FINITE ELEMENT ANALYSIS OF HEAT EXCHANGER WITH NTIW TUBESHEET

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Abstract - In order to reduce the pressure drop and mitigate vibration, No Tubes in Window (NTIW) bundle configuration is used in heat exchanger. However, this configuration has significant effect on the mechanical design of the tubesheet. Method proposed in ASME Section VIII Div.1 is widely used for designing the tubesheet with different configuration. ASME method mainly consists of analytical formulas based on consideration of equivalent properties for perforated zone and applicability of this method is limited to the plate which is uniformly perforated up to Outer tube limit (OTL) of the tubesheet. Therefore this type of configuration (NTIW) is generally design by stress analysis using finite element analysis (FEA) method.

Modeling complete heat exchanger (3D model) creates huge size FEA model and hence results in more computation time. This paper mainly discusses the advantages and limitations of axisymmetric model with respect to 3D model for designing the NTIW tubesheet of heat exchanger.

Keywords: Tubesheet, No Tubes in Window (NTIW), ASME, Heat Exchanger, Finite Element Analysis.

I. INTRODUCTION

Tubesheets in heat exchanger are designed by methods suggested by design codes, [1, 2] which are based on a uniform circular perforated area and a uniform thickness of the tubesheet. In order to reduce the pressure loss and to mitigate vibrations some portion on tubesheet is kept untubed (NTIW configurations). Design of NTIW configurations are not covered in design codes.

NTIW configuration has the tubesheet layout as shown in Figure 1. In Shell and tube heat exchangers designing NTIW configuration tubesheet requires modeling of tube, tubesheet, shell**.** Modeling entire model as 3D with all mentioned components above increases the difficulty level of analysis, also such large FEA model requires high computational capacity machines to reduce the computation time. As the perforation pattern of NTIW tubesheet is symmetric about the two axes in plane of plate and at two sides of the plate un-tubed area is present.

Therefore it is difficult cover given configuration under the axisymmetric approach. An attempt is made to validate the axisymmetric approach results with actual 3D approach by considering the stiffening effect of the un-tubed area by modifying material model. The main purpose of this paper is to explore the approximate relation between deflection and stress values obtained from axisymmetric and 3-D approach.

ASME- BPVC section VIII division 2 gives guidelines for design by analysis method. These non-habitual tubesheets are designed using design by analysis method. Design by analysis adopted FEA is an alternative method to establish tubesheet thickness.

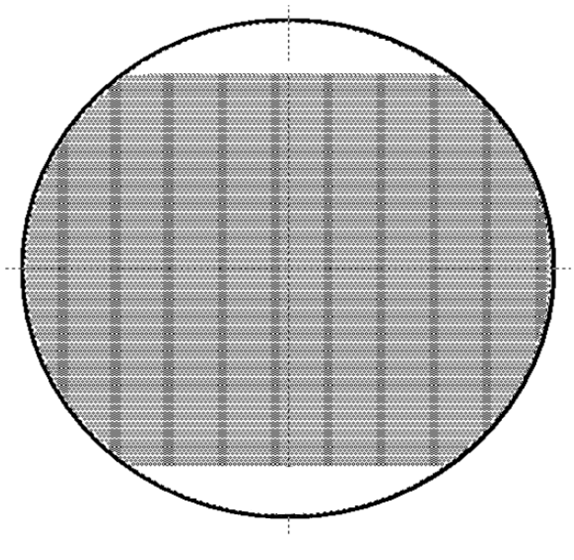

Figure 1: Perforation Layout on Tubesheet

To design a heat exchanger several codes are available. One of very simplest is TEMA code (Tubular Exchangers Manufacturers Association). This method is applicable for different constructive configurations and calculates minimum thickness for bending and shear. Depending on constructive configuration, differential thermal expansion effect between shell and tubes is also considered [2]. This code gives design guidelines for the uniform circular perforated area without large untubed areas, so TEMA code is not applicable for NTIW configuration.

