Pressure Vessel Newsletter

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INTERGRAPH

Serving the Pressure Vessel Community Since 2007

***From The Editor's Desk*:**

It has been a few months since the last issue of this newsletter was mailed in mid-January. There have been a few changes since then, the biggest one being that I have now relocated to Houston in the US. The minor interruption notwithstanding on account of this move, my goal would be to once again bring out the fresh issues of this newsletter once every month.

While the state of pressure vessel industry in India is still fresh, I would like to make one observation that has been trending for quite some time now. Retaining talent has always been a challenge acrossentire spectrum of Indian pressure vessel industry – end users, engineering companies as well as the manufacturers. Lately, however, the talent has not only been moving away from the companies at a greater frequency but also has been moving away from the country itself. And that talent is moving in droves to the Middle East – Dubai, Abu Dhabi, Kuwait and Saudi Arabia.

The reasons for this move are not entirely clear. Obviously, money is a big part of it but it is nothing new, and by itself, doesn't seem to be the catalyst. Maybe the ease of movement to the Middle East in recent years has been a big factor, and the presence of Indian community in large numbers helps the new arrivals. Whatever the reason, this a zero sum game, and the gains made in the Middle Eastern firms comes at the expense of resources available to the Indian industry.

Pressure vessel industry in India needs to tackle this menace on a war footing if it is to grow at a healthy pace. Barring the larger companies, most of the mid-size and small companies are already handicapped as they neither provide on the job training to the new recruits nor do they encourage their designers and engineers to undergo external training. The industry needs to work on a two-pronged strategy at a very minimum: 1) Offer fair compensation, and b) Make excellent training available.

With these steps, the Indian industry has a good shot at reversing the current trend, and may even benefit in the bargain by having high caliber designers and engineers that can only be helpful in the industry's long-term prospects.

In this issue…

ELLIPSOIAL HEAD UNDER INTERNAL PRESSURE

Ellipsoidal shaped heads are frequently used for the end closures of cylindrical shells for pressure vessels. Figure 1 shows the stresses generated in an ellipsoidal head with semi-major axis *a*, and semi-minor axis *b* due to internal pressure.

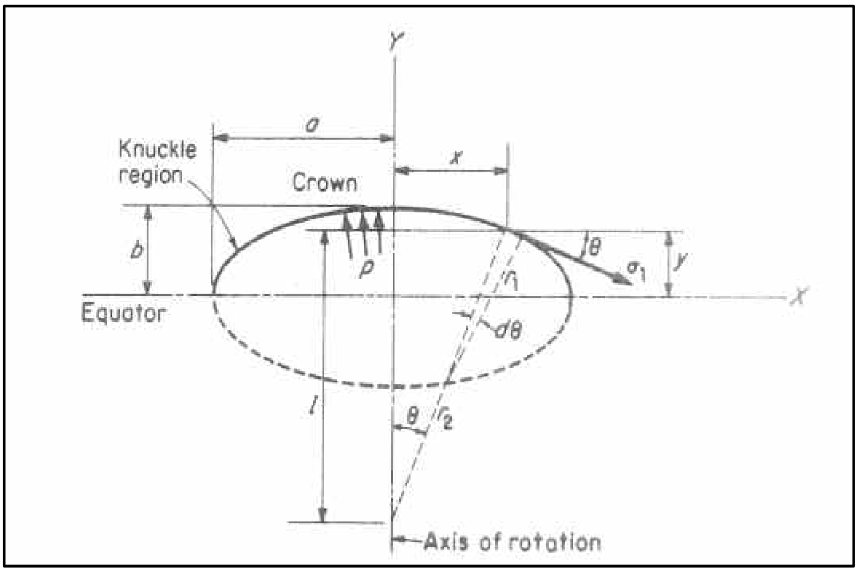

Figure 1: Stress in an Ellipsoid

The following nomenclature is used in this discussion:

σ_1	=	Longitudinal Stress
σ_2	=	Hoop Stress
h	=	Thickness of Head
r_1	=	Longitudinal Radius of Curvature
r_1	=	Radius of Curvature of Element in Hoop Direction

The equation for the longitudinal stress, σ_1, can be written as:

$$\sigma_1 = \frac{pr_2}{2h}$$

And the equation for the hoop stress, σ_2, can be written as:

$$\sigma_2 = \frac{p}{h}\left(r_2 - \frac{r_2^2}{2r_1}\right)$$

At the crown of the head, $r_1 = r_2 = a^2/b$; and $\sigma_1 = \sigma_2 = \frac{pa^2}{2bh}$

At the equator, $r_1 = b^2/a$; and $r_2 = a$.

Therefore, $\sigma_1 = \frac{pa}{2h}$

And $\sigma_2 = \frac{pa}{h}\left(1 - \frac{a^2}{2b^2}\right)$

It can be seen that when a/b exceeds 1.42, hoop stress becomes compressive. The variation in stress throughout an ellipsoid for increasing a/b ratios is shown in the Figure 2. The longitudinal stress remains tensile throughout the ellipsoid for all a/b ratios, being a maximum at the crown and diminishing in value to a minimum at the equator. The hoop stress is also tensile in the crown region but this decreases as the equator is approached where it becomes compressive for a/b ratios greater than 1.42.

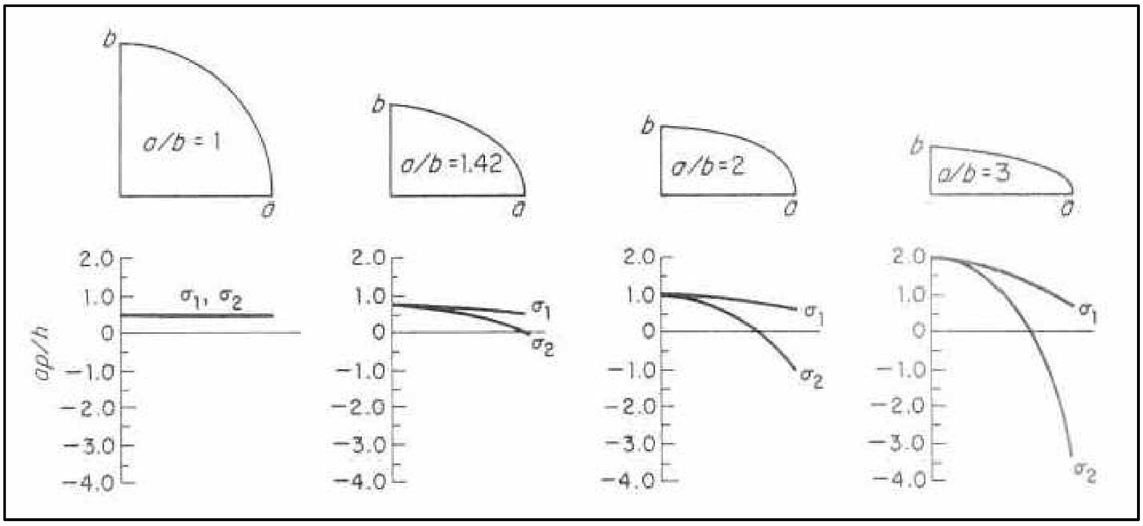

Figure 2: Variation in Stress in an Ellipsoid for Increasing a/b Ratios

Ratio a/b = 1, a sphere, gives the lowest stress. If a/b = 2, a maximum tensile stress of pa/h, which is the same as the hoop stress in a cylinder, occurs at the center of the crown and a hoop compressive stress of equal magnitude occurs at the equator.

