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ANALYSIS OF TUBESHEET STRESSES IN A SULFUR RECOVERY UNIT

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ABSTRACT

The effects of thermal gradients and pressure are analyzed for a Sulfur Recovery Unit (SRU) firetube-type waste heat recovery exchanger. Increased operating temperatures due to the advent of oxygen enriched sulfur recovery technology raised concerns regarding highly stressed areas of the tubesheet. Stayed tubesheet designs are typically used in these SRU applications. Finite Element (FE) was used due to the complexity of the geometry involved. A thermal FE model of the tubesheet, tubes, and tube ferrules was used to establish the temperature profile. The thermal model incorporated the overall gas and steam side heat transfer coefficients including temperature boundary conditions. The developed thermal profile was used as the basis for a FE stress analysis model.

The combination of pressure, differential tube expansion, and tubesheet temperature gradient results in high localized stresses in the tubesheet knuckle area. Tubes were also evaluated for compressive buckling. This paper describes FE techniques used to quantify these thermal gradients and stresses.

DEFINING THE PROBLEM

Combustion effluent temperatures entering the SRU CLAUS firetube type waste heat recovery exchanger has increased from the 2000 to 2200°F level of just a few years ago to nearly 2900°F today. The temperatures are escalating to the 2700°F range due to operator demands for higher capacity and more efficient ammonia destruction inside their upstream reaction furnaces. In addition, oxygen enriched CLAUS technology has pushed these calculated exchanger inlet temperatures even higher, to around 2900°F. These increases in temperature pose a higher risk of boiler failure resulting from sustained operations at these elevated levels. CLAUS exchanger tubesheets have had to deal with some "temperature excursions" from their first applications in these plants, but continued long term steady state exposure to these elevated temperatures poses certain design concerns that must be investigated to assure these exchangers operate safely throughout their useful design life. Potential problem areas include peak metal temperatures in excess of the 600° to 650°F range that can result in metal sulfiding and peak heat fluxes than can produce steam vapor blanketing that can result in runaway tube and tubesheet metal temperatures. In addition, more severe temperature gradients needed to be investigated for their impact on overall stresses in the exchanger components.

The increased thermal induced stresses raised concerns about localized fatigue in the tubesheet. The potential increase in tube wall temperature also raises concerns about tube buckling from expansion compressive end loads.

It was decided to investigate the temperature and stress problems in two steps utilizing finite element (FE) methods. The thermal study utilized some previous work done on the subject and additional work that utilized a small model of a portion of the tube to tubesheet area of the exchanger. First the temperature distribution would be determined, and then the stresses would be addressed. The temperature investigation would include several iterations to determine the most reasonable design conditions, which would then be addressed in the stress investigation. The stress model evaluated the overall exchanger and was more complex than the tube to tubesheet localized thermal model.

FIRETUBE EXCHANGER GEOMETRY AND THERMAL FE ANALYSIS

The area of critical importance in the design of a firetube exchanger is at the inlet thin stayed tubesheet. The tubesheet is relatively thin material on the order of 1.5" to 2" thick with tubes that when strength welded to the tubesheets act as staying members. Care must be taken in the design details of the exchanger, at this critical area, to avoid premature failure due to tube to tubesheet weld or tube wall overheating and subsequent metal sulfiding and/or stress related failures. A typical firetube exchanger and tubesheet are pictured in Figures 1 and 2.

