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# Expanded, and Welded-and-Expanded Tube-to-Tubesheet Joints

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*This paper discusses some practical aspects of expanded and welded-and-expanded tube-to-tubesheet joints. It reviews elastic-plastic tube expanding theory, which it uses as the basis for setting pressures for uniform-pressure expanding and for estimating residual interfacial stress at the tube-hole interface. It addresses uniform-pressure-expanded tube joint strength and the problem of correlating of roller-expanded joint strength with wall reduction and rolling torque. It advocates two-stage expanding: 1) applying just sufficient pressure or torque to create firm tube-hole contact over substantially the tubesheet thickness; and 2) re-expanding at full pressure or torque. It points out the advantages of segregating heats of tubing and mapping the tube-heat locations. It recommends non-TEMA Standard (and non-HEI Power Plant Standard) grooves for grooved-hole joints made by roller-expanding, when the tubes are thin-walled, high-strength, low-elastic-modulus, and for joints made by uniform-pressure expanding [1, 2]. It states what to examine when considering grooves for small-diameter tubes. It reviews tube-to-tubesheet welding requirements and discusses welding before and after expanding. It suggests TEMA revise its standards to define strength and seal welds and urges the ASME Boiler and Pressure Vessel Code Committee (the Code Committee) to incorporate the TEMA definition in Section VIII of the ASME Boiler and Pressure Vessel Code (the Code) [3]. It makes recommendations for pressure and leak-testing. The work shows why welded-and-expanded joints should be full-strength expanded and why expansion length should equal the tubesheet thickness minus 1/8 in. (approximately 3 mm) rather than the lengths the TEMA and HEI Standards permit.*

## 1 Introduction

The tubes of most tubular exchangers are connected to the tubesheets by expanding-only or by welding and expanding. Expanded joints are standard for exchangers built to the TEMA Standards [1]. The HEI Standard for Power Plant Heat Exchangers [2] requires tubes to be expanded into the tubesheet for a depth of at least 2 in. (approximately 50 mm) or within 1/8 in. (approximately 3 mm) of the shell side tubesheet face, whichever is less. The HEI Closed Feedwater Heater Standards are silent on joint design [4]. Joint holding power and tightness of tubes expanded into bare holes are functions of the surface area in contact between the tube and hole, residual interfacial pressure at the tube-tubesheet interface, $P_{ci}$, produced by the expanding process, static coefficient of friction, $\phi$, and Poisson's constant, $\nu$. Elastic-plastic and finite element analyses of the expanding process have been published in the literature [5–11].

Machining grooves into the tubesheet holes before seating the tubes, increases joint holding power and tightness. The TEMA R and B Standards require grooving. The TEMA C standard requires grooving for 5/8 in. (approximately 16 mm) and larger diameter tubes for design pressures over 300 psi (2068 kPa) and or temperatures in excess of 350°F (approximately 200°C). The HEI Power Plant Standard states, "when required, each hole shall be grooved into two 1/8 in. wide × 1/64 in. deep (approximately 3 mm wide × 0.4 mm deep) rectangular or curved grooves...". However, it does not state the conditions under which grooves are required.

Tubes are frequently *seal-welded* to the front face of the tubesheet to stop fluid that may have leaked past the tube-hole contact surface from passing into the lower-pressure side. Although seal welds confer additional strength on the attachment, the incremental capacity of the joints to bear pressure and temperature-imposed loads is neglected in calculating joint load-carrying capacity. Nevertheless, the Code requires a qualified welding procedure to be used. Seal welding does not relieve the manufacturer who conforms with the TEMA Standards from the obligation to groove the holes.

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High metal temperatures combined with cyclical operation may relieve residual stress with consequent loss of joint integrity. For these conditions, it is good practice to *strength-weld* the tubes to the tubesheets. Almost all such strength welds are fillet welds either at the front face of the tubesheet or within the hole. (In a relatively small number of exchangers, the tube holes may be drilled to match the tube inside diameter and the tubes butt-welded to the tubesheet secondary face.)[1] The TEMA standards permit considering omitting grooving and expanding strength-welded joints. Some manufacturers have built tubular exchangers with the tubes strength-welded-only to the front face of the tubesheet with no prior or subsequent tube expanding. The forthcoming discussion shows why it is better practice to expand the tubes fully into the holes after welding.

When changes in temperature from that at which the tubes were expanded cause the tube to lose contact with the hole, the benefits of post-weld expanding are lost. The methods used to set expanded-joint temperature limits are also useful for examining this problem.

## 2 Expanded Tube-to-Tubesheet Joints

Most tube expanding is done by rolling. With the development of suitable equipment and techniques, expanding by applying hydraulic pressure directly in the region to be expanded has found a place in the industry [12, 13]. Bladder uniform pressure expanding has also been performed by pressurizing a bladder in the tube end. Axially compressing an elastomer in the tube end (rubber expanding) applies radial force uniformly at any point along the tube end. However, the force may vary axially (Fig. 1). Kinetic (explosive) expanding has also become a recognized successful process for expanding tubes into thick tubesheets. For some purposes, hybrid expanding is very desirable. This consists of hydroexpanding to firm contact followed by rolling at about half the usual torque [14].

**2.1 Elastic-Plastic Tube Expanding Theory.** The objective of tube expanding is to create a residual interfacial pressure between the tube and surrounding tube plate equivalent to shrinking the plate onto the tube. The residual pressure creates equal and opposite stresses at the outside of the tube and the inside of the hole. The following summary parallels work in the aforementioned literature. It is based upon these assumptions: 1) pressure is applied uniformly inside the tube holes; 2) the tube is expanded into a hole centered in an infinitely large plate—the effect of tube holes adjacent to the hole into which the tube is being expanded is neglected; 3) stress parallel with the tube axis is zero (plane stress model). Initially, these assumptions will be applied for equal tube and plate yield stresses ($\sigma_y = \sigma_t = \sigma_s$). The situations when plate yield stress is greater than tube yield stress ($\rho = \sigma_s/\sigma_t$ greater than 1) and plate yield stress is less than tube yield stress ($\rho = \sigma_s/\sigma_t$ less than 1) will then be examined.

