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Investigation of the Expansion Requirement of Tube to Tubesheet Rolling Fitting

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ABSTRACT. Tube Expansion is the process of reducing a tube wall by displacing the outside diameter of the tube against a fixed container such as rolling tubes into tube sheets. To assure a proper tube joint to the tubesheet, the tube wall must be reduced by a predetermined percentage. Different methods have been used to estimate the necessary amount of expansion in tube to tubesheet rolling expansion for strong tube-to-tubesheet joint. The first commonly method depends on torque and pull out strength data collection by experimentation. This method fits with small and large clearance but it suffers from cost of experiments to collect the necessary data. The second method depends on theoretically calculation of the necessary inner radius displacement that to insure proper residual contact stress. That stress gives indication the strength of the joint. This method fits with small clearance as the joint strength decreases until it vanishes as the clearance increases. This paper proposes a new method that depends on the fixed outer tube surface displacement. The finite element analysis results show that, the proposed method provides good results for both low and large clearance.

Keywords**:** Tubesheet, Finite Element, ANSYS, Over-Tolerance-Clearance, Pull-Out-Strength, Roller-Expansion.

1. Introduction

Tight, durable seating of tubes in tubesheet is one of the critical functions in manufacturing heat exchangers for high-pressure steam generators, boilers for conventional power generation, food and pharmaceutical processing systems and condensers. In selecting the best method for expanding the tube within the sheet, producers must examine factors including safety, speed, cost, operator skill requirements, repeatability and overall quality. With the number of tubes in a sheet ranging from hundreds to tens of thousands, and the process that expanded manually one by one, the stakes for achieving the best balance of quality and cost are high.

The importance of expanding tube into tubesheet is to produce a residual interfacial stress between the tube and surrounding tubesheet equivalent to attenuation the tubesheet onto the tube. The residual stress creates equal and opposite stresses at the inner side of the hole and the outer side of the tube. Most of literatures [5,20] are based upon these assumptions: uniform pressure applied inside the tube holes, expanded of the tube into a centered hole in an infinitely large plate taking the account of the effect of tube adjacent holes to the center one, and assuming the stress parallel to the tube axis is zero (plane stress model).

The axisymmetric and/or plane stress finite element models have been often considered to simulate the nonlinear behavior of the expansion of tube-to-tubesheet joint. Material, boundary conditions, and geometry are major sources of nonlinearities. Analytical work by Goodier and Schoessow [1] and a companion experimental work by Grimison and Lee [2] are important contribution to understand the process of tube expansion. Soler and Hong [3] introduced a plane stress model with single tube to solve the residual contact pressure. The finite element model developed by Wang and Soler [4] presented the effect of adjacent holes and outer boundary conditions on the tube-to-tubesheet joints and determined the appropriate equivalent external annulus diameter.

In the case of a simultaneous expansion process, Kalnins et al [5] adopted a single-tube axisymmetry model to determine the residual contact stresses in the transition zone. Similarly, H. Ma [6] adopted a single tube surrounded by an annular sleeve model. Chaaban [7] also studied this problem by performing elastoplastic finite element analyses using a seven-tube model. The same type of parametric analysis was performed by Updike et al [8] but on an axisymmetric single tube model to investigate the residual stresses in the transition zone. Moreover, the plane stress model proposed by Andrieux [9] was used to identify the stress in steam generator tubes from profile measurements. Kohlpaintner [10] presented an elastic-plastic computation of the residual contact stress. Cizelj and Mavko [11] obtained an estimation of the residual hope stresses using a nonlinear finite element modeling of the tube expansion into tubesheet using rolling processes. They also investigated a scatter of the residual stresses due to the stochastic variations of dominant influencing parameters. Sherburne, et al [13] used X-ray diffraction techniques and finite element analysis to measure the residual stress distribution and cold work in a tube roll transition and in assorted rolled tube mockups. The increase of clearance between tube and tubesheet reduced the interfacial residual contact stress that sustain the joint strength as in Merah et al [14]. Moreover, the tube projection in front of tubesheet has slight effect on the interfacial residual contact stress, Merah et al [15]. Merah et al [16], Al-Aboodi et al [17, 18] and Shuaib et al [19] had investigated the over-enlarger tube-to-tubesheet using variety of input parameters such as clearance, grooves and material properties.

