INVESTIGATION OF HEAT EXCHANGER STAYED KNUCKLE TUBESHEET STRESSES

Dennis H. Martens
Black & Veatch Pritchard, Inc.
Overland Park, Kansas

Charles S. Hsieh
Black & Veatch Pritchard, Inc.
Overland Park, Kansas

Robert M. Walker
Black & Veatch Pritchard, Inc.
Overland Park, Kansas

ABSTRACT

The combination of pressure, differential tube and shell expansion and tubesheet temperature gradient results in high localized stresses in the tubesheet knuckle area and tubes near the tube sheet. This paper presents stress investigations of several stayed tubesheets utilizing knuckle designs using finite element analysis. The necessary thermal boundary information required to support the stress investigation is addressed in this paper.

The paper references previous work presented by Dennis Martens, Charles Hsieh and Christopher K. Brzon titled "Analysis of Tube Sheet Stress in a Sulfur Recovery Unit", published in ASME PVP-Vol. 336, 1996.

NOMENCLATURES APPLY TO ALL MODELS
A6-104

DEFINING THE PROBLEM

The design of a CLAUS Sulfur Recovery Unit (SRU) utilizes combustion of the hydrogen sulfide containing gases in a refractory lined furnace. The waste heat from this combustion process is captured in a shell and tube type exchanger. The typical design of the waste heat exchanger has the combustion gases on the tube side and generates steam on the shell side. The materials of construction are typically type 516-70 carbon steel and carbon steel tubes. The tubesheets are subjected to stresses resulting from pressure, differential tube and shell expansion and temperature gradient across the tubesheet. The tubesheets typically use a knuckle to provide flexibility and utilize the tubes as stays.
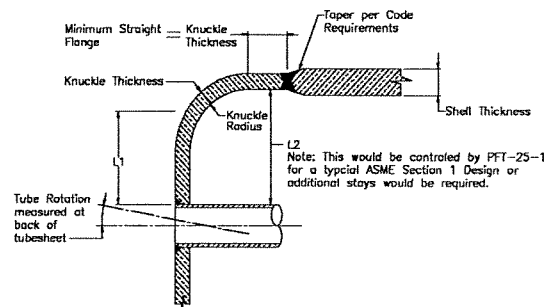
The waste heat exchangers may be designed to ASME Section VIII or Section I depending on local jurisdictional requirements and owners requirements. The design rules in Section VIII do not address the design of the knuckle for these applications. Section I does give some limited design criteria and formula for flanged heads used as tubesheets (PFT-9.22, PFT-24, PW-9, PW-13 and PG-46) but does not address the design directly as a function of the thickness or radius of the knuckle. The thickness of the knuckle is typically equal to 0.75 to 1.0 times the shell thickness with some designs as thin as 0.5 time the shell thickness, (all thickness ratios based on corroded thickness). The inside radius of the knuckle typically ranges from 3 to 8 times the knuckle thickness. The knuckle may be provided as a standard flanged only head, a special knuckle radius applied to a flanged only head or as a forged knuckle welded to the shell and to a flat plate to form the tubesheet. This paper addresses the use of a special knuckle radius applied to a flanged only head. The information

provided may be applied to the other types. The standard flanged only head utilizes a knuckle radius of 3 times the flange thickness. The special forged knuckle typically has the same thickness as the shell where it attaches to the shell and transitions to a thinner section within the radius to match the required tubesheet thickness.

The investigation of tubesheet temperature profiles and stresses was addressed in the Authors' paper referenced in the abstract. The paper provided investigation results of a thermal finite element study and resulting temperature gradients across the tubesheet and other relative temperature profiles required for the stress analysis. The temperature profiles and differential shell to tube sheet expansion used for this investigation was based on the information developed in the referenced paper.

The design of the tubesheet is usually based on the general design rules provided in ASME Section I whether the waste heat exchanger is a Section VIII or Section I. The increased operating temperatures and steam generation pressures in these exchangers have produced concerns for the application of the general design rules. The adaptability of advanced finite element software and parametrically driven modeling abilities had provided a reasonable design tool for this critical tubesheet application.

DESIGN PARAMETERS INVESTIGATED AND REPORTED

The thickness of the tubesheet knuckle area and the distance of the tubes from the knuckle tangent are critical parameters for the designer to establish. Five typical waste heat exchanger designs are investigated by varying the thickness of the tubesheet and the distance of the tubes from the tangent of the repeat knuckle radius. The pressure, thermal and combined maximum tube stresses were investigated and plotted versus tubesheet/shell thickness ratio for one model to quantify the stress level. The corresponding table provides the pertinent information utilized for the investigation. The Section I PG-46 tubesheet thickness required for the tube pitch and PFT-25 maximum unstayed distance from the shell ID to the tube hole edges is presented for comparison. The general duty clause of "application of sound engineering practice" provides the basis for a design investigated by the use of finite element which deviates from the design rules of Section I. The authors recommend the use of Section I or Section VIII Division I allowable stress values with the application of Section VIII Division II procedures for allowable stress applied to pressure and thermal loadings. In the knuckle and the tubesheet areas the FE determined stresses may be compared to 1.5 times the allowable stress for pressure only but the stress should not

exceed the yield of the material at design temperature. The authors recommend limiting the indicated pressure induced stress to less than yield since the redistribution of stresses in a plastically deformed element can not be easily predicted with current FE software. The authors recommend limiting the indicated pressure plus thermal induced stresses to 3.0 time the allowable stress or 2.0 times the yield stress at design temperature. The authors recommend the use of the cyclic loading stress provisions of Section VIII Division II when the pressure plus thermal stresses exceed 3 times the stress allowable or two times the yield stress at design temperature.

FINITE ELEMENT MODELING

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 in the circumferential direction with axial length extended to half of the exchanger length. Over 10,000 elements and 12,000 nodes were used to describe the model which took more than 3 hours of computer time to run on a 200 MHZ Pentium PC. During the initial construction of the model parameters were built into the Cosmos model files for subsequent analysis when the geometry of the exchanger changed. Initial model construction takes 40 hours of engineer's time and the subsequent modeling efforts were reduced to 4 hours. This parametrics modeling approach is discussed further in reference # 7.

