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ONCE-THROUGH STEAM GENERATOR MECHANICAL SLEEVE QUALIFICATION

Babcock & Wilcox
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1. INTRODUCTION

Some steam generator tubes have been found to have a varying amount of wall degradation after only a few years' service. If the degradation is extensive, the normal practice of plugging defective tubes may reduce the effectiveness of the steam generators and eventually reduce the performance of the nuclear steam supply (NSS) system. An alternative to tube plugging is tube sleeving. A sleeve is installed as a new pressure boundary inside the original tube to bridge the degraded area, thus permitting the tube to remain in service. Babcock & Wilcox (B&W) has developed and qualified a mechanical tube sleeve that can be installed in degraded tubes of once-through steam generators (OTSGs). This sleeve is more resistant to the expected types of corrosive attack than the original tubes. It is strong enough and sufficiently leak-free to be used as a permanent remedy to keep degraded tubes in service. This report demonstrates the technical adequacy of the OTSG mechanical sleeve for use in degraded OTSG tubes for both normal and accident load conditions.

The development of the sleeve design requirements and the actual sleeve design are discussed in the program description. This section continues to describe the qualification program and test specimens designed to evaluate the effect of service conditions on the strength and leakage of the mechanical sleeve. The results of the strength and leakage tests are presented and evaluated for both pre- and post-service conditions. Supplemental tests for corrosion resistance and the effects of tube surface condition and sleeve bending and straightening are also described. The results of an American Society of Mechanical Engineers (ASME) Code analysis of the sleeve are discussed, as well as the effect of sleeves on plant performance. The character of flow-induced vibration of a sleeved tube is compared to that of an unsleeved tube. The qualification test results are summarized, and conclusions are drawn regarding the use of the sleeve design.

2. PROGRAM DESCRIPTION

2.1. Sleeve Design

Tube degradations found in OTSGs have been located most frequently within a few inches above or below the secondary face of the upper tubesheet (UTS). There is some concentration at the 15th tube support plate (TSP) and in the 16th tube span (i.e., between the 15th TSP and the UTS), and the remainder seem to be rather randomly distributed at elevations below the 15th TSP. The radial location of degradations tends to be more frequent toward the outer periphery of the tube bundle, and there is some concentration near the open tube lane. Therefore, a sleeve that extends through the thickness of the UTS and a few inches beyond would serve the most frequent need. Extending the same sleeve through the 15th tube support plate would be most effective. The sleeve must also be installable in any tube in the steam generator.

The head clearance over the outermost tubes in the OTSG is only about 13 inches, whereas the sleeve length required to extend 6 inches beyond the UTS secondary face is 30 inches. Thus, installation requires that the sleeves be pre-bent to a gentle radius in order to clear the head and then straightened as they are fed into the tubes. After the tooling was developed to bend and straighten sleeves as they are installed, it could be modified to bend and straighten longer sleeves. Therefore, the qualified design was extended from 30 to 80 inches in order to span the entire 16th tube span and 15th TSP. This sleeve can be installed in any tube in the generator, and tooling modifications could permit installation from the lower as well as the upper tubesheet. Thus, the design which has been qualified is a 30- to 80-inch sleeve located in any tube in the OTSG.

The sleeve is made from tubing, according to standards set by ASME SB-163, that has been mill-annealed and heat treated The tube size is

As shown in Figure 2-1, there is expansion in the tubesheet, and expansion near the free end of the sleeve. The length of the sleeve could range between 30 and 80 inches overall.

The installation method qualified is roller expanding both ends of the sleeve. This method is quick, gives adequate leak tightness and pullout strength, and is easily adapted to remote operation. Both hydraulic and explosive expansion techniques were considered to obtain a mechanical seal, but the rolling technique was preferred due to the speed and simplicity of the operation and the leak-tight nature of the joint. Both brazing and welding techniques would yield sealed sleeves, but the installation rate was judged to be much too slow and expensive for large installations.

