A PROPOSED EXTENSION OF THE TEMA TUBESHEET DESIGN METHOD TO DETERMINE TUBESHEET RIM THICKNESS

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ABSTRACT

A method to compute the thickness of tubesheet extension (rims) used to serve as a flange is proposed. The method treats both fixed/floating head stationary tubesheet and U-tube tubesheets. Numerical examples illustrate the calculation method.

NOMENCLATURE

A: Tubesheet O.D.

d_o: Tube O.D.

F: Edge condition multiplier as defined by TEMA [1].

G: TEMA effective gasket diameter

M: Generic term of edge moment.

P: Generic term for pressure

P_s: Shellside design pressure

P_t: Tubeside design pressure

p: Layout pitch

r: Ratio of A to G

S_R: Stress in the tubesheet rim

S: Code allowable stress for the tubesheet material at design temperature

T: Tubesheet perforated interior thickness

T_r: Thickness of the tubesheet rim

w: Rim width [$w = (A-G)/2$]

W: Total pressure load at diameter G (W is equal to $\pi PG^2/4$

η: Tubesheet solidity factor (deflection efficiency)

1. INTRODUCTION

The current edition of TEMA [1] does not give a procedure to compute the minimum required thickness of the portion of the tubesheet extended to serve as a flange. The contribution of the flange extension in reducing (occasionally increasing) the maximum tubesheet stress in the perforated interior is incorporated in the current TEMA formula for fixed tubesheets, but not for tubesheets in U-bundle construction. Our objective herein is to

(i) Develop a conservative approach to define a minimum outer ring thickness for U-tube, fixed tube, and floating tubesheet heat exchangers, consistent with the TEMA design philosophy.

(ii) Provide for a consistent method to include the contribution of the outer ring to the thickness calculation of the tubesheet proper for U-bundle tubesheets.

Three conditions of geometry cover all possible edge support conditions for the stationary tubesheets of heat exchangers.

(i) Tubeside gasketed, shellside integral

(ii) Shellside gasketed, tubeside integral

(iii) Both sides gasketed

Since a heat exchanger tubesheet is subject to both shellside and tubeside pressures, the computations suggested by the following development should be carried out independently for both tubeside and shellside pressures, unless a "differential pressure only" design is specified.

We assume that the reader has a working level acquaintance with the TEMA Standards, and is generally familiar with the tubesheet design methods. Readers in need of such a background may consult refs. [1,2].

In the next section, working formulas to determine the tubesheet rim thickness are presented along with worked out numerical examples. The derivations of these formulas is given in subsequent sections for completeness of presentation.

The formulas are presented with reference to the tubesheet bending equation reformatted by the TEMA tubesheet technical subcommittee for possible adoption in the seventh edition of the TEMA Standards.

Since the TEMA tubesheet design formula predates the plate theory solutions which treat the tubesheet problems in a consistent manner [2, Chapters 8-11], our effort herein is directed towards developing tubesheet rim design formulas within the spirit of TEMA methodology. This has meant recourse to semi-empirical deductions which do not have a rigorous basis. The method does have the support of early works of Gardner, Miller, and others.

2. CALCULATION PROCEDURE

The calculation procedure for fixed and floating head exchangers is different from that for U-tube exchangers.

2.1 Fixed or Floating Head Exchangers:

The thickness of the portion of the tubesheet extended as a flange, t_r, is given by

$$T_r = 0.98 \left[\frac{M\,(r^2 - 1 + 3.72\,\ln r)}{S(A-G)\,(1.0 + 1.86 r^2)} \right]^{1/2} \qquad (2.1)$$

where M is the seating condition or operating condition moment, whichever is greater. A and G are tubesheet O.D., and effective gasket diameter, respectively, and S is the Code allowable stress at the pertinent design temperature. The quantity r is simply the ratio of A to G; i.e.,

$$r = A/G \qquad (2.2)$$

The quantity G has different values for different types of construction. For example, G is the shell I.D. for fixed tubesheet units. G is the shell gasket mean diameter for floating head units when shellside pressure is considered, but channel I.D. for welded channel/tubesheet construction when the tubeside pressure is considered. The user should refer to the TEMA Standards for the appropriate definition of G. Other terms are defined in the nomenclature.

It should be noted that the stationary tubesheet in a floating head exchanger may be gasketed on both sides. In such a case, M is given by

$$M = \text{Greater of } \left[\text{Abs}\,(M_{so} - M_{cs}),\ \text{Abs}\,(M_{co} - M_{ss}) \right] \qquad (2.3)$$

where:

M indicates moment and the subscripts have the following meanings.