The loading on the model included thermal gradients and pressure. The FE analysis included external pressure applied to the tubes which reflected the elongation of the tubes due to the external pressure with simultaneous shortening of the shell due to internal pressure. The results of the FE produced pressure and thermal displacements were found to be in good agreement with closed form solutions which were used to verify the validity of the FE model and analysis results.

Five basic models were constructed and each of the models had two to four different tube sheet thicknesses as a variable. This yielded fifteen individual models that were investigated. Several of the models included tube baffles to more accurately represent the stresses in the tube at the back of the tube sheet.

The authors did not attempt to achieve convergence of the finite element analysis for all the models due to time restraints. Therefore the stress values reported in the appendix are to be considered as representative of relative designs but not adequate to confirm acceptable stress levels. It can be anticipated that the stresses reported would change 5% to 20%

if adequate finite element analysis convergence was obtained. The authors did achieve convergence for model TMD8 which required double refinement of the mesh in the knuckle area. The stress increased 5% and this confirmed that the model was adequate to compare stresses without converging every case. The authors considered convergence to have been achieved when the stress distribution across any element was within 10%.

FINITE ELEMENT ANALYSIS RESULTS

The results of the analysis of the five basic models and the variations of these models are contained in the appendix. The pressure, thermal and combined maximum stresses occurring the knuckle area of each model are plotted versus the nondimensional parameter of tubesheet thickness divided by shell thickness. The maximum pressure, thermal and combined stresses occurring in the tubes at the backside of the tubesheet of model TMD0 are noted in associated table of data. It should be noted that the placement of the tube relative to the tangent of the knuckle is an important parameter for tube rotation and corresponding tube stress.

The results of the analyses indicate that the pressure retaining (non self correcting stresses) knuckle area stresses for knuckle thicknesses greater than 0.75 times the Section required shell thickness can be expected to be within 1.5 times the Section allowable design stresses (and within the yield stress of the material at design temperature). The models that were investigated with knuckle thickness at approximately 0.5 times the Section required shell thickness exceeded 1.5 times the Section allowable design stress and the yield stress of the materials. It should be noted that the greatest stress occurred in the knuckle area for all conditions . The knuckle stresses generally remained acceptable when the corroded tube sheet thickness was at approximately 0.75 times the Section required corroded shell thickness even when the maximum unstayed distance calculated by Section I PFT-25 were exceeded.

The total bending stress (combined pressure plus thermal expansion and thermal gradient stresses) induced in the knuckle are well over the yield stress for the customary carbon steel material but were considered acceptable when they did not exceed 3 times the Section allowed stress or 2 times the yield stress for the material at design temperature. When the combined stress exceeded these parameters the stresses may be evaluated based on the cyclic loading stress criteria presented in Section VIII Division II.

The tube stresses at the back of the tube sheet are influenced by the bending of the tube sheet knuckle which results in the rotation of the tube. Tube rotations in the magnitude of 0.5

degree were noted when the edge of the tube holes were approximately 1.0 times the tube sheet thickness from the tangent of the knuckle. Tube rotations are significantly reduced as the tubes are moved further from the tangent of the knuckle. The tube stresses were not excessive when the edges of the tube holes were placed at least 1.0 times the thickness of the tube sheet knuckle. The shell side baffles which support the tubes have a large effect on the tube rotation induced bending stress. The more baffles or the closer the baffle is to the tube sheet the greater the stress induced in the tube.

CONCLUSIONS

The finite element analysis of several typical waste heat exchangers confirmed that the general approach of utilizing the knuckle to absorb the thermal related expansions is acceptable. The variances applied to the tubesheet knuckle thickness and the tube hole placement relative to the tangent of the knuckle indicated that care must be utilized in selecting these parameters.

The generally accepted Section I PFT-9.2.2 requirement for the tube sheet straight flange thickness to be 0.75 times the Section required shell thickness appears to be a good rule of thumb estimation for selection of the straight flange and knuckle thickness to begin a thorough analysis. The authors did not confirm that the knuckle thickness could be approximately 0.5 times the Section required shell thickness as the indicated pressure induced stresses appear excessive for the models studied.

The stress intensity reported in the appendix is based on the maximum stress intensity located at the knuckle inside diameter. This stress was found to be distributed over a considerable amount of the interior surface of the knuckle but the stress did not extend over ¼ of the thickness of the knuckle. The authors consider this stress and its location to be applicable to the use of Section VIII Division II stress criteria methodology.

RECOMMENDATIONS

The authors recommend that the designers of this type of equipment investigate the tube sheet carefully and recommend the use of finite element software for this analysis.

The authors recommend the use of Section I or Section VIII Division I, respective to overall code stamping, allowable stress values with the application of Section VIII Division II procedures for allowable stress versus pressure, thermal and cyclic loadings. The authors recognize that the current finite

element analysis is limited to elastic (linear) applications. The authors consider this acceptable when the non self correcting stresses produced by pressure containment do not exceed 1.5 times the Section allowed stresses and does not exceed the yield of the materials at design temperature.

