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Analysis of Tube Sheets on Winding Tube Heat Exchangers

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Abstract: Spiral-wound tube heat exchanger, also known as threaded tube-type heat exchanger, screw thread heat exchanger, is a new type of high heat transfer coefficient of heat exchanger, is connected with the tube plate on the central tube to spiral Like a small number of small diameter heat transfer tube to form a tube bundle, and then put the tube into the shell of a heat exchanger. It has the advantages of compact structure, high heat transfer at the same time, high operating pressure in the tube, thermal expansion of the heat transfer tube, high heat transfer efficiency, high degree of intensification, and little demand for heat transfer equipment. Tube sheet is one of the most important parts of the shell and tube heat exchanger, it is subject to the temperature load and mechanical load, the situation is complicated. At home and abroad in the tube board design and analysis made a lot of achievements, put forward a number of practical theory, but there are some shortcomings. In this paper, according to the provisions of JB4732, through the commercial software ANSYS Workbench on the tube plate model for static analysis, respectively, to find the three conditions corresponding to the stress intensity and deformation of the largest parts, is conducive to the structural design of the tube Board safety.

Keywords: Finite element, wound heat exchanger, tube plate, stress analysis.

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1. INTRODUCTION

A device for the transfer of heat (or helium) between two or more fluids, one fluid, one solid, solid particles, or the same fluid in thermal contact and with different temperatures is called a heat exchange device. Heat exchangers are common equipment for chemical, petroleum, power, steel, food, power generation, and many other industrial sectors, and play an important role in production. Especially in chemical production, heat exchangers are related to the normal operation and operating costs of production. Usually in the construction of chemical plants, heat exchangers account for about 10%-20% of the total investment. In oil refineries, heat

exchangers account for about 35%-40% of all investment in process equipment. In the past 20 years, heat exchange equipment has been widely used in energy storage, conversion, recovery, and the use of new energy and pollution control.

In industrial production, the main function of the heat exchange equipment is to transfer heat from the fluid with higher temperature to the fluid with lower temperature, so that the temperature of the fluid reaches the specified index of the process to meet the needs of the process. In addition, the heat-generating equipment is also an effective device for recovering waste heat, waste heat, especially low-grade heat energy, increasing the total utilization of thermal energy, reducing fuel consumption and power consumption, and improving the economic benefits of industrial production.

There are many kinds of heat exchangers, including snake-tube heat exchangers, tube-type heat exchangers, shell-and-tube heat exchangers, corrugated plate heat exchangers, plate-fin heat exchangers, spiral plate heat exchangers, umbrella plate replacement Heater and so on. Different heat exchangers have different suitability and can be used in different situations. So far, there is no universal heat exchanger that can meet all different requirements.

However, the shell-and-tube heat exchanger has become the most widely used heat exchanger because of its simple structure, low cost, wide selection of materials, convenient cleaning, and strong adaptability. In the development of heat exchangers to high temperature, high pressure, and large capacity, the long history of shell-and-tube heat exchangers has added new vitality. Due to the wide variety of applications, there are generally three categories of fixed tube, floating head, and U-tubes. Inserted sleeves, which are often used as evaporators, can also be classified as such. The common feature of the shell-and-tube structure is that it has a circular outer shell, which contains a bundle of tubes consisting of parallel heat transfer tubes. The inner channel portion of the heat exchange tube is collectively referred to as the tube process, and the channel portion between the outer surface of the tube and the inner surface of the shell is collectively referred to as the shell process. Cold and hot fluids flow through the tube and shell sides, respectively, and heat transfer is achieved through the walls of the heat exchange tubes [1].

In recent years, with the advancement of finite element technology, especially the continuous improvement of the level of some commercial software, it has become possible to use the finite element technology to make the engineering design more scientific and rational, and also to provide a scientific and rational design of the heat exchanger heat exchanger tube sheet. Theoretical guidance. ANSYS software is a large-scale universal finite element analysis software that is currently popular in the world, including fusion structure, heat, fluid, electromagnetism and sound. In accordance with the provisions of JB4732, this paper conducts a static analysis of the tube plate model through commercial software ANSYS Workbench to study its stress and deformation.

