SHELL & TUBE
HEAT EXCHANGER DESIGN

Introduction

❑ Shell & tube heat exchangers are the most versatile type of heat exchangers.

 ▪ They are used in process industries, in conventional and nuclear power stations, steam generators, etc

 ▪ They are used in many alternative energy applications including ocean, thermal and geothermal.

❑ Shell & tube heat exchangers provide relatively large ratios of heat transfer area to volume.

❑ They can be easily cleaned.

Shell & Tube Heat Exchangers

❑ Shell & tube type heat exchangers are built of tubes (round or rectangular in general) mounted in shells (cylindrical, rectangular or arbitrary shape).

❑ Many variations of this basic type is available.

  ▪ The differences lie mainly in the detailed features of construction and provisions for differential thermal expansion between the tubes and the shell.
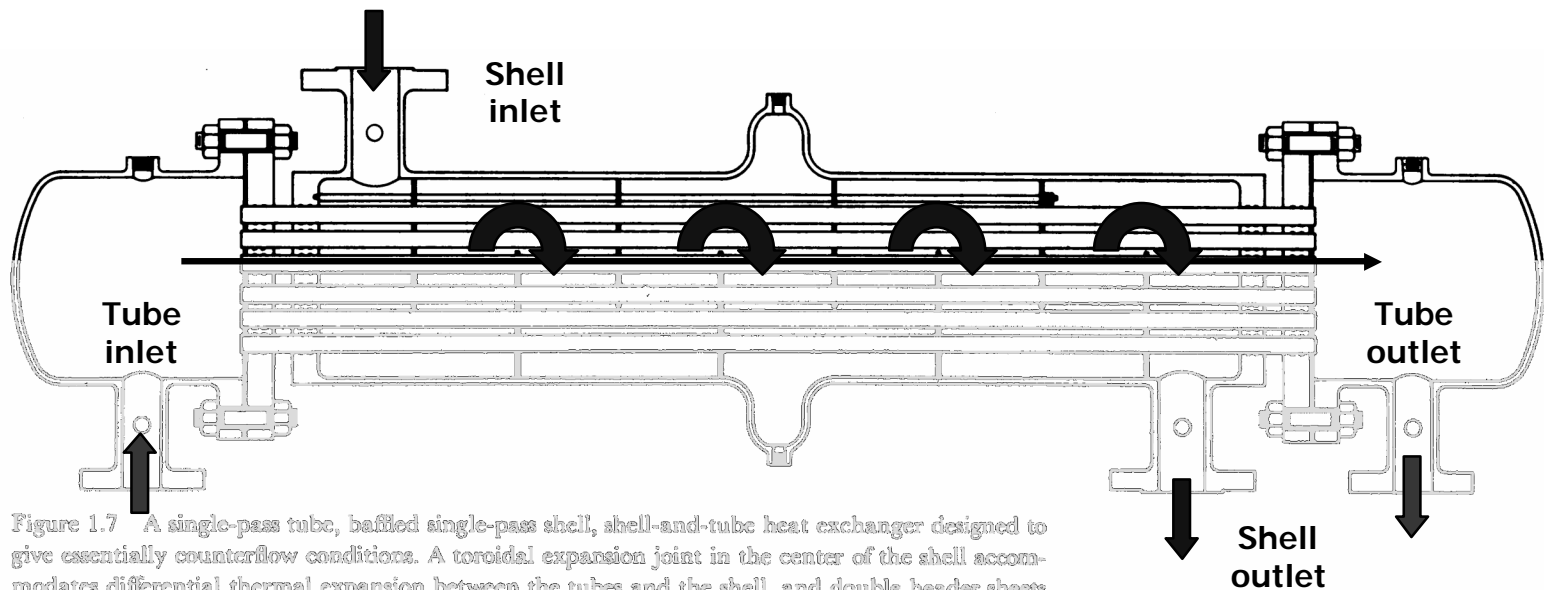

Figure 1.7   A single-pass tube, baffled single-pass shell, shell-and-tube heat exchanger designed to give essentially counterflow conditions. A toroidal expansion joint in the center of the shell accommodates differential thermal expansion between the tubes and the shell, and double header sheets assure that no fluid can leak from one circuit to the other. (Courtesy The Patterson-Kelley Co.)
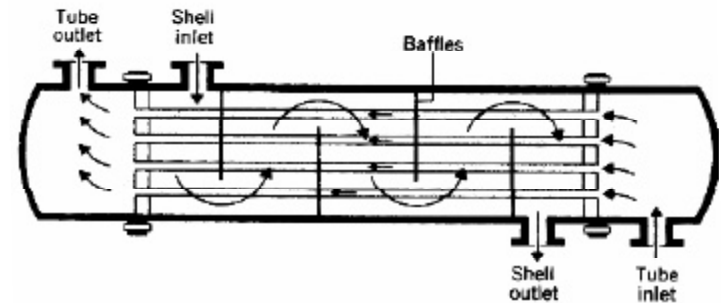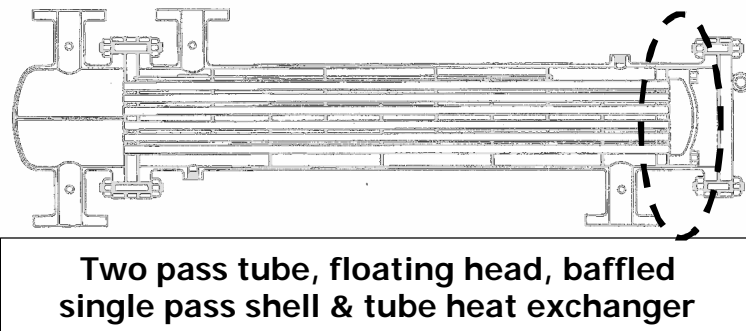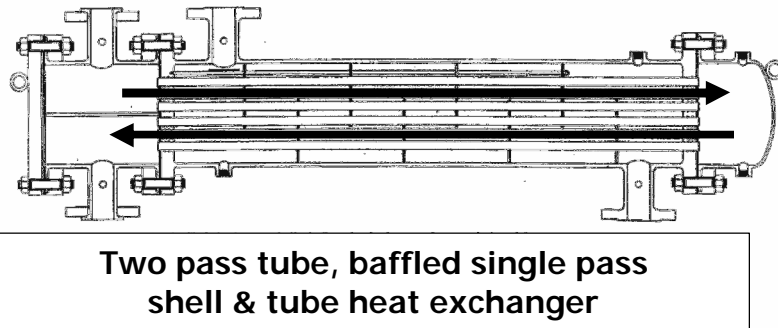
Shell & Tube Heat Exchangers

U-Tube, baffled, single pass
shell & tube heat exchanger

Figure 1.8
Patterson-

Two pass tube, baffled single pass
shell & tube heat exchanger

Two pass tube, floating head, baffled
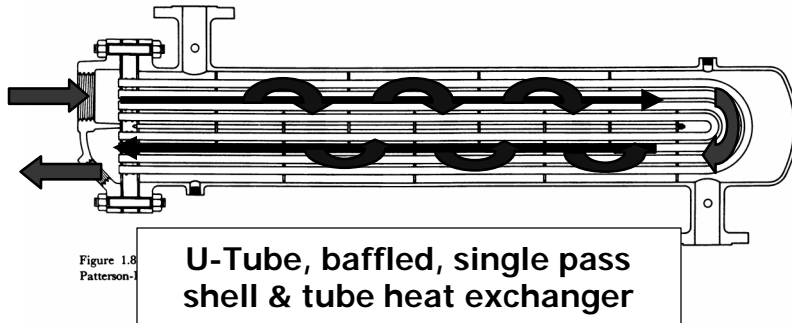single pass shell & tube heat exchanger

Figure 11.3 Shell-and-tube heat exchanger with one shell pass and one tube pass (cross-counterflow mode of operation).
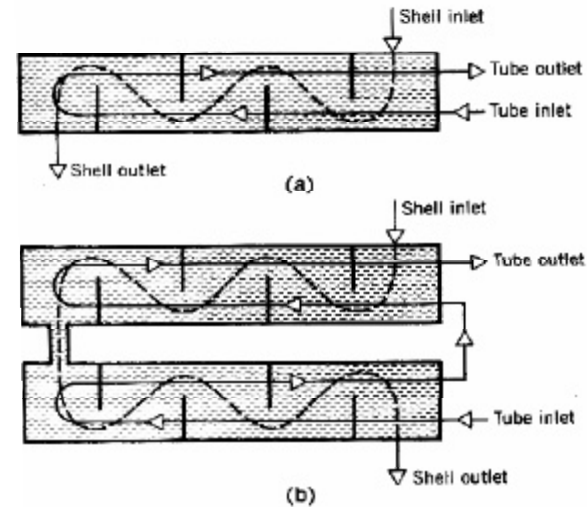
Figure 11.4 Shell-and-tube heat exchangers. (a) One shell pass and two tube passes. (b) Two shell passes and four tube passes.

Shell Types

❑ TEMA (the Tubular Exchangers Manufacturers Association) publishes standards defining how shell and tube exchangers should be built. They define a naming system that is commonly used.

