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NUMERICAL INVESTIGATION OF STRESS GENERATED IN HIGH PRESSURE HEAT EXCHANGER

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ABSTRACT

Heat Exchangers are used to transfer heat effectively from one medium to another medium. There are several aspects to study the performance of heat exchanger. This paper is concerned with thermo-mechanical issues i.e. thermal expansion due to high temperature and high pressure conditions of U-tube heat exchanger. Tubesheet is very complex part of heat exchanger which expands at high temperature. Due to high temperature difference between shell side and channel side fluids thermal stress are generated in the tubesheet which effects on the performance of the heat exchanger. 3D FEA model was modeled in ANSYS® to study the thermo-mechanical effect on heat exchanger. Mesh sensitivity analysis was performed to obtain precise results and optimum mesh size. Static structural stress analysis was performed under for two conditions, at first only mechanical loading was studied and secondly mechanical and thermal loading effects were studied. In steady state condition, tubesheet thickness was optimized using 3D parametric model in FEA. The results of the elastic stress analysis were evaluated as per ASME Section VII DIV-2 code limits. It is found that with the optimization design, the tubesheet thickness could be reduced by 20-25% without affecting the safety of the heat exchanger within the allowable limits.

KEYWORDS: Heat exchanger, Tubesheet optimization, Steady state condition, Stress analysis, FEA.

INTRODUCTION

Heat transfer equipment, such as the shell and tube heat exchanger and the waste heat boiler, is widely used in the process industry to recover heat or meet the process requirements. Heat exchangers are frequently called the workhorses in process and petrochemical plants, and more than 65% of these are (shell and tube type) tubular heat exchangers. Tubular heat exchangers exemplify many aspects of the challenges in the (mechanical) design of pressure vessels. Detail understandings of the interrelationships between the thermal and the mechanical performance of heat exchangers and many areas of mechanics are necessary for design of the exchanger [1].

The tube-sheet is the essential component of heat transfer equipments. Other three major parts of the heat transfer equipments are the shell, the header and the tube bundle which are connected to tube sheet. Due to temperature and pressure variation on both sides of tubesheet, it experiences both pressure stresses and thermal stresses [2, 3].There are several codes and standards used for design of heat exchangers. ASME Codes Section VIII DIV 1 and DIV 2, Tubular Exchanger Manufactures Association (TEMA), European Standard for Unfired Pressure vessels EN 13445 are mostly used. Heat exchangers made of expensive materials with large dimensions. The cost can be reduced by using efficient design methods and appropriate code and standards. K. Behseta had shown that the heat exchanger ASME DIV 1 codes and EN 13445 codes gives unnecessarily large thickness of the tubesheet which can be reduced using FEA [3].

ASME VIII Division 2 considers perforated tubesheet, unperforated rim and more rigorous interaction of the tubesheet with shell/channel in the new method which is more rigorous than TEMA.TEMA method allows the bending stress higher up to 2.6S whereas ASME VIII Division 2 recommends lower bending stresses up to 1.5S. Based on the stress classifications of ASME Section VIII Division 2 Appendix 4, maximum stress in the tubesheet in ASME method limited to the appropriate allowable stress. Therefore tubesheets designed according to ASME Part UHX will generally be thicker than TEMA tubesheets [4].It is noted that neither ASME nor TEMA has a design condition specifically related to the hydrostatic pressure test. However this is a design case that should be considered and included in the calculation checks. The allowable stresses will need to be higher than those for design and operating conditions. An allowable tensile stress of 90% yield is a good starting point but for compression/buckling loads the basis is less clear [5].

