Effect of Out-of-Plane Denting Loads on the Structural Integrity of Steam Generator Internals

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Research Project S169-1

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EPRI PERSPECTIVE

PROJECT DESCRIPTION

Pressurized water reactor steam generator tube denting results in tubes being locked into the support plates. This results in lower tube natural frequencies, greater susceptibility to tube buckling, and out-of-plane loading of the support plates and the tubesheet. RPS169-1, for which this is a final report, is one of several existing or anticipated Steam Generator Project Office projects which are evaluating the structural effects of denting on steam generators.

PROJECT OBJECTIVES

This report presents the results of an investigation into the effects of steam generator tubes locked at support plates due to denting. The report delineates the techniques and equations used, and identifies the conservatisms involved in the analysis.

PROJECT RESULTS

The results indicate that locked tubes do not produce unacceptable induced-tube or out-of-plane stresses, excessive tube vibration, or tube buckling.

This report should be of interest to steam generator designers and utility personnel dealing with the effects of dented steam generators.

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ABSTRACT

This report presents the results of a structural analysis of steam generator tubes, tube support plates and tubesheet when tube denting has caused tubes to lock into support plates. A section of a support plate with associated tubes was modeled by finite element techniques to determine the maximum axial load in the tubes imposed by differential thermal expansion. The tube structural integrity was addressed from the standpoint of flow induced vibrations and fatigue for the imposed axial loading. The maximum stress in the perforated plate due to locked tubes was determined by finite element methods. The axial loads in the tubes were applied to a tubesheet finite element model to determine the primary bending stresses. Axial loads in the tubes were applied as a concentrated load through the tubesheet thickness to determine maximum shear. The stresses from these analyses were compared to the ASME Code Section III allowables and were found to be acceptable.

CONTENTS

ILLUSTRATIONS

NOMENCLATURE

A	Threshold instability constant (model test)
A_1	Area of tube
A_2	Shear areas in tubesheet ligaments
A_3	Shear lug area
A_t	Area of tube material
A_s	Inside area of tube
A_p	Shear area of support plate
A_f	Outside area of tube
C_1	Tube flow constant
C_m	Virtual mass coefficient
C_L	Tube coefficient of lift
d	Diameter of tube
E^*	Effective modulus of elasticity
E	Modulus of elasticity
F_f	Uniform flow load
F_t	Axial load
F_{max}	Maximum load in tubesheet due to locked tubes
f_n	Tube natural frequency
g	Acceleration of gravity
h	Ligament width between tubes
I	Moment of inertia
k	Stress multiplier
K	Tube constant

L'_F	Length of tube exposed to cross flow
L	Length of tube
L_1	Length of tubesheet to which load is applied
M_o	Tube virtual mass
N	Transient cycles
N_1	Number of tubesheet ligaments
N_2	Number of shear lugs in generator
P_o	Pressure outside of tube
P_I	Pressure inside the tube
P	Pitch between holes
R	Radius to maximum tubesheet applied load
R_o	Outside radius of tube
R_I	Inside radius of tube
SI	Stress intensity
S	Stress at one-half ultimate
S_a	Alternating stress
S_m	Membrane stress allowable
t	Thickness of tube
t_{ts}	Tubesheet thickness
U	Usage factor
V	Maximum vertical load
V_c	Critical flow velocity of tube
V_g	Velocity of fluid in the gap
W	Uniform load on tubesheet
$\tilde{W}$	Virtual mass
ρ	Equivalent density
ρ_t	Density of tube material
ρ_s	Density of fluid in tube

ρ_f	Density of displaced fluid
δ_o	Logarithemic decrement ($\delta_o = 2\pi\zeta$)
δ	Tube deflection
ζ	Damping ratio
α	Coefficient of thermal expansion
ν	Poissons ratio
ΔT	Temperature change
β	Biaxiality ratio
τ	Shear stress
$\sigma_{x,y}$	Principle stesses
σ_s	Skin stress due to thermal gradient
σ_r	Radial stress

SUMMARY

The purpose of this study was to determine the structural integrity of steam generator tubes, support plates and tubesheet when tube denting causes tubes to lock into support plates. The loads considered in this analysis were the result of a thermal expansion mismatch between the Inconel tubes and the carbon steel structure subsequent to tubes becoming locked into support plates due to tube denting. The resulting loads in the tubes could be either tensile or compressive depending on whether the tubes are locked in cold or hot. Each of these conditions were considered in this analysis.

A portion of the support plate with associated tubes was modeled by finite element techniques to determine the maximum compressive load in the tubes. The maximum compressive load in the tubes was evaluated for (a) tube buckling, and (b) tube span natural frequency. The effect of (b) was considered in the evaluation by fluid elastic coupling and low level vibration (fatigue) in the cross flow region at the bundle entrance.

An axial load in the tubes imparts an out-of-plane load to the tube support plates and to the tubesheet. A finite element model of the tubesheet was developed to determine the membrane plus bending stresses. The stresses from this model were combined with the tubesheet design stress and compared to Section III Code allowables. The maximum load required to shear the steam generator shell to shroud shear lugs was applied to a tubesheet model and the resulting stresses compared to Code allowable. An elastic analysis of the support plate with loads applied out-of-plane, produced stresses which do not represent the maximum stress distribution in the plate. These stresses must be combined with the stresses due to in-plane loading in order to calculate a maximum bending stress in the plate. This combination would require an additional analysis of the plate which is beyond the scope of this report.

The results of this analysis indicate that the tube and tubesheet stresses do not exceed the appropriate allowables of Section III of the ASME Boiler and Pressure Vessel Code.

Section 1

INTRODUCTION

Tube denting has been widely reported in operating steam generators which have carbon steel drilled support plates. Denting can cause tubes to lock into support plates causing a significant axial load in the tubes at the shroud to support attachment points. The tube axial load also imparts an out-of-plane load to the tubesheet and support plates. This report presents the results of a structural analysis of the effects of these loads on the tubes, support plates and tubesheet.

An evaluation was conducted on two regions within the steam generator, (1) from the tubesheet to the support plate number 2, and (2) from support plate number 7 through two eggcrates to support plate number 10. The region which developed the largest axial load due to locked tubes is region 1. The tubes in this region were evaluated for buckling, fluid/elastic coupling, fatigue and low level vibrations.

A finite element analysis of the support plate and tubes in region 1 was conducted to determine the maximum axial load in the tubes and the stress in the support plate. Also of concern is the effect of locked tubes on the structural integrity of the steam generator tubesheet. Two cases are evaluated, (1) load due to locked tubes, and (2) a maximum lower shell to shroud shear lug load. Case number 2 will not physically occur in a steam generator and was considered in this report as a conservative scoping analysis. These loads were individually applied to a finite element model of the tubesheet at a location near the outer tube row. The loads are conservatively considered acting as a concentrated load per unit circumference around the tubesheet. The tubesheet primary and shear stresses are evaluated and compared to defined allowables.

Section 2

TUBE STRUCTURAL ANALYSIS

FINITE ELEMENT MODELS

A finite element analysis was conducted using the ANSYS Computer Program developed by Swanson Analysis Corporation, Reference 4. A model was developed and is depicted in Figure 2-2 to represent tubes locked or fixed at each end. The model shown in Figure 2-2 is used to analyze two regions within the steam generator as depicted in Figure 2-3, (1) from tubesheet to support plate number 2, and (2) from support plate number 7 through two eggcrates to support plate number 10. The major difference in the two regions is the length of tubes, lateral supports, (support plates and eggcrates), temperature of the tubes and the secondary side fluid densities. These regions were selected to determine the worse case location due to crossflow velocity, temperature and tube length.

