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Contact pressure in rolled tube–tubesheet joints

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The residual contact pressure in a rolled tube–tubesheet joint is calculated for several practical tube–tubesheet size and material selections. The analysis follows the elastic-plastic strain history during the tube expansion process. It accounts also for the initial gap size, reversed yielding, and material strain-hardening of both the tube and the tubesheet. The residual contact pressure is expressed as a function of the degree of expansion of the tube. The inside diameter of the tube after rolling (apparent wall reduction) is chosen as the measure of the expansion. Plots of the residual contact pressure versus the apparent wall reduction show that in all cases the contact pressure attains most of its value within 3–6% of the apparent wall reduction. Further rolling is of no benefit to the residual contact pressure.

1. Introduction

This paper is concerned with tubes that are rolled in a tubesheet of a heat exchanger. The objective of rolling is to obtain joints that are leak-tight and strong against pull-out. Both of the desired features depend on the residual contact pressure between the tube and the tubesheet. The objective of the paper is to estimate the degree of expansion of the tube that yields the highest residual contact pressure. The motivation for doing that is the assumption that the higher the residual contact pressure the better the joint. The dependence of leak-tightness and pull-out strength on the contact pressure is another problem that is not pursued in this paper.

The contribution of the present paper is the actual calculation of contact pressures for several sets of tube and tubesheet parameters. For each set, the contact pressures are plotted versus the apparent wall reduction, so that an optimum degree of expansion, which gives the maximum contact pressure, can be determined.

2. Background

The results presented in this paper were obtained with the use of an analytical model developed by Updike et al. [1] for the calculation of residual stresses in expanded tube–tubesheet joints. Previous papers on this subject go back to the 1930's. Four papers are in the 1943 volume of the ASME Transactions. The one that is most relevant here is by Grimison and Lee [2], which presents experimentally obtained contact pressures versus two measures of the degree of expansion. More recent papers are by Soler and Hu [3] and Jawad et al. [4].

3. Analytical model

Only a brief summary of the analytical model developed in [1] is given here. The tubesheet is treated as an elastic-plastic annular flat plate of inner radius a, outer radius b, and thickness h. The model of the tube

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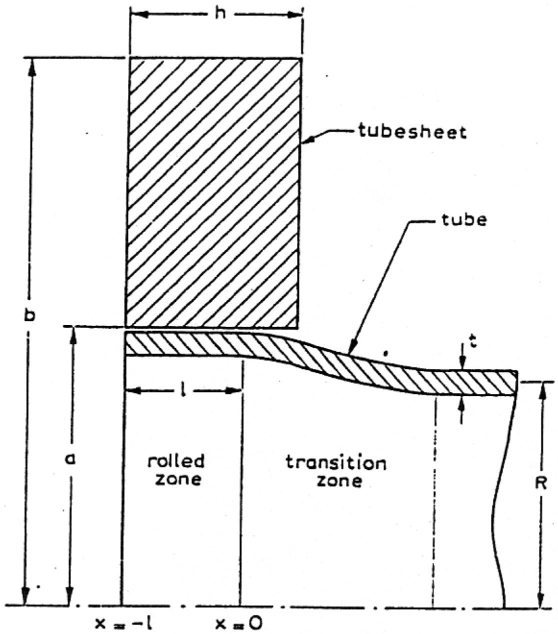

Fig. 1. Geometry of a rolled joint.

consists of the expanded (rolled) zone and the transition zone. This is shown in fig. 1.

The tubesheet is subjected during loading to a radially outward load of N per unit circumference at the inner edge while it is free at the outer edge. It is assumed that only in-plane deformation occurs and that the stress in the axial direction is zero. At the state of maximum radial expansion, the load N reaches the value $\hat{N}$ while the radial displacement at the edge of the hole is $\hat{u}$. An elastic-linearly hardening model with Young's modulus E', yield stress Y', and tangent modulus E'_{tan} is used. The behavior of the model of the tubesheet is presented in terms of the nondimensional variables a/b, E'_{tan}/E', $\hat{N}/Y'h$, and $E'\hat{u}/Y'a$. Thus, for given ratios of a/b and E'_{tan}/E', curves of $\hat{N}/Y'h$ versus $E'\hat{u}/Y'a$ are calculated by incremental elastic-plastic analysis. Points on these curves have been stored in a data base so that curves for other values of a/b and E'_{tan}/E' may be obtained by interpolation from the data base. Having the load-displacement in nondimensional form, the dimensional relationship between $\hat{N}$ and $\hat{u}$ is obtained by multiplying the nondimensional variables by the appropriate size parameters for the tubesheet.

