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Revision Memo

12/01 Abbreviation for Tube Pitch changed from TP to PT.



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SCOPE

This section presents general design considerations for shell and tube heat exchangers.
For a diagram depicting standard components of shell and tube exchangers see **Figure 1**.

REFERENCES

DESIGN PRACTICES (BESIDES OTHER SECTIONS OF SECTION IX)

- Section I** Design Economics
Section XIV Fluid Flow

GLOBAL PRACTICE

- GP 6-1-1** TEMA Type Shell and Tube Heat Exchangers

OTHER LITERATURE

Refinery Construction Materials Manual, EMRE Manual No. EETD 028
User's Manual for Hextran, General Purpose Program for the Design and Rating of Shell and Tube Heat Exchangers
Standards of Tubular Exchanger Manufacturer's Association (TEMA), Eighth Edition, 1999
 HTRI Computer Programs Support Volume Section F-3, *Flow Induced Vibration Analysis*
 HTRI Reports STV-1, *Tube Vibrations in Shell-and-Tube Heat Exchangers*, and STV-5, *Design of Shell-and-Tube Heat Exchangers to Avoid Flow-Induced Vibration Problems*

NUMBER OF SHELLS

The total number of shells necessary for an exchanger is frequently dictated by how far the outlet temperature of the hot fluid is cooled below the outlet temperature of the cold fluid. This is known as the "extent of the temperature cross." The cross as well as other variables discussed in **Section IX-B**, determine the value of F_n , the temperature correction factor; this factor should always be equal to or greater than 0.800. (The value of F_n drops slowly between 1.00 and 0.800, but then quickly approaches zero. A value of F_n below 0.800 cannot be predicted accurately from the usual information used in process designs.) In a one-shell exchanger, F_n is at least 0.800 when there is no temperature cross. Increasing the number of shells in series increases the value of F_n , thus permitting a larger temperature cross. The total number of shells also depends on the total surface required, since the size of the individual exchanger is usually limited by handling considerations. If there are no local restrictions due to capacity of tube bundle handling or cleaning equipment, the shell is usually limited to a shell inside diameter of 60 in. (1524 mm). (Note that this limitation does not necessarily apply to fixed tubesheet exchangers in clean service and for kettle type exchangers it applies to the bundle diameter.)

In special cases, such as reboilers and fixed tubesheet exchangers, very large areas per shell are occasionally used. Areas up to 25,000 ft² (2300 m²), and shell inside diameters up to 100 in. (2540 mm), have been used in fixed tubesheet exchangers. For these cases the individual refineries should be consulted to see if they are equipped to handle oversized tube bundles and shells.

TUBE SELECTION

TYPE

Exchanger tubes are commonly available with either a bare or finned outside surface. Selection of the type of surface is based on the application, availability and economics.

The conventional shell and tube exchanger is supplied with bare tubes. These are readily available in any material used in exchanger manufacture and in a wide range of wall thicknesses.

With integral fin tubes, the fins increase the outside area to approximately 2-1/2 times that of a smooth tube. Low-fin tubes should be considered, after an economic evaluation, for services with shellside fouling resistance of 0.0030 hr-ft²-°F/Btu (0.00053 m²-°C/W) or less, unless local field experience precludes their use. Low fins should never be used where the corrosion rate exceeds 2 mils per year, since the fin life will be 3 years or less. Since finned tubes cost 50 - 70% more than bare tubes for the same length and nominal wall thickness, a ratio of total outside resistance (heat transfer film and fouling) to total inside resistance of three or greater, based on bare tubes, is required to justify finned tubes. This ratio occurs frequently in steam-heated reboilers and preheaters, water coolers and condensers handling organic fluids.

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TUBE SELECTION (Cont)

Finned tubes may be the economic choice for ratios lower than three if their use requires fewer shells than would be needed for plain tubes. Finned tubes are also particularly useful for debottlenecking, expansions utilizing existing shells and cases where minimum bundle diameter rather than cost is of primary interest. For discussion on the enhancement of heat transfer of both smooth and fin tubes see [Section IX-A](#); for an estimation of tube count, see [Table 5](#) of this section.

Methods for calculating the heat transfer coefficient and pressure drop for low-fin tubes are given in [Section IX-G](#).

LENGTH

The selection of tube length is affected by availability, site preference, and economics. Tube lengths up to 24 ft (7.3 m) are readily obtainable worldwide. Longer tubes [up to 80 ft (24.4 m) for carbon steel and 70 ft (21.3 m) for copper alloys] are available in the United States. However, either 16 ft (4.9 m) or 20 ft (6.1 m) tube length is a common refinery preference. Refer to the refinery for preferable tube lengths. Keep in mind that quite often exchangers are fabricated in one location for use in another location. Some contractors and owners use standard metric lengths, e.g., 6 m instead of 20 ft (6.1 m).

