Reliability Study of the Hydraulically Expanded Tube-to-Tubesheet Joint

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1 Introduction

Shell-and-tube heat exchangers are widely used in petroleum and petrochemical industries. Connection of the tube and tubesheet has always been the weakest part in the structure and the major reasons for the failure of heat exchangers. Welding and expanding are the two methods for the connection of the tube and tubesheet and the expansion methods can be subdivided into mechanical rolling, explosive expansion, and hydraulic expansion. Hydraulic expansion is a relatively new method for the connection of tube and tubesheet but it is being more and more widely used in engineering. Yokell [1,2] discussed some practical aspects of the expanded and welded-and-expanded tube-to-tubesheet joints and reviewed elastic-plastic tube expanding theory.

Tube-to-tubesheet joints must provide a tight seal between the shell-side and the tube-side fluids, and sufficient strength to support the axial load, which have been deeply and widely studied theoretically, numerically, or experimentally. Two-dimensional models [3–6] were once used to solve the problem of the residual contact pressure between the tube and tubesheet, in which strain hardening and temperature-dependent properties of materials could be involved. One problem related to the two-dimensional models is how to choose the appropriate outer diameter of the annulus, which was addressed by Chaaban et al. [7] using nonlinear finite element method combined with statistical analysis and Kohlpaintner [8] using special Airy stress functions with confirmation of finite element analysis.

Finite element analysis and experimental measurements are more acceptable ways to investigate the reliability of the expanded tube-to-tubesheet joints. Reinhardt et al. [9] studied the transition zone of a hydraulically expanded tube which was subjected to a detailed experimental, finite element, and analytical evaluation. The influence of the expansion depth and the initial cold work on the post-expansion residual stresses was investigated. Allam et al. [10] also calculated the maximum residual stresses introduced in the transition zone of expanded tube-to-tubesheet joint. In addition, they conducted the axisymmetric two- and three-dimensional (3-D) finite analysis on the effect of thermal loading, temperature cycling, and the strain hardening of the tube material over the residual contact pressure and the maximum tensile residual stresses [11,12]. Duan et al. [13] did similar work with a 3-D finite element model. Aberl and Strohmeier [14] conducted the analysis of thermal stress in tubesheets with hydraulically expanded joints. Allam and BazerGUI [15] investigated the axial strength of tube-to-tubesheet joints numerically and experimentally. The results were compared with those obtained with previously proposed simplified analytical methods. They found that the finite element model is highly accurate for calculating the joint axial strength and the pull-out strength is overestimated when using simple analytical solution to calculate. In their study, the tubesheet holes were not grooved. Shuaib et al. [16] conducted the experimental investigation to evaluate the effect of heat exchanger tubesheet hole enlargement. Merah et al. [17] measured the tube-to-tubesheet joint strength in terms of residual contact pressure between the tube’s outer surface and the tubesheet hole surfaces. The joint integrity is affected by several design parameters, including the type of material and the initial radial clearance.

In this paper, a three-dimensional parametrized finite element model is established for the nonlinear analysis of the hydraulically expanded tube-to-tubesheet joint. Distribution of the residual contact pressure on the contact surface between the tube and the tubesheet is investigated. It is found that sealing circular bands exist on the contact surface which enhance the sealing of the joint since the residual contact pressures on the sealing circular bands are higher than on other positions. The sealing circular bands are located close to the two ends of the hole where it is not grooved, but they are located at the two brinks of the groove for a grooved hole and in the latter case, the residual contact pressures are even higher, reflecting that the joint with a grooved tube-sheet hole is more capable of sealing. Experiments and finite element calculations for the pull-out force of the joint are also performed for different expansion pressures and groove widths. Results show that with the increase of the groove width, the measured pull-out force increases significantly and becomes larger and larger than the calculated one, which is owing to the scratch on the contact surface between the tube and tubesheet. [DOI: 10.1115/1.2222375]

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transfer area in a given cylindrical shell. In this study, the triangular layout is employed. As this layout is periodically symmetric with a 60 deg of period, only 360 deg/12 range of the joint is selected and modeled as shown in Fig. 1. In the present study, the interaction between the expanded joints is not considered, or in other words, only a single tube expansion is investigated.

In order to investigate the effects of grooves on the joint properties, two finite element models are established as shown in Fig. 2 for an ungrooved model and Fig. 3 for a grooved one.

The geometrical parameters of the expansion joint are as follows (in mm).

- Inner diameter of the tube: $D_i=20$;
- Outer diameter of the tube: $D_o=25$;
- Tube length: $L=183$;
- Protrusion of the tube over the tubesheet upper surface: $\delta_1=3$;
- Thickness of the tubesheet: $W=48$;
- Diameter of the hole: $D_s=25.4$;
- Spacer between adjacent holes: 32;
- Distance from groove center line to tubesheet upper surface: $d=10$;
- Depth of the groove: $W_2=0.5$;
- Width of the groove: $W_1=6$.