2. LOAD CONDITION

Load data includes: shell pressure, tube pressure, self-gravity, temperature load. In contrast, its own gravity is small and can be ignored. Among them, the working pressure of the tube process fluid is 0.7MPa, and the working pressure of the shell process fluid is 0.3MPa.

This heat exchanger adopts hydrostatic test. The calculation of the test pressure of hydrostatic test is shown in formula (1).

$$P_T = \eta P \frac{[\sigma]}{[\sigma]^t} \tag{1}$$

in the formula:

P——the design pressure of the pressure vessel or the maximum allowable working pressure specified on the nameplate of the pressure vessel (for working pressure vessels is the working pressure), MPa;

PT——pressure test pressure; when the design considers the hydrostatic pressure, it should be added to the hydrostatic pressure, MPa;

H——pressure coefficient of pressure test; for steel and non-ferrous metals, η =1.25 in hydraulic test;

[σ] —— the allowable stress of the material at the metal temperature during the experiment, MPa;

[σ]t——The allowable stress of the material at design temperature, MPa.

Because the pressure and other data of the experiment have taken into account the temperature factor, the influence of the temperature load can be neglected in the finite element analysis, and only the pressure load is considered finally.

3. FINIT ELEMENT SIMULATION OF WINDING PIPE HEAT EXCHANGER

3.1 Simplification of Finite Element Structure Model

In this analysis, a tube and tube model including a tube sheet, a tube box, a part of a shell, and a part of a heat exchange tube was established in the analysis. When building a model, consider the following simplifications:

(1) Edge effect: Since the model is very large and the structure is relatively complicated, resulting in a large number of calculation data, which brings about considerable difficulties for calculation, it is necessary to simplify the model and reduce the number of units. At the same time, the accuracy of the calculation results must be ensured. Therefore, on the side of the shell side, the heat exchange tubes and shells of limited length are retained. According to the equation of influence of the edge effect, it can be determined that the heat exchange tubes and the shell project out of the tube along the shell side direction. The length of the board. The formula of edge effect is shown in equation (2).

$$\Delta L \geq 2.5\sqrt{RT} \tag{2}$$

In the formula,

△L—— the edge effect affects the length;

R ——outer radius of the shell or outer radius of the heat exchange tube;

T —— shell wall thickness or heat exchange tube wall thickness.

From the above formula can be obtained, the heat elongation of the tube is $\Delta L_1 \geq 2.5\sqrt{6\times 0.7}$ =2.5 × 1.71=4.29mm,The outer extent of the shell is $\Delta L_2 = 2.5\sqrt{182\times 4}$ =133.4mm,For the three-dimensional model established in this project, set the external elongation of the heat exchange tube on the side of the shell side relative to the tube sheet to ΔL_1 =40mm, Set the outer extent of the housing relative to the tubesheet to ΔL_2 =140mm.

(1) Symmetry: In the modeling, taking into account the symmetrical form of the tube plate structure and the load, a half model of the tube plate, tube box, and part of the shell is established.

(2) Integration: heat exchange tubes and tube sheets are connected together and do not move with each other. The model can be considered as the integration of heat exchange tubes and tube sheets during the creation process. Although the two units are connected to each other, there will be no contact between the heat pipe and the tube plate, nor will there be friction.

3.2 The establishment of a finite element model

This paper uses the hexahedral mesh division unit provided by ANSYS. In most CFD programs, using a hexahedral mesh can use fewer elements to solve. For example, in the fluid analysis, the same solution accuracy, the number of hexahedron nodes is less than half of the tetrahedral mesh, so that the calculation efficiency and high accuracy can be guaranteed.