The exchanger cost depends upon the tube length: the longer the tube, the smaller the bundle diameter for the same area. The savings result from a decrease in the cost of shell flanges and tubesheets with only a nominal increase in the cost of the longer shell. In the common range of tube lengths, there is no cost penalty for the longer tubes since length extras are added for steel only over 24 ft (7.3 m) and for copper alloys over 30 ft (9.1 m).

A disadvantage of longer tubes in units (such as condensers) located in a structure is the increased cost of the platforms and additional structure required. Longer tube bundles also require a greater pulling clearance, thereby possibly increasing the plot area requirements.

DIAMETER AND WALL THICKNESS*

Exchanger tubing is supplied on the basis of a nominal outside diameter and either a minimum or average wall thickness. For exchanger tubing, the outside tube diameter is fixed. The inside diameter varies with the nominal wall thickness and wall thickness tolerance. Minimum wall tubing has only a plus tolerance on the wall thickness, resulting in the nominal wall thickness being the minimum thickness. Since average wall tubing has a plus-or-minus tolerance, the actual wall thickness can be greater or less than the nominal thickness. The allowable tolerances vary as a function of the tube material, diameter and fabrication method.

For heat exchangers, ExxonMobil usually specifies minimum wall tubing, i.e., the wall thickness at any point must not be thinner than the thickness specified on the exchanger specification sheet. The effect of wall thickness on pressure drop is discussed under **PRESSURE DROP** in this section.

The following tube diameters and wall thicknesses are preferred in exchangers fabricated by rolling tubes into the tubesheets for the service indicated:

1. **Water Service** - Nonferrous tubes 3/4 in. OD, 0.065 in. wall thickness; Ferrous tubes 1 in. OD, 0.083 or 0.109 in. wall thickness.

* Exchanger tube data are listed in [Table 1](#). Thermal conductivities of tube materials are listed in [Table 2](#).



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TUBE SELECTION (Cont)

2. **Other Services** - Tube diameters, wall thicknesses, layout, and spacing are provided in the following table for various services placed on the tubeside or shellside:

SEVERITY OF SERVICE (FOULING FACTORS; hr-ft ² -°F/Btu [m ² -°C/W])	(ON TUBESIDE) OD, in. (mm)	(ON SHELLSIDE) LAYOUT AND SPACING, in. (mm)	BWG	MINIMUM WALL THICKNESS ⁽¹⁾ , in. (mm)
A. Ferrous Tubes for Oil Service				
Non-fouling or fouling < 0.003 (0.00053), mildly corrosive	3/4 (19.05)	15/16 (23.81) Δ	14	0.083 (2.11)
Non-fouling or fouling < 0.003 (0.00053), corrosive	3/4 (19.05)	1.00 (25.4) Δ or $\square$	12	0.109 (2.77)
Extremely fouling $\geq$ 0.003 (0.00053), mildly corrosive	1 (25.4)	1.25 (31.75) $\square$	12	0.109 (2.77)
Extremely fouling $\geq$ 0.003 (0.00053), corrosive	1 (25.4)	1.25 (31.75) $\square$	10	0.134 (3.4)
B. Alloy Tubes for General Service				
Water service fouling < 0.003 (0.00053)	3/4 (19.05)	15/16 (23.81) Δ	16	0.065 (1.65)
Non-fouling or fouling < 0.003 (0.00053)	3/4 (19.05)	15/16(23.81) or 1.0(25.4) Δ or $\square$	16	0.065 (1.65)
Extremely fouling $\geq$ 0.003 (0.00053)	1 (25.4)	1.25 (31.75) $\square$	14	0.083 (2.11)

Note:

- (1) When low-finned tubes are specified, the outside diameter of the unfinned portion and the wall thickness at the finned section are required. Although the thickness of the tube in the finned portion is nominally a standard wall thickness, manufacturing tolerances can result in a somewhat thinner wall. It is this minimum wall (given on [Table 1](#) of [Section IX-G](#)) that is specified.

Tubing may be supplied on either a minimum or average wall basis. However, the tabulated wall thickness is the minimum acceptable wall thickness.

For special materials (Titanium, High Nickel, etc.) thinner tube walls may be specified.

See the Refinery Construction Materials Manual for minimum thickness as a function of corrosion rate and service.

Refineries may request exceptions to this list on the basis of duplication of existing units or extended periods between exchanger cleaning.

FERRULES

Ferrules are short sleeves inserted into the inlet end of a tube. They are used to prevent erosion of the tube ends due to the inlet turbulence when erosive fluids, such as streams containing solids, are handled. When there is evidence that the tubes will be subject to erosion by solids in the tube side fluid, ferrules should be specified. Ferrule material, length and wall thickness should be given. Also, ferrules are occasionally used in cooling water service to prevent oxygen attack at the tube ends. The ferrules should be secured in place by roller expanding into the tubes. However, ferrules should not be provided if back-flushing is to be used. Assistance in selecting the proper ferrule can be obtained from the Materials Engineering Section of EMRE.

