Part UHX
White Paper

This "White Paper" is intended to describe the basis of requirements contained in Mandatory Section UHX of Section VIII Division 1 of the Boiler and Pressure Vessel Code. The discussion provides the highlights for the derivation of the analytical methodology and a discussion of the differences of between the ASME requirements and those of TEMA and other international Codes. Likewise, a general discussion is provided regarding the significance of the calculated stresses and how the rules are intended to be applied.

I   U-TUBE TUBESHEETS

I.1   Appendix AA – 1 for U-tube tubesheet appeared in Section VIII Division 1 for the first time in 1982. The rules were based on Gardner's method (1969). The same method was adopted in BS 5500 (1976) and in CODAP (1982), using for the allowable bending stress in the tubesheet: $\Omega S = 2S$ as recommended by Gardner (where Ω is a multiplier of the basic Code allowable stress).

The method was improved in 1990 by ASME, based on works of Alan Soler (1984) and Urey Miller to account for the configurations b and e (tubesheet gasketed on one side and integral on the other side). (See Figure 1.)

In year 2000, F. Osweiller proposed a more refined approach, based on the latter, to provide an analysis model to cover the 6 configurations of tubesheets shown in Figure 1.

This method, which follows the stress analysis of Section VIII – Div. 2 Appendix 4, is based on the following (see Figure 2):

a)   the perforated tubesheet is treated as a solid equivalent circular plate of effective elastic constants E^* (effective modulus of elasticity) and ν^* (effective Poisson's Ratio) depending on the ligament efficiency μ^* of the tubesheet;

b)   the unperforated tubesheet rim is treated as a rigid ring;

c)   the connection of the tubesheet with shell and channel accounts for the edge displacements and rotations of the 3 components. The effect of the pressure P_s (shell side pressure) and P_t (tube side pressure) acting on shell and channel, respectively, is considered;

d)   the maximum stresses in tubesheet, shell and channel are determined and limited to the appropriate allowable stress-based stress classifications of Section VIII Division 2 Appendix 4.

This method is the only method which accounts for all the 4 items a) through d) above.

I.2   This method has been adopted in 2001 by ASME (Section VIII – Div. 1 Appendix AA-1), by CODAP (Chapter C7) and European Standard (UPV Design Part – Clause 13). Publication is scheduled by the 3 Codes in 2002. This is also the basis of Part UHX-12 of Section VIII – Div. 1.

For more details of the specific derivation, see paper from F. Osweiller (2002), attached.

"ICPVT 2002
Vancouver Rev 11.pdf"

I.3   The TEMA rule appeared for the first time in 3rd Edition 1952 using the well-known tubesheet formula:

$$T = F \frac{G}{2} \sqrt{\frac{P}{S}} \quad \text{with} \qquad F = \begin{cases} 1.0 & \text{if the tubesheet is clamped} \\ 1.25 & \text{if the tubesheet is simply supported} \end{cases}$$

(where T is the TEMA tubesheet thickness)

The TEMA formula is based on the classical circular plate formula:

$$T = C \frac{G}{2} \sqrt{\frac{P}{\mu * \Omega S}} \quad \text{with} \quad C = \begin{cases} 0.866 & \text{if the plate is clamped} \\ 1.113 & \text{if the plate is simply supported} \end{cases}$$

where $\mu*$ is the effective ligament efficiency and ΩS the allowable stress.

If we use $\mu* = 0.5$ and $\Omega = 1.5$ we obtain approximately the values of TEMA formula which in this 3rd Edition does not consider any ligament efficiency as a direct input.

Current edition corrects that and provides a new formula:

$$T = F \frac{G}{3} \sqrt{\frac{P}{\eta S}}$$

where η is the TEMA ligament efficiency based on the mean width of the ligament (ASME ligament efficiency $\mu*$ is based on the minimum width of the ligament, which leads to lower values than TEMA).

The minimum values of η are 0.42 (triangular pitch) and 0.50 (square pitch). Coefficient 3 in the new TEMA formula has been determined so that old and new formulas led to approximately same results for these minimum values of η. In real cases η will generally range between 0.45 and 0.60, which leads to a decrease of *T* by about 10 to 15 %.
For more details see F. Osweiller's paper (2002).

I.4 **TEMA ligament efficiency** η is significantly higher than ASME ligament efficiency μ^* (generally $0.25 \le \mu^* \le 0.35$).

Comparison of TEMA new formula and classical plate formula shows that TEMA allows the bending stress in the equivalent solid plate to be 2.6 S, instead of 1.5 S recommended by ASME Section VIII – Div. 2.

Nevertheless, for about 40 years, TEMA formula did not lead to failures in U-tube tubesheets. It is likely that a value of allowable stress of 2S could be used without affecting the safety margin. This value was also recommended by GARDNER in 1969 and has been used in most European codes (BS 5500, CODAP) for about 20 years. This is also justified by limit load analysis applied to circular plates, which leads to $\Omega = 1.9$ if the tubesheet is simply supported and $\Omega = 2.1$ if the tubesheet is clamped. The SWGHTE is considering revising the rules to effectively allow $\Omega = 2.0$.

For these reasons tubesheet thickness obtained by ASME is generally thicker than TEMA, which is not the case for CODAP and UPV. See Table 1, which shows a comparison on 4 U-tube tubesheet heat exchangers.

I.5 **In conclusion** ASME method has the advantage of accounting for the perforated tubesheet, the unperforated rim, the interaction of the tubesheet with shell and/or channel, which are not treated rigorously in TEMA. The ASME method is considered more rigorous and allows the designer to more accurately account for actual geometry.