(1) In order to achieve the rule division and model establishment of the tube plate surface grid, the tube plate surface needs to be meshed several times. Because the size of the mesh will affect the number of meshes, it will have a certain impact on the accuracy of the calculation results and the size of the calculation scale. After many times of division, the Hex Dominant method was used to divide the tube plate surface and heat transfer tube surface.

(2) The entire model uses a full hexahedral mesh to divide the mesh. The mesh number is 225888 and the number of nodes is 952020. The result is shown in Figure 3.2.

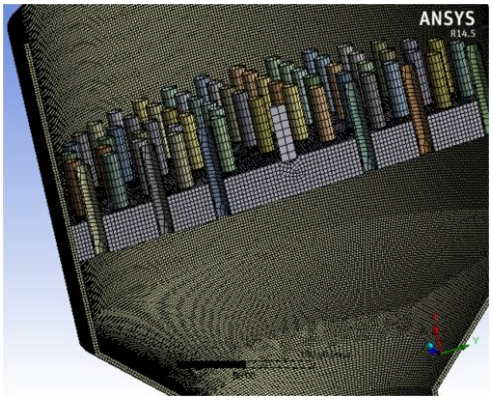

Figure 3.2. Model after meshing

3.3 Application of Displacement and Load Boundary Conditions

The constraints of all load conditions are: Limit the displacement constraint in the Z direction on the XY surface of the model and limit the displacement constraint in the X direction on the YZ surface of the model. The displacement conditions in both directions are zero.

The design pressure of the tube process is 1.0 MPa; the design pressure of the shell process is 0.5 MPa.

4. STRESS ANALYSIS OF TUBE SHEET UNDER DIFFERENT WORKING CONDITION

According to JB4732, the heat exchangers in this analysis are exempt from fatigue analysis. Therefore, there are four criteria for strength verification:

(1) The total stress intensity of a primary film, SI, shall not exceed the design stress strength value, KSm, ie, SI≤KSm, and is generally used instead of Pm;

(2) The allowable limit of a local film stress intensity SII is 1.5 KSm, ie SII≤1.5 KSm, which is generally replaced by PL;

(3) The allowable limit of the stress intensity SIII of a local film stress plus primary bending stress is 1.5 KSm, that is SIII≤1.5 KSm, which is generally replaced by (PL+Pb);

(4) The allowable stress of the stress intensity SIV of a local film stress plus secondary stress is 3Sm, that is: SIV≤3Sm, generally replaced by (PL+Q).

The K in the above formula is the load combination coefficient, which is related to the load and the combination of the containers. For the heat exchanger subjected to pressure and temperature loads, K=1, Sm is the allowable stress intensity of the material.

5. ANALYSIS UNDER VARIOUS LOAD CONDITIONS

5.1 Working Condition 1: Analysis of Tube Pressure Under Single Action (Pt)

Condition 1 is the load condition where the tube pressure (1.0 MPa) acts alone. This load condition is also an operating condition that will be encountered at the initial moment of the first drive open process. Under the effect of pressure in the process of piping, ANSYS analysis shows that the maximum stress intensity occurs at the joint of the pipe box, the maximum value is 218.73 MPa, and the reason for the maximum stress intensity here is that sharp changes in the shape of the pipe box junction occurred. This led to stress concentration; followed by the connection between the tube sheet and the tube box, with a maximum deformation of 0.083928 mm.

5.2 Condition 2: analysis of shell pressure alone (PS)

Condition 2 is the case where the shell-side pressure acts alone. The shell-side pressure is 0.5 MPa. This load condition is also a type of operating condition that will be encountered at the first instant of opening the shell. The analysis of ANSYS shows that the maximum stress intensity occurs at the connection between the tube plate and the tube box. The maximum value is 50.194 MPa. The reason for the maximum stress intensity is that the other parts of the tube

box can expand freely under the effect of pressure. The deformation of the connection between the tube plate and the tube box is limited, so the maximum stress strength appears here, with a maximum deformation of 0.019658 mm.