ASME-BPVC VIII Div. 1 in its UHX subsection gives design rules for tubesheets of heat exchanger. The method proposed is based on the consideration of an equivalent plate with modified elastic properties depending on effective ligament efficiency. Stress calculation is done by analysis based on properties of equivalent plate. In this method differential thermal expansion effect between shell and tubes and the differential radial thermal expansion between the integral shell and the tubesheet are taken into consideration. In all constructive configurations nominal uniform circular perforated area is assumed [1]. So NTIW configuration is outside of scope of UHX rules.

Most general analysis procedures found in literature are those described in codes ASME VIII Div.2 Ed.2015 (Part 5, Annex 5.E). Procedures approach is very similar based on equivalent solid plate consideration. This equivalent solid plate region is characterized by an elastic orthotropic material behavior law whose elastic constants depend on effective ligament efficiency. Even though these treatments have several similarities with ASME III Div.1 (especially on the consideration of all secondary effects), it represents an important upgrade compared to the others as it guides in the modeling and characterization of the materials involved. It also provides additional acceptance criteria depending on the kind of damage analyzed. These methods are highly orientated to DBA and are especially suitable for its use with FE analysis based tools. Therefore, as these methods exclude from their hypotheses the requirement of a circular uniform pattern (hypothesis linked to the axisymmetric approach for stress calculation) can be applied satisfactorily to NTIW designs [3]. Details of fixed tube shell and tube heat exchanger are described in Table1

Component	Specifications		Material
Shell	I. D. Thickness	3962 mm 44 mm	SA-516-70
Tubes	O. D. Thickness	25.4 mm 2.2 mm	Copper/Nickel 90/10
Tubesheet	OTL Thickness	3945 mm 225 mm	SA-216

Table 1: Heat exchanger data

II. FINITE ELEMENT MODEL

2.1 FEA OF 2-D AXISYMMETRIC MODEL

A two-dimensional axisymmetric model was developed including the components such as tubesheet as equivalent solid plate, tubes as equivalent cylinders, shell and channel shell. In axisymmetric modeling the perforated plate is replaced by equivalent solid plate with same thickness. The tubes are not axisymmetric so tubes are replaced by equivalent cylinders. Figure 2 shows the model for axisymmetric analysis. Dimensions in corroded and without weld overlay condition are considered [4]. Commercial FEM software ANSYS version 16.1 was used for model development and stress calculation.

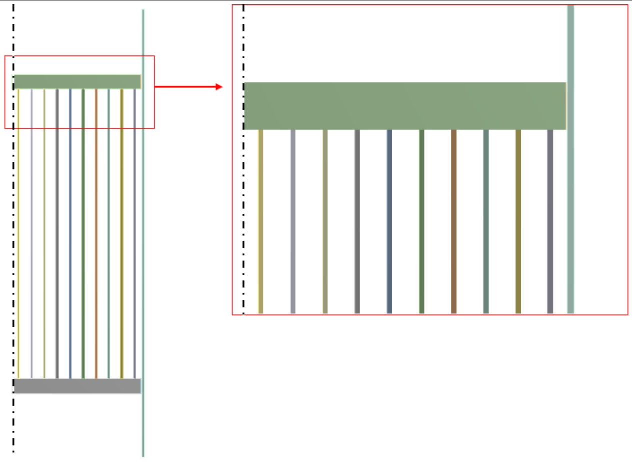

Figure 2: 2-D Axisymmetric Model

Material and model

Stresses were calculated employing elastic stress analysis without geometrical nonlinearities. Tubesheet is converted in to solid plate with effective elastic constants. The equivalent material properties for perforated plate can be determined based on the effective ligament efficiency $\mu*$ as well as ratio of tubesheet thickness to tube pitch *h/p*. The effective ligament efficiency is defined by the following equation.

$$\mu^* = \frac{p^* - d^*}{p^*}$$

When the ratio of tubesheet thickness to tube pitch *h/p* is larger than 2.0, the equivalent material constants of the perforated plate can be defined as only a function of the effective ligament efficiency $\mu*$. The material properties for perforated region of the tubesheet are calculated by means of the equation described in ASME Section VIII Division 2 - Annex 5.E. These material properties are applied to equivalent solid plate [1].

Tubes were modeled as equally spaced equivalent cylindrical shells. The number of the equivalent shells was determined to be 10 with which stiffening effect resulting from the staying action of the tubes to tubesheet can be adequately represented.