TEMA TYPES

TEMA stands for the Tubular Exchanger Manufacturer's Association, which develops and publishes the basic industrial standards for shell-and-tube heat exchangers. The standards cover heavy-duty heat exchangers (TEMA R) as well as lighter duty heat exchangers (TEMA C and TEMA B). ExxonMobil refineries typically use only the TEMA "R" heat exchangers due to the generally severe requirements of petroleum applications; however, more moderate process services, such as those encountered in Chemical Plants, may warrant consideration of TEMA B or C construction.

Each TEMA heat exchanger is designated by letters representing three major parts: the front end stationary head (commonly referred to as "channel"), the shell, and the rear end head. Each part can be designed in several modifications, commonly referred to as "types." The latest (8th) edition of the TEMA standards designates five types of channels (A, B, C or N, and D); seven types of shells (E, F, G, H, J, K and X); and eight rear end head types (L, M, N, P, S, T, U and W), refer to [Figure 2](#). The rear end head type determines whether or not the tube bundle is removable from the shell. In any ExxonMobil specification, therefore, the TEMA type heat exchanger is specified by the three single letters which refer to the type of channel, shell and rear end head, respectively; for example, "TEMA type AET."

The recommended Heat Exchanger Specification Sheet is shown in [Table 6](#).

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TEMA TYPES (Cont)

CHANNELS

The front end stationary head of shell and tube exchangers is commonly referred to as the channel. The common TEMA channel types and their applications are as follows:

Type A - Features a removable channel with a removable cover plate. It is used with fixed tubesheet, U-tube and removable bundle exchanger designs. This is the most common stationary head type.

Type B - Features a removable channel with an integral cover. It is used with fixed tubesheet, U-tube and removable bundle exchanger designs. This type is normally used only when the tube side fouling factor is less than 0.0020 hr-ft²-°F/Btu (0.00035 m²-°C/W) or when flange leaks represent a greater concern than easy access for cleaning and inspection. Note that with this channel, the mechanical cleaning of the bundle requires disassembly of the piping spool piece or elbow.

Type C and Type N - Two types of channels that are integral with the tubesheet. Type C is attached to the shell by a flanged joint and is used for U-tube and removable bundles. The N type has a tubesheet that is integral with the shell and is used with fixed tubesheet designs. Note that neither the C nor the N type is recommended in normal application because of extremely high maintenance costs for: a) assembling/testing of the heat exchanger, b) tubesheet repair, and c) retubing of the bundle.

Type D - This is a special high-pressure head used when the tube side design pressure exceeds approximately 1000 psi (6900 kPa gage). The channel and tubesheet are integral forged construction. The channel cover is attached by special high-pressure bolting.

TUBE BUNDLES

TYPES

Fixed Tubesheet Design (TEMA Rear End Head Types: L, M or N) - Fixed tubesheet exchangers have both tubesheets attached directly to the shell and are usually among the most economical exchangers for low design pressures. This type of exchanger construction should be considered when no shell side cleaning or inspection is required, or when in-place shell side chemical cleaning is available and applicable. Differential thermal expansion between tubes and shell limits applicability to moderate temperature differences. [An expansion joint may be required when there is a difference of more than 50°F (28°C) between average tube metal temperature and average shell metal temperature.] All start-up, operating, and upset conditions as well as normal operating conditions must be examined in determining the maximum temperature difference. For services that would require an expansion joint, a U-tube bundle may be more economical.

Shell and tubesheet material specifications incompatible for welding to each other (i.e., steel to aluminum or high copper alloys) might preclude fixed tubesheet construction, necessitate a change in metallurgy, or require a clad tubesheet.

U-tube Design (TEMA Rear End Head Type U) - U-tube exchangers represent the greatest simplicity of design, requiring only one tubesheet and no expansion joint or costly floating head, while permitting individual tube differential thermal expansion. U-tube exchangers are the least expensive units for high tube side design pressures. The tube bundle can be removed from the shell, but replacement of individual tubes (except for those on the outside of the bundle) is impractical.

External surfaces of the tube bundle are mechanically cleanable. When the tube side fouling factor exceeds 0.0010 hr-ft²-°F/Btu (0.00017 m²-°C/W), chemical cleaning is preferable, although mechanical cleaning with flexible end-drill shafts is possible. U-tube construction is not used when the tube side fouling factor exceeds 0.0020 hr-ft²-°F/Btu (0.00035 m²-°C/W).

Although the bend portion of the U-tube bundle provides heat transfer surface, it is relatively ineffective for sensible heat transfer compared to the straight tube length surface area. Therefore, when the effective surface area for U-tube bundles is calculated, only the surface area of the straight portions of the tubes is included, except for isothermal boiling or condensing on the shell side.

Pull-through Floating Head Design (TEMA Rear End Head Type T) - A pull-through floating head exchanger has stationary tubesheet at the channel end and a floating tubesheet with a separate floating head cover at the rear end. The bundle can be easily removed from the shell by disassembling only the channel. The floating head flange and bolt design require a relatively large clearance between the bundle and shell, particularly as the design pressures increase. Because of this clearance, the pull-through bundle has fewer tubes per given shell size than other types of construction. The bundle to shell clearance, which decreases the shell side heat transfer capability, should be blocked by sealing strips or dummy tubes to reduce shell side fluid bypassing. Mechanical cleaning of both the shell and tube sides is possible.