First subscript:

- s: shellside
- c: channel side

Second subscript:

- s: seating condition
- o: operating condition

However, if a differential pressure only design is required, then

$$M = \text{Abs}\,(M_{so} - M_{co}) \qquad (2.4)$$

2.2 U-Tube Exchangers

The proposed TEMA tubesheet bending formula to determine the required thickness of the perforated region is [3]

$$T = \frac{FG}{3} \left(\frac{P}{\eta S} \right)^{1/2} \qquad (2.5)$$

where P is shell or tubeside design pressure, and G is the corresponding effective gasket diameter. η is a term which accounts for the tubesheet solidity.

$$\eta = 1 - \frac{0.785}{(p/d_o)^2} \quad \text{(for square or rotated square pitch)} \qquad (a)$$

$$\eta = 1 - \frac{.907}{(p/d_o)^2} \quad \text{(for triangular or rotated triangular pitch)} \qquad (b)$$

(2.6)

It is herein proposed to modify the definition of P in Equation (2.4) by adding to the design pressure a term P_b due to edge moment, i.e.

$$P = P_s + P_b \text{ or } P_t + P_b \qquad (2.7)$$

where P_b is defined as follows:

$$P_b = -\frac{6.2M^*}{F^2 G^3} \qquad (2.8)$$

$$M^* = \frac{\frac{.069w}{n} F^3 P G^3 [\frac{T_r}{T}]^3 - M G - .39wPG^3}{G + \frac{1.37}{n} [\frac{T_r}{T}]^3 w} \qquad (2.9)$$

where P is either P_s or P_t.

Like F and G, edge moment is defined differently depending on whether shellside or tubeside pressure is being considered.

When shellside pressure is considered:

$$M = M_{so} - M_{cs}$$

when tubeside pressure is considered:

$$M = M_{co} - M_{ss}$$

If a joint is integral (welded connection) then the corresponding edge moment is zero. Finally, for differential pressure designs, M is the operating condition edge moment for the controlling pressure side less the operating condition moment for the other side.

The stress in the tubesheet rim due to flange action is:

$$T_r = 1.38 \left[\frac{M^* + M + .39 P G^2 w}{(A-G) S} \right]^{1/2} \qquad (2.10)$$

where P is either P_s or P_t.

For each pressure loading, the calculation procedure is as follows:

(i) Compute M^* from Eq. (2.9) assuming $T_r = T$ (rim and interior thickness equal)

(ii) Compute P_b from Eq. (2.8) and P from Eq. (2.7)

(iii) Compute the required tubesheet thickness T from Eq. (2.5) and required rim thickness from Eq. (2.10). If $T > T_r$, and both are set equal to the computed T, then the calculation may be terminated at this point.

(iv) If T_r exceeds T, or if it is desired to reduce the rim thickness below T, a new value of T_r may be selected, and used to compute a new M^* in Eq. (2.9). Next, the calculation returns to step (ii) above.

The numerical example in the next section illustrates the procedure.

Subscripts

Subscripts on M have the following meanings:

	s	c	o
First subscript	Shellside	Channel side	——
Second subscript	Seating condition	——	Operating condition

3. DERIVATION OF THE WORKING EQUATIONS

3.1 Fixed or Floating Head Exchanger:

The role of the tube bundle in providing support to the tubesheet in the manner of an elastic foundation is well-known and extensively documented in the literature. The presence of elastic foundation reduces tubesheet deflection and slope in stayed tubesheets to a fraction of what would develop in a U-bundle tubesheet under identical lateral loadings. Therefore, for the purpose of evolving the tubesheet rim thickness formula, the tubesheet may be assumed to be clamped at the shell inside diameter, G.

The flange action moment M can be replaced by a uniformly distributed lateral load W along the outer circumference of the tubesheet, such that

$$M = \frac{W (A-G)}{2} \qquad (3.1)$$

The solution for a circular plate, clamped at diameter G, and subject to a uniformly distributed lateral load W at the outer circumference is available in standard texts [4]. This leads to the expression

$$S = \frac{3M}{\pi T_r^2 (A-G)} \left[\frac{3.72 \ln r + r^2 - 1}{1.0 + 1.86 r^2} \right] \qquad (3.2)$$

where the Poisson's ratio is assumed to be 0.3.

