ENGINEERING STANDARD

FOR

PROCESS REQUIREMENTS

OF

HEAT EXCHANGING EQUIPMENT

ORIGINAL EDITION

JULY 1997

This standard specification is reviewed and updated by the relevant technical committee on Dec. 2003. The approved modifications are included in the present issue of IPS.

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0. INTRODUCTION

"Process Design of Non-Combustion Type Heat Exchanging Equipment" are broad and contain variable subjects of paramount importance. Therefore a group of process engineering standard specifications are prepared to cover the subject.

This group includes the following Standards:

STANDARD CODE	STANDARD TITLE
IPS-E-PR-771	"Process Requirements of Heat Exchanging Equipment"
IPS-E-PR-775	"Process Design of Double Pipe Heat Exchangers"
IPS-E-PR-785	"Process Design of Air Cooled Heat Exchangers (Air Coolers)"
IPS-E-PR-790	"Process Design of Cooling Towers"

This Engineering Standard Specification covers:

"PROCESS REQUIREMENTS OF HEAT EXCHANGING EQUIPMENT"

Non-combustion type heat exchanging equipment consist of various types from which the above mentioned have the most application in Oil, Gas, and Petrochemical (OGP) Industries and each item will be discussed separately.

1. SCOPE

This Engineering Standard Specification covers the minimum process design requirements, for thermal design, field of application, selection of types and hydraulic calculations for shell and tube heat exchangers, and plate heat exchangers (plate fin exchangers).

This Engineering Standard Specification consists of two parts as described below:

Part I: Process Design of Shell and Tube Heat Exchangers.

Part II: Process Design of Plate Heat Exchangers (Plate Fin Exchangers).

For mechanical design requirements, reference is made to the relevant Fixed Mechanical Equipment Standards such as IPS-G-ME-220.

Note:

This standard specification is reviewed and updated by the relevant technical committee on Dec. 2003. The approved modifications by T.C. were sent to IPS users as amendment No. 1 by circular No. 215 on Dec. 2003. These modifications are included in the present issue of IPS.

2. REFERENCES

Throughout this Standard the following dated and undated standards/codes are referred to. These referenced documents shall, to the extent specified herein, form a part of this standard. For dated references, the edition cited applies. The applicability of changes in dated references that occur after the cited date shall be mutually agreed upon by the Company and the Vendor. For undated references, the latest edition of the referenced documents (including any supplements and amendments) applies.

IPS (IRANIAN PETROLEUM STANDARDS)

IPS-E-PR-170 "Process Flow Diagram"

IPS-E-PR-230 "Piping & Instrumentation Diagrams (P & IDs)"

IPS-G-ME-220 "Shell & Tube Heat Exchangers"

IPS-E-GN-100 “Engineering Standard for Units”

TEMA (TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION)

- "Standards of the Tubular Exchanger Manufacturers Association (TEMA)"
- TEMA Class R "Exchanger", TEMA Class C "Exchanger", TEMA Class B "Exchanger"

API (AMERICAN PETROLEUM INSTITUTE)

API Std. 660, 6th. Ed., Feb. 2001, Reaffirmed Dec. 1987 "Shell & Tube Heat Exchangers for General Refinery Services"

API Std. 662, “Plate Heat Exchanger for General Refinery Services”

2nd. Ed. 2002

HEI (HEAT EXCHANGE INSTITUTE INCORPORATION)

3. DEFINITIONS AND TERMINOLOGY

3.1 Definitions of Different Types of Heat Exchangers

Heat transfer equipment can be specified either by type of construction or by service. Generally, they are designated by service. The following terminology is in general use throughout the industry.

3.1.1 Chiller

A chiller cools a fluid with a refrigerant to a temperature below that obtainable using air or cooling water as the heat sink. Common refrigerants are propane, ethylene and propylene; chilled water or brines are less frequently used.

3.1.2 Condenser

A condenser is a unit in which a process vapor is totally or partially converted to liquid. The heat sink is ordinarily a utility, such as cooling water. The term "surface condenser" refers specifically to shell and tube units, used for the condensation of exhaust steam from steam turbines or engines. A "direct contact condenser" refers to a unit in which the vapor is condensed by direct contact heat exchange with droplets of water.

3.1.3 Cooler

A cooler exchanges heat between a process stream and water or air.

3.1.4 Evaporator

Exchangers specifically designed to concentrate water solutions by vaporizing some of the water.

3.1.5 Exchanger and/or heat exchanger

In the broad sense, an exchanger is any item of unfired heat transfer equipment whose function is to change the total enthalpy of a stream. In the specific (and more usual) connotation, an exchanger transfers heat between two process streams.

3.1.6 Fouling resistance

The fouling resistance is a measure of the ultimate additional resistance to heat transfer caused by deposits on and corrosion of the heat transfer material surface.

Note:

The resistance depends on the type of fluid, the material, temperature conditions, flow velocities and the operating period between two successive cleaning actions.

3.1.7 Fouling coefficient

The fouling coefficient is the reciprocal of the fouling resistance.

Note:

The use of the fouling coefficient has generally been abandoned, since it tends to be confusing that an increase in fouling results in a decrease in fouling coefficient.

3.1.8 Reboiler

A reboiler is a vaporizer that provides latent heat of vaporization to the bottom (generally) of a fractionation tower.There are two general classes of reboilers, those which send both phases to the tower for separation of vapor from liquid and those which return only vapor. The former operate by either natural circulation (usually called thermosyphon) or forced circulation.

3.1.8.1 Thermosyphon reboilers are by far the common type. Horizontal thermosyphons with vaporization on the shell side are commonly used in the petroleum industry while vertical units with in-tube vaporization are favored in the chemical industry. In a thermosyphon reboiler, sufficient liquid head is provided so that natural circulation of the boiling medium is maintained.

3.1.8.2 Forced circulation reboilers require a pump to force the boiling medium through the exchanger. This type of reboiler is infrequently used because of the added cost of pumping the reboiler feed, but may be required to overcome hydrostatic head limitations and/or circulation problems.

3.1.8.3 Reboilers which return only vapor to the tower are called kettle reboilers. The operation of kettle reboilers would be best described as pool boiling.

3.1.9 Steam generators (waste heat boilers)

Steam generators are a special type of vaporizer used to produce steam as the vapor product. Generally, the heat source is excess heat beyond that which is required for process; this accounts for the common name of "waste heat boiler" for these Units. Like reboilers, steam generators can be kettle, pump-through, or thermosyphon type.

3.1.10 Superheater

A superheater heats a vapor above its saturation temperature.

3.1.11 Vaporizor

A vaporizor is an exchanger which converts liquid into vapor. This term is sometimes limited to units handling liquids other than water.

3.2 Definition of TEMA Classes

3.2.1 TEMA "Class R" exchanger

The TEMA Mechanical Standards for "Class R" heat exchanger specify design and fabrication of unfired shell and tube heat exchangers for the generally severe requirements of petroleum and related process application.

3.2.2 TEMA "Class C" exchanger

The TEMA Mechanical Standards for "Class C" heat exchangers specify design and fabrication of unfired shell and tube heat exchangers for the generally moderate requirements of commercial and general process application.

"Class C" units are designed for maximum economy and result in a cost saving of about 5% over "Class R".

3.2.3 TEMA "Class B" exchanger

The TEMA Mechanical Standards for "Class B" heat exchangers specify design & fabrication of unfired shell & tube heat exchangers for chemical process service.

4. SYMBOLS & ABBREVIATIONS

A	Total exchanger area, in (m^2).
A_i	Required effective inside transfer surface, in (m^2).
API	American Petroleum Institute.
Btu	British Thermal Unit.
CAF	Compressed Asbestos Fiber.
DEA	di-Ethanolamine.
DGA	di-Glycolamine.
DN	Diameter Nominal, in (mm).
DP	Design Pressure.
EPDM	Ethylene Propylene Dien Monomer.
FPM	Fine Particular Matter.
h_f	Film coefficient of tube side fluid, in ($W/m^2.°C$) or ($W/m^2.K$)
HEI	Heat Exchanger Institute Incorporation.
ID	Inside Diameter, in (mm).
LMTD	Logarithmic Mean Temperature Difference.
MAWP	Maximal Allowable Working Pressure.
max.	Maximum.
MEA	mono-Ethanolamine.
min.	Minimum.
MOP	Maximum Operating Pressure.
MOT	Maximum Operating Temperature.
OD	Outside Diameter, in (mm).
OGP	Oil, Gas and Petrochemical.
OP	Operating Pressure.
PHE	Plate Heat Exchanger.
PSV	Pressure Safety Valve.
r_i	Fouling resistance on inside surface of tubes, in ($m^2.°C/W$).
r_o	Fouling resistance on outside surface of tubes, in ($m^2.°C/W$).
RCB	Resin Cured Butyl.
RGP	Recommended Good Practice.
SS	Stainless Steel.
TEMA	Tubular Exchanger Manufacturers Association.
V	Lineal Velocity of the fluid, in meter per second (m/s).
$U\theta$	The overall heat transfer coefficient, in ($W/m^2.°C$) or ($W/m^2.K$)

WC Water Column, in (mm).

ρ **(rho)** Density, in (kg/m^3).

5. UNITS

This Standard is based on International System of Units (SI) as specified in IPS-E-GN-100.

PART I

PROCESS DESIGN OF SHELL AND TUBE HEAT EXCHANGERS

6. SHELL AND TUBE HEAT EXCHANGERS

6.1 General Considerations

6.1.1 The shell and tube type exchanger is commonly used in general petroleum processes. It is inexpensive, easy to clean, available in many sizes, and can be designed for moderate to high pressures at reasonable cost. It consists of a bundle of tubes encased in a shell.

6.1.2 Tubular units in general should have removable tube bundles and should be of the floating head type with removable shell covers.

Typical exceptions are:

a) Fixed tube sheet exchangers such as refrigeration condensers and vacuum condensers.

In this type of construction, differential expansion of the shell and tubes due to different operating metal temperatures may require the use of an expansion joint or a packed joint.

b) "U" tube type for reboilers using steam in the tube and for exchangers on hydrogen service. Removable shell covers are not required for this type.

Fluids having a fouling factor above - 0.00035m².K/W(0.002 hr.ft².°F/Btu) should be routed on the shell side of U-tube exchangers and on the tube side for floating head type exchangers. In all cases U-tubes should be located in the horizontal plane.

For chemical works, fixed head exchangers should be used when the shell side fluid is non-fouling. Where the shell side fluid is fouling. U-tubes or floating head type bundles should be used and floating head type tube bundles when both sides are fouling.

Floating head type tube bundles are to be avoided for kettle type reboilers and chillers unless agreed by the Company.

6.1.3 There are two variations of floating tube sheet units, the pull-through and the non-pull-through. In the pullthrough unit, the entire floating tube sheet and cover assembly may be drawn through the shell without disassembly. In the non-pull-through unit, the shell cover and the floating tube sheet cover must be removed before the bundle can be taken out of the shell.

This requirement is the greatest disadvantage of the non-pull-through unit. However, due to the smaller diameter tube sheet which is possible if a split ring assembly is used to fasten the floating head cover, the non-pull-through unit requires a smaller shell for the same surface.

6.1.4 Listing the above variations in shell and tube units in order of increasing cost would give the following tabulation:

1) Simple fixed tube sheet unit.

2) U-tube unit.

3) Fixed tube sheet unit with an expansion joint or packed joint.

4) Floating tube sheet unit (pull-through and non-pull-through).

Shell and tube type exchangers are usually fabricated to conform to "Class R" of the Standards of the Tubular Exchanger Manufacturers Association (TEMA).