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TUBE BUNDLES (Cont)

Split-ring Floating Head Design (TEMA Rear End Head Type S) - A split-ring floating head exchanger has a stationary tubesheet at the channel end and a floating tubesheet that is sandwiched between a split-ring and a separate floating head cover. The floating head assembly moves inside a shell cover of larger diameter than that of the shell. To remove the bundle from the shell, the front end head, rear shell cover, and floating tubesheet cover/split ring assembly must be dismantled. This requirement is the greatest disadvantage of the split-ring design since it requires considerably more time to pull the bundle. Mechanical cleaning of both the shell and tube side is possible.

Outside Packed Floating Head (TEMA Rear Head Type P) and Externally Sealed Floating Tubesheet (TEMA Rear Head Type W) Designs - There are two variations of outside packed floating head designs: the lantern ring type (W) and the stuffing box type (P). In the lantern ring design, the floating head slides against a lantern ring packing which is compressed between the shell flange and the shell cover flange. The stuffing box design is similar to the lantern ring type, except that the seal is against an extension of the floating tubesheet, and the tubesheet cover is attached to the tubesheet extension by means of a split-ring. Neither of these designs is normally specified by ExxonMobil, since the packing has a tendency to leak, allowing the streams to leak to the atmosphere. These designs should never be used for hydrocarbons or toxic fluids. Mechanical cleaning of both the shell and tubeside is possible.

See **Table 3** for guidance in TEMA head type selection.

TUBE LAYOUT

There are four types of tube layouts with respect to the shell side cross flow direction between baffle tips: square (90°), rotated square (45°), triangular (30°) and rotated triangular (60°). See **Figure 3** for illustrations. The 60° triangular layout is seldom used because its heat transfer characteristics are poor compared to the pressure drop expenditure.

Use of the triangular layout (30°) is preferred for all exchangers (except reboilers with a heat flux exceeding 6000 Btu/hr · ft² [19,000 W/m²]) with the shell side fouling factors of 0.0020 hr·ft²·°F/Btu (0.00035 m²·°C/W) or less, provided the deposit is subject to chemical cleaning. An exchanger with triangular layout costs less per sq ft and transfers more heat per sq ft than one with a square or rotated square layout. For this reason, triangular layout is preferred where applicable.

Square or rotated square tube layout should be specified for all shell and tube exchangers when the shell side fouling factor is over 0.0020 hr·ft²·°F/Btu (0.00035 m²·°C/W), whenever mechanical cleaning of the outside of the tubes is required, or when requested by the refinery. Square or rotated square tube layout is also required for reboilers with a heat flux exceeding 6000 Btu/hr·ft² (19,000 W/m²). Rotated square layouts are preferable for laminar flow because of a higher heat transfer coefficient caused by induced turbulence. In turbulent flow, especially for pressure-drop-limited cases, square layout is preferred because the heat transfer coefficient is equivalent to that for rotated square layout while the pressure drop is somewhat less.

Tube layout for removable bundles may be either square (90°), rotated square (45°) or triangular (30°). Nonremovable bundles (fixed tubesheet exchangers) are always triangular (30°) layout.

TUBE PITCH

The tube pitch (PT) is the center-to-center distance between adjacent tubes (**Figure 3**). The common pitches used are:

TUBE SIZE	TRIANGULAR	SQUARE	HEAVIEST WALL
1. 3/4 in. OD tubes (19.05 mm)	15/16 in. (23.81 mm)	—	0.095 in. (2.41 mm)
2. 3/4 in. OD tubes (19.05 mm)	1 in. (25.4 mm)	1 in. (25.44 mm)	0.109 in. (2.77 mm)
3. 1 in. OD tubes (25.4 mm)	1-1/4 in. (31.75 mm)	1-1/4 in. (31.75 mm)	0.134 in. (3.40 mm)
4. 1-1/2 in. OD tubes (38.1 mm)	1-7/8 in. (47.63 mm)	1-7/8 in. (47.63 mm)	0.165 in. (4.19 mm)
5. 1-1/4 in. OD tubes (31.75 mm)	1-9/16 in. (39.7 mm)	1-9/16 in. (39.7 mm)	0.165 in. (4.19 mm)
6. For tube sizes larger than 1-1/2 in. (38.1 mm), use 1.25 times the outside diameter.			
7. In kettle-type reboilers where the pressure is less than 50 psig (345 kPa - gage) and the heat flux is greater than 10,000 Btu/hr·ft ² (31,500 W/m ²), use 3/8 in. (9.5 mm) spacing between tubes.			