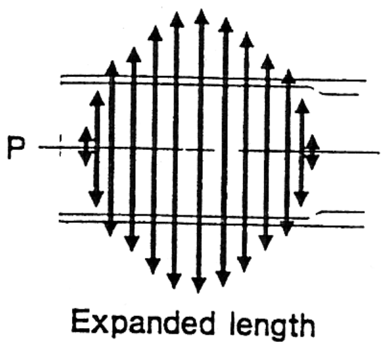


Fig. 1 Variation of radial pressure along expanded length of tube in elastomeric expanding

The assumption that expanding pressure is applied uniformly does not truly apply for tube rolling. Uniform-pressure expanding does not create localized stresses and the Poisson effect causes very slight reduction in tube-end length and about 1 to 3 percent wall thinning.[2] In contrast, rolling creates high localized stresses that affect the tube metal structure and springback [15]. These stresses cause greater wall thinning (3 to 12 percent, depending upon the tube thickness, diameter and metal properties) and sufficient tube-end extrusion to exceed the initial Poisson-effect tube-end shortening. Neglecting the effects of the tubesheet holes is suitable for approximations. In practice, they must be considered.

Figure 2 illustrates schematically uniform-pressure expanding. As pressure is applied inside the tube end, the tube deforms elastically. Increasing the pressure increases the deformation until the tube makes contact with the hole. At contact the stress

[1] See S. Yokell, *A Working Guide to Shell-and-Tube Heat Exchangers*, McGraw-Hill, New York, 1990, pp. 197–198.

[2] Analysis of and equations for calculating wall reduction and tube-end shortening in the elastic and plastic ranges were developed by Jawad et al. [8].

## Nomenclature

- $a$ = inside tube radius, in. (mm)
- $b$ = outside tube radius, in. (mm)
- $c$ = radius to interface between elastic and plastic zones, in. (mm)
- $d$ = tube outside diameter, in. (mm)
- $d'$ = tube hole diameter, in. (mm)
- $h$ = depth of groove in grooved tube holes, in. (mm)
- $L$ = joint load, lb (N)
- $L_c$ = push-out joint load, lb (N)
- $L_e$ = length of tube expanded into tubesheet, in. (mm)
- $L_t$ = pullout joint load, lb (N)
- $P$ = pressure, psi (Pa)
- $P_e$ = expanding pressure, psi (Pa)
- $P_{emax}$ = maximum expanding pressure that can be applied without axially extruding the tube, psi (Pa)
- $P_o$ = residual interfacial pressure at interface of tube and hole after expanding pressure has been released, psi (Pa)
- $p'$ = tube hole drilling pitch, in. (mm)
- $r$ = radius from tube center to any point
- $t$ = tube wall thickness, in. (mm)
- $W$ = groove width, in. (mm)
- $\beta$ = constant defined in text, dimensionless
- $\gamma$ = constant defined in text, dimensionless
- $\eta$ = ligament efficiency = $(p' - d')/d'$
- $\phi$ = static coefficient of friction
- $\rho$ = ratio of tubesheet to tube yield stress = $\sigma_s/\sigma_t$
- $\rho_L$ = limiting ratio of tubesheet to tube yield stress = $(\sigma_s/\sigma_t)_L$ at which expanding pressure $P_{emax}$ may be applied without pressure at outside of tube exceeding plate plastic limit of $(2/\sqrt{3})\sigma_s$
- $\sigma_{rad}$ = radial plastic stress
- $\sigma_s$ = tubesheet yield stress at manufacturing temperature, psi (Pa)
- $\sigma_t$ = tube yield stress at manufacturing temperature, psi (Pa)
- $\sigma_{tan}$ = tangential plastic stress, psi (kPa)
- $\sigma_y$ = yield stress, psi (kPa)
- $\nu$ = Poisson's constant

Tube in place before expanding
Assumed Temperature 70 °F
(approximately 29 °C)

Pressure applied. Tube and hole just in contact. Interfacial pressure $\sigma = 0$. Deflection may be fully elastic, fully plastic or plastic in inner tube shells and elastic in outer ones.

Tube and hole position at application of maximum expanding pressure. Interfacial pressure = equilibrium interfacial stress $\sigma$ = the lower of $(2/\sqrt{3})\sigma_t[\ln(b/a)$ or $(2/\sqrt{3})\sigma_s$.

Relaxed position after expanding pressure is released.

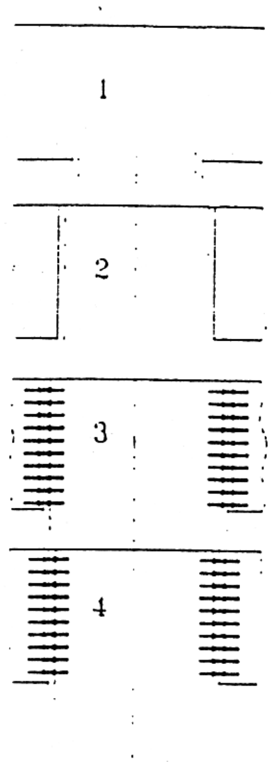


Fig. 2 Making an expanded tube-to-tubesheet joint

and deflection of a thick, high-yield-stress tube may be fully elastic; a thin, low-yield-stress tube may be fully plastic; between these extremes, there may be a plastic-elastic interface in the tube.

The requirements for creating the desired interfacial pressure are: 1) tube deformation must be fully plastic; 2) the surrounding plate must deflect under the pressure that the tube applies; and 3) upon pressure release, free plate recovery must be greater than free tube recovery. Expanding pressure beyond that required for tube/hole contact must be applied to meet these conditions. In the model used in this discussion, axial plastic stress is taken as zero. The plastic radial stress varies with plastic tangential (hoop) stress as shown in Eq. (1) [16]

$$\sigma_{rad}^2 - \sigma_{rad}\sigma_{tan} + \sigma_{tan}^2 = \sigma_y^2 \quad (1)$$

This is an equation of an ellipse.

Goodier and Schoessow [5] used an eccentric ellipse angle $\theta$ as a parameter by which to express the stresses as

$$\sigma_{rad} = \frac{2}{\sqrt{3}}\sigma_y \sin\left(\theta - \frac{\pi}{6}\right); \; \sigma_{tan} = \frac{2}{\sqrt{3}}\sigma_y \sin\left(\theta + \frac{\pi}{6}\right) \quad (2)$$

Both $\sigma_{rad}$ and $\sigma_{tan}$ are limited to lie between $\pm 2\sigma_y/\sqrt{3}$. This is the frequently referred to plastic limit or limiting stress.

Figure 3 shows graphically the radial and tangential stresses as multiples of yield stress when pressure equal to the plastic limit is applied. In this paper, $P_{emax}$ is the expanding pressure equal to the plastic limit of $(2/\sqrt{3})\sigma_y$.