II. **FIXED TUBESHEETS**

II.1 **Appendix AA-2 for fixed tubesheets appeared in 1992**. The method was validated by performing a benchmark of 10 industrial heat exchangers with comparison to TEMA Standards and to CODAP rules which have been used in France since 1980.

The design method is based on the same stress analysis as described for U-tube tubesheets. However, the tube bundle is considered as an elastic foundation, which adds considerable complications. For more details, see Alan Soler's book (1984) and the Soler/Caldwell PVP paper that is attached. CODAP method is based on the same approach, except that the unperforated rim is not considered. CODAP method has been adopted by BS 5500 (1986) and by UPV (1992).

"Soler Caldwell
PVP Paper.pdf"

It should be noted that both ASME and CODAP came to the same conclusion in the 80's (without consulting each other) that is was necessary to develop new design rules accounting better for the behavior of the tubesheets than TEMA.

II.2   TEMA rules are based on the same basic approach as ASME and CODAP, but drastic simplifications have been made in TEMA:

-    unperforated rim is not considered;
-    connection of tubesheet with shell and channel is not treated rigorously (ratio e_s / D_s (where e_s is shell thickness and D_s is the shell diameter) in coefficient *F* cannot account for the rotational stiffness of the shell or channel);
-    coefficient *F* does not account for the stiffening effect of the tube-bundle, nor for the holes which weaken the tubesheet. TEMA assumes that these 2 effects are counterbalanced.

These simplifications have a significant impact on the results obtained by TEMA.

II.3   The design of the heat-exchanger tubesheet is significantly affected by the stiffness ratio X_a :

$$X_a = \frac{\text{axial tube bundle rigidity}}{\text{tubesheet bending rigidity}}$$

which accounts for the support afforded to the tubesheet by the tube bundle which strengthen the tubesheet and for the perforations which weaken it. It may range between $X_a = 1$ (low support) and $X_a = 20$ (high support) but lies generally between 2 and 8.

The ASME formula for tubesheet stress can be written:

$$h = F_{\text{ASME}} \frac{G}{3} \sqrt{\frac{P_e(X_a)}{\eta S}} \quad \text{with:} \quad F_{\text{ASME}} = 3\sqrt{\frac{\eta}{\mu *} F_M(X_a)}$$

(Where "h" is the required ASME tubesheet thickness)
so as to compare to the TEMA formula :

$$T = F_{\text{TEMA}} \frac{G}{3} \sqrt{\frac{P_e(X_a)}{\eta S}} \quad \text{with:} \quad F_{\text{TEMA}} = \begin{cases} 0.8 & \text{if the tubesheet is clamped} \\ 1.0 & \text{if the tubesheet is simply supported} \end{cases}$$

In the TEMA formula, the equivalent pressure, P_e, accounts correctly for the tube bundle through X_a (despite some simplifications), but coefficient F_{TEMA} does not depend on X_a. The rigorous solution of the equations shows that the term F_{ASME} is a strong function of X_a. It can be demonstrated that when X_a is small, then F_{ASME} is greater than F_{TEMA}. When X_a is large, then F_{ASME} is smaller than F_{TEMA}. Since the tubesheet stress is directly proportional to F_{ASME}, then it follows that small values of X_a results in a thicker tubesheet than TEMA and large values of X_a results in a thinner tubesheet than TEMA.

Coefficient F_{ASME} is represented in Figure 3 when the tubesheet is either simply supported or clamped. Figure 3 shows that F_{ASME} varies significantly with X_a:

- for low values of X_a $(X_a < 3)$ TEMA rules generally lead to an unconservative thickness;

- for high values of X_a $(X_a > 6)$ TEMA rules will generally lead to conservative thickness

- values of X_a between 3 and 6, TEMA rules lead to tubesheet thickness that is close to ASME.

Due to simplifications mentioned above, TEMA does not ensure an overall and consistent design margin for all heat exchangers. Numerical comparison between ASME, CODAP and TEMA illustrates this in Table 2. However, it must be pointed out that the value of coefficient F_{TEMA} has been remarkably well chosen as it represents approximately the mean value of coefficient F_{ASME}.

For more details see Alan Soler's book - § 9.8 (1984) and F. Osweiller's paper (Analysis of TEMA tubesheet design rules-comparison with up to date Code methods – PVP 1986).

"Osweillers PVP Paper -1986.pdf"

III.   FLOATING TUBESHEETS

The same analytical approach applies to floating tubesheets, with the simplification that the equivalent pressures $P_e(X_a) = P_s - P_t$.

Again TEMA ignores the effect of the tube bundle and same comments as above apply.

REFERENCES

K.G. GARDNER (1969)
"Tubesheet Design: a basis for standardization" – Pressure Vessel Technology – 1969 Delft Conference – p. 621-648.

A.I. SOLER (1984)
"Mechanical Design of Heat Exchangers" – Arcturus publishers – 1047 pages.

F. OSWEILLER (1986)
"Analysis of TEMA tubesheet design rules – Comparison with up-to-date Code Methods" PVP Vol. 107 ASME Pressure Vessel & Piping Conference – Chicago – p. 1-9.

F. OSWEILLER (2000)
"Tubesheet Heat Exchangers: New Common Design Rules in UPV, CODAP and ASME "ASME Journal of Pressure Vessel Technology – Vol. 122- August 2000.

F. OSWEILLER (2002)
"New common design rules for U-tube heat-exchangers in ASME, CODAP and UPV Codes" PVP – Vancouver – August 4 to 8, 2002.