Roller expansion has been used to seal tubes into tubesheets since the 1930's. The amount of expansion is easily contolled by limiting the torque applied to the expansion tool, since there is a rapid increase in torque as the tube is squeezed against the tubesheet and begins to be extruded axially. Roller expansion of the sleeve in the free-span tube without a tubesheet back-up is an innovative application.

The adequacy of various amounts of expansion is evaluated.

2.2. Design Requirements

The sleeve design loading requirements have been established to be equivalent to those of the unsleeved tube on the premise that a sleeved tube could be totally severed without affecting the function of the tube.[1] The sleeve is considered to be a structural member that meets all normal, upset, emergency, and faulted conditions resulting from normal operation and accident transients.

The ASME loading conditions are tabulated in Table 2-1, and the design transients are tabulated in Table 2-2.

The maximum required pullout strength of the roller-expanded joint

Figure 2-2 plots acceptable combinations of axial load and joint slippage, which are dependent upon the yield strength of the parent tube material. If there were no slip (e.g., an unsleeved tube), the required axial load capability would be the combined thermal and mechanical load (3149 lb) of the worst accident condition, a main steam line break. If there were no axial load (e.g., a severed tube), the sleeve joint must be able to slip at least 1.09 inches, the maximum relative displacement between the severed ends of the tube without separation of the joint.

A sleeve leakage objective was established based on plant operating limits for primary-to-secondary leakage and an assumed quantity of sleeves which could be installed. Plant Technical Specifications normally require that plant shutdown be initiated if steam generator primary-to-secondary leakage exceeds 1.0 gpm.

The corrosion resistance of the installed sleeve must be equal to or better than the original Inconel 600 tubes as heat treated and stress-relieved for the OTSG. This is interpreted to include resistances to the following types of corrosion and mechanical damage:

1. Pitting in the presence of acid chlorides or caustic.
2. Sulfur- or caustic-induced intergranular attack.
3. Caustic stress corrosion cracking.
4. Erosion from micron-sized particle impingement.
5. Abrasive wear from platelets of debris composed of magnetite (iron oxide).
6. Corrosion with or followed by low-stress high-cycle fatigue.

The sleeve design must comply with the structural requirements of the ASME Code for the 40-year design life of the OTSG. This requires that the tube fatigue analysis be reevaluated for the effect of sleeves on the flow-induced vibration of the tubes. This analysis must consider both severed and unsevered tubes due to the worst-case assumption that the degraded area of the tube bridged by the sleeve may not be structurally sound.

The sleeve must be shown to have an adequate pressure boundary thickness according to methods outlined in the ASME Code, for installation in a severed or unsevered tube. The OTSG design conditions are listed in Table 2-3. An ASME stress analysis report is required to certify adequacy of the sleeve.

2.3. Test Plan

qualification specimens were conditioned and tested to demonstrate the strength and leakage adequacy of the sleeve design in any OTSG, both before and after service conditioning to represent 40 years of service. In order to simulate the 40-year service life, the functional specifications for all 177-fuel assembly (FA) NSS systems were evaluated with regard to the quantity and severity of design transients. The conditions that a tube sleeve joint would experience were then arranged as thermal cycles, axial load cycles, pressure cycles, and vibration cycles. The expected axial loads and service cycles were factored to establish test axial loads and cycles as required by ASME Code Section III, Appendix II, "Experimental Stress Analysis," in order to account for the effects of statistical variations, cycle rate, and test temperature. The resulting service conditions which were applied to the specimens are presented in Table 2-4.

In actual service, the sleeved tubes would simultaneously incur thermal, axial load, pressure, and vibration cycles, but the specimens were independently subjected to pressure and vibration cycles in order to assess their individual effects. Control specimens were also processed separately with no conditioning to represent as-installed pre-service sleeve performance. The largest set of samples was subjected to both thermal and axial load cycles as an expedient because the effect of the thermal cycles was expected to be minor. The specimens are identified in Table 2-5, and the processing is diagrammed in Figure 2-3.[3,4]

2.4. Supplementary Tests

Corrosion resistance was demonstrated by cold-worked samples and control samples which were maintained in a 10% sodium hydroxide solution at 550F in an autoclave with a 190 mV potential Each 24-hour day at these accelerated test conditions represents approximately one year of service in an OTSG.

