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F* TUBE PLUGGING CRITERION FOR TUBES WITH DEGRADATION IN THE TUBESHEET ROLL EXPANSION REGION OF THE DONALD C. COOK UNIT 1 STEAM GENERATORS

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ABSTRACT

An evaluation was performed to develop a plugging criterion, known as the F* criterion, for determining whether or not repairing or plugging of partial depth hardroll expanded steam generator tubes is necessary for degradation that has been detected in the expanded region of the tube located within the tubesheet. The evaluation consisted of analysis and testing programs aimed at quantifying the residual radial preload of Westinghouse 51 Series steam generator tubes hardrolled into the tubesheet. An analysis was performed to determine the length of hardroll engagement required to resist tube pullout forces during normal and faulted plant operation. It was postulated that the radial preload would be sufficient to significantly restrict leakage during normal and operating conditions. On this basis, an F* criterion value of 1.11 inch was established for the Donald C. Cook Unit 1 steam generators as sufficient for continued plant operation regardless of the extent of tube degradation below F*. The evaluation also demonstrates that application of the F* criterion for tube degradation within the tubesheet affords a level of plant protection commensurate with that provided by RG 1.121 for degradation located outside of the tubesheet region.

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1.0 INTRODUCTION

The purpose of this report is to document the development of a criterion to be used in determining whether or not repairing or plugging of partial depth hardroll expanded steam generator tubes is necessary for degradation which has been detected in the expanded region of the tube within the tubesheet. Existing D. C. Cook Unit 1 Technical Specification tube repairing/plugging criteria apply throughout the tube length, but do not take into account the reinforcing effect of the tubesheet on the external surface of the tube. The presence of the tubesheet will constrain the tube and will complement its integrity in that region by essentially precluding tube deformation beyond its expanded outside diameter. The resistance to both tube rupture and tube collapse is significantly strengthened by the tubesheet. In addition, the proximity of the tubesheet significantly affects the leak behavior of through wall tube cracks in this region, i.e., no significant leakage relative to plant technical specification allowables is to be expected.

This evaluation forms the basis for the development of a criterion which obviates the need to repair a tube (by sleeving) or to remove a tube from service (by plugging) due to detection of indications, e.g., by eddy current testing (ECT), in the expanded region of the tube within the tubesheet. This evaluation applies to the D. C. Cook Unit 1 Westinghouse Series 51 steam generators and assesses the integrity of the tube bundle, for tube ECT indications occurring in the roll expanded length of tubing within the tubesheet, relative to:

1) Maintenance of tube integrity for all loadings associated with normal plant conditions, including startup, operation in power range, hot standby and cooldown, as well as all anticipated transients.

2) Maintenance of tube integrity under postulated limiting conditions of primary to secondary (feedline break) and secondary to primary (LOCA) differential pressure,

3) Limitation of primary to secondary leakage consistent with accident analysis assumptions.

The result of the evaluation is the identification of a distance, designated F* (and identified as the F* criterion), below the bottom of the roll transition for which tube degradation of any extent does not necessitate remedial action, e.g., plugging or sleeving. The F* criterion provides for sufficient engagement of the tube to tubesheet hardroll such that pullout forces that could be developed during normal operating or faulted conditions would be successfully resisted by the elastic preload between the tube and tubesheet. The necessary engagement length applicable to the D. C. Cook Unit 1 steam generators was found to be 1.11 inch based on preload analysis. Application of the F* criterion provides a level of protection for tube degradation in the tubesheet region commensurate with that afforded by Regulatory Guide (RG) 1.121, Reference 1, for degradation located outside the tubesheet region.

2.0 EVALUATION

Tube rupture in the conventional sense, i.e., characterized by an axially oriented "fishmouth" opening in the side of the tube, is not possible within the tube/tubesheet roll expansion (RE). The reason for this is that the tubesheet material prevents the wall of the tube from expanding outward in response to the internally acting pressure forces. The forces which would normally act to cause crack extension are transmitted into the walls of the tubesheet, the same as for a non-degraded tube, instead of acting on the tube material. Thus, axially oriented linear indications, e.g., cracks, cannot lead to tube failure within the RE and may be considered on the basis of leakage effects only.

Likewise, a circumferentially oriented tube rupture is resisted because the tube is not free to deform in bending within the roll expansion. When degradation has occurred such that the remaining tube cross sectional area does not present a uniform resistance to axial loading, bending stresses are developed which may significantly accelerate failure. When bending forces are resisted by lateral support loads, provided by the tubesheet, the acceleration mechanism is mitigated and a tube separation mode similar to that which would occur in a simple tensile test results. Such a separation mode, however, requires the application of significantly higher loads than for the unsupported case.

In order to evaluate the applicability of any developed criterion for indications within the tubesheet, some postulated type of degradation must be considered. For this evaluation it was postulated that a circumferential severance of a tube could occur, contrary to existing plant operating experience. However, implicit in assuming a circumferential severance to occur, is the consideration that degradation of any extent could be demonstrated to be tolerable below the location determined acceptable for the postulated condition.

When the tubes have been hardrolled into the tubesheet, any axial loads developed by pressure and/or mechanical forces acting on the tubes are resisted by frictional forces developed by the elastic preload that exists between the tube and the tubesheet. For some specific length of engagement of the hardroll, no significant axial forces will be transmitted further along the tube, and that length of tubing, i.e., F*, will be sufficient to anchor the tube in the tubesheet. In order to determine the value of F* for application in Series 51 steam generators, a testing program was conducted to measure the elastic preload of the tubes in the tubesheet.

The presence of the elastic preload also presents a significant resistance to flow of primary to secondary or secondary to primary water for degradation which has progressed fully through the thickness of the tube. In effect, no leakage would be expected if a sufficient length of hardroll is present. This has been demonstrated in high pressure fossil boilers where hardrolling of tube to tubesheet joints is the only mechanism resisting flow, and in steam generator sleeve to tube joints made by the Westinghouse hybrid expansion joint process.

2.1 DETERMINATION OF ELASTIC PRELOAD BETWEEN THE TUBE AND TUBESHEET

Tubes are installed in the steam generator tubesheet by a hardrolling process which expands the tube to bring the outside surface into intimate contact with the tubesheet hole. The roll process and roll torque are specified to result in a metal to metal interference fit between the tube and the tubesheet.

