

APPROPRIATE CORRELATIONS FOR ASSESSING EXPANDED TUBE-TO-TUBESHEET JOINT STRENGTH

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1. ABSTRACT

This paper advocates correlating rolling torque or hydraulic expanding pressure with shear load strength instead of percent wall reduction as the criterion of expanding adequacy for thin-walled tubes. To establish a basis for such advocacy, it discusses tube expanding theory, methods and annular grooves. The paper compares the magnitudes of measurements required to estimate percent wall reduction with cumulative tolerances for hole drilling and tube manufacture. A graphic illustrates how data scatter increases with ratio of tube diameter to wall thickness. The paper comments on tube rolling procedure specifications and concludes with recommendations for two-stage and hybrid expanding.

2. INTRODUCTION

Whether to correlate percent wall reduction or rolling torque with joint pullout of thin-walled, high-strength, low-elastic modulus tubes expanded into tubesheets has proponents on both sides of the issue. The author believes that for such tubes correlating torque with pullout (shear-load) strength is the superior method. The following discussions of tube expanding theory and tube rolling practice underpins this opinion.

3. TUBE EXPANDING THEORY

The vast majority of tubes expanded into tubular heat transfer equipment are roller expanded. However, the complexity of analyzing the large numbers of variables that apply to tube rolling is insuperable. Therefore, the basis of most published investigations of expanding tubes into tubesheets is a model of applying pressure uniformly in the end of the tube.

3.1. Expanded Tube-to-Tubesheet Joint Model

A simplistic model of the tube and surrounding metal is as follows. The tube is a cylinder composed of an infinite number of infinitesimally thin shells in contact with each other. Initially, there is clearance between the outside of the cylinder and the tube hole. With application of pressure in the tube, the assembly of shells deflects. The inside tube radius increases until the outside of the tube contacts the hole. At this point, the deflection of the shells may be full elastic, fully plastic or the deformation of inner shells may be plastic and outer ones elastic with an elastic plastic interface. The state of the deflection depends upon the tube dimensions and its yield stress, σ_{yt} .

Theoretically, the tube will remain fully elastic up to a limiting applied pressure equal to $\sigma_{yt}/\sqrt{3}$ or $0.577\sigma_{yt}$, the so-called elastic limit. As the tube end undergoes elastic deformation its wall becomes thinner and its length becomes shorter in response to the Poisson effect. As pressure exceeds this limit, plastic deformation begins. Increasing the internal pressure moves the interface between elastically and plastically deformed shells outward. After the tube contacts the tubesheet, a similar process takes place in the metal of the tubesheet. The goal is to create fully plastic tube deformation, after which, upon relaxation of the tube and surrounding tubesheet, residual stress remains at the location of the tube-tubesheet interface. The analysis treats this residual stress as interfacial pressure between the tube and hole.

One analysis uses a plain-stress model based on applying pressure to a hole in a large plate to investigate tube expanding.¹ The holes that surround a hole under investigation in actual tubesheets limit the utility of this model. However, finite element analyses have shown the model of a hole under pressure in a large plate offers a practical way to examine tube expanding. In this model, when the radius of the plastic zone reaches 1.75 times the tube inside radius, it is not possible to attain further enlargement because the innermost shells begin to extrude. The corresponding expansion pressure at this point is $2\sigma_{yt}/\sqrt{3}$ ($\sim 1.155\sigma_{yt}$). Attempts to apply greater pressure spread the extrusion into successive shells.

As expanding pressure increases, the pressure that the tube exterior exerts on the hole-wall initiates deflection in the surrounding tubesheet metal, which deflects in similar fashion to the tube. The boundary between the elastic-plastic interface moves outward into the tubesheet. Upon pressure release, relaxation is approximately elastic. In order to create a tube-to-tubesheet joint, stress at the tube-hole interface in the residual stress continuum must be greater than zero after relaxation. We refer to the residual stress at the interface as interfacial pressure. The various analyses for relating and optimizing interfacial pressure to expanding pressure consider tube dimensions, tube and tubesheet yield stresses and other properties of the tube and tubesheet.