6.1.5 The selection of TEMA "Class R" or TEMA "Class C" exchangers shall be governed by the following:

6.1.5.1 TEMA "R" is required when:

a) tube side or shell side fouling factor is greater than 0.00035 m².K/W; or

b) shell side corrosion allowance is greater than 3.175 mm (1/8 inch);

c) shell side corrosion rate is greater than 0.254 mm/y (10 mils per year).

6.1.5.2 TEMA "C" may be used when exchanger is designed for chemical cleaning maintenance and fouling factor do not exceed 0.00035 m^2.K/W on both tube side and shell side.

6.1.6 Horizontal and vertical exchangers

Heat exchangers should be of the horizontal type, however, for process requirements and where cleaning and other maintenance will be infrequent and space requirements make it more attractive, the vertical arrangement may be considered and this should be discussed with the Company. Centerline elevation of the top bundle of stacked exchangers shall be limited to 3.5 m except for large exchangers which shall be limited to two stacked shells.

When horizontal arrangements are preferred, the stacking of exchangers should be considered to conserve space in the structure.

6.1.7 The use of spiral plate heat exchangers may be considered when:

a) Small overhead or vent condensers mounted directly on process vessels are required.

b) Space limitations make use of long shell and tube units impractical.

6.1.8 Manufacturer's standard for shell and tube heat exchangers may be considered upon approval of the Company and supplied as components of other equipment such as:

a) Compressor inter/after coolers.

b) Steam ejector inter/after condensers.

c) Machinery lube oil coolers.

6.1.8.1 Fig. A.1 in Appendix A shows different types of shells which has been extracted from TEMA.

6.1.9 Selection guide for heat exchanger types

Table B.1 in Appendix B is also selection guide for heat exchanger types which shows significant feature, applications best suited, limitation and relative cost in carbon steel construction of heat exchangers.

6.1.10 Selection of type

Fixed tube sheet heat exchangers should only be used in services where:

- Differential expansion between the tubes and the shell does not give rise to unacceptable stresses;
- Tube side cleaning, if required, can be done in situ;
- Shell side fluid is non-fouling, or
- Shell side fouling can be removed by chemical cleaning.

U-tube bundle heat exchangers shall only be used in services where:

- Tube side fouling resistance is less than 0.00035 (m^2.K)/W;
- Tube side fouling can be removed by chemical cleaning.

U-tube shall not be applied when tube side mechanical cleaning is required.

Floating head heat exchangers should be used in all other services expect as noted in item 6.1.2.

6.1.11 Shell selection

6.1.11.1 The single-pass shell, Type E (see Fig. A.1 in Appendix A), has the widest application and should be selected for general duties, except where significant advantage can be obtained by using one of the other shell types as indicated below:

6.1.11.2 Where the shell side pressure drop is a restricting factor, the divided flow shell Type J or cross flow shell Type X or double-split flow shell Type H, should be considered.

6.1.11.3 For horizontal shell side thermosiphon reboilers, the split flow shell Type G or Type H should be selected.

6.1.11.4 The kettle type, shell Type K, should be selected for boiling, where an almost 100% vaporization, or where a phase separation is required.

6.1.11.5 Use of the two-pass shell with longitudinal baffle Type F, should be avoided.

6.1.12 Front end and rear end selection

6.1.12.1 Front end bonnet Type B is generally used for heat exchangers where cleaning on the tube side will be infrequent.

Rear end Type S should be used for floating head heat exchangers.

6.1.12.2 Rear end Type M should be applied for fixed tube sheet design.

6.1.12.3 When frequent tube side cleaning is anticipated, and the tube design pressure is low, the front end stationary head shall be Type A, however, for the corresponding rear end, Type L may be selected.

6.1.12.4 For high-pressure and/or very toxic service, where it is desirable to limit the number of external joints, stationary

heads Type B, Type C or Type N should be selected for the front end, and the corresponding Type M or Type N for the rear end.

6.1.12.5 The outside packed floating head Type P, and externally sealed floating tubesheet type W rear ends, are not acceptable.

6.1.13 Water-cooled coolers

The following restrictions shall apply to water-cooled coolers:

6.1.13.1 Cooling water shall run upwards through the tubes in order to avoid build up of gas. The tube side velocity shall be as specified in Table 1.

6.1.13.2 The tube side shall be maintained at an atmospheric over-pressure so that air cannot separate from the water.

6.1.13.3 In open cooling water systems, the cooling water outlet temperature shall not be higher than 42°C, and to avoid scaling, the tube wall temperature on the cooling water side shall not exceed 52°C.

6.1.13.4 Internal bellows shall not be applied.

6.1.13.5 In fouling services, the following additional restrictions apply:

6.1.13.5.1 In cases where flow control of the water is required, tube side velocities should not be allowed to fall below the values specified in Table 1, in order to avoid deposits of mud, silt or salt.

6.1.13.5.2 U-tubes shall not be applied.

6.1.13.6 Shell and tube exchangers using water as the cooling medium are to be avoided when product side temperatures

exceed 200°C.

7. GENERAL REQUIREMENTS

7.1 Fluid Allocation

Fluid allocation shall be made under the following conditions.

7.1.1 Dirty fluids are passed thorough the tubes because they can be easily cleaned, particularly if the tube bundle cannot be removed, but through the shell if the tubes cannot be cleaned (hair pin bundles) or if large amounts of coke or debris are present which can be accumulated in the shell and removed by dumping the shell.

7.1.2 High pressure fluids, corrosive stock, and water are sent through the tubes because the strength of small-diameter (and thin) tube surpasses that of the shell, because corrosion-resistant tubes are relatively cheap, and because corrosion or water scale can be easily removed.

7.1.3 For the same pressure drop, higher heat-transfer coefficients will be obtained on the tube-side than the shell-side.

7.1.4 Large volume fluids (vapors) are passed through the shell because of the availability of adequate space, but small volume fluids are also passed through the shell where cross baffles can be used to increase the transfer rates without producing an excessive pressure drop.

7.1.5 Vapors that contain non-condensable gases are sent through the tubes so that the accumulation of noncondensables will be swept out.

7.1.6 If the pressure drop must be low, the fluids are sent through the shell. The same applies to viscous or low transfer rate fluids because the maximum transfer rates for a fixed pressure drop can be obtained by the use of cross baffles in the shell.

7.1.7 In fin tube equipment, high-pressure, dirty, or corrosive stock is sent through the fin tube because it is relatively cheap, can be easily cleaned, and has a higher strength than the outside tube.

7.1.8 Condensing steam is normally passed through the tubes.

7.1.9 If the temperature change of one fluid is very large (greater than approximately 145°C to 175°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 175°C temperature change cannot be passed through the shell side of a two pass shell.

If the temperatures are high enough to require the use of special alloys placing the higher temperature fluid in the tubes will reduce the overall cost. At moderate temperatures, placing the hotter fluid in the tubes will reduce the shell surface temperatures, and hence the need for lagging to reduce heat loss, or for safety reasons.

7.1.10 If one of the fluids is clean (fouling factor 0.00017 $m^2.°C/W$) and is only mildly corrosive to the material selected this fluid is passed through the tubes and U-tube construction is used, where economical.

7.1.11 Viscosity

Generally, a higher heat-transfer coefficient will be obtained by allocating the more viscous material to the shell-side, providing the flow is turbulent. The critical Reynolds number for turbulent flow in the shell is in the region of 200. If turbulent flow cannot be achieved in the shell it is better to place the fluid in the tubes, as the tube-side heat-transfer coefficient can be predicted with more certainty.

7.2 Installation

7.2.1 Vertical

a) Condensate subcooling may be accomplished more easily in a vertical unit.

b) For boiling fluids, this is usually a single tube pass type with vaporization occurring in the tubes.

7.2.2 Inclined

For tube side condensing fluids, this type of heat exchanger is sometimes employed to ensure positive drainage of the condensate from the tube. Even a few degrees inclination from the horizontal prevents the accumulation of condensate and possible redistribution, flooding, and surging effects.

7.2.3 Horizontal

Others.

7.3 Nozzle Location

The following rules are suggested as a guide for locating heat exchanger nozzles:

1) Streams being heated or vaporized should flow from bottom to top, whether on the tube side or the shell side.

2) Streams being condensed should flow from top to bottom, whether on the tube side or the shell side.

3) The direction of flow of streams being cooled should be dictated by piping economics.

7.4 Impingement Baffles and Erosion Protection

The following paragraphs should provide limitations to prevent or minimize erosion of tube bundle components at the entrance and exit areas. These limitations have no correlation to tube vibration and the designer should refer to Section 6 of TEMA for information regarding this phenomenon.

7.4.1 Shell side impingement protection requirements

An impingement plate, or other means to protect the tube bundle against impinging fluids, shall be provided when entrance line values of ñ.V^2 exceed the following: non-corrosive, non-abrasive, single phase fluids, 2230: all other liquids, including a liquid at its boiling point, 744. For all other gases and vapors, including all nominally saturated vapors, and for liquid vapor mixtures, impingement protection is required. V is the lineal velocity of the fluid in meter per second and ρ is its density in kg per cubic meter. A properly designed diffuser may be used to reduce line velocities at shell entrance.

7.4.2 Shell or bundle entrance and exit areas

In no case shall the shell or bundle entrance or exit area produce a value of ñ.V^2 in excess of 5950 where V is the lineal velocity of the fluid in meter per second and ñ is its density in kilogram per cubic meter.

7.4.2.1 Shell entrance or exit area with impingement plate

When an impingement plate is provided, the flow area shall be considered the unrestricted area between the inside diameter of the shell at the nozzle and the face of the impingement plate.

7.4.2.2 Shell entrance or exit area without impingement plate

For determining the area available for flow at the entrance or exit of the shell where there is no impingement plate, the flow area between the tubes within the projection of the nozzle bore and the actual unrestricted radial flow area from under the nozzle or dome measured between the tube bundle and shell inside diameter may be considered.

7.4.2.3 Bundle entrance or exit area with impingement plate

When an impingement plate is provided under a nozzle, the flow area shall be the unrestricted area between the tubes within the compartments between baffles and/or tubesheet.

7.4.2.4 Bundle entrance or exit area without impingement plate

For determining the area available for flow at the entrance or exit of the tube bundle where there is no impingement plate, the flow area between the tubes within the compartments between baffles and/or tubesheet may be considered.

7.4.3 Tube side

Consideration shall be given to the need for special devices to prevent erosion of the tube ends under the following conditions:

1) Use of an axial inlet nozzle.

2) Liquid ñ.V^2 is in excess of 8925, where V is the lineal velocity in meter per second, and ñ is its density in kg per cubic meter.

7.4.3.1 Shell and tube fluid velocities

High velocities will give high heat-transfer coefficients but also a high-pressure drop. The velocity must be high enough to prevent any suspended solids settling, but not so high as to cause erosion. High velocities will reduce fouling. Plastics inserts are sometimes used to reduce erosion at the tube inlet. Typical design velocities are given below:

7.4.3.2 Liquids

Tube-side, process fluids: 1 to 2 m/s, maximum 4 m/s if required to reduce fouling; water: 1.5 to 2.5 m/s as the following Table 1. Shell-side: 0.3 to 1 m/s.

TABLE 1 - COOLING WATER VELOCITIES (TUBE SIDE)

TUBE MATERIALS	AVERAGE SPEED			
	(m/s)		(ft./sec.)	
	min.	max.	max.	max.
Carbon steel	0.9	1.8	3	6
Admiralty	0.9	1.8	3	6
Aluminum brass	0.9	2.4	3	8
Aluminum bronze	1.5	3.1	5	10
Cupronickel	1.5	3.1	5	10
Aluminum	0.9	3.1	3	10
Monel	1.8	3.6	6	12
Stainless steel	2.4	3.6	8	12

7.4.3.3 Vapors

For vapors, the velocity used will depend on the operating pressure and fluid density; and allowable pressure drop.