During removal of the expander, the tubesheet springs back, so that the material unloads elastically. Letting $\overline{N}$ be the change in radial load and $\bar{u}$ be the change in radial displacement at the hole during unloading, the relation between $\bar{u}$ and $\overline{N}$ is calculated from the familiar thickwalled cylinder formula,

$$\bar{u} = (\overline{N}a/E'h)\left[(b^2 + a^2)/(b^2 - a^2) + \nu'\right],$$

from elasticity. The residual values are $\tilde{N}$ and $\tilde{u}$ given by

$$\tilde{N} = \hat{N} + \overline{N}, \qquad \tilde{u} = \hat{u} + \bar{u}.$$

For further details on the tubesheet analysis, see [1].

For the tube, the material constitutive relations are assumed to be elastic-linearly hardening. If the loading for the expansion process is simulated by uniform pressure on the inner surface, the rolled zone of the tube behaves as a cylindrical tube under internal pressure while the gap between the tube and tubesheet is still open. After the gap is closed, a contact pressure develops between the tube and the tubesheet, so that the tube is then under an internal pressure and an external pressure. Usually, the gap-closing deflection and the expansion into the tubesheet is sufficiently large to render the rolled zone of the tube fully plastic. The state of stress may then be calculated using the equilibrium equation together with the von Mises yield condition, and applying the boundary conditions. If unloading behavior of the rolled zone is assumed elastic, then the change in displacement at the outer surface is given by the usual thick-walled cylinder formula. For the details of the derivation of the complete solution of the expanded tube, see [1].

4. Selection of nondimensional parameters

All geometric lengths are reported as ratios of two lengths. The thickness-to-outside-diameter ratio, t/d_o, is used to characterize the tube geometry. All calculated stresses are normalized with respect to the yield stress of the tube, Y, which is defined as the stress at the point of discontinuity of the slope in an elastic, linearly strain hardening model of the stress–strain curve. The two slopes of the curve are E in the elastic range and E_{tan} in the elastic-plastic range. Two nondimensional parameters, E/Y and E_{tan}/E, characterize the stress–strain curve for the tube material.

The tubesheet geometry makes use of an annulus model previously used by Updike et al. [1] and Soler and Hu [3]. The annulus has an inner radius a, outer radius b, and thickness h. The annulus model assumes a traction-free boundary at the outer radius and depends on the tube spacing, or pitch. The parameters which characterize the tubesheet geometry are the ratios b/a and $h/2a$.

It is assumed that the hole diameter, $2a$, is approximately equal to the tube outside diameter, and it accounts for the initial gap between the tube and the hole by introducing the parameter c, the radial clearance, such that $2c = 2a - d_o$. The nondimensional parameter for clearance is taken to be $2c/d_o$. Values taken from a table in the TEMA [5] standards indicate that in the current heat-exchanger design practice this dimensionless clearance parameter lies between 0.00625 and 0.064.

5. Degree of expansion

Various measures can be used to represent the degree of expansion. For a theoretical analysis, it is more convenient to use a measure that is based on the inside diameter of the tube after rolling. Yokell [6] called it the apparent wall reduction. Grimison and Lee [2] define the apparent wall reduction as "...the percentage increase in the inside diameter of the tube over that calculated at first uniform contact of tube and seat...". Denoting the apparent wall reduction by m, then

$$m = [d_i' - (d_i + 2c)]/2t,$$

where d_i' is the measured, final inside diameter of the tube, d_i is the inside diameter of the unrolled tube, and $2c$ is the diametral gap between the tube and the hole. If expressed as a percentage, m is multiplied by 100.