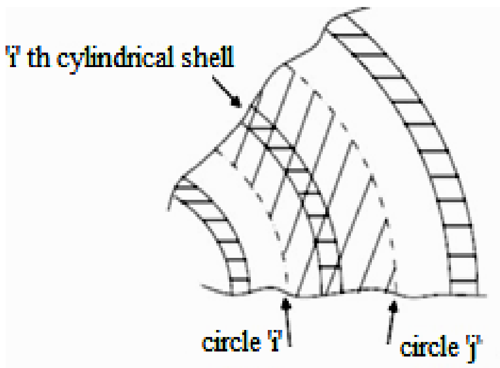

Figure 3: Equivalent Shell for Tubes.

An orthotropic material was used for elastic modulus of the equivalent shell elements. The modulus in axial direction was determined so that the total spring constant of the shell elements is equivalent to the tube bundle. The "i"th Cylindrical shell was equivalent to tubes located in the area defined by circles "i" and "j" in terms of the elongation and flexural rigidity when subjected to the tension and bending moment, respectively. In order to describe mechanical characteristic of beam elements using shell elements, the modulus in circumferential direction assumed to be negligibly small. Poisson's ratio was considered to be zero.

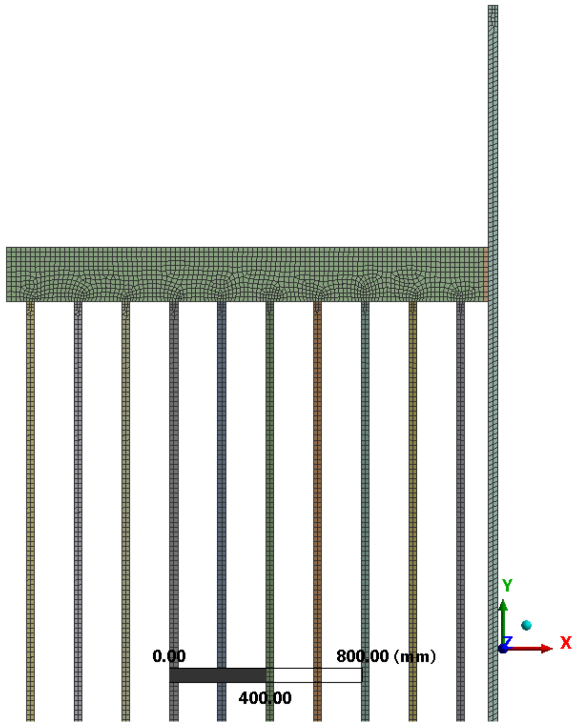

Figure 4: Discretized Axisymmetric model

Boundary conditions and loading

Figure 4 shows the discretized axisymmetric model of the heat exchanger. As the loading condition, only shell-side pressure (*Ps*) was considered for this FEA investigation. The effect of thermal loading was not considered. For the purpose of calculating tubesheet stresses due to pressure acting directly and indirectly on the tubesheet separately, the shell-side pressure was divided into two cases; equivalent pressure applied directly on the tubesheet and shell side design pressure on shell and head. An equivalent pressure which can be calculated by the equation below was considered on the perforated region of the tubesheet.

Equivalent pressure on tubesheet (channel side):

$$P_{eq.t} = P_t\left[1 - N_t\frac{d_t^2}{OTL^2}\right]$$

Equivalent Pressure on tubesheet (shellside):

$$P_{eq.s} = P_s\left[1 - \frac{\frac{\pi}{4}\,d_t^2.N_t}{\frac{\pi}{4}OTL^2 - A_1}\right]$$

