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Kinematic Simulation of Three Rollers in Circular Motion Using 2D Planar FE Modeling

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Abstract—A method is established for joining a tube to a tubesheet by expanding the tube to the tubesheet. The expansion done using different method such as rolling. The joint integrity is important to ensure the roundness of the structure of the heat exchanger, and to prevent leakage. Many parameters affect the joint integrity such as the initial clearance, wall reduction and material properties. The residual contact stress is the main parameter indicating the strength of the joint. In this paper, the two dimensional planar FE model is established for the tube to tubesheet roller expansion. The planar model is great way for our attempt to understand and analyze the problem. The planar model reveals a realistic motion of the rollers in addition to the exact geometry of the tubesheet. This eliminates the need for the assumption of uniform expansion and tubesheet sleeve diameter sets on the axisymmetric model. That, the ligament dimensional effect will be shown on this model.

Index Terms—Kinamatic, Simulation, Finite Element, Rolling, Modeling.

I. INTRODUCTION

The expanded tube joint process was derived from the French-invented process of autofrettage (1840). The first paper published on tube-to-tubesheet joint was due to P. H. Oppenheimer (1927). Grimison and Lee (1943) gave in their paper the result of an experimental investigation to determine the fundamental parameters involve in tube expanding, the various practical methods of measuring the degree of expansion, the optimum degree of expansion, and the ultimate strength of the expanded joints under various conditions of service. Scot, et al. (1984) carried out an experimental work to determine the stresses in hydraulically expanded tube. The main area of interest is the transition region between the expanded and unexpanded sections of a tube. X-ray diffraction, the stress corrosion cracking test and strain gagging were the prime measuring techniques used. Cizelj and Mavko (1995) obtained an estimate of the residual hope stresses using a nonlinear finite element simulation of the tube-to-tubesheet rolling processes. In addition, they studied a scatter of the residual stresses due to the stochastic variations of dominant influencing parameters. Sherburne, et al. (1997) initiated a project to measure the residual stress distribution and cold work in a tube roll transition and in assorted rolled tube mockups, using X-ray diffraction techniques and finite element analysis. The finite element calculates OD and ID residual stress result for tube model with and without the yield strength gradient. It indicates similar trends and magnitude to those measured for the stress relieve mockups. Allam (1997) present in his thesis an attempt to investigate the hydraulically expanded tube-to-tubesheet joints using finite element analysis.

Three different models are used, namely: 2-D plane stress/strain, 2-D axisymmetric and 3-D. The FE models are mainly used to investigate the effect of the involved design parameters on the joint strength and level of maximum tensile residual stresses. The models are also used to simulate the complex joint behavior resulting from mechanically and/or thermally induced loads. In addition, they allow the investigation of the working condition (e.g. temperature effect and mechanical loading) on both the residual contact pressure and residual stresses. Allam, et al (1998) present on their paper a simplified 2-D finite element study on the loading, unloading, and residual stresses in the tube transition zone in hydraulically expanded tube-to-tubesheet expansion. Simulation of roller expanding of tubes into tubesheet by the finite element method is associated with a number of difficulties such as roller kinematics and the absence of loading axisymmetry. In fact, the mechanical rolling process is a good example of periodic symmetry with respect to the load application. A number of researchers have shown that the complexity of the tube–tubesheet roller expansion can be simplified by reducing it to an axisymmetric quasi-static problem (Aufaure1987; Merah et al. 2010, 2009; Updike et al.1992; Merah 2008; Al-Aboodi et al 2009; Cizelj and Mavko1995; Williams 1996, 2003). Displacements are applied in steps up to the required expansion and then decreased down to zero such that the tube–tubesheet contact ensures the strength of the connection joint after transmitting a large radial stress to tube's outer surface. Andrieux and Voldoire (1995) developed an axisymmetric FE model to simulate roller expansion using five steps. Cizelj and Mavko (1995) used the axisymmetric model in ABAQUS code to evaluate the residual stresses in one-step rolled tube to tubesheet joint. Williams (1996) used three load steps to complete the final mechanical roll expansion.

