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ANALYSIS OF TUBE-TUBESHEET JOINTS WITH GROOVES

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ABSTRACT

Hydraulic expansion of a grooved tube into a tubesheet is modeled as an elastic-plastic process. A shell model of the tube is suitably modified to include the effect of radial stress in the yield condition and the effect of the resistance from contact with the tubesheet. The tubesheet is modeled as an elastic-plastic foundation. The loading and unloading analysis of the tube subjected to expansion pressure on the inside surface and tubesheet resistance on the outside surface is performed to determine the residual state in the vicinity of the grooves. The parameters investigated are the residual holding pressure between the tube and the tubesheet and the stresses in the contact and noncontact regions of the tube.

NOMENCLATURE

x axial coordinate
z thickness coordinate; $z=0$ on reference surface
θ circumferential coordinate
R radius of reference surface
z_i, z_o z-coordinates of inner and outer surfaces of tube
$R_i=R+z_i$, $R_o=R+z_o$ inner and outer radii of tube
p_i, p_o pressure at inner and outer surfaces of tube
w radial displacement of a point in the tube
u_x axial displacement of a point in the tube
β_x rotation of normal
e_x, e_θ components of strain at reference surface
κ_x, κ_θ components of curvature change
Q_x transverse shear force per unit circumference
N_x, N_θ membrane stress resultants
M_x, M_θ bending moment per unit length
$\sigma_x, \sigma_\theta, \sigma_z$ components of stress
$\varepsilon_x, \varepsilon_\theta, \varepsilon_z$ components of strain
$\varepsilon_{xe}, \varepsilon_{\theta e}$ components of elastic strain
$\varepsilon_{xp}, \varepsilon_{\theta p}$ components of plastic strain
E Young's modulus
ν Poisson's ratio
ε_p equivalent plastic strain
s_1, s_2 reduced stresses
c_1, c_2 kinematic strain hardening parameters
Y isotropic strain hardening parameter
Y_0 initial yield stress
μ mixed hardening coefficient.
E_p plastic modulus
a inner radius of annulus model of tubesheet
b outer radius of annulus model of tubesheet
p_a pressure at hole in tubesheet
u_a radial displacement at hole in tubesheet

INTRODUCTION

The transition zone at the edge of a tube-tubesheet joint is a location of significant residual stress that is induced during the fabrication process of expanding the tube [1,2,3]. A major contributor to an undesirable axial tensile component of residual stress results from the meridional bending that is associated with differential radial expansion, variation of expansion loading, and variation of springback along the tube. Another location of differential radial expansion occurs in the vicinity of a groove [4] in the tubesheet hole, where the expansion process leaves a state of residual stress that varies along the tube. The objective of this paper is to determine the magnitude and algebraic sign of these stresses.

Little information on a tube-tubesheet joint with grooves is available in the literature. Manufacturer's standards [4] state that for pressures over 300 psi and/or temperatures in excess of 350°F, the tube holes for expanded joints for tubes of 5/8" O.D. and larger shall be machined with at least two grooves, each approximately 1/8" wide by 1/64" deep. No requirements on groove spacing are stated. Experimentally determined pushout forces for tube-tubesheet joints with grooves have been reported in [5]. Information on the residual stresses in the tubes and on the distribution of the holding pressure seems to be unavailable. This paper considers a theoretical approach to the determination of these stresses.

MATHEMATICAL MODEL OF TUBESHEET

The tubesheet is modeled as a continuous distribution of annular disks surrounding the tube. It acts as an elastic-plastic foundation of the tube. The inner and outer radii of a typical disk are a and b, respectively, where b is an effective outer radius [6] at which a boundary condition of zero radial stress is applied. Each disk is assumed to be expanded by a pressure p_a, acting at the inside radius. Each disk is subjected to elastic-plastic loading, followed by unloading. A linear hardening model of plasticity has been assumed. Since the longitudinal stress σ_x is assumed to be zero throughout the tubesheet, the calculations for the behavior of a typical disk can be carried out using a computer program which performs elastic-plastic analysis of flat plates. With the aid of a plastic shell analysis program named KshelPL, a calculation is carried out to find the radial expansion u_a of the inner edge as a function of the pressure p_a at the inner edge during elastic-plastic loading. This relation is expressed mathematically as

$$p_a = P(u_a)$$

This function is stored in a data base for later use. (This data base is the same as that which was used in [1,2] for the calculation of the residual stresses in the transition zone.) To make the data base applicable to many different tubesheets, the storage uses nondimensional variables. In the tube-tubesheet joint problem, p_a and u_a are taken to be functions of the coordinate x; however, the relation between them at a given value of x is that given above.