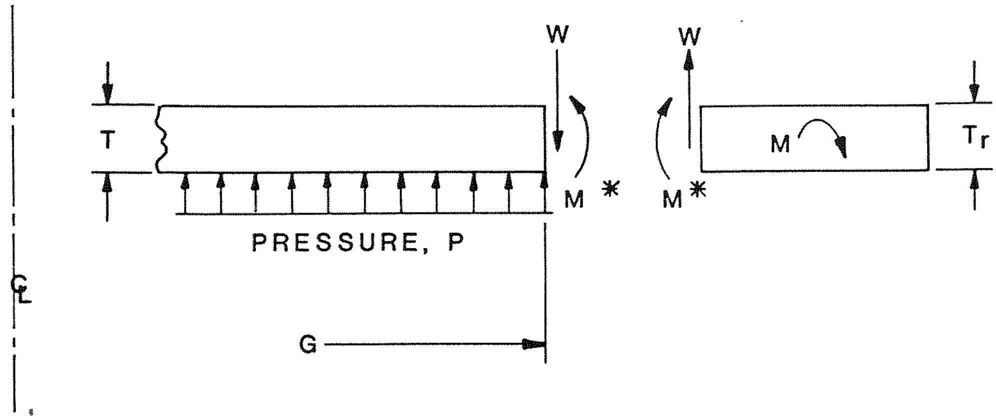

FIG. 3.1 TUBESHEET FREE BODY

Rewriting Eq. (3.2) with T_r on the left hand side gives Eq. (2.1)

3.2 Tubesheets for U-bundle units:

Fig. 3.1 shows the free body of the tubesheet. The rim is shown subjected to an edge moment M. The perforated interior of the tubesheet is modelled by an equivalent solid plate of modified Young's Modulus E^* and Poisson ratio ν^* [2, Chapters 8 or 5]. Requiring continuity of slope at the interface between the perforated region and the rim yields

$$\frac{c P G^3}{64 D^*(1 + \nu^*)} - \frac{M^*}{2\pi D^* (1 + \nu^*)} = \frac{G \left[M + M^* + \frac{\pi P G^2 w}{8}\right]}{4 \pi E I} \qquad (3.3)$$

$$\text{where } D^* = \frac{E^* T^3}{12 (1 - \nu^{*2})} \qquad (3.4)$$

In the above equation, the rotation of the rim is written using narrow ring theory. The term c represents the rotational restraint applied on the tubesheet due to its attachment to shell or channel (if such an integral attachment exists) Solving for the interface moments M^*, we have

$$M^* = \frac{\frac{\pi E I c P G^3}{16 D^* (1+\nu^*)} - M G - \frac{\pi P G^3 (A-G)}{16}}{G + \frac{2 E I}{D^* (1 + \nu^*)}} \qquad (3.5)$$

$$M^* = \frac{\frac{w f c P G^3}{16} \left(\frac{T_r}{T}\right)^3 - M G - \frac{\pi P G^3 w}{8}}{G + \frac{2 f}{\pi} \left(\frac{T_r}{T}\right)^3 w} \qquad (3.6)$$

where

$$f = \frac{\pi E (1-\nu^*)}{E^*} \qquad (3.7)$$

It remains now to express f in terms of the solidity factor η and c in terms of the edge restraint factor F.

We note that the deflection of a laterally loaded circular plate of Young's Modulus E, Poisson's ratio ν, O.D. A, and thickness T is:

$$\delta = \frac{3 W^* (1 - \nu) (5 + \nu) A^2}{64 \pi E T^3} \qquad (3.8)$$

where W^* is the uniformly distributed lateral load.

Since η is, in essence deflection efficiency, the deflection of the perforated plate is δ/η. This must equal the deflection of the identical plate with Young's modulus E^* and Poisson ratio ν^*. Hence

$$\frac{3 W^* (1 - \nu) (5 + \nu) A^2}{64 \pi E T^3 \eta} = \frac{3 W^* (1-\nu^*)(5+\nu^*) A^2}{64 \pi E^* T^3}$$

$$\frac{E^*}{E} = \frac{(1 - \nu^*)(5 + \nu^*)}{(1 - \nu)(5 + \nu)} \eta$$

TEMA tubesheet formulas are known to correlate well with $\nu^* = 0.435$ which leads to (for ν=0.3)

$$\frac{E^*}{E} = 0.827 \eta$$

Hence

$$f = \frac{\pi (1-\nu^*) E}{E^*} = \frac{2.15}{\eta} \qquad (3.9)$$

Finally, the expression for the non-simple support edge modifier is derived by noting that the edge restraint effect is built into the F factor in the TEMA formula.