Code References

- ASME Section VIII – Div. 1 – Appendix AA: July 2001 Edition (Addenda July 2002)
  – Draft UHX Chapter (May 2002, Rev. 0)

- CODAP 2000: French Code for Unfired Pressure Vessels – Section C7: January 2001 Edition

- UPV: Draft European Standard for Unfired Pressure Vessels (Pr EN 13 445) – Part 3 – Clause 13: to be published in 2002

- TEMA Standards: 3rd Edition – 1952

- TEMA Standards: 6th Edition – 1978

- TEMA Standards: 8th Edition – 1999

- BS 5500 (UK): Specifications for Unfired Fusion Welded Pressure Vessels: 2000 Edition.
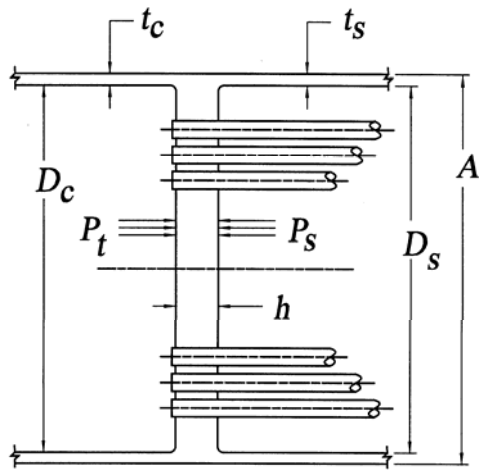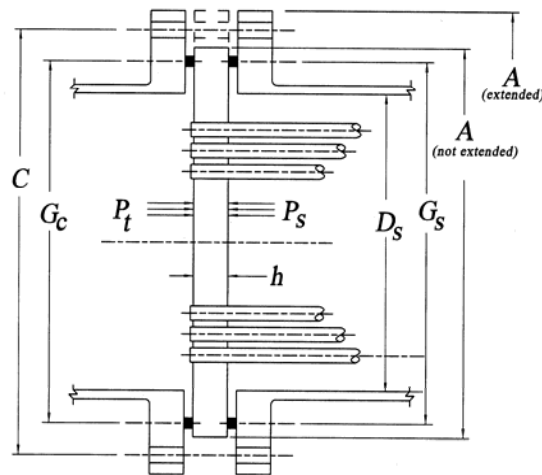
(a) Configuration a:
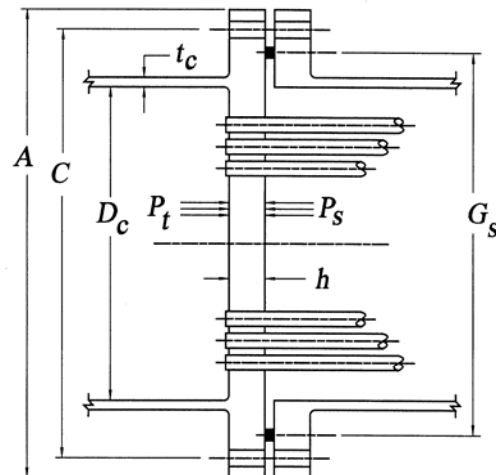Tubesheet integral with shell and channel
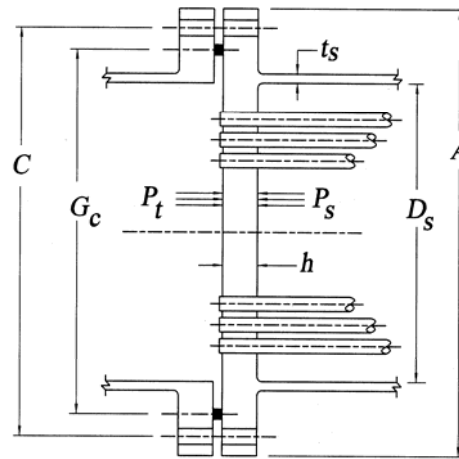
(b) Configuration b:
Tubesheet integral with shell and gasketed with channel, extended as a flange
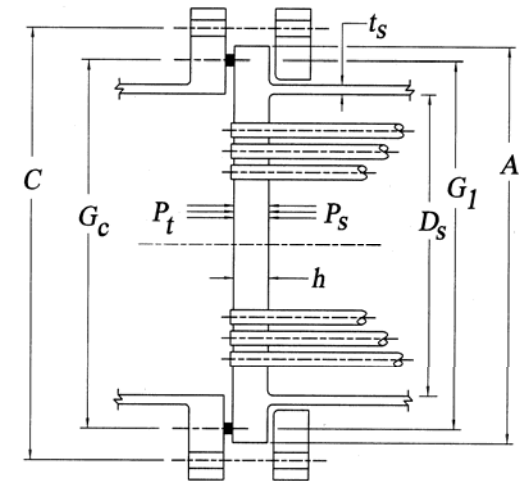
(c) Configuration c:
Tubesheet integral with shell and gasketed with channel, not extended as a flange

(d) Configuration d:
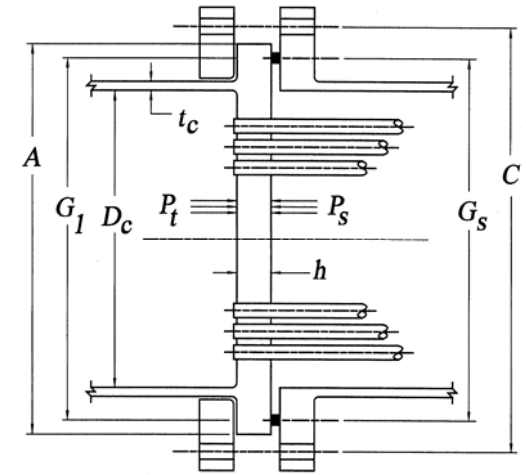Tubesheet gasketed with shell and channel