Many construction codes and specifications restrict the use of elliptical heads to those with a maximum major-to-minor axis ratio of 2.0. As the ratio a/b is further increased, the greatest stress in the crown is still in tension and lies at the center, but is far exceeded in magnitude by the compressive hoop stress in the knuckle region and the equator. It is this compressive stress that can cause:

1. Local buckling of thin heads due to the high hoop compressive stress
2. Local failure due to high shear stress developed.

Likewise, torispherical or dished heads which simulate ellipsoidal ones by a compound curve composed of a crown radius and a knuckle radius, should have a large knuckle radius in order to minimize the hoop stresses in this region. Many pressure vessel construction codes recognize this fact and specify a minimum permissibleknuckle radius. For instance, ASME Boiler and Pressure Vessel Code specifies the minimum value of knuckle radius as 6% of the crown radius.

Source: Theory and Design of Modern Pressure Vessels – John Harvey

TRAINING ANNOUNCEMENT

PRESSURE VESSEL INSPECTION: API 510

Pressure vessel inspectors must have a broad knowledge base relating to maintenance, inspection, repair, and alteration of pressure vessels. The Pressure Vessel Inspection course benefits employers and industry as a whole by helping to:

- Improve management control of unit operations, repair and maintenance
- Reduce the potential for downtime because of equipment failure
- Provide a continued high level of safety through the use of highly specialized and experienced inspectors.
- Provide knowledgeable and highly skilled inspectors

The course duration is 5 days, and the course content will follow broadly the following topics:

- Service restrictions, joint efficiencies, radiography
- Shell and head calculations
- Maximum allowable working pressure
- Hydrostatic head pressure
- Hydrostatic and pneumatic test, and test gauges
- Postweld heat treatment
- Impact testing
- Corrosion
- ASME Section IX – Welding
- Writing a welding procedure specification
- Review of WPS and PQR's
- ASME Section V – NDE
- API RP 577 – Welding Inspection and Metallurgy
- API 510 – Pressure Test and Welding Requirements
- API RP 572 – Inspection
- API RP 576 – Pressure Relieving Devices
- API RP 571 – Damage Mechanisms

This is not a course for the beginners. At least 3 years of experience in oil, gas and petrochemical industry and involvement in the design, procurement, engineering construction, operation, maintenance and inspection of pressure vessels and related facilities is required. The course is very intense – maximum benefit will be obtained by complete attention during classroom training and completing all the homework assignments in the evening. The second half of the last day will be reserved for a mock examination that will closely resemble the actual API Certification Examination.

The instructor, Ramesh Tiwari, is internationally recognized specialist in the area of pressure vessels, heat exchangers, materials, and codes and standards. He holds Bachelor's and Master's degrees in mechanical engineering from universities in India and United States. He is also a registered Professional Engineer in the State of Maryland in the United States. Ramesh has over 24 years of design engineering experience on a variety of projects spanning industries such as oil & gas, power, nuclear, chemical, petrochemical, pharmaceutical, food etc. He has also developed and conducted various pressure vessel related training programs for peers as well as clients in US, Canada and India.

The training will be held at**Houston**in the **second week of August** (10thto 14th). If interested, please email at ramesh.tiwari@codesignengg.com or call (713) 562-0368.

APPLICATION OF UHX RULES TO U-TUBE TUBESHEET GASKETED WITH SHELL AND CHANNEL

UHX rules apply to three typical types of tubular heat exchangers (See Figure 1):

1. U-tube heat exchangers
2. Fixed tubesheet heat exchangers
3. Floating tubesheet heat exchangers

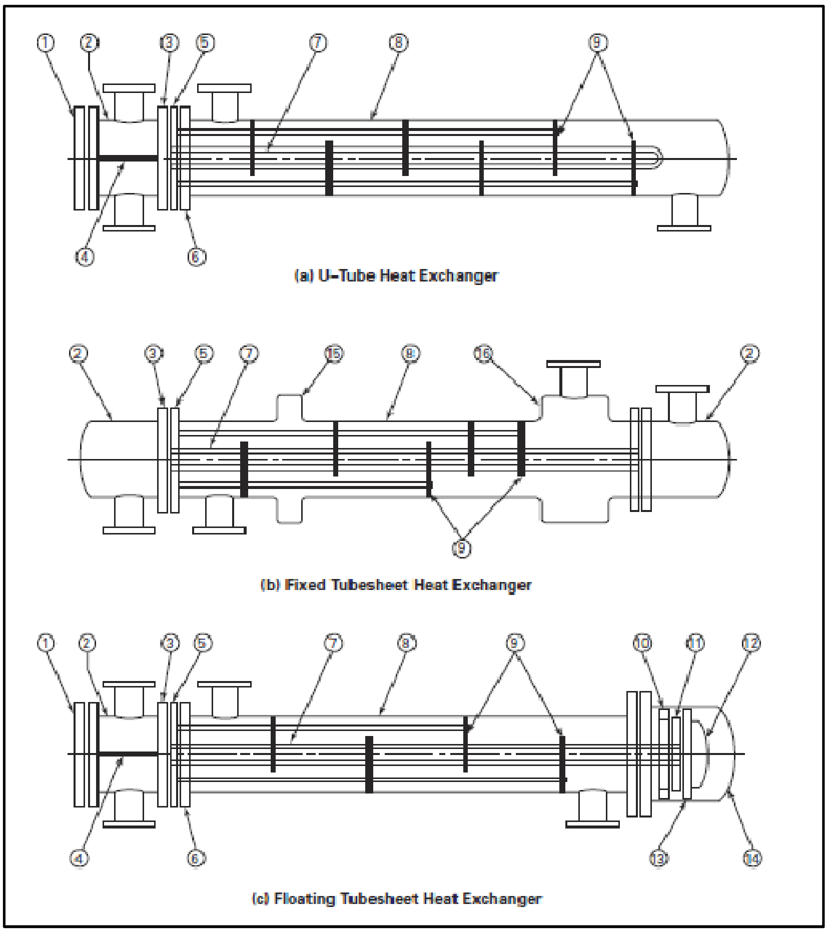

Figure 1: Types of Tubular Heat Exchanger

The tubesheet is attached to the shell and channel either by welding (integral tubesheet) or by bolting (gasketed tubesheet) according to six configurations in the industry (See Figure 2):

