evaluation
Pressurizer Instrument Nozzles - Response

ATTACHMENT

FPL Letter L-95-74 Attachment A Questions

Attachment A:	Fracture Mechanics Evaluation of St. Lucie Pressurizer Instrument Nozzles, Proprietary Version, BWNT Document 32-1235128-00.

NRC Question 1

1.1 Assumptions, Page 4 - The upper shelf value for K_{IR} assumed for the nozzle material exceeds the limit of 200 ksi$\sqrt{in}$ for the K_{IR} vs. (T- RT_{NDT}) curve. Very few data test data exist beyond that limit. Revise the evaluation by using the value of 200 ksi$\sqrt{in}$, or provide data to justify the K_{IR} value used in the submittal.

FPL Response

This evaluation utilized an upper shelf fracture toughness value of 250 ksi$\sqrt{in}$ for the SA-533 Grade B, Class 1 material. Some of the industry data supports this upper shelf toughness value for ductile materials, however, these nozzles were re-evaluated using 200 ksi$\sqrt{in}$ value since Charpy upper shelf energy value for this particular SA 533 material could not be determined. The results of the re-analysis using the upper shelf toughness value of 200 ksi$\sqrt{in}$ indicate that all the instrument nozzles in the spherical shell are acceptable by ASME Section XI code rules.

FPL Letter L-95-74 Attachment B Questions

Attachment B:	Stresses for St. Lucie Unit 2 Pressurizer LEFM, Proprietary Version, BWNT Document 32-1235127-00.

NRC Question 1

Assumptions, page 4 - Provide an estimation of the effect of thermal stratification.

FPL Response

Delete Assumption #5 on page 4 of the subject document (therefore, will not be contained in Revision 02 of the document).

NRC Question 2

Model Geometry, page 8 - To cover the stress concentration caused by nozzle insertion into the pressurizer wall, the radius of the modeled head was set to 2.5 times the actual radius. Revise the computer simulation by using models with the actual radius, then applying the concentration factors, or demonstrate that the thermal stresses due to transients would increase in proportion to the radius of the pressurizer head as the case for internal pressure.

FPL Response

(i) Stress Concentrations

For radial nozzle penetrations in spherical and cylindrical shells of large radius which are subjected to a through-wall temperature gradient, the appropriate stress concentration factor (SCF) for the penetration is that of a flat plate with a circular hole subjected to a biaxial stress state (SCF=2.0). As noted in paragraph (ii), the thermal stresses in the modeled sphere are not significantly affected by the radius. Therefore, modeling a cylinder of radius R with a radial nozzle penetration as a sphere of radius 2.5R (to account for the 2.5 SCF on hoop stresses due to internal pressure loading) results in accurate maximum pressure and thermal hoop stresses at the penetration.

For nonradial nozzle penetrations in spherical shells, the stress concentration on hoop stress at the nozzle penetration for internal pressure loading is given by the following equation from NB-3338.2 of Reference [4]

$$K_2=K_1(1+2\sin^2\phi)$$

where:
K_1 - inside stress index for radial nozzle penetration, $K_1 = 2$
K_2 - inside stress index for nonradial penetration
ϕ - angle between the nonradial penetration and the shell normal

The stress concentration on hoop stress at a nonradial nozzle penetration in a sphere (of large radius) subjected to a through-wall temperature gradient is that of an elliptical hole in a flat plate subjected to a biaxial stress state. Therefore, modeling a sphere of radius R with a nonradial nozzle penetration as a sphere of radius $(1+2\sin^2\phi)R$ with a radial nozzle penetration results in accurate maximum hoop stresses for internal pressure loading. However, the maximum thermal hoop stresses due to a through-wall temperature gradient in the subject model will only reflect an SCF of 2.0 for the radial nozzle penetration. Therefore, Revision 02 of the subject document will include an additional 1.2 factor on thermal hoop stresses to fully account for the appropriate stress concentration (SCF=2.4) at the nonradial penetration. The 2.4 SCF is based on Figure 133 in Reference [5] for stress concentrations at an elliptical hole (with major diameter/minor diameter ratio = 1.2 for the subject St. Lucie nozzle penetrations) in a flat plate subjected to a biaxial stress state.