3.1.1. Equal tube and tubesheet yield stresses

In the circumstance in which the tube and tubesheet yield stresses are equal, the plastic zone is essentially continuous. In this circumstance, the optimum expanding pressure is the maximum that will not cause the tube to begin to extrude. That pressure is equal to $1.155\sigma_{yt}$. Analysis to determine the location of the plastic-elastic interface for this circumstance has been adapted to determine the thickest tube capable of being expanded into a tubesheet.

3.1.2. Tubesheet yield stress higher than tube yield stress

If tubesheet yield stress, σ_{yp} , is higher than that of the tube, a new elastic-plastic boundary is established in the tubesheet metal after full tube plasticity has been achieved. As the ratio of tubesheet to tube yield stress, σ_{yt}/σ_{yp} increases, radius of the elastic-plastic boundary in the tubesheet moves closer to the tube. At a ratio of two, the location of the elastic-plastic interface is at the tube-tubesheet contact surface and the tubesheet metal remains fully elastic.

3.1.3. Tubesheet yield stress lower than tube yield stress

Steam surface condensers tubed with high-strength, thin-walled, low-elastic-modulus tubes expanded into clad steel or non-ferrous tubesheets exemplify this situation. Here, at ratios of σ_{yt}/σ_{yp} greater than one, the relative thickness of the tube governs the maximum permissible expanding pressure. The simplest way to understand the effect of pressure on the tube-hole structure is to imagine a tube of zero thickness in intimate contact with the hole. In this imaginary circumstance, the pressure at the tube OD would be the same as the pressure at the tube ID. Consequently, the limiting pressure in the tube would be that of the tubesheet limiting pressure equal to $2\sigma_{yp}/\sqrt{3}$. The plate yield stress would therefore govern the maximum expanding pressure.

In a tube with some thickness, which is in the fully plastic condition, there is a calculable pressure drop across the tube. The pressure that the tube exerts on the hole is the tube expanding pressure, less the pressure drop across the fully plastic tube. In order to prevent tube hole extrusion, the external tube pressure must not exceed the tube plate plastic limit of $2\sigma_{yp}/\sqrt{3}$. Therefore, the maximum expanding pressure to apply in the tube cannot exceed the sum of the plate plastic limit and the pressure drop across the fully plastic tube. This provides a means of determining the maximum permissible pressures for tube-wall, tube-diameter, and tube-plate constructions.

3.2. Effect of Tube Rolling on the Tube End

¹ See S. Yokell, "Expanded, and Welded-and-Expanded Tube-to-Tubesheet Joints" for this analysis.

When the taper mandrel of the tube rolling tool forces the hardened pins into contact with tube surface, the contact area is infinitesimally narrow and approximates a line. Theoretically, lines have no width. The contact area surface is zero and the pressure that the pins apply is infinitely large. However, after contact, as the process drives the mandrel forward, the pins first deform the tube elastically in the form of depressions with finite contact area. The positions of the pins rotate counter to the direction of mandrel rotation. At this point the tube end length becomes shorter and the wall becomes thinner because of the Poisson effect. Very rapidly, the process begins to deform the tube plastically. The unit pressure that the pins apply to the tube surface surpasses the tube plastic limit. Therefore, in the model described above, the shells that make up the tube progressively extrude as the pressure on each shell exceeds the plastic limit. The extrusion takes place at the expense of wall thickness, and the plastic thinning and thinning due to tube metal extrusion is the total wall reduction.

In a physical tube-tubesheet structure, one cannot readily measure the actual percent wall reduction. Consequently, the practice is to deduce it from measurements of the hole and tube dimensions before expanding and measurement of the tube ID after expanding. The increase in the tube ID measurement includes some unaccounted-for movement of the surrounding ligaments; hence, the practice is to call the deduced wall reduction, “apparent wall reduction”.

3.3. Correlation of Tube End Deformation with Tube Pullout Strength and Tightness.

Early attempts to correlate tube-end extrusion with joint pullout strength were abandoned because of measurement difficulties and widely scattered results. Greater success has been achieved in relatively thick tubes by correlating joint pullout strength with measurements of percent wall reduction. The percent wall reduction criterion has become the most common method used to establish tube rolling adequacy for process and power heat exchangers. However, there is no theoretical basis for a general correlation of pullout strength with percent wall reduction. Consequently, the universal practice is to perform shear load tests (pullout tests) in models to establish the percent wall reduction to use in production equipment.