Figure 4 is the part of the ellipse in which plastic radial stress varies from 0 to the limit of $(2/\sqrt{3})\sigma_y$. This is the portion of the ellipse from which the radial stresses with which we are concerned can be determined.

In the model used here, pressure greater than $P_{emax}$ cannot be applied because the tube metal simply extrudes. Under internal pressure $P_{emax}$, the stress state varies from plastic to elastic. When pressure $P_{emax}$ is applied, the radius of the interface between the plastic and elastic zones, $c$, is at a position 1.75 × the inside tube radius ($c = 1.75a$).

The variation of stress from inside of the tube to the plastic/

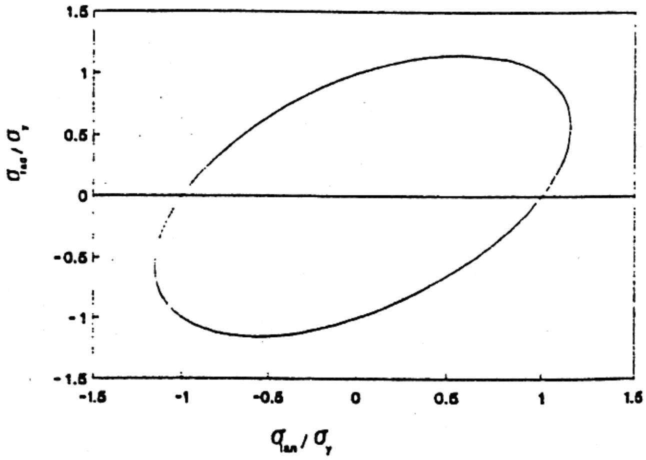


Fig. 3 $\sigma_{rad}/\sigma_y$ versus $\sigma_{tan}/\sigma_y$ during application of pressure = $2\sigma_y/\sqrt{3}$

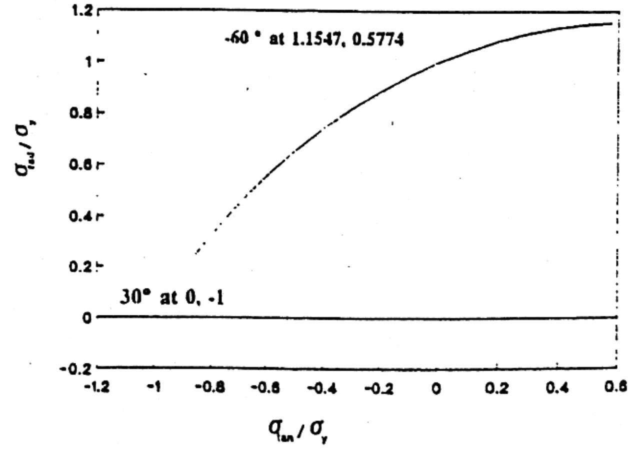


Fig. 4 $\sigma_{rad}/\sigma_y$ versus $\sigma_{tan}/\sigma_y$ over the range 0 to $2/\sqrt{3}$ during application of pressure = $2\sigma_y/\sqrt{3}$

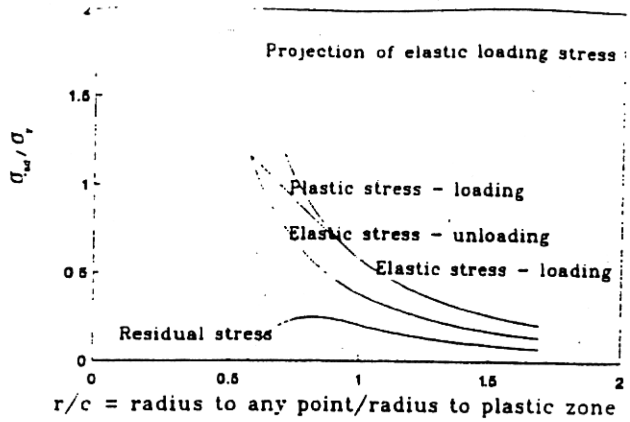


Fig. 5 Variation of $\sigma_{rad}/\sigma_y$ with $r/c$

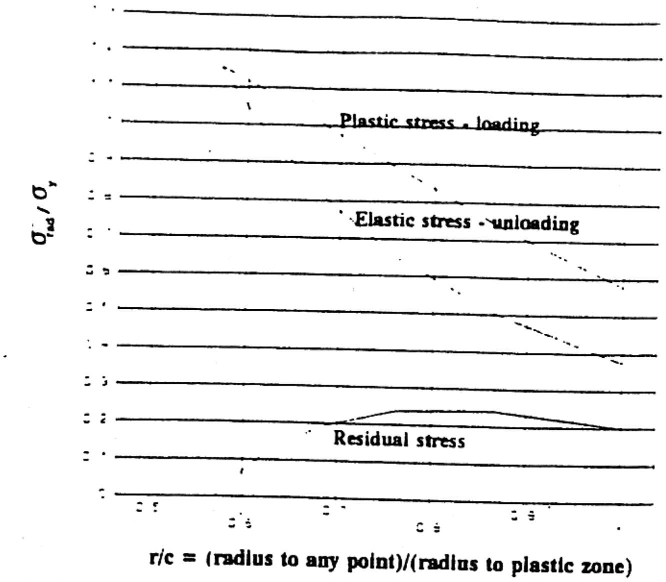


Fig. 6 $\sigma_{rad}/\sigma_y$ versus $r/c$ between inside tube radius $r = a$ and radius $r = c$

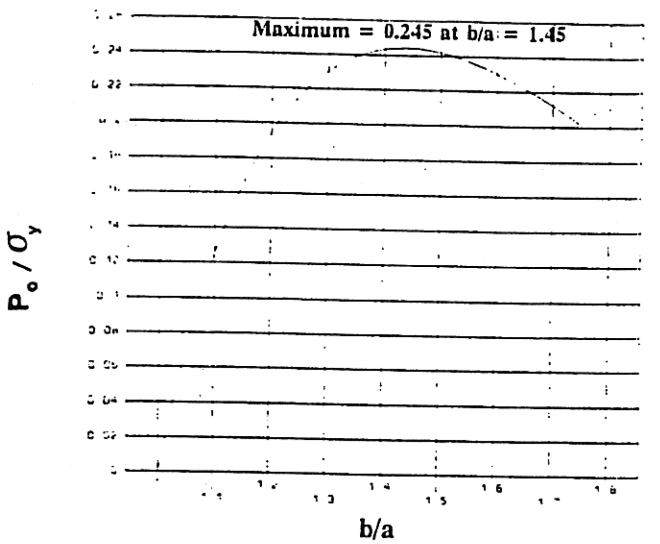


Fig. 7 $P_o/\sigma_y$ versus $b/a$ when $P_{exp} = (2/\sqrt{3})\sigma_y$

elastic interface determined, by the method of Goodier and Schoessow [5], is shown in Fig. 5.