The authors recognize that a linear finite element analysis determined stress above yield is not a true stress as the analysis does not correct for permanent strain resulting from stresses above yield. The designer should note that some linear finite element software codes have some compensation for strain redistribution of stresses and this may influence the finite element indicated stress (reference # 6). The authors recommend that the self correcting stresses such as those induced by thermal expansion or thermal gradients, when calculated by use of linear finite element software, be limited to 3 times the Section allowed stress but not exceeding 2 times the yield stress of the material at design temperature. When the combined stresses exceed the above the design may be evaluated based on the cycle stress criteria in Section VIII Division II. The authors recommend that the designer utilize reasonable conservatism in establishing design cycle life with the owner or final operator.
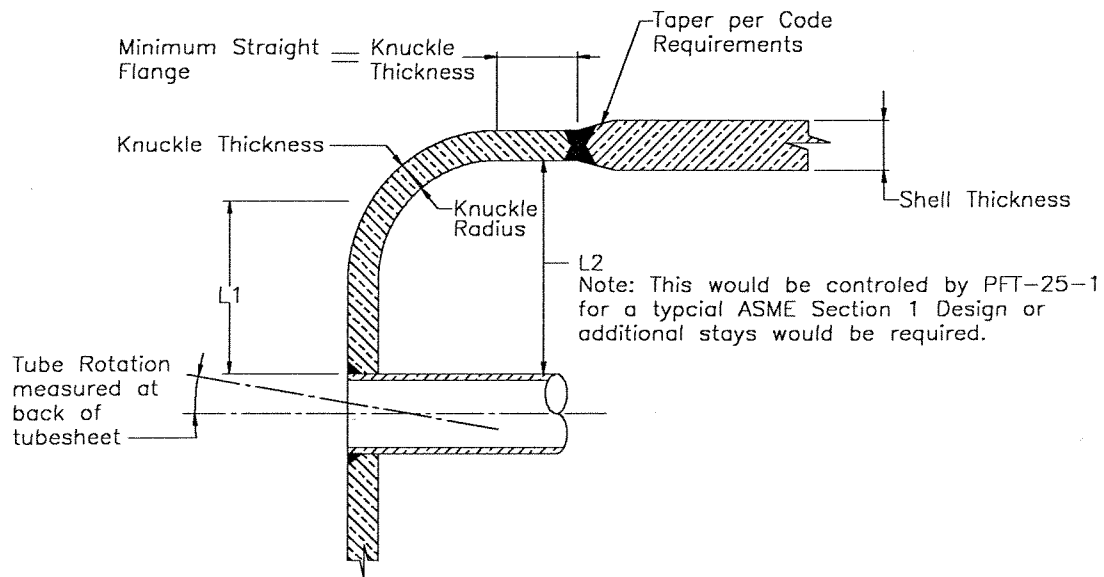
The authors recommend all FE analysis to be carefully modeled by use of 3D techniques. The model must include all pressure and thermal information to be an accurate representation of the equipment. The authors recommend that the displacements of the model for both pressure and thermal be confirmed by closed form solution to verify the correctness of the model and the analysis.

REFERENCES

1. ASME Code Section I, 1996.
2. ASME Code Section VIII Division I and II, 1996.
3. Desai/Abel, 1972, Introduction to the Finite Element Method, Van Nostrand Reinhold Company, New York, NY.
4. J. L. Hechmer & G. L. Hollinger, August 5, 1995, 3D stress Criteria: Guidelines for Application, ASME PVRC Report.
5. W. C. Kroenke, 1973 Classification of Finite Elements Stresses According to ASME Stress Categories, June 1974, The American Society of Mechanical Engineers, New York, NY., pp 107-140.
6. A. H. Primm and J. E. Stoneking, 1989, Accuracy of the Finite Element Method for Pressure Vessels, 1989 P-175.
7. Dennis Martens et al, Analysis of tubesheet stresses in a Sulfur Recovery Unit, 1996 PVP Proceedings Volume.
8. Michael Porter et al, A Comparison of Stress Results from Several Commercial Finite Element Codes with ASME Section VIII Division 2 Requirements, 1996 PVP Proceedings Volume.
9. Dennis Martens et al, Nozzle Stiffness and Stress Computation Using a Parametrically Controlled Finite Element Modeling Approach, 1996 PVP Proceedings Volume.
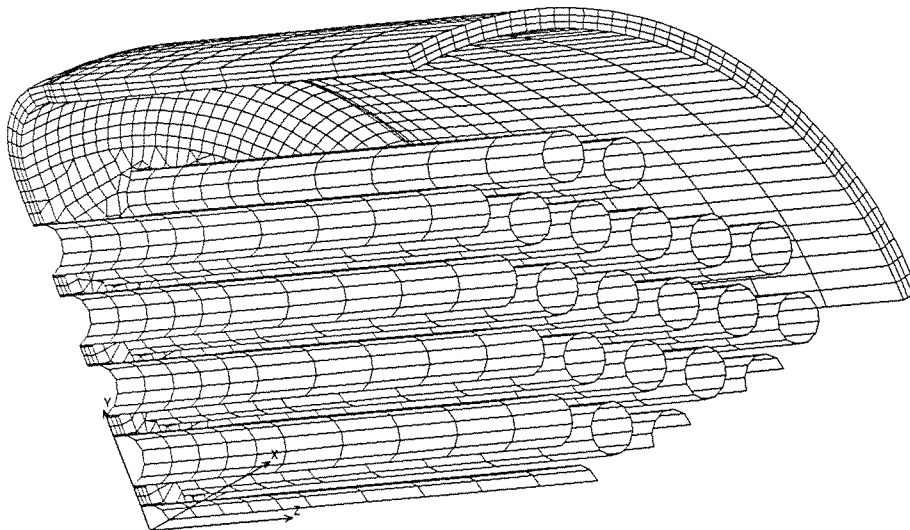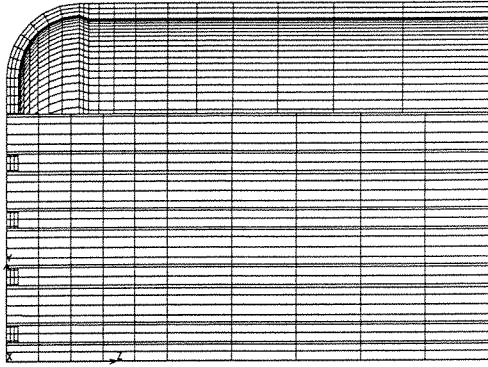
APPENDIX

1

Minimum Straight — Knuckle
Flange = Thickness

Taper per Code
Requirements

Knuckle Thickness

Knuckle
Radius

Shell Thickness

L1

L2
Note: This would be controled by PFT−25−1
for a typcial ASME Section 1 Design or
additional stays would be required.