Conventional tubesheet design is based on modified elastic-plate bending theory, in which the perforated tubesheet is treated as a thin homogeneous plate with modified material properties. The conventional approach is safe and functionally effective but may lead to over-conservative designs in which the plate thickness is greater than that required to safely contain the pressurized fluids in the heat exchanger. Difference in maximum permissible pressures results partly from different nominal design stress. Tubesheet thickness to tube pitch ratio of heat exchanger should be more than 2.0 for considering ASME VIII/2 code division. To capture the mechanical properties of the perforated tubesheet without modeling each individual hole, the tubesheet is represented by an equivalent solid plate which has the same elastic-plastic and thermal properties [6]. Due to the perforation pattern, the equivalent solid material has orthotropic (directionally dependent) properties. The elastic properties of this equivalent solid are transversely isotropic, i.e., directionally independent in the tubesheet plane, but different in the out-of-plane direction. Since an equivalent solid material is replaced the perforated region of the tubesheet, it becomes convenient to obtain a criterion for elastic shakedown (lower bound for shakedown) for this material, similar to the,$(P_M + P_B + Q) \leq 3S$ limit of the ASME Code [5].

Different approaches have been used in the FEA modeling of tubesheet and tube bundle. Tubesheet is modeled as an equivalent solid plate on elastic foundation. Shell and Tube Heat Exchanger is symmetrical in two meridional planes and considered to be symmetric with regard to cross section in the middle between the tubesheet [3].The elastic design procedures use a stress categorization methodology to guard against failure due to gross plastic deformation and progressive plastic deformation or ratcheting. In practice, 3D FEA is employed to calculate the elastic stress field, with a stress linearization procedure employed to evaluate membrane and bending stresses for design assessment. Through evaluating the stress intensity of the tubesheet, it is found that the dangerous region is located at the edge of the tube distribution region and local stress concentration along the edge of the holes on tubesheet [3]. This approach can yield a less conservative design than design by rule but does not lead to the most effective use of material.

SPECIFIC PARAMETERS OF THE HEAT EXCHANGER

The heat exchanger was considered in this paper is an ammonia synthesis loop boiler. The relevant design parameters and material considered for this investigation are as below:

Table 1: Design Parameters of the Exchanger

Tubes			Pitch	42	mm	Square Pitch		
Count	180		O.D.	32	mm	Wall thickness	3.6	mm
Design pressure at max temp	22.065	MPa	Internal Pressure	22.065	MPa	External pressure	12.356	MPa
Shell			I.D.	1340	mm	Wall thickness	55	mm
Design pressure at max temp	12.356	MPa	Internal pressure	12.356	MPa			
Channel			I.D.	1230	mm	Wall thickness	105	mm
Design pressure at max temp	22.065	MPa	Internal pressure	22.065	MPa			
Tubesheet			O.D.	1440	mm	Tubesheet thickness	300	mm
Design pressure at max temp	22.065	MPa	Channel side pressure	22.065	MPa	Shell side pressure	12.356	MPa

Table 2: Material Properties of Exchanger Parts

	Material	Design Temperature (°C)	Thermal conductivity (W/m °C)	Coefficient of thermal expansion (mm/mm °C)	Modulus of elasticity (MPa)	Poisson's Ratio
Tubesheet	SA 336 F22 CL3	430	34.88	1.40E-05	181112.22	0.3
Channel	SA 336 F22 CL3	380	35.72	1.37E-05	185458.05	0.3
Shell	SA 302 GR B	330	38.2	1.41E-05	179611.29	0.3
Tube	SA 213 T 22	380	35.72	1.37E-05	185458.05	0.3

NUMERICAL ANALYSIS

3.1 FEA Model

FEA model of the heat exchanger is generated in ANSYS 15, which consists of tubesheet, tube-channel, part of shell and tubes. Heat Exchanger is symmetrical along tubesheet axis, $1/8^{th}$ section model was modeled for simplification of analysis. All nodes at the meridional sides of the tubesheet and shells are assigned with symmetry boundary conditions. Fringing effect caused by shell, tubes, and channel should be considered in tubesheet analysis. Therefore, in FEA analysis minimum vertical length of shell, channel and tube is taken as per ASME, not less than $2.5\sqrt{Rt}$, where R and t are inside radius and thickness of vessel respectively [5]. With this requirement, the extended length of shell, channel, and tube should be more than 480 mm, 635 mm, and 17 mm respectively. For this consideration length of shell, channel and tubes are 500 mm, 650 mm and 100 mm respectively.