Loading begins at the point marked "plastic stress-loading." The tube/tubesheet structure is stressed plastically from radius $a$ to radius $c$.

This is shown in Fig. 6 as the ratio $r/c$, where $r$ is the radius to any point. At radius $a$, $r/c$ is equal to $a/1.75a = 0.57$. At radius $c$, $r/c = 1$. The stress beyond $r/c = 1$ is elastic. In the figure, the continuation of the elastic loading stress is projected from the point where $r/c = 1$.

When the pressure is released, unloading is for practical purposes elastic. The unloading stress can be determined by superposing an equal elastic radial stress at the tube/hole interface. This stress is approximate as shown in Eq. (3) [17]

$$\sigma = P\frac{a^2}{r^2} \tag{3}$$

Substituting $P_{emax}/\sigma_y$ in Eq. (3) produces the curve marked, "elastic stress-unloading" of Fig. 5. The residual stress or pressure ($P_o$) at the tube/tubesheet interface is the difference between the loading and unloading stresses. This is the bottommost curve of Fig. 5. Figure 6 is an enlargement of the curves of Fig. 5 from radius $a$, the inside of the tube, to radius $c$, the plastic/elastic radius ($r/c = 0.57$ to $r/c = 1$).

The residual interfacial pressure $P_o$ between the tube and hole is at the abscissa where radius $r$ is equal to the outside tube radius $b$ ($r/c = r/b$).

Since the radius of the plastic/elastic interface is 1.75 × the inside tube radius ($c = 1.75a$), at radius $b$, where $r/c$ is set at $b/c$, $b/a$ is $1.75r/c$. This permits calculating residual pressures for various ratios of tube outside to inside radii.

Figure 7 shows how the residual stress between the tube and tubesheet varies with the ratio of outside tube radius to inside tube radius ($b/a$) when $P_{emax}$ is applied.

**2.2 Pressures for Uniform-Pressure Expanding.** Uniform-pressure expanding theory is useful for estimating pressures to use in expanding the tubes into the tubesheets. The following considers three situations: 1) the tubes and plate have equal yield stresses ($\sigma_t = \sigma_s$); 2) the plate yield stress is greater than tube yield stress ($\sigma_t < \sigma_s$); and 3) the tube yield stress is greater than the plate yield stress ($\sigma_t > \sigma_s$). In this discussion, it is assumed that each uniform-pressure application is over the length of the tube to be embedded in the tubesheet.

*2.2.1 Equal Tube and Plate Yield Stresses $\sigma_t = \sigma_s = \sigma_y$.* When $\sigma_t = \sigma_s$, after tube-hole contact, the tube/tubesheet structure is essentially a smaller-diameter hole in a plate. In a plate without adjacent holes, the greatest residual pressure is achieved by applying pressure $P_{emax}$ ($= 2/\sqrt{3})\sigma_y$). However, if the interface between the plastic and elastic zones in a real tubesheet lies beyond tangency with adjacent drilled holes, they may be plastically deformed out of round. In addition to the difficulty of expanding tubes into out-of-round holes, already-expanded-in tubes may be loosened. In extreme cases, leaks of adjacent rolled-in joints may progress from tube joint to tube joint. This is the phenomenon that shop workers call, "a dog chasing its tail."

Two-stage expansion is advisable to avoid these troubles: 1) expand first at a pressure or torque that will just set the tubes (often called contact expanding); and 2) complete the expansion at the pressure at which the plastic/elastic radius will just be tangent to the holes.

Contact Expanding Pressure. The pressure required for contact expanding can be calculated from the pressure drop across the fully plastic tube. This is the pressure at which yield will just begin in the tubesheet. Neglecting the clearance between the tube and hole, for this condition, the pressure on the outside of the perfectly plastic tube will be $\sigma_y/\sqrt{3}$ (referred to as the elastic limit). The required pressure increase through the tube is $(2\sqrt{3})\sigma_y \ln(b/a)$. Therefore, to just set the tubes, $P_r$ must be just greater than $(\sigma_y/\sqrt{3})]1 + 2\ln(b/a)]$.

For the usual drilling pitch of 1 1/4 tube diameters ($= 2.5b$), the expanding pressure should be limited to the pressure at which $c = 1.5b$ (the tube outside radius plus the ligament width). The required pressure reduction from a pressure equal to $(2/\sqrt{3})\sigma_y$ may be determined by correlating $c = 1.5b$ with tube diameter and thickness. For consistency with the Goodier and Schoessow work [5], the following uses the ratio of inside to outside tube radii, $a/b$ as the diameter-to-thickness parameter.

At $c$ equal $1.5b$, $r/c = r/1.5b = 0.67b$. Setting $r$ equal to the inside tube diameter $a$, $a/b = 0.67$. This means that the maximum values of $r/c$ (now $= a/b$) in Fig. 6 at which expanding pressure $P_{emax} = 2\sigma_y/\sqrt{2}$ can be applied is 0.67. For thinner tubes (diameter-to-thickness ratio more than about 6), the expanding pressure must be reduced. The fact that plastic portions of the curves of Figs. 5 and 6 are approximately straight lines, with slopes of about $-1.384$, can be used to set the appropriate expanding pressure [limited to $(2/\sqrt{3})\sigma_y$]. This is shown in Eq. (4)

$$P_e = \sigma_y\left(1.945 - 1.384\frac{a}{b}\right) \tag{4}$$

To illustrate, consider the cases of 1-in. (approximately 25-mm) o.d. tubes, with wall thickness of 0.029 in. and 0.109 in. (approximately 0.74 mm and 2.77 mm), diameter-to-thickness ratios of 34.5 and 9.1, respectively. For the thicker-walled tube, the appropriate expanding pressure is about 86.3 percent of the yield stress. For the thinner-walled one, the expanding pressure should be about 64.5 percent of the yield stress.

The residual pressure calculated on the basis of an infinite plate with a pressurized hole will be reduced. However, the result will be production of joints with the highest practically attainable residual pressure at the tube/tubesheet interface. Figure 8 shows how the residual stress varies with $b/a$ when $P_e$ is just equal to the tube yield stress.