GROOVE GEOMETRY

Initially, there is a radial clearance c between the tube and the tubesheet. While the radial displacement of the tube during the initial expansion is less than c, the pressure p_a is zero. After contact between the tube and the tubesheet is made, the pressure p_a, which is also equal to p_o on the tube, begins to increase. The value of this contact pressure between the tube and the tubesheet then satisfies

$$p_a = p_o = P(w-c)$$

The grooves in the tubesheet were modeled by representing the radial clearance as step functions of the coordinate x, the step being equal to the groove depth. The present analysis was restricted to a study of a tubesheet with a repeated pattern of equally spaced grooves of equal size as shown in Figure 1. The model that was used in the analysis included one half-groove, a full ridge, and a second half-groove. Symmetry boundary conditions at the groove centers were used.

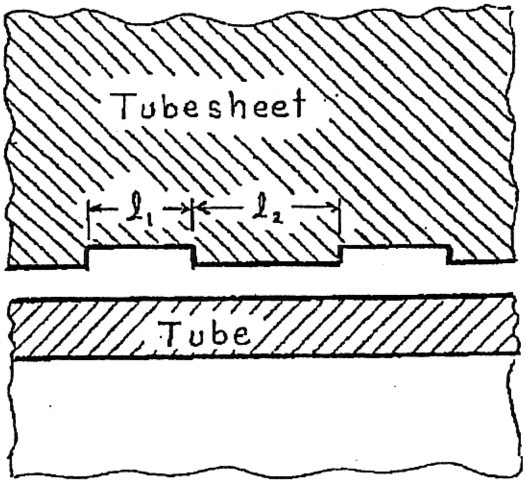

Figure 1. Repeated Pattern of Equally Spaced Grooves

MATHEMATICAL MODEL OF TUBE

A special shell theory has been developed for the study of the tube-tubesheet problem, since the usual plane stress assumption of shell theory does not apply to the expansion process, where pressure loading is on the order of the yield stress. The special theory uses the equilibrium equations taken from the theory of cylindrical shells. These equations relate the moments, forces, and surface loads as follows:

$$dQ_x/dx = N_\theta/R - (p_iR_i - p_oR_o)/R$$

$$dN_x/dx = 0$$

$$dM_x/dx = Q_x$$

The stress resultants for a cylindrical shell are

$$N_x = \int \sigma_x (1+z/R)\, dz \qquad N_\theta = \int \sigma_\theta\, dz$$

$$M_x = \int \sigma_x (1+z/R)\, z\, dz \qquad M_\theta = \int \sigma_\theta\, z\, dz$$

The equations of displacement and strain are

$$dw/dx = \beta_x$$

$$du_x/dx = e_x \qquad e_\theta = w/R$$

$$d\beta_x/dx = \kappa_x \qquad \kappa_\theta = 0$$

The distribution of strain through the thickness is given by

$$\varepsilon_x = (e_x + z\kappa_x) \qquad \varepsilon_\theta = e_\theta / (1+z/R)$$

In the special theory used here, the radial stress σ_z is assumed to vary linearly through the thickness of the tube, so that its distribution is given by

$$\sigma_z = -[p_i(z_o - z) + p_o(z - z_i)] / (z_o - z_i)$$

Following the usual practice of plasticity theory, the components of strain are decomposed into elastic and plastic portions

$$\varepsilon_x = \varepsilon_{xe} + \varepsilon_{xp} \qquad \varepsilon_\theta = \varepsilon_{\theta e} + \varepsilon_{\theta p}$$

where the elastic portions are related to the stress components by Hooke's law

$$\varepsilon_{xe} = (\sigma_x - \nu\sigma_\theta - \nu\sigma_z)/E \qquad \varepsilon_{\theta e} = (\sigma_\theta - \nu\sigma_x - \nu\sigma_z)/E$$