7.5 Geometrical

7.5.1 Tube layout and pitch

Standard tube patterns are shown in Fig. 1

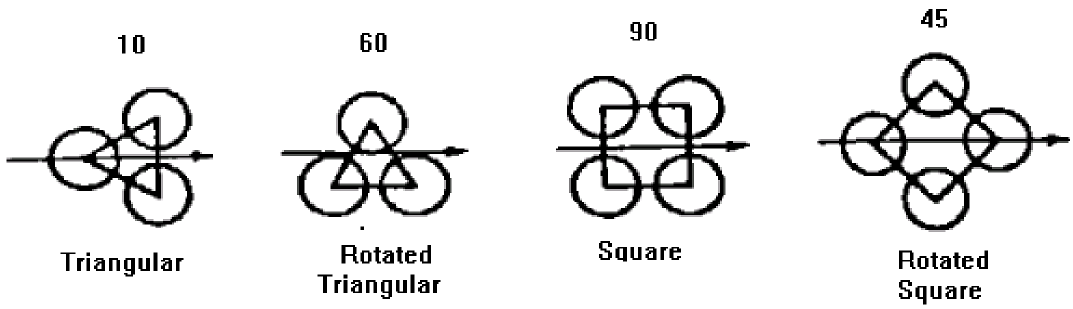


Fig. 1

a) Triangular pitch

Most popular, generally suitable for non-fouling or fouling services of chemical treatment processes medium to high pressure drop; gives better coefficients than square pitch.

b) Rotated triangular pitch

Not as popular as the staggered triangular pitch; coefficients not as high, but better than square pitch; pressure drop about medium to high; generally suitable for same fouling conditions as above.

c) Square pitch

Popular for conditions requiring low pressure drop and/or cleaning lanes for mechanical cleaning of outside of tubes; coefficient lower than triangular pitch.

d) Rotated square pitch

Popular arrangement for reasonably low pressure drop (not as low as square), mechanical cleaning requirements, and better coefficient than square pitch.

7.5.2 Tube pitch

7.5.2.1 Tube pitch (for Class R)

Tubes shall be spaced with a minimum center-to-center distance of 1.25 times the outside diameter of the tube. When mechanical cleaning of the tubes is specified by the Company, minimum cleaning lanes of DN8 (¼ inch) shall be provided.

7.5.2.2 Tube pitch (for Class C)

Tubes shall be spaced with a minimum center-to-center distance of 1.25 times the outside diameter of the tube. Where the tube diameters are DN 16 (5/8 inch) or less and tube-to-tubesheet joints are expanded only, the minimum center-tocenter distance may be reduced to 1.20 times the outside diameter.

7.5.2.3 Tube pitch (for Class B)

Tubes shall be spaced with a minimum center-to-center distance of 1.25 times the outside diameter of the tube. When mechanical cleaning of the tubes is specified by the Company and the nominal shell diameter is DN300 (12 inches) or less, minimum cleaning lanes of DN 5 (3/16 inch) shall be provided. For shell diameters greater than DN 300 (12 inches), minimum cleaning lanes of DN 8 (¼ inch) shall be provided.

7.5.3 Bypasses and block valves

Bypasses and block valves are used on an exchanger for one or more of the following reasons:

7.5.3.1 Process control

When it is necessary to control the heat transfer in an exchanger, use either:

1) A simple bypass and 2 valves or

2) A bypass and a three-way valve (or two butterfly valves) which splits the flow between the bypass and the exchanger.

A three-way valve is necessary when the exchanger pressure drop is so small that insufficient fluid diversion would take place through a simple bypass in the wide open position.

7.5.3.2 Leakage

Where leaking of one side of a heat exchanger would result in intolerable contamination of the other fluid (as in an overhead vapor-feed exchanger of a pipestill), blocks and bypasses may be installed to permit isolating the leaking unit from the system. In addition, welded tube to tubesheet joints or double tubesheets should be considered.

7.5.4 Baffles

7.5.4.1 Types

The four types of cross baffles in shell and tube type exchangers are illustrated in Fig. 2:

a) Segmental

This type is probably the most popular.

b) Double segmental

Double segmental baffles give one-third to one-half the pressure drop and 60 to 90 percent of the heat transfer for the same total flow rate compared with units with segmental baffles having the same spacing and cut. Therefore, if the pressure drop is a limitation factor at the maximum allowable segmental baffle spacing, the use of a double segmental baffle should be investigated.

c) Triple segmental

Triple segmental baffles have proved very effective in low pressure drop applications in both laminar and turbulent flows.

d) No-tubes-in-window

Support plates can be used with no-tubes-in-window type baffles. Therefore these type baffles can eliminate flow induced tube vibration.

7.5.4.2 Cut as percentage of shell in side diameter

The percent baffle cut is determined at position 1 as shown in Fig. 3:

a) Segmental

Maximum cuts allowed are 49 percent.

b) Double, triple segmental

Normally specifies the cut of baffles in a manner which gives equal net flow area for each window.

c) No-tubes-in-window

The cut is extremely important because it not only defines the window flow area but also affects the tube count and the height under the window.

7.5.4.3 Baffle cut orientation:

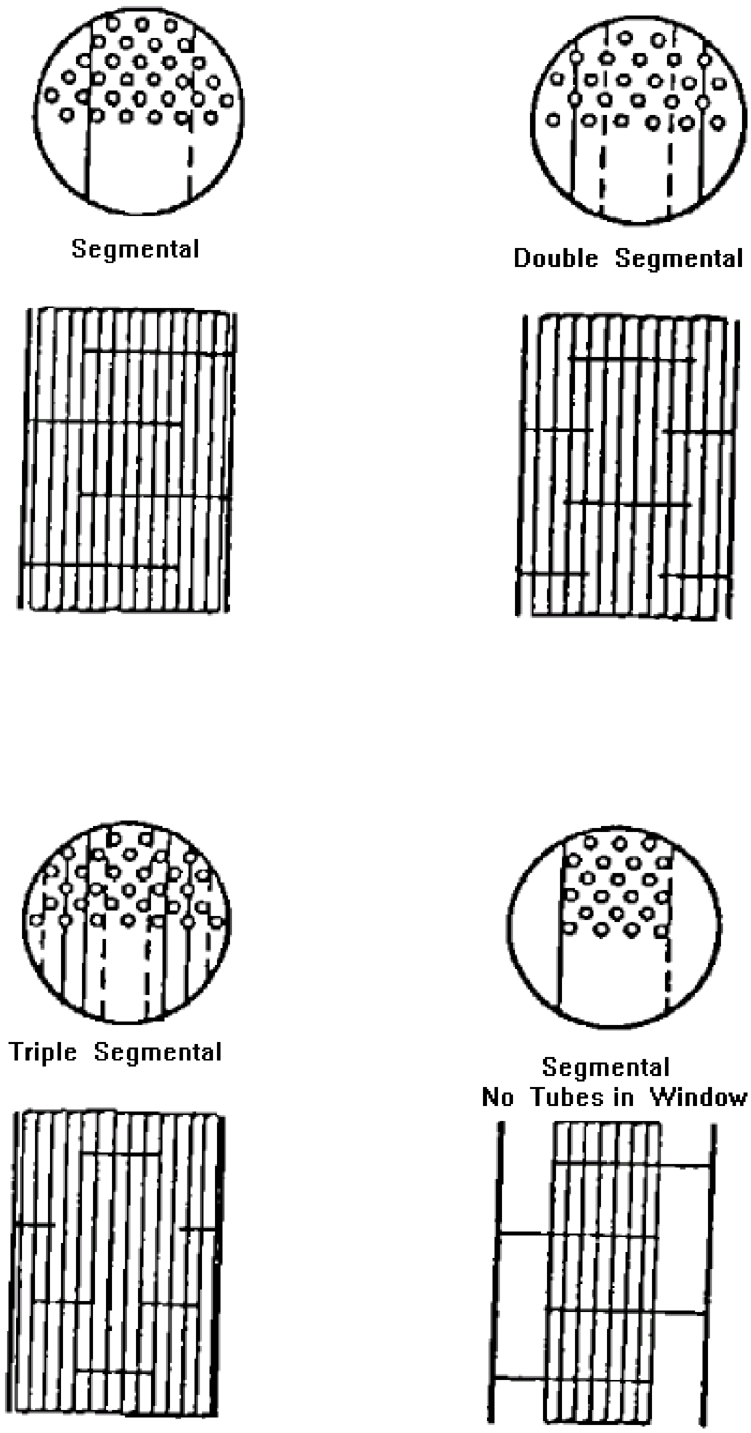
a) Vertical cut

The baffle edge is usually vertical for service in horizontal condensers, reboilers, vaporizers and heat exchangers carrying suspended matter or heavy fouling fluids. With this arrangement non-condensable vapors and inert gases can escape or flow along the top of the unit and thus prevent vapor binding or vapor lock causing a blanking-to-heat transfer of the upper portion of the shell.

Also, equally important as the passage of vapor, is the release of liquid from the lower portion of the shell as it is produced.

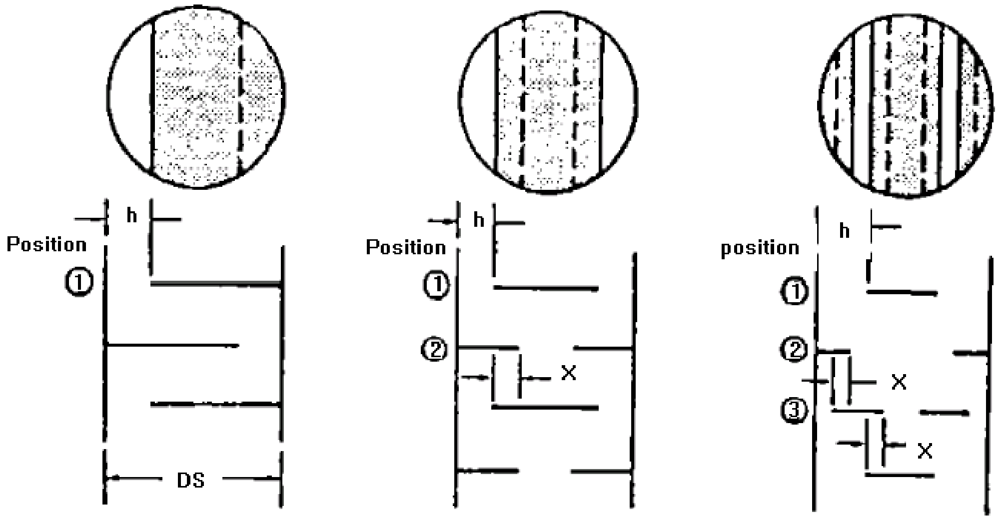
b) Horizontal cut

The horizontal cut baffles are good for all gas phase or liquid phase service in the shell. However, if there is dissolved gas in the liquid which may be released in the exchanger, this baffle should not be used, or else notches should be cut at the top for the passage of gas. Notches, will only be effective for small traces of released gas. Liquid should be clean, otherwise sediment will collect at the base of every other baffle segment and blank off part of the lower tubes to the heat transfer.



BAFFLE TYPES

Fig. 2



Note:

% Baffle cut = $\frac{h}{DS}$ **× 100**

Overlap = × (normally two tube rows overlap)

BAFFLE CUT

Fig. 3

7.5.5 Spacing of baffles and support plates

7.5.5.1 Minimum spacing

Segmental baffles normally should not be spaced closer than 1/5 of the shell ID or DN 50 (2 inches), whichever is greater. However, special design considerations may dictate a closer spacing.