❑ Shells are also typically purchased in standard sizes to control costs. Inside the shell, baffles (dividers) are installed to direct the flow around the tubes, increase velocity, and promote cross flow. They also help support the tubes. The *baffle cut* is the ratio of the baffle window height to the shell diameter. Typically, baffle cut is about 20 percent. It effects both heat transfer and pressure drop. Designers also need to specify the *baffle spacing*; the maximum spacing depends on how much support the tubes need.
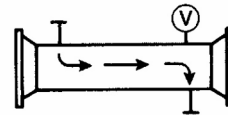
E: One-Pass Shell
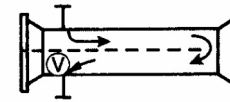
F: Two-Pass Shell with Longitude Baffle
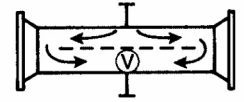
G: Split Flow
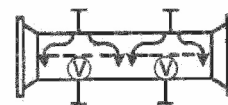
H: Double Split Flow
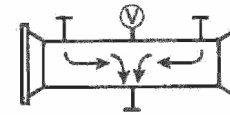
J: Crossflow (Conbined Flow for Condenser)
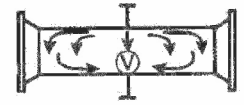
X: Crossflow

Multi Shell & Tube Passes

SHELL FLUID

2 SHELLS

TUBE FLUID

TWO SHELL PASSES
4, 8, 12, · · ·, TUBE PASSES

SHELL FLUID
MIXED

TUBE FLUID

SHELL FLUID

3 SHELLS

TUBE FLUID

THREE SHELL PASSES
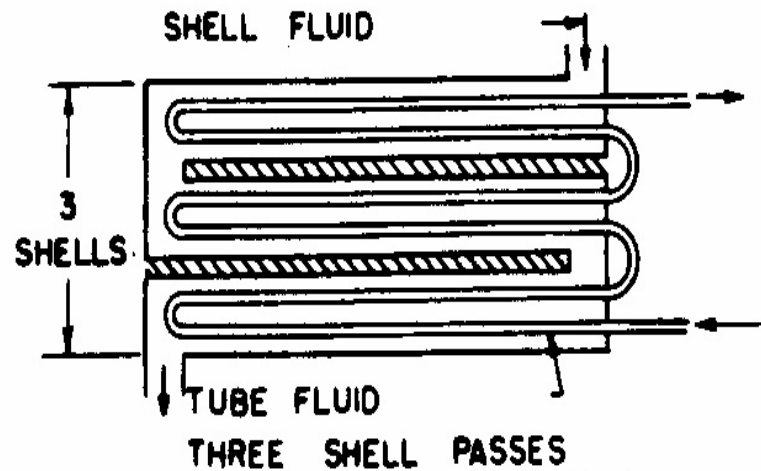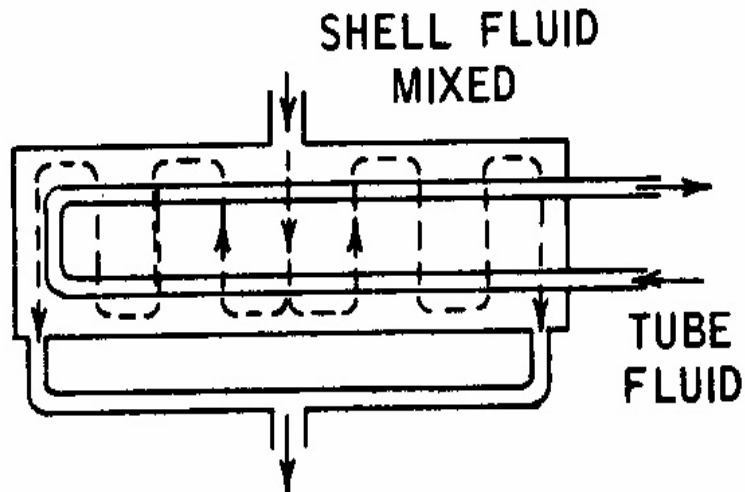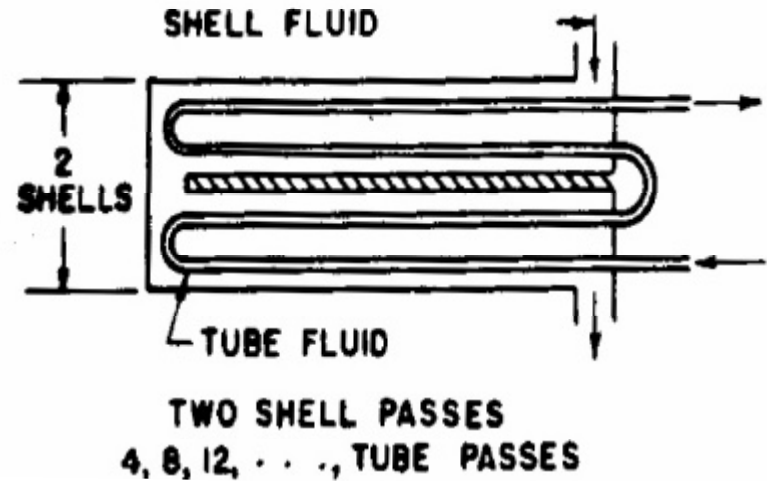
Tube to Header Plate Connection

- ❑ Tubes are arranged in a **bundle** and held in place by **header plate** (tube sheet).
- ❑ The number of tubes that can be placed within a shell depends on
  - Tube layout, tube outside diameter, pitch, number of passes and the shell diameter.
- ❑ When the tubes are to close to each other, the header plate becomes to weak.
- ❑ Methods of attaching tubes to the header plate

Header Plate

Header Plate

Tube

(a) Simple

(b) Trepanned

(c) Forged spigot

(d) Welded spigot
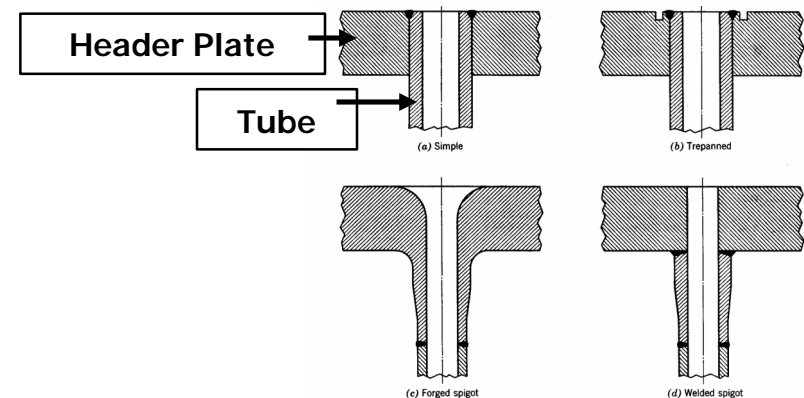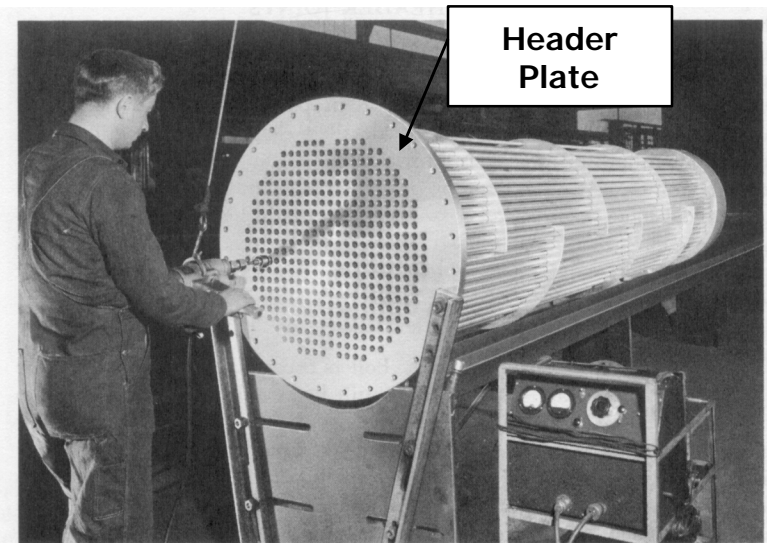
Baffle Type & Geometry

❑ Baffles serve two functions:

- **Support the tubes** for structural rigidity, preventing tube vibration and sagging

- **Divert the flow** across the bundle to obtain a higher heat transfer coefficient.

FIGURE 8.4. *Sketch of segmental cut baffles and location in shell.*

FIGURE 8.5. *Sketch of doughnut and disc baffles and location in shell.*

FIGURE 8.6. *Sketch of orifice baffle and location in shell.*
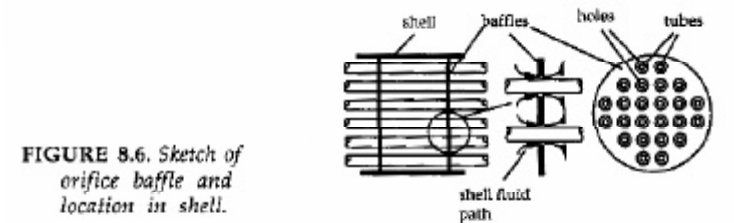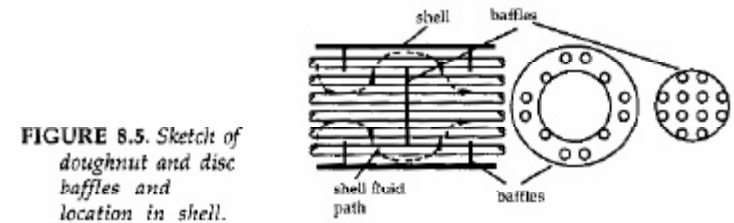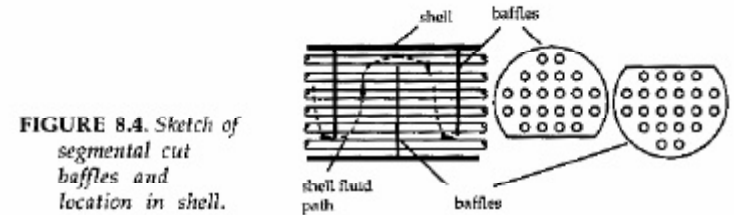
Segmental Cut Baffles
Baffle Type & Geometry

❑ The single and double segmental baffles are most frequently used. They divert the flow most effectively across the tubes.

❑ The baffle spacing must be chosen with care.

- Optimal baffle spacing is somewhere between 40% - 60% of the shell diameter.

- Baffle cut of 25%-35% is usually recommended.

❑ The triple segmental baffles are used for low pressure applications.
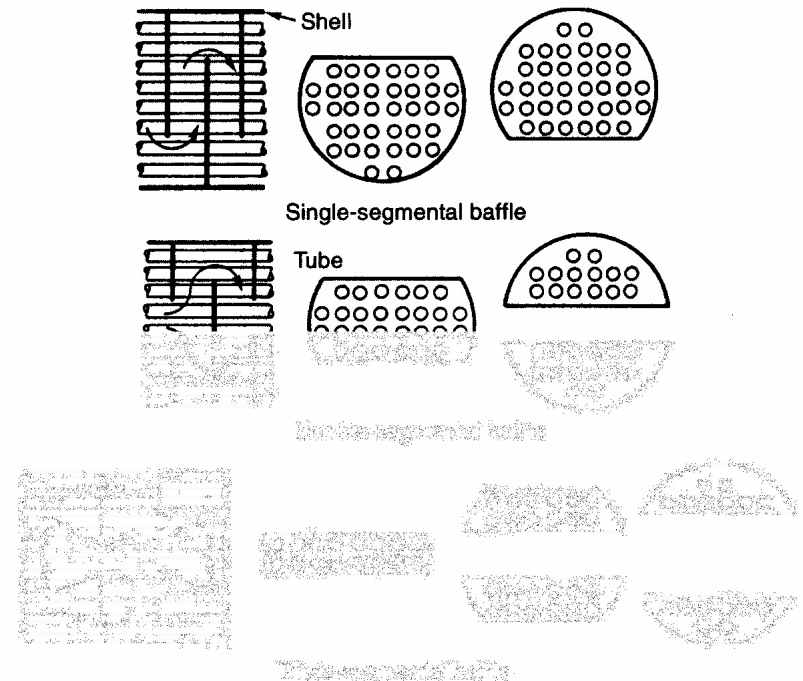
Disc & Ring Baffles
Baffle Type & Geometry

❑ Disc and ring baffles are composed of alternating outer rings and inner discs, which direct the flow radially across the tube field.