The literature on tube expanding is not replete with correlations of apparent percent-wall-reduction with tube joint tightness. HydroPro, Inc. of San Jose, Calif. has done some shear-load strength and tightness testing to examine expanded joint tightness of hybrid expanded tubes and validate the expanding procedures. (Hybrid expanding is hydraulically expanding tubes at a pressure that creates a joint with modest interfacial pressure, followed by tube rolling to increase the interfacial pressure required for the desired joint strength.) For this testing, HydroPro correlated hydraulic expanding pressure and rolling torque with joint strength and tightness. The pullout and tightness tests met the specified criteria for the validation.

3.4. Typical Percent Wall Reductions

Tool manufacturers’ suggested increases in tube diameters after the tube contacts the wall of the hole, vary with tube material, diameter and gage. A typical recommendation for a stainless steel tube 19-mm OD x 0.5-mm thick wall (18 mm ID) would be an increase of 0.1016 mm. Assuming a clearance of 0.254 mm, and neglecting tube wall thinning to the point of tube-hole contact, the tube OD at point of contact would be 19.254 mm. The ID at contact would be $18 + 0.254 = 18.254$ mm. After expanding to achieve an increase in ID of 0.1016 mm, the tube ID would be $18.254 + 0.1016 = 18.3556$ mm. The calculated new wall thickness would then be $(19.254 - 18.3556)/2 = 0.4492$ mm. The percent wall reduction would be $100(0.5 - 0.4492)/0.5 = 5.08$ percent.

3.5. Limits on the Applicability of Correlations of Percent Wall Reduction with Pullout Strength

The utility of correlating percent wall reduction with pullout strength rests on several assumptions:

- (1) The ODs and thicknesses of the tubes in each heat are uniform.
- (2) The tube hole diameters are uniform and do not vary throughout the thickness of the tubesheet.

- (3) Pullout strength varies as some definable function of tube-wall-thickness-reduction, and is constant for the specified percent wall reduction in tubes from the same heat.
- (4) The instruments used for measuring the unexpanded tube OD and ID, the hole ID, and the expanded tube ID are adequate to the task of making the measurements precisely enough to minimize data scatter.

The facts do not support these assumptions because of the following:

- (1) It is customary to require application of the drilling tolerances in the TEMA Standards for special close fit holes to feedwater heaters and surface condensers. For 19.1 mm OD tubes these are $19.25 \text{ mm} \pm 0.05$ for 96% of the tubes, with 4% of the tubes permitted to be 0.25 mm oversized.
- (2) Diameters of tube holes vary within the permissible tolerances through the thickness of the tubesheet.
- (3) Mill tolerances apply to tube manufacture; for example, Titanium tubes produced to ASME specification SB-338 in sizes under 25.4 mm may vary $\pm 0.102 \text{ mm}$ in OD from the specified diameter and $\pm 10\%$ from the specified wall thickness.
- (4) Other than one paper that presented measurements of percent wall reduction VS joint strength for boiler tubes, the author has found no theoretical or experimental data to validate that joint strength varies in some uniform way with percent wall reduction. There was considerable scatter in the results.
- (5) Different individuals experienced in using three ball snap micrometers to measure tube and hole IDs will read substantially different values, notwithstanding the typical instrument calibration to 0.00254 mm.

In practical terms the third assumption is moot because pullout loads in the models used for qualifying the process are held constant after deciding the percent wall reduction to be used for a given assembly.

