

DEP SPECIFICATION

SHELL AND TUBE HEAT EXCHANGERS; SELECTION AND APPLICATION

DEP 31.21.01.12-Gen.

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DESIGN AND ENGINEERING PRACTICE

DEM1



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PART I INTRODUCTION

1.1 SCOPE

This new DEP specifies requirements and gives recommendations for shell-and-tube heat exchangers in the refining, chemical, petro-chemical, and natural gas processing industry, for which ISO 16812 and DEP 31.21.01.30-Gen. are to be applied. This DEP is not intended to be used for small heat exchangers [i.e., having a bare tube surface area of less than 0.5 m² (5.4 ft²)] or for pressurized water-cooled (or glycol-cooled) seal-oil coolers, lube-oil coolers, jacket water coolers for rotating equipment, vacuum-operated steam surface condensers, feed water heaters, or alkylation contactor bundles of proprietary design.

This DEP was created to specify the thermal and mechanical design considerations for shell and tube heat exchangers and correspondingly assist the design team/designer of either the Principal or the Contractor. Specifications are provided to direct selection/design of cost effective heat exchangers consistent with the service conditions, intended design life, and criticality of the application.

This DEP premises user familiarity with DEP 31.21.01.30-Gen. and ISO 16812, which are primarily intended for use by the Manufacturer/Supplier.

This DEP covers the general specification requirements of the shell and tube heat exchanger data/requisition sheet with design notes and the general information typically required for a bid requisition. This DEP, therefore, is not normally intended for shell and tube heat exchanger Manufacturers/Suppliers.

This DEP contains mandatory requirements to mitigate process safety risks in accordance with Design Engineering Manual DEM 1 – Application of Technical Standards.

1.2 DISTRIBUTION, INTENDED USE AND REGULATORY CONSIDERATIONS

Unless otherwise authorised by Shell GSI, the distribution of this DEP is confined to Shell companies and, where necessary, to Contractors and Manufacturers/Suppliers nominated by them. Any authorized access to DEPs does not for that reason constitute an authorization to any documents, data or information to which the DEPs may refer.

This DEP is intended for use in facilities related to oil and gas production, gas handling, oil refining, chemical processing, gasification, distribution and supply/marketing. This DEP may also be applied in other similar facilities.

When DEPs are applied, a Management of Change (MOC) process shall be implemented; this is of particular importance when existing facilities are to be modified.

If national and/or local regulations exist in which some of the requirements could be more stringent than in this DEP, the Contractor shall determine by careful scrutiny which of the requirements are the more stringent and which combination of requirements will be acceptable with regards to the safety, environmental, economic and legal aspects. In all cases, the Contractor shall inform the Principal of any deviation from the requirements of this DEP which is considered to be necessary in order to comply with national and/or local regulations. The Principal may then negotiate with the Authorities concerned, the objective being to obtain agreement to follow this DEP as closely as possible.

1.3 DEFINITIONS

1.3.1 General definitions

The **Contractor** is the party that carries out all or part of the design, engineering, procurement, construction, commissioning or management of a project or operation of a facility. The Principal may undertake all or part of the duties of the Contractor.

The **Manufacturer/Supplier** is the party that manufactures or supplies equipment and services to perform the duties specified by the Contractor.

The **Principal** is the party that initiates the project and ultimately pays for it. The Principal may also include an agent or consultant authorised to act for, and on behalf of, the Principal.

The word **shall** indicates a requirement.

The capitalised term **SHALL [PS]** indicates a process safety requirement.

The word **should** indicates a recommendation.

1.3.2 Specific definitions

Term	Definition
BDEP	Basic Design Engineering Package
HIC	Hydrogen Induced Cracking
HTFS	Heat Transfer and Fluid Flow Services software new name is Aspen+
HTRI	Heat Transfer Research Inc.
MTD	Mean Temperature Difference
NTIW	No Tubes-In-Window
OD	Outside Diameter
SCOT	Shell Claus Offgas Treatment
SOHIC	Stress-Oriented Hydrogen Induced Cracking
SSC	Sulphide Stress Cracking
STHE	Shell and Tube Heat Exchanger
TEMA	Tubular Exchangers Manufactured Association

1.4 CROSS-REFERENCES

Where cross-references to other parts of this DEP are made, the referenced section number is shown in brackets (). Other documents referenced by this DEP are listed in (Part V).

1.5 COMMENTS ON THIS DEP

Comments on this DEP may be sent to the Administrator at standards@shell.com, using the DEP Feedback Form. The DEP Feedback Form can be found on the main page of "DEPs on the Web", available through the Global Technical Standards web portal <http://swwww.shell.com/standards> and on the main page of the DEPs DVD-ROM.

1.6 DUAL UNITS

This DEP contains both the International System (SI) units, as well as the corresponding US Customary (USC) units, which are given following the SI units in brackets. When agreed by the Principal, the indicated USC values/units may be used.

PART II THERMAL AND HYDRAULIC DESIGN AND SPECIFICATION

2.1 INTRODUCTION

1. This document is intended primarily for use by the Principal and Contractor to complete the thermal design, data/requisition sheet (DEP 31.21.00.93-Gen., or equivalent), and supplementary technical requirements necessary for a complete set of design, materials, fabrication, inspection and testing instructions to the Manufacturer/Supplier.

The basic expectation is that the Contractor will provide the complete process, thermal and mechanical design definition for the application, with the Manufacturer/Supplier providing the thermal and mechanical guarantees. When approved by the Principal, the thermal design may be delegated to the Manufacturer/Supplier. Thermal design of hairpin-style heat exchangers and shell and tube exchangers for auxiliary services used in packaged for rotating equipment (e.g., seal oil, lube oil coolers) may be performed by the Manufacturer/Supplier. In these cases, the Contractor shall validate the design through an independent check of the Manufacturer's/Supplier's proposed thermal design.