The F-factor is 1.25 for gasketed (simply supported edge condition) and linearly goes to 1.0 for integral tubesheet (pressure part connection. Thus, in effect, the TEMA tubesheet formula replaces the tubesheet diameter term by

$$\left(\frac{FG}{1.25} \right)$$

which reduces to G for gasketed construction, and to a value less than or equal to G for integral construction. Since edge slope of a simply-supported laterally loaded plate varies as the third power of diameter, it is concluded that the factor c in Eq. 3.3 should be given by

$$\left(\frac{F}{1.25} \right)^3$$

or $c = 0.512 F^3$ (3.10)

Substituting for f (Eq. 3.9), and c (Eq. 3.10) into the equation for M^* (Eq. 3.6) gives the final form for design work (Eq. 2.9).

4. APPLICATION OF THE CALCULATION METHOD

Let us consider a tubesheet bolted to a channel flange. The following data defines the tubesheet design problem:

Channel design pressure = 200 psig
Channel design temperature = 300°F
Tubesheet and mating flange O.D., A = 35.625"
Bolt circle diameter = 34.0"
Gasket effective diameter, G (as defined by the ASME Code) = 32.012"
Shell I.D. = 30.625"
Tube hole layout is rotated square.

Both the tubesheet and flange materials have Code allowable stress S = 17500 psi at ambient and design temperatures.

There are 32 body bolts, 3/4" nom. diameter, made of SA193-B7 material (S = 25000 psi).

The mating flange (weld neck type) is designed by computer code FLANGE [2, Chapter 3]. The output which lists all pertinent geometric data for the flange assembly is presented in Table I.

The seating condition and operating condition flange moments are given in Table 1 to be

$M_{co} = 0.29 \times 10^6$ lb-inch

$M_{cs} = 0.22 \times 10^6$ lb-inch

We will now compute the required tubesheet rim thickness, if the tubesheet is part of a fixed or floating tubesheet unit and if it is of U-tube unit.

(i) Tubesheet for fixed or floating exchanger:

Referring to Eq. (2.1):

A = 35.625", G = 30.625"†, hence A-G = 5"

and r = 1.163"

Therefore,

$$\frac{r^2 - 1 + 3.72 \ln r}{1.0 + 1.86 r^2} = \frac{0.916}{3.516} = 0.261$$

Hence, Eq. (2.1) yields

$$T_r = 0.98 \left[\frac{0.29 \times 10^6 \times 0.261}{(17500)(5.0)} \right]^{1/2}$$

or $T_r = 0.91"$.

The required rim thickness of 0.91" contrasts favorably with the required mating welding neck flange thickness of 1.64" (see Table I).

(ii) Tubesheet for U-tube exchanger:

Assuming that the tubesheet is integrally welded to the shell, and its dimensions are given by the preceding data for the fixed tubesheet example, we compute the required preliminary perforated region thickness by Eq. (2.5)

† According to TEMA [1] G is shell inside diameter in fixed tubesheet design exchangers.

TABLE I

****PROGRAM FLANGE REVISION 1, APRIL 16,1984****

DESIGN PRESSURE(PSI)= 200.00

	ALLOW.STRESSES(PSI)	
	AMBIENT TEMP.OF	DESIGN TEMP.OF
FLANGE	17500.00	17500.00
BOLTING	25000.00	25000.00

FLANGE GEOMETRY DATA(INCH)

INNER DIA-----------=	30.1250	OUTER DIA--------=	35.6250
EFFECTIVE GASKET DIA=	32.0120	BOLT CIRCLE DIA--=	34.0000
SHELL THICKNESS-----=	.3750	MAX HUB THICKNESS=	.7500
LENGTH OF HUB-------=	1.5000		

GASKET DATA
WIDTH(INCH)---------------= .750
DESIGN SEATING STRESS(PSI)= 3200.
GASKET FACTOR-------------= 3.00

FLANGE IS A WELD NECK

RIB AND INNER RING SEATING FORCE= 0.
GASKET SEATING FORCE ----------- = 98537.