The model shown in Figure 2-2 represents a small portion of a uniform plate with a solid rim and perforated (equivalent) plate. The equivalent plate properties for the model were determined from Reference 1, Article A-8000. The plate is modeled as flat quadrilateral shell elements and the tubes as beam elements. At the shroud attachment point, represented by nodes 427, 428, 411, 412 and 413, the model is fixed in all six degrees of freedom. Along the boundaries at the left and right sides of the figure represented by the nodes 1 through 400 and 21 through 424, the model is fixed in the Y (tangential) direction and rotations about the X and Z axix. Tubes intersecting the plate are modeled as three dimensional beams, attached at the support plate and fixed in all six degrees of freedom at the specified tube length. As noted in the figure, the tube cross-sectional areas are lumped together as effective tubes, in order to simplify the model.

In Region 1 the tube length is 48 inches and in Region 2 the length is 82.5 inches. The design average tube temperature in Region 1 is 559°F while in Region 2 it is 545°F. The shroud temperature is 504°F. These values were applied to the tubes in the model, Figure 2-2. A maximum compressive load of 2300 pounds was developed in the tube nearest to the support lug (Region 1) and 1762 pounds in Region 2.

TUBE NATURAL FREQUENCY

Analytical models shown schematically in Figures 2-4 and 2-5 represent single tube row models for two regions of the steam generator. Region 1 consists of tube fixed at the tubesheet and support plate number 2 (Figure 2-3). Region 2 is fixed at support plate number 7 and 10 and lateral restrained at eggcrates 1 and 2. In order to determine the natural frequency of the tubes, tube equivalent density is calculated for each region. The equivalent density consists of the density of the tube, density of the fluid internal to the tube and the density of the external fluid that is forced to move as the tube vibrates. The equivalent density ρ, is formulated from

$$\rho = \frac{1}{A_t} \left(\rho_t A_t + \rho_s A_s + C_m \rho_f A_f\right)$$

	Region 1	Region 2
where,	$\rho_t = 0.305$	0.305 lbs/in^3
	$\rho_s = 0.02604$	0.02604 lbs/in^3
	$\rho_f = 0.02836$	0.01040 lbs/in^3
	$A_t = 0.1059$	0.1059 in^2
	$A_s = 0.3354$	0.3354 in^2
	$A_f = 0.4418$	0.4418 in^2
	$C_m = 1.7$	1.7

A virtual mass coefficient C_m was determined from Reference 3 and applied to the vertical tube spans under consideration. The resulting mass density of 0.001524 lbs-sec^2/in^4 was calculated for Region 1. At Region 2 the tube mass density is 0.001194 lbs-sec^2/in^4.

This virtual mass coefficient used in this study was determined using the relationships developed in Reference 3. This coefficient, which is sometimes called the added mass factor or the inertia coefficient, is a function of the physical properties of the fluid external to the tube as well as the proximity and geometry of adjacent solid structures. For CE hexagonal array geometry and typical secondary fluid properties near the tubesheet the virtual mass coefficient was calculated to be $C_m = 1.7$ and was applied to the vertical tube spans under consideration.

The coefficients obtained from Reference 3 are reproduced as Figure 2-8. These values (the two lower curves) were obtained from natural frequency measurements made on a tube surrounded by rigid rods as shown on the left hand side of Figure 2-8. The values obtained were contrasted with measurements taken for the same tube but without surrounding tubes where the closest boundary was a distance of ten inches from the tube of interest. Also presented in Figure 2-8 (the two upper curves) are data obtained from an earlier study (Reference 2) which simulated a tube array by placing a concentric cylinder around the measured tube. By comparing the two sets of curves, it is obvious that the cylinder siumlation of a tube array is overly conservative. This is most probably due to the lack of "ventilation" which exists between the tubes of a real tube array but is absent in the concentric cylinder simulation.

Some concern might exist that due to the proximity of the shroud to the tubes of interest in this study that the virtual mass coefficient could be increased. In fact, however, due to the necessity to have a solid rim around the support plate and to provide space for attachment blocks between the plate and shroud, there is a gap of two inches between the shroud and the outside of the tube bundle for CE designed steam generators. There is a similar gap between the shroud and tube bundle for all operating Westinghouse units, however, the current design with 11/16 inch O.D. tubing has a considerably smaller gap. The effect of any gap would be to make the calculated virtual mass factor of $C_m = 1.7$ conservative for this region.

The models developed above were used to determine natural frequency versus tube axial load as presented in Figures 2-4 and 2-5. As shown in Figure 2-4, the natural frequency of a tube in Region 1 is 30.0 Hertz for an axial compressive load of 2300 pounds. The natural frequency for a tube with a tensile load of 2300 pounds is 69.2 Hertz. For Region 2 the natural frequency is 44.0 Hertz for an axial compressive load of 1762 pounds. For the remainder of this report, Region 1 with an applied compressive load of 2300 pounds and a natural frequency of 30.0 Hertz will be evaluated.

TUBE BUCKLING

Critical buckling of a tube occurs when the frequency of the tube reaches zero due to an applied compressive load. Any additional loading causes buckling of the tube. As shown in Figure 2-4 an axial load of 2300 pounds is below the critical buckling load of 3330 pounds. For Region 2, shown in Figure 2-5, the axial load of 1762 pounds is below the critical buckling load of 2965 pounds. Therefore, no tube buckling is expected due to heat up of a tube in a locked position.

During the fabrication of the steam generator, small scale misalignment occurs when the tubes are placed through the support plates into the tubesheet. In the installation of the tubes a misalignment of greater than 1/4 inch cannot occur without impairing the insertion of the tubes. The misalignment or eccentricity in this analysis will be conservatively considered to be 1.0 inches. A finite element model of the tube in Region 1 was developed and a uniform load of 0.748 lb/in. was applied over sixteen inches of the tube measured from the tubesheet to the bottom of the shroud. An axial compressive load of 2300 pounds was also applied along the tube centerline. No tube buckling occurred due to the eccentricity and loads developed above.

FLUID ELASTIC COUPLING

Fluid elastic coupling occurs when sufficient transverse flow velocity exists to put the tube in a motion which leads to establishment of a tube/fluid feedback mechanism. This causes increased amplitudes of vibration until either a balance is reached between fluid energy absorbed and energy dissipated through damping by the tube or impacting of tubes ensues. The following expression from Reference 5 was used to evaluate fluid elastic coupling at the downcomer to bundle entrance region.

$$V_c = (K)\ (f_n)\ (d) \left[\frac{M_o\ \delta_o}{\rho_f d^2}\right]^{1/2}$$

where, $K = AC_1 = 10.6597$

$C_1 = 2.479$ (Reference 5 for flow applied over L_F)

$A = 4.3$ (Experimentally developed constant using CE Model Test)

$L_F = 16.0$ in.

$L = 48.0$ in. (Region 1)

$M_o = 0.7476$ lb/ft

$f_n = 30.0$ hertz (Region 1)

$d = 0.0625$ ft.

$\delta_o = 2\pi\zeta = 2\pi\ (.020) = 0.1257$ (Figure 2-6 $\zeta = .020$)

$\rho_f = .02836\ \text{lb/in}^3 = 49\ \text{lb/ft}^3$

The calculated critical flow velocity (V_c) for a tube in compression is 14.0 ft/sec. The free stream gap velocity (V_g) is based on the mass flow rate in the downcomer and is 11.22 ft/sec. The gap velocity (V_g) is less than the critical velocity and therefore no fluid elastic vibration is expected to occur due to cross flow near the tubesheet.