Tube Rotation
measured at
back of
tubesheet

Nomenclatures apply to all models

3-D MODEL (TYPICAL)

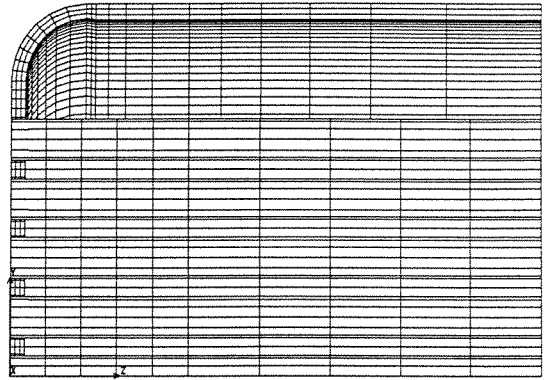|	tmd0
	2-16-97

MODEL A

Shell : 500 F
Tubes : 560 F
Int. P : 765 psig
Tubesheet Thk/Shell Thk : 0.75

ATMD0
2-18-97

MODEL B

Shell : 500 F
Tubes : 560 F
Int. P : 765 psig

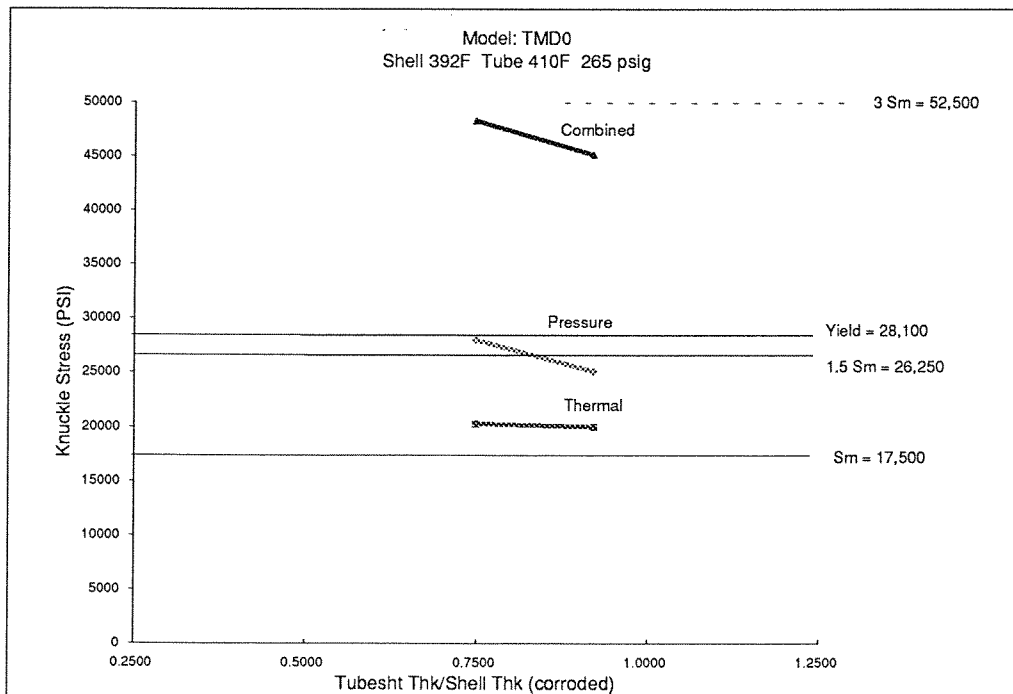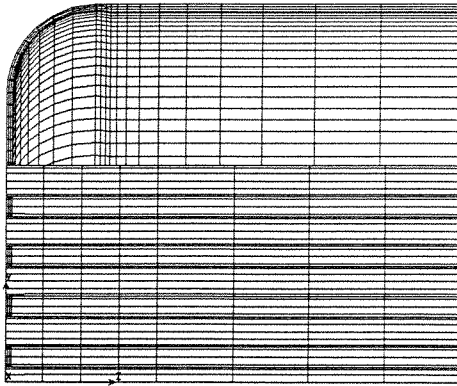Tubesheet Thk/Shell Thk : 0.923

tmd0
2-16-97

Model: TMD0
Shell 392F Tube 410F 265 psig

3 Sm = 52,500

Combined

Pressure

Yield = 28,100

1.5 Sm = 26,250

Thermal

Sm = 17,500

Knuckle Stress (PSI)

Tubesht Thk/Shell Thk (corroded)

TABLE 1   MODEL TMD0

Note: all dimensions in inches @ corroded condition CA= shell @ 1/8" and tubesheet @ 1/4" and tubes at 0""

ITEM	MODEL # A	MODEL # B
Shell OD	75.5	75.5
Shell thickness	1.625	1.625
Tubesheet thick	1.219	1.5
Tubesheet thick / Shell thickness	0.75	0.923
Knuckle radius @ ID	6.0	6.0
Knuckle radius / tubesheet thickness	4.92	4.0
Tube pitch	6.0	6.0
Tube OD	4.5	4.5
Tube thick	0.212	0.212
Tube length	336	336
L1-outside tube hole to tangent of knuckle	1.61	1.61
L1 / tubesheet thickness	1.32	1.07
L2-outside tube hole to ID of shell (corroded ID)	7.61	7.61
L3=L2 plus 0.6x Pitch for comparison to 1.5p below	11.21	11.21
1.5 p calculated per Section I par PFT-25.2	13.08	16.1
Minimum thickness of tubesheet per PG-46.1	0.839	0.839
Maximum tube pitch based on actual tubesheet thickness per PG-46.1	8.72	10.732
Tube rotation degrees	0.486 deg	0.529 deg
Tube Stress at Back of Tubesheet *		
Pressure (Hoop) Stress	7,989	7,989
Pressure Stress	28,000	25,100
Thermal Stress	20,300	20,000
Combined Stress	48,300	45,100
Calculated tube compressive load KIPs	13,800 lb	14,900 lb
Allowable tube compressive load KIPs per critical buckling	73,000 lb	73,000 lb