A test program was conducted by Westinghouse to quantify the degree of interference fit between the tube and the tubesheet provided by the partial depth hardrolling operation. The data generated in these tests have been analyzed to determine the length of hardroll required to preclude axial tube forces from being transmitted further along the tube, i.e., to establish the F* criterion. The amount of interference was determined by installing tube specimens in collars specifically designed to simulate the tubesheet radial stiffness. A hardroll process representative of that used during steam generator manufacture was used in order to obtain specimens which would exhibit installed preload characteristics like the tubes in the tubesheet.

Once the hardrolling was completed, the test collars were removed from the tube specimens and the springback of the tube was measured. The amount of springback was used in an analysis to determine the magnitude of the interference fit, which is representative of the residual tube to tubesheet radial load in Westinghouse Series 51 steam generators.

2.1.1 Radial Preload Test Configuration Description

The test program was designed to simulate the interface of a tube to tubesheet partial depth hardroll for a Series 51 steam generator. The test configuration consisted of six cylindrical collars, approximately [ ]^{a,c,e} inches in length, [ ]^{a,c,e} inches in outside diameter, and [ ]^{a,c,e} inch in inside diameter. A mill annealed, Inconel 600 (ASME SB-163), tubing specimen, approximately [ ]^{a,c,e} inches long with a nominal [ ]^{a,c,e} outside diameter before rolling, was hard rolled into each collar using a process which simulated actual tube installation conditions.

The design of the collars was based on the results of performing finite element analysis of a section of the steam generator tubesheet to determine radial stiffness and flexibility. The inside diameter of the collar was chosen to match the size of holes drilled in the tubesheet. The outside diameter was selected to provide the same radial stiffness as the tubesheet.

The collars were fabricated from AISI 1018 carbon steel similar in mechanical properties to the actual tubesheet material. The collar assembly was clamped in a vise during the rolling

process and for the post roll measurements of the tube ID. Following the taking of all post roll measurements, the collars were saw cut to within a small distance from the tube wall. The collars were then split for removal from the tube and tube ID and OD measurements repeated.

Two end boundary conditions were imposed on the tube specimen during rolling. The end was restrained from axial motion in order to perform a tack roll at the bottom end, and was allowed to expand freely during the final roll.

2.1.2 Preload Test Results: Discussion and Analysis

All measurements taken during the test program are tabulated in Table 1. The data recorded were employed to determine the interfacial conditions of the tubes and collars. These consisted of the ID and OD of the tubes prior to and after rolling and removal from the collars as well as the inside and outside dimensions of each collar before and after tube rolling. Two orthogonal measurements were taken at six axial locations within the collars and tubes. All measured dimensions given in Table 1 are in inch units. The remainder of the data of particular interest was calculated from these specific dimensions. The calculated dimensions included wall thickness, change in wall thickness for both rolling and removal of the tubes from the collars, and percent of spring back.

Using the measured and calculated physical dimensions, an analysis of the tube deflections was performed to determine the amount of preload radial stress present following the hardrolling. The analysis consisted of application of conventional thick tube equations to account for variation of structural parameters through the wall thickness. However, traditional application of cylinder analysis considers the tube to be in a state of plane stress. For these tests the results implied that the tubes were in a state of plane strain elastically. This is in agreement with historical findings that theoretical values for radial residual preload are below those actually measured, and that axial frictional stress between the tube and the tubesheet increases the residual pressure. In a plane stress analysis such stress is taken to be zero

(References 2 and 3). Based on this information the classical equations relating tube deformation and stress to applied pressure were modified to reflect plane strain assumptions.

The standard analysis of thick walled cylinders results in an equation for the radial deflection of the tube as:

$$u = C_1 * r + C_2 / r \tag{1}$$

where, u = radial deflection

r = radial position within the tube wall,

and the constants, C_1 and C_2 are found from the boundary conditions to be functions of the elastic modulus of the material, Poisson's ratio for the material, the inside and outside radii, and the applied internal and external pressures. The difference between an analysis assuming plane stress and one assuming plane strain is manifested only in a change in the constant C_2. The first constant is the same for both conditions. For materials having a Poisson's ratio of 0.3, the following relation holds for the second constant:

$$C_2 \text{ (Plane Strain)} = 0.862 * C_2 \text{ (Plane Stress)} \tag{2}$$

The effect on the calculated residual pressure is that plane strain results are higher than plane stress results by slightly less than 10 percent. Comparing this effect with the results reported in Reference 2 indicated that better agreement with test values is achieved. It is to be noted that the residual radial pressure at the tube to tubesheet interface is the compressive radial stress at the OD of the tube.

By substituting the expressions for the constants into Equation (1), the deflection at any radial location within the tube wall as a function of the internal and external pressure (radial stress at the ID and OD) is found. This expression was differentiated to obtain flexibility values for the tube deflection at the ID and OD respectively, e.g., dU_i/dP_o is the ratio of the radial deflection at the ID due to an OD pressure. Thus, dU_i/dP_o was used to find the interface pressure and radial stress between the tube and the tubesheet as:

$$S_{ro} = -P_o = -\text{(ID Radial Springback)} / (dU_i/dP_o) \tag{3}$$

The calculated radial residual stress for each specimen at each location is tabulated in Table 2. The mean residual radial stress and the standard deviation were found to be [ ]^{a,c,e} psi and [ ]^{a,c,e} psi, respectively. In order to determine a value to be used in the analysis, a tolerance factor for [ ]^{a,c,e} percent confidence to contain [ ]^{a,c,e} percent of the population was calculated, considering the [ ]^{a,c,e} useable data points, to be [ ]^{a,c,e}. Thus, a [ ]^{a,c,e} lower tolerance limit (LTL) for the radial residual preload at room temperature is [ ]^{a,c,e} psi.