(e) Configuration e:
Tubesheet gasketed with shell and integral with channel, extended as a flange

(f) Configuration f:
Tubesheet gasketed with shell and integral with channel, not extended as a flange

Figure 1 : Configurations of tubesheets in ASME, CODAP and UPV

Figure 2 : Analytical model for tubesheet integral both sides (configuration a)

Thicknesses (h, T) are in mm						$h_o = C \dfrac{G}{2} \sqrt{\dfrac{P}{\mu^* \Omega S}}$		$T_{new} = F \dfrac{G}{3} \sqrt{\dfrac{P}{\eta S}}$			No direct formula : iterative calculations to obtain the optimized tubesheet thickness h				
EXAMPLE			**LIGAMENT EFFICIENCIES**			**CIRCULAR PLATE FORMULA** (using μ*)		**TEMA** (using η)			**ASME** (using μ*)			(using η)	**CODAP/UPV** (using μ*)	
N°	Config.	Pitch (Δ, □)	μ 1- d_t/p	μ* (ASME) (CODAP) (UPV)	η (TEMA)	h_o Ω = 1.5 (ss, cl)	h_o Ω = 2.0 (ss, cl)	F (ss, cl)	T old	**T new**	F (ss, cl)	h old Ω = 1.5	**h new Ω = 1.5**	h new Ω = 1.5	**h new Ω = 2.0**	h new Ω = 3.0
1	**A**	□	0.25	0.35	0.56	22.0 (cl)	19.1 (cl)	1.21 (ss)	17.3	**15.5**	3.90 (cl)	20.6	**15.0**	11.9	**13.2**	11.0
2	**D**	Δ	0.17	0.28	0.37	40.9 (ss)	35.6 (ss)	1.25 (ss)	29.5	**32.3**	0.43 (ss)	38.1	**37.6**	33.2	**32.5**	26.4
3	**D**	Δ	0.2	0.24	0.42	131.3 (ss)	113.8 (ss)	1.25 (ss)	87.4	**89.7**	0.46 (ss)	124.2	**121.4**	92.2	**105.2**	86.3
4	**E**	□	0.25	0.39	0.56	108.7 (ss)	94.2 (ss)	1.25 (ss)	103.3	**92.2**	0.86 (ss)	109.5	**103.1**	89.4	**87.8**	69.2

ss = simply supported   Δ : triangular pitch
cl = clamped   □: square pitch

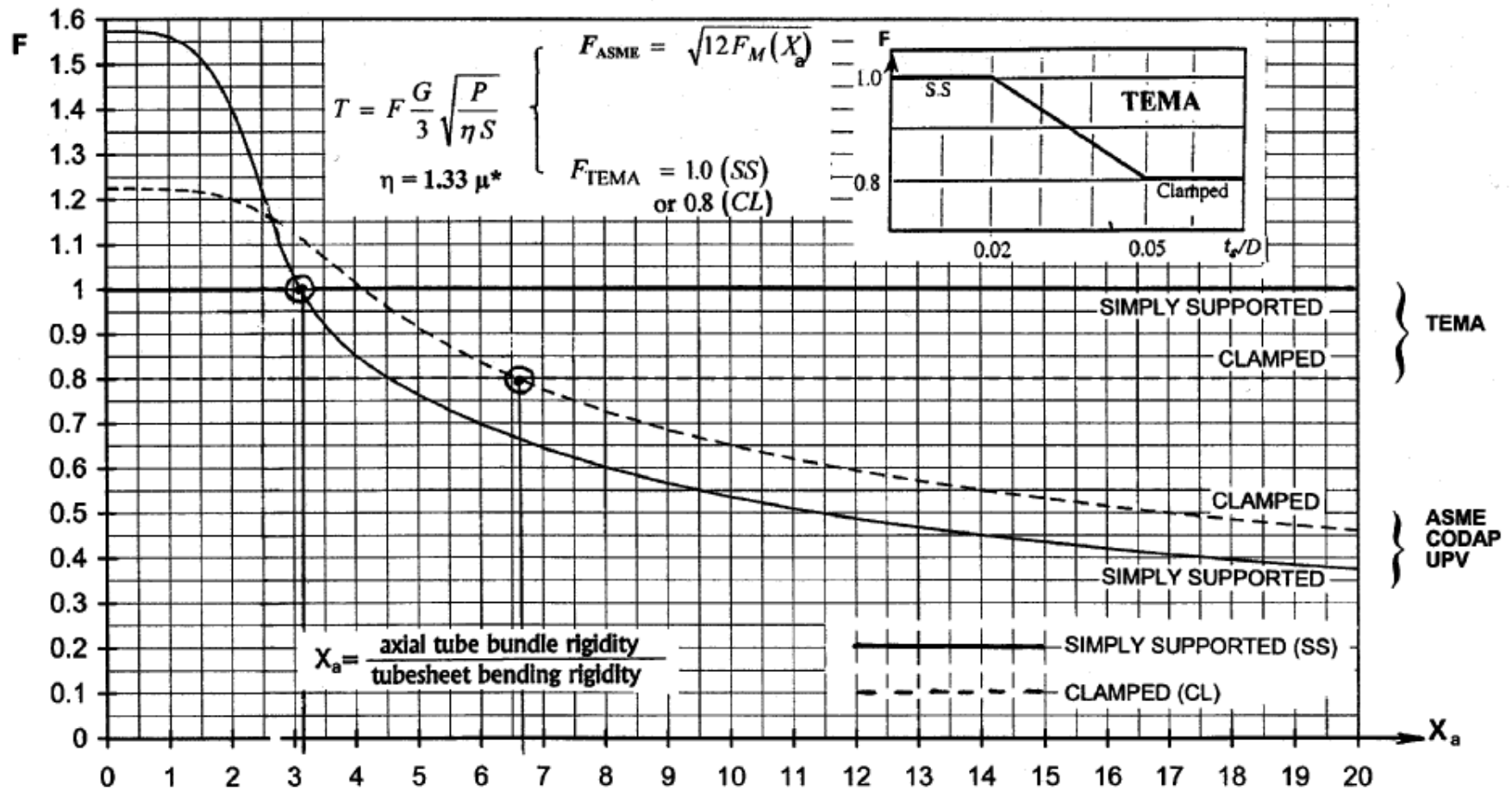**Table 1 : Comparison of TEMA, ASME and CODAP/UPV tubesheet thickness on 4 U-tube Heat Exchangers**