The critical velocity determined above was conservatively based on tubes being locked into support plates prior to heating up of the steam generator. The chemical reaction necessary in forming magnetite in the annulus between the tube and support plate normally occurs during full power operation. If the tubes are locked during full power operation and the steam generator is then cooled down, this will result in a tensile load in the tubes. As can be seen in Figure 2-4, an increase in tensile load increases the frequency of the tube. Increasing the frequency increases the difference between the critical velocity (V_c) and the gap velocity (V_g). The assumption that the load in the tubes is compressive is a very conservative method of evaluating the tubes for fluid elastic coupling.

LOW LEVEL VIBRATIONS

In addition to fluid elastic coupling it is also necessary to demonstrate that low level vibrations are within acceptable limits. An ANSYS model was developed to depict a tube loaded with an axial compressive load and a transverse load applied simultaneously. A compressive load of 2300 lbs., developed in Section 3 was applied to the beam model. In addition to the axial load, a uniform load resulting from the fluid flow across the tube was applied to the beam model. The uniform flow load is

$$F_f = C_L d \ \frac{\rho_f V_g^{\,2}}{2g}$$

where, $C_L = 1.5$ (Figure 2-7, for $\frac{U_{gap}}{V_{crit}} = .80$)

$d = 0.75$ inches

$$\frac{\rho_f V_g^2}{2g} = \frac{(49)(11.22)^2}{2\,(32.2)} = 95.8 \text{ lb/ft}^2 = 0.665 \text{ lb/in}^2$$

The uniform flow over the tube was determined to be 0.748 lbs/in. This load was applied to the tube near the shroud at the exit of the downcomer. The fluid flow is applied over sixteen inches of the tube measured from the tubesheet to the bottom of the shroud. The maximum bending stress for a healthy tube due to tube vibration in the bundle entrance region is 2.4 ksi. For a 64% degraded tube the bending stress is 12.1 ksi.

FATIGUE ANALYSIS OF TUBES

A fatigue analysis was conducted on a tube which conservatively experienced 1/32 inch degradation (31 MILS, 64% degraded tube) on the thickness of the tube. The cross sectional properties for the tube are

$$t = (.375-.03125)-(.375-.048)$$
$$t = .01675 \text{ inches}$$
$$A_1 = \pi\,(R_O{}^2+R_I{}^2) = \pi\,(.34375 - .3272^2) = .03530 \text{ in}^2$$

The fatigue analysis was conducted for 500 cycles of heatup at isothermal conditions and 14500 cycles from hot standby to 100% power. The stresses due to thermal expansion and low frequency vibration are

$\sigma_x = 32.3$ ksi (hot standby)

$\sigma_x = 65.2$ ksi (loading, 100% power)

$$\sigma_x \quad \frac{(P_O-P_I)R_I}{2t} = \frac{(2.15-.77)(.327)}{2(.01675)} = 13.5 \text{ ksi}$$

$\sigma_r = -.77$ ksi

$\sigma_x = \pm\ 12.1$ ksi (Vibration)

The skin stress due to radial gradient in the tube is

$$\sigma_s = \frac{E\alpha\Delta T}{1-\upsilon}$$

where, $\Delta T = 559\text{-}514 = 45^{o}F$

$E = 28.36 \times 10^{6}$ psi

$\alpha = 7.895 \times 10^{-6}$ in/in^{o}F

$\nu = 0.3$

The skin stress due to radial gradient through the tube thickness is 14.9 ksi. The stress intensity (SI) at hot standby defined above is

$$SI = \sigma_x - \sigma_r$$

$$SI = |(82.2 + 13.5) - (-.77)| = 46.47 \text{ ksi}$$

The stress intensity for 100% power is

$$SI = \sigma_x - \sigma_r$$

$$SI = |(65.2 + 13.5 + 14.9) - (-.77)| = 94.37 \text{ ksi}$$

The alternating stress (S_a) and resulting usage factor (U) are calculated (Reference 1) in Table 2-1.

TABLE 2-1

FATIGUE ANALYSIS

Transient	N Cycles	S_a (ksi)	U
Ambient to 100% Power	500	$S_a = \frac{1}{2}(46.47) + 12.1 =$ 35.34 ksi	$\frac{500}{103,000} = 0.0049$
Hot Standby to 100% Power	14500	$S_a = \frac{1}{2}(94.37 - 32.2) + 12.1$ 43.19 ksi	$\frac{14,500}{45,000} = 0.3222$

The maximum usage factor is 0.3271 for a 64% degraded tube, which is less than code allowable of 1.0.

SUPPORT PLATE ANALYSIS

The maximum stresses in the support plates due to locked tubes was evaluated using the finite element model shown in Figure 2-2. The plate segment shown was modeled using flat quadrilaterial shell elements. The tubes were modeled as three dimensional beams. The plate was subdivided into a solid rim at the outer periphery of the plate and as an equivalent solid plate (Reference 1, Article A-8000) over the remainder of the plate. The model is representative of a portion of the support plate and tubes from the tubesheet to support plate number 2 (Figure 2-3). The shroud and support lug were modeled as fixed points.

The maximum stress in the solid rim of the support plate is 64.6 ksi. The bending stresses in equivalent solid plate were found to be

σ_y = -11.7 ksi

σ_x = 4.5 ksi

The equivalent solid plate stresses were converted to an approximate perforated plate stress. An approximate uniform perforated plate was modeled using the ration E*/E = .144 for carbon steel. From Figure A-8131-1 of Reference 1 and Figure 2-1, the corresponding effective ligament efficiency is h/p = .22.

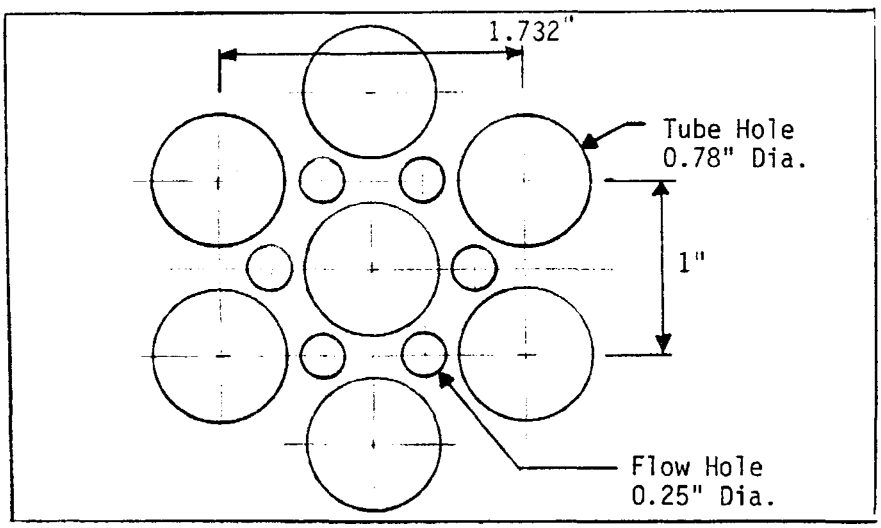


FIGURE 2-1. SUPPORT PLATE PENETRATIONS

From Figure A-8142.1 of Reference 1

$$\beta = \frac{\sigma_x}{\sigma_y} = 0.385$$

k = 1.04

The ligament stress is

$$SI = k \left(\frac{P}{h}\right) \sigma_y$$

$$SI = 55.3 \text{ ksi}$$

Maximum combined stresses in a support plate will result from out-of-plane loading (locked tubes) and in-plane loading (denting). The elastic stress of 55.3 ksi in the ligament represents only the out-of-plane load. To determine the maximum combined stress in the plate it is essential to combine the in-plane and out-of-plane loads. This would require an additional evaluation to be performed on a plate segment to determine the total stress and is beyond the scope of this project.