6. Analysis of the reference case

The computer program KshelTZ, described in [1], was used to calculate the residual stresses and contact pressure for a reference case, which is presented in detail. The dimensionless parameters for the reference case are

$$t/d_o = 0.0667, \quad E/Y = 1000.0, \quad E_{tan}/E = 0.01,$$

$$b/a = 2.0, \quad h/2a = 1.0, \quad Y'/Y = 1.0, \quad l/h = 1.0.$$

$$E'/E = 1.0, \quad E_{tan}'/E' = 0.01, \quad 2c/d_o = 0.02,$$

This is representative of many tube–tubesheet joints encountered in engineering practice. Both the tube and the tubesheet are assumed to have the same stress–strain curve in this example. The E/Y ratio is representative of low strength steels and stainless steels.

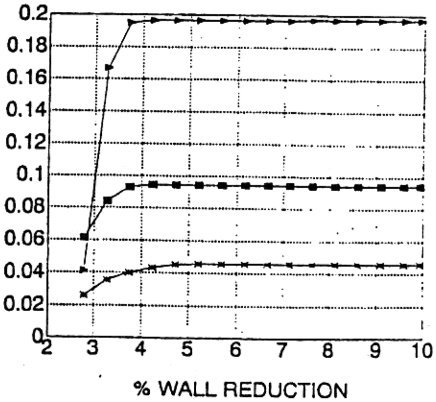

Fig. 2. Contact pressure/yield stress for different tube thicknesses (•) $t/d_o = 0.0333$, (□) 0.0667, (▽) 0.1333.

7. Effects of changing various parameters

This section presents results obtained by varying some of the tube and tubesheet parameters, one at a time. Attention is focused on how the residual contact pressure varies with the apparent wall reduction.

7.1. Effect of varying tube thickness

This is represented by the ratio of t/d_o. Does the residual contact pressure depend on the tube thickness? To answer this question, the analysis was repeated using the t/d_o ratios of 0.0333, 0.0667, and 0.1333. These values cover the range found in commercially available heat-exchanger tubes. All the other parameters were kept equal to the values given for the reference case. Figure 2 shows the nondimensional residual contact pressure as a function of the apparent wall reduction for the three t/d_0 ratios. The plots show the diminishing returns in contact pressure at increasing wall reduction. The optimum wall reduction appears to be about 4.5% for all three cases. At the optimum values of wall reduction, the residual contact pressures are $0.046Y$, $0.094Y$, and $0.196Y$. These values are roughly proportional to t/d_o.

7.2. Effect of outer radius of annulus model

This is represented by the ratio b/a, which is related to tube size and spacing in the tubesheet. The calculations were carried out by keeping all other parameters equal to those of the reference case and changing only b/a. The ratio b/a was varied over the levels 1.6, 2.0, and 3.0. Figure 3 shows the nondimensional residual contact pressure as a function of the apparent wall reduction for the three b/a ratios. The plots show the diminishing returns in contact pressure with increasing wall reduction. The optimum wall reduction appears to be about 4.0% for $b/a = 1.6$, about 4.5% for $b/a = 2.0$, and about 6% for $b/a = 3.0$. The nondimensional contact pressures are $0.046Y$, $0.094Y$, and $0.180Y$, respectively. The large variation of the contact pressure with the b/a ratio underscores the importance of having an accurate estimate of the relation between the outer radius b and the tube size and spacing. This problem has been addressed by Wang and Soler [7] and by Chaaban et al. [8].

7.3. *Effect of clearance*

The calculations were repeated using $2c/d_o = 0.00625$, 0.02, and 0.064 to cover the range which can be obtained from the TEMA Standards [5]. These values include classes 'C', 'R', and 'B' of TEMA heat exchangers. The lowest value is an extreme close fit, while the highest is an extreme loose fit. In these calculations, the other parameters remained equal to those of the reference case. For the closest fit ($2c/d_o = 0.00625$), the optimum wall reduction is about 3%, with the contact pressure being raised above that of the reference case to a value of $0.115Y$. For the loosest fit ($2c/d_o = 0.064$), the computations led to plastic strains in the tube lying outside the data base of the original elastic-plastic calculations available in the computer code KshelTZ; therefore, these results are less reliable, owing to the use of the required extrapolation from the data base. The calculations show that the optimum wall reduction for this case is 9% and that the contact pressure is $0.028Y$.