Figure 3 : Comparison of TEMA and ASME-UHX Rules for Fixed Tubesheets

SI Units	DATA					TUBESHEET OPTIMIZED THICKNESS (in mm) (exclusive of corrosion allowance and partition grooves)			
CASE	Dimensions (mm)		Pressures (MPa)	Temperatures (°C)	TEMA 1		ASME 2	CODAP 95 3	CODAP 99-EUPV 4
N° Type	$D \times L$	Yes No	P_t P_s	T_t T_s	X	T (mm)	h (mm)	e (mm)	e (mm)
1 NEN	1524 x 13995	N	23,44 0,0	149 149	1,8	314,6	330,7	360,3	324,0
2 NEN	3251 x 7010	N	6,14 5,0	260 260	7,5	230,2	185,9	169,1	149,1
3 NEN	2743 x 5994	N	5,5 5,2	275 265	8,4	234,9	113,5	126,3	114,4
4 NEN	2565 x 1702	N	-0,1 0,34	185 185	41,1	43,2	5,8	12,0	11,8
5 NEN	2007 x 4775	Y	0,55 0,86	204 232	19,2	77,3	27,4	25,4	23,2
6 NEN	635 x 4877	Y	0,0 1,38	177 93	14,6	11,9	14,2	7,6	6,0
7 AEL	1803 x 4572	N	0,28 0,28	149 149	12,3	66,6	26,2	33,3	34,0
8 AEL	914 x 4267	Y	2,76 1,03	288 288	5,4	80,8	60,2	55,2	49,6
9 AEL	610 x 5486	Y	1,72 2,59	135 186	4,0	49,5	25,9	39,4	36,3
10 AEL	152 x 3658	Y	0,86 0,86	49 163	2,5	16,7	7,4	11,4	10,5

Rows 1–6 are grouped under **INTEGRAL CHANNEL**; rows 7–10 are grouped under **GASKETED CHANNEL**.

Table 2 : Fixed tubesheet heat exchangers

IV.   Significance of Calculated Stresses

This Section discusses the significance of the stresses in a heat exchanger at the tubesheet/shell/channel in order that the heat exchanger engineer may correctly apply the Part UHX rules. In order to understand the significance of the determined stresses, one has to understand the difference between primary stresses and secondary stresses and how each may have an influence on the design.

The tubesheet rules of Part UHX are intended to generally follow the stress classification of Section VIII, Div. 2 for primary and primary plus secondary stresses. The definitions of primary and secondary stresses are defined in Section VIII Division 2, Appendix 4. The scope of Part UHX does not include any consideration of peak stresses or any requirements for fatigue.

The bending stress resulting from a pressure loading in a flat plate is a primary bending stress. Any yielding of the plate material results in a permanent deformation, and the deformation may continue to occur until the plate fails (or the deformation is so large that the plate cannot perform its intended function). When the tubesheet is extended as a flange, the loading imposed by the bolting moment is considered primary. Thus, primary bending stress limits are appropriate when considering the tubesheet bending stress resulting from pressure loading acting alone or in combination with the bolt loading when applicable.

The stresses in the shell and channel are somewhat more complex to categorize. The axial membrane stresses (the average stress across the thickness) in the shell and channel remote from the tubesheet resulting from pressure loadings are primary.

The bending stress at the shell-to-tubesheet juncture and the channel-to-tubesheet juncture result from restrained differential motion at these junctures. As such, these bending stresses have the basic characteristic of a secondary stress.   However, there is a very important distinction that has to be made regarding these stresses. This distinction is clearly defined by footnote 2 of Table 4-120.1 of Section VIII Div. 2. The bending stress at a shell to flat plate juncture may be defined as secondary unless the discontinuity bending moment at the edge of a flat plate is required to maintain the bending stress elsewhere in the plate to within its allowable stress. In this instance, the shell/channel bending stress is classified as primary bending and should be limited to the primary bending stress limit.

When an elastic stress analysis includes the rotational stiffness of the shell and channel in determining the tubesheet stress under primary loading, the discontinuity bending stress should be categorized as primary bending stress and be limited accordingly.

However, there may be instances where the design need not consider the full strengthening effect of the shell/channel. For example, if one chose to not include the stiffening effect of the shell and channel for the tubesheet analysis, the shell and channel bending stresses could be correctly categorized as secondary and be limited according to the secondary stress considerations.