  ▪ The potential bundle-to-shell bypass stream is eliminated

  ▪ This baffle type is very effective in pressure drop to heat transfer conversion

❑ Disc

No-tubes-in-window segmental baffle
Doughnut
Disk
Disk and doughnut baffle
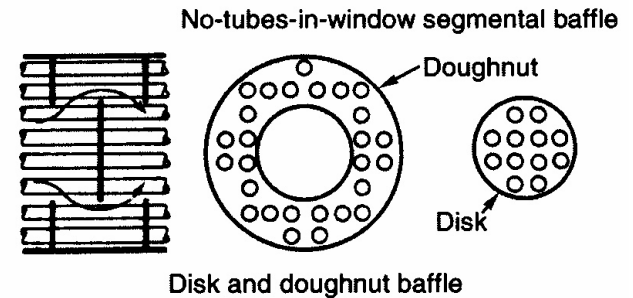
Orifice Baffle
Baffle Type & Geometry

❑ In an orifice baffle shell-side-fluid flows through the clearance between tube outside diameter and baffle-hole diameter.

Orifice baffle
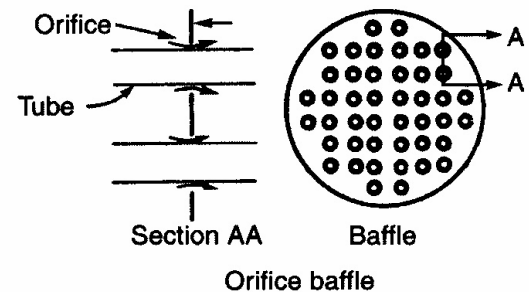
Number of Tubes

❑ The number of tubes in an exchanger depends on the

- Fluid flow rates

- Available pressure drop.

❑ The number of tubes is selected such that the

- **Tube side velocity** for water and similar liquids ranges from 0.9 to 2.4 m/s (3 to 8 ft/sec)

- **Shell-side velocity** from 0.6 to 1.5 m/s (2 to 5 ft/sec).

❑ The lower velocity limit corresponds to limiting the **fouling**, and the upper velocity limit corresponds to limiting the rate of **erosion**.

❑ When sand and silt are present, the velocity is kept high enough to prevent settling.

Tube Passes

- **A pass** is when liquid flows all the way across from one end to the other of the exchanger. We will count *shell passes* and *tube passes*.

    - An exchanger with one shell pass and two tube passes is a 1-2 exchanger. Almost always, the tube passes will be in multiples of two (1-2, 1-4, 2-4, etc.)

    - Odd numbers of tube passes have more complicated mechanical stresses, etc. An exception: 1-1 exchangers are sometimes used for vaporizers and condensers.

- A large number of tube passes are used to increase the tube side fluid velocity and heat transfer coefficient and minimize fouling.

    - This can only be done when there is enough pumping power since the increased velocity and additional turns increases the pressure drop significantly.

Tube Passes - Continued

❑ The number of tube passes depends on the available pressure drop.

  ■ Higher velocities in the tube result in higher heat transfer coefficients, at the expense of increased pressure drop.

❑ Therefore, if a higher pressure drop is acceptable, it is desirable to have fewer but longer tubes (reduced flow area and increased flow length).

  ■ Long tubes are accommodated in a short shell exchanger by multiple tube passes.

❑ The number of tube passes in a shell generally range from 1 to 10

  ■ The standard design has one, two, or four tube passes.

  ■ An odd number of passes is uncommon and may result in mechanical and thermal problems in fabrication and operation.

Tube Materials

❑ **Materials selection** and compatibility between construction materials and working fluids are important issues, in particular with regard to **corrosion** and/or operation at **elevated temperatures**.

❑ Requirement for low cost, light weight, high conductivity, and good joining characteristics often leads to the selection of aluminum for the heat transfer surface.

❑ On the other side, stainless steel is used for food processing or fluids that require corrosion resistance.

❑ In general, one of the selection criteria for exchanger material depends on the corrosiveness of the working fluid.

❑ A summary Table is provided as a reference fo rcorrosive and non-corrosive environments

Materials for Corrosive & Noncorrosive Service

Material	Heat Exchanger Type or Typical Service
Noncorrosive Service	
Aluminum and austenitic chromium–nickel steel	Any heat exchanger type, $T < -100°C$
$3\frac{1}{2}$ Ni steel	Any heat exchanger type, $-100 < T < -45°C$
Carbon steel (impact tested)	Any heat exchanger type, $-45 < T < 0°C$
Carbon steel	Any type of heat exchanger, $0 < T < 500°C$
Refractory-lined steel	Shell-and-tube, $T > 500°C$
Corrosive Service	
Carbon steel	Mildly corrosive fluids; tempered cooling water
Ferritic carbon–molybdenum and chromium–molybdenum alloys	Sulfur-bearing oils at elevated temperatures (above 300°C); hydrogen at elevated temperatures
Ferritic chromium steel	Tubes for moderately corrosive service; cladding for shells or channels in contact with corrosive sulfur bearing oil
Austenitic chromium–nickel steel	Corrosion-resistant duties
Aluminum	Mildly corrosive fluids
Copper alloys: admiralty, aluminum brass, cupronickel	Freshwater cooling in surface condensers; brackish and seawater cooling
High nickel–chromium–molybdenum alloys	Resistance to mineral acids and Cl-containing acids
Titanium	Seawater coolers and condensers, including PHEs
Glass	Air preheaters for large furnaces
Carbon	Severely corrosive service
Coatings: aluminum, epoxy resin	Exposure to sea and brackish water
Linings: lead and rubber	Channels for seawater coolers
Linings: austenitic chromium–nickel steel	General corrosion resistance

Source: Data from Lancaster (1998).

Tube Wall Thickness

❑ The wall thickness of heat exchanger tubes is standardized in terms of Birmingham Wire Gage BWG of the tube.

❑ Small tube diameters (8 to 15mm) are preferred for greater area to volume density but are limited for the purposes of cleaning.

❑ Large tube diameters are often required for condensers and boilers.

Tube Outside Diameter

❑ The most common plain tube sizes have 15.88, 19.05, and 25.40 mm (5/8, ¾, 1 inche) tube outside diameters.

❑ From the heat transfer viewpoint, smaller-diameter tubes yield higher heat transfer coefficients and result in a more compact exchanger.

❑ However, larger-diameter tubes are easier to clean and more rugged.

❑ The foregoing common sizes represent a compromise.

 ▪ For mechanical cleaning, the smallest practical size is 19.05 mm.

 ▪ For chemical cleaning, smaller sizes can be used provided that the tubes never plug completely.

Tube Length

❑ Tube length affects the cost and operation of heat exchangers.

  ▪ Longer the tube length (for any given surface area),

    • Fewer tubes are needed, requiring less complicated header plate with fewer holes drilled
    • Shell diameter decreases resulting in lower cost

❑ Typically tubes are employed in 8, 12, 15, and 20 foot lengths. Mechanical cleaning is limited to tubes 20 ft and shorter, although standard exchangers can be built with tubes up to 40 ft.

❑ There are, like with anything limits of how long the tubes can be.

  ▪ **Shell-diameter-to-tube-length ratio** should be within limits of 1/5 to 1/15

❑ Maximum tube length is dictated by

  ▪ Architectural layouts

  ▪ Transportation (to about 30m.)

    • The diameter of the two booster rockets is dictated by the smallest highway tunnel size between the location of manufacturer and Florida. Scientific hah!

Tube Length
Tube & Header Plate Deformation

❑ Thermal expansion of tubes needs to be taken into account for heat exchangers operating at elevated temperatures

❑ Tube elongation due to thermal expansion causes:

- Header plate deformation
- Shell wall deformation near the header plate

❑ Fatigue strength of the tube, header plate and shell joint needs to be considered when using

- Longer tubes
- High operating tube side temperatures
- Cyclic thermal loads

Magnified Displacement of a Shell & Tube Heat Exchanger Elements Under Thermal Load

Undeformed Shell Wall

Shell Wall Deformation

Principal Units: Custom

1.283e-01
1.202e-01
1.122e-01

Header Plate Deformation

Undeformed Header Plate Shape
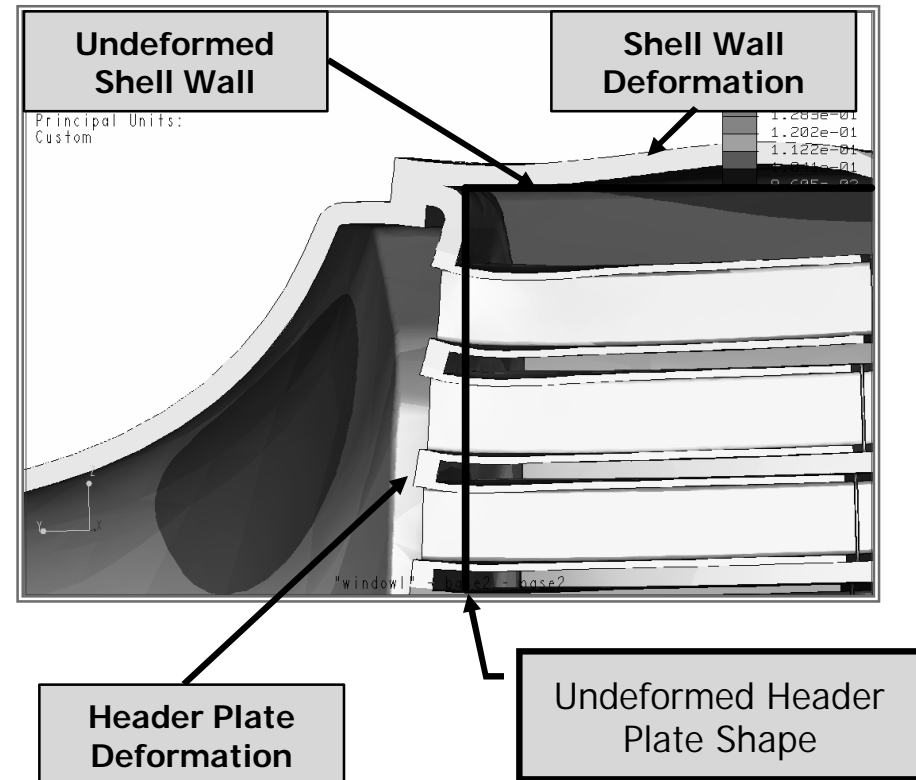
Tube Layout

❑ Tube layout is characterized by the included angle between tubes.