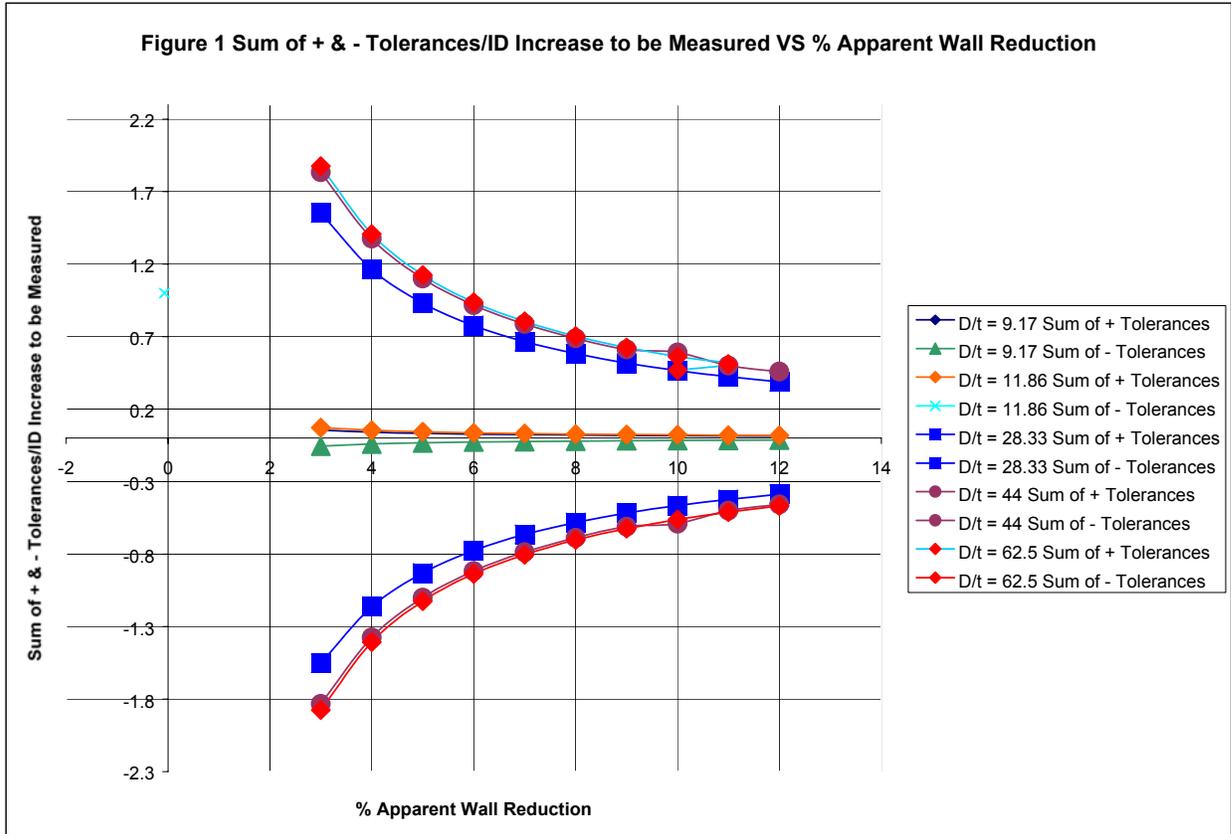
Although the percent-wall-reduction criterion is widely used, as Figure 1 shows, it becomes less reliable as the tube wall thickness declines relative to diameter. Most process heat exchangers have wall thickness in the range of 1.25 to 2.8 mm thick. Depending upon feedwater pressure and material selection, feedwater heater tubes may vary from 0.712 to 2.8 mm thick. For very thin-walled tubes, the range of data scatter is very wide between the sum of plus and minus tolerances as a fraction of ID to be measured to determine apparent percent wall reduction. As the wall thickness increases, and D/t decreases, the range of data scatter narrows. This shows graphically that correlating percent apparent wall reduction with pullout loads is a reasonably good practice for most process exchangers where typical D/t values range from ~ 13 to ~ 7 . However, it is not a suitable way to determine the adequacy of tube expansion for thin-walled tubes, such as those used in low-pressure feedwater heaters and in steam surface condensers.

It is noteworthy that Table 5 of ASME Code Section II Part B, Specification SB-338, from which the dimensional data used in constructing Figure 1 is abstracted, is titled, "Permissible Variations in Outside Dimensions Based on *Individual* Measurements" (Italics added). Figure 1 illustrates the factor by which the sum of maximum tube diameter and wall thickness, and hole drilling tolerances exceeds the increase in tube ID that must be measured for five ratios of tube OD/t. It shows that the cumulative plus or minus tolerances can be a substantial multiple of the increase in diameter required to create the specified apparent percent wall reduction. The ratio (sum of tolerances)/(tube ID increase to be measured), decreases with increasing percent average wall reduction. The combinations of plus and minus tolerances for individual tubes and the difficulty of making accurate measurements of the tube and tube hole are what cause often-observed wide data scatter.