With 3/4 in. (19.05 mm) OD tubes on triangular layout, 15/16 in. (23.81 mm) pitch should be used unless the pitch is limited by the wall thickness required, since this results in a smaller unit due to the higher coefficient and closer spacing. The column "Heaviest Recommended Wall" is based on the maximum allowable tubesheet distortion resulting from rolling the indicated tube into a tubesheet having the minimum permissible ligament width at the listed pitch. The ligament is that portion of the tubesheet between two adjacent tube holes.

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TUBE BUNDLES (Cont)

CROSS BAFFLES

Cross baffles support the tubes, restrain tube vibration and direct fluid flow on the shell side. Three types of baffles are generally used: segmental, double segmental, and rod baffles. The first two baffle types are illustrated in **Figure 4**. The rod baffle concept is discussed and illustrated in **Section IX-A**. In addition, full circle support baffles are occasionally used for tube support in kettle reboilers.

Baffle cut defines the segment of the baffle "cut" away to provide for fluid flow past the chord of the baffle. For segmental baffles, this is expressed as the ratio of the segment height to shell diameter in percent. Segmental baffle cuts are usually about 25%, although the maximum practical cut for tube support is approximately 45%. Test work (single phase) by Heat Transfer Research, Inc. (HTRI), on segmental baffles indicates that optimum baffle cuts range from 15 - 30% with a slight peak at 25%.

Double segmental baffle cut is expressed as the ratio of total window area to exchanger cross-sectional area in percent. Normally the window areas for the single central baffle and the area of the central hole in the double baffle are equal and are 40% of the exchanger cross-sectional area. This allows a baffle overlap of approximately 10% of the exchanger cross-sectional area on each side of the exchanger. However, there must be enough overlap so that at least two rows of tubes are supported by adjacent segments.

Baffle pitch is the longitudinal spacing between baffles. The maximum baffle pitch is a function of tube size and metallurgy and, for no change of phase flow, of shell diameter. See **Table 4** in **Section IX-D** which gives maximum allowable values as a function of tube diameter. If there is no change of phase in the shell side fluid, the baffle pitch should not exceed the shell inside diameter. Otherwise, the fluid would tend to flow parallel with the tubes, rather than perpendicular to them, resulting in a poorer heat transfer coefficient. With shell side condensation or vaporization, the maximum baffle spacing is a function of tube diameter only, as indicated in **Table 4** of **Section IX-D**. Also, refer to **TEMA Table RCB-4.52** for maximum unsupported straight tube spans.

The minimum allowable baffle pitch is 20% of the shell inside diameter or 2 in. (50 mm), whichever is greater. A very tight baffle pitch tends to force the shell side fluid into leakage and bypass streams. This decreases the effective cross flow stream, resulting in a decreased heat transfer coefficient.

Position of Baffle Chord

1. Square and Rotated Square Tube Layout

- a. The chord should be vertical (usually with a large baffle cut) in condensers, vaporizers and units handling fluids containing suspended solids. This arrangement minimizes "pockets" which would trap vapors or sediment, reducing the effective surface area.
- b. The baffle chord should be horizontal when sediment-free, single phase shell side fluid is being cooled or heated through a wide temperature range [200 to 300°F (110 to 165°C)] in one shell. This avoids fluid stratification.
- c. The baffle chord can also be placed at a 45° angle to grade. This position is termed "on the bias". When sedimentation and stratification possibilities co-exist, a 45° chord at a suitable baffle cut provides a reasonable compromise solution to both problems. A 45° chord is also useful to prevent the accumulation of heavy components in an exchanger when a wide boiling range fluid is being partially vaporized. With horizontal pass partitions, the 45° chord mitigates the effect on heat transfer of fluid flow through the pass lanes.

2. 30° Triangular Tube Layout

- a. When a sediment-free shell fluid is being cooled through a wide temperature range [200 - 300°F (110- 165°C)] in one shell, the chord of the baffle should be horizontal, to avoid fluid stratification (see **Par. 1. b.** above).
- b. The chord of the baffle should be vertical for all other services (see **Par. 1. a.** above).

SEAL STRIPS

Seal strips are flat strips of metal (or tie rods) that extend the length of the shell in order to prevent the shell side fluid from flowing through the clearance between the tube bundle and the inner wall of the shell. The seal strips are located between the baffle chords of adjacent baffles, in the clearance between the tube bundles and the shell. **Figure 6** illustrates a typical seal strip installation. Note that seal strips are normally installed in pairs, a strip on each side of the bundle.

Seal strips are normally installed on pull-through and split ring bundles because the clearance between the tube bundle and the shell is quite large [2 to 5 in. (50 to 125 mm)]. They are also installed on other types of tube bundles when the clearance between the outer tubes and the shell exceeds one half the tube pitch. Seal strips are not used in kettle reboilers or in units with isothermal condensation occurring on the shell side. Refer to **GP 6-1-1** for specific requirements concerning seal strip installation.