  ■ Two standard types of tube layouts are the **square** and the **equilateral triangle**.

    • Triangular pitch (30° layout) is better for heat transfer and surface area per unit length (greatest tube density.)

    • Square pitch (45 & 90 layouts) is needed for mechanical cleaning.

  ■ Note that the 30°,45° and 60° are staggered, and 90° is in line.

❑ For the identical tube pitch and flow rates, the tube layouts in decreasing order of shell-side heat transfer coefficient and pressure drop are: 30°,45°,60°, 90°.

Triangular

Rotated Square

Flow →

P_T

30°

Flow →

45°

P_T

Flow →

Square

90°

Flow →

60°

Rotated Triangle

(a) square pitch   (b) square pitch rotated   **Triangular**
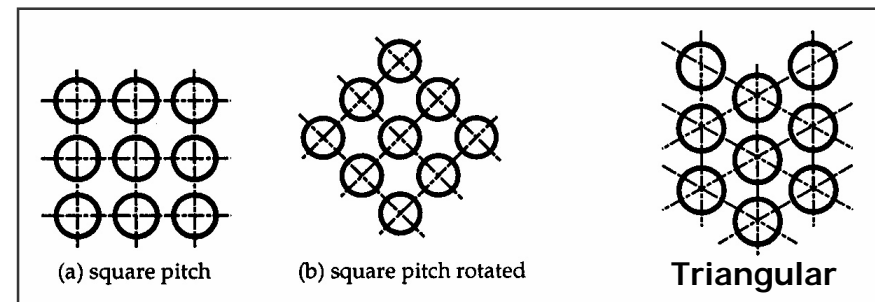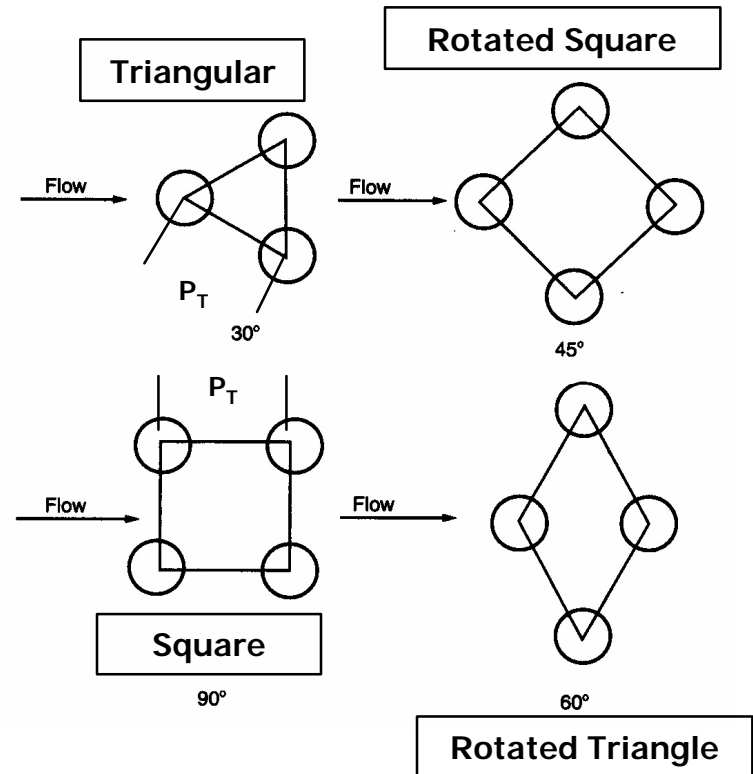
Tube Layout - Continued

❑ **The 90° layout** will have the lowest heat transfer coefficient and the lowest pressure drop.

❑ **The square pitch** (90° or 45°) is used when jet or mechanical cleaning is necessary on the shell side. In that case, a minimum cleaning lane of ¼ in. (6.35 mm) is provided.

▪ The square pitch is generally not used in the fixed header sheet design because cleaning is not feasible.

❑ **The triangular pitch** provides a more compact arrangement, usually resulting in smaller shell, and the strongest header sheet for a specified shell-side flow area.

▪ It is preferred when the operating pressure difference between the two fluids is large.

Tube Pitch

- ❑ **The selection of tube pitch** is a compromise between a
  - ▪ **Close pitch** (small values of P_t/d_o) for increased shell-side heat transfer and surface compactness, and an
  - ▪ **Open pitch** (large values of P_t/d_o) for decreased shell-side plugging and ease in shell-side cleaning.
- ❑ Tube pitch P_T is chosen so that the **pitch ratio** is $1.25 < P_T/d_o < 1.5$
  - ▪ When the tubes are to close to each other (P_t/d_o less than 1.25) , the header plate (tube sheet) becomes to weak for proper rolling of the tubes and cause leaky joints.
- ❑ Tube layout and tube locations are standardized for industrial heat exchangers.
  - ▪ However, these are general rules of thumb and can be "violated" for custom heat exchanger designs.

Tube & Shell Exhaust Gas Cooler
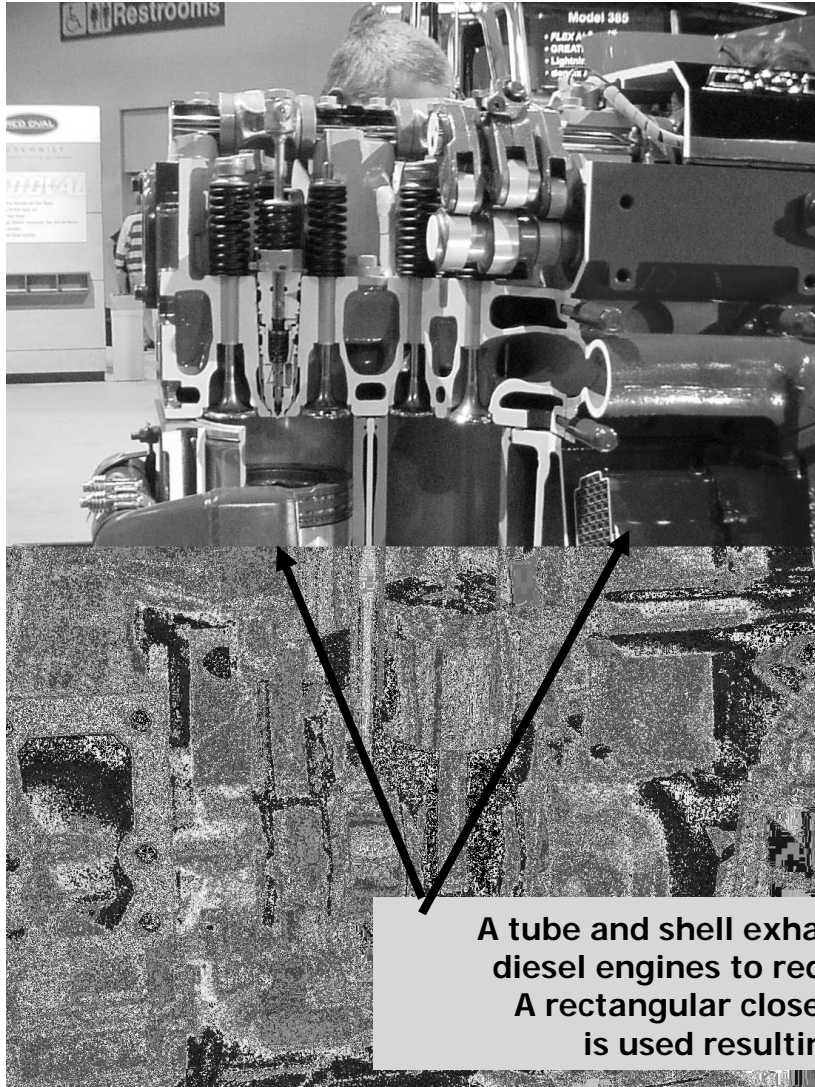
A tube and shell exhaust gas cooler is used on diesel engines to reduce the NOx emissions.
A rectangular closely packed tube arrangement is used resulting in a rectangular shell.

Fluid Allocation

- ❑ Tube side is preferred under these circumstances:
  - ▪ Fluids which are prone to foul
    - • The higher velocities will reduce buildup
    - • Mechanical cleaning is also much more practical for tubes than for shells.
  - ▪ Corrosive fluids are usually best in tubes
    - • Tubes are cheaper to fabricate from exotic materials
    - • This is also true for very high temperature fluids requiring alloy construction
  - ▪ Toxic fluids to increase containment
  - ▪ Streams with low flow rates to obtain increased velocities and turbulence
  - ▪ High pressure streams since tubes are less expensive to build strong
  - ▪ Streams with a low allowable pressure drop
- ❑ Viscous fluids go on the shell side, since this will usually improve the rate of heat transfer.

  - ▪ On the other hand, placing them on the tube side will usually lead to lower pressure drops. Judgment is needed

Basic Design Procedure

❑ Heat exchanger must satisfy the

- Heat transfer requirements (design or process needs)

- Allowable pressure drop (pumping capacity and cost)

❑ Steps in designing a heat exchanger can be listed as:

- Identify the problem

- Select an heat exchanger type

- Calculate/Select initial design parameters

- Rate the initial design
  - Calculate thermal performance and pressure drops for shell and tube side

- Evaluate the design
  - Is performance and cost acceptable?