The elastic-plastic analysis of Kasraie and others [6] offers a closed-form equation for directly calculating $P_o$. It is paraphrased here in Eq. (4).

$$P_o = P_e\left[1 - \left(\frac{a}{b}\right)^2\right] - \frac{2}{\sqrt{3}}\sigma_y\left[\ln\frac{b}{a}\right] \tag{5}$$

*2.2.2 Plate Yield Stress Higher Than Tube Yield Stress* ($\sigma_s > \sigma_t$). When the plate yield stress is higher than the tube yield stress, the radius of the plastic/elastic interface for any expanding pressure moves inward. In the unlikely circumstance that $\sigma_s = 2\sigma_t$ the plate will always remain fully elastic when the tube is fully plastic. This is because when $P_{emax}$ [$= (2/\sqrt{3})\sigma_t$] is applied inside the tube, the pressure on the outside of the tube is $(2/\sqrt{3})\sigma_t[1 - \ln(b/a)]$, and for a tube of no thickness, $b/a = 1$. Ln(1) being 0, the maximum pressure on the outside of the tube is $(2/\sqrt{3})\sigma_t$, which is equal to $(1/\sqrt{3})\sigma_s$.

If we now consider the hole under the pressure applied by the tube and again set the plastic radius to the point of tangency to the next hole, the ratio of the inside hole radius to radius $c$ is $b/1.5b = 0.67$. The expanding pressure that will produce this result is in proportion to $t$ and $\sigma_t$, and $1.5b$ and $\sigma_s$.

$$P_e = \sigma_t\frac{t + b\left(\dfrac{\sigma_s}{\sigma_t}\right)}{t+b}\left(1.945 - 1.384\frac{a}{b}\right) \tag{6}$$

Defining a yield-stress ratio $\rho = \sigma_s/\sigma_t$, Eq. (6) becomes

$$P_e = \sigma_t\frac{1+b\rho}{t+b}\left(1.945 - 1.384\frac{a}{b}\right) \tag{6a}$$

$P_e$ is limited to $P_{emax}$, based upon the tube yield stress.

*2.2.3 Plate Yield Stress Lower Than Tube Yield Stress* ($\sigma_s < \sigma_t$). This discussion follows the Goodier and Schoessow [5] practice of expressing radial stresses and pressures as multiples of yield stress.

In the single-hole model, when the plate yield stress is lower

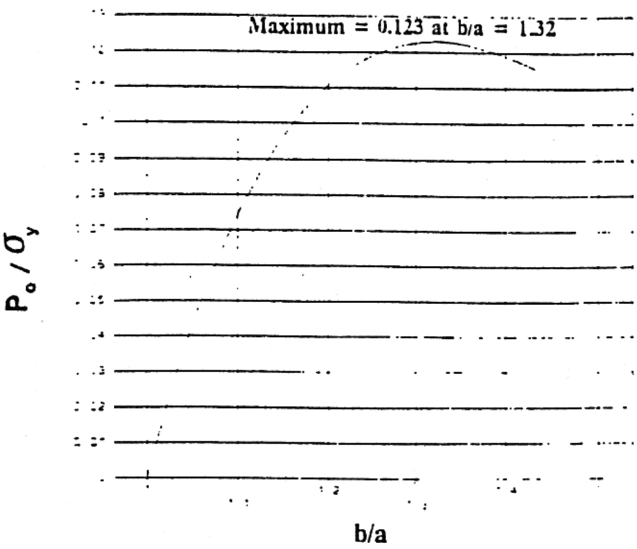


**Fig. 8** $P_o/\sigma_y$ versus $b/a$ when $P_{exp} = \sigma_y$

than the tube yield stress, $P_e$ must be reduced to a level at which the pressure at the outside of the tube (= the pressure on the hole) equals, but does not exceed, the plate plastic limit $(2/\sqrt{3})\sigma_s$. After the onset of plasticity, the plastic radial stress $\sigma_{rad}$ at $a$ is equal to the expanding pressure $P_e$. $P_e$ relative to $\sigma_t$ is calculated for various ratios of $\rho$ over the range from $r/b$ ($= a/b$) of 1 (a tube of no thickness) to the $r/b$ ($= a/b$) at which $P_e = P_{emax}$; that is, $P_e = (2/\sqrt{3})\sigma_t$. Figure 9 is the curve of the variation of $P_e/(2/\sqrt{3})\sigma_t$ versus $r/b$ for $\rho$ of approximately 0.6. Figure 10 is the curve for $\rho = 0.0$. The curves for all ratios of $\rho$ terminate at the ordinate $2/\sqrt{3}$. Note that when $\rho = 0$, the curve terminates at $r/b = 0.335$.

*2.2.4 The Limit Line.* The values of $a/b$ coincident with ordinates of $2/\sqrt{3}$ for each ratio $\rho$ are the limiting values of $\rho$, called here $\rho_L$, at which $P_{emax} = (2/\sqrt{3})\sigma_t$ applied in the tube produces the pressure on the hole of $(2/\sqrt{3})\sigma_s$. Figure 11 is the curve of $a/b$ versus $\rho_L$. This is the limit line introduced in an earlier paper [9]. For any value of $a/b$ at which $\rho$ is less than $\rho_L$, $P_e$ must be reduced in proportion to $\rho/\rho_L$ as shown in Eq. (7)

$$P_e = \sigma_t\left(1.945 - 1.384\frac{a}{b}\right)\frac{\rho}{\rho_L} \tag{7}$$

**2.3 Maximum Tube Thickness for Expanding.** The curve of $\rho = 0$ (Fig. 10) terminates at the point $\sigma_{rad}/\sigma_t = 2/\sqrt{3}$, $a/b = 0.335$. This means that when $P_{emax}$ is applied in the tube there is zero radial plastic stress (or pressure) at $b$ for $a/b = 0.335$. The material beyond that point will always remain elastic. The limiting thickness for tube expanding in terms of $d/t$ is then approximately 3.

The $d/t$ limit may be established alternatively by setting the pressure drop through the fully plastic tube to $P_{emax}$, thereby producing zero pressure at $b$. For this condition $(2/\sqrt{3})\sigma_t - 0 = (2/\sqrt{3})\sigma_t\ln(b/a)$ or $\ln(b/a) = 1$, from which $b/a = 2.719$, which corresponds with $d/t$ of 3.16. This is in substantial agreement with the graphic method.