2.1.3 Residual Radial Preload During Plant Operation

During plant operation the amount of preload will change depending on the pressure and temperature conditions experienced by the tube. The room temperature preload stresses, i.e., radial, circumferential and axial, are such that the material is nearly in the yield state if a comparison is made to ASME Code, Reference 4, minimum material properties. Since the coefficient of thermal expansion of the tube is greater than that of the tubesheet, heatup of the plant will result in an increase in the preload and could result in some yielding of the tube. In addition, the yield strength of the tube material decreases with temperature. Both of these effects may result in the preload being reduced upon return to ambient temperature conditions, i.e., in the cold condition. However, as documented in Reference 5, for a similar investigation, tube pullout tests which were preceded by a very high thermal relaxation soak showed the analysis to be conservative.

The plant operating pressure influences the preload directly based on the application of the pressure load to the ID of the tube, thus increasing the amount of interface loading. The pressure also acts indirectly to increase the amount of interface loading by causing the tubesheet to bow upward, i.e., placing the roll expansions near the bottom of the tubesheet in compression for normal operating and FLB conditions. For the LOCA event, the tubesheet bows in the opposite direction, producing dilation of the tubesheet holes and reducing the amount of tube to tubesheet preload. Each of these effects may be quantitatively treated.

The maximum amount of increase in preload due to tubesheet bow for primary-to-secondary pressure differential will occur at the bottom, central part of the tubesheet. Since F* is measured from the bottom of the hardroll transition (BRT) and leakage is to be restricted by the F* region of the tube, the potential for the tube section within the F* region to experience a net tightening or loosening during operation is evaluated. However, the central location case is not the most stringent case for normal operation and FLB; rather, the most stringent case for normal operation and FLB involves a peripheral tube, which experiences little or no increase in radial preload due to tubesheet bending. The effects of the three identified mechanisms affecting the preload are considered in the following sections.

2.1.4 Calculation of F* Based on Limiting Operating Conditions

An evaluation was performed to determine the effect of variations in operating parameters on residual preload at the tube/tubesheet expansion zone, the axial pullout force on the tube, and the resulting value of F*. For D. C. Cook Unit 1, a total of 15 cases including various parameters approved for Cook-1 by the Power Capability Working Group (PCWG) were evaluated; the cases represent various conditions at levels of plugging ranging from 0 to 30%. The residual preload in the roll expansion is affected by differential thermal expansion, internal pressure and tubesheet bowing; T_{hot}, T_{cold}, and the primary and secondary side pressures are used to evaluate these effects. The primary to secondary ΔP affects the axial pullout load on the tube. These parameters are summarized in Tables 3 and 4 for each PCWG condition evaluated. For the PCWG cases, the primary side pressure may range from 2100 psia to 2250 psia. Table 3 shows calculations of F* for the 2100 psia cases; Table 4 shows results for the 2250 psia cases. Case 1-1 of Tables 3 and 4 corresponds to the original plant operating conditions with 0% plugging. Cases 1-3 to 1-11 corresponded to plugging levels of 10% to 15% (PCWG Case 1-2 applies only to D. C. Cook Unit 2 and was not evaluated). Cases 2-1 to 2-4 corresponded to postulated 30% steam generator tube plugging conditions. Case 3-1 represents the bounding plant operating conditions with respect to D. C. Cook Unit 1 Evaluation for Tube Vibration Induced Fatigue (Reference 7).

Calculations were also performed for faulted conditions. The feedline break event is used as the limiting faulted condtion. While the steamline break (SLB) event provides the most stringent radiological conditions for postulated accidents involving a loss of pressure or fluid in the secondary system, the FLB pressure differential of 2650 psia maximizes the axial end loading on the tube for tube pullout considerations.

Based on the calculated values of F* for normal operation in Tables 3 and 4, and separate calculations for faulted conditions, the maximum value of F* is obtained for Case 1-9 of normal operation with P_p = 2250 psi, which is the case with the maximum primary-to-secondary ΔP and the lowest T_{cold}. The results for this limiting case are presented in the following sections.

2.1.5 Increase in Radial Preload Due to Thermal Expansion Tightening

For conservatism in determining the total residual preload for normal operating conditions, tightening of the tube/tubesheet joint due to differential thermal expansion is minimized by applying the SG outlet temperature to the tubing. For the limiting PCWG case from Section 2.1.4, this corresponds to a cold leg temperature of 511°F. The mean coefficient of thermal expansion for the Inconel tubing between ambient conditions and 511°F is approximately $7.71*10^{-6}$ in/in/°F. That for the steam generator tubesheet is $7.27*10^{-6}$ in/in/°F. These values were reconciled as conservative with respect to the 1968 ASME Boiler and Pressure Vessel Code, which was the code of construction for the Cook-1 SGs. Thus, there is a net difference of $0.44*10^{-6}$ in/in/°F between the expansion properties of the two materials. Considering a temperature difference of (511 - 70) = 441°F between ambient and operating conditions, the increase in preload between the tube (t) and the tubesheet (ts) was calculated as:

$$S_{rT} = (0.44E\text{-}6)*(441)*(\text{Collar ID}) / 2 / ((dU_i/dP_i)_{ts} - (dU_o/dP_o)_t) \qquad (4)$$

The results indicate that the increase in preload radial stress due to thermal expansion is [ ]^{a,c,e} psi. It is to be noted that this value applies for both normal operating and faulted conditions.

2.1.6 Increase in Radial Preload during N.O. and FLB Due to Differential Pressure

The normal operating (N.O.) differential pressure from the primary to secondary side of the steam generator during the most limiting PCWG condition is 1674 psi. The internal pressure acting on the wall of the tube will result in an increase of the radial preload on the order of the pressure value. The increase was found as:

$$S_{rP} = - P_o = - P_i \, (dU_o/dP_i) \, / \, (\, (dU_i/dP_i)_{ts} - (dU_o/dP_o)_t) \qquad (5)$$

In actuality, the increase in preload will be more dependent on the internal pressure of the tube since water at secondary side pressure would not be expected between the tube and the tubesheet. However, the primary to secondary ΔP is used for conservatism.

The increase in radial contact pressure due to differential pressure was evaluated for both normal operating (ΔP = 1674 psi) and faulted (ΔP = 2650 psi) conditions. The results indicate that the increase in preload radial stress is [ ]^{a,c,e} psi for normal operating conditions and [ ]^{a,c,e} psi for faulted (FLB) conditions.