If the shell/channel bending stresses resulting from pressure/bolting loads only do not satisfy the primary stress limits, then full credit cannot be taken for the stiffness of that component. In such cases, it is deemed appropriate to apply a "knockdown" factor to the stiffness of the shell or channel component by reducing its modulus of elasticity when the primary bending stress criteria has not been satisfied. The "knockdown" factor used in the Part UHX "Elastic-Plastic" analysis is based on evaluations of the extent of strengthening offered by the fully plastic moment at the shell and channel junction when the primary stress limits are not satisfied.

Thus, when considering pressure/bolting loads only, the shell/channel stresses are considered primary bending stresses in the initial calculation. If the shell/channel stress membrane plus bending stress does not meet the primary bending stress allowable, then the engineer may, at his option:

- conduct simplified, pseudo elastic-plastic analysis to determine if reduced strengthening effect results in acceptable tubesheet design.
- increase shell/channel thickness adjacent to tubesheet
- increase tubesheet thickness to lower shell/channel stress
- increase the tubesheet thickness and shell/channel thickness

The stresses resulting from the temperature differential are secondary in that they are self-limiting. The Code limits on secondary stress are derived to accomplish "shakedown to elastic action". The Part UHX rules consider the tubesheet, shell, channel, and tube stresses to be secondary stresses under the action of thermal load.

It has been a long, standing practice of TEMA to divide the loads resulting from thermal expansion by a factor of two, including the tube loads. Thus, the TEMA allowable tube tensile stress for thermal load cases is effectively increased by a factor of two. This practice has not led to any noted problems or deficiencies in allowed tube loads, and this practice is continued for the Part UHX rules for allowable tensile stress when considering any thermal load conditions. The maximum axial compressive load in Part UHX parallels that as given in TEMA. For compressive stresses, tube buckling may restrict the tube's load carrying ability. This is true for either pressure or thermal load conditions. If a substantial number of tubes were above their buckling limit, it is possible that the bundle could not sustain the required loading. For this reason, no distinction is made between primary and secondary allowable compressive loads in the tubes.

V. How to Use the Rules

Based on the above discussion, the rules of Part UHX for tubesheet should generally be used in the following manner.

- Establish the geometry for the initial set of calculations.
- Calculate the tubesheet stress based on the full support of an integrally attached shell and/or channel using Load Cases 1, 2 and 3. (See Part UHX for definition of Load Cases. Load Cases 1,2, and 3 consider pressure loads only.)
- Once the tubesheet stress is shown to be acceptable using an allowable stress based on 1.5 S (where S is the basic allowable stress from Section II Part D), calculate the shell/channel stresses as appropriate for Load Cases 1, 2, and 3.
- If the shell/channel stresses do not exceed their respective allowable stress based on 1.5 S, then the geometry is acceptable for Load Cases 1, 2, and 3, and it is not necessary to conduct a plastic analysis.
- If the shell/channel stress exceeds its primary stress allowable of 1.5 S, but less than the secondary stress allowable (greater of 3S or $2S_y$), then the simplified elastic-plastic analysis may be used for Load Cases 1, 2, and 3. For the elastic-plastic analysis, it is required that the tubesheet stress be less than the primary bending allowable stress (of 1.5 S) after reducing the strengthening effect of the adjacent cylinder(s). It is not required to recheck the shell/channel stress for the elastic-plastic analysis. If the tubesheet stress from the elastic-plastic analysis exceeds the allowable primary bending stress, then a new geometry shall be established and it is necessary to start over at the first step.
- If the shell/channel stress exceeds its primary plus secondary stress limit (larger of 3S or $2S_y$), then the geometry is not acceptable and must be revised. It will then be necessary to start over and return to the first step
- Determine the tube loading for Load Cases 1, 2, and 3. The maximum tube stress shall not exceed its allowable in either tension or compression using the primary stress limits.
- For Load Cases 4, 5, 6, and 7, (which consider pressure plus restrained differential thermal expansion) determine tubesheet, shell/channel, and tube stresses using the unaltered elastic properties for the shell and/or channel (use the elastic analysis parameters). If any of the stresses exceed their respective allowable stress of the largest of 3S or $2S_y$, then the geometry is not acceptable and shall be reconsidered and it is necessary to restart and return to the first step. Use the allowable buckling stress for determining the tube allowable compressive stresses. If the allowable stresses are satisfied, then the design is considered acceptable.

Because of the complexity of the procedure in Appendix AA of the Code, it is likely that users will computerize the solution. The criteria and logic presented in this Section will facilitate the understanding of the correct application of the rules.

VI.   Effect of Ligament Efficiency in Calculation Procedure

The TEMA and Part UHX tubesheet design methods both define and use a ligament efficiency. The TEMA method is based on the average width of the ligament between the tube holes, and is different for triangular pitch vs. square pitch. The Part UHX method uses the minimum ligament width; however, if the tubes are expanded into the tubesheet, then tube wall may be considered as part of the effective ligament.