7.5.5.2 Maximum spacing

Tube support plates shall be so spaced that the unsupported tube length does not exceed the value indicated in the project specification for the tube material used.

7.5.6 Finned tube

It has been recognized for some time that the use of integral finned tubes in shell and tube type heat exchangers can have significant economic advantages in certain applications. On the other hand, there are also conditions under which these tubes would provide no advantage over plain tubes. In order to make proper use of their potential, therefore, each case should be carefully evaluated. Some of the factors which limit the use of finned tubes are as follows:

7.5.6.1 Low tubeside heat transfer coefficient

If the tubeside resistance is a controlling factor, the outside finned surface is almost completely ineffective, and plain tubes should be used instead.

7.5.6.2 High tubeside fouling factor

The additional outside area does nothing to decrease the tubeside fouling resistance and, if this factor is controlling, finned tubes should be used.

7.5.6.3 High shellside fouling factor

The effect of fouling on the outside of finned tubes has long been a controversial subject. It is obvious that if fin valleys become filled with fouling deposit, heat transfer cannot take place at the finned surface.

7.5.6.4 Extended surface

Extended surface exchangers are characterized by tubes with either longitudinal or transverse helical fins. This type of surface is best employed when the heat transfer properties of one fluid result in a high resistance to heat flow and those of the other fluid have a low resistance. The fluid with the high resistance to heat flow contacts the fin surface.

7.5.6.5 Brazed plate fin

Brazed plate fin heat exchangers are made up of a stack of layers which consist of a corrugated fin between plate metal sheets, sealed off on two sides by channels or bars to form a passage for the flow of fluid. Maximum design conditions are about 41 bar(ga) at 38°C. Typical design conditions are for lower pressure and subzero temperatures. Plate-fin type exchangers in applicable services in some recent ethylene plant designs have been used.

7.5.6.6 Spiral wound (Hampson Coil)

Spiral tube heat exchangers consist of a group of concentric spirally wound coils, which are connected to tube sheets.

Features include countercurrent flow, elimination of differential expansion problems, compactness, and provision for more than two fluids exchanging heat. In general, these units are used in cryogenic applications where the process pressure is 45 bar (ga) or greater.

8. BASIC RELATIONS

8.1 Fluid Temperature Relations

8.1.1 Logarithmic mean temperature difference (LMTD)

For cases of true countercurrent or cocurrent flow, the logarithmic mean temperature difference should be used if the following conditions substantially apply:

- Constant overall heat transfer coefficient.
- Complete mixing within any shell cross pass or tube pass.
- The number of cross baffles is large.
- Constant flow rate and specific heat.
- Enthalpy is a linear function of temperature.
- Equal surface in each shell pass or tube pass.
- Negligible heat loss to surroundings or internally between passes.

8.1.2 Correction for multipass flow

In multipass heat exchangers, where there is a combination of cocurrent and countercurrent flow in alternate passes, the mean temperature difference is less than the logarithmic mean calculated for countercurrent flow and greater than that based on cocurrent flow. The correct mean temperature difference may be evaluated as the product of the logarithmic mean for countercurrent flow and an LMTD correction factor. For these factors refer to TEMA.

8.2 Fouling

8.2.1 Types of fouling

Currently five different types of fouling mechanisms are recognized. They are individually complex, often occurring simultaneously, and their effects may increase pressure drop, accelerate corrosion and decrease the overall heat transfer coefficient. The five different types are:

a) Precipitation fouling.

b) Particular fouling.

c) Chemical reaction fouling.

d) Corrosion fouling.

e) Biological fouling as mentioned below.

For further information refer to TEMA.

8.2.2 Considerations in evaluating fouling resistance

The determination of appropriate fouling resistance values involves both physical and economic factors, many of which vary from user to user, even for identical services. When these factors are known, they can be used to adjust typical base values.

8.2.2.1 Physical considerations

Typical physical factors influencing the determination of fouling resistances are:

- Fluid properties and the propensity for fouling.
- Heat exchanger geometry and orientation.
- Surface and fluid bulk temperatures.
- Local fluid velocities.
- Heat transfer process.
- Fluid treatment.
- Cathodic protection.

8.2.2.2 Economic considerations

Typical economic factors influencing the determination of appropriate fouling resistances are:

- Frequency and amount of cleaning costs.
- Maintenance costs.
- Operating and production costs.
- Longer periods of time on stream.
- Fluid pumping costs.

- Depreciation rates.
- Tax rates.
- Initial cost and variation with size.
- Shut down costs.
- Out-of-service costs.

8.2.2.3 Further explanation to physical considerations

a) Surface and bulk temperatures

For many kinds of fouling, as the temperatures increase, the amount of fouling increases. Lower temperatures produce slower fouling build-up and deposits that often are easier to remove.

b) Local velocities

Normally, keeping the velocities high reduces the tendency to foul. Velocities on the tube side are limited by erosion, and on the shell side by flow-induced vibration. Stagnant and recirculation regions on the shell side lead to heavy fouling.

c) Tube material, configuration and surface finish

The selection of tube material is significant when it comes to corrosion. Some kinds of biological fouling can be lessened by copper-bearing tube materials. There can be differences between finned and plain tubing. Surface finish has been shown to influence the rate of fouling and the ease of cleaning.

d) Heat exchanger geometry and orientation

The geometry of a particular heat exchanger can influence the uniformity of the flows on the tube side and the shell side. The ease of cleaning can be greatly influenced by the orientation of the heat exchanger.

e) Heat transfer process

The fouling resistances for the same fluid can be considerably different depending upon whether heat is being transferred through sensible heating or cooling, boiling, or condensing.

f) Place the more fouling fluid on the tube side

There are two benefits from placing the more fouling fluid on the tube side. There is less danger of low velocity or stagnant flow regions on the tube side, and, it is generally easier to clean the tube side than the shell side. It is often possible to clean the tube side with the exchanger in place while it may be necessary to remove the bundle to clean the shell side.

g) Fouling is usually not severe below 121°C.

h) Fouling is more severe when heating hydrocarbons than when cooling them. In the case of pipestills, this is due to "salting out". To minimize this sort of fouling, a crude preheat train sometimes includes a desalter or a flash drum to remove water before the crude reaches the "salting out" temperature.

i) Vaporization in an exchanger can cause severe fouling.

j) High velocities reduce fouling. This is especially true in the case of cooling water that contains salt.

k) The feed to catalytic reformers and catalytic cracking plants is sometimes severely fouled due to organic reactions with oxygen while the feed is in intermediate tankage. Inert gas blanketing of the tankage is often used to reduce this fouling.

l) Bottoms from a crude distillation tower, even though heavy and at a high temperature will not normally cause much fouling (provided flash zone temperatures are not excessive).

8.2.3 Application of lower fouling resistances

Lower fouling resistances may be appropriate if one or more of the circumstances described below apply. However, such lower values may be applied only where specifically approved by the Company in writing.

8.2.3.1 In services where the surface requirements are significantly influenced by the magnitude of fouling, it may be advantageous to specify a lower resistance if a reduced period between two successive shutdown is feasible. This can be achieved for instance by the installation of a spare exchanger in parallel with the one in operation, thus enabling cleaning at any time, without plant shut-down. This is especially important where controllability/stability is influenced by fouling, e.g., thermosyphon reboilers.

8.2.3.2 The maximum allowable pressure drop generally limits the fluid velocity. This means that for designs where low pressure drops have to be applied fluid velocities will often become low. If the specified fouling resistance is also high, resulting in the installation of considerable oversurface in clean condition, the maximum attainable velocities will appreciably reduce, which will increase the tendency to fouling. By taking a small fouling resistance, a smaller heat exchanger will be adequate, thus making it possible to apply a higher velocity and still stay within the limits of allowable pressure drop.

Though some construction materials can have a beneficial influence on fouling, there is generally inadequate information available. An exception can be made for titanium in cooling water service, where some relaxation of the specified fouling values may be considered for each separate case.

8.2.4 The best design fouling resistances, chosen with all physical and economic factors properly evaluated, will result in a minimum cost based on fixed charges of the initial investment (which increase with added fouling resistance) and on cleaning and down-time expenses (which decrease with added fouling resistance). By the very nature of the factors involved, the manufacturer is seldom in a position to determine optimum fouling resistances. The user, therefore, on the basis of past experience and current or projected costs, should specify the design fouling resistances for his particular services and operating conditions. In the absence of specific data for setting proper resistances as described in the previous paragraphs, the user may be guided by the values tabulated in the section of TEMA standards. In the case of inside surface fouling, these values must be multiplied by the outside/inside surface ratio.

8.2.4.1 Design fouling resistances ($m^2.°C/W$)

The user should attempt to select an optimal fouling resistance that will result in a minimum sum of fixed shut-down and cleaning costs. The following tabulated values of fouling resistances allow for oversizing the heat exchanger so that it will meet performance requirements with reasonable intervals between shut-downs and cleaning. These values do not recognize the time related behavior of fouling with regard to specific design and operational characteristics of particular heat exchangers.

8.2.4.2 The normal fouling factors for a variety of process services are recommended by TEMA. The tabulated fouling factors are intended to prevent the exchanger from delivering less than the required process heat load for a period of about a year to a year and a half. That table is only a guide, however, and if specific data is available which can be used to determine a more accurate fouling factor for a particular service, that data should be used in preference to Table C.1 of Appendix C.

8.2.4.3 The actual importance of the fouling factors depends on the value of the clean coefficient Uo, that is the better the coefficient, the more important is the fouling factor.

8.2.4.4 After making a preliminary calculation of Uo, it is easy to determine the effect of doubling (or halving) the assumed fouling factors on the size of the exchanger. If this effect is small (5% or less), it is not worthwhile trying to determine a more accurate fouling factor. There are many occasions, however, when Uc is so large that the size of the

exchanger depends almost entirely on the value of the fouling factor. For these cases, all available plant data should be closely examined.

9. THERMAL DESIGN

9.1 Pressure Drop

Maximum acceptable pressure drops indicated in the process data sheet shall be understood for fouled exchangers and as inclusive of the pressure drops through inlet and outlet nozzles. In cases of alternate conditions these shall apply to the worst operating condition.

9.2 Design Velocities

Design velocities in tubes for cooling water shall be kept within the undermentioned operating range (see Table 1).

9.3 Exchanger Design Pressures and Temperatures

9.3.1 Design pressures

9.3.1.1 Design pressures shall be as shown on the individual process data sheet.

9.3.1.2 Unless otherwise specified, design pressure for heat exchangers shall be established as follows:

Max. Operating Pressure (MOP)	Design Pressure (DP) [6), 7)]
- Atmospheric pressure	Hydrostatic (water) pressure + 35 mm WC min.(see Note 4)
- Vacuum (see Note 3)	Absolute vacuum and 3.5 bar (ga) min.
- Between 0 and 1.5 bar (ga)	3.5 bar (ga) min.
- Between 1.5 and 20 bar (ga)	max. oper. pressure + 2 bar min.
- Between 20 bar (ga) and 80 bar (ga)	max. oper. press. + 10% min.
- Between 80 bar (ga) and 140 bar (ga)	max. oper. press. + 8 bar min. (see Note 5)
- Above 140 bar (ga)	max. oper. press. + 5% min. (see Note 5)

Notes:

1) In defining the design temperature due consideration shall be given to the start-up, shutdown upset or any other condition that could result in a temperature lower than the normal operating. However, for all of the above conditions, the corresponding pressure shall be considered too.