2.1.7 Change in Radial Preload due to Tubesheet Bow

An analysis of the Series 51 tubesheet was performed to evaluate the change in preload stress that would occur as a result of tubesheet bow for interior tubes. The analysis was based on performing finite element analysis of the tubesheet and SG shell using equivalent perforated plate properties for the tubesheet (Reference 3). Boundary conditions from the results were then applied to a smaller, but more detailed model, in order to obtain results for the tubesheet holes. Basically the deflection of the tubesheet was used to find the stresses active on the bottom surface and then the presence of the holes was accounted for. For the location where the increase of preload is a maximum, the radial preload stress would be increased by [ ]^{a,c,e} psi during normal operation and [ ]^{a,c,e} psi during faulted (FLB) conditions.

However, the interior tubes are not the limiting case for primary-to-secondary pressure differential. The limiting case involves peripheral tubes where tubesheet bowing has a negligible effect on tube-to-tubesheet preload. Therefore, the N.O. and FLB analyses address only tubes in the peripheral region of the tubesheet. During LOCA, the differential operating pressure is from secondary to primary. Thus, the radial preload will decrease by [ ]^{a,c,e} psi as the tubesheet bows downward. However, the action of the differential pressure is such that the tube is pushed toward the tube-to-tubesheet well. This case is of no consequence to the determination of F*.

2.1.8 Net Preload in Roll Transition Region for N.O. and FLB Conditions

Combining the room temperature hardroll preload with the thermal and pressure effects results in a net operating preload of [ ]^{a,c,e} psi during normal operation and [ ]^{a,c,e} psi for faulted conditions. In addition to restraining the tube in the tubesheet, this preload should effectively retard leakage from indications in the tubesheet region of the tubes.

2.2 DETERMINATION OF REQUIRED ENGAGEMENT DISTANCE

The calculation of the value of F* recommended for application to the D. C. Cook Unit 1 steam generators is based on determining the length of hardroll necessary to offset the applied loads during the maximum normal operating conditions or faulted conditions, whichever provides the largest value. Thus, the applied loads are balanced by the load carrying ability of the hardrolled tube for both of the above conditions. In performing the analysis, consideration is made of the potential for the ends of the hardroll at the hardroll transition and the assumed severed condition to have a reduced load carrying capability.

2.2.1 Applied Loads

The applied loads to the tubes which could result in pullout from the tubesheet during all normal and postulated accident conditions are predominantly axial and due to the internal to external pressure differences. For a tube which has not been degraded, the axial pressure load is given by the product of the pressure with the internal cross-sectional area. However, for a tube with internal degradation, e.g., cracks oriented at an angle to the axis of the tube, the internal pressure may also act on the flanks of the degradation. Thus, for a tube which is conservatively postulated to be severed at some location within the tubesheet, the total force acting to remove the tube from the tubesheet is given by the product of the pressure and the cross-sectional area of the tubesheet hole. The force resulting from the pressure and internal area acts to pull the tube from the tubesheet and the force acting on the end of the tube tends to push the tube from the tubesheet. For this analysis, the tubesheet hole diameter has been used to determine the magnitude of the pressure forces acting on the tube. The forces acting to remove the tube from the tubesheet are [ ]^{a,c,e} pounds and [ ]^{a,c,e} pounds respectively for normal operating and faulted conditions. Any other forces such as fluid drag forces in the U-bends and vertical seismic forces are negligible by comparison.

2.2.2 End Effects

For a tube which is postulated to be severed within the tubesheet there is a material discontinuity at the location where the tube is severed. For a small distance from each assumed discontinuity the stiffness, and hence the radial preload, of the tube is reduced relative to that remote from the ends of the roll expansion. The analysis of end effects in thin cylinders is based on the analysis of a beam on an elastic foundation. For a tube with a given radial deflection at the end, the deflection of points away from the end relative to the end deflection is given by:

$$u_{rx} / u_{ro} = e^{-\lambda x} * \text{cosine} (\lambda * x) \qquad (6)$$

where, λ = [ ]^{a,c,e} = end effect constant.

x = distance from the end of the tube.

For the radially preloaded tube, the distance for the end effects to become negligible is the location where the cosine term becomes zero. Thus, for the roll expanded Series 51 tubes the distance corresponds to the product of "λ" times "x" being equal to (π/2) or [ ]a,c,e inch. Figure 1 shows a roll expansion which is postulated to be severed at the bottom of the F* region. For a distance of [ ]a,c,e inch above the severed end and below the bottom of the roll transition, the expanded joint has a reduced radial load carrying capability relative to the remainder of the F* length. The effective radial preload carried by these "end-affected" regions is calcualated as follows.

The above equation can be integrated to find the average deflection over the affected length to be 0.384 of the end deflection. This means that on the average the stiffness of the material over the affected length is 0.616 of the stiffness of the material remote from the ends. Therefore, the effective preload for the affected end lengths is 61.6 percent of the preload at regions more than [ ]a,c,e inch from the ends. For example, for the normal operating net preload of [ ]a,c,e psi or [ ]a,c,e pounds per inch of length, the effective preload for a distance of [ ]a,c,e inch from the end is [ ]a,c,e pounds per inch or [ ]a,c,e pounds.

2.2.3 Calculation of Engagement Distance Required, F*

The calculation of the required engagement distance is based on determining the length for preload frictional forces to equilibrate the applied operating loads. The axial friction force was found as the product of the radial preload force and the coefficient of friction between the tube and the tubesheet. The value assumed for the coefficient of friction was [ ]a,c,e, from Reference 5. For normal operation the radial preload is [ ]a,c,e psi or [ ]a,c,e pounds per inch of engagement. Thus, the axial friction resistance force is [ ]a,c,e pounds per inch of engagement. It is to be noted that this value applies away from the ends of the tube. For any given engagement length, the total axial resistance is the sum of that provided by the two ends plus that provided by the length minus the two end lengths. From the preceding section the axial resistance of each end is [ ]a,c,e pounds. Considering both ends of the presumed severed tube, i.e., the hardroll transition is considered one end, the axial resistance is [ ]a,c,e pounds plus the resistance of the material between the ends, i.e., the

total length of engagement minus [ ]a,c,e inch. For example, a one inch length has an axial resistance of,

[ ]a,c,e

Conversely, for the maximum normal operating pressure applied load of [ ]a,c,e pounds, considered as [ ]a,c,e pounds with a safety factor of 3, the length of hardroll required is given by,

F* = [ ]a,c,e

Similarly, the required engagement length for faulted conditions can be found to be 0.75 inch using a safety factor of 1.43 (corresponding to a ASME Code safety factor of 1.0/0.7 for allowable stress for faulted conditions).