* The FEA modeling techniques for this tube sheet and tube junction are similar to modeling performed in ref. #8. These stresses occur within 2t of the discontinuity and should be considered similar to that reference.
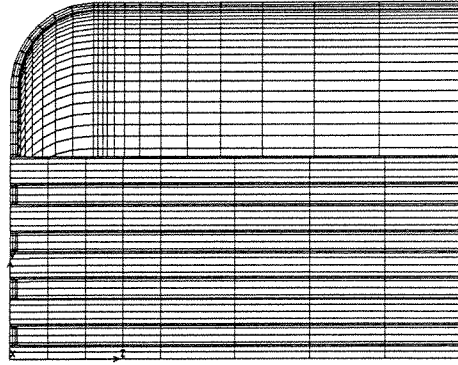
MODEL A

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig
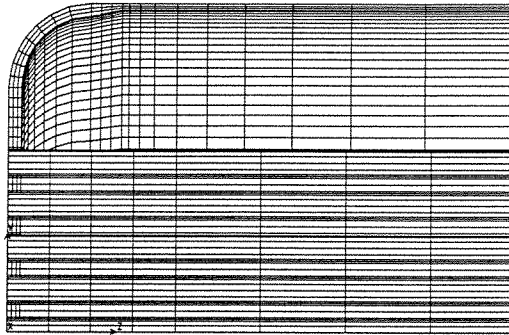
Tubesheet Thk/ Shell Thk : 0.75

BTMD2
2-18-97

MODEL B

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig

Tubesheet Thk/Shell Thk : 1.0

ATMD2
2-18-97

MODEL C

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig
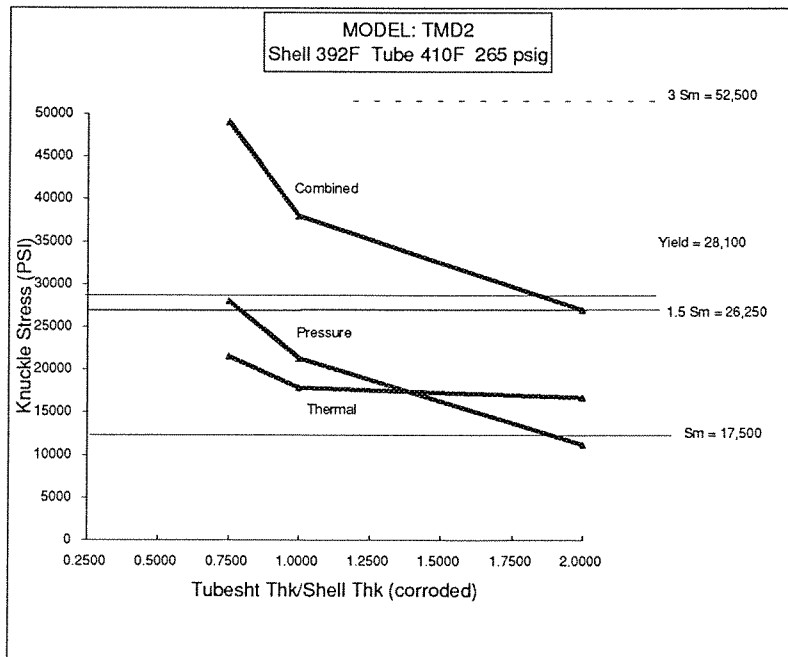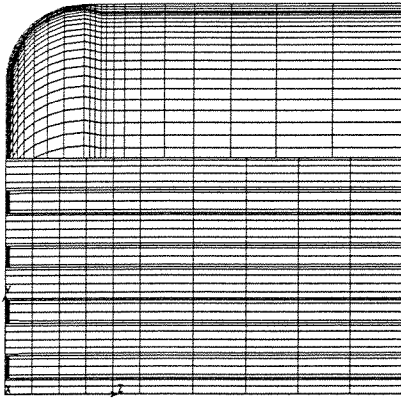
Tubesheet Thk/Shell Thk : 2.0

TMD2
2-16-97

MODEL: TMD2
Shell 392F Tube 410F 265 psig

3 Sm = 52,500

Combined

Yield = 28,100

1.5 Sm = 26,250

Pressure

Thermal

Sm = 17,500

Knuckle Stress (PSI)

Tubesht Thk/Shell Thk (corroded)

Table 2 Model TMD2

Note: all dimensions in inches @ corroded condition CA= shell @ 1/8", tubesheet @ 1/4" and tubes at 0"

| ITEM	MODEL # A	MODEL # B	MODEL # C
Shell OD	52.25	52.26	52.26
Shell thickness	0.5	0.5	0.5
Tubesheet thick	0.375	0.5	1.0
Tubesheet thick / Shell thickness	0.75	1.0	2.0
Knuckle radius @ ID	6.12	6.12	6.12
Knuckle radius / tubesheet thickness	16.32	12.24	6.12
Tube pitch	3.375	3.375	3.375
Tube OD	2.375	2.375	2.375
Tube thick	0.25	0.25	0.25
Tube length	360	360	360
L1-outside tube hole to tangent of knuckle	3.61	3.61	3.61
L1 / tubesheet thickness	9.62	7.22	3.61
L2-outside tube hole to ID of shell (corroded ID)	9.73	9.73	9.73
L3=L2 plus 0.6x Pitch for comparison to 1.5p below	11.75	11.75	11.75
1.5 p calculated per Section I par PFT-25.2	6.84	9.12	12.16
Minimum thickness of tubesheet per PG-46.1	0.278	0.278	0.278
Maximum tube pitch based on actual tubesheet thickness per PG-46.1	4.56	6.08	18.24
Tube rotation degrees	.0.209 deg	0.448 deg	0.587 deg
Calculated tube compressive load KIPs	3,570 lb	8,460 lb	10,900 lb
Allowable tube compressive load KIPs per critical buckling	63,000 lb	63,000 lb	63,000 lb