**2.4 Strength of Expanded-Only Joints.** Tube joint strength is measured in terms of the force required to pull or push the tube out of the hole into which it has been expanded. Because of the Poisson effect, pulling the tube reduces slightly the tube diameter and pushing it increases it. In either case, the hole deflects elastically to meet the tube deflection. Goodier and Schoessow [5] analyzed the pullout and push-out loads, taking into consideration the length of tube embedded in the tubesheet, the plate elastic modulus, Poisson's constant for

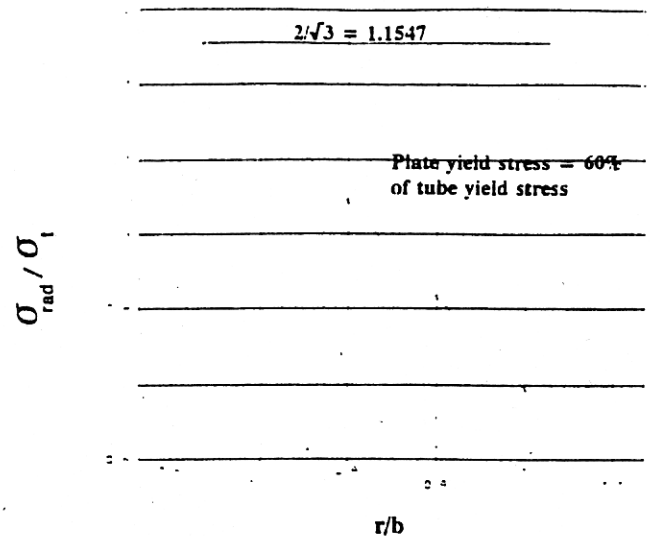


Fig. 9 Plastic $\sigma_{rad}/\sigma_t$ versus $a/b$ for $\rho$ = 0.6

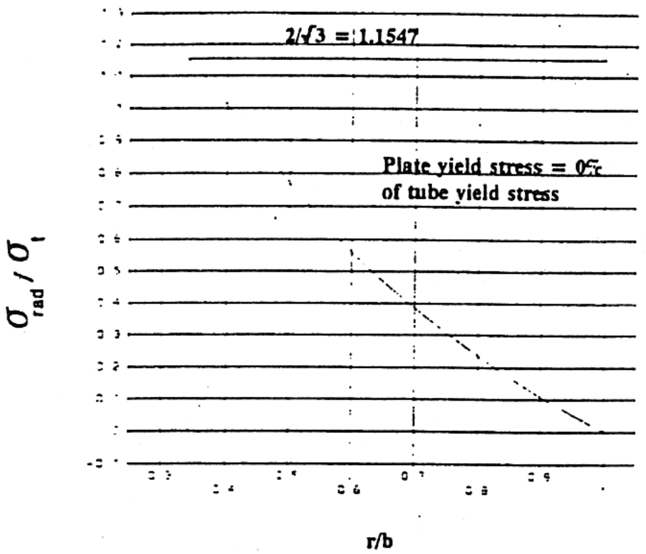


Fig. 10 Plastic $\sigma_{rad}/\sigma_t$ versus $a/b$ for $\rho$ = 0.0

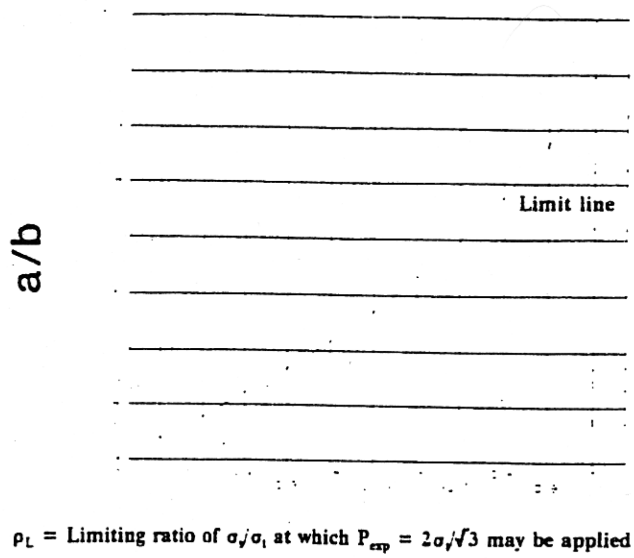


Fig. 11 $a/b$ versus $\rho_L$

the tube material, coefficient of friction between the tube and hole and tube wall thickness.

They treated the tube-hole assembly as being equivalent to a tube in a thick sleeve with its outside radius at the plastic/elastic interface. Equations (8) and (9) restate their results in the terms used in this paper.

For pullout loading

$$L_T = P_o \frac{2\pi b}{\gamma} (L_e - e^{-\gamma\phi L_e}) \tag{8}$$

For push-put loading

$$L_c = P_o \frac{2\pi b}{\beta} (e^{\beta\phi L_e} - 1) \tag{9}$$

where

$$\gamma = \nu \frac{c^2 - a^2}{b(c^2 - b^2)};\ \beta = \nu \frac{c^2 - b^2}{b(c^2 - a^2)}$$

These equations would be useful if the coefficient of friction could be precisely determined. However, the range of $\phi$, which is usually assumed to be 0.3 to 0.7 with the average value of 0.5 used for most estimates, may actually be as great as 0.3 to 1.0. Furthermore, the frictional behavior of the structure probably is different when it is hot and when it is cold [10]. Equation (10) is a practical approximation that neglects the Poisson effect of the tube and elastic response of the tubesheet.

$$L = P_o 2\pi b L_e \tag{10}$$

**2.5 Tube Rolling.** Rolling plastically strains and elastically relaxes the tube/hole structure cyclically. To examine the effect of cyclical loading it helps to conceive of the tube as a series of concentric shells in intimate contact. As the rolls traverse the tube, momentarily, they apply force on an infinitesimally small contact surface, thereby applying pressure far greater than the plastic limit on the innermost shell. In doing so, they create a slight depression in the inner concentric shells. Consequently, the rolls are always pushing a wave of metal ahead of them. As with the uniform-pressure model, any pressure greater than $(2/\sqrt{3})\sigma_t$ wall cause tube-end extrusion. The pressure on successive shells is reduced by the pressure drop through the inner shells. The tube-end shortening that results from the initial elastic and subsequent plastic overall tube deformation is overcome by tube-end extrusion. This is the reason for the well-known phenomenon of tube-end growth during rolling. If the tubes are not appropriately tack-rolled by lightly rolling in a few tubes at six or eight equally spaced peripheral locations and at the center, the tube extrusion can cause a tubesheet to bow and tilt out of perpendicular. Plastic radial deformation of the ligaments also causes overall radial tubesheet growth, which must be accounted for in removable-bundle exchangers.