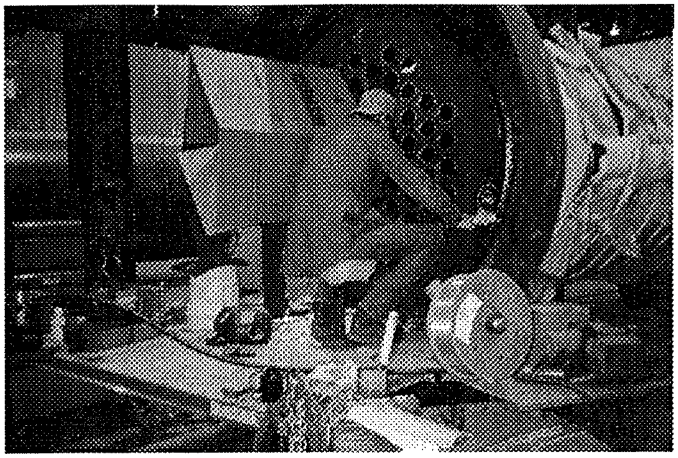


Figure 1

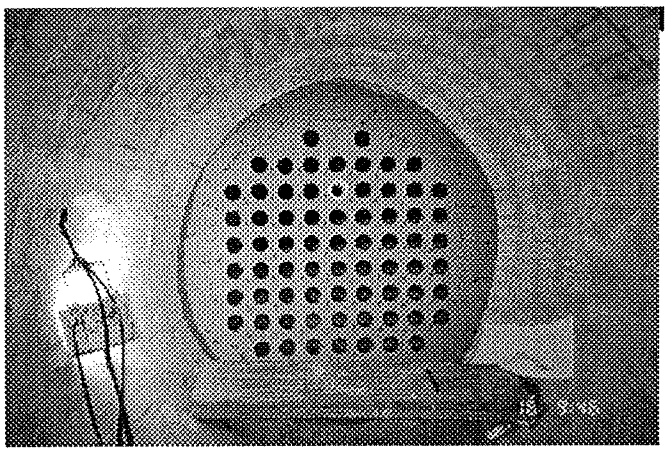


Figure 2

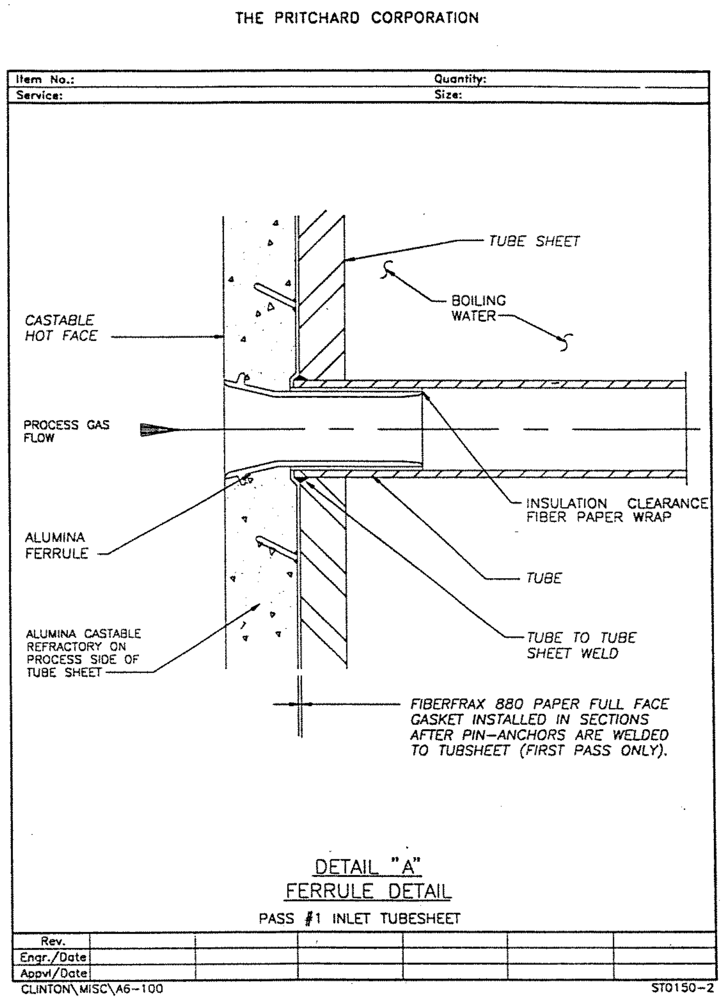


Figure 3

Typical design at the inlet tubesheet includes the use of ceramic tube ferrules and dense tubesheet castable refractory to shield this area from direct contact with the furnace process effluent. The potential for excessive temperatures and heat fluxes can be reduced by limiting the design mass flux entering the tubes of the boiler. Reduced mass fluxes reduce the convective transfer coefficients. This in turn reduces peak temperatures and heat fluxes.

The thermal FE model consisted of one complete tube to midway to an adjacent tube in order to determine the influence of nearby tubes. The tubesheet steel, tube steel, ceramic ferrule, ceramic fiber paper ferrule wrapping, ceramic fiber tubesheet paper facing and dense castable tubesheet refractory are all modeled into the thermal FE geometry. Refer to Figure 3.

Several thermal FE analysis investigations have verified that the majority of the steady state heat being transferred to the critical tube to tubesheet joint is coming from the inside of the ferrule and not from the face of the refractory covered tubesheet. Adding more refractory or insulating material to the face of the tubesheet proper yielded diminishing results.

Significant tubesheet temperature and flux reductions were achieved by modifying the ceramic ferrule inserts which form the entrance to the firetubes. These ferrules are typically wrapped with multiple layers of a very low thermal conductivity ceramic fiber paper. Increasing this paper thickness has dramatic results on calculated tube to tubesheet joint metal temperatures and associated fluxes. Refer to Table 1 for a listing of various paper wrap thicknesses studied. A key consideration in assigning paper wrap thickness is maximizing the ferrule "throat" I.D. This will minimize the pressure drop from the ferrule. It also minimizes the increase in heat transfer coefficient resulting from increasing gas velocities in this area. Other factors that define ferrule design included the velocity transition from the ferrule to the bare tube. This velocity reduction can be made gradually by using a shallow divergence angle on the ferrule outlet so as to reduce turbulence at the ferrule exit. Turbulence increases heat transfer in this area which is not desirable.

THERMAL FE MODEL BOUNDARY CONDITIONS

The most important aspect of the FE thermal model is to have the thermal boundary conditions accurately defined. The most important thermal FE boundary conditions are:

- Process gas bulk temperature
- The tubesheet refractory hot face temperature (assumed to be the process effluent bulk temperature for conservatism)
- Ferrule and bare tube inside combined convective and radiant heat transfer coefficients
- Boiling water heat transfer coefficient on the steam side of the tube to tubesheet joint

The ferrule and bare tube inside heat transfer convective coefficient is calculated by methods outlined by Sieder-Tate with a radiant component applied per the article "Including Radiative Heat Transfer and Reaction Quenching in Modeling a Claus Plant Waste Heat Boiler" by Karan, Mehrotra and Behie. Typical combined convective and radiant gas coefficients range from 20 to 40 Btu/hr/sq ft/°F. The combined coefficients are a function of gas temperature and mass flux rate.