A metallurgical examination of the samples after the autoclave exposure permitted the comparison of stress corrosion cracking found in the various cold-worked samples with the unworked samples. The comparison permits a conclusion regarding the effect on corrosion resistance due to the cold-working by roller expansion or bending and straightening.

A group of free-span joint specimens was leak and pull tested in order to evaluate the effect of the tube surface on the joint strength and leakage.

Four complete samples were assembled to determine whether the sequence of roller expansion has any significant effect upon the quality of the rolled joints or the strain in the sleeve and tube.

sleeves were bent and straightened to represent the worst-case (i.e., most curved) sleeves which would be inserted into tubes. These sleeves were assembled in tubes with tubesheets and then leak and pull tested to determine whether the bending and straightening affects the quality of the rolled joint.

The effect of rolling sleeves in adjacent tubes was demonstrated by rolling plugs in a sample tubesheet test block.

2.5. Analyses

The effect of sleeve installation on the performance of the steam generators was analyzed for flow restriction and heat transfer capacity. In order to scope the worst case, it was assumed that as many as 10,000 80- inch long sleeves are installed in one OTSG. Temperature distribution and effect on steam outlet temperature were also analyzed.

The likelihood of flow-induced vibration of a sleeved tube was evaluated. It was assumed that an 80-inch or a 30-inch sleeve was installed in any tube in the generator, and that the tube may be severed or unsevered. The sleeve adds mass which tends to reduce the natural frequency, but it also adds stiffness which tends to increase the damping. The net effect of the sleeve is not obvious, and all combinations were considered to assure that the worst case was identified.

A sleeved severed tube was analyzed to confirm compliance with Nuclear Regulatory Commission (NRC) and ASME Code requirements. This included the calculation of required thickness for normal operating loads, faulted conditions, and primary plus secondary (thermal) stresses according to ASME Section III Appendix F, and NRC Regulatory Guide 1.121. The resulting minimum sleeve wall requirement establishes a degradation limit which indicates when a sleeved tube must be plugged or removed from service.

The potential for a sleeve to collapse from external pressure was evaluated. The bending and straightening operations may leave the sleeve with a slightly oval cross section. Under some accident conditions, there could be a secondary-to-primary pressure difference. If a sleeved tube is severed, an oval sleeve could conceivably be subjected to this external pressure. If a leaking tube subsequently self-seals, water trapped between the sleeve and tube may also tend to collapse an oval sleeve. The consequences of a collapsed sleeve were also evaluated.

The variables of the roller expansion process were analyzed to assure that adequate roller expansions could be reliably reproduced.

The result was an engineering requirements document which established limiting conditions for the rolling process which are dependent upon the dimensions of the tube to be sleeved.

Table 2-1. Load Conditions

Loading case	Static	Transients	Seismic and LOCA
Design	DW	DP	OBE
Level A/B	DW	OP, ML, HL (consideration of all specified level A/B thermal transient loading combinations)	OBE
Level D 1	DW	OP, HL	SSE
2	DW	OP, HL	LOCA

DP -- Design pressure
DW -- Dead weight
ML -- Mechanical loads
HL -- Hydraulic loads

OP -- Operating pressure
SSE -- Safe shutdown earthquake
OBE -- Operating basis earthquake

Table 2-3. Steam Generator Design Parameters

	Primary side	Secondary side
Design pressure, psig	2500	1050
Design temperature, F	650	600
Full-load flow, 10^6 lb/h	65.6	5.6
Full-load operating pressure, psia	2200 (OTSG inlet)	925 (Steam outlet)
Full-load inlet temperature, F	604	460
Full-load outlet temperature, F	554	570
Full-load pressure drop, max. psi	33	50

3. RESULTS AND CONCLUSIONS

3.1. Leakage Tests

The expected leakage per sleeved tube was determined by measuring the leakage of each sample over a range of axial loads, fitting regression lines to the observed data, and factoring the control or as-installed specimen leakage by the ratio of service cycle leakage to control leakage. This results in an end-of-life leakage curve that can be compared to the as-installed leakage curve. The maximum expected leakages for normal operation and accident conditions are read from these curves at the appropriate axial loads. They are conservative because the samples were made using low-yield tube material, which is the worst case, and conservative applied service cycles.