BOLTING DATA
NUMBER-------= 32.
NOM DIA(INCH)= .750

THE NECESSARY TOTAL CROSS SECTIONAL AREA = 7.9169
THE ACTUAL TOTAL CROSS SECTIONAL AREA----= 9.6640

OPERATING LOAD= 197921.97
SEATING LOAD--= 98537.02
OPERATING MODE MOMENT= .29E+06
SEATING MODE MOMENT--= .22E+06

INTEGRAL FLANGE FACTORS

QUANTITY	FIG(APPENDIX11)	VALUE
F	3240-2	.83765
V	3240-3	.24328
FS	3240-6	1.41385
H/HO	-	.44628

MIN.THK.OF HUBBED FLANGE= 1.6376

	STRESSES			
CONDITIONS	HUB	RADIAL	TANGENL	AVERAGE
OPERATING	26250.07	6013.63	6467.02	16358.54
SEATING	20016.89	4585.67	4931.40	12474.15

$$T = \frac{F\,G}{3}\left(\frac{P}{\eta\, s}\right)^{1/2}$$

Assuming p/d_o = 1.25 gives η = 0.497. For gasketed condition, TEMA gives F = 1.25, G = effective gasket diameter = 32.012"†, we have

$$T = 0.143\,(P)^{1/2} \qquad (4.1)$$

Setting M = 0.29 x 10^6 lb-inch, F = 1.25, η = .497, P = 200 psi, A-G = 35.625 - 32.012 = 3.613" in Eq. (2.9) gives the simplified expression for M^* for this example problem.

$$M^* = \frac{3.214 \times 10^6 \left(\frac{T_r}{T}\right)^3 - 13.90 \times 10^6}{32.012 + 4.972\left(\frac{T_r}{T}\right)^3} \qquad (4.2)$$

Similarly, the expression for P_b becomes

$$P_b = \frac{-M^*}{8267.4} \qquad (4.3)$$

We can now perform the calculations in the manner of the steps laid out in Section 2 of this paper.

Step I

Assuming $T = T_r$, we obtain

$$M^* = -\,0.289 \times 10^6$$

which gives

Step II

P_b = 34.95 psi

Step III

Hence P = 200 + 34.95 = 234.95 psi

Therefore, from Eq. (4.1)

$$T = 0.143\,(234.95)^{1/2} = 2.192"$$

† The current edition (sixth) of TEMA, in a slight anamoly with the Code, defines G as the average gasket diameter. We use the Code definition of G in the example problem.

Step IV

Required rim thickness T_r is given by

$$T_r = 1.38\left[\frac{0.29\times10^6 - .289\times10^6 + 40\times3.613(32.012)^2}{(3.613)(17500)}\right]^{1/2}$$

or T_r = 2.119"

$T > T_r$; therefore, both the tubesheet should be made from a 2.192" minimum thickness plate.

Let us assume that the designer wishes to examine whether a 2.5" thick tubesheet plate, and a 2" thick rim will be adequate for this application.

We have

$$\frac{T_r}{T} = \frac{2}{2.5} = 0.8$$

From Eq. (4.2), M^* = -.355 x 10^6 lb-inch which yields, P_b = 42.87 psi. Hence P = 200 + 42.87 = 242.87 psi and

$$T\ (\text{required}) = 0.143\,(242.87)^{1/2} = 2.229"$$

$$T_r(\text{required}) = 1.38\left[\frac{0.29\times10^6 - .355\times10^6 + .148\times10^6}{(3.613)(17500)}\right]^{1/2}$$

$$= 1.581"$$

Therefore, the proposed 2.5" thick interior and 2" thick outer rim are quite adequate.

In this example the tubesheet plate thickness is controlled by the required thickness of the perforated region. However, the rim transmits moment to the interior which increases the "effective pressure" on the perforated interior. In the next example, we will consider the case where the rim moment helps reduce the required thickness of the perforated region.

(iii) Tubesheet for U-tube exchanger: This example illustrates the condition where the edge moment reduces the interior thickness.

Consider a TEMA type AEU unit where the tubesheet is extended as a flange and bolted to a flanged shell. Both shellside and tubeside design conditions are identical as follows:

Design pressure 75 psig, design temperature 300°F, inside diameter 42 inch, shell/channel thickness 0.5 inch. The allowable stress of the tubesheet and flange material is 17500 psi at both ambient and design temperature. Flange details are as shown in Table II.

Assume $\frac{P}{d_0}$ = 1.25 which gives η = 0.497

TABLE II

Item - SHELL/CHANNEL INTERNAL Type

Design Pressure 75. Design Temp. 300. Corrosion Allowance 0.000

	MATERIAL	ALLOW. STRESS(ATM)	ALLOW. STRESS(DES)
FLANGE---------	SA-105	17500.	17500.
BOLT-----------	SA-193-B7	25000.	25000.