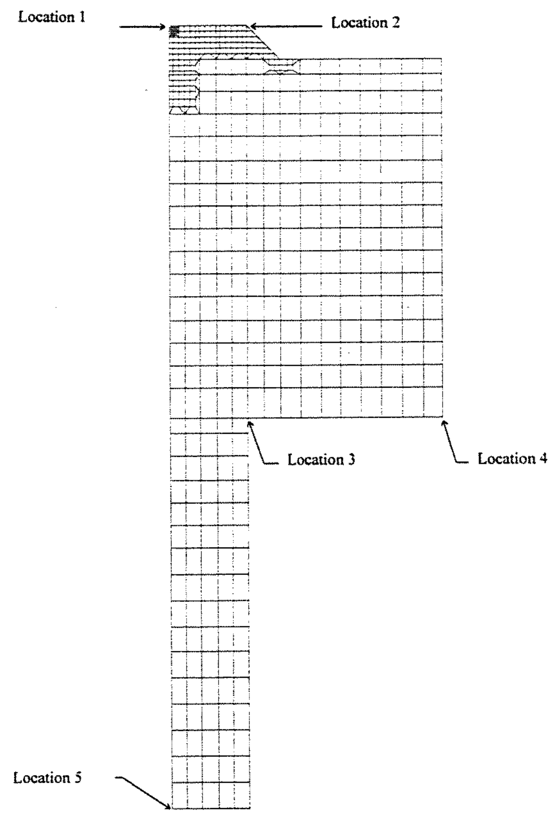
The inside heat transfer coefficient will be greater for the turbulence or flow eddys that occur at the outlet of the ferrule where the flow transitions into the bare tubes. An assumed multiplier of 1.38 times the bare tube transfer coefficient (calculated using the methods indicated above) can be used was used for the first 3 to 4 diameters of the bare tube length beyond the end of the ferrule to account for this turbulence.

The water side coefficient is obtained from Kern's "Process Heat Transfer" with a factor applied to consider shellside fouling in the boiler. A reasonable value for water convective coefficient in free boiling is 2000 to 4000 Btu/hr/sq ft/°F. The 2000 Btu/hr/sq ft/°F used in this particular model accounts for bubbles from surrounding tubes and tube fouling. The gas coefficient has a greater influence on the heat transfer in the thermal model than the water side coefficient.

THERMAL FE RESULTS

The main areas of concern for the thermal FE analysis are the calculated peak metal temperature and calculated peak heat flux into the boiling water. Peak metal temperatures should not exceed the 600 °F to 650°F range in order to avoid H_2S related metal sulfiding corrosion and other sulfur/metal phenomena that occur at temperatures in excess of this limit. Refer to "New Hydrogen Sulfide Corrosion Curves" Petroleum Refining by Sjoberg.

Peak temperatures as calculated for this particular unit are listed in Table 1 for the locations as defined by Figure 4 and an example temperature profile is depicted in Figure 6.

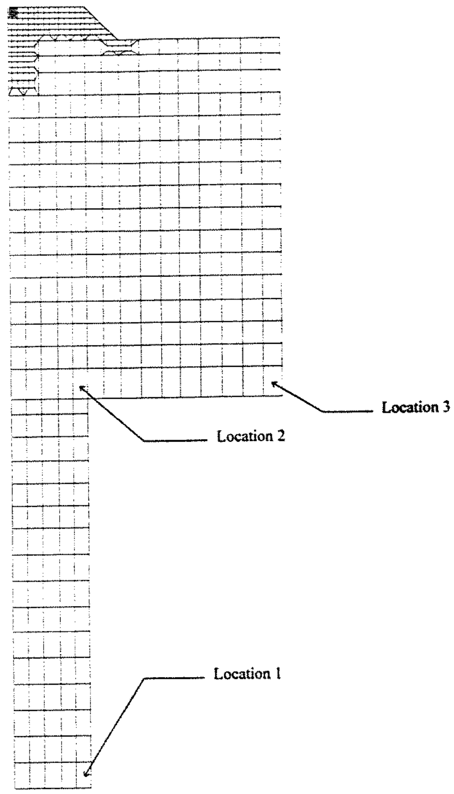


Tube Temperature Reporting Locations

Figure 4

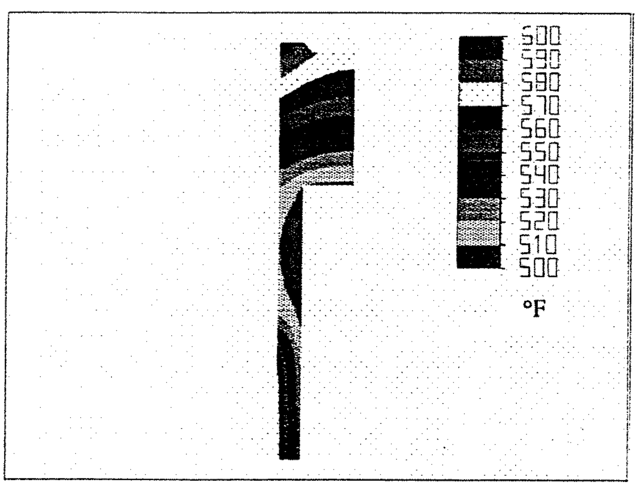
For thermosyphon type steam generators, peak fluxes should be held to less than 50,000 to 70,000 Btu/hr/sq ft range depending on pressure. This limit will avoid vapor blanketing which can result in runaway tube metal temperatures, excessive metal sulfidization, corrosion erosion and stress rupture failures. Refer to the articles "Critical Flux on a Tube in a Horizontal Tube Bundle" by Dykas and "Evaluate Waste Heat Steam Generators" by Knight as well as API 534 on "Heat Recovery Steam Generators".