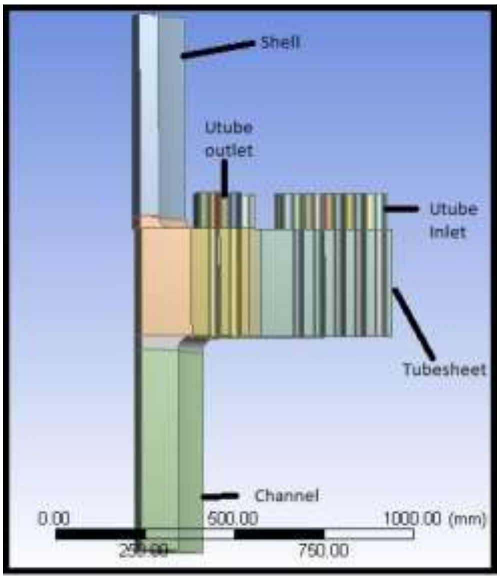

Figure 1: Computational Model of Heat Exchanger

3.2 Mesh Model

Solid eight node quad elements were preferred while meshing. Along the cross-section of tube, channel, shell minimum two numbers of nodes were generated. Midside nodes of element are dropped because it creates large number of nodes which leads model more complex to solve for simple computers and takes lot of time. Transition is kept slow for slow transformation meshing from larger element to adjacent small size elements for better results and vice versa. Element size is kept default for all type of mesh. Span angle center should be medium in this case, because small angle leads to very large number of elements and large angle leads very less elements leading error in solution. In Tetrahedral Element midside nodes are useful, but for Solid Quad Nodes there is need of midside nodes. Therefore, midside nodes are dropped for mesh generation.

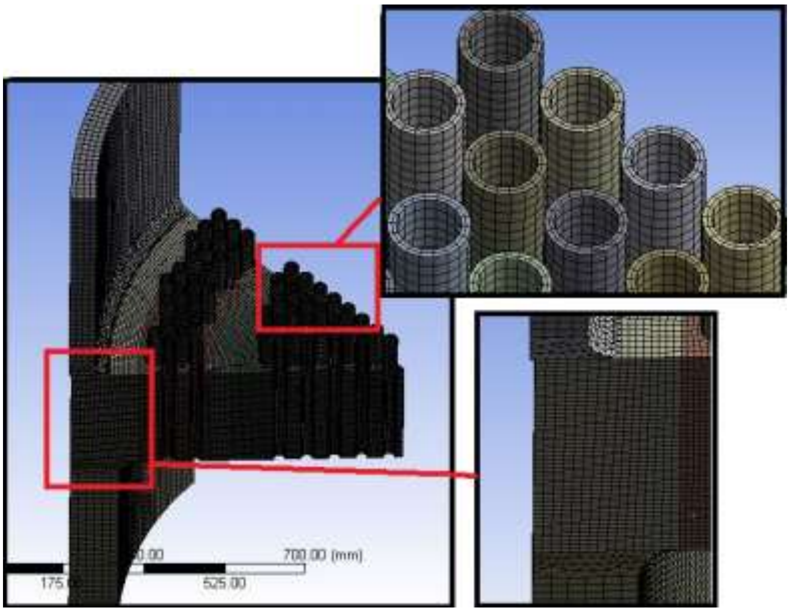

Figure 2: Mesh of FEA model

3.3 Boundary Conditions

For checking effect of pressure on exchanger without considering effect of temperature, thermal loads in the process were kept at atmospheric condition and some boundary constraints along with pressure forces are applied at exchanger surface. Base of exchanger was assumed fixed therefore at the base vertical displacement was restricted. Circumferential displacement along the tubesheet and other part cut section was restricted by giving frictionless support at the cut cross section. Shell side pressure was applied on shell side and tube side pressure was on tube side. An equivalent axial pressure force had applied on upper cross section of shell and tubes and is calculated as below.

$$P_a = \frac{P_e d_i}{4t}$$

Where, P_e −Effective pressure acting on vessel surface, d_i − Internal Diameter of vessel and t − Wall thickness of vessel.