### 2.2 Load and Boundary Conditions

The only load in this analysis is hydraulic expansion pressure. Two loading steps are considered in this study: (1) expansion pressure loading step. In this step, expansion pressure is increased from zero to a prescribed value $p_{exp}$; (2) expansion pressure unloading step. In this step, expansion pressure is decreased from $p_{exp}$ back to zero again. Because the analysis deals with nonlinearities, such as gaps and material yielding, each of the two loading steps has to be divided into many sub-loadsteps.

Referring to Fig. 1, the boundary conditions are set as follows. Boundaries 1 and 6 are symmetric boundaries. They can be defined in a cylindrical coordinate system as $U_y=0$ for all nodes on the boundaries. Boundary 2 is the inner surface of the tube section inside the tube hole, and it is a pressure boundary subjected to the hydraulic pressure during expanding process. Boundary 3 is the tubesheet hole surface that is not expanded, thus, it is a free boundary. Boundary 4 represents the inner and outer surfaces of the tube sections outside the tubesheet hole, and the upper and lower surfaces of the tubesheet. They are free boundaries. All nodes on the boundary for the lower end-surface of the tube are axially fixed, i.e., $U_z=0$ in a cylindrical coordinate system. Boundary 5 is a virtual cylindrical boundary cut from the tubesheet. Its radial displacement boundary condition is actually very complicated, but should be between two extreme boundary conditions, i.e., the free and fixed conditions. Because this boundary is far away from the tube-to-tubesheet contact area where research interest is concentrated, it is reasonable that the conditions of Boundary 5 will have only slight effects on the tube-to-tubesheet contact results. Thus, as an acceptable approximation, this boundary condition can be simplified to be either free or fixed in radial direction. In this study, free boundary condition is applied.

The internationally widely used commercial finite element software ANSYS is employed in this study. Solid45 (3-D solid element) is used to mesh all the components of the joint. Contact173 and Target170 are used to simulate the contact pair of the tube and tubesheet. The meshed expansion joint is shown in Fig. 4.

In this study, the material of the tubesheet is 16Mn (low alloy steel) and the material of the tubes is No. 20 (low carbon steel). These materials are most commonly used to construct heat exchanger. In this paper, double linear model has been used to represent the stress-strain relations of the joint materials.

### 3 Tightness Analysis of the Expansion Joint

The mechanism of hydraulic expansion can be expressed as follows. Under the action of the hydraulic pressure, the tube section inside the tubesheet hole expands elastically and then elastically-plastically. The outer surface of the tube comes into contact with

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**Fig. 1 Selection of the expansion joint**

**Fig. 2 Ungrooved joint geometry**

**Fig. 3 Grooved joint geometry**
the surface of the tube hole on the tubesheet. Increasing the expansion pressure, the tubesheet is also elastically or even elasto-plastically deformed. When unloading the pressure, the deformed tube and tubesheet rebound elastically. But as the tube is serious plastic deformed, the rebounded tubesheet tightly binds the tubes and, thus, forms expansion joints. From this process, it is seen that sufficient residual contacting pressure on the contact surface between the tube and tubesheet is necessary to ensure the tightness and strength of the joint. So in this section, we will concentrate on the investigation of the residual contacting pressure on the contact surface between the tube and tubesheet.

Figure 5 is the distribution of residual contacting pressure in a cylindrical coordinate system on the contact surface between the tube and tubesheet in which the tubesheet hole is not grooved and the expansion pressure is $P_{\text{exp}} = 170$ MPa. It is seen that the distribution of residual contacting pressure on the contact surface is not uniform, instead, near the tubesheet surfaces, there are two circular bands on which the residual contacting pressure is high. We call these bands “tightness bands,” because they play key roles in keeping the tightness of the joint and this is explained as follows. Fluids usually leak along the least contacted path. So if the expansion joint is leaking, the fluids must pass through the bands along the path where the residual contacting pressure is the smallest. Therefore, the smallest residual contacting pressure on the tightness bands determines the tightness of the joint. The larger the smallest residual contacting pressure is, the better the tightness of the joint will be. It is very difficult to measure the leakage of the joint, but qualitatively we can use the smallest residual contacting pressure to describe the tightness of the joint. For a given joint geometry and expansion pressure, the total contact force on the contact surface should be constant. Therefore, existence of the tightness bands will enhance the tightness of the expansion joint.