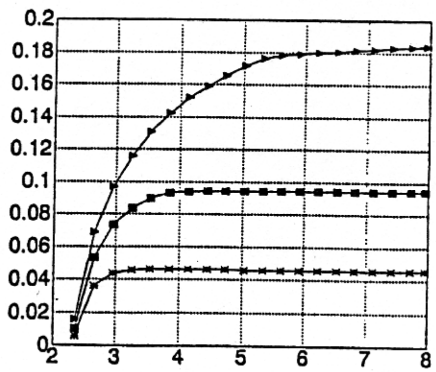

Fig. 3. Contact pressure/yield stress for different outer annulus radii (•) $t/d_o = 1.6$, (□) 2.0, (▽) 3.0.

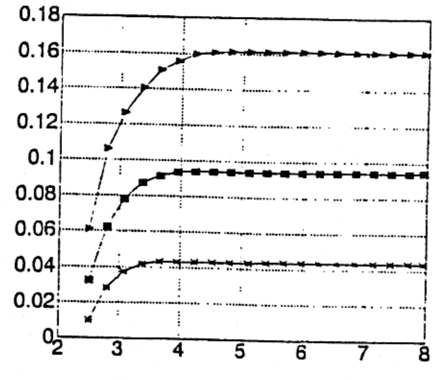

Fig. 4. Contact pressure/yield stress for different yield stresses (•) $Y'/Y = 0.8$, (□) 1.0, (▽) 1.25.

7.4. *Effect of tubesheet thickness*

The tubesheet thickness parameter $h/2a$ was varied over the levels 0.5, 1.0, and 2.0, while keeping all other input the same as that of the reference case. The results show that the tubesheet thickness has a negligible effect on the residual contact pressure, provided the roller is inserted to a depth equal to the tubesheet thickness. Of course, the necessary roller forces and the pullout force are approximately proportional to the tubesheet thickness.

7.5. *Effect of yield stress ratio*

What happens to the residual contact pressure if the tubesheet is stronger or weaker than the tube? If the ratio Y'/Y is less than 1.0, the tubesheet is weaker than the tube. If it is greater than 1.0, the tubesheet is stronger. Both cases were considered by performing calculations for $Y'/Y = 0.80$ and $Y'/Y = 1.25$ in addition to those for the reference case of equal yield strengths. For these calculations, all other parameters were kept at the values of the reference case.

Figure 4 shows how the residual contact pressure varies with the apparent wall reduction for the three yield stress ratios. The optimum wall reduction is about 4% for $Y'/Y = 0.80$, about 4.5% for $y'/Y = 1.0$, and about 5% for $Y'/Y = 1.25$. It is seen that the strength ratio has a large effect on the residual contact pressure. An increase of 25% in the yield stress of the tubesheet results in a 70% increase in residual contact pressure. On the other hand, a 20% decrease in tubesheet yield stress results in a 55% decrease in residual contact pressure.

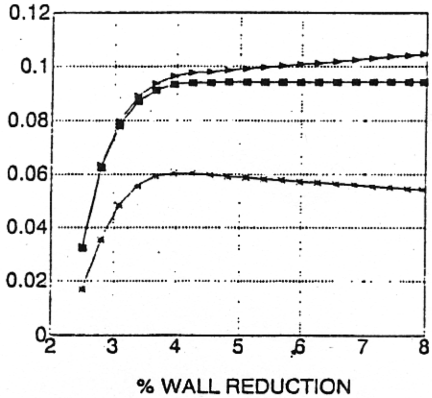

Fig. 5. Contact pressure/yield stress for different strain hardening (*) $E'_{tan}/E_{tan} = 0.5$, (□) 1.0, (▽) 2.0.

7.6. *Effect of strain-hardening*

Two questions were posed. First, what is the effect of increasing the strain-hardening slope of the stress–strain curve for the tube from $E_{tan}/E = 0.01$ to 0.02 while keeping all other parameters at the values of the reference case? Both the tube and the tubesheet were assumed to have the same yield stress. Only the hardening slopes were different. Figure 5 shows that an increase in the plastic slope of the tube relative to that of the tubesheet results in a significant decrease of the residual contact pressure. Doubling the strain-hardening slope decreased the optimum contact pressure by 35%. For the case of a tube that hardens faster than the tubesheet, the residual contact pressure decreases slightly with apparent wall reduction beyond the optimum point.