II. KINEMATICS OF THE PROBLEM

The roller has three motions, first: a rotation about its own axis, a planetary rotation around the inner surface of the tube, and a radial translation that deforms the tube. The three motions are related to each other. In this analysis, we will assume that the roller moves by one degree in its motion around the inner surface of the tube, and calculate the other two motions that correspond with the specified one. The mandrel has a uniform taper by an angle = 1.859 degree. The mandrel length is 122 mm, the right side radius is 6.76 mm, and the left side radius is 2.8 mm. the time of expansion was selected as12 seconds, The calculated axial speed being v = 122/12 mm/s which is equal to 10.167 mm/s.

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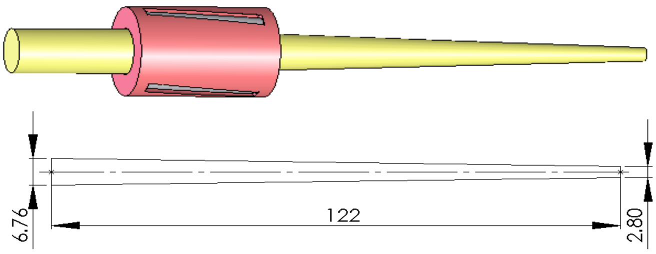

Figure 1. Mandrel dimensions

The mandrel rotates with a speed of 360 rpm. Thus the time of one revolution is 60/360 = 1/6 sec. and for one degree of motion is t = 1/(6×360) sec. The radial displacement for each angle of rotation could calculate using the equation:

$$\Delta r = \Delta z \tan\beta = tv \tan\beta \qquad (1)$$

Thus Δr mm/degree.

For clearance cl = 0.127 mm, tube thickness = 2.1 mm and wall reduction WR = 5%.

$$\Delta r_{tot} = cl + WR \cdot t \qquad (2)$$

Then Δr_{tot} = 0.232 mm.

The total angle of rotation is the total displacement divided by the displacement of one degree.

$$\theta_{tot} = \frac{\Delta r_{tot}}{\Delta r} = \frac{0.232}{1.528\times10^{-4}} = 1521.3^{0} \qquad (3)$$

Now, we have for each increment of rotational angle the radial displacement. This will give us the trajectory of the circumferential rotation around the inner side of the tube. The roller also rotates around itself. Its rotation is direct proportion with the circumferential motion.

The roller moves around the inner surface of the tube by angle θ. At this angle, the contacted portion curvature of the inner service of the tube is equal to the contacted roller curvature. The roller will rotate around itself by angle α. For very small motion, curvatures approximated by multi lines. The equation of equality is:

$$(R + r + \Delta r)\sin\theta = (r + \Delta r)\sin\alpha \qquad (4)$$

For small angle we can approximate $\sin\theta = \theta$. In addition, $\Delta r = 1.528\times10^{-4}$ mm/° which is very small and we can reduce it for both the nominator and dominator. Then Equation (4) will be:

$$\alpha = \frac{R + r + \Delta r}{r + \Delta r}\theta \approx \frac{R + r}{r}\theta \qquad (5)$$

Where

r : Roller radius (constant)

R: is the inner radius side of the tube (variable)

α: Self roller rotational angle

θ: Roller rotational around the inner side of the tube

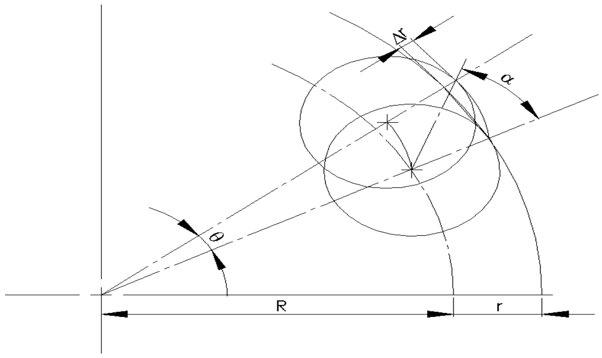

Figure 2. Roller Motion

III. MODEL AND MESH

The 2D planar model will be used with full circle to overcome the lack of loading symmetry. The rectangular element will used for the tube mesh and triangular element are used for tubesheet to overcome the non-uniform geometry of the tubesheet.