Peak fluxes as calculated for this particular model are listed in Table 1 for locations as defined by Figure 5 and an example heat flux profile is depicted in Figure 7.



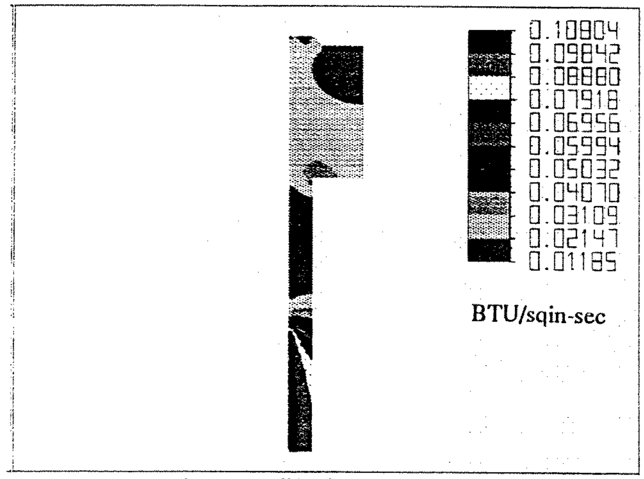
Tube Flux Reporting Locations

Figure 5



Temperature profile in tube -- Z2

Figure 6



Flux profile in tube -- Z2

Figure 7

To support the stress FE analysis, other temperature gradients and values had to be obtained. This work included the two dimensional conventional heat transfer calculation at the edge of the tube sheet and tubesheet knuckle where no tubes were present. In this area, the temperature gradient varies from the tube field temperature gradient to the water saturation temperature some distance back in the shell. The FE stress model temperature profile was based on the fin effect along the tubesheet knuckle radius with the gradient uniformly declining through a distance of five times the tubesheet thickness. The FE stress model required a prediction of the tube mean metal temperature for differential expansion compressive stress analysis. This mean metal temperature was easily obtained from computer thermal design software output for the exchanger.

FIRE TUBE EXCHANGER FINITE ELEMENT STRESS ANALYSIS

Concerns about overstress, long term thermal induced fatigue problems, and tube buckling required the stress analysis to be performed using COSMOS/M FE software. As shown in Figure 8, the finite element model was constructed using 3-D solid elements for the tube sheet, the knuckle radius, and the shell. Tubes are modeled using thick shell element due to diameter to thickness ratio of the tube. Taking advantage of symmetrical conditions of the exchanger, the FE model was constructed to represent 1/4 of the exchanger with axial length such that the critical stress area would be free from influence by the boundary restraints of the FE model. The tube length and the shell length were extended to half of the exchanger length using beam elements to reduce the total number of elements and nodes. The model dimensions are shown in Table 3. Over 5500 elements and 7000 nodes were used to describe the model which took more than one hour of computer time to run on a Pentium PC.

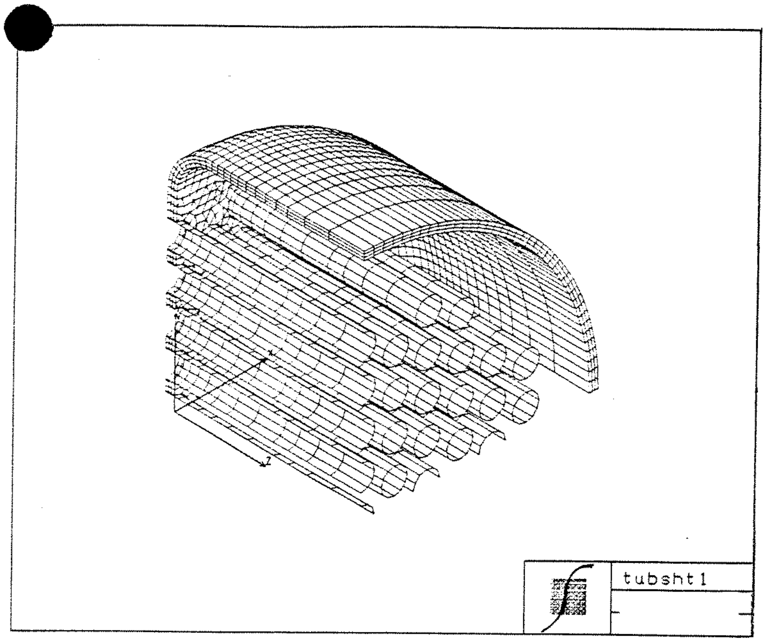


Figure 8

STRESS FE MODEL CONDITIONS

Internal pressure (750 + FV = 765 psig) was applied on the shell side. Based on the heat transfer analysis results, the temperature was set at 500°F on the inside wall of the shell, the tube sheet, the knuckle radius, and outside wall of the shell. The "hot face" (channel side) of the tube sheet was set at 600°F. The average temperature of tubes were set at 560°F. The 560°F is a mechanical design value based on 535°F operating plus a 25°F design margin. The temperature profiles for the tube array were incorporated. With these reference temperature boundary conditions, the model was analyzed with the FE program to complete the temperature profiles.

STRESS FE RESULTS

Linear static analysis with small displacement theory is the basis of this FEA based on the COSMOS/M User Guide and Introduction to the Finite Element Method by Desai and Abel. The maximum pressure stress and maximum thermal stresses are located in the knuckle radius region.