The calculation of the above values is summarized in Table 5. The F* value thus determined for the required length of hardroll engagement below the BRT, for normal operation is sufficient to resist tube pullout during both normal and postulated accident condition loadings.

Based on the results of the testing and analysis, it is concluded that following the installation of a tube by the standard hardrolling process, a residual radial preload stress exists due to the plastic deformation of the tube and tubesheet interface. This residual stress is expected to restrain the tube in the tubesheet while providing a leak limiting seal condition.

2.3 LIMITATION OF PRIMARY TO SECONDARY LEAKAGE

The allowable amount of primary to secondary leakage in each D. C. Cook Unit 1 steam generator during normal plant operation is limited by plant technical specifications, generally to 0.1 gpm (150 gpd). This limit, based on plant radiological release considerations and implicitly enveloping the leak before break consideration for a throughwall crack in the free span of a tube, is also applicable to a leak source within the tubesheet. In evaluating the primary to secondary leakage aspect of the F* criterion, the relationship between the tubesheet

region leak rate at postulated FLB conditions is assessed relative to that at normal plant operating conditions. The analysis was performed by assuming the existence of a leak path; however, no actual leak path would be expected due to the hardrolling of the tubes into the tubesheet.

2.3.1 Operating Condition Leak Considerations

In actuality, the hardrolled joint would be expected to be leak tight, i.e., the plant would not be expected to experience leak sources emanating below F*. Because of the presence of the tubesheet, tube indications are not expected to increase the likelihood that the plant would experience a significant number of leaks. It could also be expected, that if primary to secondary leakage is detected in a steam generator it will not be in the tube region below F*. Thus, no significant radiation exposure due to the need for personnel to look for tube/tubesheet leaks should be anticipated, i.e., the use of the F* criterion is consistent with ALARA considerations. As an additional benefit relative to ALARA considerations, precluding the need to install plugs below the F* criterion would result in a significant reduction of unnecessary radiation exposure to installing personnel.

The issue of leakage within the F* region up to the top of the roll transition (RT) includes the consideration of postulated accident conditions in which the violation of the tube wall is very extensive, i.e., that no material is required at all below F*. Based on operating plant and laboratory experience the expected configuration of any cracks, should they occur, is axial. The existence of significant circumferential cracking is considered to be of very low probability. Thus, consideration of whether or not a plant will come off-line to search for leaks a significant number of times should be based on the type of degradation that might be expected to occur, i.e., axial cracks. Axial cracks have been found both in plant operation and in laboratory experiments to be short, about 0.5 inch in length, and tight. From field experience, once the cracks have grown so that the crack front is out of the skiproll or transition areas, they arrest.

Axial cracks in the free span portion of the tube, with no superimposed thinning, would leak at rates compatible with the technical specification acceptable leak rate. For a crack within the F* region of the tubesheet, expected leakage would be significantly less. Leakage through cracks in tubes has been investigated experimentally within Westinghouse for a significant number of tube wall thicknesses and thinning lengths (Reference 6). In general, the amount of leakage through a crack for a particular size tube has been found to be approximately proportional to the fourth power of the crack length. Analyses have also been performed which show, on an approximate basis for both elastic and elastic-plastic crack behavior, that the expected dependency of the crack opening area for an unrestrained tube is on the order of the fourth power, e.g., see NUREG CR-3464. The amount of leakage through a crack will be proportional to the area of the opening, thus, the analytic results substantiate the test results.

The presence of the tubesheet will preclude deformation of the tube wall adjacent to the crack, i.e., the crack flanks, and the crack opening area may be considered to be directly proportional to the length. The additional dependency, i.e., fourth power relative to first power, is due to the dilation of the unconstrained tube in the vicinity of the crack and the bending of the side faces or flanks of the crack. For a tube crack located within the tubesheet, the dilation of the tube and bending of the side faces of the crack are suppressed. Thus, a 0.5 inch crack located within the F* region up to the top of the roll transition would be expected to leak, without considering the flow path between the tube and tubesheet, at a rate less than a similar crack in the free span, i.e., less than the D. C. Cook Unit 1 technical specification limit of 150 gpd (~0.1 gpm). Additional resistance provided by the tube-to-tubesheet interface would reduce this amount even further, and in the hardroll region the residual radial preload would be expected to eliminate it. This conclusion is supported by the results of the preload testing and analysis which demonstrated that a residual preload in excess of [ ]^{a,c,e} psi exists between the tube and the tubesheet at normal operating conditions.

2.3.2 Postulated Accident Condition Leak Considerations

For the postulated leak source within the RE, increasing the tube differential pressure increases the driving head for the leak and increases the tube to tubesheet loading. For an initial location of a leak source below the BRT equal to F*, the FLB pressure differential results in an insignificant leak rate relative to that which could be associated with normal plant operation. This small effect is reduced by the increased tube to tubesheet loading associated with the increased differential pressure as well as the tightening contribution of the tubesheet bending. Thus, for a circumferential indication within the RE which is left in service in accordance with the pullout criterion (F*), the existing technical specification limit is consistent with accident analysis assumptions. For postulated accident conditions, the preload testing and analysis showed that a net radial preload of about [ ]^{a,c,e} psi would exist between the tube and tubesheet.

For axial indications in a partial depth hardrolled tube below the BRT of the roll transition zone (which is assumed to remain in the tubesheet region), the tube end remains structurally intact and axial loads would be resisted by the remaining hardrolled region of the tube. For this case, the leak rate due to FLB differential pressure would be bounded by the leak rate for a free span leak source with the same crack length, which is the basis for the accident analysis assumptions.