A somewhat subtle phenomenon is that unless the tube end is fixed in the hole before rolling begins, the "bite" of the hardened rolls twists the tubes. If the tubes are not fixed in place before rolling, they may be loaded torsionally along their length. The tubesheets of two-tubesheet exchangers may be rotated relative to each other as much as 15 deg. Restraining the tube twist, for example in U-tube bundles, applied tensile stress to the straight lengths and ends that promotes stress corrosion cracking (SCC). This is one of the reasons that controlled-tensile-stress tubes, often specified for closed feedwater heaters, fail from SCC.

Well-known to manufacturers, is that the maximum length of tube that can be roller expanded in one rolling-tool application is about 2 in. (about 50 mm). Accordingly, rolling is done in steps. If the step-rolled lengths do not overlap, there is a series of transitions between the rolled and unrolled lengths instead of just one at the shell-side face region. For SCC-sensitive materials it creates that many more regions that can fail. The previously advocated hybrid expanding mitigates this problem.

**2.6 Wall Reduction, Rolling Torque and Joint Strength.** The common way to try to achieve uniform tube rolling is to control rolling torque so as to achieve uniform wall reduction. Some analysis has been done to try to corelate wall reduction with joint strength [6]. However, to the best of the author's knowledge, there are no published correlations of interfacial pressure with wall reduction. Moreover, rolled-joint interfacial pressure is not measurable in a production exchanger. There is also no evidence that wall reduction correlates directly with pull or push-out strength. Accurate wall-reduction measurements in relatively large-diameter, thin-walled tubes are difficult to make. When such tubes are of high-strength materials, for example Ti Gr III, only a modest amount of wall reduction is attainable. Repeatability from tube to tube is hard to achieve. Consequently, some manufacturers try to correlate joint strength directly with rolling torque. This requires all tubes rolled at a given torque value to have the same physical properties.

**2.7 Tube Heat Mapping.** Exchangers that have more than about 150 tubes will have tubes from more than one heat. It is troublesome to segregate the heats into discrete groups when loading them into the bundle and map the heat locations. However, heats of the same diameter, gage and alloy, produced by one manufacturer may vary very substantially from the specified *minimum* yield and tensile strengths tabulated in the ASME Code's tables of permissible stress. Wall reduction or rolling torque established on the basis of specimens from one heat may be too high or too low for tubes from another one. Heat treating the tube ends to a fully annealed condition may eliminate this problem provided that: 1) the heat treatment does not affect the corrosion resistance; 2) the heat-treated end is fully rolled to re-establish the required tube strength. The latter is required despite the fact that the tubesheet ligaments support the tube end.

In addition to variations in physical properties, tube-metal chemistry varies somewhat within the limits of the tube specifications. By not segregating the several heats of tubing, when there are tube failures, the ability to connect tube chemistry with the failure is lost.

**2.8 Two-Stage Expanding.** Except for hybrid expanding and expanding of thick tubes, most tubes are expanded in one application of the expanding device. This is also true for each step in step-rolling. However, in two-stage expanding, after the first roll, the tubes stiffen the ligaments, which makes it less likely that adjacent tubes will be knocked loose.

**2.9 Nonstandard Grooves—Grooves for Small-Diameter Tubes.** The two 1/8 in. wide × 1/64 in. deep (approximately 3 mm wide × 0.4 mm deep) grooves of the TEMA and HEI Power Plant Standards [1, 2] are adequate for most rolled joints. However, for thin-walled, high-strength tubes, a series of smaller, shallower grooves of the same total width produces better results [18]. Such grooves were described in the previously-footnoted "Working Guide to Shell-and-Tube Heat Exchangers." The author has achieved good results with 60 deg or smaller included-angle grooves having a depth $h = 0.35t$, width $W' = t$ spaced on $2.5t$ centers. For example, a 0.029-in. (about 0.74-mm) thick tube would be grooved with eight or nine 0.029-in. (about 0.74-mm) wide × 0.01 in. (about 0.26 mm) deep grooves on 0.07-in (about 1.8-mm) centers.

Analysis of the stresses and deflections of rolled and hydroexpanded tubes in various grooves seems to substantiate that wider-than-TEMA-standard grooves are desirable for hydroexpanding [19, 20]. The analysis of Lindike and others [16] supports the conclusions of Jawad and others [8] and Yoshitomi et al. [15]. Idealized widths for hydroexpanding based upon the Jawad modification of the Yoshitomi work, and assuming $\nu = 0.3$, are as shown in Eq. (11)

$$W = 1.1\sqrt{(a+b)t} \tag{11}$$

When an exchanger is built with small-diameter tubes, the designer should consider: 1) the percentage of ligament width removed by the tube grooving; and 2) the percentage of tubesheet thickness that the grooves occupy. For such tubes, better results might be achieved by using a coarse hole finish—in the range of 250 rms or grooves similar to those described for thin, high-strength tubes.

## 3 Welded-and-Expanded Tube-to-Tubesheet Joints

**3.1 Requirements for Welding Tubes to Tubesheets.** It is well known that to achieve tube-to-tubesheet welds free of cracks, inclusions and porosity, all the surfaces to be joined must be spotlessly clean and dry. Extreme measures taken at the outset may eliminate costly and often unsuccessful repairs after hydrostatic testing.

Tubesheets, to which tubes are to be welded must have a granular structure fine enough to permit consistently uniform weld-metal deposits. Carbon steel plate for tubesheets should always be grain refined. This is why SA-517 plate is not desirable for joining tubes by fusion welding. For the same reason forged-steel tubesheets should be clad with reasonably thick fine-grain weld deposit by submerged-arc, plasma or gas-tungsten-arc-welding.

The tube metal, tubesheet metal and weld filler metal (when used) must be compatible. This means not only that they are capable of fusing with each other and producing a tough, crack-free weld nugget, but that the thermal coefficients of expansion of the tube, the part of the tubesheet to which it is joined and the weld metal deposit must be close to each other. Consider, for example, an austenitic tube gas-tungsten arc welded to a carbon steel tubesheet with high-alloy filler metal. The fusion temperature is in the range of 2800°F (1730°C). The thermal coefficient of expansion of the austenitic components is about 50 percent greater than that of the carbon steel. As the welds solidify (which occurs very rapidly), the difference in the rates of expansion engender high levels of thermal stress. Pressure stresses superimposed on the thermal stresses may cause permature failures. If the shrinkage stresses could be relieved, heating would create thermal stresses due to restrained differential expansion. Therefore, the region of the tubesheet to which the tubes are welded should have a thermal coefficient of expansion close to that of the tube and filler metal. It is noteworthy that carbon steel weld-wire deposits on forged-steel tubesheets may have measurably different thermal expansion coefficients than the tubes.