SUPPORT PLATE SHEAR

The thermal expansion of locked tubes results in axial loads in the tube which could shear through the support plate. It was assumed in this analysis that the magnetite formed in the annulus between the tube and support plate has the shear strength of carbon steel (SA-36). This assumption is probably conservative since magnetite may not have any strength in shear. However, there has been no test conducted, to our knowledge, to substantiate this statement. The maximum axial load in the tube, as defined in this section, is 2300 pounds (2.3 kips). For this analysis it will be conservatively assumed that the tube has been degraded 1/32 inch (31 MILS, 64% degraded tube). The maximum shear in the plate is

$$\tau = \frac{F_t}{A_p}$$

where, $A_p = \pi \, |.75 - 2(.03125)| \, .75 = 1.620 \text{ in}^2$

$F_t = 2.3$ kips

The shear stress in the support plate due to an out-of-plane axial load is 1.42 ksi. The shear is less than the allowables defined in Paragraph NB-3227.2 of the ASME Code, Reference 1.

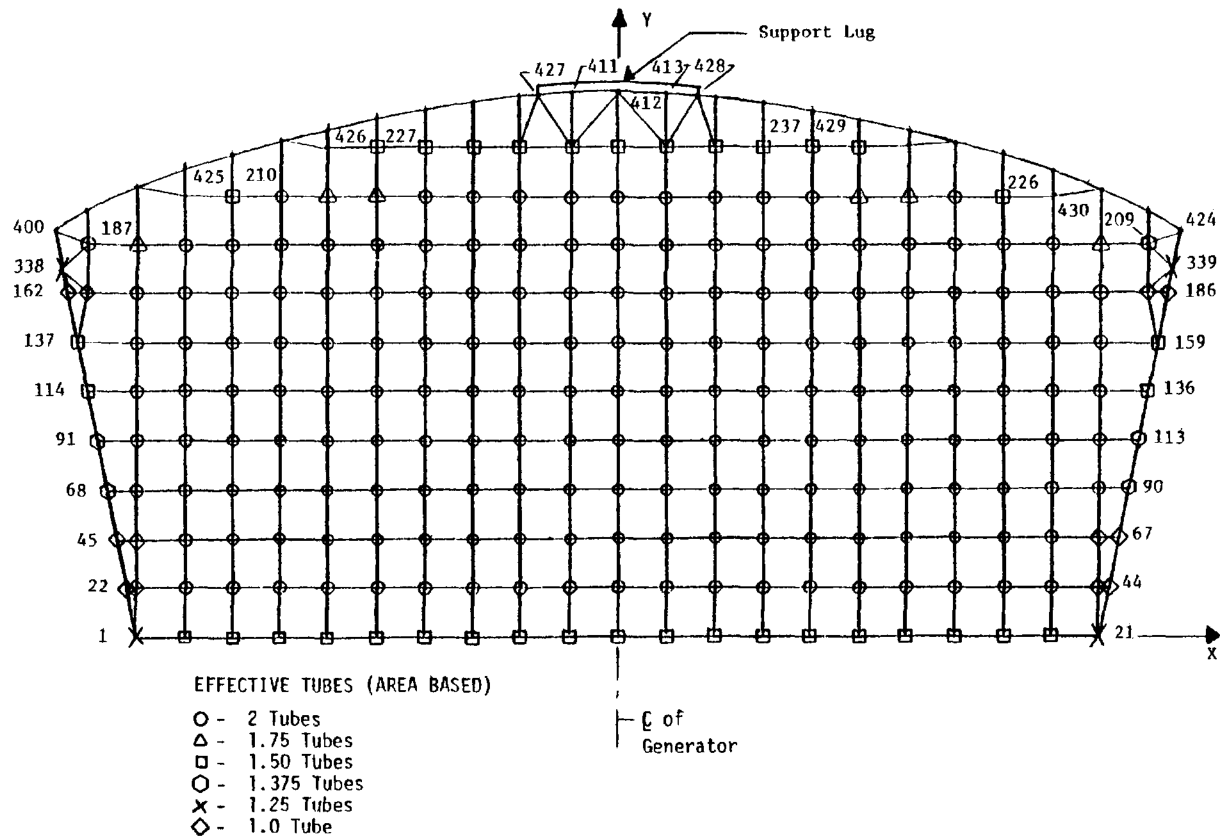