The mean value of measured leakage for each applied axial load is plotted in scatter diagrams in Figure 3-1. The control-, axial/thermal-, pressure-, and vibration-cycled groups of specimen data are each represented by a linear regression line fitted by minimizing the sum of the squares of deviations from the line.

Leakages for zero axial load were not available because internal hydrostatic pressure on the tube end- plugs created an axial load of almost 400 lbs.

The regression lines of Figure 3-1 indicate that leakage is

Table 3-1, these relationships have been expressed as factors that are ratios of conditioned leakage to control leakage at each test load. The cumulative effect of all the conditioning could be expressed as the product of these three factors, but the healing effect of cycling has not been confirmed in conjunction with vibration cycling. Rather than underestimate the predicted leakage, all ratios of less than 1.0 were arbitrarily increased to 1.0 in order to assure conservatism. The resulting overall service factors and cumulative leakage after 40 years' simulated service are presented in Table 3-1 and plotted in Figure 3-2. Within the range of normal operation the maximum leakage is well within the 2.5 ml/h average leakage objective.

The tightness of a rolled joint is highly dependent upon the yield strengths of the materials used. In this qualification, both tubes and sleeves were type 600 Inconel per ASME Specification SB-163, which permits a minimum yield strength of 35,000 psi. All sleeves were made from a single heat of material which had a yield strength of

Figure 3-3 includes a scatter diagram of the mean measured leakage of these high-yield samples at various axial loads and a linear regression line fit to these points. A comparison with the regression line for the low-yield tubes, also plotted in Figure 3-3, confirms that the stronger tubes are also much tighter. Table 3-2 tabulates specific values from these two curves and the ratios of these leakages are used as factors to predict the leakage of high-yield service-conditioned sleeve joints at various axial loads.

Two samples were tested to assess the effects of test temperature on tube leakage. When the specimens were maintained at 388F, the mean leakage with no axial load was At the ambient temperature of 67F, the corresponding mean leakage was Although it was not practical to leak test at the full design temperature, the hot test results show Since both sleeve and

In service, the annulus between sleeve and tube may tend to insulate the tube so that the sleeve is hotter

Samples were tested to evaluate the effect of tube surface condition on the quality of the joint.

The measured leakage of these specimens is tabulated in Table 3-3. The tube samples all show very little leakage with no significant differences The mean leakage

Space limitations require that the sleeves be bent into a gentle arc outside the OTSG and straightened as they are fed into the tubes.

Two sample sleeves which had been bent and straightened were expanded into tubes and leak tested to verify that the insertion process does not degrade the quality of the expansion joint. The measured leakage is tabulated in Table 3-4. As these expansions were made in high-yield tubes, the leakage in similar high-yield samples which had not been bent and straightened is also listed for comparison.

The measured leakages are quite low at all axial loads,

The axial load on a tube during operation is a function of the pressures, the position of the tube in the OTSG, and the tube and shell temperature difference, which in turn is a function of the service transient. For normal operation, the transient that results in the greatest total tube load is a cooldown from 15% power. During this transient, the load on a center tube reaches 649 lb while the load on a peripheral tube reaches 1107 lb.[12] The worst accident condition is a main steam line break (MSLB), which results in a 3140-lb total load on a peripheral tube of high-yield strength,[12] or 2620 lb if the tube has a low-yield strength, according to Figure 2-2. For a central tube, the maximum load of 1585 lb is a result of a loss-of-coolant accident (LOCA). These loads are plotted in Figure 3-4 against expected leakage curves for high- and low-yield tubes. This indicates that under maximum operating load, the greatest expected leakage for a peripheral tube would range from

It also indicates that under maximum accident load, the greatest leakage for a peripheral tube

In order to predict the leakage in a sleeved OTSG, it is necessary to make some assumptions regarding the location and yield strength of the sleeved tubes. Figure 3-5 shows the radial distribution of upper tubesheet tube eddy current indications at a typical plant.