- Configuration a: Tubesheet integral with shell and channel
- Configuration b: Tubesheet integral with shell and gasketed with channel, extended as flange
- Configuration c: Tubesheet integral with shell and gasketed with channel, not extended as flange
- Configuration d: Tubesheet gasketed with shell and channel
- Configuration e: Tubesheet gasketed with shell and integral with channel, extended as flange
- Configuration f: Tubesheet gasketed with shell and integral with channel, not extended as flange.

Structure of Part UHX

UHX-1 to UHX-8:	General consideration common to the three types of heat exchangers. Includes scope, materials, fabrication, terminology etc.
UHX-9:	Design rules for tubesheet flange extension
UHX-10:	“Conditions of Applicability: specifies under which conditions the rules are applicable

UHX-11: "Tubsesheet Characteristics" is common to the three types of heat exchangers. It provides design formulas for the ligament efficiency and the effective elastic constants.

UHX-12: Design rules for U-tube heat exchangers

UHX-13: Design rules for fixed tubesheet heat exchangers

UHX-14: Design rules for floating tubesheet heat exchangers

UHX-15 to UHX-19: These paragraphs provide considerations on tube-to-tubesheet welds, expansion bellows, pressure tests and Markings

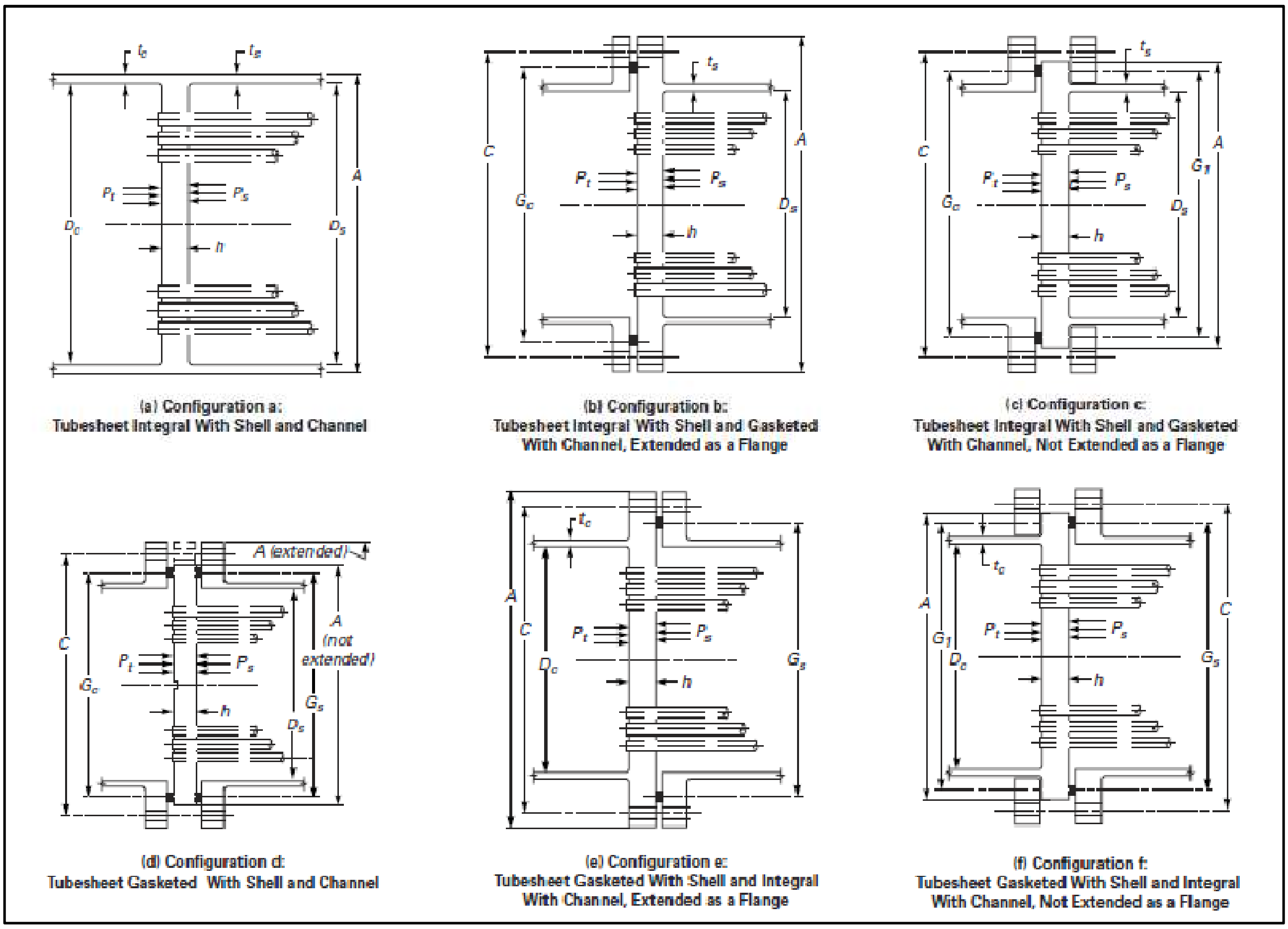

Figure 2: Tubesheet Configurations

ASME PTB-4 provides design examples for each type of heat exchangers. We have reproduced below the example for a U-tube tubesheet gasketed with shell and channel (Configuration d in Figure 2).

<u>Design Conditions</u>:

Shell Side: 10 psi/ FV @300^{o}F

Tube Side: 135 psi @300^{o}F

Materials: Tube – SB111 C44300 (Admiralty), 5/8" OD, 0.065" thick (16 gage), expanded for the full thickness of the tubesheet.

Tubesheet: SA285 Grade C, 1/8" corrosion allowance on tube side. No pass partition grooves. 20"OD, 386 tube holes, ¾" equilateral triangular pattern with one centerline pass lane. The largest center-to-center distance between adjacent tube rows is 1.75", and the radius to the outermost tubehole center is 8.094 in.