2. METHODOLOGY

Many industrial companies such as Airetools [23] has used the interference method in roller expansion. Interference method assumes that the tube is perfectly round and requires precise measurement, further; it works better while the tubesheet hole has not been enlarged before.

The interference fit method requires four measurements that are tube inner diameter I.D., tube wall thickness, tubesheet holes diameter, tube outer diameter O.D. In this method, the increase in the I.D. of the tube is used to determine the theoretical decrease in wall thickness [22]. Then the inner tube displacement is calculated as:

$$u_i = c + \%WR \cdot t \qquad (1)$$

Where u_i is the inner wall reduction; c is the tube-tubesheet clearance; $\%WR$ percentage of wall reduction; and t is the tube thickness. Equation after rearranging will be:

$$u_i - c = \%WR \cdot t \qquad (2)$$

The proposed method depends on volume constancy that the tube volume after and before expansion remains constant because of the volume constancy as:

$$V_1 = V_2 \rightarrow A_1 = A_2 \frac{l_2}{l_1} \qquad (3)$$

Assume no expansion on the lateral direction then $\frac{l_2}{l_1} = 1$ or:

$$r_{o1}^2 - r_{i1}^2 = r_{o2}^2 - r_{i2}^2 \qquad (4)$$

The subscript (1) means before expansion, (2) means after expansion, (i) means inner, and (o) means outer.

Let $u_i = r_{i2} - r_{i1}$ and $u_o = r_{o2} - r_{o1}$

Rearrange $u_i + r_{i1} = r_{i2}$ and $u_o + r_{o1} = r_{o2}$

Substitute in (4) to be:

$$r_{o1}^2 - r_{i1}^2 = (u_o + r_{o1})^2 - (u_i + r_{i1})^2 \qquad (5)$$

Simplifying equation (5) to be:

$$(u_o + r_{o1})^2 = (r_{o1}^2 - r_{i1}^2) + (u_i^2 + 2u_i\, r_{i1} + r_{i1}^2) \qquad (6)$$

Or

$$(u_o + r_{o1})^2 = r_{o1}^2 + 2u_i\, r_{i1} + u_i^2 \tag{7}$$

Taking the root of the both sides

$$u_o + r_{o1} = \pm\sqrt{r_{o1}^2 + u_i^2 + 2u_i\, r_{i1}} \tag{8}$$

Ignoring the negative root yield:

$$u_o = \sqrt{u_i^2 + 2u_i\, r_{i1} + r_{o1}^2} - r_{o1} \tag{9}$$

Using r_{i1}=7.42, and r_{o1}=9.53 and using variable clearance with 5% wall reduction produce the curves shown on figure (1) It shows (the inner displacement – clearance) verse clearance on the dotted line which is constant for the clearance spectrum. The solid line is the (outer tube displacement – clearance) verse clearance spectrum which is decrease until clearance around 0.4 mm it reach zero then goes to negative. Thus, there is no contact or penetration between tube inner surface and tubesheet hole after expansion for clearance higher than 0.4 leading to confirm the tube will not bond (joint) with the tubesheet. Furthermore, there is a higher penetration between tube and tubesheet for the small clearance which means a higher torque than necessary one has been used that my defect the tube.

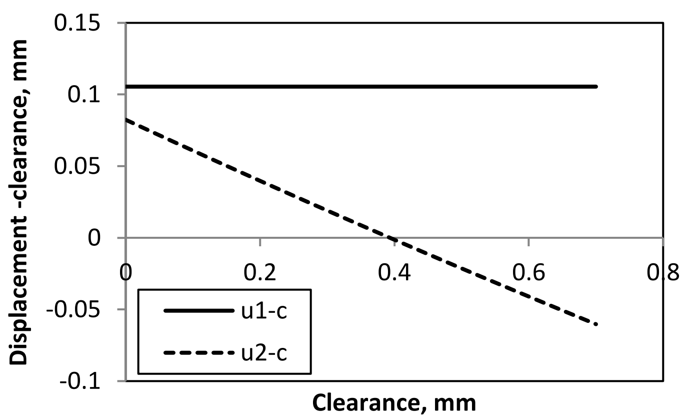

Fig. (1). (Inner Displacement – Clearance) verse Clearance for the old method

The old method fits with TEMA [3] for small clearance ranges, In contrast, TEMA does not support data for the over tolerance clearance.