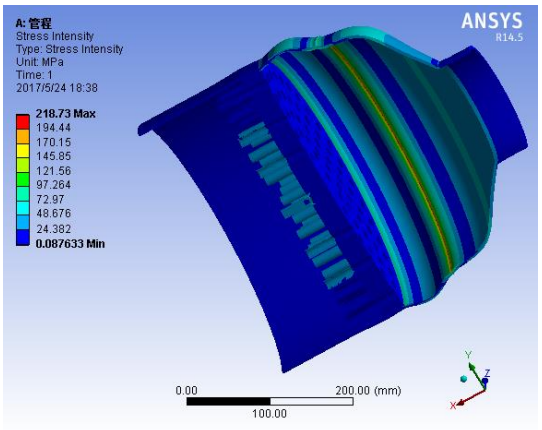

Figure 5.1. Stress cloud diagram under pressure alone

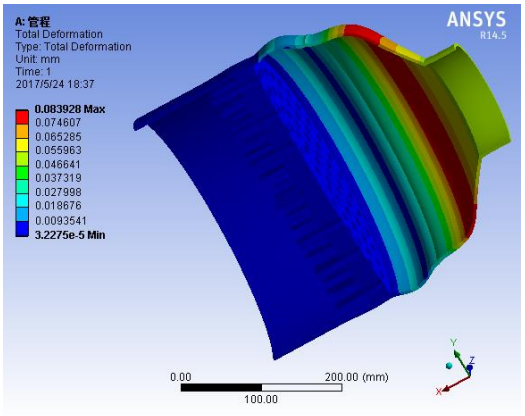

Figure 5.2. Total strain cloud diagram under pressure alone

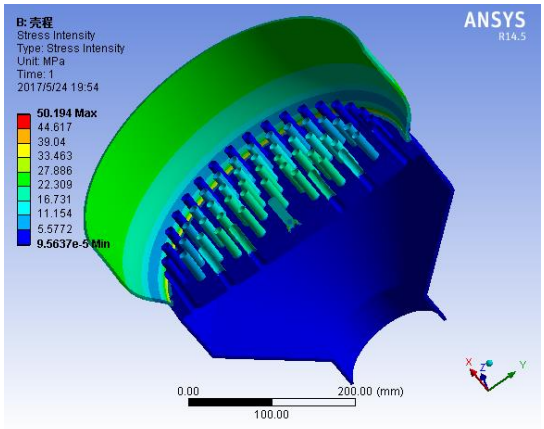

Figure 5.3. Stress cloud diagram under the influence of shell pressure alone

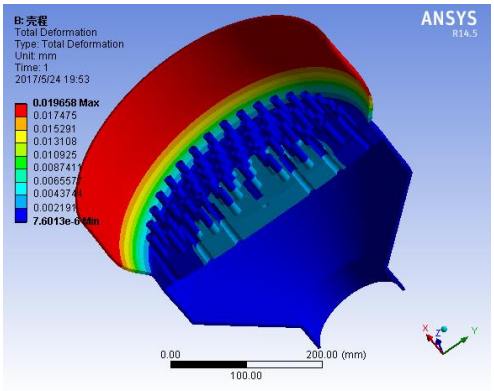

Figure 5.4. Total strain cloud diagram under the action of shell pressure alone

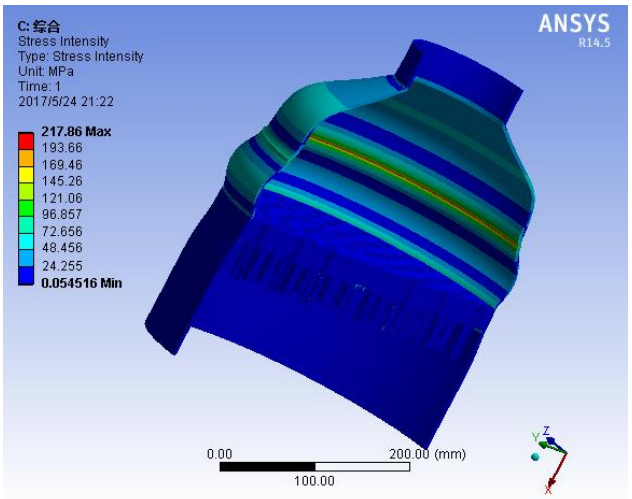

Figure 5.5. Stress cloud diagram under the pressure of tube pressure and shell pressure