Diameter of Shell Gasket Load Reaction: 19"

Shell Flange Design Bolt Load: 147,000 lb

Diameter of Channel Gasket Load Reaction: 19”

Channel Flange Design Bolt Load: 162,000 lb

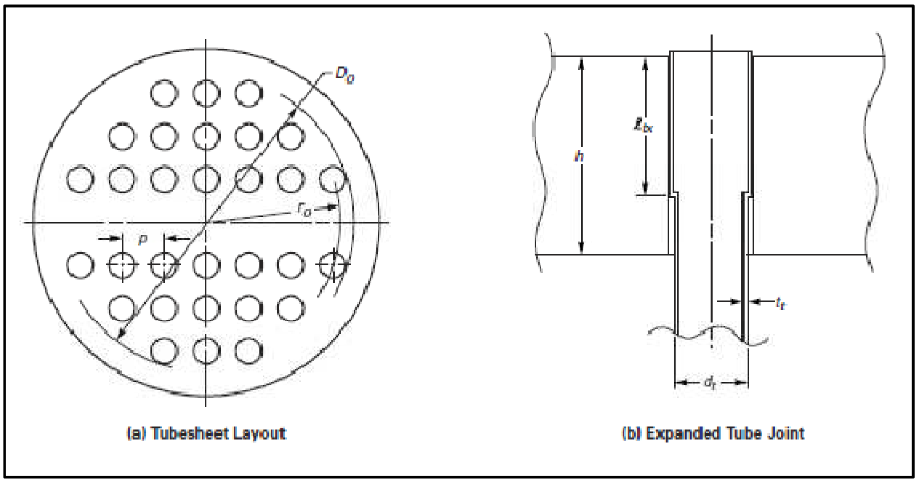

Figure 3: Tubesheet Geometry

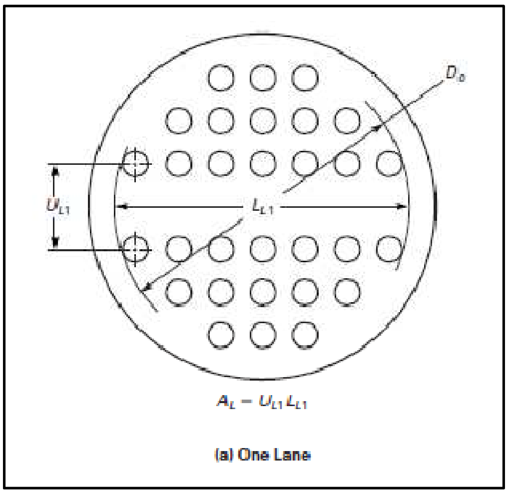

Figure 4: Untubed Lane Configuration

Data Summary:

The data for paragraph UHX-11.3 is as follows:

c_t = tubesheet corrosion allowance on tube side (0.125 in)

d_t = nominal outside diameter of tubes (0.625 in)

E = Modulus of elasticity of tubesheet material (28.3E10^6 psi from Table TM-1 of Section II, Part D at 300^{o}F)

E_t = Modulus of elasticity of tube material (15.4E10^6 psi from Table TM-3 of Section II, Part D at 300^{o}F)

h_g = Tubeside pass partition groove depth (0 in)

p = Tube pitch (0.75 in)

r_o = Radius to the outermost tubehole center (8.094 in)

S = Allowable stress for tubesheet material at tubesheet design temperature (15,700 psi from Table 1A of Section II, Part D at 300^{o}F)

S_t = Allowable stress for tube material at tube design temperature (10,000 psi from Table 1B of Section II, Part D at 300°F)

t_t = Nominal tube wall thickness (0.065 in)

U_{L1} = Center-to-center distance between adjacent tube rows of untubed lane (1.75 in)

ρ = Tube expansion depth ratio (1.0 for a full length tube expansion)

Table TM-1
Moduli of Elasticity *E* of Ferrous Materials for Given Temperatures

	Modulus of Elasticity E = Value Given × 10^6 psi, for Temperature, °F, of																	
Materials	−325	−200	−100	70	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500
Carbon steels with C ≤ 0.30%	31.4	30.8	30.3	29.4	28.8	28.3	27.9	27.3	26.5	25.5	24.2	22.5	20.4	18.0	...	...	...	...
Carbon steels with C > 0.30%	31.2	30.6	30.1	29.2	28.6	28.1	27.7	27.1	26.4	25.3	24.0	22.3	20.2	17.9	15.4	...	...	...
Ductile cast iron	...	...	24.5	23.4	22.5	21.9	21.2	20.5	19.9	19.2	...	...	...	...	...	...	...	...
Material Group A [Note (1)]	31.1	30.5	30.0	29.0	28.5	28.0	27.6	27.0	26.3	25.3	23.9	22.2	20.1	17.8	15.3	...	...	...
Material Group B [Note (2)]	29.6	29.0	28.6	27.8	27.1	26.7	26.2	25.7	25.1	24.6	23.9	23.2	22.4	21.5	20.4	19.2	17.7	...
Material Group C [Note (3)]	31.6	30.9	30.5	29.6	29.0	28.5	28.0	27.4	26.9	26.2	25.6	24.8	23.9	23.0	21.8	20.5	18.9	...
Material Group D [Note (4)]	32.6	31.9	31.4	30.6	29.9	29.4	28.8	28.3	27.7	27.0	26.3	25.6	24.7	23.7	22.5	21.1	19.4	...
Material Group E [Note (5)]	33.0	32.4	31.9	31.0	30.3	29.7	29.2	28.6	28.1	27.5	26.9	26.2	25.4	24.4	23.3	22.0	20.5	...
Material Group F [Note (6)]	31.2	30.7	30.2	29.2	28.4	27.9	27.3	26.8	26.2	25.5	24.5	23.2	21.5	19.2	16.5	...	...	...
Material Group G [Note (7)]	30.3	29.7	29.2	28.3	27.5	27.0	26.4	25.9	25.3	24.8	24.1	23.5	22.8	22.0	21.2	20.3	19.2	18.1
Material Group H [Note (8)]	...	...	30.2	29.0	28.2	27.5	27.0	26.4	26.0	25.5	25.1	...	...	...	...	...	...	...
Material Group I [Note (9)]	27.8	27.1	26.6	25.8	25.1	24.6	24.1	23.6	23.1	22.6	22.1	21.6	21.1	20.6	20.1	19.6	19.1	18.6
Material Group J [Note (10)]	31.1	30.3	29.7	28.6	27.8	27.2	26.6	26.0	25.4	24.7	24.1	23.5	22.9	...	...	...	...	...
S13800 [Note (11)]	31.5	30.9	30.3	29.4	28.7	28.1	27.5	26.9	26.3	25.7	25.0	24.4	...	...	...	...	...	...
S15500 [Note (12)]	30.5	29.9	29.4	28.5	27.8	27.2	26.7	26.1	25.5	24.9	24.3	23.7	...	...	...	...	...	...
S45000 [Note (13)]	31.6	31.0	30.4	29.5	28.8	28.2	27.6	27.0	26.4	25.8	25.1	24.5	...	...	...	...	...	...
S17400 [Note (14)]	30.5	29.9	29.4	28.5	27.8	27.2	26.7	26.1	25.5	24.9	24.3	23.7	...	...	...	...	...	...
S17700 [Note (15)]	31.6	31.0	30.4	29.5	28.8	28.2	27.6	27.0	26.4	25.8	25.1	24.5	...	...	...	...	...	...
S66286 [Note (16)]	31.0	30.6	30.2	29.2	28.5	27.9	27.3	26.7	26.1	25.5	24.9	24.2	...	...	...	...	...	...