The proposed method uses a fixed outer tube displacement instead of fixed inner tube displacement. Subsequently, the inner tube displacement is calculated using the volume constancy criteria. This method assures a fixed amount of penetration between the tube and tubesheet.

Equation of the outer displacement – clearance could be written as:

$$u_o - c = Pn \cdot t \tag{10}$$

Where $u_o$the outer wall displacement, c is: tube-tubesheet clearance, t is the tube thickness, and Pn is neither apparent nor actual percentage of wall reduction but it is the penetration amount required for strong joint, which goes from 0.01 to 0.05.

Rearranged we get

$$u_i = \sqrt{u_o^2 + 2u_o r_{o1} + r_{i1}^2} - r_{i1} \tag{11}$$

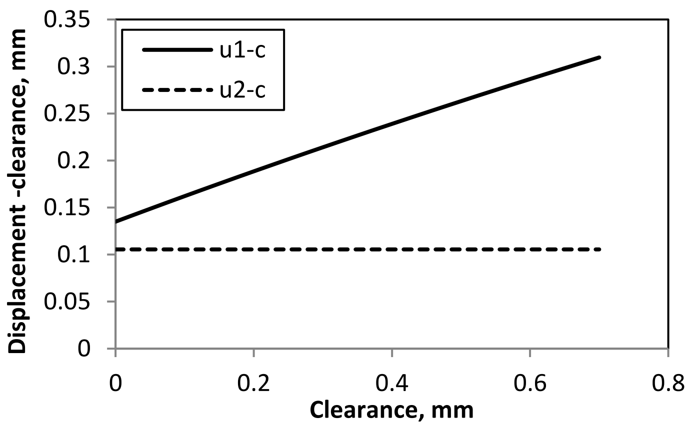

Fig. (2). (Inner Displacement – clearance) verse Clearance for proposed method

Using r_{i1}=7.42, and r_{o1}=9.53 and using variable clearance with 5% tube thickness penetration of tube to tubesheet. Figure 2 shows the (inner displacement – clearance) vs. clearance on the solid line, which is increased for the clearance spectrum. The dotted line is the outer tube (displacement – clearance) verse clearance spectrum, which is constant. This behavior means at all the clearance spectrum; there is enough amount of tube penetration to tubesheet. Moreover, it guarantees an enough strength joint at over tolerance clearance.

3. Finite Element Analysis

The equivalent sleeve diameter has been used in a number of studies [10, 11, 13-16] that used to include the ligament effect on joint strength. The equivalent sleeve is a single hole model that will produce contact stress distribution or deflection, depending on the objective of the study, around the hole equivalent to the average of those around holes to the test hole on the real tubesheet configuration. Since this study concerns the same configuration of the stabilizer feed/bottom exchanger used in the work by Merah et al. [11] and Shuaib et al. [19], the sleeve dimensions will remain unchanged. As in Figure 1 (a), the tube inner and outer radii are 7.425 mm, 9.525 mm respectively, and the tubesheet inner and outer radii are 7.425 + c mm and 36 mm respectively, where c is the radial clearance, which will be varied here from 0 to 0.5 mm. It should be mentioned that for this tube dimension the TEMA allowable radial clearance is about 0.16 mm.

The tube and tubesheet areas are meshed using the 2-D VISCO108 element as in Figure 3-b. (VISCO108) is quadratic element defined by eight nodes having two degrees of freedom at each node: translations in the nodal x and y directions. From its special features; it has a rate-dependent plasticity, stress stiffening, large deflection, large strain, and adaptive descent. Element could be used with axisymmetric or plane strain problems. Because of expected large deflections due to large over tolerances used in this study NLGEOM, which accounts for geometric nonlinearities is activated in order to update the geometry at each sub step. CONTA172 and TARGE169 elements are used to represent the 2-D contacting and deformable surfaces respectively. They have the same geometric characteristics as the solid element face with which they are connected.

The elastic-plastic behavior tube and tubesheet material is represented by bilinear curves. Tube is seamless cold-drawn low carbon steel type ASTM 179 and the tubesheet material is carbon steel type ASTM A5 16 G70. Their mechanical properties were measured in the lab by performing tensile tests. Measurements results show that the tube and tubesheet's average yield stress was approximately 250 MPa (36.3 ksi) and the modulus of elasticity was 207 GPa (30,000 ksi). These properties were specified for the tube and tubesheet materials in the finite element analysis.