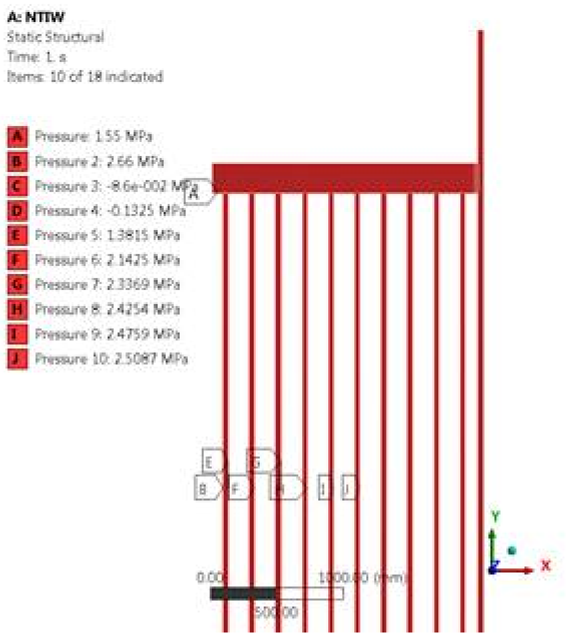

Figure 5: Boundary conditions and loading.

Actual design pressure is to be applied on shell (P_s) and channels (P_t).

In order to take effect of axial loadings due to tube expansion by pressure were applied in the form of effective temperature to the equivalent cylinders.

$$\text{Strain in tubes} = 2v_t\,\frac{d_i^2.\;P_t - d_o^2.P_s}{E_t(d_i^2 - d_o^2)} = \alpha\,.\Delta t$$

From above formula the value of Δt is to be found out and applied to equivalent cylinders. As these equivalent pressure and temperature correction for tube expansion are shown in the Figure 5.

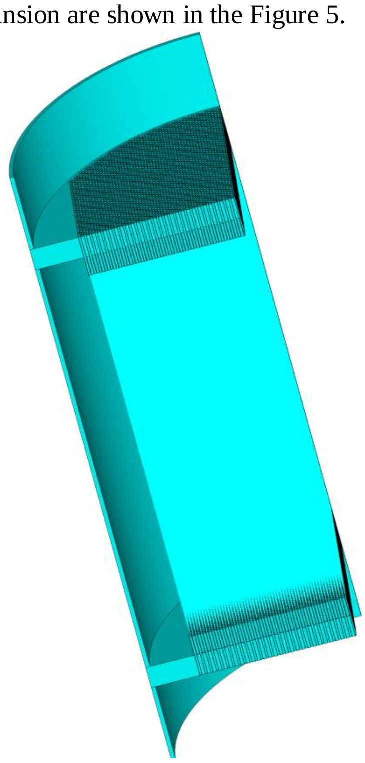

Figure 6: Quarter model of heat exchanger.

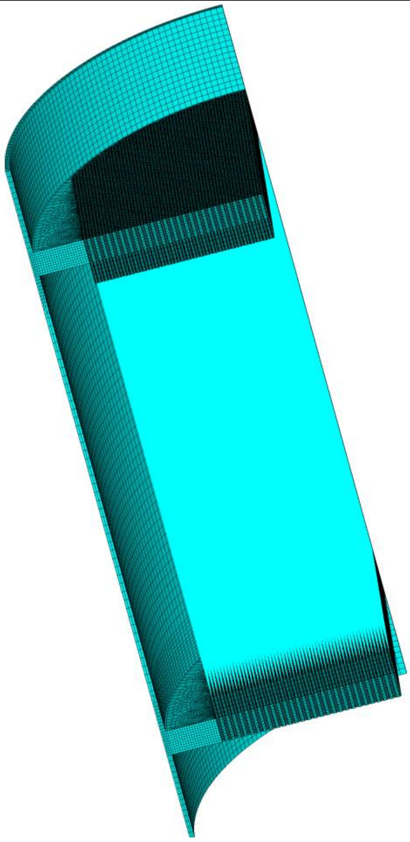

Figure 7: Discretized model of heat exchanger.

2.2 FEA OF 3-D MODEL

The tubesheet is symmetric about the two axes so quarter part was modeled. Quarter of the model including the tubesheet, heat exchanger tube, part of the shell was established shown in Figure 6.

In this model perforated plate, shell, and channel are modeled as present and tubes are modeled as beam element with equivalent axial stiffness of the tube. FE model is built using solid elements which represents the actual geometry of the heat exchanger.

The boundary condition is as follows: the axial displacement constraint was applied on the end section of the channel, the equivalent axial force was applied on the cross section of the shell. Symmetrical restriction was imposed on plane of symmetry. Shell side and tube side pressure are applied.