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┌─────────────────┐
│     Problem     │
│  Identification │
└─────────────────┘
         │
         ▼                    The Elements Within this
┌─────────────────┐           Box May Be Done By
│ Selection of a  │           Hand or Computer
│ Basic Heat      │
│ Exchanger Type  │
└─────────────────┘
         │
         ▼
┌─────────────────┐
│ Selection of a  │
│ Tentative Set   │          ┌──────────────┐
│ of Exchanger    │          │ Modification │
│ Design          │          │ of the Design│
│ Parameters      │          │ Parameters   │
└─────────────────┘          └──────────────┘
         │
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┌─────────────────┐
│ Rating of the   │
│ Design:         │
│ Thermal         │
│ Performance     │
│ Pressure Drops  │
└─────────────────┘
         │
         ▼
┌─────────────────┐
│ Evaluation of   │
│ the Design:     │
│ Q, ΔP           │    Unacceptable
│ Acceptable?     │
└─────────────────┘
         │ Acceptable
         ▼
┌─────────────────┐
│ Mechanical      │
│ Design, Costing,│
│ Etc.            │
└─────────────────┘
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Size of Heat Exchanger

❑ The **initial size (surface area)** of a heat exchanger can be estimated from ➡

$$A_o = \frac{q}{U_o \Delta T_m} = \frac{q}{U_o F \Delta T_{lm,cf}}$$

 ▪ where

 • A_o Outside tube surface area
 • q Heat duty – heat exchange between tube and shell side
 • U_o Overall heat transfer coefficient
 • F Correction factor F=1.0 for cross flow heat exchanger
 • $?T_m$ True mean temperature → $?T_m$ = F $?T_{lm}$
 • $?T_{lm}$ Log mean temperature difference (Est of true mean temperature)

  ▪ **Correction Factor F** is be covered in module TFD-HE4 Log-Mean Temperature Difference

Overall Heat Transfer Coefficient

❑ The **overall heat transfer coefficient U$_o$** based on the outside diameter of tubes can be estimated from:

▪ The individual heat transfer coefficients (h)

▪ Shell wall, outside & inside tube fouling resistances (R$_w$, R$_{fo}$, R$_{fi}$)

▪ Overall surface efficiency (?$_i$ & ?$_o$)

$$\frac{1}{U_o} = \frac{A_o}{A_i}\left(\frac{1}{\boldsymbol{h}_i h_i} + \frac{R_{fi}}{\boldsymbol{h}_i}\right) + A_o R_w + \frac{R_{fo}}{\boldsymbol{h}_o} + \frac{1}{\boldsymbol{h}_o h_o}$$

Heat Balance of
Shell & Tube Heat Exchanger

❑ Heat load of a heat exchanger can be estimated from heat balance:

$$q = \left(\dot{m}c_p\right)_c \left(T_{c,o} - T_{c,i}\right) = \left(\dot{m}c_p\right)_h \left(T_{h,i} - T_{h,o}\right)$$

■ If three of the temperatures are given, the fourth can be calculated using the above equation.

■ The above equation assumes no phase change in any of the fluids.

$T_{c,i}$

Shell
inlet

Tube
inlet

Tube
outlet

$T_{h,i}$

$T_{c,o}$ Shell
outlet

$T_{h,o}$
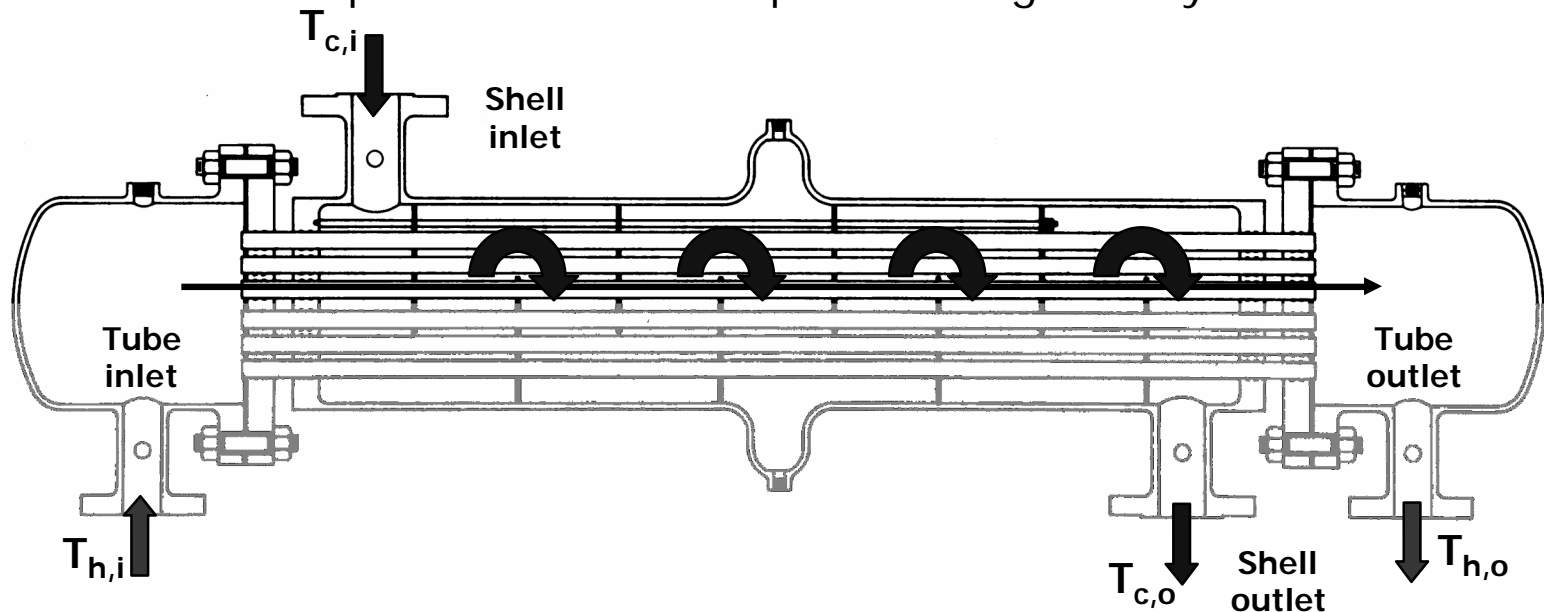
Other TFD Modules Supporting Shell & Tube Heat Exchangers

❑ Overall heat transfer coefficient is covered in module TFD-HE01

❑ Log-mean temperature difference is covered in module TFD-HE4

❑ Heat transfer from finned surfaces is covered in module TFD-HE11

Total Number of Tubes

❑ Once the total tube outside surface area A_o is estimated a cost effective heat exchanger configuration needs to be calculated.

❑ **Number of tubes N_t** is dependent on tube side flow conditions. It is related to the **shell diameter (D_s)**, **tube length (L)** and **tube diameter (d_o)** together with the allowable pressure drop and the total tube side flow rate hence the heat transfer coefficient.

$$A_o = \pi d_o N_t L$$

Total Number of Tubes

❑ The total number of tubes can be predicted as a function of the shell diameter by taking the shell circle D_s and dividing it by the projected area of the tube layout pertaining to a single tube A_1

$$N_t = (CTP)\frac{pD_s^2}{4A_1} \qquad\qquad A_1 = (CL)P_T^2$$

❑ CTP is the tube count constant which accounts for the incomplete coverage of the shell diameter by the tubes due to necessary clearances between the shell and the outer tube circle.

- CTP=0.93   One tube pass
- CTP=0.90   Two tube passes
- CTP=0.85   Three tube passes

CL - Tube Layout Constant

- CL=1.00   for 90 & 45 square pitch
- CL=0.87   for 30 & 60 equilateral tri pitch

Shell Diameter

❑ Shell diameter in terms of main constructional diameters can be expressed as:

$$N_t = 0.785 \left(\frac{CTP}{CL} \right) \frac{D_s^2}{\left(\frac{P_T}{d_o} \right)^2 d_o^2}$$

$$A_o = \boldsymbol{p} d_o N_t L$$

$$D_s = 0.637 \sqrt{\frac{CL}{CTP}} \left[\frac{A_o \left(P_T / d_o \right)^2 d_o}{L} \right]^{\frac{1}{2}}$$

Rating of the Heat Exchanger Design

❑ **Rating an exchanger means** to evaluate the thermo-hydraulic performance of a **fully specified** exchanger.

❑ Input to the rating process is heat exchanger **geometry** (constructional design parameters), **process conditions** (flow rate, temperature, pressure) and **material/fluid properties** (density, thermal conductivity)

❑ **First output** from the rating process is either the outlet temperature for fixed tube length or the tube length itself to meet the outlet temperature requirement.

❑ **Second output** from the rating process is the pressure drop for both fluid streams hence the pumping energy requirements and size.
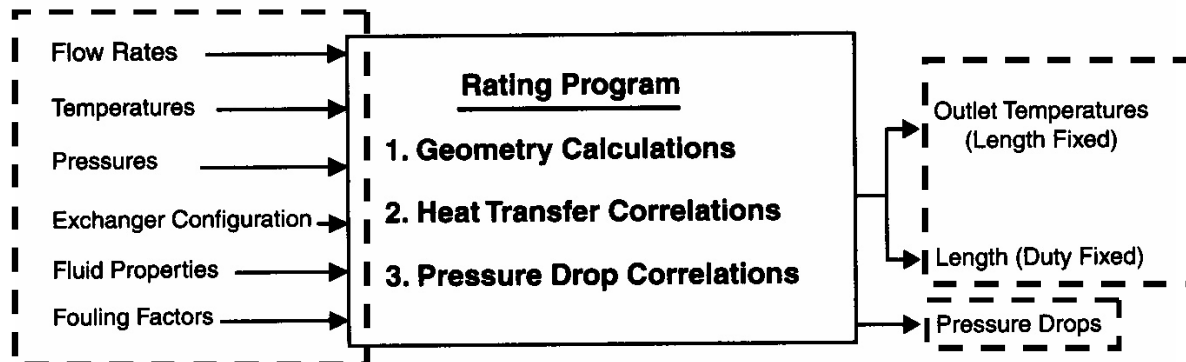
Insufficient Thermal Rating

❑ If the output of the rating analysis is not acceptable, a geometrical modification should be made

❑ If the required amount of heat cannot be transferred to satisfy specific outlet temperature, one should find a way to increase the heat transfer coefficient or increase exchanger surface area

- One can increase the tube side heat transfer coefficient by increasing the fluid velocity - Increase number of tube passes

- One can increase the shell side heat transfer coefficient by decreasing baffle spacing and/or baffle cut

- One can increase the surface area by
    - Increasing the heat exchanger length
    - Increasing the shell diameter
    - Multiple shells in series

Insufficient Pressure Drop Rating

- ❑ If the pressure drop on the tube side is greater than the allowable pressure drop, then
    - ▪ the number of tube passes can be decreased or
    - ▪ the tube diameter can be increased which may result to
        - • decrease the tube length – (Same surface area)
        - • increase the shell diameter and the number of tubes
- ❑ If the shell side pressure drop is greater than the allowable pressure drop then baffle spacing, tube pitch, and baffle cut can be increased or one can change the baffle type.