Table TM-3
Moduli of Elasticity *E* of Copper and Copper Alloys for Given Temperatures

	Modulus of Elasticity E = Value Given × 10^6 psi, for Temperature, °F, of									
Material	−325	−200	−100	70	200	300	400	500	600	700
C93700	11.6	11.4	11.3	11.0	10.7	10.5	10.3	10.1	9.8	9.4
C83600	14.8	14.6	14.4	14.0	13.7	13.4	13.2	12.9	12.5	12.0
C92200	14.8	14.6	14.4	14.0	13.7	13.4	13.2	12.9	12.5	12.0
C28000	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C36500	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C46400	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C65500	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C66100	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C95200	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C95400	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8
C44300	16.9	16.7	16.4	16.0	15.6	15.3	15.0	14.7	14.2	13.7
C44400	16.9	16.7	16.4	16.0	15.6	15.3	15.0	14.7	14.2	13.7
C44500	16.9	16.7	16.4	16.0	15.6	15.3	15.0	14.7	14.2	13.7
C64200	16.9	16.7	16.4	16.0	15.6	15.3	15.0	14.7	14.2	13.7
C68700	16.9	16.7	16.4	16.0	15.6	15.3	15.0	14.7	14.2	13.7
C10200	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C10400	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C10500	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C10700	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C11000	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C12000	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C12200	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C12300	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C12500	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C14200	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C23000	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C61000	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C61400	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C65100	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C70400	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5
C19400	18.5	18.2	18.0	17.5	17.1	16.8	16.5	16.1	15.6	15.0
C60800	18.5	18.2	18.0	17.5	17.1	16.8	16.5	16.1	15.6	15.0
C63000	18.5	18.2	18.0	17.5	17.1	16.8	16.5	16.1	15.6	15.0
C70600	19.0	18.7	18.5	18.0	17.6	17.3	16.9	16.5	16.0	15.4
C97600	20.1	19.8	19.6	19.0	18.5	18.2	17.9	17.5	16.9	16.2
C71000	21.2	20.8	20.6	20.0	19.5	19.2	18.8	18.4	17.8	17.1
C71500	23.3	22.9	22.6	22.0	21.5	21.1	20.7	20.2	19.6	18.8

Paragraph UHX 12 provides the rules for the design of U-tube tubesheets. The data for the paragraph UHX-12.3 are as follows:

A = Outside diameter of the tubesheet (20 in)

E = Modulus of elasticity for tubesheet material at design temperature ($28.3E10^6$ psi from Table TM-1 of Section II, Part D at 300^oF)

G_c = Diameter of channel gasket load reaction (19.0 in)

G_s = Diameter of shell gasket load reaction (19.0 in)

P_s = Shell side design pressure (10 psi/ FV)

P_t = Tube side design pressure (135 psi)

S = Allowable stress for tubesheet material at tubesheet design temperature (15,700 psi from Table 1A of Section II, Part D at 300^oF)

W^* = Tubesheet effective bolt load determined in accordance with UHX-8 (162,000 lb)

Table UHX-8.1
Tubesheet Effective Bolt Load, W^*

Configuration	Loading Case 1	2	3	4–7
a	0	0	0	0
b	W_{m1c}	0	W_{m1c}	W_c
c	W_{m1c}	0	W_{m1c}	W_c
d	W_{m1c}	W_{m1s}	W_{m1max}	W_{max}
e	0	W_{m1s}	W_{m1s}	W_s
f	0	W_{m1s}	W_{m1s}	W_s
A	0	0	0	0
B	W_{m1c}	0	W_{m1c}	W_c
C	W_{m1c}	0	W_{m1c}	W_c
D	0	0	0	0

In the above table, the configuration is d, and the loading case is 3 for the tube side pressure and the shell side pressure acting simultaneously. W_{m1max} refers to the greater of channel flange design bolt load and shell flange design bolt load for the operating condition.

Calculation Procedure:

The calculation procedure for a U-tube heat exchanger tubesheet is given in paragraph UHX-12.5. The calculation results are shown for loading case 3 where P_s = -15 psi and P_t = 135 psi since this case yields the greatest value of stresses.

Step 1: Determination of effective dimensions and ligament efficiencies

D_o = Equivalent diameter of outer tube limit circle (See Figure 4)

= (2 x 8.094) + 0.625

= 16.813 in

$$\mu = \frac{p - d_t}{p}$$

$$= \frac{0.75-0.625}{0.75}$$

= 0.167

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

$=\quad \mathrm{MAX}\left\{\left[0.625-2*0.0625*\left(\frac{15{,}400{,}000}{28{,}300{,}000}\right)\left(\frac{10{,}000}{15{,}700}\right)*1.0\right],[0.625-2*0.065]\right\}$

= 0.580 in

A_L = Total area of untubed lanes

= $U_{L1} * D_o$

= 1.75 * 16.813

= 29.423 in^2

$$p^* = \frac{p}{\left(1-\frac{4\,\mathrm{MIN}[(A_L),(4D_o p)]}{\pi D_o^2}\right)^{\frac{1}{2}}}$$

$$= \frac{0.75}{\left(1-\frac{4\,\mathrm{MIN}[(29.423),(4*16.813*0.75)]}{\pi*16.813*16.813}\right)^{\frac{1}{2}}}$$

= 0.805 in

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

$$= \frac{0.805-0.580}{0.805}$$

= 0.280

h'_g = MAX[(h_g-c_t),0]

= 0 in

Step 2: Calculate diameter ratios ρ_s, ρ_c and moment M_{TS} due to pressures P_s and P_t acting on the unperforated tubesheet rim.

$$\rho_s = \frac{G_s}{D_o}$$

$$= \frac{19.0}{16.813}$$

= 1.130

$$\rho_c = \frac{G_c}{D_o}$$

$$= \frac{19.0}{16.813}$$

= 1.130

$$M_{TS} = \frac{D_o^2}{16}\left[(\rho_s-1)(\rho_s^2+1)P_s-(\rho_c-1)(\rho_c^2+1)P_t\right]$$

$$= \frac{16.813*16.813}{16}\left[(1.13-1)(1.13*1.13+1)(-15)-(1.13-1)(1.13*1.13+1)135\right]$$

= -785 in-lb/in

Step 3: Assume a value for tubesheet thickness, h, and calculate h/p, and then determine E^*/E and ϑ^* relative to h/p

h = 1.28 in

h/p = 1.28/ 0.75

= 1.71

Refer to the Figure 5 below to determine E^*/E and ϑ^*.