3.4 Stress Categorization Lines

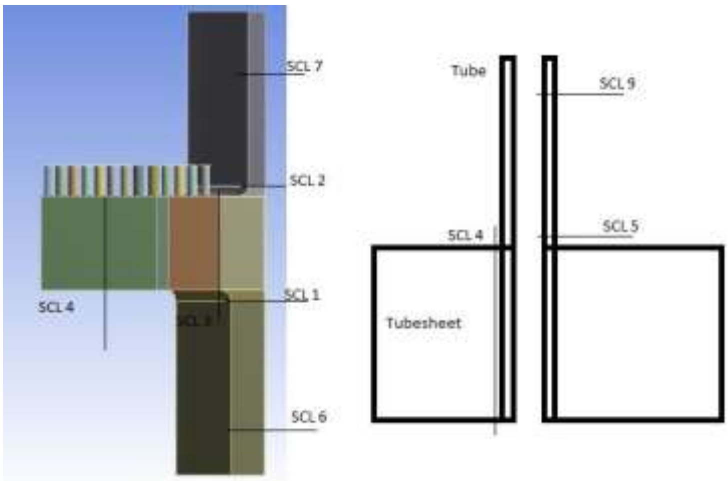

Figure 3: Stress Categorization Lines

SCL 1 is horizontal line on channel near tubesheet and channel weld, at this SCL local stresses in channel are evaluated. SCL 2 is horizontal line on shell near tubesheet and shell weld, at this SCL local stresses in shell are evaluated. SCL 3 and SCL 4 are vertical lines on Tube sheet near tubesheet-channel weld and tubesheet-tube joint respectively; at this tubesheet local stresses are evaluated. SCL 5 is horizontal line on tube near tubesheet-tube joint; here local stresses in tube are evaluated. SCL 6, SCL 7 and SCL 9 are horizontal lines on channel, shell, and tube away from any discontinuity respectively and SCL8 is vertical line on tubesheet where very less discontinuity present. SCL1, SCL2, SCL3, SCL4, and SCL5 are at structural discontinuity area, therefore 1.5S and 3S criteria is applied to membrane stress and membrane stress plus bending stress, respectively. For other paths, 1S and 1.5S is applied to membrane stress and membrane plus bending stress, respectively.

ASME codes along with FEA method had been used for design and optimization of tubesheet. Tubesheet thickness is optimized by FEA. Four models of heat exchangers were generated in ANSYS. First model was with original designed tubesheet thickness i.e. 300 mm, and other three models were with reduced tubesheet thickness by 10%, 20%, and 30%. Above four models were analyzed for two loading case, in first loading case only mechanical pressure forces are applied. In the second case along with pressure forces and thermal loads is applied. Results are obtained for these three models were compared with original tubesheet model.