GASKET AND BOLTING CALCULATIONS

n= 0.5000	H= 111463.	Am= 13.666
b= 0.2500	Hp= 15374.	Ab= 16.912
y= 10000.0	Wm1= 126837.	W= 422800.
m= 3.0000	Wm2= 341649.	

CONDITION	LOAD	X	LEVER ARM	=	MOMENT
	HD= 103908.	X	hD= 1.5000	=	MD= 155862.
OPERATING	HG= 15374.	X	hG= 1.1250	=	MG= 17296.
	HT= 7555.	X	hT= 1.5000	=	MT= 11332.
			TOTAL MOMENT		MO = 184490.
GASKET SEAT	HG=W= 422800.	X	hG= 1.1250	=	MG = 475650.

SHAPE CONSTANTS

go	0.5000	ho	4.6	h/ho	0.33	t	1.813
g1	0.7500	K	1.137	g1/go	1.50	alpha	1.346
h	1.5000	T	1.86	F	0.87	beta	1.461
mo=	4392.62	Z	7.84	V	0.36	gamma	0.722
mg=	11325.0	Y	15.16	f	1.07	delta	0.112
G	43.5000	U	16.66	e	0.19	lambda	0.834
blt. fact.	1.000000	d	53.0				

STRESSES

			OPERATING		SEATING	
			Allowable	Actual	Allowable	Actual
E	1.0000	SH	26250.	10042.	26250.	25891.
R	1.1250	SR	17500.	2343.	17500.	6041.
		ST	17500.	1915.	17500.	4937.
		SAV	17500.	6193.	17500.	15966.

Shell O.D.--------	43.0000	Gasket Material	S/S SPRL WND
Shell Thickness--	0.5000	Gasket O.D.----	44.0000
Flange Thickness-	1.8125	Gasket I.D.----	43.0000
Flange O.D.------	47.7500	Gasket thickness	0.1750
Flange I.D.------	42.0000		
Flange (B.C.)----	45.7500		
Flange Bolting---(	56)0.750 Dia Bolts		

$M_{so}=M_{co}=0.184\times10^6$ lb.in. $M_{ss}=M_{cs}=0.476\times10^6$ lb.in.

$M = (0.184 - 0.476)10^6 = -0.292 \times 10^6$ lb.in.

For gasketed condition G = 43.5" and F = 1.25".

Then A-G = 4.25" and w = 2.125"

Substituting in Eq. (2.9) gives, upon simplification

$$M^* = \frac{3.557 \times 10^6 \left[\frac{T_r}{T}\right]^3 + 7.586 \times 10^6}{43.5 + 5.858 \left[\frac{T_r}{T}\right]^3}$$

From Eq. (2.8) $P_b = \frac{6.2M^*}{F^2 G^3}$ Hence $P_b = \frac{-M^*}{20744}$

Assuming $T = T_r$ gives $M^* = 0.226 \times 10^6$

then $P_b = -10.88$ psi

Hence P = 75-10.9 = 64.1 psig

Therefore, from Eq. (2.5); $T = \frac{FG}{3} \left(\frac{P}{\eta S}\right)^{1/2}$

$$T = \frac{1.25 \times 43.5}{3} \left(\frac{64.1}{0.497 \times 17500}\right)^{1/2}$$

Hence T = 1.556"

From Eq. (2.10), T_r is calculated to be 1.15".

In the case of this example, the result of the interface moment reduces the term P below the design pressure resulting in a reduced interior tubesheet thickness. The interior thickness governs and the minimum tubesheet thickness, neglecting any additional allowances for pass partition groove and corrosion allowance, is 1.556".

5. CLOSURE

A TEMA-consistent method for computing tubesheet rim thickness is proposed. The procedure is based on the thin plate/ring theory, but uses TEMA's concept of deflection and stress efficiencies. The interaction of the rim thickness and tubesheet interior thickness in unstayed tubesheet is clearly brought out in the design equations. Numerical examples illustrate application of the method. This procedure outlined herein is currently under consideration by the TEMA Technical Committee for possible inclusion in the upcoming 7th edition of the TEMA Standards. Comparison with the stress analysis codes [2] shows reasonable agreement in the practical range of geometric parameters and pressure loadings.

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