For this purpose, the Part UHX calculation procedure defines the effective tube hole diameter d^*, used to calculate μ^* (the effective ligament efficiency) as follows.

$$\mu^* = \frac{p - d^*}{p}$$

$$d^* = MAX \left\{ \left[d_t - 2t_t \left(\frac{E_t}{E} \right) \left(\frac{S_t}{S} \right) \rho \right], \left[d_t - 2t_t \right] \right\}$$

In these equations:

μ^* = effective ligament efficiency
p = tube pitch, in.
d^* = effective tube hole diameter, in.
d_t = nominal outside diameter of tubes, in.
t_t = nominal tube wall thickness, in.
E_t = modulus of elasticity for tube material at design temperature, psi
E = modulus of elasticity for tubesheet material at design temperature, psi
S_t = allowable stress for tube material at design temperature, psi For a welded
        tube, use the allowable stress for an equivalent seamless tube, psi
S = allowable stress for tubesheet material at design temperature, psi
ρ = tube expansion depth ratio = ℓ_{tx}/h , $(0 \le \rho \le 1)$
ℓ_{tx} = expanded length of tube in tubesheet $(0 \le \ell_{tx} \le h)$
h = tubesheet thickness, in.

The Part UHX calculation procedure also takes into account differences in material properties of the tube and tubesheet. It allows the Manufacturer to take advantage of the stiffening effect of a tube expanded into a tubesheet for all tubesheet configurations, whether U-tube or straight tube.

To illustrate the difference between the TEMA method and the Part UHX methods for determining ligament efficiency, consider a 1 inch diameter, 16 BWG tube that is the same material as the tubesheet. If the tube is expanded the full depth of the tubesheet, then $\mu^* = 0.304$, but if the tube is welded and not expanded at all, then μ^* = 0.20. However, the TEMA ligament efficiency is 0.420 and 0.498 for triangular and square pitch layouts respectively regardless of whether the tubes are expanded or not.

The ligament efficiency has a direct bearing on the calculated tubesheet stress. A smaller ligament efficiency results in a larger predicted tubesheet stress and a larger ligament efficiency results in a smaller predicted tubesheet stress. Thus, as may be seen, if the same basic theory is used to determine the stress in a plate, then the TEMA ligament efficiency would result in a smaller calculated stress as compared to the ASME method, even when the full tube wall is considered. This difference is exacerbated when the tube is not expanded. The Part UHX method for determining ligament efficiency is consistent with that found in Section VIII Division 2 Appendix 4. Inclusion of the tube wall in contributing to the ligament efficiency is considered a rational consideration as opposed to using a greater ligament efficiency that does not consider how the tube is joined to the tubesheet.

In order to maintain joint integrity, the Manufacturer's design and tube expanding procedure must be adequate to ensure that there will be tube-hole contact under all operating conditions such as start-ups, shut-downs, normal operation, and upsets. Therefore, action being developed under Item BC02-2366 will require that Manufacturers have written, qualified expanding procedures for joints (whether welded and expanded or expanded only) to demonstrate that the expanded joint is capable of providing the required properties for its intended application. Qualification under BC02-2366 shall consist of the Manufacturer demonstrating to the Authorized Inspector a record of having produced satisfactory expanded joints using an existing written procedure, or by shear-load testing specimens produced using a proposed procedure.

VII. Tube Stresses and Tube-to-Tubesheet Joint Loads

VII.1 General

Both ASME and TEMA require that the design of the tubes and tube joints consider various loading cases. For the calculation of the tube stresses and the tube joint loads, the ASME (TEMA) loading cases are correlated as follows using P_t (P_2) for the tube side pressure, P_s (P_3) for the shell side pressure, and P_γ (P_d) for the pressure due to the differential thermal expansion γ:

ASME Loading Case	ASME Loads	TEMA RCB-7.23 P_t^*
1	P_t	P_2
2	P_s	$-P_3$
3	P_t, P_s	$P_2 - P_3$
4	P_γ	P_d
5	P_t, P_γ	$P_2 + P_d$
6	P_s, P_γ	$-P_3 + P_d$
7	P_t, P_s, P_γ	$P_2 - P_3 + P_d$

ASME Loading Cases 1, 2 and 3 consider only the effects of pressure loading (TEMA P_2 & P_3) and are referred to as the **pressure loading** cases. For tube-to-tubesheet joint loads (UHX-15), ASME also describes these cases as those due to pressure-induced axial forces.

ASME Loading Cases 4, 5, 6 and 7 include the effects of thermal expansion P_γ (TEMA P_d) and are referred to as the **thermal loading** cases. For tube-to-tubesheet joint loads, ASME also describes these cases as those due to thermally-induced or pressure plus thermally-induced axial forces.

VII.2 Tube Stresses

1) TEMA

The TEMA calculation for the tube longitudinal stress S_t at the periphery of the tube bundle is given in RCB-7.23.

When the tubes are in tension (P_t^* positive), and a thermal loading case (the equivalent differential expansion pressure P_d is included in the P_t^* term) is being considered, the calculated tensile stress is divided by two ($C_t = 0.5$).

TEMA requires that the calculated stress S_t not exceed 1.0 times the Code allowable tensile stress. By dividing the longitudinal tensile stress S_t by two for the thermal loading cases, TEMA effectively permits the allowable stress to be 2.0 times the Code allowable tensile stress for the thermal loading cases.

When the tubes are in compression (S_t negative), the stress must not exceed the buckling stress limit (allowable tube compressive stress) calculated in RCB-7.24.

2) ASME

The ASME calculation for the tube longitudinal stress $\sigma_{t,o}$ is given in UHX-13.5.9 for fixed tubesheet heat exchangers and UHX-14.5.9 for floating tubesheet heat exchangers.

The ASME equation does not require a C_t term because $\sigma_{t,o}$ is compared to 1.0 times the allowable stress for the pressure loading cases and to 2.0 times the allowable stress for cases that include thermal loading.