2.3.3 Operating Plant Leakage Experience for Within Tubesheet Tube Cracks

A significant number of within-tubesheet tube indications have been reported for some non-domestic steam generator units. The present attitude toward operation with these indications present has been to tolerate them with no remedial action relative to plugging or sleeving. No significant number of shutdowns occurring due to leaks through these indications have been reported.

2.4 TUBE INTEGRITY UNDER POSTULATED LIMITING CONDITIONS

The final aspect of the evaluation is to demonstrate tube integrity under the postulated loss of coolant accident (LOCA) condition of secondary to primary differential pressure. A review of tube collapse strength characteristics indicates that the constraint provided to the tube by the tubesheet gives a significant margin between tube collapse strength and the limiting secondary to primary differential pressure condition, even in the presence of circumferential or axial indications.

The maximum secondary to primary differential pressure during a postulated LOCA is [ ][a,c,e] psi. This value is significantly below the residual radial preload between the tubes and the tubesheet. Therefore, no significant secondary to primary leakage would be expected to occur. In addition, loading on the tubes is axially toward the tubesheet and could not contribute to pullout.

2.5 CHEMISTRY CONSIDERATIONS

The concern that boric acid attack of the tubesheet due to the presence of a through wall flaw within the hardroll region of the tubesheet may result in loss of contact pressure assumed in the development of the F* criterion is addressed below. In addition, the potential for the existence of a lubricated interface between the tube and tubesheet as a result of localized primary to secondary leakage and subsequent effects on the friction coefficient assumed in the development of the F* criterion is also discussed.

2.5.1 Tubesheet Corrosion Testing

Corrosion testing performed by Westinghouse specifically addressed the question of corrosion rates of tubesheet material exposed to reactor coolant. The corrosion specimens were assembled by bolting a steel (A336) coupon to an Inconel Alloy 600 coupon. The coupon dimensions were 3 inches x 3/4 inch x 1/8 inch and were bolted on both ends. A torque wrench was used to tighten the bolts to a load of 3 foot-pounds. The performance of A508 in

testing of this nature is expected to be quite comparable to the performance of A336 (Gr. F-1) steel (a material used for tubesheet construction prior to A508). The arguments used in supporting the F* case relative to corrosion of tubesheet material due to minute quantities of primary coolant contacting the carbon steel were so conservative and had such margin that minor differences in material composition or strength would not change the conclusion.

The specimens were tested under three types of conditions:

1. Wet-layup conditions
2. Wet-layup and operating conditions
3. Operating conditions only

The wet-layup condition was used to simulate shutdown conditions at high boric acid concentrations. The specimens were exposed to a fully aerated 2000 ppm boron (as boric acid) solution at 140 degrees F. Exposure periods were 2, 4, 6, and 8 weeks. Test solutions were refreshed weekly.

While lithium hydroxide is normally added to the reactor coolant as a corrosion inhibitor, it was not added in these tests in order to provide a more severe test environment. Previous testing by Westinghouse has shown that the presence of lithium hydroxide reduces corrosion of Inconel Alloy 600 and steel in a borated solution at operating temperatures.

Another set of specimens were used to simulate startup conditions with some operational exposure. The specimens were exposed to a 2000 parts per million boron (as boric acid) solution for one week in the wet-layup condition (140 degrees F), and 4 weeks at operating conditions (600 degrees F, 2000 psi). During wet layup, the test solution was aerated but at operating conditions the solution was deaerated. The high temperature testing was performed in an Inconel autoclave. Removal of oxygen was attained by heating the solution in the autoclave to 250 degrees F and then degassing. This method of removing the oxygen results in oxygen concentrations of less than 100 parts per billion.

Additional specimens were exposed under operating conditions only for 4 weeks in the autoclave as described above.

High temperature exposure to reactor coolant chemistry resulted in steel corrosion rates of about 1 mil per year. This rate was higher than would be anticipated in a steam generator since no attempt was made to completely remove the oxygen from the autoclave during heatup. Even with this amount of corrosion, the rate was still a factor of nine less than the corrosion rate observed during the low temperature exposure. This differential corrosion rate observed between high and low temperature exposure was expected because of the decreasing acidity of the boric acid at high temperatures and the corrosive effect of the high oxygen at low temperatures.

These corrosion tests are considered to be very conservative since they were conducted at maximum boric acid concentrations, in the absence of lithium hydroxide, with no special precaution to deaerate the solutions, and they were of short duration. The latter point is very significant since parabolic corrosion rates are expected in these types of tests, which leads one to overestimate actual corrosion rates when working with data from tests of short duration.

Also note that the ratio of solution to surface area is high in these tests compared to the scenario of concern, i.e., corrosion caused by reactor coolant leakage through a tube wall into the region between the tube and the tubesheet.

2.5.2 Tubesheet Corrosion Discussion

At low temperatures, e.g., less than 140 degrees F, aerated boric acid solutions comparable in strength to primary coolant concentrations can produce corrosion of carbon steels. Deaerated solutions are much less aggressive and deaerated solutions at reactor coolant temperatures produce very low corrosion rates due to the fact that boric acid is a very much weaker acid at high temperature, e.g., 610 degrees F, than at 70 degrees F.

In the event that a crack occurred within the hardroll region of the tubesheet, as the amount of leakage would be expected to be insufficient to be noticed by leak detection techniques and is largely retained in the crevice, then a very small volume of primary fluid would be involved. Any oxygen present in this very small volume would quickly be consumed by surface reactions, i.e., any corrosion that would occur would tend to cause existing crevices to narrow due to oxide expansion and, without a mode for replenishment, would represent a very benign corrosion condition. In any event the high temperature corrosion rate of the carbon steel in this very local region would be extremely low (significantly less than 1 mil per year).

Contrast the proposed concern for corrosion relative to F* with the fact that Westinghouse has qualified boric acid for use on the secondary side of steam generators where it is in contact with the full surface of the tubesheet and other structural components made of steel. The latter usage involves concentrations of 5 - 10 ppm boron, but, crevice flushing procedures have been conducted using concentrations of 1000 to 2000 ppm boron on the secondary side (at approximately 275 degrees F where boric acid is more aggressive than at 610 degrees F).