Although the maximum stress intensities are localized in the knuckle region, they occurred at different points for pressure, thermal, and combined pressure and thermal conditions. Therefore, the combined condition was chosen to find the critical stress area which was identified as node 3032. Stress Classification Line recommendations (SCL, Ref. 9) were used to find stress locations to compare against ASME Code Sec. 8, Div. 2 allowables. It should be noted that these stress categories are estimates due to lack of clearly defined linearization procedures and supporting technical literature.

In the pressure load case, as indicated in Figure 9, the maximum stress intensity of 18,240 psi is found at node 3032 of the knuckle region. A Stress Classification Line was drawn at this point through the thickness of the wall and the primary local membrane stress intensity (P_L)was estimated at 6,500 psi; primary local "membrane+bending" (P_L+P_b) stress intensity of 10,900 psi was estimated. Based on the ASME Code Sec. VIII, Div. 2, the allowable stress intensity is 1.5 times the design stress intensity for both P_L and P_L+P_b, which in this case, is 30,750 psi at the operating temperature of 500°F.

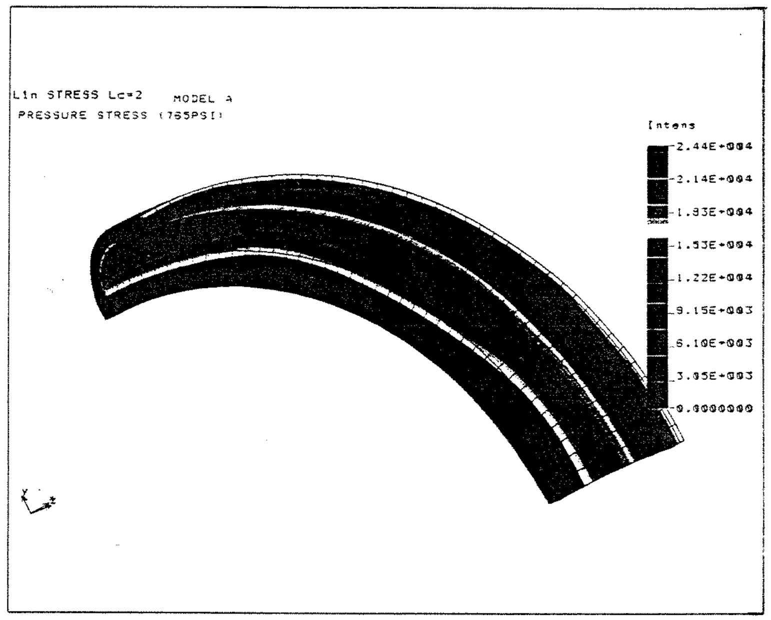


Figure 9

In the thermal load case, the maximum stress intensity of 142,000 psi is found at node 3032 in the knuckle region as shown in Figure 10. Using similar steps as in the pressure load case for constructing SCL, secondary "membrane+bending" (Q) stress intensity of 52,000 psi and "peak" (F) stress intensity of 90,000 psi were estimated.

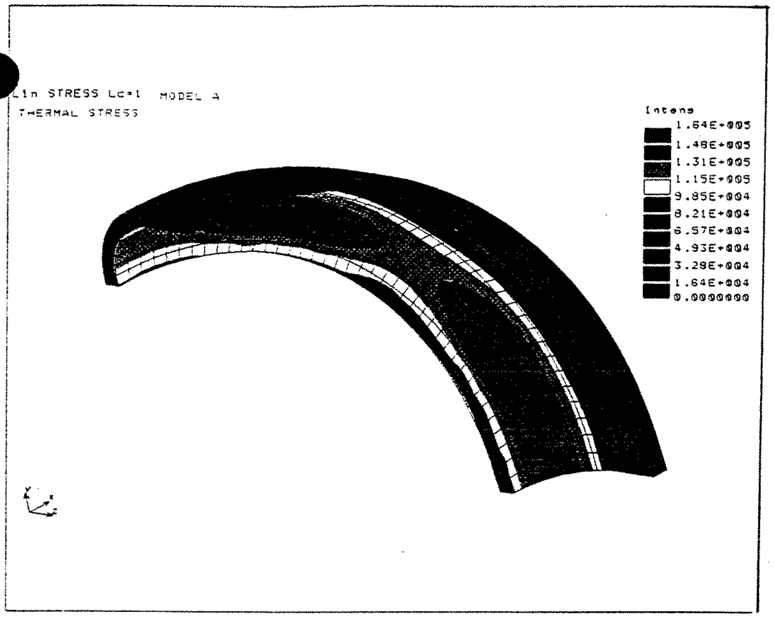


Figure 10

Again, using the same procedures as described above for the combined pressure and thermal case, the maximum stress intensity at node 3032 is 160,000 psi ($P_{L}+P_b+Q+F$), and the "$P_{L}+P_b+Q$" stress of 59,050 psi was estimated. The allowables for $P_{L}+P_b+Q$ is 3 times the design stress intensity or, 61,500 psi. The allowable stress "Sa" obtained from fatigue curves was 165,000 psi based on design service life of 150 cycles. Therefore the design is acceptable based on the cycle fatigue criteria (see Table 2).

EFFECTS OF TUBE LAYOUT ON STRESSES

It is apparent that differential thermal expansion between shell and tubes contributes to the high local stresses in the knuckle region, especially when tubes are located very close to tangent line of the adjacent knuckle radius. To study the effects, model "B" was constructed by modifying the original model "A". Two (2) tubes were removed from the model (Figure 8), as shown in Figure 13, and the same analysis was performed. The results, shown in Table 2, indicated that pressure stress increased by 43% while thermal stress decreased by 33%. Combined pressure and thermal stress decreased by 24%. The through thickness thermal and pressure stress profiles for Model A are shown in Figure 11 and Model B are shown in Figure 12. Note Table 2 stress values are adjusted for modulus of elasticity at 500°F. Additional Model B information shown in Figures 14 and 15.