TUBESHEET OPTIMIZATION AND RESULTS

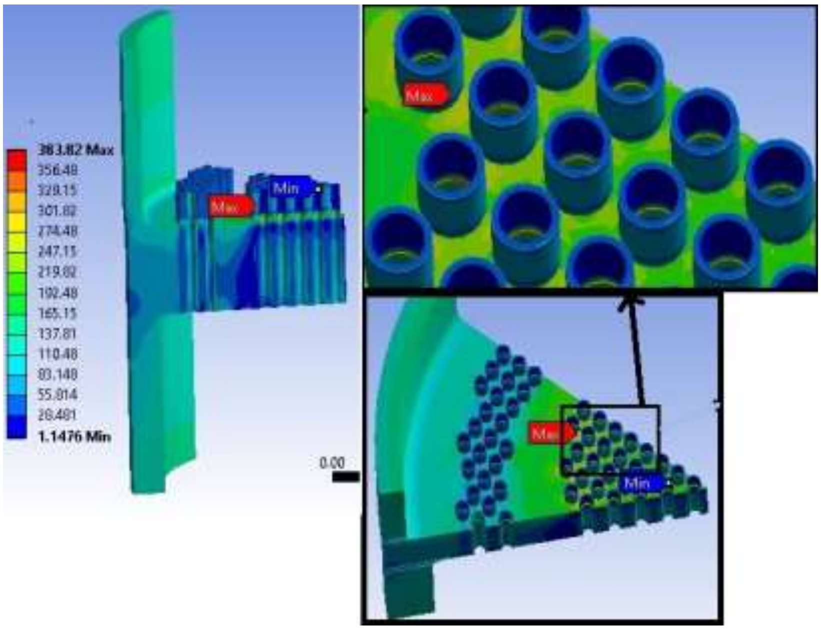

Figure 4: Equivalent (Von Misses) Stress Distribution of Load Case 2 of original model

From the stress distribution contour, it is cleared that maximum stress intensity was at tube to tubesheet joints. Because tubesheet is in contact with the shell side fluid and tube side fluid. Pressure from both sides of fluids and temperature difference of both sides creates more stresses at tubesheet. At the inner portion of the tubesheet more variation observed in the stress distribution compared to outer portion.

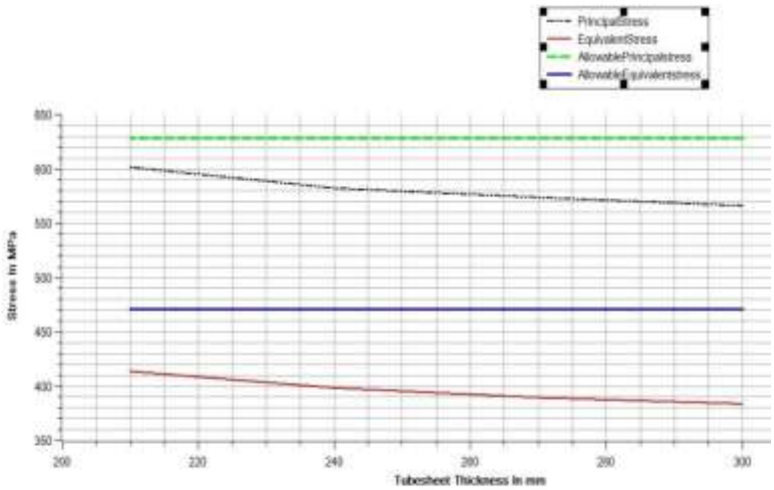

Figure 5: Principal stress and equivalent von misses Stress

It is cleared (Figure 5) that as tubesheet thickness decreases maximum principal stress in the heat exchanger increases. Maximum Equivalent Stress was calculated based on Von Misses stress theory. Von Misses stress was generated in the exchanger. Von Misses stress increases as tube sheet thickness decreases. For the optimized heat exchanger model principal stress and equivalent stress are much nearer to the allowable stress limit, whereas for original model it was far away of the allowable stress limit.