Relative to the lubricating effects of boron, the presence of boric acid in water may change the wetting characteristics (surface tension) of the water but Westinghouse is not aware of any significant lubricating effect. In fact, any corrosion that would occur would result in oxides that would occupy more space than the parent metals, thus reducing crevice volume or possibly even merging the respective oxides.

3.0 SUMMARY

On the basis of this evaluation, it is determined that tubes with eddy current indications in the tubesheet region below the F* pullout criterion of 1.11 inch can be left in service. Tubes with circumferentially oriented eddy current indications of pluggable magnitude and located a distance less than F* below the bottom of the hardroll transition should be removed from service by plugging or repaired in accordance with the plant technical specification plugging limit. The conservatism of the F* criterion was demonstrated by preload testing and analysis commensurate with the requirements of RG 1.121 for indications in the free span of the tubes.

For tubes with axial indications, the criterion which should be used to determine whether tube plugging or repairing is necessary should be based on leakage since the axial strength of a tube is not reduced by axial cracks. Under these circumstances it has been demonstrated that significant leakage would not be expected to occur for through wall indications greater than [ ][a,c,e] inch below the bottom of the hardroll transition. Therefore, an F* distance of 1.11 inches achieves a leak limiting condition.

4.0 REFERENCES

1. United States Nuclear Regulatory Commission, Regulatory Guide 1.121, "Bases for Plugging Degraded PWR Steam Generator Tubes," August, 1976.

2. Goodier, J. N., and Schoessow, G. J., "The Holding Power and Hydraulic Tightness of Expanded Tube Joints: Analysis of the Stress and Deformation," Transactions of the A.S.M.E., July, 1943, pp. 489-496.

3. Grimison, E. D., and Lee, G. H., "Experimental Investigation of Tube Expanding,", Transactions of the A.S.M.E., July, 1943, pp. 497-505).

4. ASME Boiler and Pressure Vessel Code, Section III, "Rules for Construction of Nuclear Power Plant Components," The American Society of Mechanical Engineers, New York, New York, 1983.

5. WCAP-11241, "Tubesheet Region Plugging Criterion for the Central Nuclear de ASCO, ASCO Nuclear Station Units 1 and 2 Steam Generators", October, 1986. (Proprietary)

6. WCAP-10949, "Tubesheet Region Plugging Criterion for Full Depth Hardroll Expanded Tubes," Westinghouse Electric Corporation, September, 1985. (Proprietary)

7. WCAP-13814, "Donald C. Cook unit 1 Evaluation for Tube Vibration Induced Fatigue, Westinghouse Electric Corporation, December 1993. (Proprietary)

Model 51 Steam Generator Tube Roll Pre-Load Test

TABLE 1. TEST DATA

Test No.	Location No.	Collar ID Pre-Roll			Collar OD Pre-Roll			Tube ID Before Roll			Tube OD Before Roll		
		0 Deg.	90 Deg.	Avg.	0 Deg.	90 Deg.	Avg.	0 Deg.	90 Deg.	Avg.	0 Deg.	90 Deg.	Avg.
1	1												
	2												
	3												
	4												
	5												
	6												
	Average												
2	1												
	2												
	3												
	4												
	5												
	6												
	Average												
3	1												
	2												
	3												
	4												
	5												
	6												
	Average												
6	1												
	2												
	3												
	4												
	5												
	6												
	Average												
7	1												
	2												
	3												
	4												
	5												
	6												
	Average												
8	1												
	2												
	3												
	4												
	5												
	6												
	Average												
Col. Avgs:													

a,c,e

Model 51 Steam Generator Tube Roll Pre-Load Test.

TABLE 1. TEST DATA (Cont.)

Test No.	Location No.	Pre-Roll Thickness	Collar OD Post-Roll 0 Deg.	90 Deg.	Avg.	Collar Delta	Tube ID Post-Roll 0 Deg.	90 Deg.	Avg.	Tube ID Growth	Tube ID Post-Roll Collar Removed 0 Deg.	90 Deg.	Avg. a,c,e
1	1												
	2												
	3												
	4												
	5												
	6												
	Average												
2	1												
	2												
	3												
	4												
	5												
	6												
	Average												
3	1												
	2												
	3												
	4												
	5												
	6												
	Average												
6	1												
	2												
	3												
	4												
	5												
	6												
	Average												
7	1												
	2												
	3												
	4												
	5												
	6												
	Average												
8	1												
	2												
	3												
	4												
	5												
	6												
	Average												
Col. Avgs:													

Model 51 Steam Generator Tube Roll Pre-Load Test

TABLE 1. TEST DATA (Cont.)

a,c,e

Test No.	Location No.	Tube OD Post-Roll Collar Removed 0 Deg.	90 Deg.	Avg.	Post-Roll Thick	Thick-ness Red.	Collar Flex. dUi/dPi	Radii Ratio (4)	Tube ID Spring-Back
1	1								
	2								
	3								
	4								
	5								
	6								
	Average								
2	1								
	2								
	3								
	4								
	5								
	6								
	Average								
3	1								
	2								
	3								
	4								
	5								
	6								
	Average								
6	1								
	2								
	3								
	4								
	5								
	6								
	Average								
7	1								
	2								
	3								
	4								
	5								
	6								
	Average								
8	1								
	2								
	3								
	4								
	5								
	6								
	Average								
Col. Avgs:									

Notes: 1. All measured dimensions are in inches.
2. The OD stress is calculated using the measured ID springback.
3. The radii ratio is a term that appears frequently in the analysis and is found as (OD^2+ID^2)/(OD^2-ID^2).

Model 51 Steam Generator Tube Roll Pre-Load Test

TABLE 2. STRESS ANALYSIS RESULTS

a,c,e

Test No.	Location No.	Tube ID Spring-Back	Tube Flex. dUi/dPo	Tube Flex. dUo/dPo	OD Radial Stress	OD Hoop Stress	OD Axial Stress	Thermal Exp. Radial Stress	Tube Flex. dUo/dPi	Oper. Pressure Radial Stress	Total Radial Stress	Total vonMises Stress
1	1											
	2											
	3											
	4											
	5											
	6											
	Average											
2	1											
	2											
	3											
	4											
	5											
	6											
	Average											
3	1											
	2											
	3											
	4											
	5											
	6											
	Average											
6	1											
	2											
	3											
	4											
	5											
	6											
	Average											
7	1											
	2											
	3											
	4											
	5											
	6											
	Average											
8	1											
	2											
	3											
	4											
	5											
	6											
	Average											
Col. Avgs:												

Notes: 1. The OD stress is calculated using the measured ID springback.