TUBE BUCKLING

Buckling analysis was carried out for tube axial compression. The results are shown in Table 2, which indicated that, by locating tubes away from knuckle region, the tube compressive load was reduced by 72% as shown in model A and B (see Table 2). This reduced axial load also reduced the weld stresses at the critical tube to tube sheet junction. The buckling loads are well under allowables.

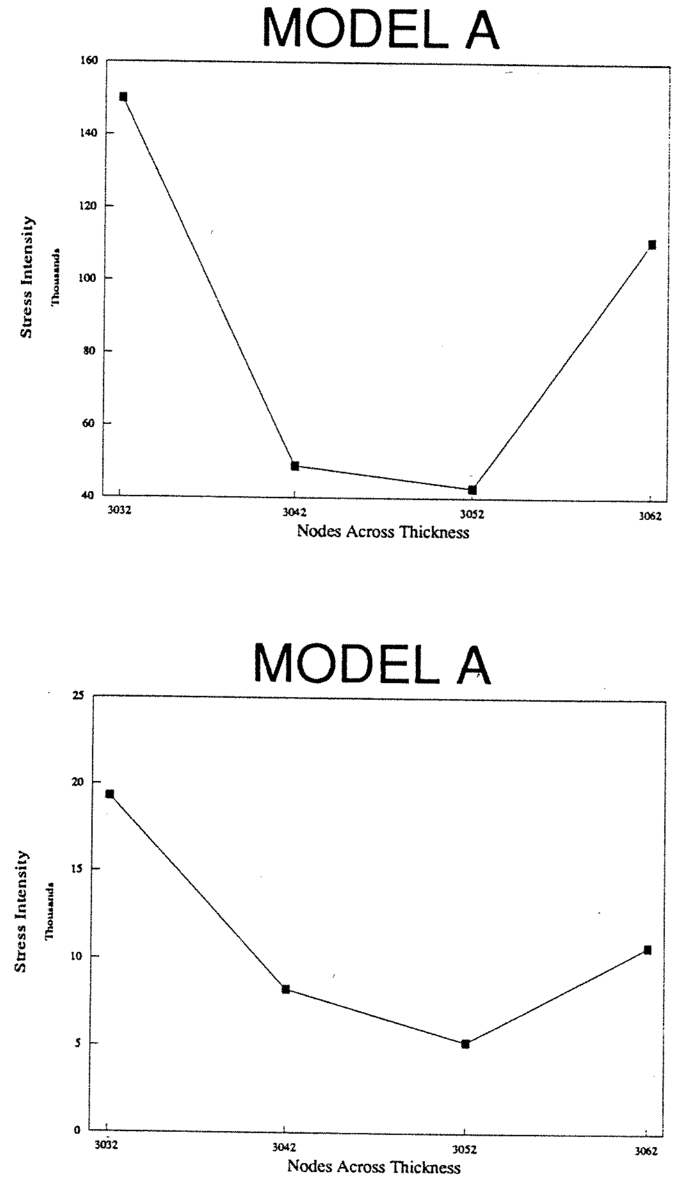
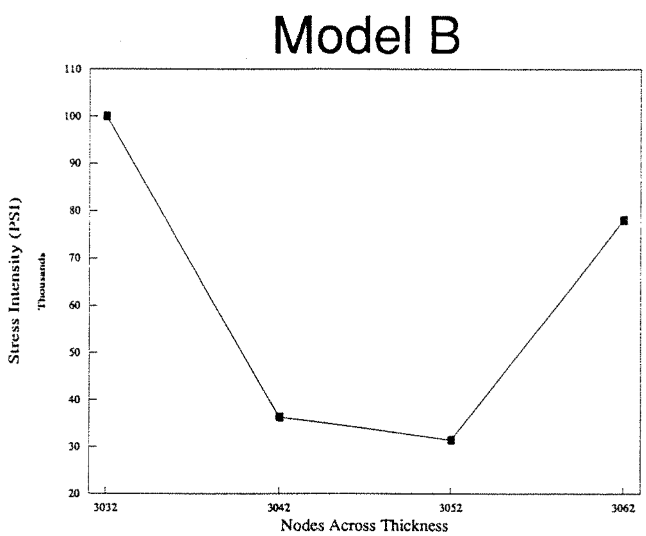