MODEL A

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig

Tubesheet Thk/Shell Thk : 0.5
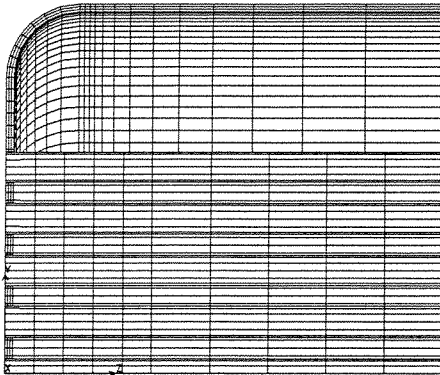
CTMD4
2-17-97

MODEL B
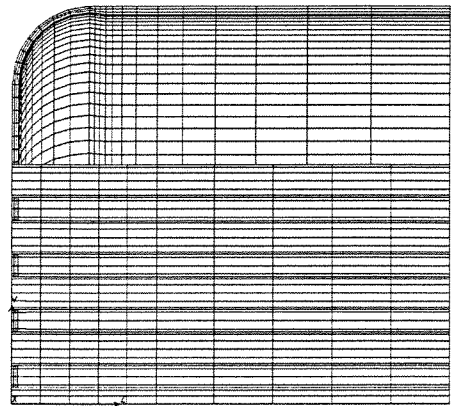
Shell : 392 F
Tubes : 410 F
Int. P : 265 psig
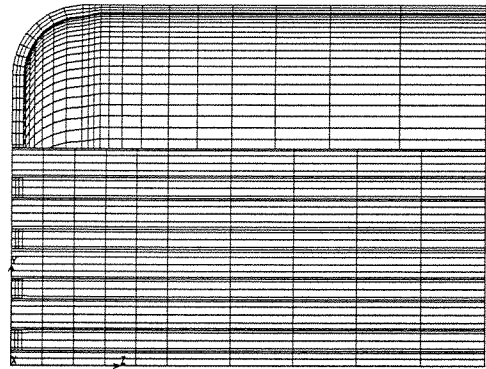Tubesheet Thk/Shell Thk : 0.75

atmd4
3- 2-97

MODEL C

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig

Tubesheet Thk/Shell Thk : 1.0

BTMD4
2-17-97

MODEL D

Shell : 392 F
Tubes : 410 F
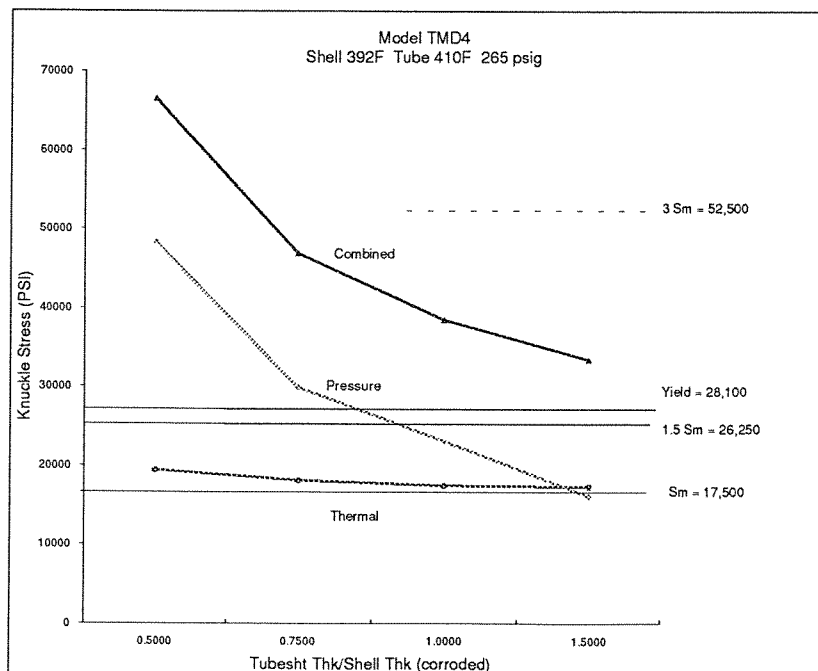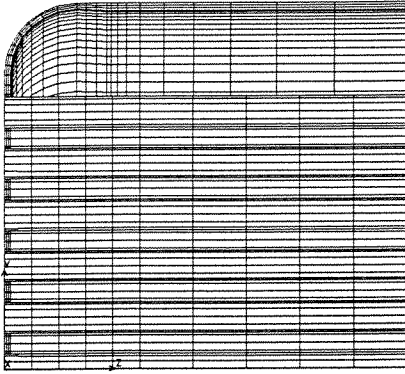Int P : 265 psig

Tubesheet Thk/Shell Thk : 1.5

tmd4
2-17-97

Model TMD4
Shell 392F Tube 410F 265 psig

Knuckle Stress (PSI) vs Tubesht Thk/Shell Thk (corroded)

3 Sm = 52,500

Combined

Pressure

Yield = 28,100

1.5 Sm = 26,250

Thermal

Sm = 17,500

Table 3 Model TMD4

Note: all dimensions in inches @ corroded condition CA= shell @ 1/8", tubesheet @ 1/4" and tubes at 0"

| ITEM	MODEL #A	MODEL #B	MODEL #C	MODEL #D
Shell OD	48.25	48.25	48.25	48.25
Shell thickness	0.5	0.5	0.5	0.5
Tubesheet thick	0.25	0.375	0.5	0.75
Tubesheet thick / Shell thickness	0.5	0.75	1.0	1.5
Knuckle radius @ ID	4.25	4.25	4.25	4.25
Knuckle radius / tubesheet thickness	17.0	11.33	8.50	5.67
Tube pitch	3.375	3.375	3.375	3.375
Tube OD	2.375	2.375	2.376	2.375
Tube thick	0.25	0.25	0.25	0.25
Tube length	360	360	360	360
L1-outside tube hole to tangent of knuckle	3.5	3.5	3.5	3.5
L1 / tubesheet thickness	14	9.33	7	4.67
L2-outside tube hole to ID of shell (corroded ID)	7.75	7.75	7.75	7.75
L3=L2 plus 0.6x Pitch for comparison to 1.5p below	9.78	9.78	9.78	9.78
1.5 p calculated per Section I par PFT-25.2	3.04	4.56	6.08	9.12
Minimum thickness of tubesheet per PG-46.1	0.278	0.278	0.278	0.278
Maximum tube pitch based on actual tubesheet thickness per PG-46.1	4.56	6.84	9.12	13.68
Tube rotation degrees	0.289 deg	0.184 deg	0.137 deg	0.258 deg
Calculated tube compressive load KIPs	4,220 lb	11,400 lb	8,950 lb	14,300 lb
Allowable tube compressive load KIPs per critical buckling	63,000 lb	63,000 lb	63,000 lb	63,000 lb