In Figure 3-4, dash lines are shown connecting these mean loads between the high- and low-yield leakage lines. It is reasonable to assume that the yield strengths of all of the tubes in an OTSG are

for a typical OTSG in Table 3-5. Thus, the predicted maximum leakage for

under normal operating loads

If there were 10,000 sleeved tubes in a plant (and all of the tubes leaked), the predicted leakage in normal operation would be

of the usual Technical Specification plant shutdown limit. For accident conditions, the rate would be

These predicted leak rates are conservative for normal expansions. It is difficult to quantify the amount of conservatism, but factors that contribute to the overall conservatism are as follows:

1. It is assumed that all the sleeved tubes have through-wall defects. If a degraded tube remains leak-tight, sleeve leakage would not contribute to overall primary-to-secondary leakage.

2. The maximum operating load is not continuous. It occurs only during

   The axial loads during steady state and the lesser transients are much lower.

3. The leakage at is considerably less than However, all of the leakage predictions are based upon

4 tube surfaces tend to result in tighter rolled joints, but some of the qualification tests used

In the event of a complete failure of a rolled joint, such as a full circumferential tube crack at the lower roll transition, the tube and sleeve have been designed to remain engaged under worst-case accident conditions The maximum leakage which could occur in such a failed tube has been calculated as For comparison, the maximum leakage for an unsleeved ruptured tube has been calculated

3.2. Joint Strength Tests

The expected joint strength was determined by measuring under a range of axial loads, fitting regression lines to the observed data, This results in an end-of-life curve which can be compared to the as-installed curve.

The mean value of measured each applied axial load is plotted in scatter diagrams in Figure 3-6. The groups of specimen data are each represented by a linear regression line fitted by minimizing the sum of the squares of deviations from the line. The sample data are also plotted in Figure 3-7

The regression lines of Figures 3-6 and 3-7 indicate that joint strength is reduced

the individual conditioning samples at each axial load is tabulated in Table 3-6 and plotted in Figure 3-7. This represents the cummulative effect of all the conditioning.

The strength of a rolled joint is dependent upon the yield strength of the materials used. In this qualification, most of the tubes were made from

All sleeves were made from material. Figure 3-8 includes a scatter diagram of the samples at various axial loads and a linear regression line to fit these points.

Table 3-7 tabulates specific values from these curves,

Samples were tested to evaluate the effect of tube surface condition on the strength of the joint. Table 3-8 reports the various surface samples at several axial loads.

tube joints made with sleeves that had been bent and straightened were measured at various axial tube loads. The results are presented in Table 3-9 and compared to joints where the sleeve had always been straight.

The expected displacement of the rolled joint after 40 years of service is compared to the design requirement in Figure 3-9. The acceptance limits for high- and low-yield tube material from Figure 2-2 have been replotted, and the maximum operating loads for center and peripheral tubes have been added in the manner of Figure 3-5.

It is evident that the expected joint is well within the acceptance limits for both maximum operating and accident loads.

The are based upon low-yield tubes on the outer periphery of the OTSG undergoing the maximum cooldown rate or maximum hypothetical accident conditions. They also presume that the sleeved tube is totally severed so that the rolled sleeve joint must carry the entire axial load on the tube. In actual practice, the tube would normally share the load with the sleeve because a severed tube is quite rare. As previously discussed in section 3.1,

Thus, it would be a relatively rare occurrence for a sleeve in a low-yield peripheral tube to be subjected to the full maximum operating loads. However, when this happens, the rolled joint is capable of carrying the load without failure.

Although no acceptance criteria were established for the ultimate failure of a rolled joint, most of the test specimens were pulled to failure. These failures included

failed specimens had a mean failure load

maximum load that a joint could experience under accident conditions.

