IOP Conference Series: Materials Science and Engineering

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To cite this article: Q Guo *et al* 2019 *IOP Conf. Ser.: Mater. Sci. Eng.* **504** 012060

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IOP Conf. Series: Materials Science and Engineering **504** (2019) 012060 doi:10.1088/1757-899X/504/1/012060

Numerical simulation of shell-and-tube heat exchanger and study of tube bundle support

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Abstract. In this paper, the method for numerical simulation of fixed tubesheet heat exchangers was analyzed and the support of the tube bundle was studied. It is found that for low or medium pressurized shell-and-tube heat exchangers, no matter which type of solid elements, solid-shell elements or shell elements is utilized for meshing the cylinder and tubes, the axial stresses at the cylinder and tubes as well as the tubesheet deflection are almost the same. The number of tubes has no effects on the hoop stress at the shell, but has significant effects on the axial stress. The less the number of the tubes, the larger the axial stress at the cylinder is. The tubesheet deflection and the action of the pressure on the outside surface of the tubes and inside surface of the cylinder affect the axial stress at the cylinder remarkably. Accurate stress calculation for the combined tubesheet, cylinder and tubes system should be performed using finite element method. The error would be large if the axial stress were simply calculated with the axial force divided by the axial cross section area bearing the load.

1. Introduction

Fixed shell-and-tube heat exchangers are widely used in petro and petrochemical industries. The axial stresses at the cylinder and heat exchange tubes are usually the controlling factors for the failure of the heat exchanger under the action of pressures and temperatures. Therefore, it is a must to calculate and assess the axial stresses at both the cylinder and tubes and deflection of the tubesheet when design a fixed-head shell-and-tube heat exchanger.

For the numerical simulation of the heat exchangers, Liu[1], Zhang[2] and Chen[3] built finite element models using solid elements, shell element and beam elements, stress and deformation at the tubesheet and axial stress at the tubes were analyzed and compared between different models to find the more reasonable method to simulate heat exchangers. Yu et al.[4] and Qian et al. [5] performed finite element analysis and experimental investigation on the shell-and-tube heat exchanger and found that the support of heat exchange tubes are much larger than that considered in the tubesheet design. Shan[6] found that the principal stress at the top point of the cylinder of the heat exchanger is smaller than that without tubes, implying that the cylinder is enforced by the tubes. Sang[7] proved the axial stress at the cylinder could be induced as a result of deformation compatibility in the tubesheet system composed of the cylinder, tubesheet and tubes, especially when the stiffness of the tube bundle is close to that of the cylinder.

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In this paper, different finite element models were built with different elements. By comparing the axial stress at the cylinder and tubes and tubesheet deflection, the reasonability of the models for the simulation of heat exchangers are discussed. In addition, the support effects of the tube bundle on the tubesheet were also investigated in this paper.

2. Establishment of the finite element models of a heat exchanger

2.1. Geometric and grid models

Fig. 1 illustrates the structure of a fixed shell-and-tube heat exchanger. Some geometrical parameters of the heat exchanger are listed in Table 1.

Table 1. Some structural parameters of the heat exchanger.

Parameters (mm)	Values
Inner diameter of the shell	159
Thickness of the shell	4
Length of the shell	400
Outer diameter of the tubes	14
Thickness of the tubes	1.5
Length of the tubes on the shell-side	400
Number of the tubes	29
Distance between adjacent tubes	19
Diameter of the tubesheet	285
Thickness of the tubesheet	26

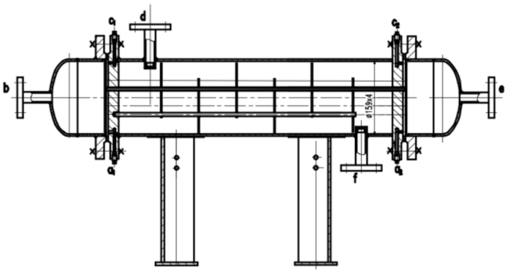

Figure 1. Illustration of the structure of the heat exchanger.

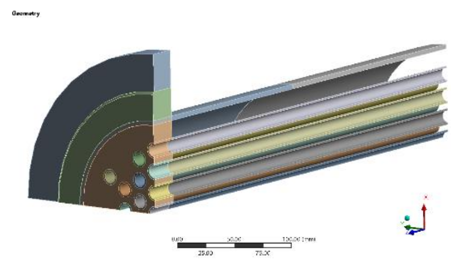

Figure 2. Geometrical model of the heat exchanger.