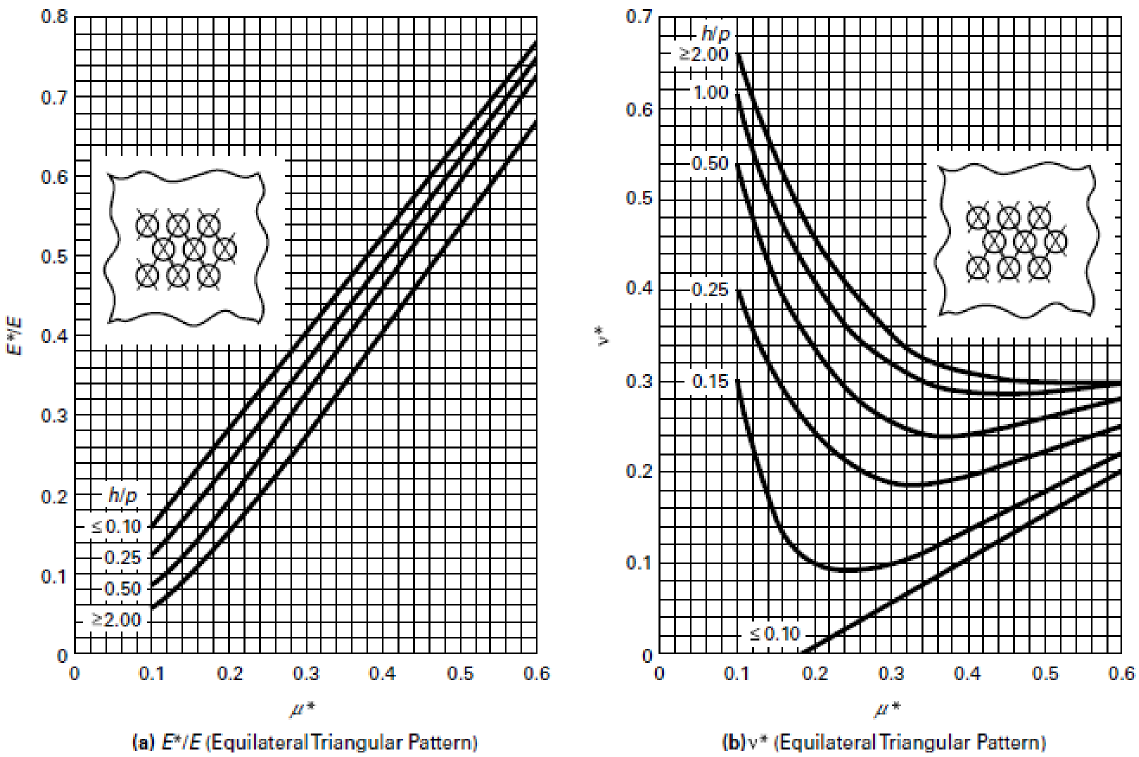

Figure 5: Curves for Determination of E^*/E and ϑ^* (Equilateral Triangular Pattern)

E^*/E = 0.265

ϑ^* = 0.358

E^* = 0.265 x 28,300,000

= $7.5E10^6$ psi

Step 4: For configuration d, skip STEP 4 and proceed to STEP 5.

Step 5: Calculate K and F for configuration d.

$$K = \frac{A}{D_o}$$

$$= \frac{20}{16.813}$$

$$= 1.19$$

$$F = \frac{1-\vartheta^*}{E^*}(E \ln K)$$

$$= \frac{1-0.358}{7{,}500{,}000}(28{,}300{,}000 \ln 1.19)$$

$$= 0.420$$

Step 6: Calculate moment M^* acting on the unperforated tubesheet rim

$$M^* = M_{TS} + \frac{(G_c - G_s)}{2\pi D_o} W^*$$

= -785 in-lb/in

Step 7: Calculate M_p, M_o and M.

$$M_p = \frac{M^* - \frac{D_o^2}{32} F (P_s - P_t)}{1+F}$$

$$= \quad \frac{-785 - \frac{16.813 \times 16.813}{32} \times 0.420\,(-15-135)}{1+0.420}$$

= -160 in-lb/ in

$$M_o \quad = \quad M_p + \frac{D_o^2}{64}(3+\vartheta^*)(P_s - P_t)$$

$$= \quad -160 + \frac{16.813 \times 16.813}{64}(3+0.358)(-15-135)$$

= -2,380 in-lb/ in

M = MAX[$|M_p|$, $|M_o|$]

= 2,380 in-lb/ in

Step 8: Calculate the tubesheet bending stress σ and check the acceptance criterion.

$$\sigma \quad = \quad \frac{6M}{\mu^*(h-h_g')^2}$$

$$= \quad \frac{6 \times 2{,}380}{0.280\,(1.28-0)^2}$$

= 31,200 psi ≤ 2 x 15,700 psi = 31,400 psi

Step 9: Calculate the average shear stress, τ and check the acceptance criterion.

$$\tau \quad = \quad \frac{1}{4\mu}\left(\frac{1}{h}\left\{\frac{4A_p}{C_p}\right\}\right)|P_s - P_t|$$

= 2,960 psi ≤ 0.8S = 12,600 psi

C_p in the equation for perimeter of the tube layout measured stepwise in increments of one tube pitch from the center-to-center of the outermost tubes (See Figure 6). A_p is the total area enclosed by C_p.