(ii) Effect of Radius on Thermal Stresses

For transient conditions with thermal gradients in the shell, the thermal stresses in the shell are primarily a function of the thickness of the shell and not the radius of the shell. Therefore, modeling the shell radius as the actual radius or 2.5 times the actual radius will result in essentially identical thermal stresses. The following example calculations for steady state heat flow in a sphere (for application to the spherical heads of the St. Lucie Unit 2 pressurizer) from pages 924 and 925 of Reference [2] show that the radius has negligible effect ($\Delta=3\%$) on the thermal stresses. The hoop stress is given by equation:

$$\sigma_h = \frac{\alpha E \Delta t}{2(1-\nu)}\left(1+\frac{2m}{3}\right)$$

where: α - coefficient of thermal expansion
E - Young's Modulus
Δt - the temperature difference across the thickness
ν - Poisson's ratio
m - (Ro/Ri) - 1
Ro - outside radius
Ri - inside radius

For the actual radii of Ri=48.4375" and Ro=52.3125"

$$\sigma_h = \frac{\alpha E \Delta t}{2(1-\nu)}\left(1+\frac{2(52.3125/48.4375-1)}{3}\right) = 1.053\left(\frac{\alpha E \Delta t}{2(1-\nu)}\right)$$

For 2.5 times the actual radii of Ri=121.094" and Ro=124.969"

$$\sigma_h = \frac{\alpha E \Delta t}{2(1-\nu)}\left(1+\frac{2(124.969/121.094-1)}{3}\right) = 1.021\left(\frac{\alpha E \Delta t}{2(1-\nu)}\right)$$

NRC Question 3

Finite Element Model, Pages 11 and 29 - The boundary conditions for the finite element method (FEM) model shown in Figure 6.2 are only true for an airborne sphere subjected to radially dependent structural or thermal load. Demonstrate that your simplified partial FEM model is adequate for the analyses conducted in this evaluation.

FPL Response

Zero displacement in the tangential direction (theta) of the cylindrical coordinate system is specified as the boundary condition of the partial axisymmetric model of the sphere as shown on page 29 of the subject document. This restraint in the hoop direction (remote from the hole

at the centerline of the axisymmetric model) prevents rigid body motion of the model. Note that the axis of the nozzle penetration is the axis of symmetry in the axisymmetric model.

Since the instrumentation nozzles are small compared to the vessel shell, the forces and moments imparted by the nozzle on the shell are insignificant and do not significantly influence the stresses in the shell or the amount of the structure that is required to be modeled. The only characteristic of the structure that affects the amount of the structure that is modeled is the hole in the shell at the nozzle penetration. A sufficient portion of the structure was modeled to attenuate the stress concentration effects at the nozzle penetration as evident in the model verification of hoop stress remote from the hole on page 38 of the subject document. The stresses 3" from the edge of the hole (4.3"sin45°) are within 1.6% of theoretical values for stresses in a sphere (without a hole) due to internal pressure and the total arc length of the model is approximately 10.5" ($R\theta = 120.5"(5°)(\pi/180°)$).

Therefore, the portion of the structure that was modeled and the boundary conditions used in the analysis are appropriate to yield accurate results for the stresses in the subject pressurizer shell.

NRC Question 4

Thermal Analysis, Pages 12 - The heat transfer coefficient (h) used for the reactor vessel is 471 Btu/Hr-Ft^2-°F (SECY-82-465); the coefficient for feed water nozzle is 2000 Btu/Hr-Ft^2-°F (NUREG-0619); and the number used in your first evaluation is 10,000 Btu/Hr-Ft^2-°F. Revise the evaluation by using the test supported value of 2,000 Btu/Hr-Ft^2-°F, or provide data to justify the heat transfer coefficient used in the submittal.

FPL Response

Heat transfer coefficients vary greatly depending on fluid parameters such as fluid type and the mode of heat transfer. Typical values of heat transfer coefficients listed in Table 2-2 of Reference [3] are shown in the following table:

Typical Heat Transfer Coefficients

Condition		h (Btu/Hr-Ft^2-°F)
Free Convection	Gases	1-5
	Water	20-150
Forced Convection	Water	50-2000
Phase Change	Boiling Liquids	500-10,000
	Condensing Vapors	1000-20,000

From the discussion in Section 5.0 of the subject document, the heat transfer coefficient used for the thermal analyses (461 Btu/Hr-Ft^2-°F) is realistic or conservative for all nozzle locations and transients evaluated with the exception of the water space nozzles during the 53°F step-down transient and the loss of secondary pressure transient. For these fast cooling transients, a heat transfer coefficient representative of boiling should be used to conservatively determine stresses. Therefore, a heat transfer coefficient of 10,000 Btu/Hr-Ft^2-°F will be used in Revision 02 of the subject document to conservatively determine the thermal gradients and resulting stresses for these fast cooling transients. For all other transients, the heat transfer coefficient of 461 Btu/Hr-Ft^2-°F discussed above is appropriate and will be used in Revision 02 of the subject document.

References

1) BWNT 32-1235128-00, "FM Analysis of St. Lucie Pressurizer Instrument Nozzles."
2) Faupel, Joseph H., "Engineering Design, A Synthesis of Stress Analysis and Materials Engineering," Second Edition, Wiley-Interscience, New York, 1981.
3) Arpaci, Vedat S., "Conduction Heat Transfer," Addison-Wesley Publishing Co., 1966.
4) Section III, Subsection NB of the ASME Boiler & Pressure Vessel Code, 1986 Edition.
5) Peterson, R. E., "Stress Concentration Factors," Wiley-Interscience, New York, 1974.