2) Design temperatures lower than 85°C are allowed only for insulated equipment for which a design temperature of 60°C shall be selected.

3) Steam drums shall be designed for full vacuum conditions.

- Due consideration shall be taken to establish external design pressure for vessels subject to internal pressure but connected to the suction of compressor or other evacuating equipment.

- Vacuum design conditions shall not be required as consequence of equipment block in after steam out operation.

4) Same criteria is applied for the design of atmospheric storage tanks.

5) When design pressure lower than 110% max. operating is specified, safety valves blowdown shall be selected accordingly.

6) Exchangers in gas turbine driven compressor discharge circuits shall have the setting of the PSV (Pressure Safety Valve) in that circuit. In low pressure systems (less than 20-25 bar), the relief valve setting should be equal to the compressor's maximum case working pressure.

In higher pressure systems, the PSV setting should be considered on a case by case basis.

7) Exchangers in refrigerant service shall have a minimum DP based on vapor pressure of refrigerant maximum temperature.

8) Indicate maximum sun temperature for uninsulated exchangers. Insulated case should be calculated.

9.3.1.3 In case of equipment connected in series, without block valves in between, the design pressure for the upstream equipment shall be the same as the design pressure for the downstream equipment (equipped with safety valve) increased by 110% of the pressure drop foreseen between the two equipment, under safety valve discharge conditions.

9.3.1.4 Design pressure for heat exchangers, low pressure side, (when technically justified) will be not less than 1.5 time the high pressure side design pressure to avoid the installation of PSV for tube rupture. This higher value of design pressure should be extended until the first block valves.

9.3.1.5 Exchangers operating under a vacuum shall be designed for full vacuum.

9.3.1.6 Tube plates may be required to withstand differential pressure in high pressure exchanger when specified on the process data sheet.

9.3.2 Design Temperatures

9.3.2.1 Design temperature shall be as shown on the individual process data sheet.

9.3.2.2 Unless otherwise specified, design temperature for heat exchangers, shall be established according to the following criteria:

Operating Temperature (OT)	**DesignTemperature (DT)**$^{4)}$
- Less than -100°C	min. oper. temp./85°C min. (see Note 2)
- Between -40°C and	-100°C -100°C/85°C min. (see Note 2)
- Between -30°C and -39°C	-45°C/85°C min. (see Note 2)

- Between -29°C and +60°C	min. oper. temp. /85°C min. (see Note 2)
- Between 60°C and 343°C	max. oper. temp. +25°C. (see Note 8)
- Above 343°C	To be specified according to selected material and process requirement.

Note:

For note explanation see Article 9.3.1.2.

9.3.2.3 The design temperature is determined for the maximum temperature coincident with the design pressure as determined above. Indicate any higher temperatures as alternate design conditions.

9.3.2.4 Exchangers which will operate at temperatures 0°C and below shall be designed for minimum anticipated operating temperature.

9.3.2.5 Maximum water outlet temperatures on coolers and condensers shall be based on the water characteristics.

9.3.2.6 When, due to the possible loss of flow of the cooling medium, the tubes, tube sheets and floating heads may be subject to the full inlet temperature, it shall be indicated on the individual process data sheet and these components shall be designed for the maximum anticipated operating temperature of the hotter medium.

9.3.2.7 The design temperatures for multiple exchangers in series shall be selected in accordance with the maximum temperatures likely to occur on each exchanger in both clean and fouled condition. The design temperature indicated on the process data sheet is the temperature of the hottest exchanger.

Intermediate design temperatures shall be calculated assuming the highest heat transfer coefficient with fouled surface and the lowest heat transfer coefficient with fouled surface for the colder and hotter sections respectively.

If irregular heat profiles are indicated on the process data sheet, design data will be supplied on which the Vendor shall base all calculations, which shall be submitted to the Company for approval.

9.3.2.8 For fixed tubesheet exchangers without expansion joints, the differential between the average shell metal temperature and the average metal temperature of any one tube pass shall not exceed 28°C. When temperature differentials exceed 28°C an expansion joint shall be furnished.

For two-pass-shell exchangers the differential between the inlet and the outlet temperature of the shell side fluid shall not exceed 194°C.

PART II

PROCESS DESIGN OF PLATE HEAT EXCHANGERS

(PLATE FIN EXCHANGERS)

10. PLATE FIN EXCHANGERS

As the name implies, a plate-fin exchanger consists of a series of: parallel metal (usually aluminum) plates between which are sandwiched corrugated metal (usually aluminum) sheets. The corrugations act as fins providing extended surface area for heat transfer, giving the unit mechanical strength and forming a large number of parallel flow channels.

The sides of each sandwich are sealed with metal (usually aluminum) bars thereby forming the overall flow passage and the entire construction is brazed in a molten salt bath. Metal (usually aluminum) headers are then welded to the ends of the core.

11. APPLICATION

11.1 Typical petrochemical processes utilizing plate-fin exchangers are:

- air separation;
- helium extraction from natural gas;
- ethylene recovery;
- natural gas liquefaction;
- hydrogen purification and liquefaction; and,
- refrigeration systems used in conjunction with any of these processes.

11.2 Suitable for fouling services and where a high degree of sanitation is required, as in food, dairy, brewing industries and pharmaceutical processing.

11.3 Treating Crudes

The low cost of titanium in the PHE (Plate Heat Exchanger) has made it a favorite in treating crudes. Highly corrosive crudes, produced water, desalter water and brackish and sea water coolants have little effect on the plates of a titanium PHT. Fouling factors only 1/10 that of shell and tube heat exchangers lend extra reliability in these services.

11.4 Gas Absorption Systems

A number of PHEs has been installed in gas processing absorption systems. PHE amine water and amine amine interchangers are common. Common amines such as MEA (mono-Ethanolamine) and DEA (di-Ethanolamine) as well as proprietary absorbents such as DGA (di-Glycolamine). Sulfinol and Selexol are all handled by PHEs in gas absorption service.

11.5 Some of these call for very long temperature programs. To cool Selexol from 99 to -12°C, for example, would have required 13 two-pass shell and tube units. Instead two PHEs were installed. These PHEs had nine passes on each side. Use of brine as a coolant indicated titanium plates, which would be impractical in conventional heat exchangers.

11.6 Tail Gas Treatment

Sulfidity removal from tail gas is another increasingly common PHE application. The PHE easily met the 35 kPa limit on pressure drop across each side.

11.7 Water-to-Water Applications

In the refinery, water-to-water temperature control has been the most frequent application of the PHE. Indirect cooling, often using brackish or natural salt water once through and discharged, is common. Tempered water, fresh water, condensate and process water are cooled. PHEs also have been used for heat and cool glycol and glycol water in a number of refineries.

11.8 Most application for plate heat exchangers are for liquid/liquid duties at operating pressure below 2,100 kPa (ga) or 21 bar (ga) and temperatures below 150°C, although some models can operate at temperature up to 275°C.

11.8.1 Pressure limitation

Maximum Allowable Working Pressure (MAWP) may be determined by frame strength, gasket retainment, or plate deformation resistance . It is often the frame that limit operating pressure.

All Plate Heat Exchanger used in chemical industries are capable of operating at 600 kPa (ga) or 6 bar (ga), most at 1000 kPa (ga) or 10 bar (ga), many at 1600 kPa (ga) or 16 bar (ga) and some at 2100 kPa (ga) or 21 bar (ga).

11.8.2 Temperature limitation

Normally it is the gasket that limit the Maximum Operating Temperature (MOT) for Plate Heat Exchanger.

In the absence of chemical attack, following may serve as a rough indication guide:

- Natural Rubber, Styrene Resin, Neoprene.	max.	70°C
- Nitrile, Viton (FPM) Resin*.		190°C
- Cured Butyl.		120°C
- Ethylene/Propelene, Silicone.		140°C
- Compressed Asbestos Fiber (CAF).		200°C

Operating temperature may also be limited by plate corrosion effect.

*** Note:**

Viton is trademark for a series of fluoroelastomers based on the copolymers of vinylidene fluoride and hexafluoropropylene, with the repeating structure Possibly - CF_2 - CH_2 - CF_2 - CF (CF_3) -.

It is non-flammable and resistant to corrosive liquids and chemicals up to 315°C. Useful continuous service at 204 - 232°C. It is further resistant to ozone, weather, flame, oils, fuels, lubricants and many solvents. Further has a good radiation resistance.

12. MATERIAL

12.1 Plates can be pressed in many different metals, including Aluminum, (usually) Stainless Steel (304SS, 316SS), Titanium, Hastelloy Alloys, Nickel, Monel, Incolloy 825, Inconel 600 and 625, Aluminum Brass and Hastelloy B & C.

12.2 Gaskets are available in nitrile, Resin Cured Butyl (RCB), viton (FPM) resin, EPDM, silicon, and fluorocarbon rubbers and natural rubber, styrene resin; in addition, certain plates can be supplied with gaskets of Compressed Asbestos Fiber (CAF).

13. CONSTRUCTION

13.1 Unlike shell and tube units which can be custom built to conform to virtually any capacity and operating conditions, plates are mass produced in thicknesses range from 0.6 to 1 mm by complex and expensive press tools. They are therefore available only in a limited number of types and sizes, each of which has its own clearly defined specification with regard to performance and operating conditions.

14. ADVANTAGES

Some of the advantages inherent in the plate-fin construction are as follows:

14.1 A very high degree of compactness can be achieved. Surface area to volume ratios of 1476 m^2/m^3 are quite common and values up to 2526 m^2/m^3 have been reported. In comparison, conventional shell and tube units have ratios of 164 to 246 m^2/m^3.

14.2 Three or four process streams can be easily accommodated in a single unit with the plate spacing and fin construction optimized for each of the streams. Such multi-stream units are ideal for operating as reversing units for the removal of impurities.

14.3 Cores can be used individually or connected in series and/or parallel as manifolded assemblies.

14.4 Small size and light mass permit compact installations with minimum foundations and supporting structures.

14.5 Pumping costs per unit of heat transfer are said to be lower than for shell and tube equipment.

14.6 Plate heat exchangers achieve high heat transfer rates that greatly reduce the surface area required. Since these low surface areas are on thin plates, plate heat exchangers need much less material than comparable conventional units.

The plate heat exchangers take less space in the refinery and cost less even when expensive materials are used. Noncorrosive titanium has become a standard for plate heat exchangers in oil and gas processing.

15. DISADVANTAGES

Against these advantages several disadvantages and limitations must be kept in mind.

15.1 Maximum operating pressures are limited to 4500 kPa (ga) or 45 bar (ga) under steady conditions and up to 2100 kPa (ga) or 21 bar (ga) under reversing conditions.

15.2 Plate-fin exchangers cannot be used where one or more of the process streams have a tendency to foul.

15.3 Internal leaks between passes are difficult to locate and correct in the field. Equipment for different alloy welding is necessary, and highly skilled personnel are required.

16. DESIGN CONSIDERATIONS (PLATE FIN EXCHANGERS)

16.1 Exchanger Geometry

The core of a plate-fin exchanger is built up of a number of elemental sandwiches of the type shown in Fig. 4. Several types of flow patterns are possible and with any of these patterns the size and type of corrugation may be varied for each stream.

16.1.1 Flow patterns

Plate fin exchangers have two basic flow patterns, crossflow and counterflow which are illustrated in Fig. 4. These basic patterns are then built up to form simple crossflow, multi-pass crossflow, counterflow and multi-stream units by using suitable internal seals, distributors and external header tanks. Some typical arrangements are illustrated in Fig. 5.