2. The initial thermal/hydraulic performance design or rating of a shell and tube heat exchanger will normally be performed during the project design phase, e.g., Project Specification, Basic Design Engineering Package (BDEP) etc., by the Contractor. The responsibility for the final thermal/hydraulic design and guarantee shall be provided by the exchanger Manufacturer/Supplier regardless of the party that completes the initial design, unless otherwise agreed by the Principal.
3. A data/requisition sheet (DEP 31.21.00.93-Gen. or the Contractor's equivalent) shall be completed.
4. Items listed in the Shell and Tube Heat Exchanger (STHE) Checklist (see Annex B, ISO 16812) are covered in DEP 31.21.01.30-Gen. except for drawing requirements. The Contractor shall specify the number of copies of drawings, data, etc., required for the project.
5. When providing specifications for shell-and-tube heat exchangers and filling in data/requisition sheets, the items in ISO 16812 marked with a bullet (●) shall be provided by the Contractor; Standard Form DEP 31.21.01.83-Gen. may be utilized for this purpose.
6. Decisions related to the design class performance categories for shell-and-tube heat exchangers are intended to be made by the Principal in the project "Define" phase and detailed in project scope defining documentation e.g., equipment BDEP, shell-and-tube heat exchanger data/requisition sheets, shell-and-tube heat exchanger checklist, project specific specifications, etc. This DEP is generally written for design class 2 / 3 facilities, with options identified based on design class where appropriate. When projects opt not to implement the design class practice, options set forth within this DEP based on design class shall default to design class 3 selection unless otherwise directed by the Principal.
7. Equipment Numbering

Heat transfer equipment shall be identified by a specific equipment number in accordance with DEP 31.10.03.10-Gen.

Equipment consisting of two or more heat exchangers connected in series or parallel and performing a common duty is termed a unit. In such cases, the unit as a whole is identified by the specific equipment number, and each heat exchanger within the unit is separately identified by the addition of a capital letter in alphabetical progression, e.g., E-101-A, E-101-B, E-101-C, etc.

2.2 PROCESS SPECIFICATION

1. Stream physical properties and data shall be provided as specified in the following paragraphs to allow design and analysis using the Principal's preferred methods. Transport properties for well-known components such as air or water do not need to be specified.
2. For single phase cooling, transport properties (viscosity, density, thermal conductivity, specific heat) shall be provided at the inlet and exit conditions. For cooling of liquid streams with a viscosity greater than 10 cP at the specified outlet temperature, a viscosity curve with intermediate temperatures shall be provided, spanning from the hot side inlet temperature down to a temperature equal to the cold side fluid inlet temperature, so that the viscous boundary layer at the tube wall can be properly characterized.
3. For condensing and boiling streams, transport properties shall be provided for each phase at the inlet, outlet, and several intermediate temperature points based on equal enthalpy intervals, such that purely-interpolation-based errors do not exceed 5%. The transport properties shall be based on the total composition of each phase including water, air and/or steam when these components are part of a homogeneous phase. Liquid properties at the dew point and vapour properties at the bubble point should be included when such points occur within the heat exchanger. If water is present, the water dew-point at the operating pressure shall also be specified for condensing services. Additionally, plots of heat release (or total stream enthalpy), weight fraction vapour, percent (by weight) of the second liquid phase (if present) and vapour molecular weight versus temperature at inlet pressure and the inlet pressure minus the allowable pressure drop shall be provided. The above information shall be calculated under the assumption of thermodynamic equilibrium.
4. The physical properties of the fluid and the mechanical requirements/limitations shall be specified on data/requisition sheet DEP 31.21.00.93-Gen., sheets 1, 2 and 3. If the data provided is considered to be insufficient for design purposes, it is the responsibility of the heat exchanger designer to request further information from the process engineering specialist.
5. If condensate has immiscible components, liquid properties are to be specified separately and completely for each liquid phase, along with the 'mixed phase' liquid properties.
6. Special attention shall be paid to liquid phase properties of systems comprised of both a heavy oil and a water phase which may form an emulsion, with particular attention to the apparent and/or emulsion viscosity.
7. Process fluids which exhibit non-Newtonian behaviour shall be identified. The process engineering specialist and heat exchanger designer shall discuss and agree upon the property data to be provided for any such cases.
8. If liquid disengages from vapour in an amount sufficient to alter vapour-liquid equilibrium implicit in (2.2 para. 3) above, the heat release profiles and vapour/liquid transport properties shall be generated based on a differential condensation method, where the liquid and vapour phases are considered to separate as formed and do not remain in thermodynamic equilibrium, when requested by the heat exchanger design engineer and should be considered in the thermal and hydraulic performance design.
9. For boiling streams, surface tension of the liquid phase, the boiling range (i.e., dew point temperature minus bubble point temperature at constant pressure), the mole fraction average critical pressure and critical temperature of the liquid, and the number of components (present in at least 5% by mole) in the reboiler feed shall be specified.
10. For condensing streams, the surface tension of the liquid phase shall be provided, along with the mole fraction of non-condensable components at the exchanger's inlet conditions when these are present in an amount greater to or equal to 5%.

11. For condensing or boiling (non-thermosiphon applications) at operating pressures greater than 2,100 kPag (300 psig), a single heat release curve may be provided with the pressure varying linearly from the inlet to outlet condition, assuming that all of the allowable pressure drop will be utilized.
12. For condensing or boiling (non-thermosiphon applications) at operating pressures less than or equal to 2,100 kPag (300 psig), at least two isobaric heat curves shall be included on the data sheet and in the thermal design simulation, with one at the inlet pressure and one at the expected outlet pressure. For exchangers operating under vacuum conditions, a minimum of three isobaric heat curves shall be provided at the inlet pressure, expected outlet pressure, and at the average pressure between the inlet and outlet.
13. For thermosiphon reboilers, multiple isobaric heat curves shall be provided in order to properly estimate the onset of boiling for the feed to the reboiler, which is sub-cooled due to the static head introduced by the elevation difference between the column sump and the reboiler. Three (3) isobaric heat curves shall be provided: (a) at the column sump pressure; (b) at the column sump pressure plus 3.75 m (12 ft) of liquid head; and, (c) at the column sump pressure plus 7.5 m (24 ft) of liquid head. If additional heating curves at other isobaric pressures are determined to be necessary for the design of the exchanger, the heat exchanger designer should obtain this information from the process engineer.
14. When actual hydraulic information is available (static head and piping arrangements), the process curves should be provided at the sump pressure, a pressure equal to the actual head and at a pressure midway between the sump and actual head value.
15. The nature, size (e.g., microns) and quantity (in wt% or ppm) of solids in the stream such as silt, sand, fines, catalyst fines, etc., shall be defined on the equipment data sheet when applicable. This is particularly important for design of pressure equipment in upstream environments (e.g., production coolers, tailings, slurry services in mining, froth treatment facilities) and catalytic cracking units to prevent plugging and/or erosion. Where the size distribution and shape of the solids are known, this information shall also be provided on the data sheet.
16. The quantity of free water (wt% or ppm) shall be defined on the equipment data sheet as well as any other compounds (e.g., ammonia, H₂S, chlorides, etc.) which when combined with water may form a corrosive environment. The Contractor SHALL [PS] indicate if the service is in sour or wet H₂S service and shall clearly identify the sulphide stress cracking (SSC), hydrogen blistering, hydrogen induced cracking (HIC), stress-oriented hydrogen induced cracking (SOHIC), or carbonate cracking severity ratings on the equipment datasheet, as determined in accordance with DEP 30.10.02.15-Gen. or DEP 30.10.02.17-Gen., as may be applicable.
17. Any limitations with respect to the tube wall or bulk fluid temperatures shall be specified on the process data sheet. This may include a thermal degradation/coking point, the freezing point, the pour point, the wax point, the dew point (if condensation may result in increased corrosion), the hydrate formation point, or any other temperature which can create operating difficulties for a heat exchanger. All that apply shall be specified.
18. The Contractor SHALL [PS] indicate any exchangers subject to hydrogen service on the heat exchanger data sheet along with the hydrogen partial pressure. Hydrogen service is defined as any service in which the hydrogen partial pressure is greater than 700 kPa absolute (100 psia) at any temperature.
19. The Contractor SHALL [PS] indicate any exchangers subject to very toxic service on the heat exchanger data sheet. Very toxic substances are those which are as specified in DEP 01.00.01.30-Gen.
20. Data sheets and curve-specified data shall represent the actual design conditions. The heat duty specified should be consistent with the flows and temperatures specified. A heat duty margin alone should not be specified without also specifying the flow rate and/or terminal temperature conditions that are consistent with that heat duty.