Plasticity is modeled by introducing the loading function (von Mises yield condition with strain hardening) given by

$$f(\sigma_x, \sigma_\theta, \sigma_z; c_1, c_2, Y) = (s_1^2 + s_2^2 - s_1 s_2)^{1/2} - Y$$

where

$$s_1 = \sigma_x - \sigma_z - c_1, \qquad s_2 = \sigma_\theta - \sigma_z - c_2$$

The yield condition is

$$f(\sigma_x, \sigma_\theta, \sigma_z; c_1, c_2, Y) = 0$$

The parameters c_1 and c_2 are the kinematic strain hardening parameters which indicate the translation of the yield surface. Initially, their values are zero. The parameter Y is the isotropic hardening parameter. It indicates the expansion of the yield surface. It is initially equal to the yield stress Y_0 in tension and compression. The increments of these strain-hardening parameters are related to the plastic strain increments by

$$dc_1 = \mu E_p (s_1/Y)\, d\varepsilon_p$$

$$dc_2 = \mu E_p (s_2/Y)\, d\varepsilon_p$$

$$dY = (1-\mu) E_p\, d\varepsilon_p$$

where

$$d\varepsilon_p = [4(d\varepsilon_{xp}^2 + d\varepsilon_{\theta p}^2 + d\varepsilon_{xp} d\varepsilon_{\theta p}) / 3]^{1/2}$$

is the equivalent plastic strain increment. The coefficient μ is equal to 0.0 if the strain hardening is isotropic and equal to 1.0 if it is kinematic. An intermediate value of μ may be used.

EXAMPLE

As an example, we consider a tube and tubesheet composed of materials having the same elastic-plastic material behavior. We have assumed ratios of material properties as $E/Y_0 = 1000$, $\nu = 0.3$, $E_p/E = 0.01$. These properties are representative of a number of materials. All calculated stresses are normalized with respect to the common yield stress Y_0. The following geometry, which is the same as that used in [2], is assumed:

Tube outside diameter = 1.000
Tube wall thickness = 0.0667
Radial clearance at ridge = 0.0100
Radial clearance at groove = 0.0256
Depth of groove = 0.0156
Width of groove = 0.125 (l_1 in Fig. 1)
Width of ridge = 0.250 (l_2 in Fig. 1)
Radius of hole = 0.510
Effective outer radius of annulus = 1.020

In order to expand the tube into the grooved hole, the pressure p_i (the hydraulic forming pressure) must be increased to some peak value $p_{i,max}$ and then be reduced to zero. This load-unload cycle has been simulated in the model, where we have, in this example, assumed that

$$p_{i,max} = 1.10\, Y_0$$

Distributions of stress components in the tube and contact pressure at the state of peak internal pressure and at the residual state have been calculated and plotted in Figures 3-8.

RESULTS

Figure 2 shows a plot of the hydraulic forming pressure as a function of radial displacement of the tube at the center of the groove. Each mark on the curve indicates one load step taken in the elastic-plastic process. There are four steep slope regions and three plateaus on the curve. The first steep portion and plateau represents the initial elastic-plastic expansion of the tube alone prior to its contacting the ridge. The second steep portion and plateau shows the additional resistance introduced when contact is made at the ridge, the plateau indicating yielding of the tubesheet at the ridge. The third steep portion indicates the stiffening effect due to contact between the tube and the valley of the groove. The presence of the third plateau indicates the tubesheet is also yielding in this region. The final steep portion indicates the unloading. Regarding the unloading curve, it may be noted that, even though further yielding is not indicated until the pressure is almost completely removed, the curve shows a slight nonlinear effect that is induced by the reduction in the size of the contact zone in the groove.