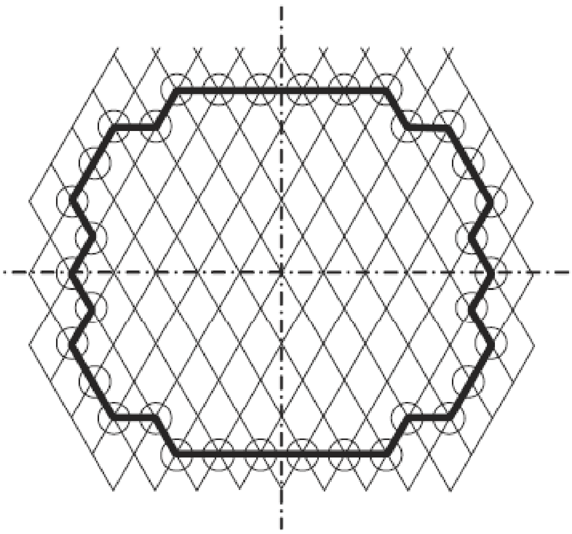

Figure 6: Tube Layout Perimeter

Source: ASME PTB-4-2012 Section VIII, Division 1 Example Problem Manual

ASME Section VIII, Division 1, Subsection C, Part UHX

THERMAL EXPANSION IN A FLOATING HEAD SHELL-AND-TUBE HEAT EXCHANGER

Shell-and-tube heat exchangers comprise nearly 99% of all heat exchangers used in chemical plants and petroleum refineries. Figure 1 shows one such heat exchanger with the floating head configuration.

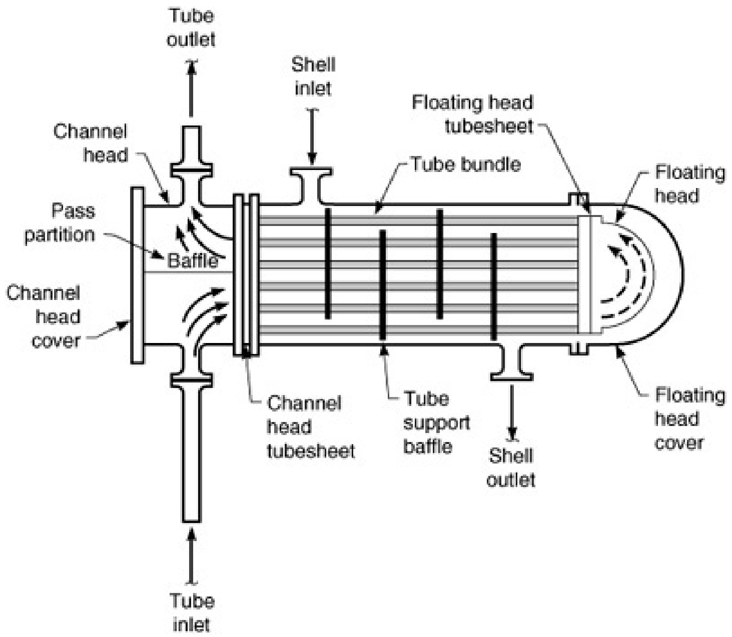

Figure 1: Floating Head Shell and Tube Heat Exchanger

Tube-side Flow

Referring to the figure, we can see how a floating head heat exchanger works. The tube side flow enters the bottom of the channel head. Inside the heat exchanger's head, we have the pass partition baffle which divides the head into two equal portions. This baffle forces the total flow only through the bottom half of the tubes. The tubes are mostly ¾" or 1" outside diameter (OD); the front end of each tube is slipped into a slightly larger hole drilled into the head tubesheet. This tubesheet is a disk about 2" thick and slightly larger than the inner diameter (ID) of the shell.

The tubes are firmly attached to the tubesheet by "rolling". After a tube is pushed into the tubesheet, a tapered tool is inserted into the open end of the tube, and forcefully rotated. The tube's diameter is thus slightly expanded. While rolling is quite effective in sealing the tube inside the tubesheet, the rolls have been known to leak.

The tube side flow now flows into the floating head which acts as a return header for the tubes. The tubeside flow now makes 180^o turn and flows back through the top half of the floating-head tubesheet. The floating head is firmly attached to the floating-head tubesheet. It is this floating head that allows the differential rate of thermal expansion between the tubes and the shell.

Not all shell-and-tube heat exchangers designed for differential rate of thermal expansion between the tubes and the shell have floating heads. Manu heat exchangers have individual U-bends for each tube. Then, each of the U-bends functions like a mini-floating head for each tube.

Similar to the U-tube heat exchangers, the floating head heat exchangers create a rather unpleasant process problem. The most efficient way to transfer heat between two fluids is to have a true countercurrent flow. This means that the shell side fluid and the tube-side fluid must flow through the heat exchanger in opposite directions. However, in the floating head, the tube side fluid reverses direction and flows back to the top half of the channel head in a concurrent flow with the shell-side fluid. A typical heat exchanger may lose 5-30% of its capacity because of the non-true countercurrent flow.

Shell-side Flow

The shell-side flow enters the heat exchanger through the top inlet nozzle. Not shown on this is an impingement plate which is simply a square piece of metal somewhat larger than the inlet nozzle. Its function is to protect the tubes from the erosive velocity of the shell-side feed. The plate lies across the upper row of tubes.

The four tube support baffles shown in this heat exchanger serve a dual function:

- They serve to support the tubes
- More importantly, they promote high cross-flow velocity

The concept of cross-flow velocity is very important in understanding how shell-and-tube heat exchangers work. This concept is related to a flow phenomenon called *vortex shedding*.

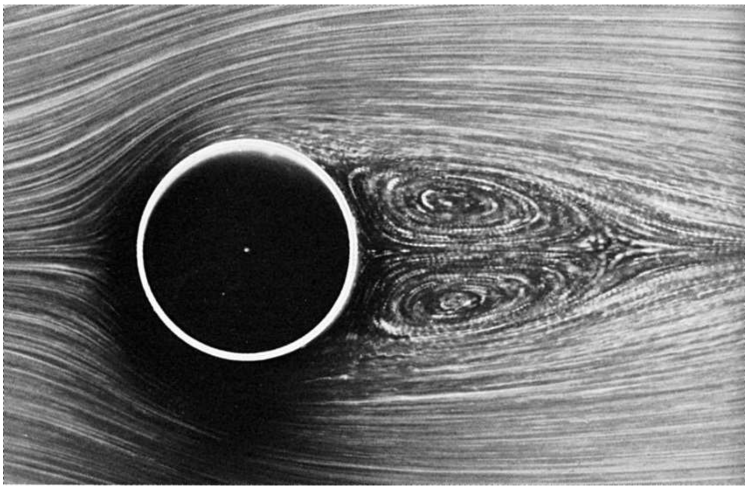

Figure 2: Vortices and Turbulence behind a Tube

When a fluid such as water flows perpendicularly across a tube, vortices are created. The resulting turbulence that forms behind the tube promotes good heat transfer. Therefore, we want to encourage vortex shedding and turbulence on the shell side; this can best be done by increasing the cross flow velocity. A good cross-flow velocity for water is 3 to 5 ft/s. For fluids other than water, a reasonable cross flow velocity, in ft/s, is $\frac{30}{(Density)^{1/2}}$.