III. RESULTS AND DISCUSSION

3D model and 2D axisymmetric model were analyzed to study the stresses and deflections in tubesheet and shell under loading.

3.1 STRESSES IN HEAT EXCHANGER

As the method contained in ASME VIII Div.2 Ed.2015 is based on elastic analysis, a stress breakdown has to be done for stress check, having therefore membrane, bending and peak stresses and considering them as primary or secondary stresses. At various locations stress classification lines are drawn to find membrane and bending stresses. Evaluated stresses are compared with limits given by protection against plastic collapse in ASME VIII Div. 2.

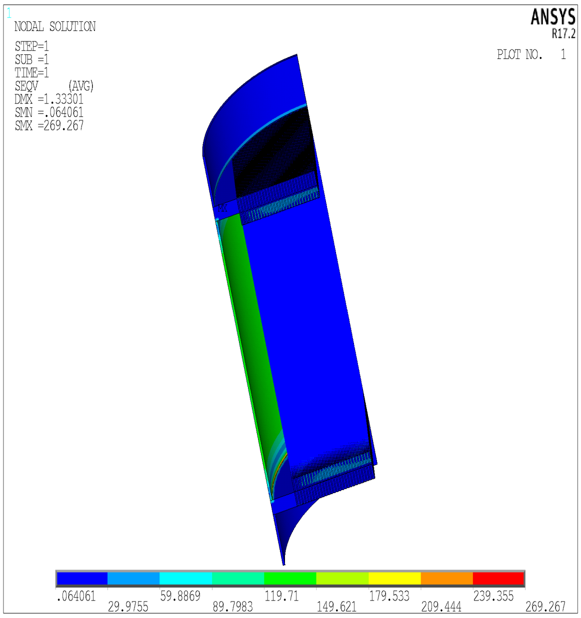

Figure 8: Stress plot for 3-D model

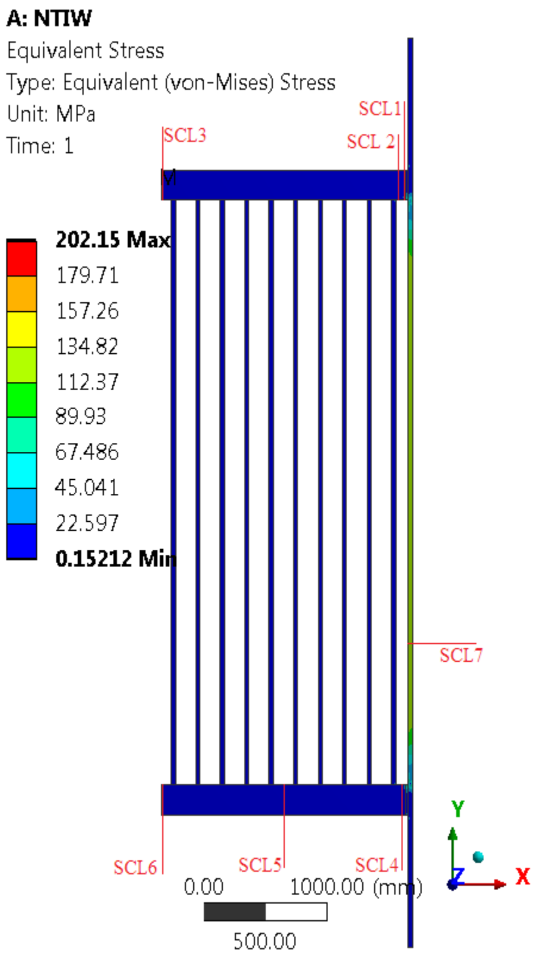

Figure 9: Stress classification lines.

SCL No.	Stress category	Stresses (MPa)	
		2-D model	3-D model
1	P_L+P_b	79.1533	76.542
2	P_L+P_b	103.625	83.2514
3	P_L+P_b	20.468	16.84
4	P_L+P_b	100.54	75.8265
5	P_L+P_b	11.77	10.65
6	P_L+P_b	14.8818	11.942
7	P_m	129.33	129.8

Table 2: Stresses at different locations

In 3- D model of heat exchanger the stresses in the tubesheet are maximum in ligament and shell to tubesheet junction. This is because of spring effect of tubes. Tubesheet stresses in axisymmetric model are more by 20-25% as compared to stresses in 3-D model.