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TUBE BUNDLES (Cont)

IMPINGEMENT PROTECTION

Impingement plate or other protection devices are normally installed on the top row of tubes under the shell inlet nozzle or in shell side inlet nozzles to protect the bundle against impingement by the incoming fluid when the fluid: (a) is condensing, (b) is a liquid vapor mixture, (c) contains abrasive material, or (d) is entering at high velocity. In addition, TEMA requires bundle impingement protection when nozzle values of pV^2 exceed:

1. 1500 lb/ft-sec² (2230 kg/m-s²) for noncorrosive, nonabrasive, single phase fluids.
2. 500 lb/ft-sec² (744 kg/m-s²) for all other liquids, including liquid at its boiling point.

The minimum bundle entrance area should equal or exceed the inlet nozzle area and should not produce a value of pV^2 greater than 4000 lb/ft-sec² (5950 kg/m-s²) per TEMA. An impingement baffle can be a flat plate, a curved plate or an angle bar; angle bars must be installed at each tube individually. The use of perforated impingement baffles is prohibited by **GP 6-1.1**. In order to maintain a maximum tube count, the impingement plate is sometimes located in a conical nozzle opening or in a dome cap above the shell. The impingement protection device material should be of at least the quality of the tube material.

INTERCHANGEABILITY OF TUBE BUNDLES

There is an incentive to standardize tube bundle design within a project, to reduce the refinery's expensive inventory of spare bundles and other exchanger parts. Although desirable, standardization from project to project is less subject to control since the exchanger fabrication may not be by the same manufacturer. It is not considered desirable to overdesign a heat exchanger solely for bundle interchangeability, unless the required surfaces are nearly equal (within approximately 5%) and the design pressures and temperatures are similar.

Tube bundles must be of the same general design and must have identical tubesheets to be physically interchangeable, including the same grooving for pass partitions even though the two units are designed for a different number of passes.

SHELL TYPES

SINGLE-PASS SHELL (TEMA E)

The single-pass shell is the most common shell construction for shell and tube heat exchangers. The shell side inlet and outlet nozzles are located at opposite ends of the shell. The nozzles can be placed on opposite or same sides of the shell, depending on the number and type of baffles used. A typical one shell-pass exchanger with horizontal cut segmental baffles is illustrated in **Figure 5A**.

TWO-PASS SHELL (TEMA F)

A two-pass shell requires the use of a longitudinal baffle to direct the shell side flow. An exchanger with two shell passes is illustrated in **Figure 5C**. Note that both the shell inlet and outlet nozzles are adjacent to the stationary tubesheet. To avoid excessively thick longitudinal baffles, two-pass shells should not be used with a shell side pressure drop greater than 10 psi (70 kPa) [5 psi (35 kPa) is preferred]. Also, a shell side temperature range exceeding 350°F (195°C) should be avoided since greater temperature ranges result in excessive heat leakage through the baffle, as well as thermal stresses in the baffle, shell and tubesheet.

The longitudinal baffle can be of either welded or removable design. Since there are severe design difficulties and cost penalties associated with the use of welded baffles in floating head exchangers, this type of design should be used only with fixed tubesheet units that do not require expansion joints. A welded baffle may be used in a four-pass U-tube design while retaining the ability to remove the bundle from the exchanger. Any welded baffle is limited to a minimum shell ID of 16 in. (406.4 mm) due to welding feasibility. If an ID of less than 16 in. is desired, a double pipe design could be considered. If a longitudinal baffle is to be used in a floating head exchanger, it should be a removable design. Removable longitudinal baffles require the use of flexible, light gage sealing strips or a packing device between the baffle and the shell, to reduce fluid leakage from one side to the other.

Care must be taken during maintenance operations in removal or insertion of bundles with removable longitudinal baffles to avoid damaging the sealing device. Damaged seals will result in poor heat transfer performance and it is recommended that they be replaced every time the bundle is removed. The shell ID must be ground smooth during fabrication to assure a good seal between the sealing device and the shell.

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SHELL TYPES (Cont)

Recent developments in large diameter multi-tube hairpin exchangers have turned hairpins into viable alternatives to F-shells and should be considered where applicable.

A two-pass unit may be used when the LMTD correction factor F_n is less than 0.8 for a one-shell pass unit. A two-shell pass unit with a welded longitudinal baffle is satisfactory if F_n for two-shell passes is equal to or greater than 0.85, and a unit with a removable baffle is satisfactory if F_n for two-shell passes is 0.90 or greater. These limitations on F_n are the result of heat loss through the longitudinal baffle and, in the case of a removable baffle, fluid leakage around it.

DIVIDED FLOW SHELL (TEMA J)

A divided flow shell has a central inlet nozzle and two outlet nozzles, or vice versa. A divided flow exchanger is illustrated in **Figure 5B**. Typically, this type of shell is used to reduce pressure drop and to avoid vibration damage in a condensing service. The following list shows exchanger types ranked in order of decreasing pressure drop:

1. E shell with segmental baffles.
2. E shell with double segmental baffles.
3. J shell with segmental baffles.
4. J shell with double segmental baffles.
5. E shells in parallel with segmental baffles.
6. E shells in parallel with double segmental baffles.
7. J shells in parallel with segmental baffles.
8. J shells in parallel with double segmental baffles.