**3.2 Strength and Seal Welding.** The meanings of strength and seal-welding tubes to the front-faces of tubesheets are frequently misunderstood. This has been exacerbated by lack of general agreement about joint configurations and weld sizes required for strength welds. The confusion may be eliminated with these definitions proposed for inclusion in the Code: "Tube-to-tubesheet seal welds are ones used to prevent fluid transfer between shell side and tube side—no strength is associated with it. Tube-to-tubesheet strength welds are ones used to transfer all longitudinal mechanical and/or thermal loads from the tubes to the tubesheet." These definitions do not conflict with Code sections that specify minimum leak paths in the weld as a percentage of tube wall thickness.

The suggested definition of strength weld leads to the following considerations: 1) using the joint efficiencies specified in the Code's Paragraph UW-15(c), the total weld cross-sectional area times the joint efficiency defined it in the cited paragraph must equal the tube cross-sectional area; 2) weld strength must be based upon the lower of the tube or tubesheet allowable stress; and 3) differential thermal expansion between the tube, tubesheet and weld metal must be considered.

**3.3 Welding Before Expanding or Expanding Before Welding.** To this day there is ongoing controversy about whether to expand first or weld first. This is what determines the sequence: 1) the requirement that the surfaces be very clean; 2) the need for a path of escape of welding-generated gases that will not cause porosity in the weld; 3) the maximum desirable root gap for the material being welded; and 4) the need to be able to repair welds that have failed in service.

Except for joining titanium tubes to titanium tubesheets, these requirements indicate that tubes should be welded to the tubesheets prior to expanding. If it were not for the necessity of keeping the root gap as small as possible in titanium joint, it would also be desirable to weld it first.

In order to set the tubes in place, it has been customary to "lightly" roll (presumably to create tube/hole contact that is not hydraulically tight). To avoid contamination, no lubricant should be used. Without lubricant, there is a possibility that heat of friction will cause some of the hardened roll and cage material to flake off and contaminate the welds. Alternatives are to set the tubes in place with a drift pin, or to use one of the commercially available expanders that compresses a polymer in the tube end to produce radial pressure.[3] These can be set to lock the tube in place without creating a hydraulically tight seal.

**3.4 Reasons for Full-Depth, Full-Strength Expanding After Welding.** When the welds are at or near the tubesheet front face, it is desirable to full-strength expand the tubes for approximately the whole tubesheet thickness. Here is why:

1 Expanding the tubes into the holes isolates the welds from the effects of tube vibration, which is always present to some degree.

2 Contact between the tube and hole permits heat to flow between the tube and hole wall. Therefore, the metal temperature of the length of tube embedded in the tubesheet is much closer to the temperature of the tubesheet metal than if there were no contact.

3 When the tubes are in hydraulically tight contact with the holes for the full tubesheet thickness, the ligament efficiency, upon which the elastic constants used in tubesheet design is based, is much greater than when there is no contact. This can be illustrated by considering 3/4 in. o.d. × 0.065 in. (approximately 19 mm × 1.6 mm) tubes in holes drilled to a TEMA special close fit on 15/16 in. (approximately 23 mm) pitch. If the tubes are simply welded to the tubesheet and not subsequently expanded, the ligament efficiency is

$$\eta = \frac{0.9375 - 0.758}{0.9375} = 0.191$$

However, if the tube is always in tight contact with the hole, the hole size is effectively the inside diameter of the tube. Therefore, the ligament efficiency is

$$\eta = \frac{0.9375 - 0.628}{0.9375} = 0.330$$

*3.4.1 Reason for Full-Strength Expanding.* The reasons for recommending full-strength expanding are: 1) interfacial pressure may be lower at joint operating temperature than at manufacturing temperature; 2) the required locked-in stresses in the tube and tubesheet may be creep relieved; and 3) residual stress declines drastically with reduced expanding pressure (compare Figs. 7 and 8). From these considerations it is apparent that if pressures less than those recommended here are applied, upon shock cooling or other untoward event, the tubes may pull away from the holes. This will reduce the tubesheet ligament efficiency on which the tubesheet thickness was based. In U-tube units it will also permit the welds to bear the full hydrostatic end load of the pressure in the tubes. This may be a frequent cause of weld failures. If the tubes separate from the holes, the metal temperatures of the tubesheet and tube end may also be far different. This may intensify the effects of differences in thermal coefficients of expansion between the tubes, tubesheet and weld metal.

## 4 Conclusions

1 Rubber or polymeric expanding is an ideal way to set tubes prior to welding because it can be adjusted to fasten the tube in place axially but without hydraulic tightness. This permits welding-generated gases to escape.

2 The variations of plastic radial stress with application of expanding pressure and the elastic stress relaxation have been graphically illustrated.

3 Graphic means, closed-form equations and empirical methods have been provided for estimating residual stress after tube expanding.

4 A basis and equations for optimizing expanding pressures for uniform-pressure expanding has been provided for the cases where tube and plate properties are the same, plate yield is higher than tube yield and tube yield higher than plate yield. The limit line concept and its use have been explained.

5 Plastic/elastic theory has been used to establish maximum practicable tube thickness for expanding.

6 A justification for using approximate means for calculating expanded joint pullout strength has been provided.

7 The effects of tube rolling on the structure have been examined. Practical suggestions for minimizing harmful ones have been made. The benefits of hydroexpanding have been outlined. Reasons have been advanced for questioning the validity of tube wall reduction and rolling torque as determinants of pullout and push-out strength.

8 The recommendations has been made to segregate and map tubes in exchangers in which more than one heat of tubing is installed. The basis for the recommendation has been explained.

9 A two-stage expanding technique has been suggested as a way to avoid manufacturing problems during expanding.

10 The use of non-TEMA-Standard grooves has been advocated for high-strength, thin-walled tubes and an empirical means of selecting groove configuration for small-diameter tubes has been presented. The equation for grooves for rubber expanding recommended by Yoshitomi and others, as modified by Jawad and others for grooves for hydroexpanding has been provided. It is based upon a typical Poisson constant of 0.3. Rough tube holes instead of grooving have been suggested for small-diameter tubes.

11 Requirements for successfully welding tubes to tubesheets have been outlined. Strength and seal welding have been defined and the basis for calculating weld sizes for strength welding outlined.

12 Reasons for welding before expanding for most materials have been submitted.

13 The advantages of full-depth strength expanding have been explained.

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