21. All expected operating scenarios/conditions shall constitute separate alternative design cases and be identified by separate data sheets. Predictable upset conditions and full range of flows, including turn-down, shall also be indicated.
22. Alternate modes of operation such as e.g., steam-out, etc. SHALL [PS] also be defined on the data sheet. This information is needed to ensure an adequate mechanical design, including the specification of expansion joint requirements.
23. Abnormal flows and/or temperature conditions that are expected to impose either additional duty requirements or require additional heat transfer surface area or result in a higher pressure drop shall be specified. An alternate operating case shall not be dismissed from analytical review simply because it is a lower heat duty case.
24. Fouling resistances or design margins should be specified based either on DEP 20.21.00.31-Gen., facility experience for a specific service, project specific information or licensor requirements. When no other data or experience is available, TEMA recommended values for the most similar fluid may be used.
25. Where cooling water systems are such that significantly lower water temperatures may be experienced during winter operation (e.g., once-through cooling water systems), the lower water temperature shall be identified on the data/requisition sheet (i.e., through specification of separate summer and winter cases) and the exchanger evaluated to ensure that operating or control problems will not exist during winter mode of operation (e.g., low velocity and high fouling potential due to excessive flow through a control bypass line). The Principal should be consulted to ensure proper operation and control can be maintained.
26. Heat exchangers should not be specified to have or require "excess surface" for a process margin. If excess flow, duty or heat exchanger capability is required, the required temperatures, flows and duty for the condition shall be specified on the datasheet accordingly to ensure the heat exchanger is designed appropriately.
27. Where a bypass for temperature control is used, it should be applied to the side of the exchanger which exhibits the lower fouling tendency, wherever possible. Bypassing high fouling fluids for temperature control, leads to lower velocities and potential for increased fouling rates. The process design specification for the heat exchanger shall consider a minimum of 10% of the heat and material balance flow rate to be bypassed for the bypassed stream.

2.3 GENERAL THERMAL DESIGN

1. Unless otherwise agreed with the Principal, the initial thermal design for shell and tube exchangers shall be performed by the Contractor. When agreed and delegated to the Manufacturer/Supplier, the Contractor shall check rate the Manufacturer's/Supplier's thermal designs during evaluation of the Manufacturer's/Supplier's bids. The Manufacturer's/Supplier's design shall be adjusted to meet the specification requirements. The thermal and mechanical performance guarantee shall be provided by the Manufacturer/Supplier, regardless of the party that completes the initial thermal design.
2. Validation of thermal and hydraulic design of shell-and-tube and multi-tube hairpin heat exchangers shall be based on design methods which have been proven in practice. In this respect, the design procedures and computer programs published by Heat Transfer Research Inc. (HTRI) or Aspen+/UNISIM (the former Heat Transfer and Fluid Flow Services (HTFS) software) shall be used. Discrepancies in the performance prediction results between different design methods shall be referred to the Principal's heat transfer specialist for resolution. The applicable versions of such computer programs used for thermal design shall be agreed upon between the Contractor and the Principal during the initial stage of the project.
3. In the case of a proprietary design, the calculation method and the degree of responsibility shall be agreed between the Manufacturer/Supplier, Contractor, and Principal.

4. Heat exchanger thermal rating files (including the native electronic input files) reflecting the as-built design shall be turned over to the Principal at the end of the project, using the same version of the program agreed with the Principal at the beginning of the project. The printout of the computer runs shall correspond to the input data. No editing of the final results/ print out is allowed to incorporate any changes or deviations after the computer run has been made.
5. The heat exchanger design shall most economically satisfy the process performance requirements (heat duty and allowable pressure drop) stipulated on the process data sheets for any defined operating case while meeting project requirements and expectations for reliability and maintainability. In general, the most economical design will provide for the lowest surface area and/or smallest diameter shell within the prescribed length restriction and the fewest number of shells. The thermal/hydraulic design shall be optimized for the controlling case(s), noting that one operating case may control heat transfer performance while another operating case may control from a pressure drop consideration. Any controlling case shall have its thermal/hydraulic performance completely specified on the mechanical data sheet. Non--controlling cases should also be check-rated and specified with their respective differential fouling resistances.
6. In general, heat exchangers shall be provided with effective surface areas between 100% and 105% of the required surface area for the design performance condition, including the prescribed allotment for fouling.

NOTE: The design performance condition may already require a process margin for example on flow rates and heat duty.

Additional margin may be required for special service considerations when specified by the Principal.

7. Unless otherwise specified by the Principal, TEMA Class R shall be specified for all exchangers. TEMA Class 'R' shall apply even if TEMA class 'B' has been specified (e.g., as sometimes occurs with process licensors).
8. Heat exchangers should normally be of the horizontal configuration; however, the vertical arrangement may be considered for certain process requirements, or if cleaning and other maintenance will be infrequent or if there is limited space available.
9. The Contractor's design shall satisfy all restrictions on bundle size (diameter, length, weight) and any other special provisions for bundle maintenance based on the bundle pulling, handling, or cleaning equipment used at the operating site, considering applicable economic considerations for exchanger foundations, structures, and required maintenance access.
10. Where a tube is specified or purchased to a minimum tube wall thickness specification, the tube wall thickness used for thermal and hydraulic calculation purposes shall be 1.1 times the minimum tube wall thickness. This is required to more accurately reflect the furnished thickness supplied by the tube mill.
11. The minimum velocities for single phase non-erosive liquid streams, excluding water, shall be as specified in Table 1. For erosive streams, the Principal's heat transfer specialist shall be consulted.