3.3. Light Expansion Tests

The tightness of a rolled joint depends upon the amount of roller expansion. In the qualification tests, all sleeves which were rolled at the tubesheet end had a

regardless of tube dimensions and strength. The free-span expansions were

Figure 3-10A plots test data which confirm

The amount of the freespan diameter increase depends on

In the event that

the rolled joint may have less strength and leak more.

assess the effect of light expansion. Figures 3-11 and 3-12 show the scatter diagrams of the light expansion mean measured leakage and joint strength and the regression lines for both light and normal expansion. Tables 3-10 and 3-11 tabulate specific values from these curves, and the

determine the leakage and strength of service-conditioned lightly expanded joints at various axial loads.

The effect of the amount of roller expansion on the quality of the joint is also shown in Figures 3-13 and 3-14,

Again,

they show that

they clearly indicate that the effect of light expansions on both leakage and displacement is

The sensitivity of the joint to light expansions is

The roller expansion tool

However, these dimensions vary as illustrated by the sample distributions of Figure 3-15. The normal expansion that has been qualified

The effect of light expansions on an OTSG can be estimated by applying factors to the expected portion of light expansions. Figure 3-16 indicates

Figure 3-17, which is a modified version of Figure 3-10, shows within the strength acceptance limits. If the expected leakage curves of Figure 3-4 are

as shown in Figure 3-18. As described in section 3.1, these

the predicted leakage for the total plant would be of the usual plant Technical Specification limit. Under the worst normal operation loads, the expected mean leakage

3.4.1. Corrosion Tests

Accelerated stress corrosion cracking tests were performed on sleeved tube mockup specimens to determine whether residual stresses from the sleeving process are sufficient to cause stress corrosion cracking of the sleeve or the OTSG tube. The mockups were rolled at the tubesheet joint,

In the free-span expansion, the tube and sleeve walls were respectively, which is greater than expected in a normal expansion. In the tubesheet expansion, the tube was in the initial roll and the sleeve was by the sleeve rolling, which is the normally expected amount at Specimens were removed from the tubesheet sleeve joints, free-span sleeve joints, and bent and straightened sleeve samples and mounted in autoclaves along with control specimens from both U-bend and virgin sleeves and tubes. In the autoclaves, they were exposed to a 10% sodium hydroxide solution at 550F with a +190 mV applied potential

The results of the corrosive tests are listed in Table 3-12.

All U-bend control specimens cracked which demonstrates that the test was rigid enough to produce severe cracking in highly strained specimens.

Use of the accelerated caustic corrosion test results to predict failure in service with all-volatile-treated (AVT) water is based

In autoclave tests, the time to crack initiation was measured at various strains for specimens subjected to 650F AVT water and 550F 10% caustic solution with an applied electric potential.

a linear curve was fit to define the relationship between failure times

Although the data are limited, this correlation indicates that a

These correlations are for well controlled AVT water and may not apply to other water treatments or water with impurities which may be present in steam generator operations.

Actual steam generator water chemistries and loading conditions may reduce this prediction

3.4.2. Flow-Induced Vibration Analysis

The effect which a sleeve has on the vibration characteristics of a tube is not obvious because the sleeve stiffens the tube and tends to increase the system damping. However, the additional mass tends to reduce the natural frequency. The characteristics of the sleeved tube depend upon

These averaged curves are plotted in Figure 3-20.

Finite element models were set up with different assumed rotational spring constants at the lower face of the tubesheet in order to match the computed and the observed frequency in the mockup. The modal frequencies compare as follows:

	Computed Hz	Mock-up Hz
Unsleeved tube		
Sleeved tube		
Sleeved tube severed at tubesheet		

The response plots from the in-air tests (Figure 3-20) were used to determine approximate critical damping ratios These values were rounded off to conservative test damping values, and then more conservative operational damping values for the analysis were obtained

These damping values are as follows:

	Half-power method	Test (rounded)	Assumed for analysis
Unsleeved tube			
Sleeved tube			
Sleeved tube severed at TS			
Sleeved tube severed at TSP			

A NASTRAN finite element model was used for the analysis with a

at the tubesheet face and damping as listed above. Secondary side crossflow velocities of OTSGs with both internal and external auxiliary feedwater headers were evaluated. The resulting worst-case generic fluid-elastic stability margins and random vibration and vortex shedding responses are as follows:[17]