Selection of the proper flow pattern for a particular application depends on several factors including flow rates, pressure levels and the temperature effectiveness of each stream. Temperature effectiveness determines the LMTD correction factor and therefore the size penalty associated with flow patterns other than counterflow.

In the simple crossflow exchanger shown in Fig. 5, the fins run throughout the full length of each passage and no internal distributors are necessary. This configuration is often used in liquefiers where the warm stream is condensed, with little temperature change, while exchanging heat with a large throughput of a low pressure gas. Temperature effectiveness is generally greater than 60 percent if an excessive size penalty is to be avoided.

In the multi-pass unit shown in Fig. 5, one stream flows straight through while the other is guided by internal seals and external tanks to make the required number of passes. The unit basically consists of several crossflow sections assembled in counterflow formation with a mean effective temperature difference approaching that of counterflow.

In most low temperature applications counterflow is generally specified. Header arrangements must be matched to the type of service. The counterflow units shown in Fig. 5 include:

Type 1: which is mainly used in low pressure applications;

Type 2: a symmetrical arrangement suitable for reversing duty and for high pressure units; and

Type 3: which may be used for three or more streams.

16.1.2 Corrugations

To satisfy widely differing requirements, several types of corrugations have been developed. The more commonly used corrugations are listed as follows along with their alternate designations:

1) Plain - (straight).

2) Lanced - (strip, serrated, multi-entry).

3) Ruffled - (wavy, herringbone).

4) Perforated.

Plain fin surfaces are characterized by long uninterrupted flow passages with performance similar to that obtained inside circular tubes. Plain-fin surfaces include those with rectangular passages, triangular passages and passages with rounded corners. In general, the lanced, ruffled, and perforated corrugations offer enhanced heat transfer and pressure drop characteristics. When compared to the plain fin, their use results in a reduction in length with some increase in cross-sectional area for a given thermal load and pressure loss.

Within each general category of corrugation there are variations in specific fin geometry. For industrial applications fin dimensions generally lie within the following ranges:

- height	3.81	to	11.43 mm;
- thickness	0.153	to	0.635 mm;
- pitch	8	to	18 fins per 25.4 mm.

The percentage of fin surface area removed in perforated fins generally lies between 10 and 25 percent. The resulting surface area per unit core volume lies between 820 and 1476 m^2/m^3.

16.1.3 Nomenclature specific to plate-fin exchanger geometry is given in Table 2. Also given in Table 2 are relations for calculating the geometrical properties of a plate-fin surface from the fin dimensions. Note that although the calculated geometrical properties of plain and lanced fins having the same dimensions are the same, their heat transfer and pressure drop characteristics are markedly different as will be seen later in this subsection.

A partial list of the many industrial corrugations available from the principal U.S. manufacturers is given in Table 3.

Included in this table are the geometrical properties calculated using the relationships of Table 2.

16.1.4 Fin selection criteria

16.1.4.1 The selection of the optimum fins for a particular application is at best a difficult trial procedure due to the large number of process variables involved and the large number of available surfaces. However, some general preliminary fin selection criteria can be given based on the nature of the fluid stream and on the working pressure.

The most widely used fin is the lanced fin. It is generally suitable for application in all gas, all liquid, condensing, and vaporizing services. It is the first fin that should be considered when selecting surfaces for a particular application. The use of plain fins is generally limited to special cases of liquid and condensing flow and to cases where the free passage of contaminating solids is desired. Perforated fins are often used in condensing and vaporizing service and in the distribution sections of counterflow units.

16.1.4.2 Fin height and thickness are subject to working pressure limitations. The use of fins that are 9.52 mm and taller and less than 0.3048 mm thick is limited to working pressures below 21 bar (ga). At higher pressures, shorter and thicker fins are necessary. Pressure considerations permitting, the taller corrugations are used for gas streams while those with heights 6.35 mm and smaller are used for liquids. Wavy fins are generally at least 9.52 mm high and as such are sometimes used with low pressure gas streams.

16.1.5 Core size limitations

The maximum size of a single core is limited by mechanical design considerations and by manufacturing facilities.

Pressure loadings limit core cross-sections to between 0.232 and 0.836 m². The size of brazing ovens limit core lengths to about 3.048 m although recently some 6.096 cores have become available for low pressure operation. A list of the maximum size cores for various non-reversing pressure levels is given in Table 4. For reversing operation, the maximum working pressure for each of the cores listed should be taken as one half the value given in the table.

Included in Table 4 are the thicknesses of the separation sheets required at the various pressure levels.

16.1.6 Dummy passages and outside sheets

To protect the exchanger core during shipping and installation, dummy layers of 6.35 mm fins and outside sheets 6.35 mm thick are brazed to the top and bottom of the core. In some cases the dummy passages are not necessary.

16.1.7 Distribution section

The counterflow units shown in Fig. 5 require distributor sections to uniformly spread the flow from the headers over the width of the core. These sections are simply plate-fin arrangements installed at an angle to the core fin direction. In addition, there are suitable internal seals to help guide the flow.

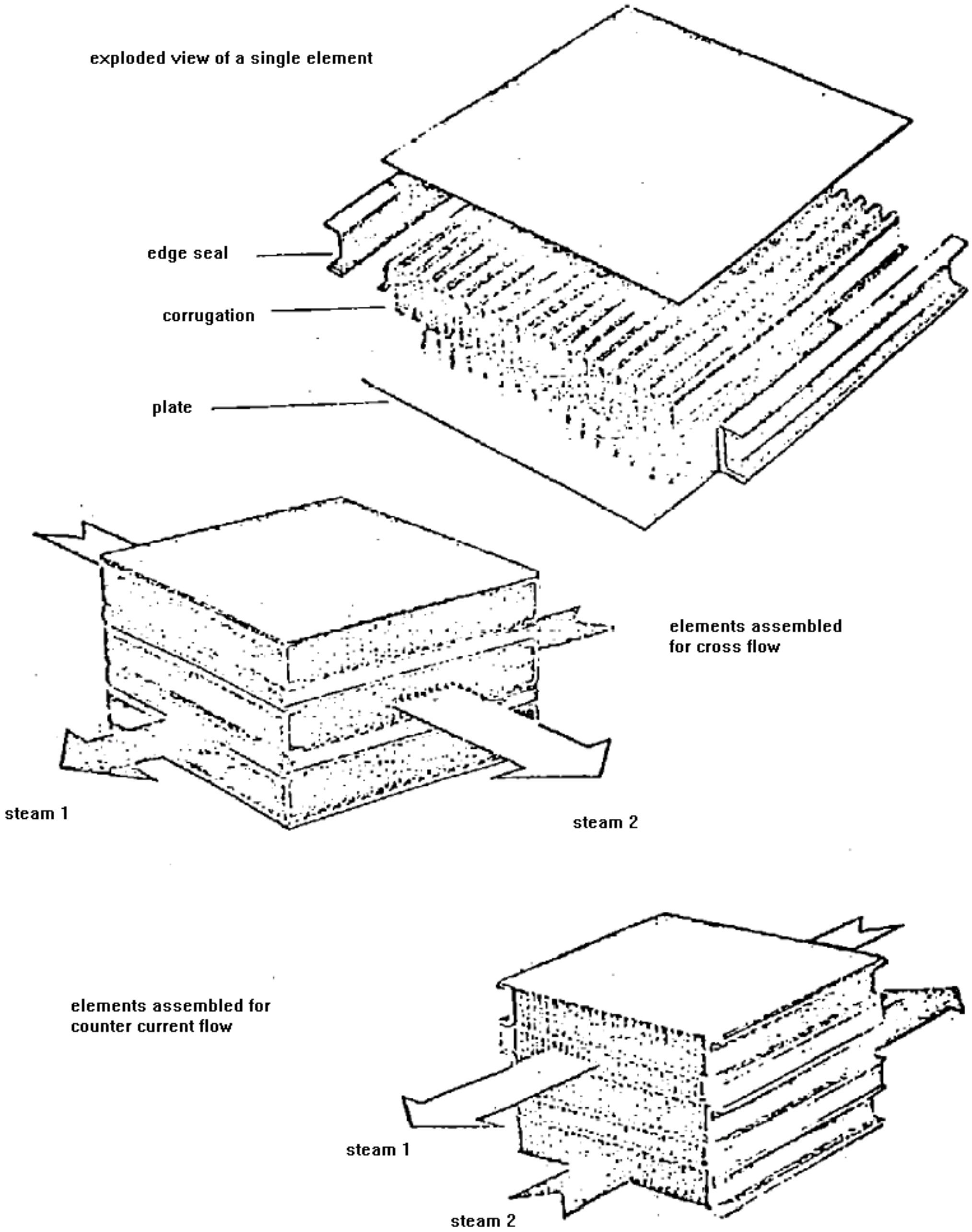


PLATE FIN EXCHANGER ASSEMBLY

Fig. 4

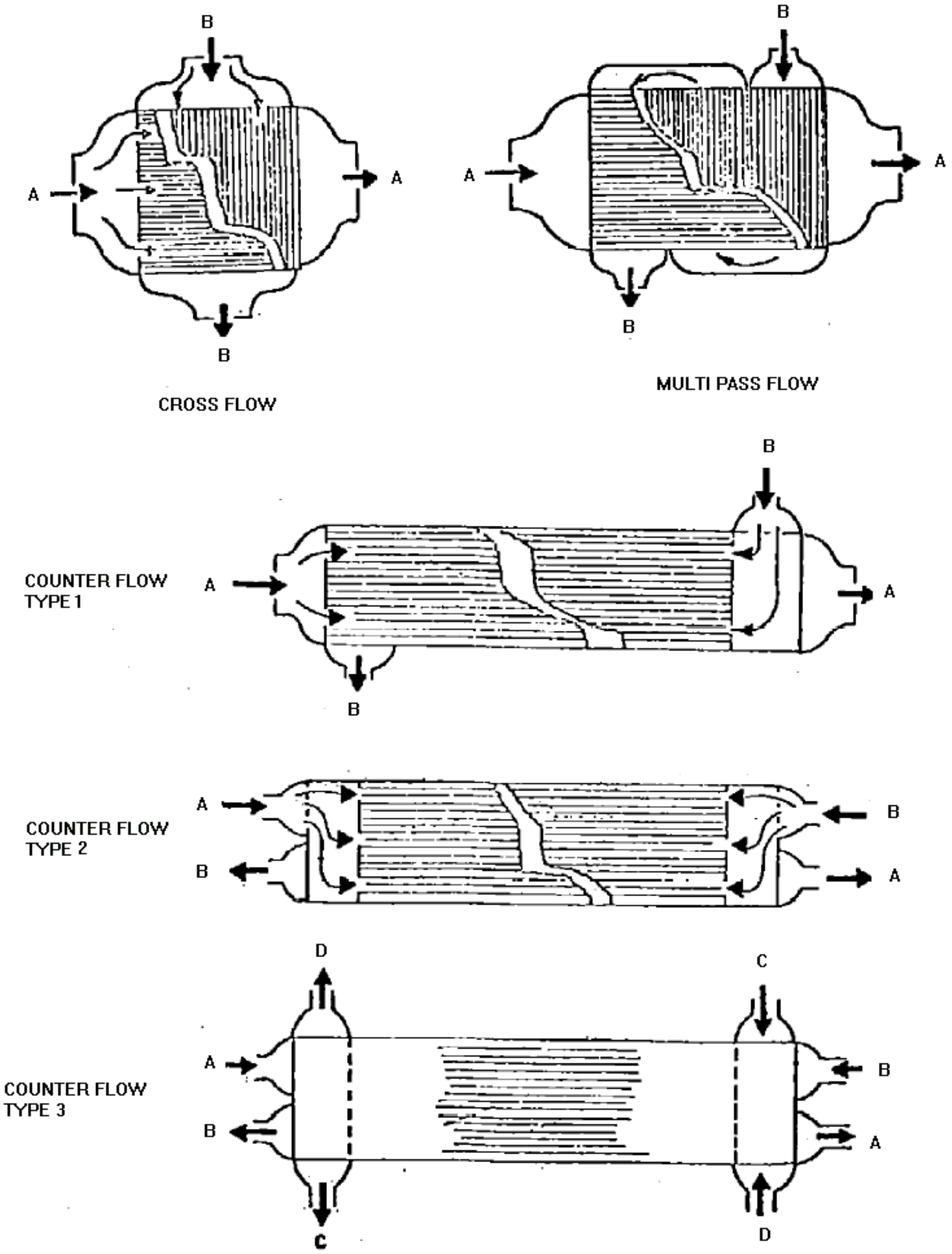


PLATE FIN EXCHANGER FLOW ARRANGEMENTS

Fig. 5

TABLE 2 - GEOMETRICAL RELATIONSHIPS OF PLATE FIN SURFACES

The following are the basic geometry data for a plate fin surface:

b	=	fin height = plate spacing	25.4 mm (inch)
n	=	fin pitch or spacing	fins/25.4 mm (fins/inch)
X_f	=	fin thickness	25.4 mm (inch)
Y	=	fraction of fin perforated (for perforated fins only) .	___