When the tubes are in compression ($\sigma_{t,o}$ negative), the stress must not exceed the buckling stress limit S_{tb} calculated in UHX-13.5.9(b) or UHX-14.5.9(b).

CONCLUSION

The TEMA and ASME calculated tube stresses will not yield the same results due to differences in the their tube loading models, but both compare these stresses to the same allowable stress limits.

TEMA and ASME calculate the buckling stress limit (allowable tube compressive stress) the same; however, F_q may be different because it is calculated using a different model. Both TEMA and ASME do not permit the allowable buckling stress to exceed 1.0 times the Code allowable stress.

Note that ASME paragraphs UHX-13.4(b) and UHX-14.4(c) permit the allowable stresses for the thermal loading cases to be taken at the operating temperature.

TEMA and ASME only calculate the tube longitudinal stress and the buckling stress limit at the periphery of the tube bundle, because it is presumed that this is the location of the highest tube stress. SWGHTE, under BC94-439, is working on developing the maximum stress of the interior tubes, because it has been shown that sometimes the interior tube stress is higher than the peripheral tubes stress. CODAP presently has rules for calculating the interior tube stress.

VII.3   Tube-to-Tubesheet Joint Loads

1) TEMA

The TEMA calculation for the tube-to-tubesheet joint load W_j at the periphery of the tube bundle is given in RCB-7.25. This load only considers the effect of pressure; because, TEMA assumes that the joint loads caused by the thermal loading are within acceptable limits if the tube stresses calculated for pressure loading are within the allowable stress limits. This assumption could lead to joint loads for the thermal loading cases that are greater than the joint strength.

TEMA directs the user to calculate the allowable joint load in accordance with the Code or by other means.

2) ASME

ASME does not give the tube joint load equation, but it can easily be calculated as follows:

$$W_j = \sigma_{t,o} A_t$$

W_j =   tube-to-tubesheet joint load, lb (N)
$\sigma_{t,o}$ =   axial tube stress from UHX-13.5.9 or UHX-14.5.9, psi (MPa)
A_t =   tube cross-sectional area, in. (mm)

The ASME Code specifies the allowable tube joint load L_{max} in either UHX-15 (formerly UW-20) for strength welded tube joints or Appendix A for all other tube joints. For the pressure loading cases, L_{max} cannot exceed 1.0 times the tube joint strength. For the thermal loading cases, L_{max} cannot exceed 2.0 times the tube joint strength; except, that for welded-only tube joints where the thickness through the throat of the weld is less than the nominal tube thickness, L_{max} is limited to 1.0 times the tube joint strength for **all** loading cases.

ASME uses the following guidelines:

a) For full strength welds meeting the requirements of UHX-15.4, the tube joint strength is tube strength F_t.

b) For partial strength welds meeting the requirements of UHX-15.5, the tube joint strength is the weld design strength F_d, which is the fillet weld strength F_f, or the groove weld strength F_g, or a combination of both ($F_f + F_d$).

c) For partial strength welds meeting the requirements of UW-18(d), the tube joint strength is the allowable load on the fillet weld.

d) For all other tube joints, the tube joint strength is the allowable load for tube-to-tubesheet joints as established by meeting the requirements of Appendix A.

CONCLUSION

The ASME method permits the designer to calculate the tube joint load for every loading case, whereas the TEMA method only requires this calculation for pressure loading.

TEMA assumes that the tube joint strength will be at least equal tube joint load for all loading case just by meeting the ASME allowable load criteria for the pressure loading cases. A review of the ASME allowable tube joint load described above shows that the tube joint loads should be calculated for every loading case and properly compared with the corresponding allowable tube-to-tubesheet joint load.

VIII. In-service Implications for Existing Exchangers

There has been some concern expressed regarding the in-kind replacement of heat exchanger bundles if new rules are adopted that are different from those of the original fabrication.

Repairs and alterations are not affected by UHX, because the Codes and Standards in effect at the time of construction may be used. In substantiation, it is noted that Paragraph RC-1020 of the National Board Inspection Code (NBIC) states:

"When the standard governing the original construction is the ASME Code, repairs and alterations shall conform insofar as possible to the section and edition of the ASME Code most applicable to the work planned."

The rules of nearly all North American political jurisdictions require repairs and alterations to be made in accordance with either the NBIC or API-510 Pressure Vessel Inspection Code: Maintenance, Inspection, Rating, Repair, and Alteration. API 510 has provisions similar to those of the NBIC. These Inspection Codes are used worldwide.

For replacement heat exchanger bundles in-kind, it is a common and acceptable practice for an owner to ship a drawing of a tube bundle (or for that matter, a drawing of a shell section) to fabricator with no consideration at all given to whether design calculations are required or not. The fabricator builds the bundle (or part of a shell) to the current code rules, and, if needed, applies a Part Stamp and sends it back to the owner who fills out the necessary post construction paper work to the satisfaction of the jurisdiction. (The Part Stamp holder is not required by Section VIII to certify any design.) This does not violate any ASME requirements and is, in fact, what the NBIC suggests be done. Since a U Stamp is not normally required for replacement heat exchanger bundles, Section VIII does not prohibit "fabrication to the drawing" because design calculations are required by Section VIII for Parts or materials. For Section VIII construction, the design responsibility belongs to the U Stamp Certificate holder; however, for a post construction replacement bundle, there is not necessarily a U Stamp Certificate holder involved.

If there is a replacement bundle that requires a U Stamp, then all mandatory rules, including the design rules, in effect must be satisfied. This is not any different than for any other component or piece of equipment going into an operating facility and any differences in geometry from the old to the new has to taken into consideration.