Table 1 Minimum velocities

Fouling Characterization	Fouling Factor $\text{m}^2\text{-K/W}$ ($\text{hr-ft}^2\text{-}^\circ\text{F/Btu}$)	Tube Side Velocity; m/s (ft/s)	Shell Side Velocity; m/s (ft/s)
Very Low	≤ 0.00009 (0.0005)	1.0 (3.0)	0.3 (1.0)
Low	≤ 0.00026 (0.0015)	1.0 (3.0)	0.3 (1.0)
Medium	≤ 0.00052 (0.003)	1.5 (5.0)	0.5 (1.6)
High	≤ 0.0012 (0.0068)	2.0 (6.5)	0.6 (2.0)
Severe	> 0.0012 (0.0068)	2.2 (7.2)	0.7 (2.3)

12. Velocity should always be maximized for the available pressure drop to provide the highest heat transfer coefficient and to reduce fouling tendency. For shell side streams, the velocity should be maximized to the extent practical while maintaining the velocity criteria and flow fraction requirements as provided in (2.7 para. 49) and (2.7 para. 50) and avoiding vibration issues.
 13. Refer to (2.19) and DEP 20.21.00.31-Gen. for cooling water velocity requirements.
 14. The maximum in-tube velocity for lean Amine and Sulfinol streams shall be 2.4 m/s (8 ft/s). The maximum in-tube velocity for rich Amine and Sulfinol streams shall be 1.5 m/s (5 ft/s) for carbon steel materials and 2.4 m/s (8 ft/s) for 316L stainless steel materials.
 15. Consideration shall be given to the need for special devices to prevent erosion of the tube ends for saturated vapours and two-phase fluids, if the velocity head (ρV^2) in the tubes exceeds $8,930 \text{ kg/m-s}^2$ ($6,000 \text{ lb/ft-s}^2$). For non-erosive liquids, the maximum allowable velocity head shall be calculated using the equivalent maximum water velocities as specified in DEP 20.21.00.31-Gen. for the different tube materials. For dry non-erosive gasses, no specific tube velocity limits apply.
- NOTE: In the above calculation of velocity head, V is the linear velocity in m/s (ft/s) and ρ is the fluid density in kg/m^3 (lb/ft^3). For two-phase fluids, the homogeneous velocity should be used.
16. The allowable pressure drop specified by the process engineer on the heat exchanger's process data sheet shall be applied as the maximum that may be consumed in the exchanger's "fouled" condition. For vapour and condensing streams, the allowable pressure drop shall be determined by the process engineer based on process and pressure profile requirements. For pumped liquid streams the allowable pressure drop shall be sufficient to achieve target design velocities in Table 1 to ensure optimum exchanger design, sizing and fouling minimization. Allowable pressure drops per heat exchanger for pumped liquids should not be less than the guideline values provided in Tables 2 and 3. The heat exchanger designer shall communicate to the Principal any opportunity to reduce the size or cost of a design by >10% with more allowable pressure drop than specified in Table 2 or Table 3, as the specifics for certain cases may extend beyond the assumption premises of the table.

Table 2 Allowable tube side pressure drop, kPa (psi)^{1, 2, 5}

Fouling Characterization	Fouling Factor $\text{m}^2\text{-K/W (hr-ft}^2\text{-}^\circ\text{F/Btu)}$	Liquid Viscosity at the Process Outlet Temperature		
		$\leq 1 \text{ cP}$	$\leq 2 \text{ cP}$	$\leq 5 \text{ cP}^3$
Very Low	$\leq 0.00009 (0.0005)$	75 (11)	110 (16)	165 (24)
Low	$\leq 0.00026 (0.0015)$	75 (11)	110 (16)	165 (24)
Medium	$\leq 0.00052 (0.003)$	120 (17.5)	150 (22)	210 (30.5)
High	$\leq 0.0012 (0.0068)$	200 (30)	225 (32.5)	300 (43.5)
Severe	$> 0.0012 (0.0068)$	275 (40)	315 (45)	350 (50)

Table 3 Allowable shell side pressure drop, kPa (psi)^{1, 2, 5}

Fouling Characterization	Fouling Factor $\text{m}^2\text{-K/W (hr-ft}^2\text{-}^\circ\text{F/Btu)}$	Liquid Viscosity at the Process Outlet Temperature			
		$\leq 5 \text{ cP}$	$\leq 10 \text{ cP}$	$\leq 25 \text{ cP}$	$\leq 50 \text{ cP}^4$
Very Low	$\leq 0.00009 (0.0005)$	35 (5)	35 (5)	50 (7.5)	75 (11)
Low	$\leq 0.00026 (0.0015)$	35 (5)	50 (7.5)	75 (11)	100 (15)
Medium	$\leq 0.00052 (0.003)$	50 (7.5)	75 (11)	100 (15)	125 (18)
High	$\leq 0.0012 (0.0068)$	75 (11)	100 (15)	125 (18)	150 (22)
Severe	$> 0.0012 (0.0068)$	100 (15)	125 (18)	150 (22)	175 (25)

NOTES FOR TABLE 2 AND TABLE 3:

1. Values are estimated based on experience to allow the required minimum velocities to be achieved and to provide an optimized heat exchanger design. The values shown are per heat exchanger when multiple exchangers in series are used. When the allowed pressure drops are insufficient to achieve the target velocities, the Principal's heat transfer specialist shall be consulted.
 2. The appropriate values from the tables above shall be included on the process datasheet for heat exchanger thermal design. Values shall be specified as the allowable pressure drop.
 3. For tube side fluids with liquid viscosity $> 5 \text{ cP}$, consult the Principal's heat transfer specialist for further review. In such instance, the pressure drop will be as required to achieve the minimum velocity, and to avoid laminar and/or transitional flow where possible.
 4. For shell side fluids with liquid viscosity $> 50 \text{ cP}$, consult the Principal's heat transfer specialist. The pressure drop will be as required to achieve the minimum velocity, and to avoid laminar flow, where possible.
 5. Values are estimated based for a single shell. Where multiple shells in series are required, either to achieve the required shell side velocity, or because of a hot and cold side outlet temperature cross-over, these values should be increased accordingly.
17. Pressure drop in the "fouled" condition shall be estimated for use in hydraulic system calculations, including pump sizing. Since the thickness and surface roughness of fouling deposits are not usually known, it is necessary to estimate the increase in pressure drop above a "clean" condition due to fouling, as it cannot be accurately calculated by simulation.