K	=	ratio of actual length to projected length . (for ruffled fins only this fraction is difficult to calculate and can be taken as approximately 1.07 for most ruffled fins)	___

The following items describe an exchanger core:

A_x	=	free flow area of each stream	m² (sq ft)
A_T	=	total heat transfer area of each stream	m² (sq ft)
L	=	effective passage length	m (ft)
W	=	effective passage width	25.4 mm (inch)
X_p	=	plate thickness	25.4 mm (inch)
N	=	number of passages of each stream .	___

From the basic fin dimensions the following geometrical properties can be calculated for a plate fin passage.

A'_x	=	free flow area per passage per 25.4 mm (inch) of passage effective width	m²/mm (sq ft/inch)
A_p	=	primary (plate) surface area per passage	m² (sq ft)
A_p''	=	primary (plate) surface area per passage per 0.3048 m (foot) of length per effective width	25.4 mm (inch) of m²/mm (sq ft/inch)
A_f	=	secondary (fin) surface area per passage	m² (sq ft)
A_f''	=	secondary (fin) surface area per passage per of length per 25.4 mm (inch) of effective width	0.3048 m (foot) m²/mm (sq ft/inch)
A_T''	=	total surface area per passage per 0.3048 m (foot) of length per 25.4 mm (inch) of effective width	m²/m.mm (sq ft/ft inch)
r_h	=	flow passage hydraulic radius, A_xL/A_T	m (ft)
D_h	=	hydraulic diameter = 4 r_h	m (ft)
β	=	ratio of total transfer area on one side of the exchanger to volume between plates on that side	m²/m³ (sq ft/cu ft)

$$\frac{A_f}{A_T} = \frac{A_{f''}}{A_{T''}} \text{ = ratio of fin surface area to total surface are} \quad \textbf{(Eq. 1)}$$

$$A'_x = A'_x = \frac{\mathrm{A_x}}{\mathrm{W.N}} = \frac{(bX_f)(\frac{1}{n} - X_f)^n}{144} \quad \textbf{(Eq. 2)}$$

$$A''_P = \frac{A_P}{W.L} = \left(\frac{1}{n} - X_f\right)\frac{n}{6} \quad \textbf{(Eq. 3)}$$

$$A''_f = \frac{A_f}{W.L} = (b - X_f)\frac{n}{6} \times K(1-Y) \quad \textbf{(Eq. 4)}$$

$$A''_T = \frac{\mathrm{A_T}}{\mathrm{N.W.L}} = A''_P + A''_f \qquad \textbf{(Eq. 5)}$$

$$r_h = r_h = \frac{\mathrm{A_x\,.L}}{\mathrm{A_T}} = \frac{A'_X}{A_T} = \frac{1}{24} \times \frac{(b - X_f)\left(\frac{1}{n} - X_f\right)}{\left(\frac{1}{n} - X_f\right) + (b - X_f)K(1 - Y)} \qquad \textbf{(Eq. 6)}$$

$$\beta = \frac{24\left(\frac{1}{n} - X_f\right) + (b - X_f)K(1 - Y)}{b \times \frac{1}{n}} \qquad \textbf{(Eq. 7)}$$

TABLE 3 - FIN GEOMETRY DATA

TYPE	COMPANY	HEIGHT h mm	(inch)	PITCH η fins/25.4 mm	(fins/inch)	THICK. X_t mm	(inch)	A_x'	A_P'	A_F'	A_T'	A_F/A_T	D_h	APPROX MAX PRESSURE bar (ga).	(psig)
Plain	*	5.080	(0.200)	14	(14)	0.076	(0.003)	(0.001184)	(0.148)	(0.448)	(0.596)	(0.751)	(0.00794)	14	(200)
(Straight)	*	5.080	(0.200)	14	(14)	0.305	(0.012)	(0.001086)	(0.138)	(0.439)	(0.577)	(0.760)	(0.00753)	21	(300)
	*	6.350	(0.250)	10	(10)	0.635	(0.025)	(0.001172)	(0.150)	(0.375)	(0.500)	(0.750)	(0.00937)	49	(700)
		7.112	(0.280)	14	(14)	0.635	(0.016)	(0.001422)	(0.130)	(0.515)	(0.745)	(0.825)	(0.00763)	35	(500)
	*	5.842	(0.230)	15	(15)	0.305	(0.012)	(0.001335)	(0.137)	(0.595)	(0.732)	(0.813)	(0.00741)	3	(300)
	*	9.525	(0.375)	15	(15)	0.203	(0.008)	(0.00224)	(0.147)	(0.917)	(1.064)	(0.862)	(0.00843)	14	(200)
	*	9.525	(0.375)	11.5	(11.5)	0.305	(0.012)	(0.00217)	(0.144)	(0.695)	(0.833)	(0.828)	(0.01038)	21	(300)
	*	9.525	(0.375)	8	(8)	0.635	(0.025)	(0.001944)	(0.134)	(0.466)	(0.600)	(0.778)	(0.01296)	14	(200)
Lanced		6.350	(0.250)	15	(15)	0.305	(0.012)	(0.001355)	(0.137)	(0.595)	(0.732)	(0.813)	(0.00742)	24	(300)
(Serrated)		6.350	(0.250)	14	(14)	0.508	(0.020)	(0.001150)	(0.120)	(0.535)	(0.655)	(0.817)	(0.00700)	49	(700)
		9.525	(0.375)	15	(15)	0.203	(0.008)	(0.00224)	(0.145)	(0.919)	(1.034)	(0.862)	(0.00843)	14	(200)
		7.874	(0.310)	15	(15)	0.152	(0.006)	(0.00192)	(0.152)	(0.760)	(0.912)	(0.833)	(0.00843)	14	(200)
		9.525	(0.375)	10.5	(10.5)	0.305	(0.012)	(0.00220)	(0.146)	(0.635)	(0.781)	(0.813)	(0.01128)	21	(300)
		5.080	(0.200)	Data		Not	Available							49	(700)
		7.112	(0.280)	Data		Not	Available							35	(500)
Ruffled		11.303	(0.445)	18	(18)	0.152	(0.006)	(0.00272)	(0.147)	(1.318)	(1.465)	(0.899)	(0.00744)	14	(200)
		9.525	(0.375)	12	(12)	0.076	(0.008)	(0.00230)	(0.151)	(0.734)	(0.885)	(0.830)	(0.01040)	14	(200)
		10.820	(0.426)	12	(12)	0.152	(0.006)	(0.00271)	(0.155)	(0.840)	(0.995)	(0.939)	(0.01090)	14	(200)

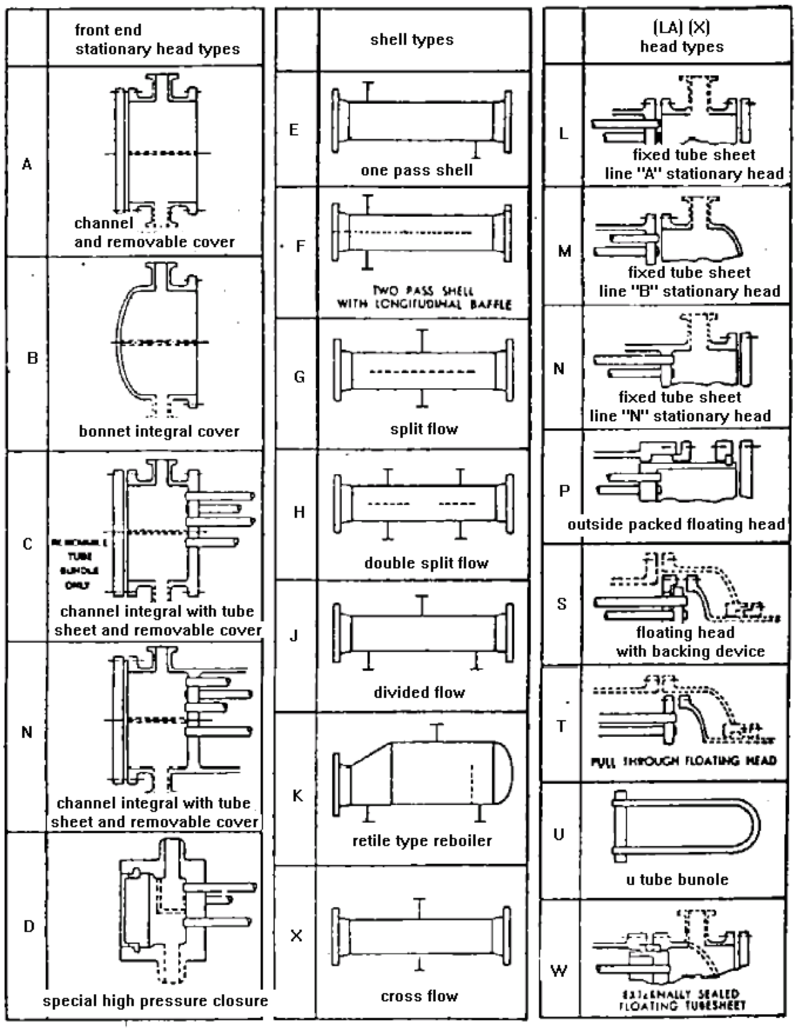
*** = Also available perforated.**

TABLE 4 - MAXIMUM CORE DIMENSIONS

MAX. ASME WORKING PRESS bar (ga), (psig)	MAX. OVERALL WIDTH mm (inch)	MAX. EFFECTIVE WIDTH mm (inch)	MAX. OVERALL HEIGHT mm (inch)	MAX. OVERALL LENGTH mm (inch)	SEPARATOR SHEET THICKNESS mm (inch)
14 (200)	914.4 (36)	885.8 (347/8)	914.4 (36)	762 (30)	0.813 (0.032)
24 (300)	914.4 (36)	885.8 (347/8)	914.4 (36)	3657.6 (144)	1.626 (0.064)
35 (500)	635 (25)	606.4 (237/8)	535 (211/16)	3048 (120)	1.626 (0.064)
49 (700)	457.2 (18)	431.8 (17)	514.4 (20¼)	3048 (120)	1.626 (0.064)
14 (200)	762 (30)	730.3 (28¾)	762 (30)	3168.7(124 ¼)	0.813 (0.032)
31 (450)	660.4 (26)	628.7 (24¾)	762 (30)	3168.7(124¾)	1.270 (0.050)
49 (700)	450 (17¾)	419.1 (16½)	762 (30)	3168.7(124¾)	1.626 (0.064)