3.6. Precautions for Correlating Pullout Load with Rolling Torque

There are some caveats for using correlations of rolling torque with pullout load as the determinant of joining adequacy.

- (1) The torque-sensing device in the rolling equipment cannot sense if resistance to rotation stems from the effort to squeeze the tube wall, or if foreign matter in the tool or tool-to-hole region is



impeding rotation. Consequently, to achieve reliable correlations the surfaces and tool must be extremely clean.

- (2) Torque required to dry-roll tubes into tubesheets is considerably higher than that required for rolling with lubrication. Therefore, qualification tests made with lubricated rolling do not apply to dry rolling.
- (3) Table 1 shows SB-338's minimum and maximum allowable yield stresses for three grades of annealed Titanium tubes. It includes the ratios of the maximum to the minimum allowable stresses to highlight the magnitude of the range. Therefore, to use torque-pullout-strength correlations, there must be a reasonably close match of the yield stresses reported for the tubes of the surface condenser with the yield stresses of the tubes used to prepare the test models. Depending upon the number of tubes to be installed several heats of tubing may be provided, and a box may contain tubes from more than one heat. For this reason, it is prudent to require manufacturers to provide heat maps of the installed tubes keyed to the tube heat numbers,

bearing in mind that one box of tubes may have several heats of tubing. Such heat maps are also valuable for examining the potential causes of tube failures.

3.7. Comments on Rolling Procedure for Steam Surface Condensers Tubed with Thin Titanium Tubes

Subject to the caveats stated above, the author's opinion is that correlating pullout strength with rolling torque of high-strength, thin-walled, low elastic modulus tubes is a superior method to correlating pullout strength with percent wall reduction.

3.8. Comments on Tube Rolling Specifications

Specifications should include the following

- (1) The specification should state, how the manufacturer proposes to clean the holes in the tube plate before tubing. Common methods are to steam clean and blow-dry with filtered, dry air, or alternatively, to wipe the holes with acetone-soaked felt plugs.
- (2) The specification should require cleaning the tube ends before expanding and should require the manufacturer to describe the proposed cleaning method.
- (3) Wording like, "suitable protection", is insufficiently specific. Specifications should require workers not to open boxes of tubes until just before tubing the condenser. The specification should require that tubes in boxes from which workers have loaded tubes into the condenser and when loading is interrupted, be covered with plastic sheeting to prevent settling of foreign matter on the tube surfaces.
- (4) The specification should set a maximum time between removal of tubes from the box and their installation. Workers should not open boxes of tubing until they are ready to install them in the condenser. However, if the workload and schedule requires a period of more than one hour to elapse after opening the box, the workers should cover the tubes with plastic sheeting.
- (5) The specification should describe how the manufacturer proposes cleaning the tubes after expanding in some detail.
- (6) The specification should include a plan for the distribution throughout the tube field of bands or clusters of tubes expanded into the tubesheet to stabilize the tubesheet before commencing production rolling.
- (7) The specification should have a description of how the manufacturer will calibrate the torque controller, and the frequency of calibration.
- (8) The specification should provide standards for maintaining the condition of the tube rolling equipment.
- (9) The specification should include a system for identifying tube positions.
- (10) The specification should include a system for locating tube heats.

3.8.1. Rolling tools and equipment

The author is of the opinion that parallel 5-pin, hydraulically operated tube-rolling equipment is superior to inclined-roll (self-feeding) 3-pin tube rolling equipment. Such equipment is available from Asian, North American, and European manufacturers. The reason for this opinion is that, although the frequency with which the rolls traverse the tube surface is 67% greater, the amplitude of the strains is much lower. Therefore, there is less likelihood of fatiguing the tube metal (See Figure 1). Because, it may be impractical to manufacture 5-pin rollers for use in tubing of inside diameter smaller than 18 mm, for tube IDs 18 mm and smaller, the tube rolling tool should be 3-pin.