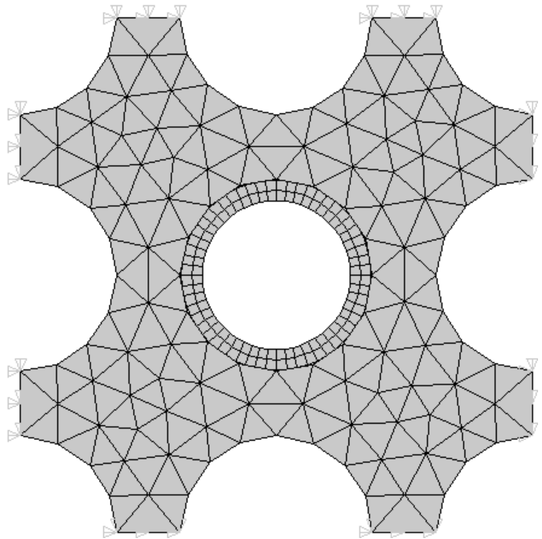

Figure 3. Mesh of the 2D planar model.

The eight nodded VISCO108 elements are used for both tube and tubesheet. The rectangular mesh was used for the tube because of it geometric uniformity, and the triangular mesh were used for the non-uniform geometry of the tubesheet mesh. Contact algorithms were built for each roller and the inner side of the tube with surface-to-surface contact.

The rollers were used as a rigid target and the inner tube surface is a fixable contact. The contact was built between the outer side of the tube and the inner side of the hole of the tubesheet with surface-to-surface contact. The key option were manipulated to get the best performance of analysis and preventing the non-convergent and the highly distorted elements. The contact stiffness were updated each step using the key option (10) sited equal to 1 for each algorithm. The key option (5) was set to equal 3 to close gap and reduce penetration with auto CNOF. The element used for contact is CONTA172 and TARGE169 for the target.

IV. RESULTS AND DISCUSSIONS

The planar model (plane stress) was used in this analysis with assumption of unit depth of the rollers, tube and tubesheet. The model gives more helpful information about the stress distribution at the tube and tubesheet. Our interest is the contact stress between tube and tubesheet and the stress distribution at the ligament. The three models of strain hardening materials are used with tube tangent plastic modulus E_{tt} of 0, 0.5 and 1 GPa. Wall reduction is 5% of tube wall reduced after its contact with the tubesheet. This means the deformation of the tube is the total of clearance and the percentage of wall reduction.

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It is found that the contact stress distributions are not unformed around the tube resulting from the geometry of the tubesheet, which is thick in some parts and thin on other parts. This phenomenon could be bowed to the ligament size at the axial and 45° inclination directions.

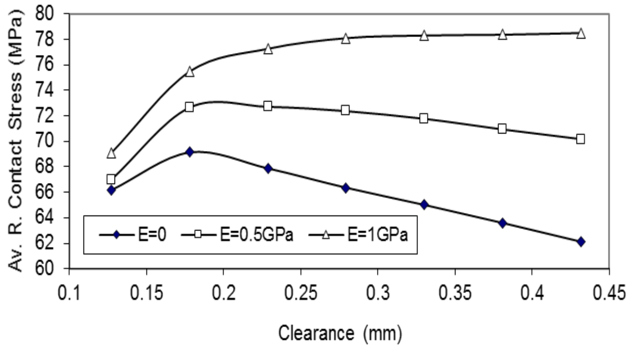

Figure 4 Residual contact stresses vs. clearances with variable tangent modulus of plasticity.