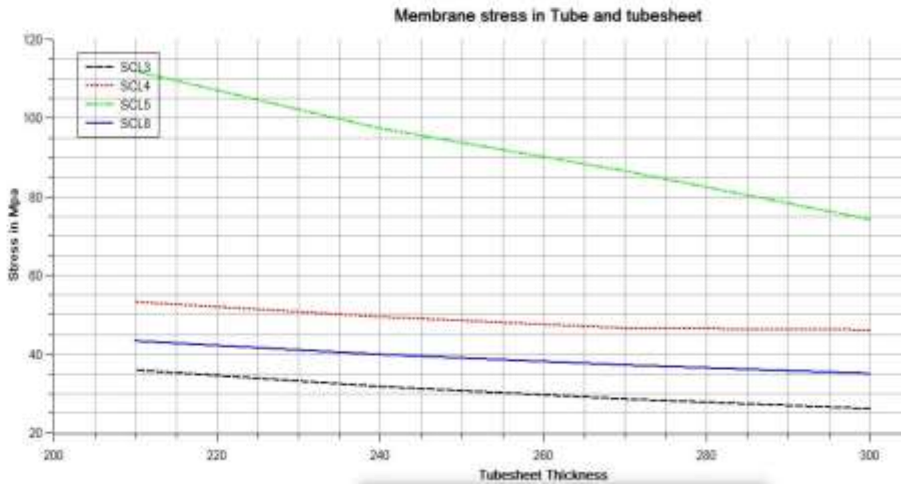

Figure 6: Maximum membrane Stress intensity at SCL 3, 4, 5 and 8 for Loading Case 1

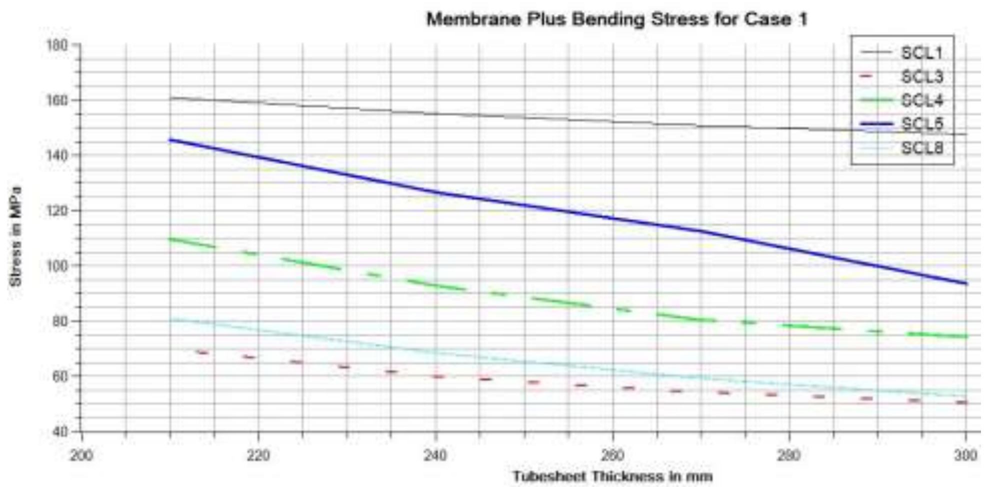

Figure 7: Maximum membrane plus bending stress intensity at SCL 1, 3, 4, 5 and 8 for loading Case1

In case 1, for all four models Membrane stresses and Membrane plus bending stresses were within the allowable limit as per ASME codes. Membrane stresses at SCL's 1, 2, 6, 7 and 9 were constant for loading case 1, as these locations were not on tubesheet and away from the discontinuity. For SCL's 3, 4, 5, and 8 it increases with decrease in tubesheet thickness as shown in Figure 5. Membrane plus bending stresses at SCL's 6, 7, and 9 were constant. For SCL's 1, 3, 4, 5, and 8 membrane plus bending stresses were increased with decrease in tubesheet thickness which is shown in Figure 6.