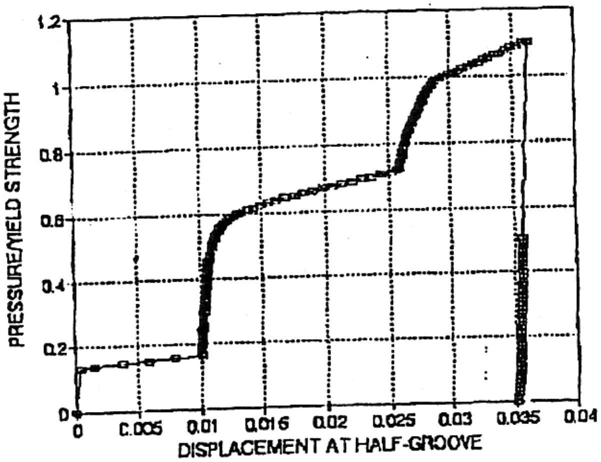

Figure 2. Hydraulic Forming Pressure Versus Radial Deflection at the Center of the Groove

Figures 3-6 show the distributions of residual stresses, normalized with respect to the initial yield stress, along the tube. The left and right ends of the abscissa represent the centers of two consecutive grooves, while the center of the abscissa is the center of the ridge. The largest residual tensile stress at the inside of the tube, where stress corrosion cracking might be a problem, is the axial stress at the center of the groove. The magnitude of this stress is about 32% of the initial yield stress. At the outside surface of the tube, the largest tensile residual stress is the hoop stress at the center of the groove, where it is 26% of the initial yield stress.

Figures 7 and 8 show the distribution of the residual contact pressure between the tube and tubesheet at the start of unloading and at the final residual state. Figure 7 also shows the applied hydraulic forming pressure on the inside of the tube (straight line at the level 1.1). The absence of regions of zero contact pressure indicates that, at the end of the load-up, the contact between the tube and tubesheet has been made over the whole length of the groove. Figure 8 shows that, after unloading, the tube and tubesheet have separated over the whole length of the groove.

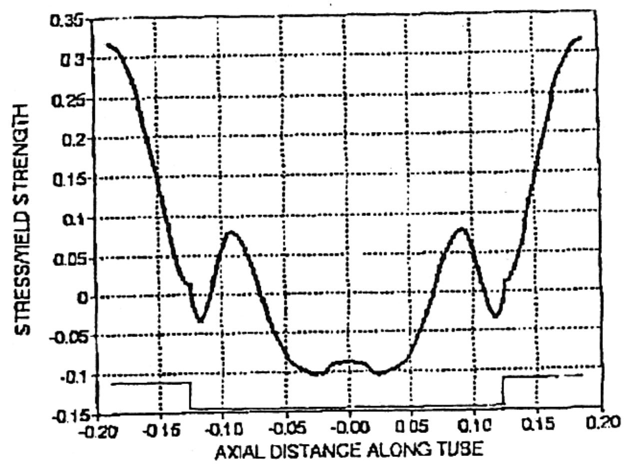

Figure 3. Residual Axial Stress at Inside Surface of Tube

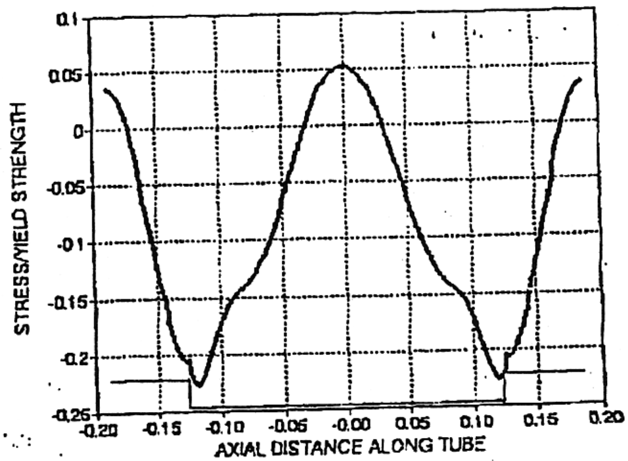

Figure 4. Residual Hoop Stress at Inside Surface of Tube

DISCUSSION

It is useful to compare the residual stresses in the groove with those in the transition zone at the edge of the tube-tubesheet joint. This information is available from reference [2], where a tube-tubesheet joint of the same dimensions and material properties was treated. In reference [2], the maximum residual stress at the inside surface of the transition zone was found to be 87% of the initial yield stress, compared with maximum of 32% found at the groove.