Figure 2-2. Finite Element Model of a Plate Segment

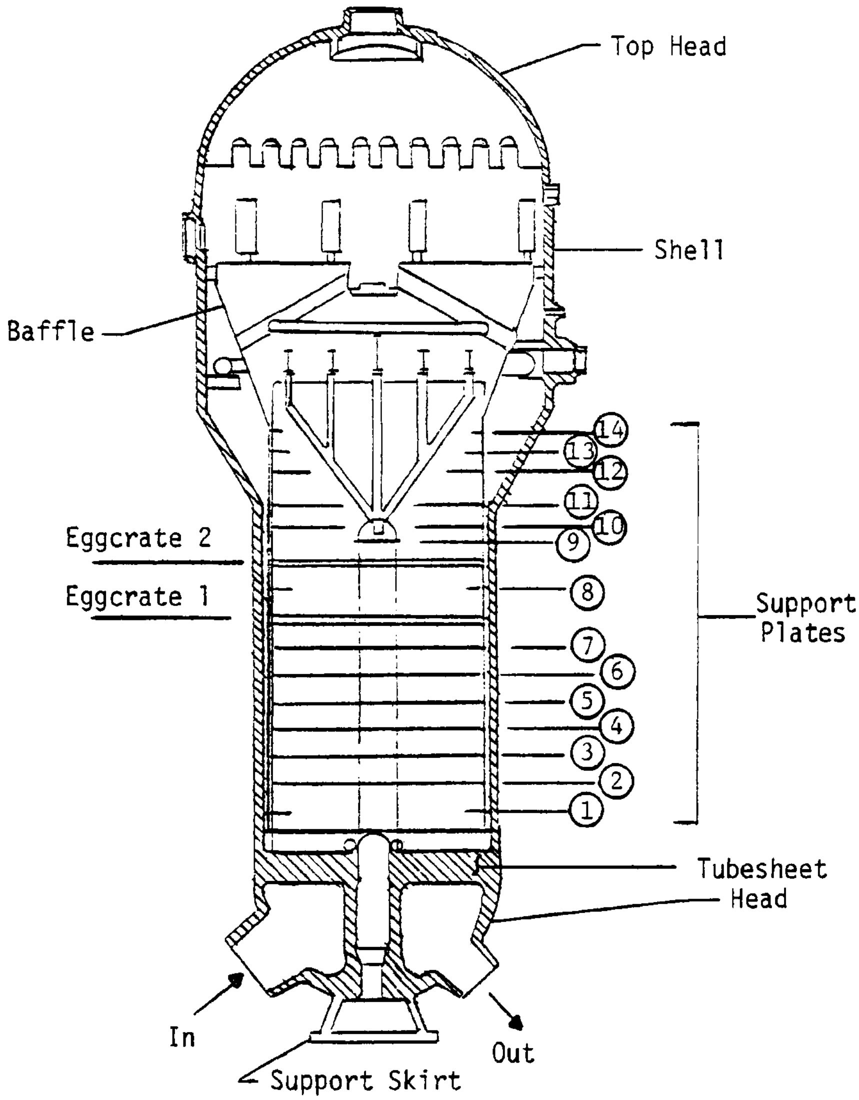


Figure 2-3. Tube Support Plate General Locations

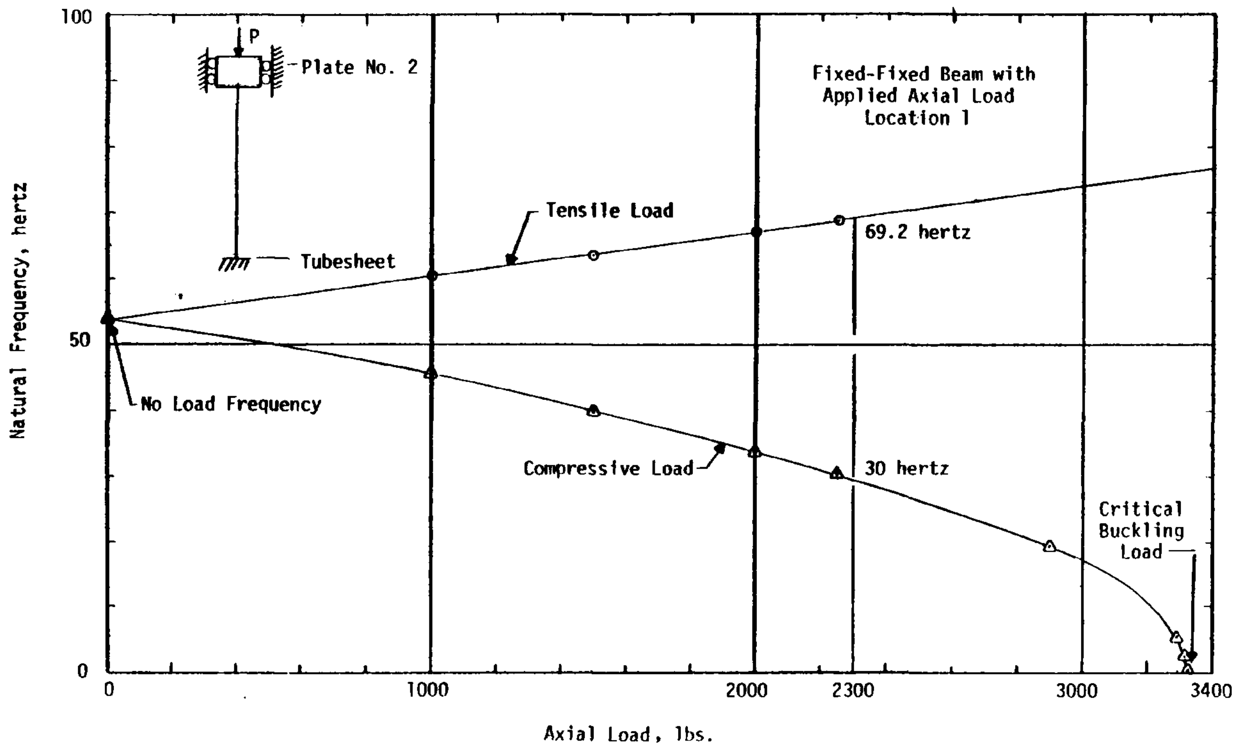


Figure 2-4. Tube Natural Frequency - Region 1

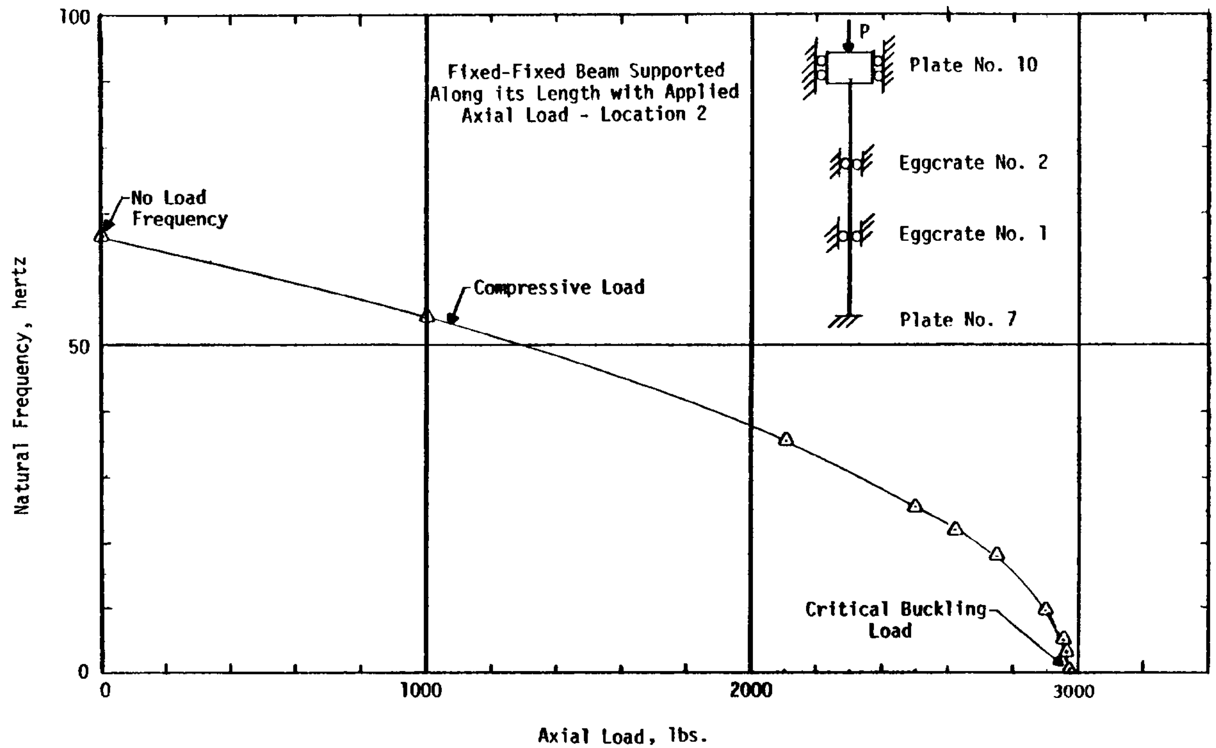


Figure 2-5. Tube Natural Frequency - Region 2

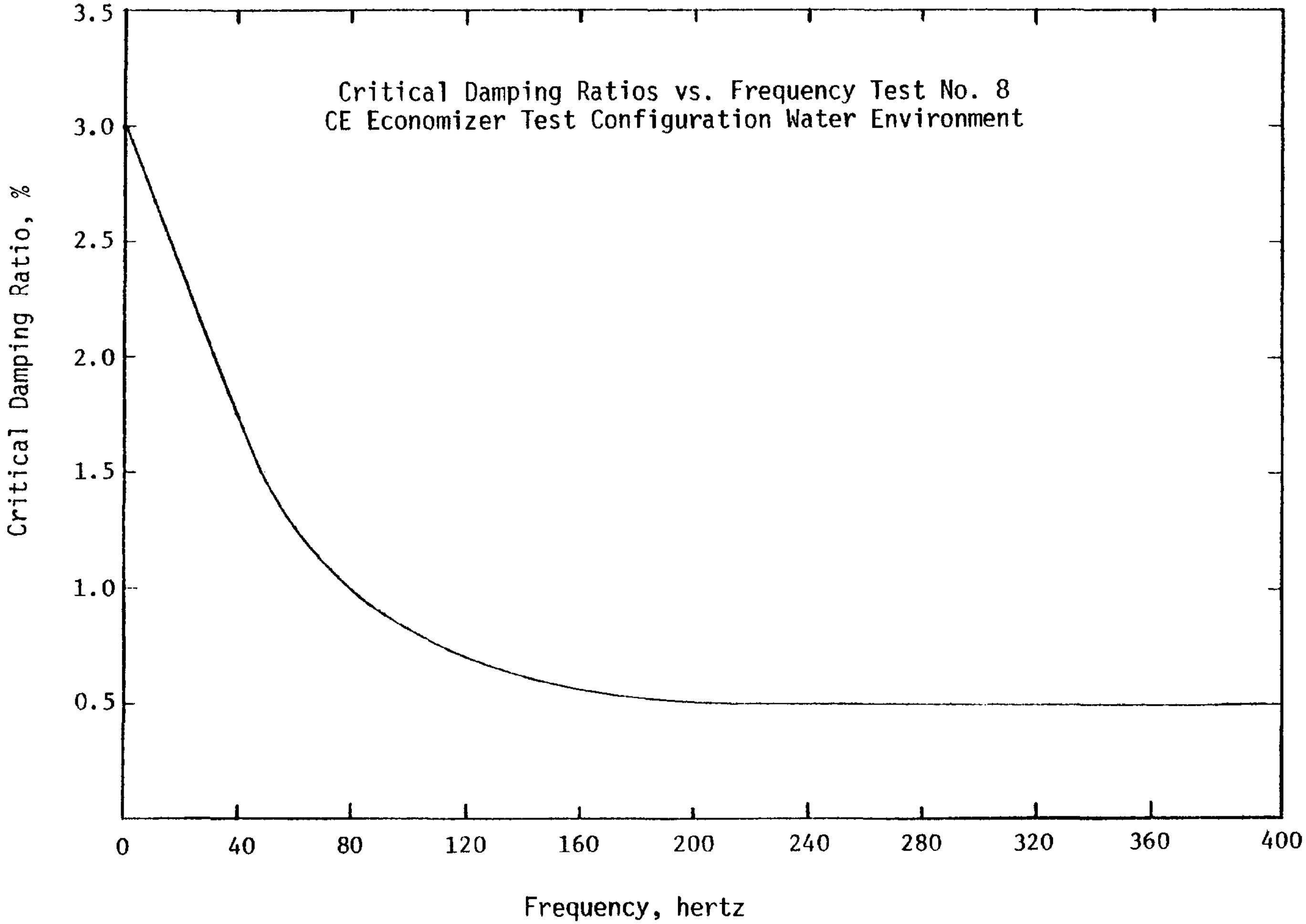


Figure 2-6. Critical Damping Ratio

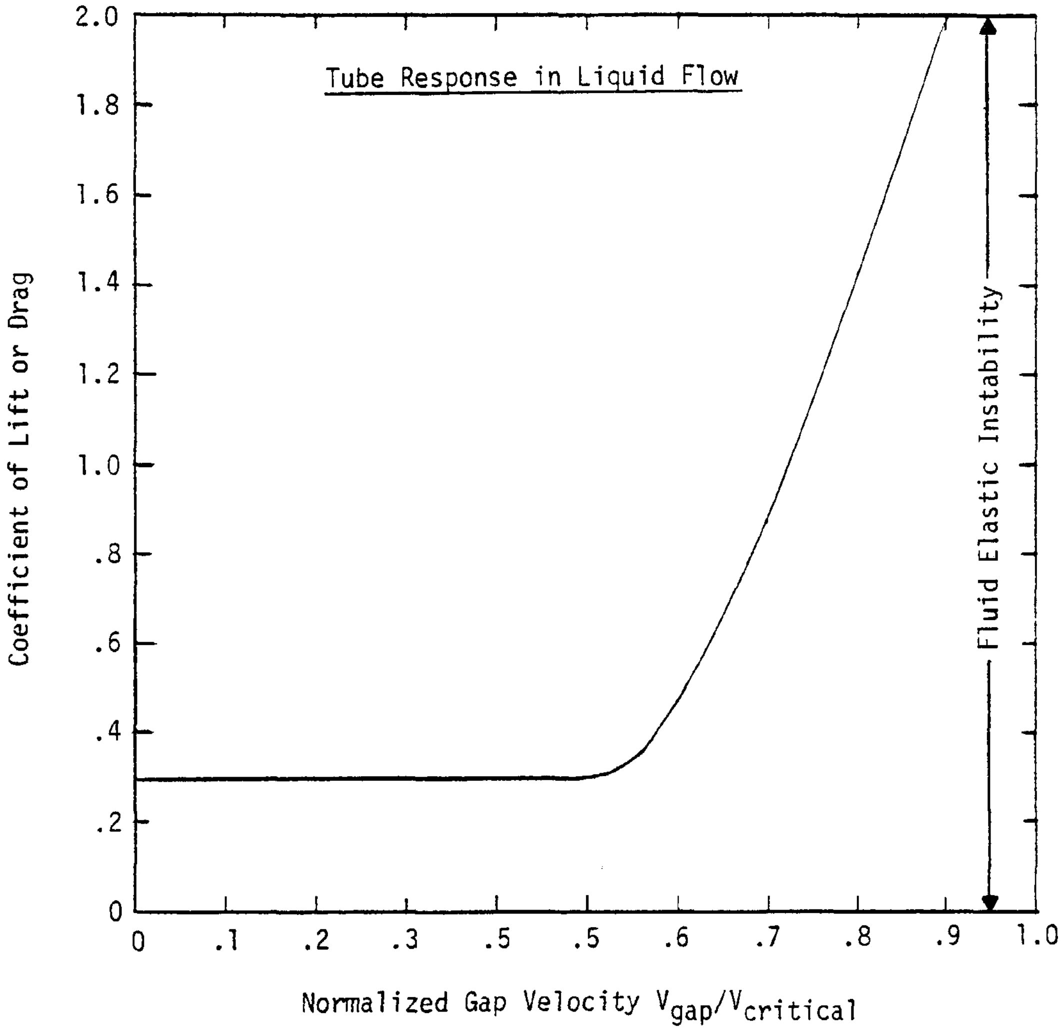
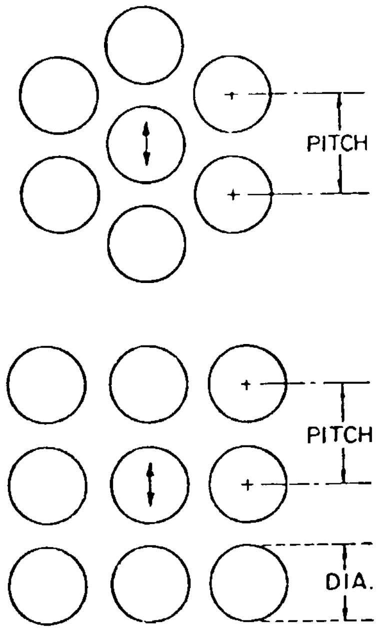
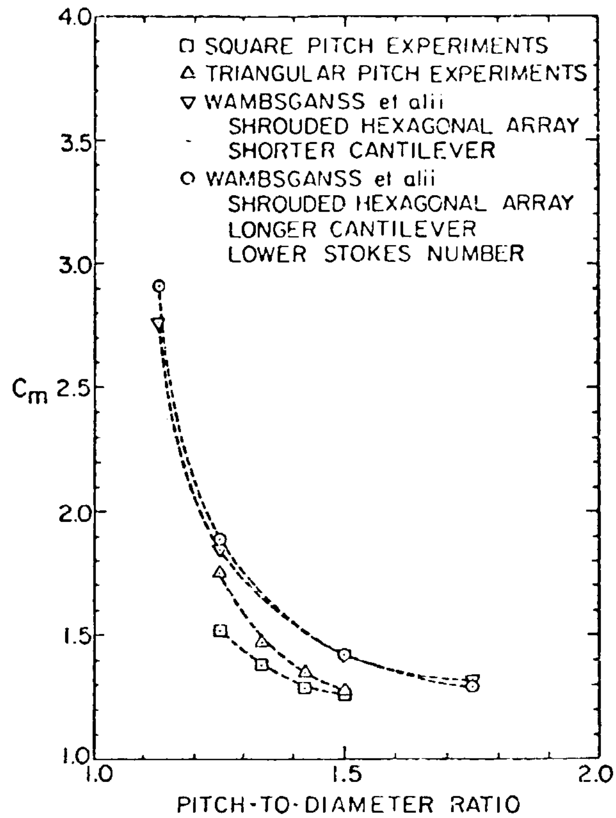