	Unsleeved tube	Sleeved tube	Sleeved tube severed at TS	Sleeved tube severed at TSP
Minimum fluid-elastic stability margin				
Random vibration response				
max. displacement rms, in.				
max. stress rms, psi				
Vortex shedding response				
max. displacement, in.				
max. stress, psi				

Therefore, it is concluded that flow-induced vibration will not be detrimental in any OTSG tube sleeved in the upper span, even if the tube is completely severed anywhere between the upper tubesheet and the number 15 tube support plate.

3.4.3. Strain Tests

Specimens were processed to evaluate the sleeve and tube elongation due to sleeve rolling. During the tubesheet roll, the sleeve is

3.4.4. Adjacent Tube Tests

There was some concern that rolling a sleeve in a tube adjacent to a tube which previously had been sleeved may loosen the first expansion, causing increased leakage.

it is conservative when applied to rolled sleeves in the OTSG tubesheet.

3.5. Analysis

3.5.1. Performance

The installation of a significant number of sleeves in the OTSG could reduce the OTSG's thermal performance due to the insulating effect of the sleeve (especially the annulus between sleeve and tube) and the change in primary flow distribution caused by higher flow resistance. The net result of these effects

The analysis of thermal and hydraulic effects assumed that 5000 80-inch long sleeves were installed in the peripheral tubes of each OTSG. These worst-case assumptions reduce primary flow by

The effect of this reduction in superheat temperature on plant operation is considered to be minimal. The first OTSG put into operation was warranted to produce a minimum of 35F superheat steam at full power,

The new operating point for the OTSGs, turbines, and feedwater control system would be

3.5.2. Structural

The minimum acceptable wall thickness for degraded sleeves was determined in accordance with the allowable stress and pressure limits of ASME Section III and NRC draft Regulatory Guide 1.121. Primary membrane stress, burst pressure, and fatigue analysis were considered for normal operation, and primary membrane stress, burst pressure, collapse pressure, and primary membrane plus bending stresses were considered for postulated accident conditions. In addition, primary plus thermal stresses were evaluated. The minimum sleeve wall thickness was calculated for these eight different acceptance criteria. For the expected type of defects, the greatest required minimum wall was found to be

a 70% through-wall defect would require that a sleeve be removed from service. This compared to a 69% defect limit for the OTSG tubes.

The sleeve must be bent and straightened for installation in the outermost OTSG tubes. This results in a slightly elliptical cross section, which was evaluated for buckling pressure. The maximum expected ovality (i.e., difference in extreme ODs at any one cross section) was found to be inch based on sample dimensions.[2] The critical external pressure depends on the material yield strength.

Under the maximum secondary pressure of 1050 psi with no primary pressure, neither tube nor sleeve would collapse.

In the Obrigheim steam generator, tube blisters were found inside the tubesheet between tubesheet rolls. This plastic deformation of the tube was attributed to water that had leaked through roll transition cracks and become trapped in the annulus between the tube OD and the hole ID by corrosion products. Upon heatup, the water expanded more rapidly than it could leak out, causing the tube deformation. Should water be trapped in a similar manner between the OTSG sleeve OD and tube ID,

the annular pressure increase is more likely to blow out the corrosion products which plugged the leak than to collapse the sleeve. Thus, the likelihood of sleeve collapse is very small.

3.5.3. Process Control

The parameters of the free-span expansion were evaluated to assure that the amount of expansion can be controlled within a qualified range. A general equation was developed to define the relationships among tube, sleeve, and expansion tool parameters before and after expansion.[13] This equation is:

This process was qualified for a range

Most of the qualification samples were rolled to represent the normal range of expansion. to evaluate the effect of light expansion on leakage and strength, and another set was rolled at an average to evaluate the effect of cold working on corrosion resistance.

The range of normal expansion is while the range of

assure that any tube diameter will be expanded within the qualified range.