In reference [2], the optimum forming pressure was found to be about 78% of the initial yield stress. For higher values, little change in final contact and maximum residual stress was observed. It was observed in the present analysis that, when the same level of forming pressure was used for the tube-tubesheet joint with grooves, the tube just barely made contact with the tubesheet in the groove. This can be seen clearly from Figure 2 where the third steep portion begins. Therefore, in the present analysis, a much higher forming pressure (110% of yield stress) was used.

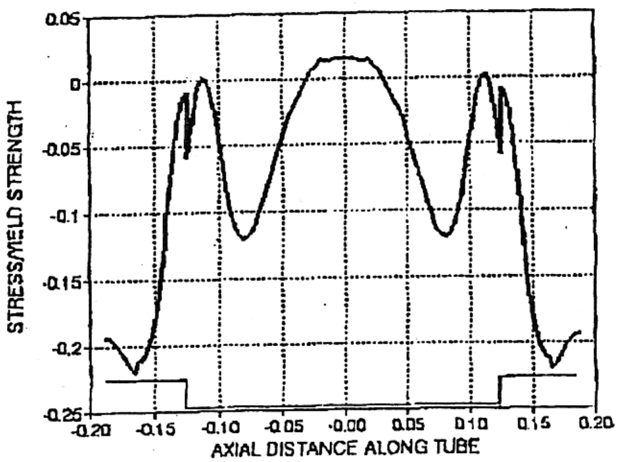

Figure 5. Residual Axial Stress at Outside Surface of Tube

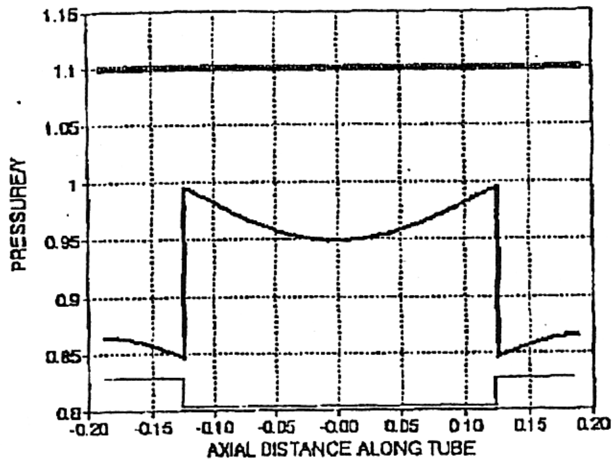

Figure 7. Contact Pressure (outside surface) and Forming Pressure (inside surface) at Condition of Peak Forming Pressure