For most designs, double segmental baffles are used with J shells.

Double segmental baffles in a divided flow exchanger normally have a vertical cut since J shells are most often in condensing or vaporizing service. A full circle support baffle is usually provided at the center nozzle. The baffle arrangement requires that there be an odd total number of baffles, as well as an odd number of baffles in each end of the shell. The baffles on each side of the central baffle and the last baffle at the ends of the shell have solid centers with cut-away edges to optimize flow through regions of the tube bundle at the shell nozzles.

KETTLE (TEMA K)

This type is used for vaporizing services (reboilers, steam generators and refrigeration services). A dome area is provided to separate the vapor from the entrained liquid. Process reboilers are provided with a weir plate to maintain a constant liquid level 1 in. (25mm) to 2 in. (50mm) above the top row of tubes.

CROSSFLOW SHELL (TEMA X)

This type is used when low pressure drop is required or when the shell nozzles are unusually large and is a possible solution to vibration problems since it can be designed for low cross flow velocity.

SPLIT FLOW (TEMA G) AND DOUBLE SPLIT FLOW (TEMA H)

These types are normally used for thermosyphon reboilers. G type shells are typically used for tube lengths up to 10 ft (3.05 m) and H type shells are typically used for tube lengths between 10 ft (3.05 m) and 20 ft (6.1m).

TUBE SIDE FLOW

Whichever fluid appears higher on the following list will ordinarily be passed through the tubes:

1. Cooling water.
2. Corrosive fluid (especially those requiring higher alloy materials) or a fluid likely to deposit coke, sediment or other solids.
3. A fluid which is fouling.
4. The less viscous of the two fluids.
5. The fluid under higher pressure [for extremely high pressures, 1000 psi (6900 kPa gage), it may be economical to treat this item as Number 1 on this list].
6. The hotter fluid.
7. The fluid of less volume.

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TUBE SIDE FLOW (Cont)

Several exceptions to this list are:

1. Condensing vapors are normally passed through the shell.
2. Condensing steam is normally passed through the tubes.
3. If the temperature change of one fluid is very large [greater than approximately 300 to 350°F (165 to 195°C)], that fluid is usually passed through the shell, rather than the tubes, if more than one tube pass is to be used. This minimizes the construction problems caused by thermal expansion. Also, to avoid thermal stress problems, fluids with greater than 350°F (195°C) temperature change cannot be passed through the shell side of a two-pass shell (**GP 6-1-1**).
4. If one of the fluids is clean [fouling factor 0.0010 hr-ft²-°F/Btu (0.00018 m²-°C/W) or less] and is somewhat corrosive (see the *Refinery Construction Materials Manual*), this fluid is passed through the tubes and U-tube construction is used since it is typically more economical to alloy up the tubeside versus the shellside to easily handle the corrosive fluid.

PRESSURE DROP

The calculation of pressure drop is covered in greater detail under *Calculation of Surface*, **Sections IX-D, IX-E and IX-F**. General considerations applicable to shell and tube exchangers are given in the following paragraphs.

CORRECTION FOR REDUCED FLOW AREA

Tube Side - GP 6-1-1 requires that exchanger vendors provide tubes having no point of the wall thinner than the specified thickness. For this reason, actual inside diameters are always smaller than standard diameters. This deviation from the standard diameter is greater for carbon steel than for copper alloy tubes. The tube side pressure drop equations used in this manual include a factor that corrects for the increased wall thickness and for fouling on the inside of the tubes. In other words, tube side pressure drop calculations are based on minimum wall tubes with a 10 to 20% increase (depending on tube inside diameter) for fouling deposits. **Table 4** of **Section IX-D** shows the values for this factor. An additional method to correct for reduced flow area using improved correlations can be found on **Figure 4** and **4A** in **Section IX-B**.

Low-fin tubes require special consideration. The inside diameters listed in **Table 1** of **Section IX-G** must be used for pressure drop and heat transfer calculations.

Shell Side - The equation for shell pressure drop (**Section IX-D**) includes a factor that accounts for fouling of the outside of the tubes. This fouling reduces the free flow area, thereby increasing the pressure drop. Typical values of the factor are given in **Table 4** of **Section IX-D**.

Note that fouling affects both pressure drop and heat transfer. The corrections discussed here address the effect on pressure drop only. The effect of fouling on heat transfer is discussed in **Section IX-B**.

ECONOMICS CONSIDERATION OF PRESSURE DROP

The design pressure drop is usually determined by an economic balance between: (1) the increased cost of pumping and of certain exchanger components, and (2) the decreased surface area. As the number of tube passes is increased or the baffle pitch is decreased, the fluid velocity becomes greater, thereby increasing the pressure drop and the convective heat transfer coefficient. Increased velocity also decreases the rate of fouling.