Table 3

Calculation of F* Length for T/TS Hardroll Interface - D. C. Cook 1
Minimum RCS Pressures for all PCWG Cases

Ref.-Case	% Tube Plugging	RCS Pressure (psia)	Steam Pressure (psia)	Pri-Sec Delta P (psia)	Thot (deg. F)	Tcold (deg. F)	Limiting (1) Diff Temp (deg. F)	Delta P Ratio (2)	Tcold T ratio (3)	N.O. Srp (4)	Srt (5)	Hardroll Preload	Total Sr (6)	N.O. Pr (7)	N.O. Pend (8)	Pa (9)	3Pa	N.O. F* (10)
1-1	0	2250	758	1492	599.3	536.3												
1-3	15	2100	807	1293	607.5	545												
1-4	10	2100	603	1497	580.7	513.1												
1-5	10	2100	820	1280	611.2	546												
1-6	15	2100	806	1294	611.2	546												
1-7	10	2100	587	1513	582.3	511.4												
1-8	10	2100	820	1280	615.2	547.1												
1-9	15	2100	576	1524	582.3	511.4												
1-10	15	2100	808	1294	615.2	547.2												
1-11	15	2100	607	1493	579.1	514.6												
2-1	30	2100	595	1505	586.8	518.9												
2-2	30	2100	749	1351	609.1	543.2												
2-3	30	2100	589	1511	588.5	517.2												
2-4	30	2100	742	1358	610.8	541.6												
3-1	Current	2250	821	1429	608.1	542.9												

a,c,e

a,c,e

References:

1) 1989 PCWG parameters for 10 avg. and 15% peak plugging.
2) 30% SGTP parameters from Letter PCWG-1983, 9/7/93.
3) Current operating parameters used for Cook-1 U-bend fatigue analysis, WCAP-13814, Dec. 1993.

Table 4

Calculation of F* Length for T/TS Hardroll Interface - D. C. Cook 1
Maximum RCS Pressures for all PCWG Cases

Ref.-Case	% Tube Plugging	RCS Pressure (psia)	Steam Pressure (psia)	Pri-Sec Delta P (psia)	Thot (deg. F)	Tcold (deg. F)	Limiting (1) Diff Temp (deg. F)	Delta P Ratio (2)	Tcold T ratio (3)	N.O. Srp (4)	Srt (5)	Hardroll Preload	Total Sr (6)	N.O. Pr (7)	N.O. Pend (8)	Pa (9)	3Pa	N.O. F* (10)
1-1	0	2250	758	1492	599.3	536.3												
1-3	15	2250	807	1443	607.5	545												
1-4	10	2250	603	1647	580.7	513.1												
1-5	10	2250	820	1430	611.2	546												
1-6	15	2250	806	1444	611.2	546												
1-7	10	2250	587	1663	582.3	511.4												
1-8	10	2250	820	1430	615.2	547.1												
1-9	15	2250	576	1674	582.3	511.4												
1-10	15	2250	806	1444	615.2	547.2												
1-11	15	2250	607	1643	579.1	514.6												
2-1	30	2250	595	1655	586.8	518.9												
2-2	30	2250	749	1501	609.1	543.2												
2-3	30	2250	589	1661	588.5	517.2												
2-4	30	2250	742	1508	610.8	541.6												
3-1	Current	2250	821	1429	608.1	542.9												

a,c,e

a,c,e

References:

1) 1989 PCWG parameters for 10 avg. and 15% peak plugging.
2) 30% SGTP parameters from Letter PCWG-1983, 9/7/93.
3) Current operating parameters used for Cook-1 U-bend fatigue analysis, WCAP-13814, Dec. 1993.

Table 5

Preload Analysis Summary

Material Properties:	
Elastic Modulus	28.7E+06
Poisson's Ratio	0.30
I600 Coeff. of Therm. Exp.	7.71E-06 in/in/°F
TS Coeff. of Therm. Exp.	7.27E-06 in/in/°F
Operating ΔT	411°F
N.O. ΔP	1674 psi
Faulted ΔP	2650 psi

Tube/Tubesheet Dimensions (Tested):	a,c,e
Init. Avg. Tube OD	
Init. Avg. Tube Thickness	
Init. Avg. Tubesheet ID	
Actual Thinning	
Apparent Thinning	
Coefficient of Friction	
End Effects:	
Mean Radius (Rolled)	
Thickness (Rolled)	
λ	
End Effect Length	
Load Factor	

Additional Analysis Input:

Tubesheet Bow Stress Reduction	a,c,e
N.O.	
FLB	
Lower Tolerance Limit Factor	
95/95 LTL	2.16 (N=36)

EVALUATION OF REQUIRED ENGAGEMENT LENGTH, F*

Elastic Analysis:	N.O.	FLB	a,c,e
RT Preload (LTL)			
Thermal Expansion Preload			
Pressure Preload			
Tubesheet Bow Loss			
Net Preload			
Net Radial Force			
Net Axial Resistance			
Applied Load			
Analysis Load			
End Effect Resistance			
Net Analysis Load			
End Effect Length			
Add. Length Required			
Total Length Required, F*	1.11in.	0.75 in.	

Table 5 (continued)

Preload Analysis Summary

NOTES:
1) 95/95 Lower Tolerance Limit rolled preload used.
2) For Normal Operation, a safety factor of 3.0 is used.
3) For Faulted Conditions, a safety factor of 1.43 is used (corresponding to ASME Code use of 0.7 on ultimate strength.
4) The required length does not include eddy current inspection uncertainty for the location of the bottom of the hard roll.
5) Preload stresses used were for the most stringent case, i.e., cold leg, peripheral location. This minimizes the thermal expansion preload and eliminates the preload due to tubesheet bowing.

Figure 1

"End-Affected" Regions in a Tube Postulated to be Severed at the Bottom of F* Length