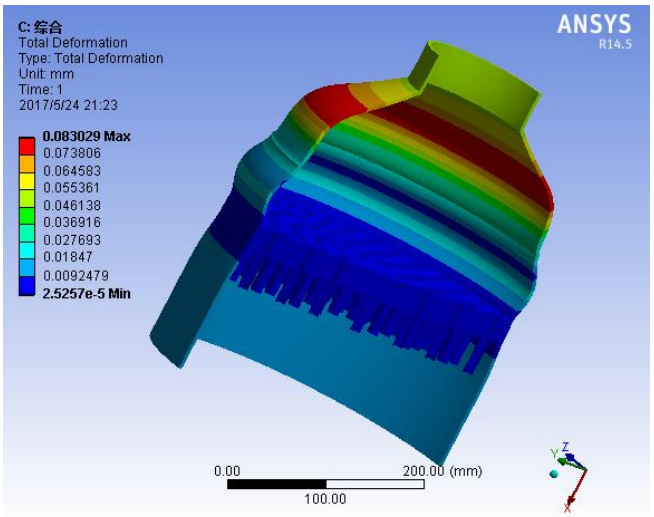

Figure 5.6. Total strain cloud diagram under pressure of shell pressure

5.3 Condition 3: Analysis of Pt+PS under simultaneous action of tube pressure and shell pressure

The load condition where the pressure of the tube process is applied alone is added to the shell load and the operating conditions when the tube load and the shell load act simultaneously are

obtained. This operating condition will occur when the vehicle is started and the tube and shell valve are opened at the same time.

The maximum stress intensity occurs at the splice of the pipe box, with a maximum of 217.86 MPa. The reason for the maximum stress intensity here is that a sharp change in the shape of the splice at the pipe box causes stress concentration; the second is the pipe. The connection between the plate and the tube box, the maximum deformation is 0.083029mm.

6. STRESS CHECK

In order to obtain various stress strengths, it is necessary to stress linearize the results of the finite element analysis. When linearizing the stress, it is necessary to select the stress linearization path according to the following principles: The stress linearization path must pass the stress intensity of the equipment The largest node is set perpendicular to the wall thickness direction.

In order to carry out the strength check in accordance with JB 4732 stress analysis standard in post-processing, according to the analysis to get the stress intensity distribution, 3 paths are selected on the model. Each path is defined by two points on the body. of. As shown in Figure 6.1, Path 1 is the path through the thickness of the tube box splice; Path2 is the path through the thickness of the tube plate where it is connected to the tube box; and Path3 is the path through the thickness of the tube plate where it is connected to the shell. The direction is from inside to outside. The following analysis results are mainly to check the stress intensity of each path taken, and to evaluate the strength status of the structure according to these strength check results. The check results are shown in Table 6.1 to Table 6.3.