Second, what if the tubesheet stress–strain curve has the steeper plastic slope? The analysis was repeated with E'_{tan}/E' for the tubesheet changed from 0.01 of the reference case to $E'_{tan}/E' = 0.02$. The other curves in fig. 5 show that this effect is quite small. Doubling the strain-hardening slope for the tubesheet results in about a 5% increase in the residual contact pressure.

8. Discussion

The foregoing parameter study has shown that the residual contact pressure depends greatly on the selected geometry and material for the joint. Thicker tubes (greater t/d_o ratios) and greater tube spacing (b/a ratios) result in greater contact residual pressures, provided adequate wall reduction is used in fabricating the joint. In all cases studied, there is a value of apparent wall reduction, from about 3 to 6%, beyond which the improvement in contact pressure is negligible. This result is confirmed in a tube-expander manufacturer's catalog [9], where the question of the correct degree of expansion is addressed. As shown in fig. 6, reprinted from the Airetool catalog, page AA-2, the recommended apparent wall reduction is from 4 to 5%, which fits very well within the range shown in the other figures.

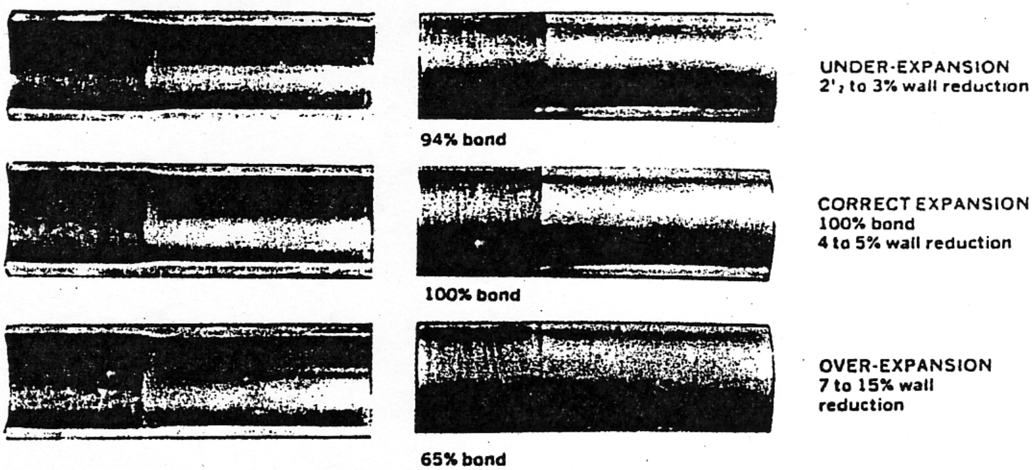

Fig. 6. Instructions for correct expansion in Airetool catalog, with as covering text: What is correct expansion of tubes? Correct expansion of tubes is the forming of a 100% bond between the tube and tube sheet, a result of reducing the tube wall by 4 to 5%. Anything less or more will result in under- or over-expansion.

9. Conclusions

The parametric study of this paper has investigated the effects of various parameters on the residual contact pressure. The major finding is that for any choice of parameters the maximum contact pressure is reached at 3–6 percent of the apparent wall reduction. This is also stated by Grimison and Lee [2] as a result obtained by experience and is also included as a recommendation in a recent Airetool catalog [9] (fig. 6). This result has now been confirmed by the mathematical model used in this paper.

Furthermore, it was found that the contact pressure:

- is roughly proportional to tube thickness;
- increases with clearance between tube and hole;
- increases with the radius of the annulus model;
- increases with yield stress of tubesheet;
- decreases with strain hardening of tubesheet.

Nomenclature

a Radius of hole in tubesheet,
b outer radius of annulus model,
h thickness of tubesheet,
l length of contact with roller,
d_o outside diameter of tube,
d_i inside diameter of tube,
t thickness of tube,
c initial radial clearance,
m apparent wall reduction,
E Young's modulus of tube,
Y yield stress of tube,
E_{tan} tangent modulus for plastic range of tube,
E' Young's modulus for tubesheet,
Y' yield stress of tubesheet,
E'_{tan} tangent modulus for plastic range of tubesheet.

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