The curve in the plastic region was approximated by a linear relationship. The slope of the approximated line (or lines) in the plastic region of the true stress-strain diagram defines the tangent modulus of plasticity, E_{tt}. An elastic-perfectly-plastic material is that having zero tangent modulus. The approximate value of the tangential modulus of plasticity, E_{tt}, for the tube material used in the current stabilizer feed/bottom exchanger was 733 MPa (106.4 ksi). However, to investigate the effect of tube and tubesheet material strain hardening on contact stresses, E_{tt} values ranging from 0 to 1.2 GPa will be considered. This range covers most of steel materials in heat exchanger tubes and tubesheets.

The tube and tubesheet were constrained from translation in the axial direction on the primary side (Fig. 3-b). The exact roller profile is represented as a rigid bodyline (Figure 3-a). Loading is performed in three steps the first, consists on displacing the roller radially in small increments until contact is established. The targeted percent wall reduction (5% in this study) is reached by performing fifty sub steps in the second loading step. The third step consists on the simulation of the retraction of the roller.

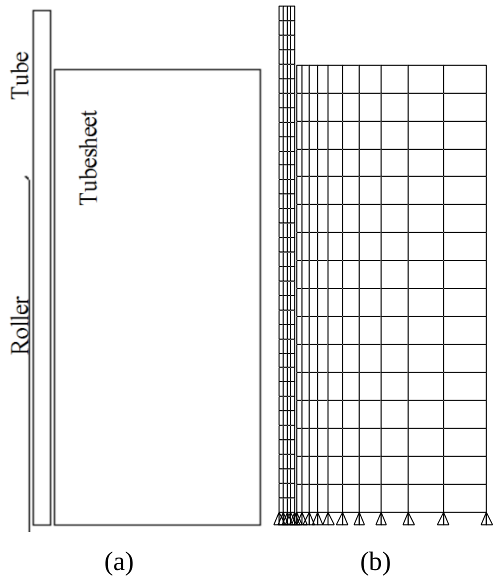

Fig. (3). a) Geometry and b) meshed axisymmetric model.

The expanded length, which is equal to roller length shown in Figure 3-a, was 47.25 mm (1.872 in.); this represents about 75% of the tubesheet thickness. The total displacement load was specified with the knowledge of the required percentage wall reduction (5%WR), tube thickness t, and initial clearance c, using the following equation[19]:

$$u_r = \left(\frac{\%WR.t}{100} \right) + c \tag{8}$$

4. RESULTS AND DISCUSSIONS

Finite element simulation was applied for both methods, old and proposed one. Figure (4) shows the distribution of residual contact stresses between tube and tubesheet. The contact stress remains constant across the contact area with 61 MPa except the end region of enlarging tool and beyond it. At the end region of enlarging tool, the contact stress increased with a hump shape and it vanished beyond that. This case will be repeated with increasing of clearance to draw a curve between clearance and residual contact stress.

The distribution of residual contact stress vs. clearance is shown in Figure (5). It is found that for small clearance i.e. c < 0.3 mm, the contact stress is similar for both methods with small different. Furthermore, for c > 0.3 the contact stress in the proposed one increased gradually while the old method, the contact stress is decreasing until it vanished at clearance beyond 0.55 mm. In addition, the trend of the proposed method is similar to the experimental work by Shuaib et al [4] that the torque criterion of expansion has been used.