MODEL A

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig
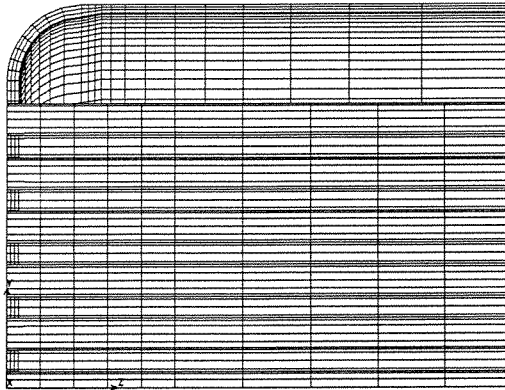
Tubesheet Thk/Shell Thk : 0.75

	BTMD6
	2-17-97

MODEL B
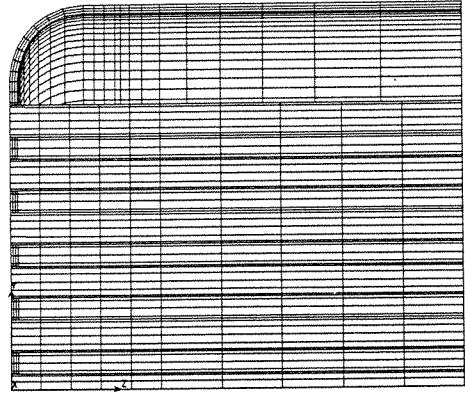
Shell : 392 F
Tubes : 410 F
Int. P : 265 psig

Tubesheet Thk/Shell Thk : 1.0

	ATMD6
	2-17-97

MODEL C

Shell : 392 F
Tubes : 410 F
Int. P : 265 psig

Tubesheet Thk/Shell Thk : 1.5

	TMD6
	2-17-97

Model: TMD6
Shell 392F Tube 410F 265 psig

Knuckle Stress (PSI) vs Tubesht Thk/Shell Thk (corroded)

3 Sm = 52,500

Combine

Yield = 28.100

1.5 Sm = 26,250
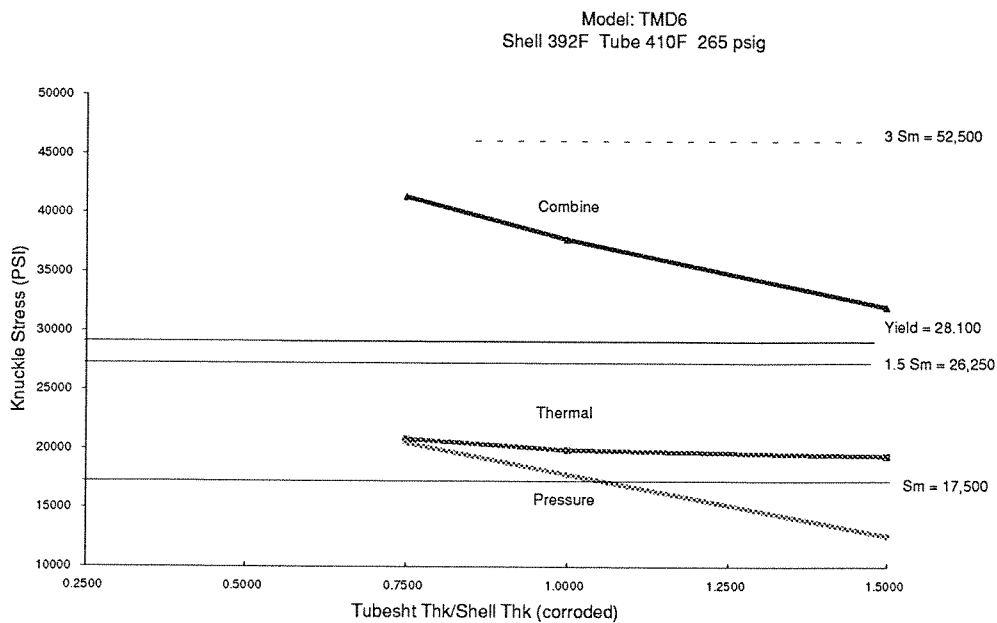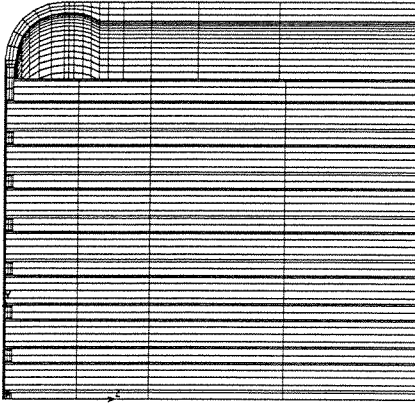
Thermal

Pressure

Sm = 17,500

Table 4 MODEL TMD6

Note: all dimensions in inches @ corroded condition CA= shell @ 1/8", tubesheet @ 1/4" and tubes at 0"