The large commercial software ANSYS17.0 is used to build the finite element models of the heat exchanger. In order to improve computation efficiency without changing the nature of the concerned problems, 1/4 structure of the heat exchanger is modeled as the structure is symmetric as a whole. Fig. 2 shows a geometrical model of the heat exchanger which includes tubesheet and part cylinder and tubes.

Heat exchanger is a special structure in which the cylinder and tubes are thin in thickness while the tubesheet is thick. For building grid models, it is appropriate to mesh the tubesheet with solid elements. But for cylinder and tubes, except the solid elements, the solid-shell elements and shell elements can also be used if fatigue analysis is necessary. Of course, MPC connections between solid elements and solid-shell elements or shell elements are needed to ensure the deformation compatibility.

As the first part of the study, four grid models are established using different type of elements. These models are: Model 1 in which all the tubesheet, cylinder and tubes are meshed with solid elements (SOLID185) and at least three layers are employed along the thickness, Model 2 in which all the tubesheet, cylinder and tubes are meshed with solid elements (SOLID185) but only one layer is

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employed along the thickness, Model 3 in which the tubesheet is meshed with solid elements and the cylinder and tubes are meshed with solid-shell elements (SOLSH190) and Model 4 in which the tubesheet is meshed with solid elements and the cylinder and tubes are meshed with shell elements (SHELL181). Fig. 3 shows the Model 1 grid of the heat exchanger.

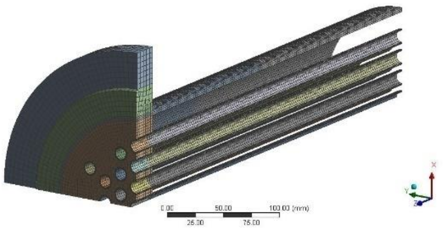

Figure 3. Model 1 grid of the of the heat exchanger.

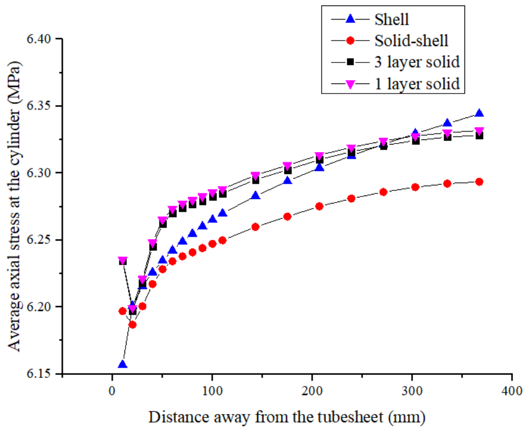

Figure 4. Average axial stress at the cylinder along its length.

2.2. Loadings and boundary conditions

The loadings and boundary conditions are set as follows.

- The shell-side pressure of 0.937MPa is applied on the shell-side tubesheet surface, internal surface of the cylinder and outside surface of the tubes.
- The axial displacements at the cut end of the cylinder and tubes are fixed.
- Symmetric constraints are applied on the symmetric surfaces of the models.

2.3. Analysis results and comparison of stresses

For shell-and-tube heat exchangers under low or medium pressures, the axial stresses at the cylinder and tubes and the tubesheet deflection are the main parameters to be considered for the strength design. The following will be focused on the analysis results of these parameters with comparison between them.

For the models built with solid or solid-shell elements, membrane stress and membrane plus bending stress at the cylinder can be obtained from the stress linearization along the cylinder thickness. For the models built with shell elements, however, the membrane stress and membrane plus bending stress at the cylinder can be directly obtained from calculation results.

Table 2 lists the axial stresses at the outer, middle and inner surfaces of the cylinder on the middle section (far away from the tubesheet).

Table 2. Axial stress on the middle section of the cylinder.

Element type	Outer surface (MPa)	Middle Surface (MPa)	Inner surface (MPa)
3 layers solid	6.224	6.285	6.346
1 layer solid	6.08	6.29	6.499
Solid-shell	6.041	6.25	6.458
Shell	6.262	6.27	6.279

It is seen from Table 2 that whatever the cylinder and tubes are meshed by solid, solid-shell and shell elements, the axial stresses are almost the same with relative differences less than 4% at the inner surface and outer surface and even less at the middle surface.

Fig. 4 shows the average axial stress (i.e. the membrane stress at the middle surface of the cylinder) along the cylinder length beginning at the tubesheet. It is seen that for the axial stresses, no clear differences are found whether the cylinder thickness is meshed with one layer or three layers solid elements. But because of the structural discontinuity at the connection point of the cylinder with the tubesheet, the stresses here are in some extent different.