4. SUMMARY AND RECOMMENDATIONS

Mechanical tube sleeves have been qualified for use in degraded OTSG tubes by a series of tests and analyses. The design is strong enough and sufficiently leak-free to be used as a permanent remedy to keep degraded tubes in service. The criteria for this qualification have been summarized in Table 4-1 along with the results, which show that all the criteria are satisfied.

It is recommended that up to 10,000 of these mechanical sleeves be installed in the OTSGs of any plant as needed to correct or prevent tube degradation which would otherwise require that the tube be removed from service.

Table 4-1. OTSG Sleeve Qualification Criteria

Sleeving criterion	Justification	Results	Reference section
Installation			
Capable of installation in any tube in an OTSG at depths up to 80 in. below the top of the tubesheet.	Degradation has been found in tubes across the face of the OTSG, concentrated at elevations in or near the upper tubesheet and at the 15th tube support plate.	Tooling designed to permit the installation of an 80-in. sleeve in any tube in an OTSG.	2.1
Leakage			
Average leakage of no more than per sleeve under normal operating conditions.	10,000 sleeves with this average leak rate could be installed in a plant, and the overall steam generator primary to secondary leakage would be only one-tenth of the leakage which requires plant shutdown.	The average leakage for a sleeved tube under maximum operating loads was found to be under maximum loads.	3.1
Strength			
Pullout strength of at least	Accident conditions could load a sleeve up to the yield limit of the tube in which it is installed	No joints failed at loads below the tube yield strength,	3.2
Process Control			
The sleeve installation process must be controllable so that predictable joint quality is maintained.	The limits of the amount of expansion must be established in order to adequately control the expansion process.	light expansions are expected, and these would have adequate strength	3.3
Corrosion			
The corrosion resistance of the installed sleeve is to be at least as good as that of the original OTSG tubes when subjected to conditions expected in the OTSG.	The tube/sleeve assembly is required to remain functional in the OTSG.	Accelerated corrosion tests representing primary and secondary coolant tests indicated sleeved tubes will not crack	3.4.1

Table 4-1. (Cont'd)

Sleeving criterion	Justification	Results	Reference section
Vibration			
A sleeved tube, either severed or unsevered, is to be adequate for the 40-year design life as confirmed by an ASME III fatigue analysis.	Tube/sleeve integrity is required under normal operating conditions.	The sleeved tube, whether severed or not, has a smaller maximum displacement and a greater fluid-elastic stability margin than the unsleeved tube, and the maximum stresses are well below allowable.	3.4.2
Tube Strain			
Sleeving installation should not leave a tube in compression or permit tube to tube contact.	Differential expansion in postulated accident conditions must not permit adjacent tubes to contact each other.	Sleeving will increase the tension in a tube. Maximum slippage would not bow the tube in heatup enough to contact an adjacent tube.	3.4.3
Adjacent Sleeves			
Sleeve installations must not damage a prior sleeve installation in an adjacent tube.	Quantity sleeve installations must not be restrained by sequence or location limitations.	Adjacent sleeve installations are unaffected by a new sleeve installation.	3.4.4
Performance			
The effect of sleeving up to 10,000 tubes must have a tolerable effect upon plant performance.	The thermal/hydraulic effects of sleeving must be acceptable.	10,000 sleeved tubes would reduce primary flow and reduce full power steam superheat The effect of this would have a minimal effect on plant operation.	3.5.1
Code			
Tube/sleeve funtional integrity must be maintained under stress and pressure limits of ASME Section III and NRC Regulatory Guide 1.121.	Tube/sleeve integrity is required under postulated accident conditions.	The minimum required sleeve wall for normal and accident conditions permits sleeve defects less than 70% through-wall.	3.5.2
Collapse			
A sleeve must not collapse under external pressure.	Sleeve collapse in a leaking tube would create a primary to secondary leak path.	The sleeve is about as strong as the tube under external pressure, and collapse is unlikely.	3.5.2

5. REFERENCES

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17. Flow-Induced Vibration Analysis of Sleeved Tubes in the 177 OTSG, B&W Document 32-1151170, Babcock & Wilcox, Lynchburg, Virginia, June 1984.