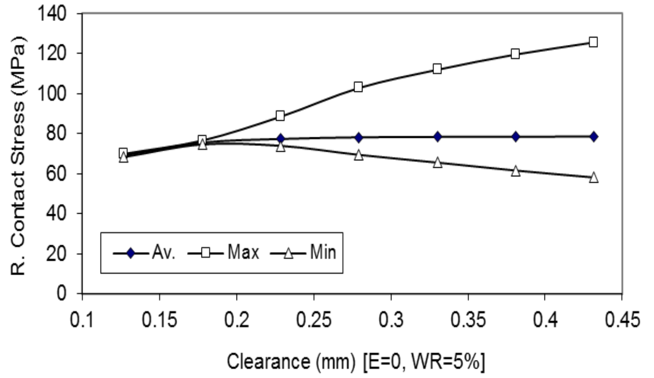

Figure 5 Residual contact stresses vs. clearances for E_{tt} =0 GPa

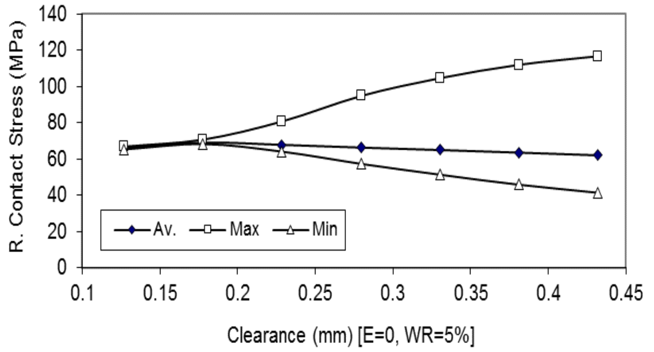

Figure 6 Residual contact stresses vs. clearances for E_{tt}=0.5GPa

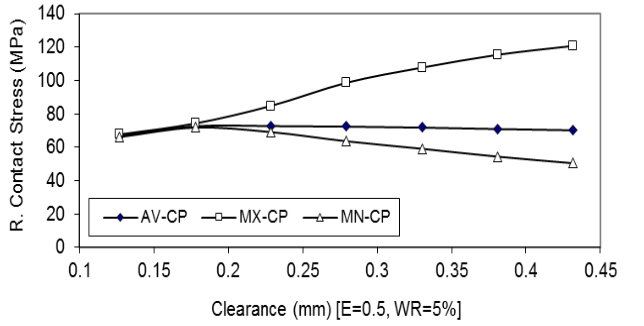

Figure 7 Residual contact stresses vs. clearances for E_{tt}=1GPa

The stress distributions along the x-axis and 45° inclination are presented in the following figures. The radial stress is in compression and its maximum at the contact between the tube and tubesheet. The circumferential stress is in tension all the time and its maximum stress at the tube. It suddenly decreases at the tube to the tubesheet transition which is continuing slightly decreasing afterward until meddle of the ligament then the stress slightly increased.

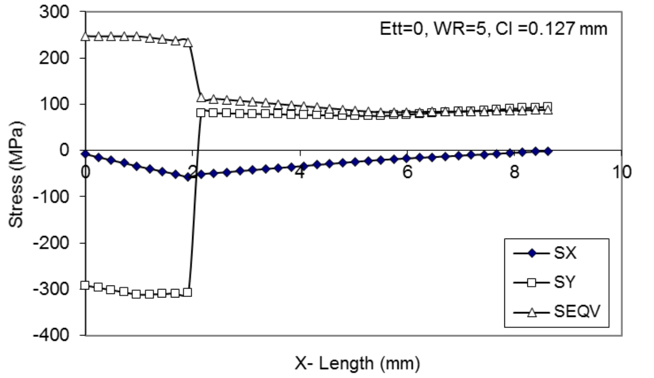

Figure 8 Radial, tangential and equivalent (Von Mises) stress distributions at x- axis ligament

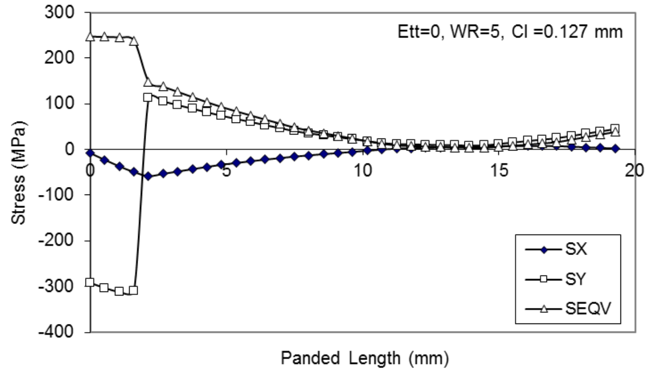

Figure 9 radial, tangential and equivalent (Von Mises) stress distributions at inclined 45° ligament

The stress distribution at the inner and outer circumference of the tube are represented on Figure 10 which shows that the stress start increasing from 0° until 10° then it continue at constant value until 72°, after that it drops down. The kind of symmetry is seen at 90° and it shown each 45° angles.