The pressure drop for the fouled condition shall be estimated by multiplying the calculated pressure drop for the clean condition by the appropriate factor from Table 4. Both the "clean" and "fouled" pressure drops shall be shown on the heat exchanger data sheet.

The “fouled” pressure drop shall be less than the allowable pressure drop. The “fouled” pressure drop may be used in the final hydraulic system calculations.

Table 4 Hydraulic multipliers for fouling

Fouling Characterization	Fouling Factor $\text{m}^2\text{-K/W}$ (hr-ft²- °F/Btu)	Pressure Drop Multiplier
Very Low	≤ 0.00009 (0.0005)	1.1
Low	≤ 0.00026 (0.0015)	1.1
Medium	≤ 0.00052 (0.003)	1.2
High	≤ 0.0012 (0.0068)	1.3
Severe	> 0.0012 (0.0068)	1.4

NOTE: For cooling waters other than closed loop cooling waters, a hydraulic multiplier of 1.5 shall be applied. For closed loop cooling waters, a hydraulic multiplier of 1.1 shall be applied.

18. The calculated pressure drop shall include the effect of any special distribution devices and/or tube inserts or ferrules that may be present. Where tube inserts are present, the pressure drop shall consider the actual inside diameter of the insert.
19. To provide a heat exchanger with the minimum area requirement and lowest cost, the allowable pressure drop on the side with the lowest heat transfer film coefficient should be fully utilized, where possible.
20. Through the course of detailed design, whenever the thermal resistance due to fouling exceeds 50% of the overall thermal resistance, the Contractor, in consultation with the Principal's heat transfer specialist, should further evaluate the design for a reduction in area. Large amounts of excess area for fouling margins are discouraged as this can lead to poor exchanger design, control problems and an increase in fouling tendency.
21. Except for kettle type vapour generators, geometries that promote liquid/vapour separation shall be avoided but may be allowed in some instances for horizontal shell side condensing services, when approved by the Principal.
22. For horizontal shell side condensers, liquid/vapour separation is generally not as critical as for boiling heat transfer. Consideration should be given, however, for the need to prevent the gross separation of liquid and vapour phases or otherwise incorporate a differential condensing curve when all of the following conditions are present:
 - The condensing range exceeds 70°C (125°F);
 - The temperature approach at the process outlet is less than 8°C (15°F);
 - Greater than 22°C (40°F) of temperature range occurs in a gravity flow zone.
23. When the tube side fluid in a horizontal heat exchanger is either a multi-component condensing, boiling, or two-phase service, heat exchanger designs shall utilize either a single tube pass or two-pass U-tube configurations, unless it may be assured that the flow regime in pass turnaround and entry zones will be shear dominated for any operating condition.
24. The Contractor shall review the thermal design in the clean case, for normal and alternate defined modes of operation, including turndown, to ensure problems will not occur with these modes of operation. Performance shall be evaluated for problems such as film boiling, increased fouling potential due to low flows, excessive tube wall temperatures, vibration, water hammer, etc., with consideration given to the type of process control employed. If a problem exists, a revision to the design or method of control may be required. Some typical problems that can occur are as follows.

- a. When throttling or bypassing flow, low velocity and fouling can occur. A larger temperature difference will also occur across the shell or channel when controlled in this fashion, and this can be detrimental (especially for F or G shells; see (2.5 para. 5)) due to the higher or lower outlet temperature than specified on the data/requisition sheet;
 - b. For steam-heated heat exchangers with a level control valve at the condensate outlet, inconsistent process control and/or water hammer can occur, particularly when the amount of surface area needed for the clean condition is less than approximately 25% of the available surface;
 - c. For steam-heated heat exchangers with control valves at steam inlet, vibration (for shell side steam allocation) or erosion (for either shell or tube side steam allocation) may occur in the clean case, due to a decreased steam pressure, and associated increase in steam velocity;
 - d. Heat exchangers in series, particularly those with cascaded materials specifications, may experience temperatures or pressures beyond the design conditions. For example, high temperature hydrogen services (e.g., hydrotreater feed effluent) shall ensure that materials are appropriate (HTHA) at the most severe expected operating conditions.
25. When calculating skin temperature or tube wall temperature, all applicable cases that are pertinent, including alternate operating cases, shall be considered. The worst case operating condition considering clean operation, fouled operation, and operation of either side fouled with the other side clean shall be taken into account.
 26. The skin temperature shall also be below the coking temperature of hydrocarbons, where possible; generally conservatively considered to be 315°C (600°F).
 27. When the sub-cooling duty constitutes 10% or more of the overall duty for the condensing stream, a separate exchanger is recommended to handle the sub-cooling duty when using a horizontal shell and tube exchanger. For vertical heat exchanger arrangements, there are no limitations on sub-cooling duties with proper design considerations.
 28. For sub-cooling duties up to 10% of the overall duty, the sub-cooling duty may be accomplished in a single shell. Consideration shall be given to use of loop seal piping, dam baffles, or process controls to effect the flooding of heat transfer surface for sub-cooling.
 29. Special attention should be given to services which require 100% vaporization with in-tube boiling whenever either the superheat duty required is less than 20% of the total heat duty for the exchanger or when the superheated outlet temperature is less than 50°C (90°F) above the dew point temperature. Considerations may include the use of tube inserts that promote wetted wall flow regimes at high vapour qualities and mist flow conditions.
 30. Where condensation occurs when two immiscible liquid phases exist, liquid mixture properties may be used for the thermal and hydraulic design calculations. When the aqueous phase comprises less than 20% of the total liquid by mass, the thermal conductivity of the condensate film shall be based on the hydrocarbon phase only. When the aqueous phase comprises at least 20% of the total liquid by mass, the mixed phase liquid thermal conductivity may be used.
 31. Line pipe dimensions shall be used for carbon and low alloy steel shells up to a nominal shell diameter of DN 600 mm (NPS 24), and up to a nominal shell diameter of DN 300 mm (NPS 12) for high alloys. For shells to be fabricated from rolled plate, the nominal diameter is the shell inside diameter.
 32. When line pipe is to be used for the shell cylinder, the heat exchanger designer shall calculate the required wall thickness for pressure plus corrosion allowance (excluding considerations of nozzle reinforcement), to ensure that the shell diameter and outer tube limit specified on the heat exchanger data/requisition sheet can be achieved.