Figure 11



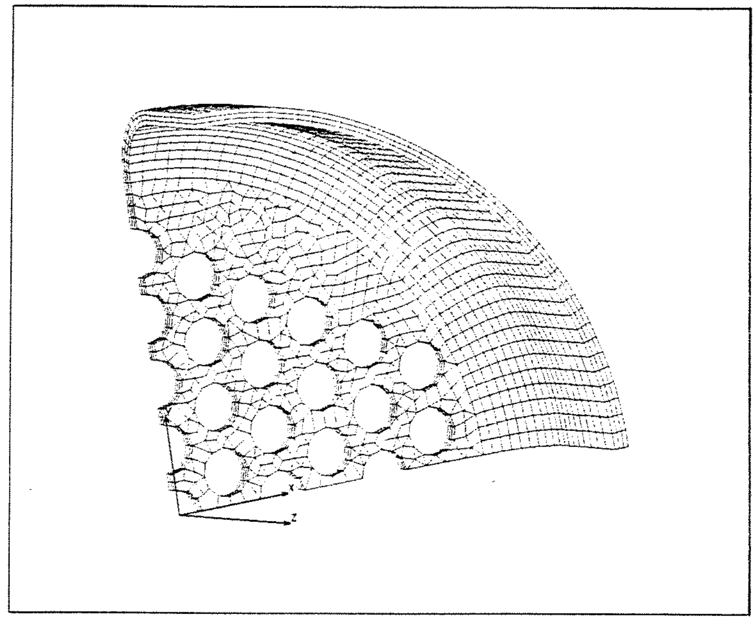


Figure 13 - Model B

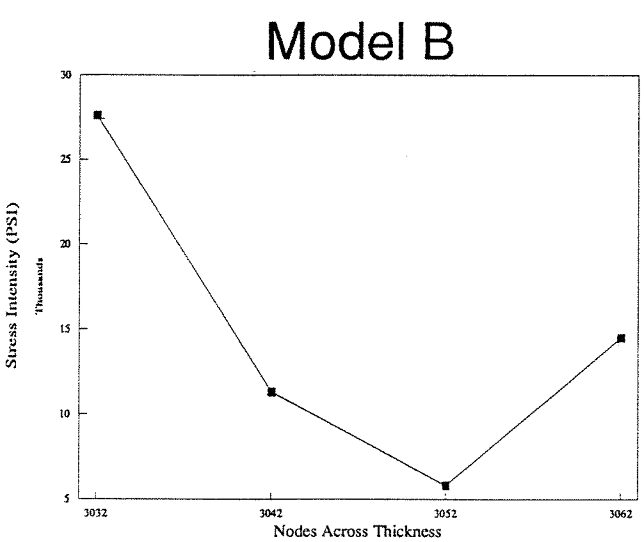


Figure 12

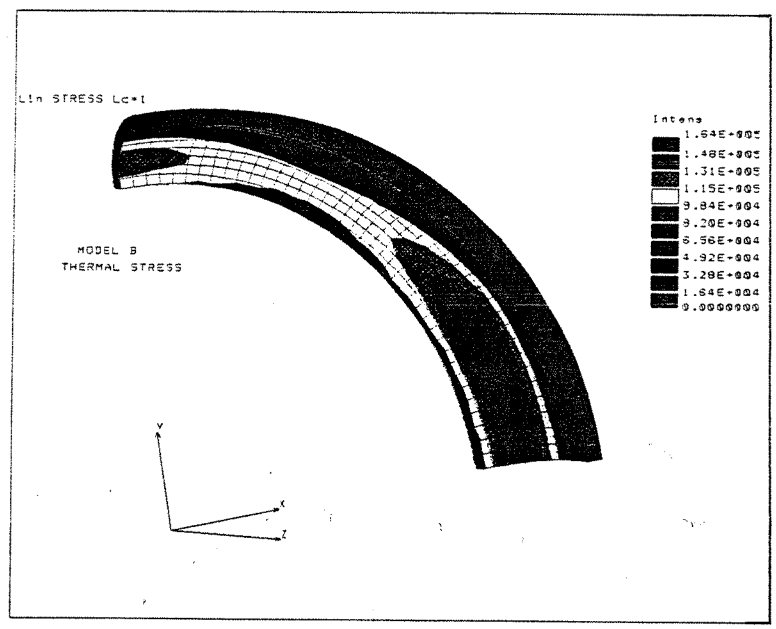


Figure 14 - Model B Thermal Stress

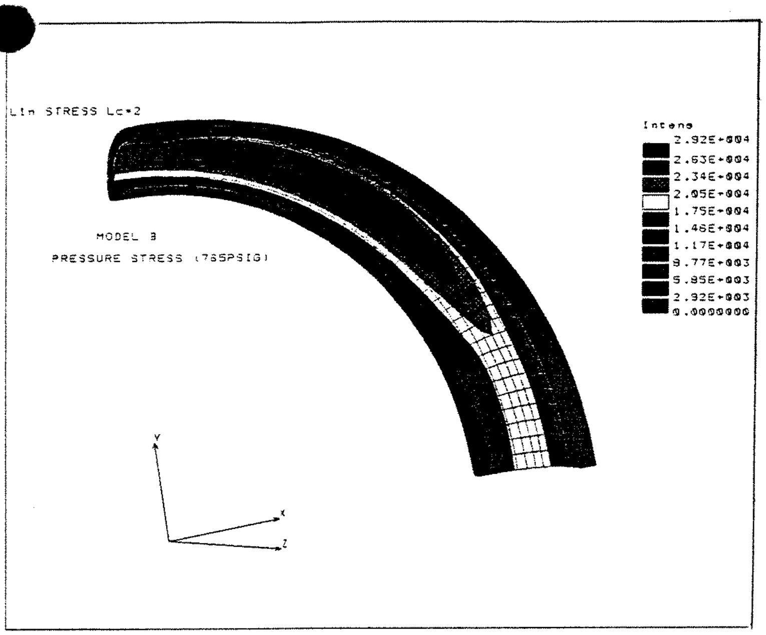


Figure 15 - Model B Pressure Stress

CONCLUSIONS

THERMAL FE

All temperature and flux parameters noted above were met for the boiler under investigation. In order to meet the temperature and flux parameters, it was necessary to reduce the mass flux rate from 4.5 lb/sec/sq ft to 2.5 lb/sec/sq ft and slightly re-design the tube ferrules to accommodate insulating paper from 1/8" nominal thickness wrapping to 1/4" nominal thickness wrapping.