Figure 6 is the distribution of residual contacting pressure on the contact surface between the tube and tubesheet in which the tubesheet hole is grooved and the expansion pressure is $P_{\text{exp}} = 170$ MPa. It is seen that the residual contacting pressure distribution is not uniformly distributed on the contact surface either. But the tightness bands are located on the two brinks of the groove instead of near the tubesheet surfaces as shown in Fig. 5. It is found that for a same expansion pressure, the residual contacting pressure on the tightness bands for a grooved joint is much larger than that for an ungrooved joint. This is the reason why the grooved expansion joint is also good at tightness.

The effects of the expansion pressure on the tightness of the joint are shown in Fig. 7 in which the expansion pressure is changed from 150 MPa to 220 MPa. It is clear that the smallest residual contacting pressure increases with increasing expansion pressures, meaning the larger the expansion pressure, the better the joint tightness. The grooved joint gives much higher smallest residual contacting pressure, or the grooved joint has better tightness.
4 Strength Analysis of the Expansion Joint

Another parameter to reflect the reliability of the expansion joint is the strength or the pull-out force. Different from the joint tightness, the pull-out force depends not only on the distribution of the residual contact pressure on the contact surface, but also on the friction coefficient between the two materials forming the joint. So the above-defined tightness bands and the smallest residual contacting pressure are not enough to describe the strength of the expansion joint. Besides, the friction coefficient must be experimentally determined prior to any numerical calculation of the pull-out force. Therefore, the pull-out tests were performed in this study.

The specimen for the tests is shown in Fig. 8. The thickness of the tubesheet is 48 mm and the material is 16Mn (low alloy steel). To be consistent with numerical model, triangular layout was selected with a spacer of 32 mm between any two adjacent holes. The size of the tubes are $d_{25} \times 2.5$ mm and the material is No. 20 (low carbon steel). Every other hole is expanded so that the interaction between the expanded joints is eliminated. There are five expansion joints under the same expansion pressure in a specimen and all are performed pull-out tests. Pull-out force is measured for each joint and the mean value from the five measurements is taken as the experimental value for the pull-out force. The tests were performed on a universal tension machine with a specially designed grasp.

Figure 9 shows the pull-out forces against expansion pressures for the ungrooved joints. As expected, the pull-out forces increase with increasing the expansion pressures. In order to calculate the pull-out force numerically, the friction coefficient is determined by the experimental pull-out force at $P_{\text{exp}}=160$ MPa using try-and-error method. This friction coefficient will be used in other cases including different expansion pressure and different groove widths. It is seen from Fig. 9 that the calculated pull-out forces are in good agreement with the experimental measurements.

Figure 10 shows the experimental and calculated results of the
pull-out forces for different groove widths under the expansion pressure of 160 MPa. It is seen that if the groove width is smaller than 6 mm, the calculated pull-out force is very close to the experimental one. But if the groove width is larger than 6 mm, the experimental pull-out force substantially increases with increasing the groove width and the difference between the experimental and calculated results becomes larger and larger. Comparing Figs. 9 and 10 tells that the pull-out force for a grooved joint is much larger than that for an ungrooved joint, meaning that a grooved joint has a large strength, this is because for a grooved joint, the tube metal was squeezed into the groove when forming the joint and, thus, it is harder to pull it out.

To explain Fig. 10, fracture surfaces were examined. For the ungrooved joint, there are no scratches on the pull-out tube surface and groove surface. But for the grooved joint, there are scratches on the pull-out tube surface and groove surface, and the larger the groove width, the more serious the scratches are, as shown in Figs. 11 and 12 in which a whole brink of groove has been taken off. This phenomenon indicates that during the expansion, the tube section corresponding to the groove of the tube hole is squeezed into the groove and scratched with the groove brinks when pulling-out. These scratches increase the pull-out force, which is hard to calculate, so the difference between the experimental and calculated results becomes larger and larger with increasing groove width.

5 Conclusions

In this study, finite element analysis and experimental measurements are carried out to investigate the reliability of the hydraulically expanded tube-to-tubesheet joints. Conclusions are drawn as follows.

(1) The residual contacting pressure on the contact surface between tubes and the tubesheet is not uniformly distributed and two tightness bands are found near the surfaces of the tubesheet or at the two brinks of the groove on the tubesheet hole. The tightness bands enhance the tightness of the joint.

(2) Compared with the ungrooved joint, the residual contacting pressure on the tightness bands for the grooved joint is much higher, indicating a grooved joint has better tightness.

(3) Both experiments and numerical calculations show that the larger the expansion pressure, the greater the pull-out force. As a result of tube metal squeezing into the groove, the pull-out force for a grooved joint is much larger than that of an ungrooved joint.

(4) Under the same expansion pressure, increasing the
groove width, the measured pullout force increases significantly and becomes larger and larger than the calculated one, which is owing to the more serious scratches on the contacted tube and the hole surfaces as more tube metal squeezed into the groove for a larger groove width.

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References