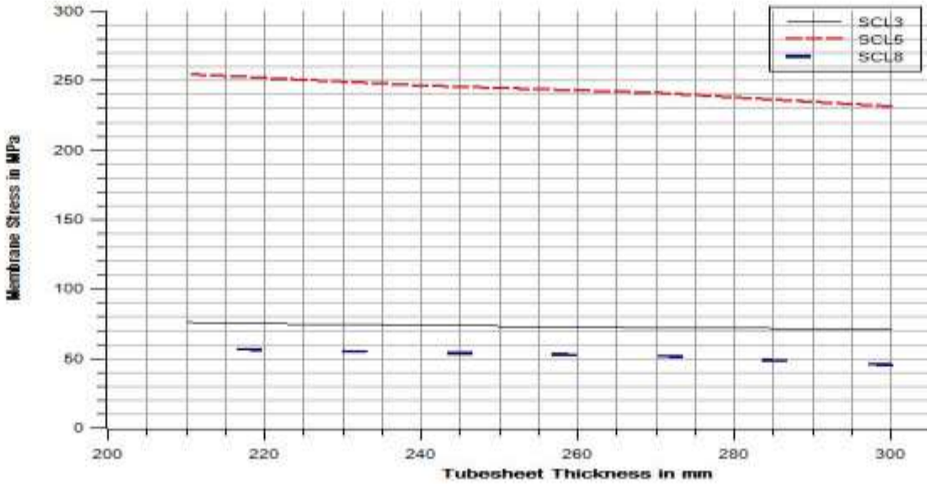

Figure 8: Maximum membrane stress intensity at SCL3, 5and 8 for loading Case 2

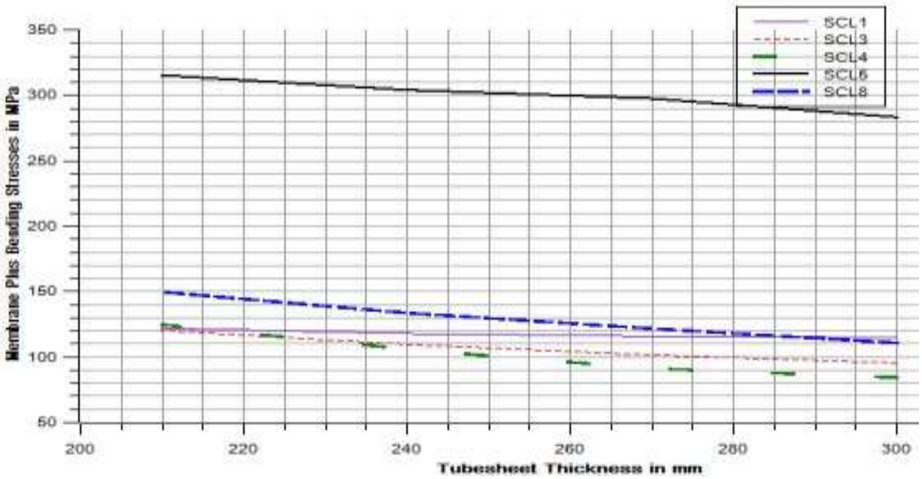

Figure 9: Maximum membrane plus bending Stress intensity at SCL 1, 3, 4, 5, and 8 for case 2

In case 2, for all four models Membrane stresses and Membrane plus bending stresses were within the allowable stress limit as per ASME codes. Membrane Stresses increases at SCL 3, 5 and 8 but within allowable limit. At SCL1, 3, 4, 5 and 8 Membrane plus bending stress increases as tubesheet thickness decreases. For the optimum model membrane stresses are within allowable limit. From the above graphs it is cleared that, design of tubesheet using ASME code gives excess tubesheet thickness. FEA analysis method was used for optimization of tubesheet.

CONCLUSION

Tubesheet design is more critical by using Codes and Standards. In conventional method perforated tubesheet is converted in to equivalent solid plate for design purpose, which causing to increase tubesheet thickness. Tubesheet designed by ASME codes was observed with more thickness. Optimization of the original tubesheet using FEA analysis was performed in ANSYS. Tubesheet thickness based on optimization method could be reduced by 20-25%. Therefore while designing tubesheet of heat exchanger using conventional codes along with finite element analysis should be used for optimum design.

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