Fig. 5 shows the average axial stresses at the tubes along their length beginning at the tubesheet. It is seen that the differences between stresses obtained with different models are not significant with relative difference between the largest and smallest being less than 8%.

Similarly, the axial stresses obtained from the models meshed by solid elements are almost the same whether the cylinder thickness is divided with one layer or three layers.

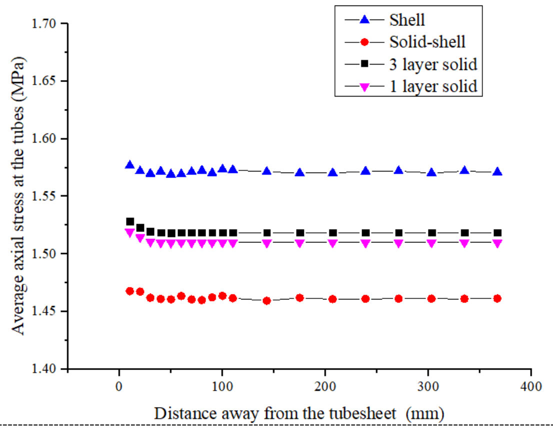

Figure 5. Average axial stress at the tubes along its length.

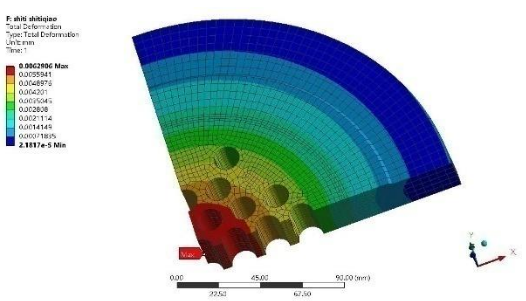

Figure 6. Tubesheet deflection distribution.

Fig. 6 shows the tubesheet deflection distribution. Table 3 lists the deflection at the tubesheet center obtained from different simulation models. Clearly it is seen that differences between them are very small.

Table 3. Tubesheet deflections for different type of elements.

Type of elements	Tubesheet deflection (mm)
Solid	0.0062347
Solid-shell	0.0062525
Shell	0.00596654

From above analysis, it is reached that for low and medium pressure shell-and-tube heat exchangers, solid-shell elements can be used for meshing cylinder and tubes if fatigue analysis is needed and shell elements can be used if fatigue analysis is not needed. This result is very meaningful in engineering because for large heat exchangers, the difficulties to build the finite element model and the compute time can be greatly reduced.

3. Support analysis of the tube bundle

In shell-and-tube heat exchangers, tubesheet is supported by tubes. In this section, the axial stress is taken into as an index to analyze these support effects by different number of tubes. Model 1 described in previous section is used and the loadings and boundary conditions are the same as in subsection 1.2.

In order to investigate the support effects of tubes on the tubesheet, several finite element models of the heat exchanger with different number of tubes are established. Table 4 lists the axial and hoop stresses at the middle section of the cylinder for different number of tubes. It is seen that the number of the heat exchange tubes has no effects on the hoop stress at the cylinder, but has great effects on the axial stress. For the heat exchanger studied here, the less the number of tubes, the larger the axial stress at the cylinder is. This result is understandable because more tubes give larger cross area to bear loadings.

Table 4. Axial and hoop stresses at the cylinder for different number of tubes.

Number of tubes	Axial stress σ_z/MPa	Hoop stress σ_θ/MPa
29	6.42	19.24
25	6.59	18.99
19	6.83	19.10
0	9.08	19.10

Fig. 7 shows the axial stress distribution at the heat exchanger. It is seen that the axial stress at the cylinder is larger than that at the tubes. It is also found the because of the tubesheet deflection, stresses at the tubes in different radical places are not the same.

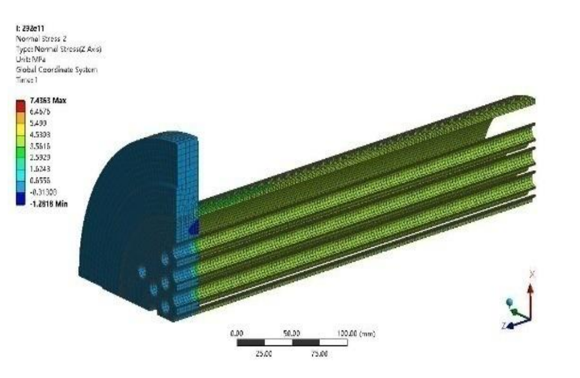

Figure 7. Axial stress distribution at the heat exchanger.