THERE IS ALWAYS A TRADE-OFF BETWEEN THERMAL & PRESSURE DROP RATINGS!

The Trade-Off
Between Thermal Balance & Flow Loss

❑ Heat transfer and fluid friction losses tend to compete with one another.

❑ The total energy loss can be minimized by adjusting the size of one irreversibility against the other .

❑ These adjustments can be made by properly selecting physical dimensions of the solid parts (fins, ducts, heat exchanger surface).

❑ It must be understood, however, that the result is at best a thermodynamic optimum.

  ■ Constraints such as cost, size, and reliability enter into the determination of truly optimal designs.

Shell Side Heat Transfer Coefficient

❑ There are three **rating methods** to calculate the shell side heat transfer coefficient:

- ▪ **Kern method** is a simplified approach suitable for shell side flow without baffles

- ▪ **Taborek** method

- ▪ **Bell Delaware** method is the most complex but accurate way of rating a heat exchanger with baffles

SHELL SIDE
HEAT TRANSFER COEFFICIENT
WITH BAFFLES

Shell Side Heat Transfer
Baffled Flow

❏ **When the tube bundle employs baffles**, the heat transfer coefficient is higher than the coefficient for undisturbed flow around tubes without baffles.

❏ For a baffled heat exchanger

  ▪ the higher heat transfer coefficients result from the **increased turbulence**.

  ▪ the **velocity of fluid fluctuates** because of the constricted area between adjacent tubes across the bundle.

❏ **Only part of the fluid takes the desired path** through the tube bundle (Stream B), whereas a potentially substantial portion flows through the 'leakage' areas (Streams A, C, E & F)

  ▪ However, these clearances are inherent to the manufacturing and assembly process of shell-and-tube exchangers, and the flow distribution within the exchanger must be taken into account.

Main & Leakage Flow Streams
Baffled Heat Exchanger

❑ There are five different shell side flow streams in a baffled heat exchanger

- **Stream A** is the leakage stream in the orifice formed by the clearance between the baffle tube hole and the tube wall.

- **Stream B** is the main effective cross-flow stream, which can be related to flow across ideal tube banks.

- **Stream C** is the tube bundle bypass stream in the gap between the tube bundle and shell wall.

- **Stream E** is the leakage stream between the baffle edge and shell wall.

- **Stream F** is the bypass stream in flow channel partitions due to omissions of tubes in tube pass partitions.

Main & Leakage Flow Streams
Baffled Heat Exchanger

Pass 2

Pass 1

B
B
B

B
B
B
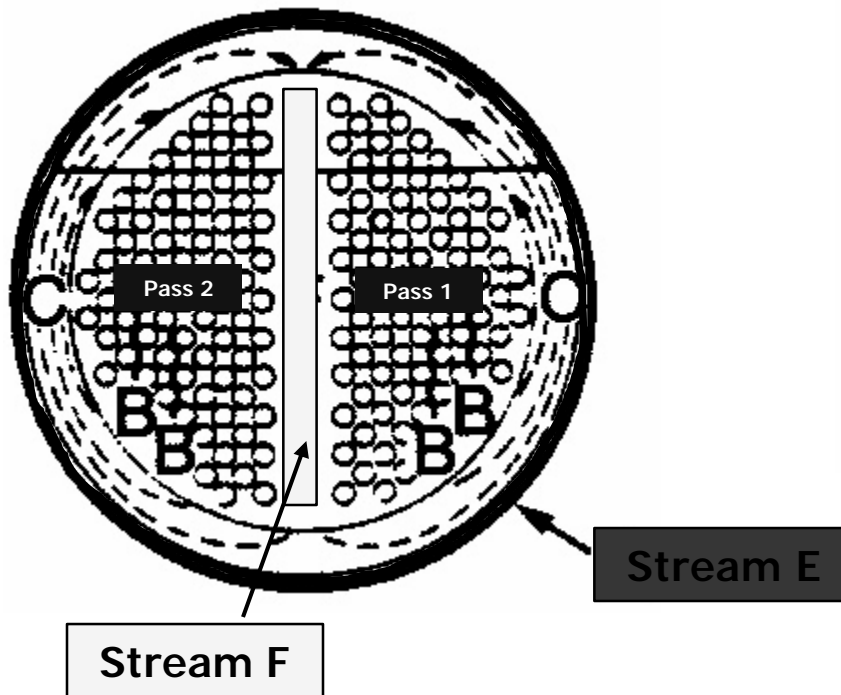
Stream E

Stream F

Stream F happens in a multiple pass (1-2, 1-4) heat exchanger

shell

baffles

holes

tubes

Stream A

Stream C

shell

ba

shell fluid path

ba
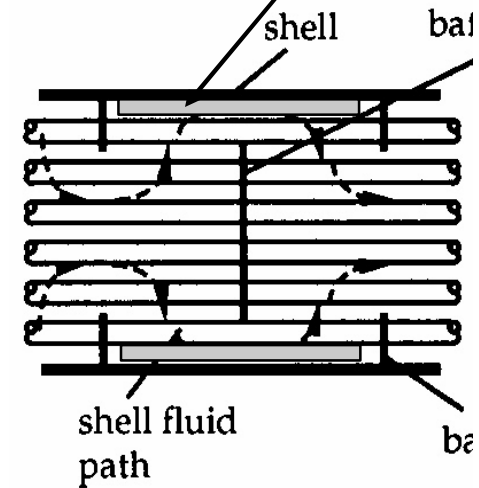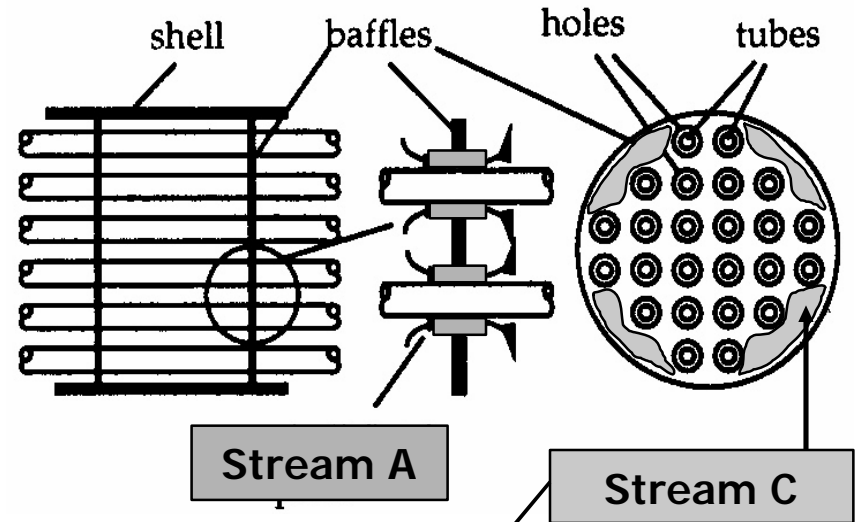
Bell Delaware Method
Heat Transfer Coefficient & Correction Factors

❑ In the Delaware method, the fluid flow in the shell is divided into a number of individual streams A through F as defined before.

❑ Each of the above streams introduces a correction factor to the heat transfer correlation for ideal cross-flow across a bank of tubes.

$$h_o = h_{ideal} J_c J_l J_b J_s J_r \quad \Longleftarrow \quad h_{ideal} = j_i c_{ps} \left(\frac{\dot{m}_s}{A_s} \right) \left(\frac{k_s}{c_{ps} \mu_s} \right)^{2/3} \left(\frac{\mu_s}{\mu_{s,w}} \right)^{0.14}$$

- h_{ideal} heat transfer coefficient for pure cross-flow in an ideal tube bank
- J_c for baffle cut and spacing
- J_l for leakage effects
- J_b bundle bypass flow C & F streams
- J_s for variable baffle spacing in the inlet and outlet sections
- J_r for adverse temperature gradient build-up

- j_i Colburn j-factor
- A_s Cross flow area at the centerline of shell for one cross flow between two baffles
- s   Stands for shell
- w   Wall temperature

The combined effects of all these correction factors for a reasonable well-designed shell-and-tube heat exchanger is of the order of 0.60

Bell Delaware Method
J_c *Correction Factor*

❑ J_c is the correction factor for baffle cut and spacing. This factor takes into account the heat transfer in the window and calculates the overall average heat transfer coefficient for the entire heat exchanger.

❑ It depends on the shell diameter and the baffle cut distance from the baffle tip to the shell inside diameter.

  ■ For a large baffle cut, this value may decrease to a value of 0.53

  ■ it is equal to 1.0 for a heat exchanger with no tubes in the window

  ■ It may increase to a value as high as 1.15 for small windows with a high window velocity.

Bell Delaware Method
J_l *Correction Factor*

❑ J_l is the correlation factor for baffle leakage effects including tube-to-baffle and shell-to-baffle leakage (A- and E-streams).

❑ If the baffles are put too close together, then the fraction of the flow in the leakage streams increases compared with the cross flow.

❑ J_l is a function of the

  ▪ ratio of total leakage area per baffle to the cross flow area between adjacent baffles

  ▪ ratio of the shell-to-baffle leakage area to the tube-to-baffle leakage area.

❑ A typical value of J_l is in the range of 0.7 and 0.8.

Bell Delaware Method
J_b *Correction Factor*

❑ J_b is the correction factor for bundle bypassing effects due to the clearance between the outermost tubes and the shell and pass dividers (C- and F-streams).

- For relatively small clearance between the outermost tubes and the shell for fixed tube sheet construction, $J_b = 0.90$.

- For a pull-through floating head, larger clearance is required, $J_b = 0.7$.

- The sealing strips (see figure8.14) can increase the value of J_b

Stream F
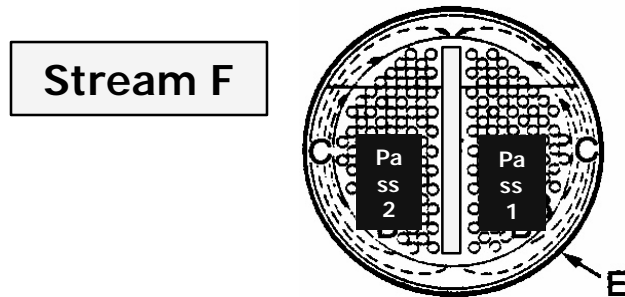
Bell Delaware Method
J_s & J_r *Correction Factors*

❑ J_s is the correction factor for variable baffle spacing at the inlet and outlet. Because of the nozzle spacing at the inlet and outlet and the changes in local velocities, the average heat transfer coefficient on the shell side will change.