ITEM	MODEL # A	MODEL # B	MODEL # C
Shell OD	48.25	48.25	48.25
Shell thickness	0.5	0.5	0.5
Tubesheet thick	0.375	0.5	0.75
Tubesheet thick / Shell thickness	0.75	1.0	1.5
Knuckle radius @ ID	4.25	4.25	4.25
Knuckle radius / tubesheet thickness	11.33	11.33	5.67
Tube pitch	3.375	3.375	3.375
Tube OD	2.375	2.375	2.375
Tube thick	0.25	0.25	0.25
Tube length	360.00	360.00	360.00
L1-outside tube hole to tangent of knuckle	1.3125	1.3125	1.3125
L1 / tubesheet thickness	3.5	2.625	1.75
L2-outside tube hole to ID of shell (corroded ID)	5.56	5.56	5.56
L3=L2 plus 0.6x Pitch for comparison to 1.5p below	7.58	7.58	7.58
1.5 p calculated per Section I par PFT-25.2	6.84	9.12	9.12
Minimum thickness of tubesheet per PG-46.1	0.278	0.278	0.278
Maximum tube pitch based on actual tubesheet thickness per PG-46.1	4.56	6.08	13.68
Tube rotation degrees	0.058 deg	0.096 deg	0.119 deg
Calculated tube compressive load KIPs	2.540 lb	4,880 lb	6,210 lb
Allowable tube compressive load KIPs per critical buckling	63,000 lb	63,000 lb	63,000 lb

MODEL A

Shell : 516 F
Tubes : 543 F
Int. P : 790 psig
Tubesheet Thk/Shell Thk : 0.566

ctmd8
3-11-97

MODEL B

Shell : 516 F
Tubes : 543 F
Int. P : 790 psig
Tubesheet Thk/Shell Thk : 0.75

BTMD8
2-17-97

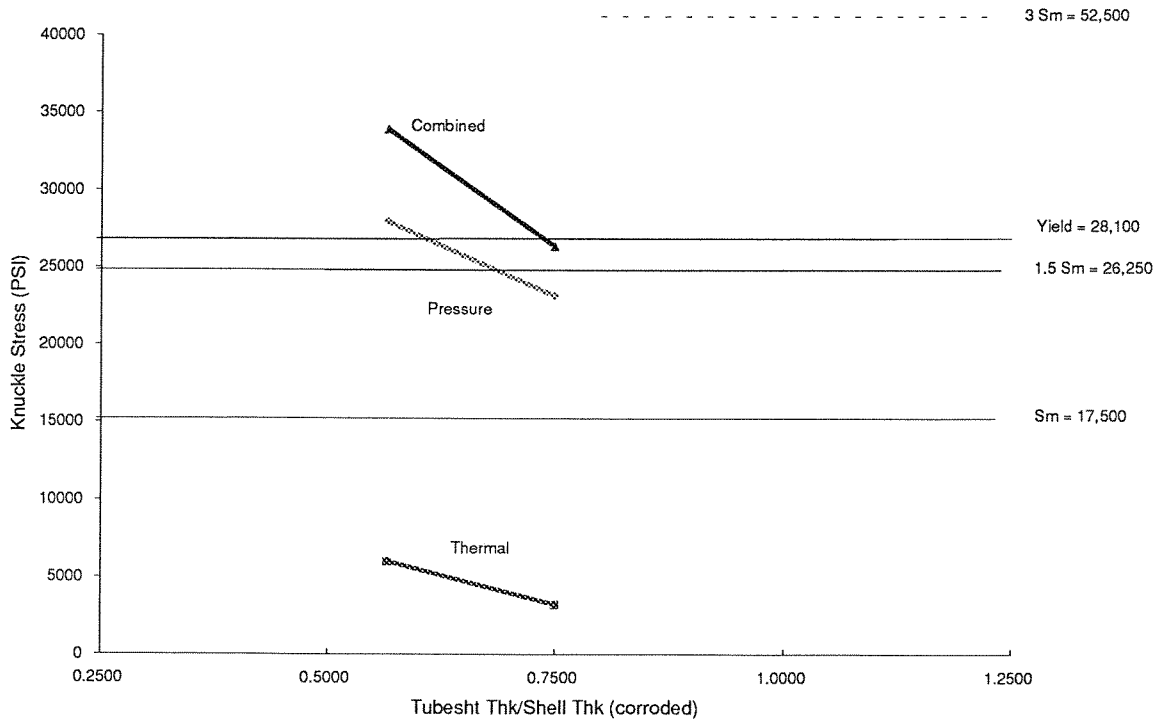Model: TMD8
Shell 516F Tube 543F 790 psig

3 Sm = 52,500

Combined

Yield = 28,100

1.5 Sm = 26,250

Pressure

Sm = 17,500

Thermal

Knuckle Stress (PSI)

Tubesht Thk/Shell Thk (corroded)

Table 5 Model TMD8

Note: all dimensions in inches @ corroded condition CA= shell @ 1/8", tubesheet @ 3/8" and tubes at 0"

ITEM	MODEL #A	MODEL #B
Shell OD	120.8665	128.9375
Shell thickness	2.867	2.9945
Tubesheet thick	1.625	2.1875
Tubesheet thick / Shell thickness	0.566	0.75
Knuckle radius @ ID	7.125	8.625
Knuckle radius / tubesheet thickness	4.4	4
Tube pitch	6.625	6.625
Tube OD	5.0	5.0
Tube thick	0.48	0.48
Tube length	252	252
L1-outside tube hole to tangent of knuckle	1.902	4.375
L1 / tubesheet thickness	1.02	2.03
L2-outside tube hole to ID of shell (corroded ID)	9.03	16.98
L3=L2 plus 0.6x Pitch for comparison to 1.5p below	13.0	20.955
1.5 p calculated per Section I par PFT-25.2	17.16	22.44
minimum thickness of tubesheet per PG-46.1	0.941	0.941
maximum tube pitch based on actual tubesheet thickness per PG-46.1	11.44	14.96
Tube rotation degrees	0.0865 deg	0.0504 deg
Calculated tube compressive load KIPs	5,430 lb	4,610 lb
Allowable tube compressive load KIPs per critical buckling	130,000 lb	130,000 lb
Stress Intensity @ Knuckle		
Thermal	5,970 psi	3,000 psi
Pressure	26,200 psi	23,100 psi
Combined	32,170 psi	26,300 psi