For pressured streams, the pressure drop should be maximized. Where there are no material or process restrictions on maximum velocity, a reasonable limitation for liquids is about 10 to 15 ft/sec (3 to 5 m/sec). For gases and vapors, velocities up to 100 ft/sec (30 m/sec) are common. Typical pressure drops are shown in **Table 4** in **Section IX-B**. Designs are typically based on pressure drop allowances set by experience, and many systems exist where pressure drop is dictated by process concerns.

For pumped streams, unless otherwise limited, the preliminary design pressure drop is that required to fully load the pump driver. The pressure drop should be finalized by checking the return on incremental investment by comparing the first case with that using the next larger size driver, loaded either fully or to the extent permitted by velocity limitations. Items to be considered in the comparison are a possible change in line rating or pump type.

Tube Side (Number of Passes) - For coolers and condensers using water, specify enough tube passes to maximize the utilization of the available pressure drop as limited by materials. The permissible maximum and minimum water velocities vary with the tube material and water type as shown in **Table 4** of **Section IX-D**. Note that for cooling water service care must be taken to avoid using excessive pressure drop in elevated exchangers due to the need to ensure that vacuum conditions do not occur.

For hydrocarbon services, specify sufficient passes to make the tube velocity high enough to prevent laminar or transitional flow (or as shown on **Figure 1.8** of **Section IX-D**, (Re) > 3700).

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PRESSURE DROP (Cont)

In some cases the exchanger terminal conditions will be such that the type of flow could change from laminar to turbulent (or vice versa) within the unit. This must be avoided, since prediction of heat transfer coefficients in the transition region is unreliable. For some services, such as heavy fuel oil, this may be overcome by fluxing with a suitable fluid of lower viscosity.

Ordinarily no fewer than two nor more than eight tube passes are used. With more than eight or fewer than two passes, the construction becomes complicated and fabrication costs tend to become higher. Note that two, four, or eight tube-pass arrangements are sometimes easily interchanged as long as the exchangers remain within the allowable pressure drops. However, in special services, one pass or more than eight passes can sometimes be justified.

See **Table 4** for the maximum number of tube passes, normally used for various shell diameters. Restrictions on tube pass arrangements for particular exchanger designs are as follows:

1. **Fixed Tubesheet Exchangers** - Any number of tube passes, odd or even, is possible. The most common arrangements are single pass or an even number of multiple passes.
2. **U-tube Exchangers** - Any even number of tube passes is possible, but normally the maximum recommended is six, because of construction considerations.
3. **Pull-through and Split-ring Floating Head Exchangers** - Any even number of tube passes is possible. Single-tube-pass designs require special expansion joints, and are not generally used.
4. **Lantern Ring (TEMA Type W) Externally Sealed Floating Tubesheet Exchangers** - Only single or double tube-pass arrangements are possible for this type of construction.
5. **Stuffing Box (TEMA Type P) Outside Packed Floating Head Exchangers** - Any even or odd number of tube passes is possible.

SHELL SIDE

1. **Number of Shell Passes** - The vast majority of exchangers are of one-pass shell (Type E) design, or variants thereof (Types J, K or X). Two-pass shells (Type F), or variants (Types G and H), are less frequently utilized. Multiple pass shells require a longitudinal baffle of either welded or removable design (see previous discussion of two-pass shells).

Where there are very large surface requirements, especially in services requiring multiple exchangers in series to meet F_n requirements, multiple shell-pass units employing longitudinal baffles may be the most economical choice. However, it is difficult to estimate accurately the cost of maintaining exchangers with longitudinal baffles. In addition, if the longitudinal baffle seal strips are damaged, the shellside bypass streams could result in large performance debits. Therefore, it is recommended that the exchanger be specified as a one-pass shell unit if possible. If specific project conditions require the features of a two-pass shell, multi-tube hairpin designs should also be considered.

2. **Cross Baffles** - Segmental baffles are normally specified, unless shell side pressure drop is excessive, even at maximum baffle spacing. In such cases, double segmental baffles should be used because they result in lower pressure drop with only a slight decrease in the shell side heat transfer coefficient.
3. **"No-Tubes-in-Window" Design** - This design is a bundle with segmental baffles that have no tubes in the "windows" left by the baffle cut. Each tube is supported by every baffle, whereas, in conventional bundles, the tubes in the window region are supported only by every other baffle. The "no-tube-in-window" bundle is used either to minimize pressure drop or to minimize vibration damage to tubes. Intermediate tube support plates may be inserted between NTIW baffles to further increase tube natural frequency, without impacting exchanger performance.
4. **Nozzles** - Use two shell outlet nozzles (one at each end of the shell) and a center inlet nozzle, or vice versa, if other means of reducing pressure drop are ineffective. This is termed a "divided flow" (TEMA J) shell.

For condensers or other exchangers handling vapors, an enlarged shell section at the inlet nozzle ("vapor belt") is sometimes used. Vapor belts are recommended if the inlet nozzle is large compared to the shell diameter. This design will prevent deleting tubes to provide the required flow area between the shell and the vapor impingement baffle and will provide inlet vapor distribution to the bundle.