❑ The J_s value will usually be between 0.85 and 1.00.

❑ J_r applies if the shell-side Reynolds number, Re_s, is less than 100.

   ▪ If $Re_s < 20$, it is fully effective.

   ▪ This factor is equal to 1.00 if $Re_s > 100$.

The combined effect of
all these correction factors
for a well-designed shell-and-tube heat exchanger
is of the order of 0.60

Bell Delaware Method
Heat Transfer Coefficient - Colburn j-factor

❑ Colburn-j factor is used in heat transfer in general and free and forced convection calculations in particular.

  ▪ It is equivalent to (St.Pr$^{2/3}$) where St is Stanton number where Stanton Number is defined as

$$S_t = \frac{h}{Gc_p} = \frac{h}{(rV_{max})c_p} = \frac{h}{\dfrac{\dot{m}}{A_{min}}c_p}$$

▪ G is the mass velocity

▪ A_{min} is the min free flow x-sec area regardless where it occurs

❑ Colburn j-factor is a function of:

  ▪ **Shell side Reynolds number** based on the outside tube diameter and on the minimum cross section flow area at the shell diameter

$$\left.\begin{array}{c} \\ \\ \\ \end{array}\right\} \quad \mathrm{Re}_s = \frac{d_o \dot{m}_s}{m_s A_s}$$

  ▪ Tube layout

  ▪ Pitch size

Bell Delaware Method
Numerical Forms of Colburn (j) & Friction (f) Factors

❑ Although the ideal values of j and f are available in graphical forms, for computer analysis, a set of curve-fit correlations are obtained in the following forms:

Colburn j-factor

$$j_i = a_1 \left(\frac{1.33}{P_T / d_o} \right)^a (\mathrm{Re}_s)^{a_2}$$

$$a = \frac{a_3}{1 + 0.14(\mathrm{Re}_s)^{a_4}}$$

Friction factor

$$f = b_1 \left(\frac{1.33}{P_T / d_o} \right)^b (\mathrm{Re}_s)^{b_2}$$

$$b = \frac{b_3}{1 + 0.14(\mathrm{Re}_s)^{b_4}}$$

TABLE 8.6

Correlation Coefficients for j_i and f_i Equations (8.25) and (8.26)

Layout Angle	Reynolds Number	a_1	a_2	a_3	a_4	b_1	b_2	b_3	b_4
30°	10^5–10^4	0.321	−0.388	1.450	0.519	0.372	−0.123	7.00	0.500
	10^4–10^3	0.321	−0.388			0.486	−0.152		
	10^3–10^2	0.593	−0.477			4.570	−0.476		
	10^2–10	1.360	−0.657			45.100	−0.973		
	<10	1.400	−0.667			48.000	−1.000		
45°	10^5–10^4	0.370	−0.396	1.930	0.500	0.303	−0.126	6.59	0.520
	10^4–10^3	0.370	−0.396			0.333	−0.136		
	10^3–10^2	0.730	−0.500			3.500	−0.476		
	10^2–10	0.498	−0.656			26.200	−0.913		
	<10	1.550	−0.667			32.00	−1.000		
90°	10^5–10^4	0.370	−0.395	1.187	0.370	0.391	−0.148	6.30	0.378
	10^4–10^3	0.107	−0.266			0.0815	+0.022		
	10^3–10^2	0.408	−0.460			6.0900	−0.602		
	10^2–10	0.900	−0.631			32.1000	−0.963		
	<10	0.970	−0.667			35.0000	−1.000		

SHELL SIDE
HEAT TRANSFER COEFFICIENT
WITHOUT BAFFLES

SHELL-and-TUBE HEAT EXCHANGER

Shell Side Heat Transfer Coefficient
Without Baffles – Flow Along the Tube Axis

❑ The heat transfer coefficient outside the tube bundle is referred to as the shell-side heat transfer coefficient.

❑ If there are no baffles, the flow will be along the heat exchanger inside the shell. Then, the heat transfer coefficient can be based on the equivalent diameter, D_e (Same as a double-pipe heat exchanger)

$$\frac{h_o D_e}{k} = 0.36 \, \mathrm{Re}^{0.55} \, \mathrm{Pr}^{1/3} \left[\frac{m_b}{m_w} \right]^{0.14}$$

❑ D_e Equivalent shell diameter

❑ G_s Shell side mass velocity

$$\frac{h_o D_e}{k} = 0.36 \left[\frac{D_e G_s}{m} \right]^{0.55} \left[\frac{c_p m}{k} \right]^{1/3} \left[\frac{m_b}{m_w} \right]^{0.14}$$

❑ b   Bulk fluid temperature

❑ w   Wall temperature

$$2 \times 10^3 < \mathrm{Re}_s = \frac{D_e G_s}{m} < 10^6$$

Equivalent Shell Diameter - D_e

❑ The equivalent diameter of the shell is taken as four times the net flow area as layout on the tube sheet (for my pitch layout) divided by the wetted perimeter:

$$D_e = \frac{4 \times \text{free flow area}}{\text{wetted perimeter}}$$

Rectangular Pitch
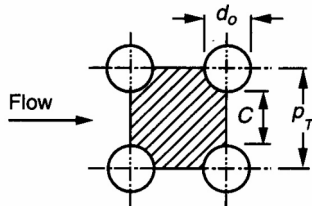
$$D_e = \frac{4(P_T^2 - \boldsymbol{p}d_o^2/4)}{\boldsymbol{p}d_o}$$

Triangular Pitch
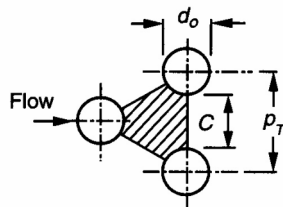
$$D_e = \frac{4(P_T^2\sqrt{3} - \boldsymbol{p}d_o^2/8)}{\boldsymbol{p}d_o/2}$$

Shell Side Mass Velocity - G_s

❑ There is no free-flow area on the shell side by which the shell-side mass velocity, G_s, can be calculated.

❑ For this reason, fictional values of G_s can be defined based on the bundle cross flow area at the hypothetical tube row possessing the maximum flow area corresponding to the center of the shell.

❑ Variables that affect the velocity are:

- Shell diameter **(D_s)**   Clearance between adjacent tubes **(C)**;
  Pitch size **(PT)**   Baffle spacing **(B)**

❑ The width of the flow area at the tubes located at center of the shell is (D_s/P_T) C and the length of the flow area is taken as the baffle spacing, B.

- Therefore, the bundle cross flow area A_s, ➡️ at the center of the shell is

$$A_s = \frac{D_s C B}{P_T}$$

❑ Shell side mass velocity is ➡️

$$G_s = \frac{\dot{m}}{A_s}$$

Shell Side Pressure Drop

❑ The shell-side pressure drop depends on the number of tubes the fluid passes through in the tube bundle between the baffles as well as the length of each crossing.

- ■ If the length of a bundle is divided by four baffles, for example, all the fluid travels across the bundle five times.

❑ A correlation has been obtained using the product of distance across the bundle, taken as the inside diameter of the shell, Ds and the number of times the bundle is crossed.

$$\Delta p_s = \frac{fG_s^2\left\{\left(\dfrac{L}{B}-1\right)+1\right\}D_s}{2\rho D_e\left(\mu_b/\mu_w\right)^{0.14}}$$

- ■ L is the heat exchanger length, B is the baffle spacing

❑ **Shell side friction coefficient** f includes the entrance and exit losses

$$f = \exp\{0.576 - 0.19\ln(\mathrm{Re}_s)\}$$

TUBE SIDE
HEAT TRANSFER COEFFICENT
&
FRICTION FACTOR

SHELL-and-TUBE HEAT EXCHAGER

Tube Side Heat Transfer Correlations

❑ **Extensive experimental and theoretical efforts** have been made to obtain the solutions for turbulent forced convection heat transfer and flow friction problems in ducts because of their frequent occurrence and application in heat transfer engineering.

❑ There are a **large number of correlations available** in the literature for the fully developed (hydro-dynamically and thermally) turbulent flow of single-phase Newtonian fluids in smooth, straight, circular ducts with constant and temperature-dependent physical properties.

❑ The objective of this section is to **highlight some of the existing correlations** to be used in the design of heat exchange equipment and to **emphasize the conditions or limitations** imposed on the applicability of these correlations.

❑ Extensive efforts have been made to obtain empirical correlations that represent a best-fit curve to experimental data or to adjust coefficients in the theoretical equations to best fit the experimental data.

Flow Maldistribution & Header Design

❑ One of the common assumptions in basic heat exchanger design theory is that fluid be distributed uniformly at the inlet of the exchanger on each fluid side and throughout the core.

▪ However, in practice, flow maldistribution is more common and can significantly reduce the desired heat exchanger performance.

❑ Flow maldistribution can be induced by heat exchanger

▪ Geometry - mechanical design features such as the basic geometry, manufacturing imperfections, and tolerances

▪ Operating conditions - viscosity or density induced mal distribution, multi phase flow, and fouling phenomena

Tube-to-Tube Velocity Variation

❑ In most cases, geometric flow entry & exit conditions to the headers promote a tube-2-tube velocity variation

Flow velocity distribution over the header plate before tube entrance for a rectangular x-sec heat exchanger

X-sectional area of the inlet pipe to the header plate may be smaller compared to the header plate area

Figure 1.8 A U-tube, baffled single-pass Patterson-Kelley Co.)