Figure 2-7. Coefficient of Lift or Drag



Experimental geometries



Experimental C_m values in tube bundles

Figure 2-8. Experimental Data from Reference 3

Section 3

TUBESHEET STRUCTURAL ANALYSIS

LOCKED TUBE AXIAL LOADS

Shear Stress

The axial load developed in the tubes due to locking of tubes into the support plate were applied to the tubesheet. The maximum tube axial load occurred at a location near the shroud to shell support. This load was developed in Section 2 and is 2300 pounds. Due to the flexibility of the support plate this load decreases toward the center of the support plate. In this analysis the load which is triangular in shape will be applied as a concentrated load applied uniformally around the circumference of the tubesheet. The concentrated load is determined by assuming that the tube load of 2300 pounds/inch is a triangular load, thus

$$W = 1/2\ WL_1$$

where, W = 2300 lbs/in

L_1 = 5 inches (load applied over a portion of tubesheet)

The load W is determined to be 5750 pounds. This load is conservatively considered as a 5750 lbs/in load around the circumference of the tubesheet. The load is carried by the ligament area contained in the tubesheet circumference at 70.33 inch radius (Figure 3-2, Nodes 41 thru 59). The maximum load is,

$$F_{max} = 2\ \pi RW$$

where, W = 5750 lb/in = 5.75 Kips/in

R = 70.33 in.

The maximum load applied to the tubesheet is 2541 Kips. The total ligament area around the circumference is conservatively contained in 155 liagments for each quarter of the tubesheet. The total ligament area for a minimum ligament width of 0.216 inches and a tubesheet thickness of 20.25 inches is

$$A_2 = 4 N_1 h t_{ts}$$

where, $N_1 = 155$

$h = 0.216$ inches

$t_{ts} = 20.25$ inches

The total shear area is 2711.9 in^2. The shear due to locked tubes is

$$\tau = \frac{F_{max}}{A_2}$$

The shear in the tubesheet is 0.92 ksi. For design conditions the maximum shear in the tubesheet occurs in the shell and is 17.88 ksi (Reference 2, Page C-189). The maximum shear resulting from design conditions and locked tubes is

$$\tau_{max} = 17.88 + 0.92 = 18.8 \text{ ksi} \quad < S_m = 26.7 \text{ ksi}$$

The maximum shear is less than the allowables defined in Paragraph NB-3221 of Reference 1, ASME Code.

Membrane Plus Bending Stress

The primary plus bending stress in the tubesheet were determined using the ANSYS finite element model shown in Figure 3-2. The tubesheet was modeled as quadrilateral shell elements pinned at the inside radius (stay cylinder) and outside radius. In this simplified model a load representing the locked tubes was applied as a force per unit circumference near the outside radius of the tubesheet. This load, as developed in this section is 5750 pounds/inch at a radius of 70.33 inch. The equivalent solid plate stresses were determined to be

$$\sigma_x = 0.214 \text{ ksi}$$

$$\sigma_y = 0.584 \text{ ksi}$$

From Reference 1, Figure A-8142.1

$$\beta = \frac{\sigma_x}{\sigma_y} = 0.366$$

$k = 1.05$

$P = 1.0$

$h = .216$

The stress intensity from Reference 1 is

$$SI = k \frac{P}{h} \sigma_y = 2.8 \text{ ksi}$$

The maximum membrane plus bending ($P_L + P_B$) stress intensity in the tubesheet for design conditions is 27.8 ksi. Adding this stress intensity to the stress resulting from locked tubes above

$$SI = 27.8 + 2.8 = 30.5 \text{ ksi} < 1.5\ S_m = 40.1 \text{ ksi}$$

The primary plus bending stress intensity is less than the Section III Code allowables.

MAXIMUM SHROUD SUPPORT LUG LOAD

The maximum load that could be imparted to the tubesheet by the tubes is the load required to shear the shroud support lugs. Under normal operating conditions, with locked tubes, this load would not occur and therefore is considered conservative. This analysis was conducted to show that the tubesheet structure meets the code requirements when exposed to extremely conservative out-of-plane loads.

Shear Stress

The maximum vertical load developed in the shroud support lugs necessary to cause the lugs to fail in shear was developed around the circumference of the tubesheet. In the steam generator analyzed there are sixteen (16) lugs located at the bottom of the shroud. The cross-sectional area of a lug is found from Figure 3-1

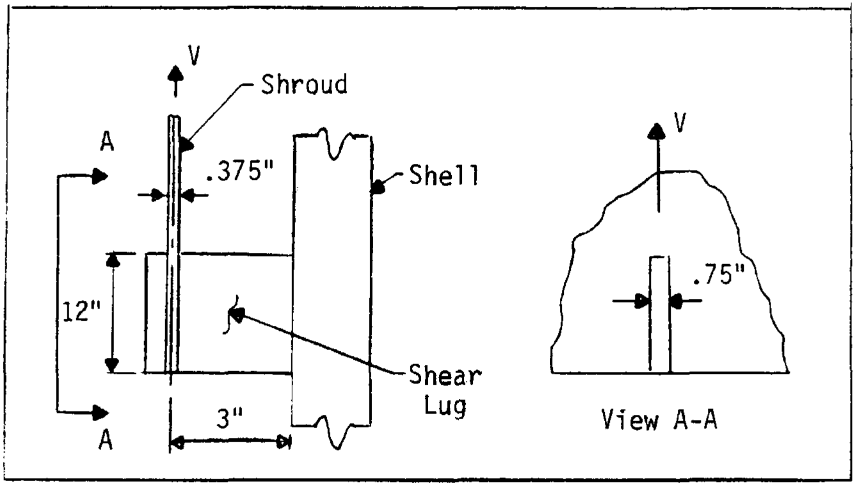


FIGURE 3-1. SHEAR LUG

The maximum load which a lug may transmit is determined by assuming that failure occurs at one-half of material ultimate. The maximum vertical load required to shear the sixteen shear lugs is then