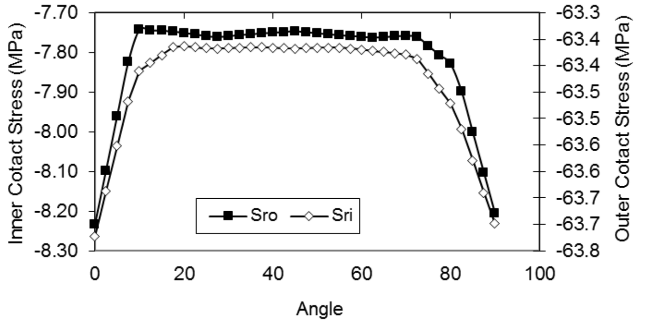

Figure 10 Radial stress distributions on the inner and outer surface of the tube

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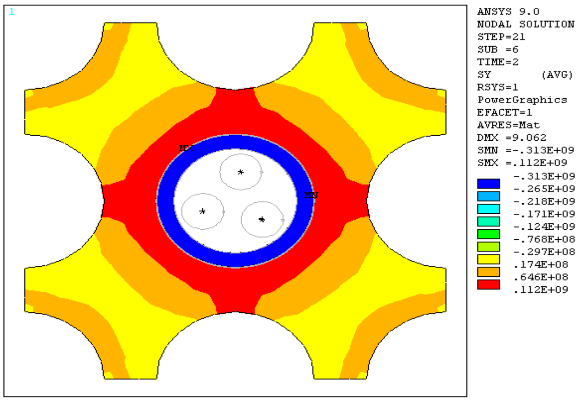


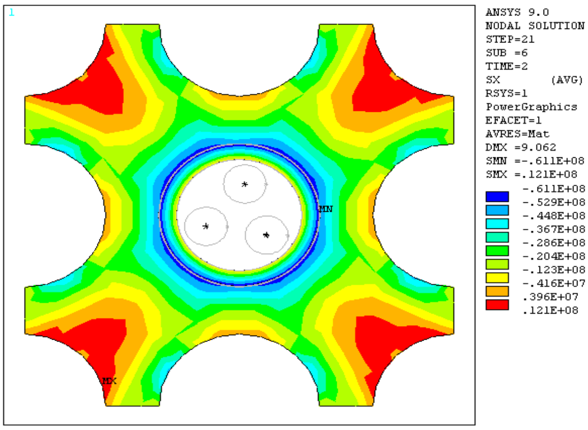
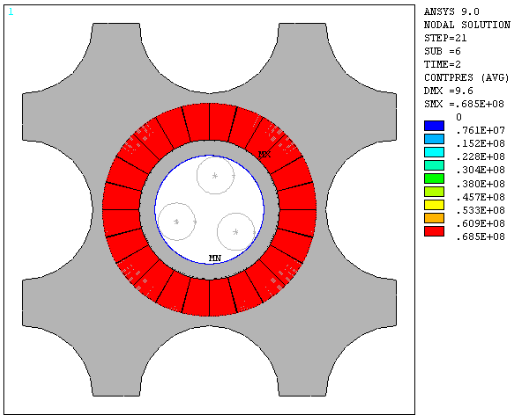


Figure 11 Contour of the radial (top) and tangential (bottom) stress for clearance 0.127 mm

The contour output results shows the radial and circumferential stress distribution at Figure 11(top) for the clearance of 0.127 mm and 5% wall reduction with elastic perfectly plastic material. It emphasize that the maximum radial stress at the contact region between the tube and the tubesheet. In addition the stresses distribution are not uniform around the tube because of the tubesheet holes that make some part of the tubesheet around the tube thick and other thin. The circumferential stresses at Figure 11(bottom) show that the tube has the maximum stress with approximately same amount of compressive stress. The circumferential stress distribution at the tubesheet is starting with tensile and move to compression outward.