Header Plate

90 degree flow turn creates non-uniform velocity distribution inside the tubes
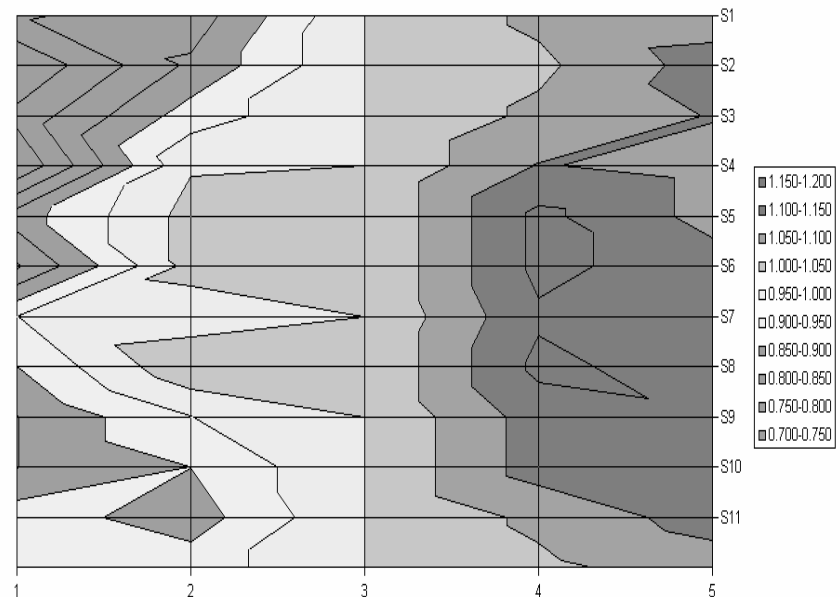
❑ **Nusselt correlations presented in this module assume an equally distributed flow between tubes**

▪ **Same velocity in each tube!**

Tube Side Heat Transfer Coefficient

❑ **Petukhov & Popov's** theoretical calculations for the case of fully developed turbulent flow with constant properties in a circular tube with constant heat flux boundary conditions fielded a correlation, which was based on the three-layer turbulent boundary layer model with constants adjusted to match the experimental data.

- Petukhov also gave a simplified form of this correlation as

$$Nu_b = \frac{(f/2)\operatorname{Re}_b \operatorname{Pr}_b}{1.07 + 12.7(f/2)^{0.5}\left(\operatorname{Pr}^{2/3} - 1\right)}$$

Where the friction factor f is defined as: ➡ $f = \left(1.58 \ln \operatorname{Re}_b - 3.28\right)^{-2}$

❑ This equation predicts results in the range of

- 10^4 < Re < 5×10^6 & 0.5 < Pr < 200 with 6% error
- 10^4 < Re < 5×10^6 & 0.5 < Pr < 2000 with 10% error

Tube Side Pressure Drop

❑ The **tube-side pressure drop** can be calculated by knowing the

- Number of tube passes, N_p

- Length of the heat exchanger, L

- Mean fluid velocity inside the tube, u_m

$$\Delta p_t = 4f \frac{LN_p}{d_i} \times \frac{1}{2} \rho u_m^2$$

$$\Delta p_t = 4f \frac{LN_p}{d_i} \times \frac{G_{tube}^2}{2\rho}$$

❑ The change of direction in the passes introduces an additional pressure drop, $?P_r$ due to sudden expansions and contractions that the tube fluid experiences during a return

$$\Delta p_r = 4N_r \times \frac{1}{2} \rho u_m^2$$

- This is accounted with four velocity heads per pass

❑ Total pressure drop than becomes

$$\Delta p_{total} = \left[4f \frac{LN_p}{d_i} + 4N_p \right] \times \frac{1}{2} \rho u_m^2$$

Roadmap To Increase Heat Transfer

❑ Increase heat transfer coefficent

 ▪ Tube Side

 • Increase number of tubes

 • Decrease tube outside diameter

 ▪ Shell Side

 • Decrease the baffle spacing

 • Decrease baffle cut

❑ Increase surface area

 ▪ Increase tube length

 ▪ Increase shell diameter → increased number of tubes

 ▪ Employ multiple shells in series or parallel

❑ Increase LMTD correction factor and heat exchanger effectiveness

 ▪ Use counterflow configuration

 ▪ Use multiple shell configuration

Roadmap To Reduce Pressure Drop

- ❑ Tube side
    - ▪ Decrease number of tube passes
    - ▪ Increase tube diameter
    - ▪ Decrease tube length and increase shell diameter and number of tubes
- ❑ Shell side
    - ▪ Increase the baffle cut
    - ▪ Increase the baffle spacing
    - ▪ Increase tube pitch
    - ▪ Use double or triple segmental baffles

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APPENDIX

Dimensional Data For Commercial Tubing

TABLE 8.1
Dimensional Data for Commercial Tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.²)	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Steel, (lb.)	ID Tubing (in.)	OD/ID
1/4	22	0.028	0.00296	0.0655	0.0508	0.066	0.194	1.289
1/4	24	0.022	0.00331	0.0655	0.0539	0.054	0.206	1.214
1/4	26	0.018	0.00350	0.0655	0.0560	0.045	0.214	1.168
3/8	18	0.049	0.00503	0.0982	0.0725	0.171	0.277	1.354
3/8	20	0.035	0.00731	0.0982	0.0798	0.127	0.305	1.233
3/8	22	0.028	0.00799	0.0982	0.0835	0.104	0.319	1.176
3/8	24	0.022	0.00860	0.0982	0.0867	0.083	0.331	1.133
1/2	16	0.065	0.1075	0.1309	0.0969	0.302	0.370	1.351
1/2	18	0.049	0.1249	0.1309	0.1052	0.236	0.402	1.244
1/2	20	0.035	0.1452	0.1309	0.1126	0.174	0.430	1.163
1/2	22	0.028	0.1548	0.1309	0.1162	0.141	0.444	1.126
5/8	12	0.109	0.1301	0.1636	0.1056	0.602	0.407	1.536
5/8	13	0.095	0.1486	0.1636	0.1139	0.537	0.435	1.437
5/8	14	0.083	0.1655	0.1636	0.1202	0.479	0.459	1.362
5/8	15	0.072	0.1817	0.1636	0.1259	0.425	0.481	1.299
5/8	16	0.065	0.1924	0.1636	0.1296	0.388	0.495	1.263
5/8	17	0.058	0.2035	0.1636	0.1333	0.350	0.509	1.228
5/8	18	0.049	0.2181	0.1636	0.1360	0.303	0.527	1.186
5/8	19	0.042	0.2298	0.1636	0.1416	0.262	0.541	1.155
5/8	20	0.035	0.2419	0.1636	0.1453	0.221	0.555	1.136
3/4	10	0.134	0.1825	0.1963	0.1262	0.894	0.482	1.556
3/4	11	0.120	0.2043	0.1963	0.1335	0.809	0.510	1.471
3/4	12	0.109	0.2223	0.1963	0.1393	0.748	0.532	1.410
3/4	13	0.095	0.2463	0.1963	0.1466	0.666	0.560	1.339
3/4	14	0.083	0.2679	0.1963	0.1529	0.592	0.584	1.284
3/4	15	0.072	0.2894	0.1963	0.1587	0.520	0.606	1.239
3/4	16	0.065	0.3019	0.1963	0.1623	0.476	0.620	1.210
3/4	17	0.058	0.3157	0.1963	0.1660	0.429	0.634	1.193
3/4	18	0.049	0.3339	0.1963	0.1707	0.367	0.652	1.150
3/4	20	0.035	0.3632	0.1963	0.1780	0.269	0.680	1.103
7/8	10	0.134	0.2892	0.2291	0.1589	1.061	0.607	1.441
7/8	11	0.120	0.3166	0.2291	0.1662	0.969	0.635	1.378
7/8	12	0.109	0.3390	0.2291	0.1720	0.891	0.657	1.332
7/8	13	0.095	0.3685	0.2291	0.1793	0.792	0.685	1.277
7/8	14	0.083	0.3948	0.2291	0.1856	0.704	0.709	1.234
7/8	16	0.065	0.4359	0.2291	0.1950	0.561	0.745	1.174
7/8	18	0.049	0.4742	0.2291	0.2034	0.432	0.777	1.126
7/8	20	0.035	0.5090	0.2291	0.2107	0.313	0.805	1.087
1	8	0.165	0.3526	0.2618	0.1754	1.462	0.670	1.493
1	10	0.134	0.4208	0.2618	0.1916	1.237	0.732	1.366
1	11	0.120	0.4536	0.2618	0.1990	1.129	0.760	1.316
1	12	0.109	0.4803	0.2618	0.2047	1.037	0.782	1.279
1	13	0.095	0.5153	0.2618	0.2121	0.918	0.810	1.235
1	14	0.083	0.5463	0.2618	0.2183	0.813	0.834	1.199
1	15	0.072	0.5755	0.2618	0.2241	0.714	0.856	1.167
1	16	0.065	0.5945	0.2618	0.2278	0.649	0.870	1.119

Dimensional Data For Commercial Tubing

TABLE 8.1 (CONTINUED)

Dimensional Data for Commercial Tubing

OD of Tubing (in.)	BWG Gauge	Thickness (in.)	Internal Flow Area (in.²)	Sq. Ft. External Surface per Ft. Length	Sq. Ft. Internal Surface per Ft. Length	Weight per Ft. Length, Steel (lb.)	ID Tubing (in.)	OD/ID
1	18	0.049	0.6390	0.2618	0.2361	0.496	0.902	1.109
1	20	0.035	0.6793	0.2618	0.2435	0.360	0.930	1.075
1-1/4	7	0.180	0.6221	0.3272	0.2330	2.057	0.890	1.404
1-1/4	8	0.165	0.6648	0.3272	0.2409	1.921	0.920	1.359
1-1/4	10	0.134	0.7574	0.3272	0.2571	1.598	0.982	1.273
1-1/4	11	0.120	0.8012	0.3272	0.2644	1.448	1.010	1.238
1-1/4	12	0.109	0.8365	0.3272	0.2702	1.329	1.032	1.211
1-1/4	12	0.095	0.8825	0.3272	0.2773	1.173	1.060	1.179
1-1/4	14	0.083	0.9229	0.3272	0.2838	1.033	1.084	1.153
1-1/4	16	0.065	0.9852	0.3272	0.2932	0.823	1.120	1.116
1-1/4	18	0.049	1.042	0.3272	0.3016	0.629	1.152	1.085
1-1/4	20	0.035	1.094	0.3272	0.3089	0.456	1.180	1.059
1-1/2	10	0.134	1.192	0.3927	0.3225	1.955	1.232	1.218
1-1/2	12	0.109	1.291	0.3927	0.3356	1.618	1.282	1.170
1-1/2	14	0.083	1.398	0.3927	0.3492	1.258	1.334	1.124
1-1/2	16	0.065	1.474	0.3927	0.3587	0.996	1.370	1.095
2	11	0.120	2.433	0.5236	0.4608	2.410	1.760	1.136
2	13	0.095	2.573	0.5236	0.4739	1.934	1.810	1.105
2-1/2	9	0.148	3.815	0.6540	0.5770	3.719	2.204	1.134

Source: Courtesy of the Tubular Exchanger Manufacturers Association.