For most Titanium-tube surface condensers, the tubes are harder than the tubesheets. In this circumstance, self-feeding (inclined) rolling tools create an hour glass configuration of the metal deformation. With such a configuration, the interfacial pressure varies throughout the thickness of the tube end, effectively producing a weaker, less tight joint than a parallel roll tool would produce. Note

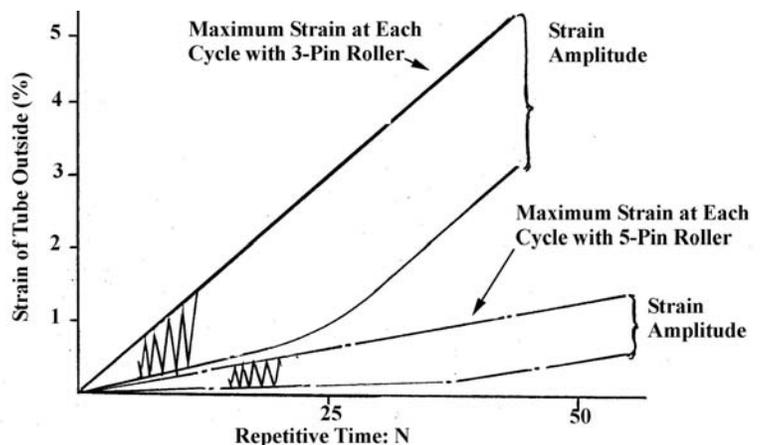


Figure 2 Tube outside strain amplitude VS Frequency (Adapted from U. S. Patent No. 4,142,581, Issued March 6, 1979 to Yuji Yoshitomi and others)

that parallel tools are not exactly parallel but are slightly inclined to assist the hydraulic ram in inserting and retracting the tapered mandrel.

3.9. Suggested Alternative Method for Tube Expanding (Hybrid Expanding)

As the curve reproduced from U.S. Patent 4,142,581 shows, the greater number of pins is less likely to cause tube ends to fatigue. If the number of pins were to be infinite, the tendency to fatigue would be minimal. This (impossible) situation mimics what would happen if hydraulic pressure were to be applied in the tube end. Applying hydraulic pressure directly in the tube end expands the tube without causing tube-end extrusion. However, it has the disadvantage that it does not strain harden the inner shells of the tube end sufficiently to restrain tube spring-back.

To the author's knowledge, a major manufacturer of naval shipboard surface condensers has recently used hybrid expanding to qualify models of tube-to-tubesheet joints. It consists of hydraulically expanding to initial tightness, followed by tube rolling. Hybrid expanding has these advantages:

- (1) The initial expanding increases the tube ligament efficiency thereby enhancing tubesheet stiffness. Eq. 1 defines ligament efficiency.

$$h = \frac{p - d}{d} 100 \quad (1)$$

In this equation,

- | | | |
|--------|---|-----------------------------|
| η | = | Ligament efficiency percent |
| p | = | Tube pitch, mm |
| d | = | hole ID, mm |

The increase in ligament efficiency for 25-mm x 0.5-mm wall tubes is modest. For smaller diameter and thicker-walled tubes, it is substantial. At either end of the wall-thickness range, such two-stage expanding improves tubesheet stiffness.

- (2) Control of hydraulic expanding pressure currently available as standard equipment from the manufacturer whose equipment the author is familiar is $\pm \sim 70$ Bar. For a premium price, the manufacturer can furnish a machine with a precision of approximately 35 Bar. In correlating tube yields with hybrid expanding, the same requirements regarding tube yields apply as apply for correlations of torque with pullout strength.
- (3) The transition from expanded to unexpanded tube is more gradual than with tube rolling. Therefore, the residual stress in the transition zone is lower than with tube rolling.

3.10. Annular Tube-hole Grooves for Expanding Titanium Tubes

3.10.1. Grooves for rolling

TEMA standard grooves, 3.18 mm wide x 0.4 mm deep are not suitable for expanding thin-walled titanium tubes and other materials with similar physical properties. This is because the tubes may crack at the corners of the grooves as the tube metal deforms to form keys. Multiple shallow, Vee-shaped grooves adequately enhance expanded joint strength and tightness for such materials.² One firm that specializes in retubing surface condensers, has found that refinishing the tube holes with serrations provides excellent strength and leak tightness.³

² Such grooves are pictured in Page 189 of S. Yokell, "A Working Guide to Shell-and-Tube Heat Exchangers", McGraw-Hill, New York, 1990.