$$V = N_2 \, S \, A_3$$

where, $N_2 = 16$

$S = 1/2 \, Su = 1/2(70) = 35$ ksi

$A_3 = (.75)(12) = 9.0 \text{ in}^2$

The vertical load is determined to be 5040 kips. The total ligament area calculated in Section 3 is 2711.9 inches square. The shear is

$$\tau = \frac{V}{A_2}$$

where, $V = 5040$ kips

$A_2 = 2711.9 \text{ in}^2$

The shear stress in the tubesheet due to a maximum shear load is 1.86 ksi. The shear due to design conditions, as defined above, is 17.88 ksi. The maximum shear due to design conditions plus the maximum support lug load is

$$\tau_{max} = 17.88 + 1.83 = 19.71 \text{ ksi} < S_m = 26.7 \text{ ksi}$$

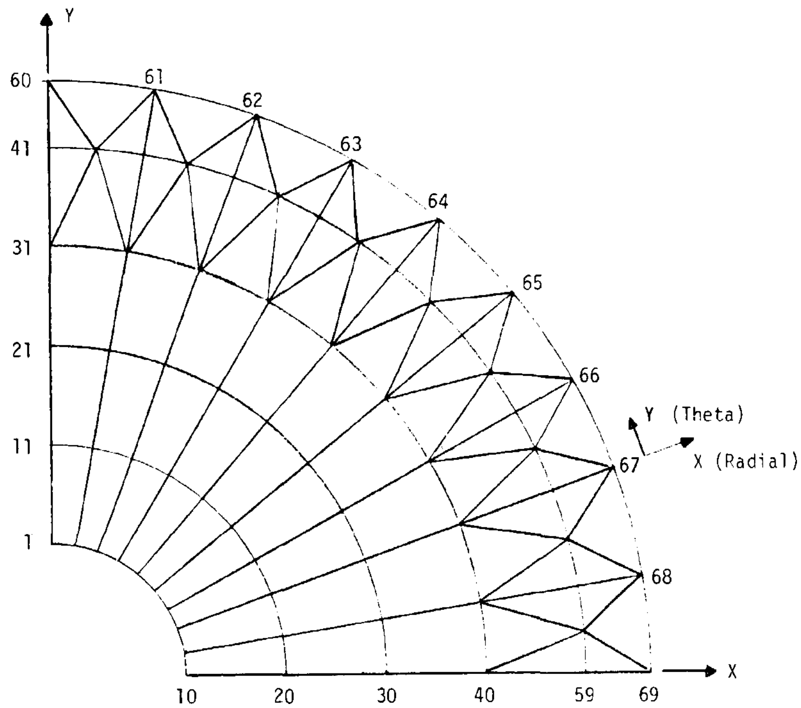
The maximum shear is less than the allowables defined in Paragraph NB-3221.1 of the ASME Code, Reference 1.

Membrane Plus Bending Stress

The lug load developed above is 5040 kips/in at a radius of 70.33 inches. When this load is applied to the ANSYS model presented in Figure 3-2, a maximum stress intensity of 6.12 ksi was obtained. Adding this to the stress due to design conditions results in a maximum stress of

$$SI = 27.8 + 6.12 = 33.92 \text{ ksi} < 1.5\ S_m = 40.1 \text{ ksi}$$

The primary plus bending stress intensity is less than the Section III ASME Code allowables.



BOUNDARY CONDITIONS

1. Nodes 1 through 10 - UX, UY, UZ, ROTX, ROTZ = 0
2. Nodes 60 through 69 - UX, UY, UZ, ROTX, ROTZ = 0
3. Nodes 11, 20, 21, 30, 31, 40, 41, 59 - UX, UY, ROTX, ROTZ = 0

FIGURE 3-2. FINITE ELEMENT MODEL OF TUBESHEET

Section 4

CONCLUSIONS AND RECOMMENDATIONS

An evaluation of the tubes, support plates and tubesheet was performed assuming the tubes are locked into drilled support plates. The following results were obtained:

1. The maximum compressive load of 2300 pounds for a single tube compares favorably with the critical buckling load of 3330 pounds.

2. The worst case compressive load of 2300 pounds results in a tube natural frequency of 30 hertz. The calculated critical flow velocity for fluid elastic coupling is 14.0 ft/sec. In the bundle entrance region, the cross-flow velocity of 11.22 ft/sec in less than this critical velocity.

3. The fatigue usage factor for 64% degraded tube was determined to be 0.34 which is less than the ASME Code allowable of 1.0.

4. The bending stress in the support plate due to out-of-plane loading is 55.3 ksi. This stress is not predominate in the structural behavior of the plate and must be combined with the in-plane stress due to denting to determine the maximum stress in the plate.

5. The maximum load developed by locked tubes of 2300 pounds was applied to the tubesheet model as a triangular load around the circumference. The resulting shear stress due to locked tubes is 0.92 ksi. The maximum shear resulting from design conditions and locked tubes is 18.8 ksi which is less than the allowable of 26.7 ksi. The membrane plus bending stress due to locked tubes and design conditions is 31.0 ksi which is less than the ASME design allowable of 40.1 ksi.

6. The load of 5040 kips required to shear the shroud lugs was developed across the tubesheet. The maximum shear due to design conditions and maximum shear lug load is 19.71 ksi which is less than the allowable of 26.7 ksi. The membrane plus bending stress in the tubesheet due to these load conditions is 33.92 ksi which is less than the ASME design allowable of 40.1 ksi.

7. The elastic results for the support plate due to out-of-plane loads represent a portion of the total stress in the plate. This stress must be combined with stress due to an in-plane load to determine the maximum stress in the plate. It is recommended that an elastic/plastic analysis be conducted on a support plate ligament model to determine the maximum stress due to the combined in-plane and out-of-plane loading.

Summarized below are some of the conservatism used in this report:

1. For fabrication of a steam generator, drawing tolerance for tube misalignment are set at 1/8 inch. A misalignment of 1/4 inch would impair installation and damage the tube. A 1.0 inch misalignment or eccentricity is a conservative value to determine if tube buckling occurred when tubes are locked into support plates.

2. Chemical reaction necessary in forming magnetite in the annulus between the tube and support plate normally occurs during full power operation. If the tubes are locked into the support plate during full power operation, then cooling down the generator would result in tension in the tubes. As can be seen in Figure 2-4, an increase in tensile load increases frequency of the tube. The assumption that the load in the tube in compressive is a conservative method of evaluating fluid elastic coupling in the tube.

3. The tube degradation experienced in generators today are relatively small approximately in the order of 1 to 2 MILS. This is due mostly to change in water chemistry in the generator. A 31 MIL, 64% degradation of a tube is therefore considered a conservative representation of a degraded tube.

4. The thermal expansion of locked tubes results in axial load in the tube which could shear through the support plate. It was conservatively assumed that the tube was degraded 1/32 inch or 64% at the tube to support plate interface. This conservatism results in a smaller cross sectional area and a larger shear stress.

5. The maximum tube axial load due to locking of tubes in the support plate occurred at a location near the shroud to shell supports. In this analysis it was conservatively assumed that this load acted or loaded the tubesheet around the circumference of the tubesheet.

6. The shear area in the tubesheet ligaments between tube holes was conservatively assumed to be in 155 ligaments for each quarter of the tubesheet. This area is conservative since it does not include the large area contained between the hot and cold side of the tubesheet.

7. The maximum load that could be imparted to the tubesheet by the tubes is the load required to shear the shroud support lugs. This is a very conservative load and could not occur under normal operating conditions.

SECTION 5

REFERENCES

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