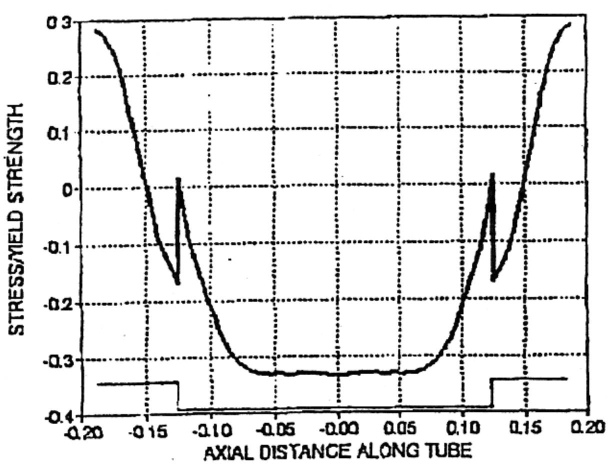

Figure 6. Residual Hoop Stress at Outside Surface of Tube

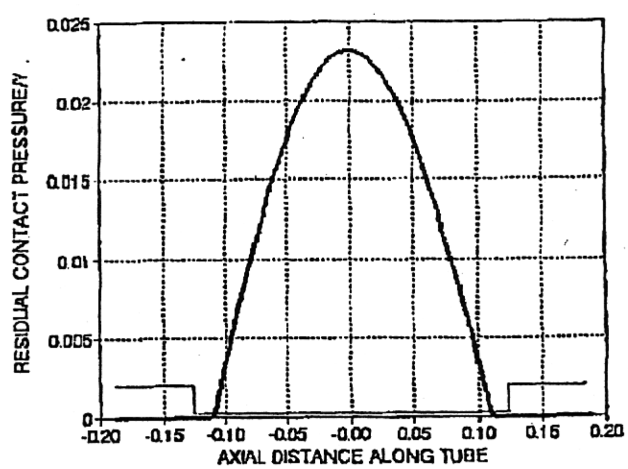

Figure 8. Distribution of Residual Contact Pressure

The plots of the stress distributions show the occurrence of discontinuity in the hoop stress. This is an expected result. The reason is that the transverse stress to the tube wall, σ_z, is discontinuous at these points, and satisfaction of the yield condition on both sides of the discontinuity requires discontinuity of some other stress component, namely the hoop stress.

Since the width of the groove is not large in comparison with the tube wall thickness, the use of a shell theory that neglects the effect of transverse shear stresses in the mechanical behavior may appear suspect. To check that this is not the case, we calculated the average transverse shear stress, i.e. transverse shear force divided by wall thickness, for the sample problem. Its value turned out to be an order of magnitude below the other calculated stresses; thus, the use of shell theory was judged acceptable.

Figure 9 shows the deflection curve of the outer surface of the tube at the maximum forming pressure, superposed on the cross section of the undeformed tubesheet and the outer surface of the tube. From the distribution of the contact pressure (Figure 7), it can be seen that there is complete contact between the tube and tubesheet, even at the re-entrant corners of the groove. This means that the deformed tubesheet touches the deflection curve of the tube, shown in Figure 9, at every point. This implies a discontinuous radial displacement in the tubesheet at the boundary of the ridge and the groove. Such a discontinuity represents an infinite shear strain at the corner of the groove. This is a defect in the mathematical model of the tubesheet used in this paper. The model assumes thin, independently-acting rings, with no shear stress between them (elastic-plastic foundation). On the other hand, the model of the tube assumes zero transverse shear strains, which makes it insensitive to the infinite strain in the tubesheet. For this reason, and since this paper is concerned with the stresses in the tube, and not in the tubesheet, it is expected that this defect does not alter our main results significantly.

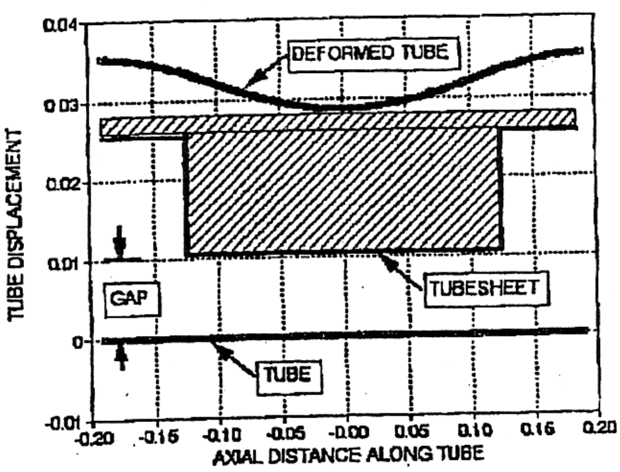

Figure 9. Deflection of deformed tube and cross sections of undeformed tube and tubesheet.

CONCLUSIONS

The results of this investigation lead to the following conclusions:

1. For the sample problem considered, the highest tensile residual stress at the inside surface of the tube was the axial stress at the center of the groove. Its value was about 32% of the yield stress, which is small in comparison with that found in the transition zone just outside the joint, where a residual stress of 87% of the yield stress is expected.

2. A joint with grooves requires a higher forming pressure than a joint without grooves. This is due to the additional effort required to bend the tube wall into the grooves. No attempt was made in this paper to optimize this value.

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