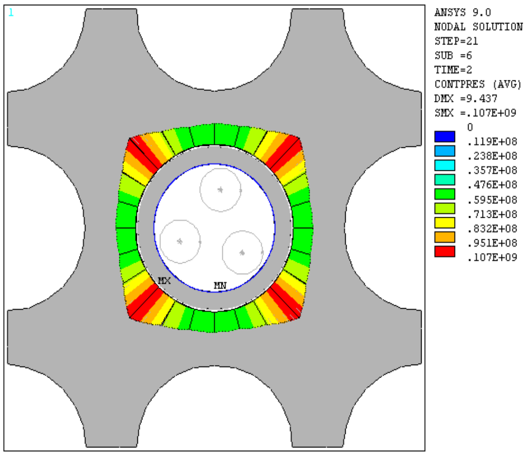


Figure 12 Residual contact stress for clearance 0.127 mm (top) and 0.3302 mm (bottom)

Figure 12 shows the contact stresses with clearance of 0.127 mm (top) and 0.3302 mm (bottom). The different between two figures is the uniformity of the stress. Small clearance means more uniform stress around the tube; while large clearance shows greater non-uniform stess.

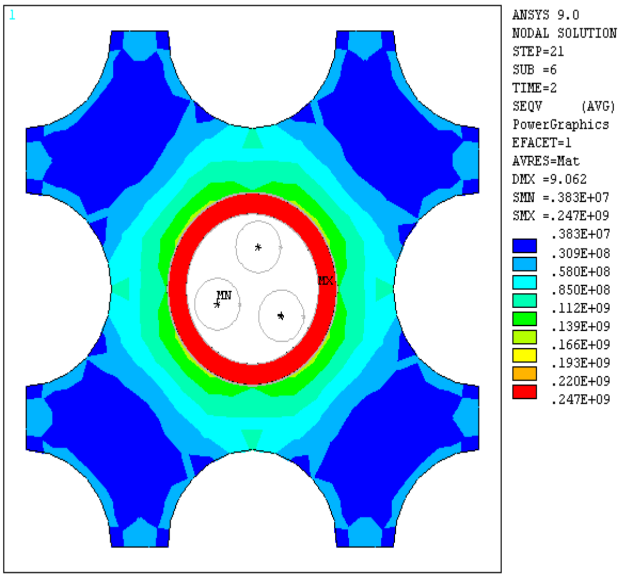


Figure 13 Equivalent Von Mises stress for clearance 0.127 mm E_{tt} =0 and wall reduction 5%

Figure 13 shows the equivalent Von Mises stress distribution at the tube and tubesheet. The tube stress is the same at the yielding stress equal to 247 MPa. This is applicable because the material is perfectly elastic plastic.

The geometric symmetry is shown at the 45o angle. This means the geometry repeats itself every 45o angle. The load symmetry resulting from the roller load does not exist, because the load moves around the inner surface at very fast rotational speed. At 360 rpm, the one roller revolution takes 1/6 second. Thus, the inner tube node will be loaded and unloaded each 1/16 second. This will make the assumption of uniform loading a valid assumption if the geometry is uniform. After releasing the load, we will have only the geometric symmetry.

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V. CONCLUSION

In finite element modeling, the planar model represents the realistic motion of the roller in tube to tubesheet expansion. The joint integrity is the main part of this study which is affected by the method and parameters of expansion. The most important parameter is the initial clearance, wall reduction and material properties. Results show that, for perfectly elastic plastic; the increase in initial clearance increases the amount of residual contact stress up to a certain threshold value after which stress starts to decrease. For strain hardening, the amount of residual contact stress is more than the perfectly elastic plastic material. In addition, the behavior is similar to the perfectly elastic plastic for the first region, but in the second region the decreasing slope of plastic material increases until stress increases for large strain hardening material. Symmetry of the process is at 45^o or 90^o angle, which follows the geometric not the loading symmetry, the loading symmetry is at 120^o.

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