³ Personal communication from Bob Hahn of Atlantic Group, Norfolk, Virginia.

3.10.2. Grooves for hydraulic expanding

The current edition of the TEMA Standards requires grooves for hydraulic expanding to be 6.35 mm wide. Although this width is suitable for most materials, use Eq. 2 to optimize the groove width.

$$W = 1.56\sqrt{Rt} \quad (2)$$

In this equation, W is the optimal width, R is the mean tube radius, and t is the tube wall thickness.

4. CONCLUSIONS

The following conclusions are drawn from the foregoing discussions of expanding theory and the difficulty of making methods adequate to predict deduction of percent wall reduction.

- (1) For thin-walled Titanium tubes, correlating shear loads (pullout loads) with torque provides a superior way to achieve consistent results.
- (2) Designers should be aware that rolled joints with thin-walled Titanium tube may require annular groove configurations different from TEMA standard grooves. They should be aware that grooves for hydraulic expanding should be at least 6.35 mm wide.
- (3) Two-stage expanding is desirable because of the stiffening effect of the first stage on the tubesheet.
- (4) Hybrid expanding takes advantage of fixing the tubes firmly in place and stiffening the tubesheet in the hydraulic expanding stage, and strain hardening the inner shells of the tube end sufficiently to overcome spring back. This results in tube-to-tubesheet joints less prone to fatigue failures and with adequate strength for the service.

5. ACKNOWLEDGEMENTS

The author wishes to acknowledge the information that HydroPro, Inc. of San Jose, California provided about their hydroexpanding equipment.

Tables

Table 1							
Permissible variations in SB338 tubes							
Nominal OD mm	Diameter Variation		Nominal Wall mm	Variation			
	Maximum mm	Minimum mm		Maximum mm	Minimum mm		
17.000	17.102	16.898	0.400	0.440	0.360		
18.000	18.102	17.898	0.500	0.550	0.450		
19.500	19.602	19.398	0.508	0.559	0.457		
21.000	21.102	20.898	0.600	0.660	0.540		
22.000	22.102	21.898	0.700	0.770	0.630		
22.225	22.327	22.123	0.711	0.782	0.640		
22.500	22.602	22.398					
24.000	24.102	23.898	Range of Permissible Stress Variations				
25.000	25.102	24.898	Grade	Min Mpa	Max Mpa	Max/Min	
25.400	25.527	25.273					
28.575	28.702	28.448	1	170	310	1.82	
			2	275	450	1.64	
			3	380	550	1.45	

Table 2		
Relative Thickness of Tubes		
Tube OD D mm	Tube Wall t mm	Relative Thickness
17.000	0.400	42.50
17.000	0.500	34.00
17.000	0.600	28.33
18.000	0.400	45.00
18.000	0.400	45.00
18.000	0.500	36.00
18.000	0.600	30.00
19.000	0.400	47.50
19.000	0.500	38.00
19.000	0.600	31.67
20.000	0.400	50.00
20.000	0.500	40.00
20.000	0.600	33.33
20.000	0.700	28.57
21.000	0.400	52.50
21.000	0.500	42.00
21.000	0.600	35.00
21.000	0.700	30.00
22.000	0.400	55.00
22.000	0.500	44.00

Table 2		
Relative Thickness of Tubes		
Tube OD D mm	Tube Wall t mm	Relative Thickness
22.000	0.600	36.67
22.000	0.700	31.43
23.000	0.400	57.50
23.000	0.500	46.00
23.000	0.600	38.33
23.000	0.700	32.86
24.000	0.400	60.00
24.000	0.500	48.00
25.000	0.400	62.50
25.000	0.500	50.00
25.000	0.600	41.67
25.000	0.700	35.71
25.400	0.500	50.80
25.400	0.600	42.33
25.400	0.700	36.29
28.575	0.500	57.15
28.575	0.600	47.63
28.575	0.700	40.82