33. When the heat exchanger design requires the installation of tube ferrules at the tube inlets, either for erosion protection of the tube ends, high temperature protection to the tubesheet, or for fluid distribution (e.g., falling film evaporators), the specific requirements concerning ferrule material, dimensions, insulation materials (if any) and installation details, shall be specified by, or otherwise agreed to, by the Principal.

2.4 FLUID ALLOCATION

1. The tube and shell side stream allocation selection shall in general be guided by Table 5.

Table 5 Stream placement

Service	Shell	Tube
Sea or brackish water		X
Cooling tower water		X
Lower allowable pressure drop	X	
Larger flow w/similar properties	X	
High pressure		X
Corrosive fluids		X
Slurry service		X
High viscosity laminar flow	X	
Suspended solids		X
Streams requiring water wash		X
Oxygen service		X

2. Swapping streams may be considered where there is an economic incentive to do so.
3. Traditionally high fouling fluids are placed on the tube side, particularly where mechanical cleaning or highly effective hydroblasting is required. However, where the high fouling stream or stream with suspended solids also has a high viscosity, it is preferable to place that stream on the shell side to reduce the surface area or number of shells required. A higher wall shear stress and U value can be achieved with the higher viscosity stream on the shell side, potentially reducing fouling tendency with a proper design. Attention to baffle design and maintaining low leakage stream flow fractions is critical in these situations.

2.5 SHELL TYPE SELECTION

1. In general, the selection of the TEMA shell type shall be by the thermal designer. Refer to TEMA for definitions of the various TEMA shell types.
2. The shell type for most applications will be an E-type with segmental baffles. Other shell types may be selected as follows:
- Where the shell side pressure drop is a constraint, the divided flow shell Type J, cross-flow shell Type X or double-split flow shell Type H should be considered.
 - G shells may also be considered for slight amounts of temperature cross between the hot and cold fluids that could not otherwise be achieved through the use of an E shell.
 - J, X, H or G type shall be used for horizontal thermosiphons. E shell type thermosiphons may be considered when approved by the Principal.

- d. K type (kettles) shall be selected for pool boiling applications where almost 100% vaporization (typically with 0% to 5% liquid entrainment) is required or where phase separation is required
 - e. F type may be considered where the temperature profile of the process fluids requires two or more heat exchangers in series (temperature crossovers), or to increase the shell side velocity, subject to the restrictions below.
3. For design of new facilities, multiple shell pass designs shall only be applied in services with non-removable bundles or removable bundles that will not require frequent disassembly for mechanical cleaning; i.e., shell side fluid has a very low, low, or medium fouling tendency with fouling resistances $\leq 0.00034 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$).
 4. For retrofit or debottlenecking of existing facilities, multiple shell side pass designs may be considered for removable bundles with high or severe fouling fluids on the shell side, when approved by the Principal.
 5. An F shell (or baffled G or H shells requiring sealing of the longitudinal baffle against the shell) may only be selected in the following cases:
 - The longitudinal baffle is continuously welded to the shell and the maximum allowable shell side pressure drop is less than 100 kpa (15 psi);
 - For removable bundles, the maximum allowable shell side pressure drop shall not exceed 50 kpa (7.5 psi). The gap between the longitudinal baffle and the shell shall be closed with a double-sided, multi-layered clip-on spring set (Kempchen T4 type or Principal approved equivalent). Alternate sealing devices, such as single-sided multiple ply flexible seals may be provided when approved by the Principal;
 - The difference between the shell side inlet and outlet temperature does not exceed 110°C (200°F).
 6. Longitudinal baffles in fixed tubesheet or floating head exchangers shall be welded to the tubesheet. U-tube exchangers with removable longitudinal baffles having no more than two tube side passes shall also have their longitudinal baffles welded to the tubesheet.
 7. U-tube bundles may be specified with non-removable longitudinal baffles when four (4) tube passes are utilized with the U-bends arranged in the horizontal plane to permit installation and removal of the bundle. U-tube bundle tubesheets for this configuration shall include a longitudinal baffle partition groove and gasket on the shell side of the tubesheet. This arrangement shall only be applied in services that will not require frequent disassembly for mechanical cleaning; i.e., the shell side fluid has a very low, low, or medium fouling tendency with fouling resistances $\leq 0.00034 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$).
 8. The designer shall ensure that the pass partition lane width will be adequate for the thickness of the longitudinal baffle and any required skid bars. For un-insulated baffles, the minimum thickness for longitudinal baffles shall be the thickness calculated per TEMA RCB-9-132, using twice the calculated clean shell side pressure drop, plus twice the shell side corrosion allowance or the minimum thickness allowed by TEMA, whichever is greater.
 9. Thermal leakage across the longitudinal baffle shall be considered in the thermal design calculations.
 10. An insulated longitudinal baffle (baffle composed of two plates, separated by spacers with a dead fluid zone between them or a single baffle with an insulation layer properly attached) should be considered when the thermal leakage exceeds 10% of the mean temperature difference. The selection and design of an insulated longitudinal baffle shall be subject to the approval of the Principal.
 11. For two-phase shell side flow, HTRI rating software does not calculate thermal leakage across a longitudinal baffle. When used in two-phase flow, the baffle shall be an

insulated type or the thermal leakage shall be manually calculated and considered in the heat exchanger design.