The baffle cut (shown in Figure 3) is usually about 20 to 30 percent of the diameter of the baffle. The smaller the baffle cut, the more perpendicular the flow across the tubes. Perpendicular flow encourages desirable cross flow velocity and vortex shedding. But a smaller baffle cut will also increase the pressure drop on the shell side.

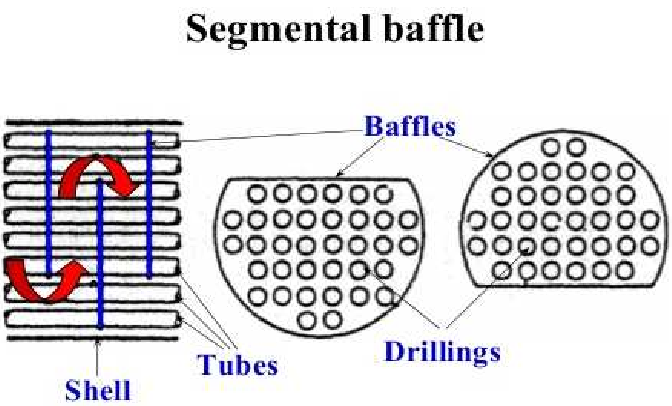

Figure 3: Tube Support Baffle

Effect of Shell-side Pressure Drop

Reducing the baffle spacing increases cross flow velocity and improves heat transfer. But similar to the effect of reducing the baffle cut, it also increases the shell side pressure drop. Too much shell-side pressure drop can create a problem which is the flow through the bypass area. The bypass area is caused by two factors:

1. The tube support baffles must have a diameter somewhat smaller than the ID of the shell.
2. The holes drilled in the baffles for the tubes cannot be drilled too close to the edge of the baffles.

The gap thus created between the shell ID and the outer row of tubes will permit the shell-side fluid to bypass around the tubes. This is obviously bad for heat transfer. And as the shell-side ΔP increases, the percent of fluid that is squeezed through the bypass area increases (because there is less resistance to the flow). If the baffle spacing gets too small, the shell-side heat transfer rate will actually worsen. This happens even though the crossflow velocity increases.

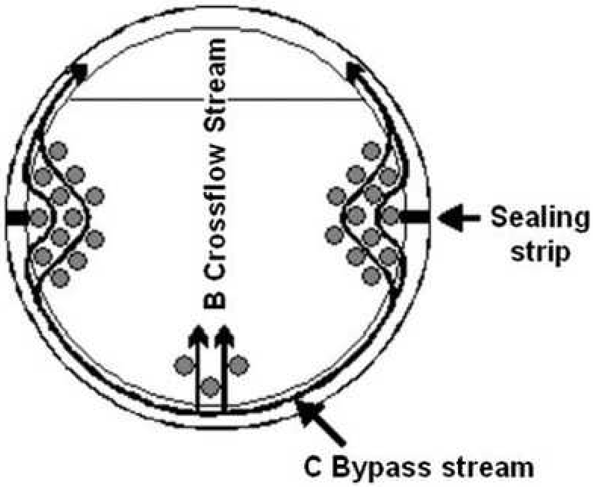

Figure 4: Seal Strips to Reduce Bypassing around Tube Bundles

In order to reduce this bypass, many tube bundles have pairs of metal strips set around the edge of the tube bundle (See Figure 4). These metal strips are typically ¼" thick and 4" wide and extend down the length of the tubes. They are inserted in grooves cut in the tube support baffles. Their function is to interfere with, and reduce, the fluid flow through the bypass area. These seal strips often increase the heat transfer efficiency by 5 to 10 percent.

Source: A Working Guide to Process Equipment *by* Norman Lieberman and Elizabeth Lieberman

NEWS AND EVENTS

2015 ASME Power & Energy

June 18 – July 2, 2015 | San Diego, California

In 2015, four of ASME's major conferences (ASME Power Conference, ASME Energy Sustainability Conference, ASME Fuel Cell Conference, and ASME Nuclear Forum) come together to create an event of major impact for the Power and Energy sectors.

POWER-GEN Natural Gas

August 18-20, 2015 | Columbus, Ohio

Annual conference and exhibition targeting gas-fired generation related to the development of natural gas reserves in the Marcellus and Utica shales of the Appalachian Basin.

GASPRO AMERICAS and LNG 360^{o} AMERICAS

September 9-11, 2015 | Houston, Texas

GasPro 2015 will focus on gas supply, procurement, purchasing, transportation, trading, distribution, operations, safety, the environment, regulatory affairs, technology development, business analysis, and more. All segments of the gas processing industry will be discussed: upstream, midstream and downstream.

2015 API Tanks, Valves and Piping Conference and Expo

October 12-15, 2015 | Las Vegas, Nevada

This event will give attendees an opportunity to learn about new and existing industry codes and standards and to hear about emerging trends from industry experts. The conference offers over 65 sessions in three conference tracks, addressing the needs of individuals involved in productions systems, pipelines, terminals, refining and chemical manufacturing, and storage facilities.

Gastech Conference and Exhibition

October 27-30, 2015 | Singapore

The Gastech conference program has been designed to be at the core of the global LNG and gas business and to fully mirror the industry's needs. In total, 13 business critical session themes will shape this year's program.

The Abu Dhabi International Petroleum Exhibition & Conference

November 9-12, 2015 | Abu Dhabi

The Abu Dhabi International Petroleum Exhibition and Conference (ADIPEC) is the world's new meeting point for Oil & Gas professionals. Held in Abu Dhabi, a natural crossroads between east and west, the city is fast becoming one of the world's most influential energy hubs of the 21st century.

2015 International Mechanical Engineering Congress & Exposition

November 13-19, 2015 | Houston, Texas

ASME's International Mechanical Engineering Congress and Exposition (IMECE) is the largest interdisciplinary mechanical engineering conference in the world. IMECE plays a significant role in stimulating innovation from basic discovery to translational application. Among the 4,000 attendees from 75+ countries are mechanical engineers in advanced manufacturing, aerospace, advanced energy, fluids engineering, heat transfer, design engineering, materials and energy recovery, applied mechanics, power, rail transportation, nanotechnology, bioengineering, internal combustion engines, environmental engineering, and more.

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