APPENDICES

APPENDIX A



TYPE DESIGNATION BY TEMA

Fig. A.1

APPENDIX B

TABLE B.1 - SELECTION GUIDE FOR HEAT EXCHANGER TYPES

TYPE DESIGNATION	SIGNIFICANT FEATURE	APPLICATIONS BEST SUITED	LIMITATIONS	RELATIVE COST IN CARBON STEEL CONSTRUCTION
Fixed Tube Sheet	Both tube sheets fixed to shell	Condensers: liquid-liquid; gas-gas; gas-liquid; cooling and heating, horizontal or vertical, reboiling	Temperature difference at extremes of about 93°C due to differential expansion	1.0
Floating Head or Tube Sheet (Removable and non-removable bundles)	One tube sheet "floats" in shell or with shell, tube bundle may or may not be removable from shell, but back cover can be removed to expose tube ends.	High temperature differentials, above about 93°C extremes; dirty fluids requiring cleaning of inside as well as outside of shell, horizontal or vertical.	Internal gaskets offer danger of leaking. Corrosiveness of fluids on shell side floating parts. Usually confined to horizontal units.	1.28
U-Tube; U-Bundle	Only one tube sheet required. Tubes bent in U-shape. Bundle is removable.	High temperature differentials which might require provision for expansion in fixed tube units. Clean service or easily cleaned conditions on both tube side and shell side. Horizontal or vertical.	Bends must be carefully made or mechanical damage and danger of rupture can result. Tube side velocities can cause erosion of inside of bends. Fluid should be free of suspended particles.	1.08
Kettle	Tube bundle removable as U-type or floating head. Shell enlarged to allow boiling and vapor disengaging.	Boiling fluid on shell side, as refrigerant, or process fluid being vaporized. Chilling or cooling of tube side fluid in refrigerant evaporation on shell side.	For horizontal installation. Physically large for other applications.	1.2 - 1.4
Double Pipe	Each tube has own shell forming annular space for shell side fluid. Usually use externally finned tube.	Relatively small transfer area service, applications. Especially suited for high pressures in tube above 27.6 bar (ga).	Services suitable for finned tube. Piping-up a large number often requires cost and space.	0.8 - 1.4
Pipe Coil	Pipe coil for submersion in coil-box of water or sprayed with water is simplest type of exchanger.	Condensing, or relatively low heat loads on sensible transfer.	Transfer coefficient is low, requires relatively large space if heat load is high.	0.5 - 0.7
Open Tube Sections (Water Cooled)	Tubes require no shell, only end headers, usually long, water sprays over surface, sheds scales on outside tubes by expansion and contraction. Can also be used in water box.	Condensing, relatively low heat loads on sensible transfer.	Transfer coefficient is low, takes up less space than pipe coil.	0.8 - 1.1
Open Tube Sections (Air Cooled) Plain or finned tubes	No shell required, only end headers similar to water units.	Condensing, high level heat transfer.	Transfer coefficient is low, if natural convection circulation, but is improved with forced air flow across tubes.	0.8 - 1.8
Plate and Frame	Composed of metal-formed thin plates separated by gaskets. Compact, easy to clean.	Viscous fluids, corrosive fluids slurries, High heat transfer.	Not well suited for boiling or condensing; limit 177-260°C by gaskets. Used for Liquid-Liquid only; not gas-gas.	0.8 - 1.5
Spiral	Compact, concentric plates; no bypassing, high turbulence.	Cross-flow, condensing, heating	Process corrosion, suspended materials.	0.8 - 1.5
Small-tube Teflon	Chemical resistance of tubes; no tube fouling.	Clean fluids, condensing, cross-exchange.	Low heat transfer coefficient.	2.0 - 4.0

APPENDIX C

TYPICAL TEMA RECOMMEND FOULING RESISTANCES FOR INDUSTRIAL FLUIDS

TABLE C.1 - TYPICAL FOULING RESISTANCES FOR INDUSTRIAL FLUIDS

Oils:	$M^2.°C/W$
Fuel Oil #2	0.00035
Fuel Oil #6	0.00085
Transformer Oil	0.00017
Engine Lube Oil	0.00017
Quench Oil	0.0007
Gases and Vapors:	
Manufactured Gas	0.0017
Engine exhaust Gas	0.0017
Steam (Non-Oil Bearing)	0.000085
Exhaust Steam (Oil Bearing)	0.000255 - 0.00035
Refrigerant Vapors (Oil Bearing)	0.00035
Compressed Air	0.00017
Ammonia Vapor	0.00017
CO2 Vapor	0.00017
Chlorine Vapor	0.00035
Coal Flue Gas	0.0017
Natural Gas Flue Gas	0.00085
Liquids:	
Molten Heat Transfer Salts	0.000088
Refrigerant Liquids	0.00017
Hydraulic Fluid	0.00017
Industrial Organic Heat Transfer Media	0.00035
Ammonia Liquid	0.00017
Ammonia Liquid (Oil Bearing)	0.000528
Calcium Chloride Solutions	0.000528
Sodium Chloride Solutions	0.000528
CO2 Liquid	0.00017
Chlorine Liquid	0.00035
Methanol Solutions	0.00035
Ethanol Solutions	0.00035
Ethylene Glycol Solutions	0.00035

FOULING RESISTANCES FOR CHEMICAL PROCESSING STREAMS

Gases and Vapors:	
Acid Gases	0.00035 - 0.000528
Solvent Vapors	0.00017
Stable Overhead Products	0.00017
Liquids:	
MEA and DEA Solutions	0.00035
DEG and TEG Solutions	0.00035
Stable Side Draw and Bottom Product	0.00017 - 0.00035
Caustic Solutions	0.00035
Vegetable Oils	0.000528

FOULING RESISTANCES FOR NATURAL GAS-GASOLINE PROCESSING STREAMS

Gases and Vapors:	
Natural Gas	0.00017 - 0.00035
Overhead Products	0.00017 - 0.00035
Liquids:	Liquids:
Lean Oil	Lean Oil
Rich Oil	Rich Oil
Natural Gasoline and Liquefied Petroleum Gases	Natural Gasoline and Liquefied Petroleum Gases

(to be continued)

TABLE C.1 (continued)

TYPICAL FOULING RESISTANCES FOR OIL REFINERY STREAMS

Crude and Vacuum Unit Gases and Vapors:						
Atmospheric Tower Overhead Vapors					0.00017	
Light Naphthas					0.00017	
Vacuum Overhead Vapors					0.00035	
Crude and Vacuum Liquids:						
Crude Oil						
	17 to 121°C Velocity (m/s)			121 to 176°C Velocity (m/s)		
	< 0.6	0.6 - 1.2	> 1.2	< 0.6	0.6 - 1.2	> 1.2
Dry	0.00053	0.00035	0.00035	0.00053	0.00035	0.00035
Salt*	0.00053	0.00035	0.00035	0.00088	0.000704	0.000704
0.00053	176 to 232°C Velocity (m/s)			232°C and over Velocity (m/s)		
	< 0.6	0.6 - 1.2	> 1.2	< 0.6	0.6 - 1.2	> 1.2
Dry	0.704	0.00053	0.00053	0.00088	0.000704	0.000704
Salt*	1.056	0.00088	0.00088	0.00123	0.00106	0.00106
* Assumes desalting @ approx. 121°C						
Gasoline					0.00035	
Naphtha and Light Distillates					0.00035 - 0.000528	
Kerosene					0.00035 - 0.000528	
Light Gas Oil					0.00035 - 0.000528	
Heavy Fuel Oils					0.000528 - 0.00088	
Heavy Gas Oil					0.00088 - 0.00123	
Asphalt and Residuum:						
Vacuum Tower Bottoms					0.00017	
Atmosphere Tower Bottoms					0.00123	
Cracking and Coking Unit Streams:						
Overhead Vapors					0.00035	
Light Cycle Oil					0.00035 - 0.000528	
Heavy Cycle Oil					0.000528 - 0.000704	
Light Coker Gas Oil					0.000528 - 0.000704	
Heavy Coker Gas Oil					0.000704 - 0.00088	
Bottoms Slurry Oil (1.35 m/s minimum)					0.000528	
Light Liquid Products					0.00035	
Catalytic Reforming, Hydrocracking and Hydrodesulfurization Streams:						
Reformer Charge					0.000264	
Reformer Effluent					0.000264	
Hydrocracker Charge and Effluent*					0.00035	
Recycle Gas					0.00017	
Hydrodesulfurization Charge and Effluent*					0.00035	
Overhead Vapors					0.00017	
Liquid Product Over 50° API					0.00017	
Liquid Product 30 - 50° API					0.00035	
* Depending on charge, characteristics and storage history, charge resistance may be many times this value.						
Light Ends Processing Streams:						
Overhead Vapors and Gases					0.00017	
Liquid Products					0.00017	
Absorption Oils					0.00035 - 0.000528	
Alkylation Trace Acid Streams					0.00035	
Reboiler Streams					0.00035 - 0.000528	

(to be continued)

TABLE C.1 (continued)

Lube Oil Processing Streams:	0.00035
Feed Stock	0.00035
Solvent Feed Mix	0.00017
Solvent	0.000528
Extract*	0.00017
Raffinate	0.00088
Asphalt	0.000528
Wax Slurries*	0.00017
Refined Lube Oil	
* Precautions must be taken to prevent wax deposition on cold tube walls.	
Visbreaker:	
Overhead Vapor	0.000528
Visbreaker Bottoms	0.0017
Naphtha Hydrotreater:	
Feed	0.000528
Effluent	0.00035
Naphthas	0.00035
Overhead Vapors	0.000204
Catalytic Hydro Desulfurizer:	
Charge Effluent	0.000704 - 0.00088
Effluent	0.00035
HT Sep Overhead	0.00035
Stripper Charge	0.000528
Liquid Products	0.00035
HF Alky Unit:	
Alkylate, Deprop. Bottoms, Main Fract. Overhead Main Fract. Feed	0.000528
All Other Process Streams	0.00035

FOULING RESISTANCES FOR WATER

Temperature of Heating Medium*	Up to 116°C		116 to 204°C	
Temperature of Water	51°C		Over 51°C	
	Water Velocity (m/s)		Water Velocity (m/s)	
	0.9 and Less	Over 0.9	0.9 and Less	Over 0.9
Sea Water	0.000088	0.000088	0.00017	0.00017
Brackish Water	0.00035	0.00017	0.000528	0.00035
Cooling Tower and Artificial Spray Pond:				0.00035
Treated Make Up	0.00017	0.00017	0.00035	0.000704
Untreated	0.000528	0.000528	0.00088	0.00035
City or Well Water	0.00017	0.00017	0.00035	
River Water:				
Minimum	0.00035	0.00017	0.000528	0.00035
Average	0.000528	0.00035	0.000704	0.000528
Muddy or Silty	0.000528	0.00035	0.000704	0.000528
Hard (over 258 g/m³)	0.000528	0.000528	0.00088	0.00088
Engine Jacket	0.00017	0.00017	0.00017	0.00017
Distilled or Closed Cycle				
Condensate	0.000088	0.000088	0.000088	0.000088
Treated Boiler Feedwater	0.00017	0.000088	0.00017	0.00017
Boiler Blowdown	0.00035	0.00035	0.00035	0.00035
* If the heating medium temperature is over 204°C and the cooling medium is known to scale, these ratings should be modified accordingly.				