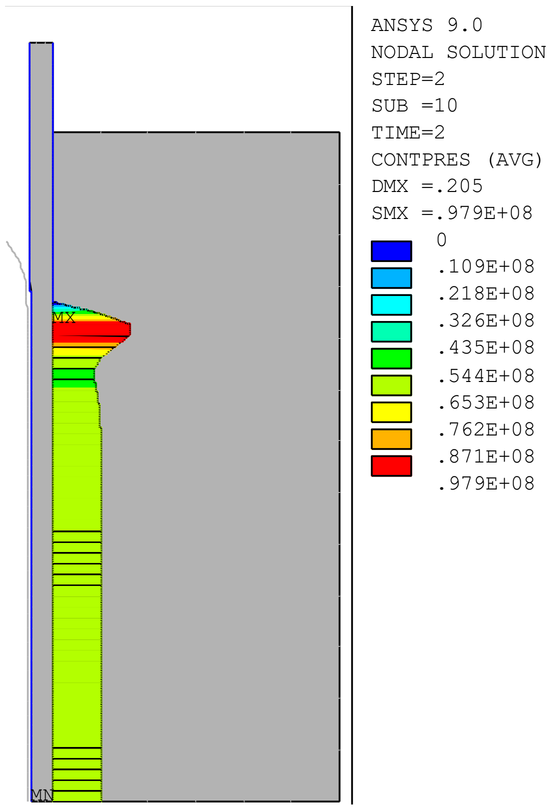

Fig. (4). Contact stress, radial stress and equivalent stress for projection =0, clearance =0 friction =0 Ett=0.8GPa, WR =5%

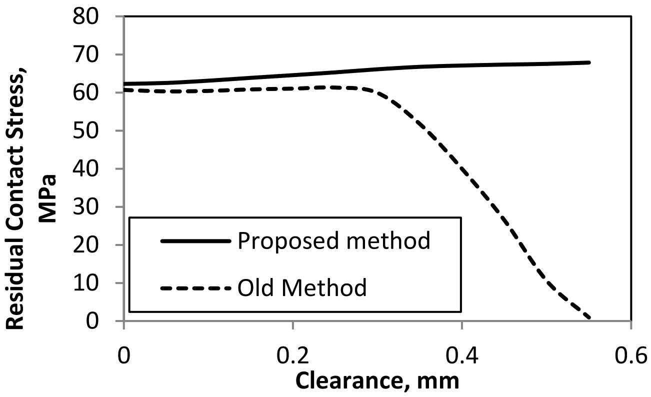

Fig. (5). Residual Contact Stress verse Clearance

5. CONCLUSION

There are two common methods used to assure a sufficient expansion and strong joint in tube to tubesheet rolled expansion. One of them depends on torque and pullout strength of collected data. This method suffers of its cost effective because the data necessary gathered using experimental practice. The second method depends on fixed inner radius displacement and that is calculated theoretically. This method fits for lower clearance but it is not suitable for large clearances. The proposed method depends on the fixed outer tube surface displacement, which also the sufficient expansion calculated theoretically. This method gives good results for low and large clearance, in addition to its similar trend to the experimental data used in torque criteria.

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دراسة التوسيع المطلوب للأنابيب داخل صفائح الأنابيب باستخدام الدرفلة المناسبة

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ملخص البحث. توسيع الأنابيب هي طريقة لخفض سماكة جدار الأنبوب بواسطة ازاحة القطر الخارجي للأنبوب داخل حاوية. ومثال ذلك درفلة الأنابيب داخل ثقوب الصفائح في المبادلات الحرارية. لضمان التصاق الأنابيب داخل ثقوب الصفائح، يجب خفض جدار الأنبوبة بنسبة مئوية محددة سلفا. وقد تم استخدام أساليب مختلفة لحساب المقدار اللازم لتوسيع أقطار الأنابيب بالدرفلة داخل ثقوب الصفائح لوصلة قوية بين الأنابيب والصفائح. أول طريقة يكثر استخدامها، كانت تعتمد على العزم و بيانات قوة السحب والتي يتم الحصول عليها بالتجريب. يناسب هذا الأسلوب مع السماحيات الصغيرة والكبيرة بين الأنابيب والثقوب، ويكمن عيبها في تكلفة تجارب جمع البيانات اللازمة. الطريقة الثانية يعتمد على الحساب النظري لمقدار إزاحة القطر الداخلي للأنبوب ليعطي قدرا مناسبا من الإجهاد المتبقي الإتصالي بين الأنبوب والثقب. الإجهاد المتبقي يعطي دلالة على مقدار قوة الوصلة. هذه الطريقة مناسبة مع السماحيات الصغيرة بين الأنابيب والثقوب وتعطي نتائج مناسبة، ولكن الإجهاد المتبقي يختفي مع السماحيات الكبيرة. لذلك تقترح هذه الورقة أسلوب جديد يعتمد على ازاحة سطح الأنبوب الخارجي الثابتة. باستخدام طرق تحليل العناصر المحددة تم حساب الجهد المتبقي الالتصاقي وقد أعطى نتائج جيدة مع السماحيات الصغيرة وكذلك الكبيرة.