STRESS FE

The stress analysis confirmed that model A and model B meet the ASME stress allowable requirements and was an acceptable design. While tubes become stays for the tube sheet for the pressure induced load, the tubes expand more than the shell. This differential expansion results in high stresses in the knuckle region which is displaced by the pushing of tubes and the pulling of shell. The effect will amplify when tubes are located near the knuckle region as discussed above. Comparing results between model A and model B, it was determined that if tube wall was located away from knuckle tangent line at a minimum distance of 2.5 times the tube sheet thickness localized stresses in the knuckle region and weld stresses at the tube and tube sheet junction would be reduced significantly.

RECOMMENDATIONS

Firetube type waste heat exchangers should be evaluated to determine the effects of temperature gradient profiles on stress levels in the equipment to ensure the structural integrity of the equipment.

The authors recommend that consideration be given to limiting the mass flux entering the firetube exchanger to control metal temperatures. For this application, the mass flux entering the tubes was limited to 2.5 lb/sec/sq ft. This minimized the heat flux, temperature, and temperature gradients observed at the hot inlet tubesheet and the first few feet of bare tube in the exchanger.

Tube ferrules for the first pass waste heat exchanger tube inlets must be carefully designed. For this application ferrule design includes 1/4" thick circumferential wrap of insulating paper to minimize the heat transferred into the critical tube to tubesheet area.

The exchanger tube layout should be considered with respect to stress levels and flexibility of the tubesheet. Tubes located too close to the knuckle of the tubesheet will increase stress levels and reduce available fatigue life. Placing the outermost tubes of the bundle no closer than two and a half (2.5) times the tubesheet thickness away from the tubesheet knuckle tangent line can be expected to reduce the stress intensity and provide near optimum fatigue life.

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Table 1 - Heat Transfer Models Summary

	Insulation Thickness		
Model	Ferrule to Tube	Face of Steel Tubesheet	Maximum Tube Temperature
Z1	1/8”	1/8”	630
Z2	1/4”	1/8”	599
Z3	1/8”	1/4”	621
Z4	3/16”	1/4”	602
Z5	3/16”	1/8”	611
Z6	3/16”	1/8”	606

								BTU/sqin-sec			BTU/sqft-hr						
Model	Tubesheet Thickness	Tube Diameter	Tube Pitch	HTC	Gas Temp	Water Temp	T1 Max Tube Temp	Heat Flux 1	Heat Flux 2	Heat Flux 3	Heat Flux 1	Heat Flux 2	Heat Flux 3	T2	T3	T4	T5
	(Inches)	(Inches)	(Inches)	B/hr/ sqft/°F	°F	°F	°F							°F	°F	°F	°F
Z1	1.750	4	6	23.3	2925	500	630	9.01E-02	5.46E-02	3.94E-02	46,707	28,310	20,431	616	516	510	580
Z2	1.750	4	6	23.3	2925	500	599	9.01E-02	3.95E-02	2.95E-02	46,692	20,464	15,300	590	511	508	580
Z3	1.750	4	6	23.3	2925	500	621	9.00E-02	5.16E-02	3.65E-02	46,661	26,758	18,941	606	515	510	580
Z4	1.750	4	6	23.3	2925	500	602	9.01E-02	4.25E-02	3.06E-02	46,700	22,019	15,868	592	512	508	580
Z5	1.750	4	6	23.3	2925	500	611	9.01E-02	4.63E-02	3.38E-02	46,700	24,001	17,509	601	513	509	580
Z6	1.750	4	6	23.3	2850	500	606	8.73E-02	4.41E-02	3.22E-02	45,255	22,887	16,699	596	513	508	577

Table 2

Tubesheet Knuckle Stress (Intensity)					
		Stress Category	Allowable (@ 500°F)	**Model A** **	**Model B** **
Pressure	(KSI)	P_L+P_b Maximum	30.75 ---	10.9 18.24	16.0 26.08
Thermal	(KSI)	Q F Maximum	--- --- ---	52.0 90.0 142	42.0 54.1 94.5
Pressure+ Thermal *	(KSI)	P_L+P_b+Q	61.5	59.05	47.25
		P_L+P_b+Q+F Cycles	186.0 100	160.0 160	120.9 310

* Including differential expansion of tubes and shell
** Adjusted for E @ 500°F

Tube Stress (Axial)				
		Allowable	**Model A** Dist = 1.25 x t*	**Model B** Dist = 3.8 x t*
Pressure	(KSI)	---	3.1	3.3
Pressure+ Thermal**	(KSI)	-18.9	-18.7	-5.3
Tube Load (Axial)				
Pressure+ Thermal**	(KIP)	-429.0***	-82.4	-23.36

* Distance of tube wall to knuckle tangent
** Including differential expansion of tubes and shell
*** Based on critical buckling load

*t = Thickness of tubesheet (Inches)

Table 3

Tubesheet / Shell

Material	SA-516-70N
Shell I.D.	72 inch
Corrosion Allowance	0.125 inch
Tubesheet Thickness	1.75 inch
Design Pressure (shell side)	750 psig / FV

Tubes

Material	SA-106-B
O.D.	4.5 inch
Thickness	0.337 inch
Length	28'-0"
Design Pressure (tube)	75 psig / FV