2.6 FRONT AND REAR HEAD TYPE SELECTION

1. The use of removable tube bundles is generally preferred. Fixed tubesheet heat exchangers shall be used only in services where all of the following conditions are satisfied:
 - Stresses caused by differential expansion between the tubes and the shell shall not exceed the design stress limits for all design operating scenarios. A floating-head type or U-tube type heat exchanger should be selected if flexibility is required to avoid overstressing. The use of a flexible shell element is allowed only with the approval of the Principal.
 - Shell side fluid is designated as very low or low fouling tendency with a fouling resistance $\leq 0.00026 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.0015 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$), not requiring mechanical cleaning or when the shell side foulant can be removed by chemical cleaning, as specified in DEP 20.21.00.31-Gen.
 - Shell side entry is not required for shell/nozzle inspection, maintenance, etc.; e.g., some sour, wet H_2S , or other potentially corrosive or materials degrading fluids may require shell internal inspection techniques and/or associated internal repairs, as equipment sizing, etc. allows.
2. Fixed tubesheet exchangers should not be used when the expected tube bundle life is less than twice the anticipated process unit run lengths between turnarounds. The bundle cage components shall be selected to provide a life no less than the shell life.
3. When considering the acceptability of use for a fixed tubesheet exchanger, the stresses caused by differential expansion during upset shall be considered. Also, alternate operating conditions including start-up, turn down, steam-out, low ambient temperature conditions and other operating scenarios shall be considered.
4. U-tube bundle heat exchangers may be used only in services where at least one of the following conditions are satisfied.
 - The tube side fluid's fouling tendency is not high or severe, with a fouling resistance not exceeding $0.00035 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$).
 - Tube side mechanical cleaning is not required or, if it is required, then mechanical cleaning shall be possible and the tube side fouling deposit is relatively easy to remove via high pressure water lancing. In this respect, tube side mechanical cleaning is considered possible with a minimum nominal tube diameter of 25.4 mm (1 in) with a centre-to-centre distance between the parallel legs of the U-tube is at least 100 mm (4 in), or with 19.05 mm ($\frac{3}{4}$ in) nominal tube diameters with a centre-to-centre distance between the parallel legs of the U-tube is at least 150 mm (6 in).
 - tube-side fouling can be removed by chemical cleaning
5. U-tube bundles should be considered, however, for all high pressure applications typically greater than 6200 kPag (900 psig) when it is economically attractive to do so.
6. Floating-head heat exchangers shall be used in all services for which fixed tubesheet or U-tube bundles are not allowed.
7. Shell diameters exceeding 1525 mm (60 in) for exchangers with removable bundles shall be subject to the approval of the Principal. The Contractor shall indicate the maximum bundle size for each project on the STHE Checklist, DEP 31.21.01.83-Gen.
8. The selection of the TEMA front and rear head type shall be made on a case-by-case basis with the use of Table 6 as a general guide. The term "fouled" means mechanical cleaning access is expected and has a fouling resistance exceeding $0.00035 \text{ m}^2 \text{ }^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$), or fouling margins of 'Medium' and higher.

Table 6 Front and rear head selection guidance

Shell Side	Clean		Fouled	
Tube Side	Clean	Fouled	Clean	Fouled
	B/M or B/U	N/N or A/T (Note)	B/T (Note) or U	A/T (Note)

NOTE: A split ring (S) type floating head may be used in lieu of a pull-through (T) type as noted in (2.6 para. 16).

9. The need for in-situ cleaning, inspection, testing and maintenance activities shall also be considered in the selection process. Site maintenance/cleaning practices will typically determine the decision between A or B type for the front head; e.g., exchanger tube bundles which will only be accessed for cleaning and/or inspection and/or repair after being extracted should normally be specified as having a B type head. Flat cover plates may be prohibitive in larger diameter and/or higher pressure services if the required thickness becomes excessive. The preferred form of head type shall be specified by the Contractor on the STHE Checklist form (DEP 31.21.01.83-Gen.).
10. For water-cooled exchangers where frequent tube side cleaning is anticipated and the tube side design pressure is less than 1035 kPag (150 psig), the front-end stationary head shall be Type A.
11. Rear end head Type M should be used for fixed tubesheet designs; however, for heat exchangers with a Type A front end stationary head and an odd number of tube passes, Type L shall be selected.
12. TEMA Type C designs may be used when it is desirable to eliminate the channel-to-tubesheet gasketed joint and/or improve the reliability of the shell-to-tubesheet gasketed joint by decoupling the tube and shell side loads on the tubesheet girth flange.
13. TEMA Type C designs shall be considered for tube side design pressures greater than 10.3 MPag (1500 psig).
14. TEMA Type C designs shall be used when the temperature gradient across the tube side face of the tubesheet exceeds 140°C (250°F). In such cases, an internally gasketed pass partition box shall be supplied so that the channel cover and bolting operate at a uniform temperature, typically at the lower tube side operating temperature. A TEMA N type may be used for non-removable tube bundles.
15. If the temperature gradient across the thickness of a tubesheet at any location exceeds 140°C (250°F), designs which eliminate a dual-gasketed seal at the tubesheet between the channel flange-to-tubesheet-to shell flange should be considered. When high temperature differences can exist, consideration shall be given to the location of the shell inlet and outlet nozzles along the shell so as to minimize the temperature gradient across the tubesheet.
16. A split ring (S) type floating head may be used in lieu of a pull-through (T) type whenever the main cross-flow (B-stream) flow fraction with a T type is less than 0.5, or when a B-stream flow fraction advantage of 0.1 or greater can be achieved through use of an S type.
17. Rear end head Type T shall be used for a kettle type heat exchanger with floating heads.
18. TEMA S type floating heads should be limited to services with tube side design pressures less than 4100 kPag (600 psig) and shell side design pressures less than 6200 kPag (900 psig).
19. TEMA T type floating heads shall be limited to a design pressure on either side of 6200 kPag (900 psig).

20. TEMA Type D closures shall be considered in high pressure tube side services, where they may be more economical than A, B, C or N types. The Manufacturer's/Supplier's proposed closure shall be reviewed in detail prior to purchase of the exchanger. See (3.9) for further information on high pressure closures.
21. Where single pass floating head style exchangers are to be used, the tailpipe-to-shell cover seal shall be of the "flanged" or "packed" type. Use of the packed-type joint is limited to non toxic services with non-flammable liquids and gases below 150°C (300°F) and 1035 kPag (150 psig) on the shell side. All gaskets in the flanged design shall be accessible from the exterior of the exchanger without removing the tube bundle.
22. Floating head tail-pipe designs shall comply with DEP 31.21.01.30-Gen.

2.7 BAFFLE, TUBE BUNDLE AND TUBE LAYOUT SELECTION AND SPECIFICATION

1. The permissible types of baffles are segmental, double segmental, and the segmental type with No Tubes-in-Window (NTIW) area. The use of other types (EMBaffle®, helical baffles, rod baffle, disk-and-doughnut, triple segmental, etc.) requires approval of the Principal.
2. Baffle cuts for single segmental baffles shall normally be maintained between 15 to 35% of the shell diameter. Larger cuts are especially detrimental in single phase flow. A cut up to 45% of the diameter may be used for vertical cut baffles for pure component condensing services to aid removal of condensate from the shell.
3. Baffle cuts for NTIW baffles shall normally be maintained between 10 to 25% of the shell diameter, but in no case shall the velocity head in the window area exceed $7,440 \text{ kg/m-s}^2$ ($5,000 \text{ lb/ft-s}^2$).
4. Baffle cuts for double segmental baffles shall have equal nominal window flow areas and shall be specified in terms of the number of tube rows between baffle cuts and/or percent cut on an area basis. The minimum number of tube rows overlapped shall be in accordance with the following:
 - two (2) for bundle diameters up to and including 1000 mm (39 in);
 - four (4) for bundle diameters between 1005 and 1525 mm (40 and 60 in); and
 - six (6) for bundle diameters exceeding 1525 mm (60 in).
5. Baffles acting as a dam or weir in condensers to maintain a liquid level for sub-cooling the condensate shall not be used when an external loop seal with a bypass can be applied.
6. A full support baffle shall be provided at the shell inlet/exit nozzles for TEMA shell types G and H and at the central nozzle for TEMA shell type J. The baffle shall be located at the centreline of the nozzle.
7. A full support shall be provided at the centreline of each shell side nozzle in TEMA Type X shells. This full support, as well as any other tube supports in an X shell, shall be cut 3 mm ($\frac{1}{8}$ in) above and below the top and bottom tube rows, respectively.
8. Except when used in TEMA Type G, H, X and J shells, any full support plate near the tangent line of U bends shall have cut outs to allow nominal fluid circulation through the U bend area. The support plate cut shall be located such that the outermost tube rows are fully encompassed and supported by the full support plate.
9. TEMA Type G, H, X and J shells in U-tube exchangers shall be provided with a full support baffle at the U-bend. The support baffle shall be provided with a 12 mm ($\frac{1}{2}$ in) notch on the top and bottom.
10. Floating heads in shells with removable shell covers shall be supported by a doughnut baffle. Unless required for anchoring sealing strips, the doughnut baffle need not be provided with TEMA Type T floating heads provided without removable shell covers.