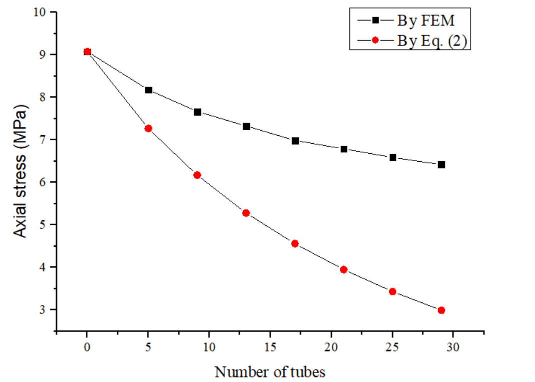

Figure 8. Variation of the axial stress at the cylinder with the number of tubes.

For simplicity, for the heat exchangers constructed with the cylinder and tubes having similar material properties, the hoop elastic stress and axial elastic stress may be calculated by the following formulas:

$$\sigma_\theta = \frac{PD_{\mathrm{i}}}{2t} \tag{1}$$

$$\sigma_{\mathrm{z}} = \frac{PS}{S_{\mathrm{s}} + S_{\mathrm{t}}} \tag{2}$$

where,

σ_{z} —Axial stress at the cylinder, MPa;

σ_θ —Hoop stress at the cylinder, MPa;

P —Shell-side pressure, MPa;

D_i—Inner diameter of the cylinder, mm;

t—Thickness of the cylinder, mm;

S —Shell-side pressure acting area on the tubesheet, m^2;

S_s—Cross area of the cylinder, m^2;

S_t—Cross area of the tubes, m^2.

It turns out that the hoop stress at the cylinder calculated by Eq. (1) is very close to that obtained by finite element method (FEM). If there are no tubes, the axial stresses obtained by Eq. (2) and finite element method are the same. But as liasted in Table 5 and shown in Fig. 8, with the tube support, the axial stress obtained by finite element method is larger than that calculated by Eq. (2). The larger the number of tubes, the bigger the differences are.

Table 5. Variation of the axial stress at the cylinder with the number of tubes.

Number of tubes	Axial stress σ_z /MPa	
	By FEM	By Eq. (2)
29	6.42	2.99
25	6.59	3.43
21	6.79	3.95
17	6.99	4.56
13	7.33	5.28
9	7.67	6.17
5	8.18	7.27
0	9.08	9.08

Further study finds that tubesheet deflection as well as the action of the shell-side pressure on the inner surface of the shell and outer surface of the tubes is the cause for the axial stress differences. In fact, the tubesheet deflection makes the tubes deform unevenly from the edge tube to the center tube. The pressure acting on the outside surfaces of the tubes would enforce the tubes extend because of the Poisson's effects, which reduces the support force or even applies additional push force on the tubesheet and, thus, increases the axial force acting on the cylinder. Similarly, the radical deformation of the cylinder also changes its axial force. It turns out that if the stiffness of the tubesheet is strong enough and tubesheet deflection is very small, the axial stresses at all tubes are uniform. In addition, if the shell-side pressure is only acted on the tubesheet surface, the axial stress at the cylinder calculated by FEM is almost the same as by Eq. (2). So for the combined tubesheet, cylinder and tubes system, it is very complicated to calculate the stresses analytically. An effective and accurate method for the stress evaluation is FEM. If simply calculating the axial stress by dividing the axial force with whole cross metal area, the error would be large and for the cases studied here, it is not conservative.

4. Conclusions

(1) For low and medium pressure shell-and-tube heat exchangers, whatever the cylinder and tubes are meshed by solid, solid-shell and shell elements, the axial stresses at the cylinder and tube and the tubesheet deflections obtain by different models are almost the same with relative error between stresses less than 4% and deflections less than 3%. So for large shell-and-tube heat exchanger, solid-shell elements are suggested to be used for meshing cylinder and tubes if fatigue analysis is needed and shell elements are suggested if fatigue analysis is not needed. Thus, the difficulties to build the finite element models and the compute time can be greatly reduced.

(2) The number of the heat exchange tubes has no effects on the hoop stress at the cylinder, but has great effects on the axial stress. For the heat exchanger studied here, the less the number of tubes, the larger the axial stress at the cylinder is.

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(3) The tubesheet deflection and shell-side pressure applied on the internal surface of the cylinder and outside surfaces of the tubes affects the axial stress at the cylinder. Finite element method is effective to accurately calculate this axial stress. If simply calculating the axial stress by dividing the axial force with cross metal area, the error would be large and for the case studied here, it is not conservative.

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