3.2 DEFLECTIONS IN HEAT EXCHANGER

Components	Axisymmetric	3-D model
Top Tubesheet	0.17785	0.180264
Bottom Tubesheet	0.47406	0.473618
Shell	1.3416	1.33301

Table 3: Deflections in different parts (mm)

For simply supported or fixed at the ends the deflection in the plate is maximum at center. But in heat exchanger tube bundle act as elastic spring support, deflection at center may not be maximum. The deflection in major parts of the heat exchanger is given in table 2.

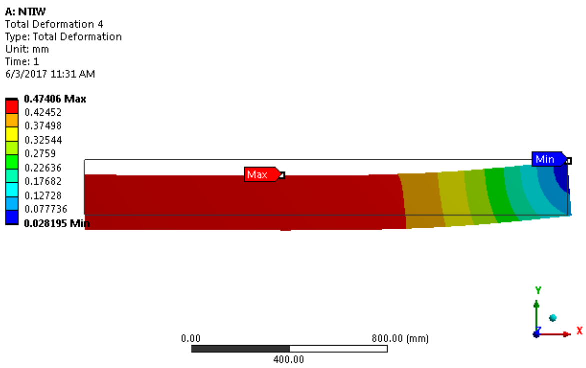

Figure 10: Deflection in Bottom Tubesheet in axisymmetric model.

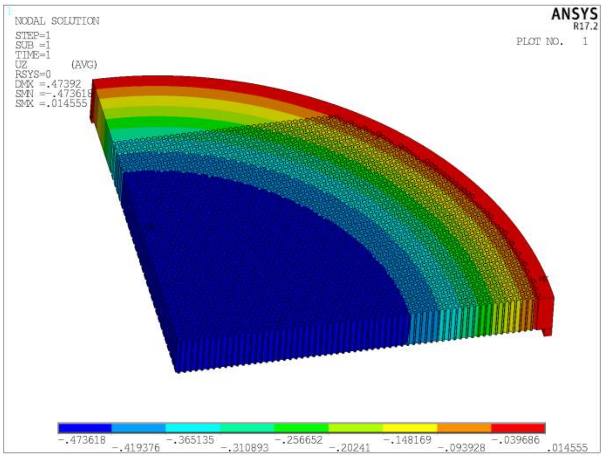

Figure 11: Deflection in Bottom Tubesheet in 3D model.

The error in the deflection values of both models is less than 1%.

CONCLUSIONS

Design of the NTIW tubesheet for the fixed tubesheet type heat exchanger was discussed in this paper.
Stresses in tubesheet due to pressure loading are higher in axisymmetric model. Stresses are high because of bending stiffness of the plate is higher due to presence of untubed area and also the untubed area is not axisymmetric. Untubed area present on tubesheet is about 9.8% and variation in tubesheet stresses is found to be 20-25%. Deflection in the tubesheet and shell are very close value.
The present FEA study confirms that employing an axisymmetric method stress analysis, faster and economical design of the tubesheet can be realized.

NOMENCLATURE

d_t = tube O.D [mm]
d_o = tube I.D [mm]
N_t = the number of tubes
p = tube pitch [mm]
$p*$ = effective tube pitch [mm]
D_o = outer tube limit (OTL) / diameter of perforated region of tubesheet [mm]
d* = Effective Diameter of tube
A_L = Area of Un-tubed lane
μ = ligament efficiency
$\mu*$ = Effective ligament efficiency
h = tubesheet thickness [mm]
= elastic modulus of tubesheet at design temperature [MPa]
= effective elastic modulus for perforated region of tubesheet [MPa]
= effective Poisson's ratio for perforated region of tubesheet
= elastic modulus for tube at design temperature [MPa]
ν_t = Poisson's ratio for tube
α_t = coefficient of thermal expansion for tube material
P_t = Pressure inside the Tubes (Tube side)

P_s = Pressure outside the Tubes (Shell Side)
P_L = Local Membrane stress
P_m = Primary Membrane stress
P_b = Primary Bending stress

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