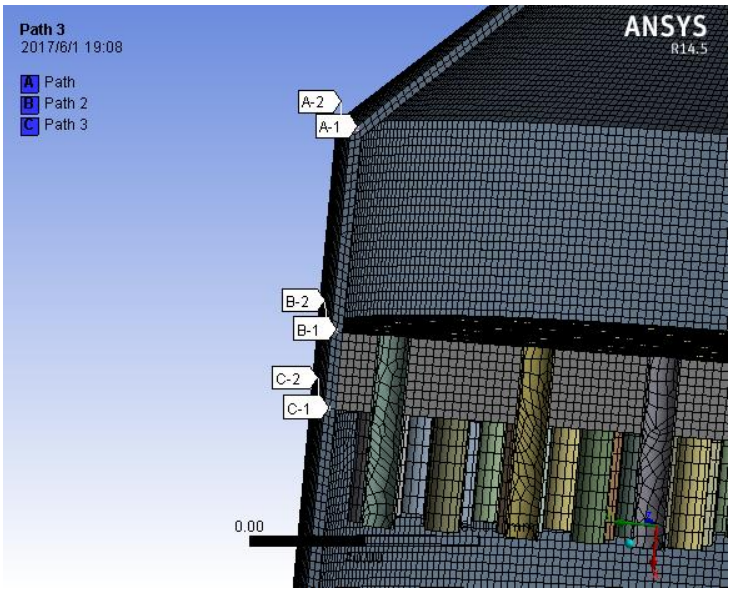

Figure 6.1. Path Settings

Table 6.1. Pipeline pressure strength check

Stress strength and combined stress strength	path	material	Stress intensity calculation(MPa)	Stress strength allowable limit(MPa)	result
A local film stress S_{II}	1	S30408	37.926	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	1	S30408	96.64	3Sm=411	
A local film stress S_{II}	2	S30408	24.736	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	2	S30408	60.848	3Sm=411	
A local film stress S_{II}	3	S30408	10.588	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	3	S30408	9.2011	3Sm=411	

Table 6.2. Checking the shelling pressure strength

Stress strength and combined stress strength	path	material	Stress intensity calculation(MPa)	Stress strength allowable limit(MPa)	result
A local film stress S_{II}	1	S30408	0.0512	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	1	S30408	0.1223	3Sm=411	
A local film stress S_{II}	2	S30408	1.3154	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	2	S30408	2.9317	3Sm=411	
A local film stress S_{II}	3	S30408	4.5733	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	3	S30408	29.124	3Sm=411	

Table 6.3. Simultaneous pressure strength check of tube process and shell process

Stress strength and combined stress strength	path	material	Stress intensity calculation(MPa)	Stress strength allowable limit(MPa)	result
A local film stress S_{II}	1	S30408	38.055	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	1	S30408	194.16	3Sm=411	
A local film stress S_{II}	2	S30408	12.318	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	2	S30408	90.736	3Sm=411	

A local film stress S_{II}	3	S30408	8.5406	1.5Sm=205.5	pass
Primary + secondary stress intensity S_{IV}	3	S30408	24.765	3Sm=411	

7. CONCLUSION

In this paper, the analysis and design criteria of pressure vessels are adopted to perform static analysis and strength check on the tube plates of spirally wound heat exchangers with threaded tubes. Through the above analysis of the finite element model of the tube sheet, the following conclusions can be drawn:

(1) The difficulty in realizing the analysis of the solid model tube sheet using the finite element method is mainly due to the complexity of the structure and load boundary conditions of the tube sheet, resulting in a large number of units, and the application of various loads takes time and trouble.

(2) Under the current computer hardware and software analysis conditions, the finite element method can be used on the computer to analyze the tube plate of the physical model. Moreover, this finite element tube plate analysis technology is currently the best tube plate analysis technology and can realize the analysis of any number of tube passes and shell side number heat exchanger tube plates. It can better mold the stress and deformation of the plate under the pressure load.

(3) Hazardous conditions of the heat exchanger do not necessarily occur under normal operating conditions where the tube load and the shell load act simultaneously, but may occur under a transient operating condition. Therefore, in order to ensure safety, it is very necessary to analyze various operating conditions during heat exchanger analysis and then compare them to find dangerous working conditions.

(4) Taking into account several dangerous operating conditions existed in the heat exchanger, in order to avoid danger, the heat exchanger should reasonably arrange the sequence of valve opening when driving or parking. Under conditions permitting, the equipment should be opened at the lower pressure side of the valve, and then open the valve on the higher pressure side, so as to avoid the danger of high-pressure side load alone. When the equipment is parked, on the contrary, the valve on the higher pressure side should be closed first, and then the valve on the lower pressure side should be closed, so as to avoid dangerous working conditions in which the high pressure load and the temperature load act simultaneously.

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