11. Vertically cut baffles with large baffle cuts and/or large un-tubed heights under nozzles may be furnished with "ears" as necessary to seal the un-tubed void area created for bundle entrance or escape area.
12. Tube holes for all baffles shall be specified as 0.4 mm (1/64 in) over the tube outside diameter.
13. For single-phase fluid flow with a shell side fluid fouling tendency not high or severe, with a fouling resistance not exceeding $0.000352 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$), horizontally cut baffles should be used when a main cross-flow B-stream flow fraction advantage of 0.1 or greater can be achieved through use of horizontal cut baffles, else a vertical cut orientation is preferred.
14. Baffle cut shall be vertically oriented in the following cases:
 - a. double segmental baffle types;
 - b. shell-side fluid that normally contain or may contain suspended solids;
 - c. shell-side crude oil and residuum streams that are or have been exposed to temperatures above 315°C (600°F);
 - d. shell-side fluid is a condensing vapour; and
 - e. TEMA Type F, G or H shells
15. Baffle designs for shell side two-phase (no phase change) and flow boiling (excluding thermosiphon applications) requires special attention to ensure maximum mixing of the phases and wetting of the heat transfer surface area. The following guidelines shall be applied.
 - a. Bundle configurations should achieve a vapour shear flow regime over the predominant amount of heat transfer surface; typically 2/3rds of the surface area.
 - b. Horizontal baffle cuts are generally preferred when a vapour shear dominated flow regime can be achieved over the predominant amount of heat transfer surface area. Baffle cuts, particularly when no tube in window designs are utilized, may be smaller than otherwise recommended in (2.7 para. 2) and (2.7 para. 3) in order to prevent phase separation in the window region. Horizontal baffle cuts however are not advised in services susceptible to coking, such as the high temperature end of the oil feed train in some Hydrocrackers, as they tend to experience accelerated pressure drop escalation versus vertical baffle cut.
 - c. Designs with vertical cut baffles may be used when the resulting shell side flow regime is everywhere vapour shear controlled, and when the two-phase frictional pressure drop through the tube bundle is at least two times the calculated liquid phase static pressure at the bundle diameter (i.e., ΔP_{frict} greater than or equal to $2 \times \rho_{\text{liq}} \times \text{gravitational acceleration constant} \times \text{bundle diameter}$, in consistent units).
 - d. Alternatively, a vertical exchanger orientation or horizontal orientation shell side with non-baffled TEMA shell types H, X or G may be used.
 - e. For applications in a horizontal exchanger using horizontal cut baffles, where the boiling or two-phase fluid is not in shear controlled flow for the entire length of the exchanger, a safety multiplication factor of 1.35 shall be applied to the calculated pressure drop to allow for pressure surges.
16. Shell side services, which may be subject to laminar or transition to turbulent flow with temperature ranges (inlet minus outlet temperatures in a single shell) greater than 20°C (36°F), shall be specified with horizontal cut baffles to prevent thermal stratification.
17. The effective outside surface area shall be based on length of tubes measured between inner faces of the tubesheets. The U bend area shall not be considered effective area for any service where:
 - a. the shell side stream is of medium or higher fouling characterization, or
 - b. its fouling resistance exceeds $0.00026 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.0015 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$), or

- c. a full support baffle blocks flow in the U bend area.

Where floating head exchangers are provided with a floating head support, the surface area located between the support and the floating tubesheet shall not be considered as effective.

18. The baffle located adjacent to the U-bend shall be located a distance of approximately two tube diameters in front of the U-bend tangent point.
19. When the baffle located at the U-bend is a flowing baffle, the baffle cut shall be on the opposite side of the shell from the nozzle located either over or beyond the U-bends.
20. The maximum size of U tube or floating head bundles, that will routinely be pulled for mechanical cleaning, shall normally be 1525 mm (60 in) diameter and 7315 mm (24 ft) nominal tube length. Certain facilities have cleaning and/or handling equipment that may limit the maximum dimensions to lower values. Thus, the thermal design engineer shall always verify that the dimensions of "large" bundles are acceptable to the particular Principal's location.
21. No size limits are placed on fixed tube sheet exchangers.
22. For straight tubes, standard tube lengths of 2440 (8), 3050 (10), 3660 (12), 4880 (16), 6100 (20) and 7315 (24) mm (ft) should be used. Alternatively tube lengths of 2.5 m, 3.0 m, 4.0 m, 5.0 m, 6.0 m and 7.5 m may be used. For U-tubes, the maximum tube length, from the tube end to the U-bend tangent point, shall be 7315 mm (24 ft). Longer tube lengths may be considered for special services, subject to the approval of the Principal.
23. Vertical thermosiphon reboilers should be limited to a maximum nominal tube length of 4880 mm (16 ft), unless approved by the Principal.
24. Acceptable tube diameters are 19.05 mm (3/4 in), 25.4 mm (1 in), 31.75 mm (1-1/4 in), 38.10 mm (1-1/2 in) and 50.80 mm (2 in). Minimum tube wall thicknesses shall comply with Table 1 of DEP 31.21.01.30-Gen.
25. Tubes with an outside diameter of 15.87 mm (5/8 in) may be used when approved by the Principal, and generally should only be considered for fluids designated as very low fouling per DEP 20.21.00.31-Gen.
26. Tubes with an outside diameter of 19.05 mm (3/4 in) may be used for:
 - Heat exchangers where the tube-side fluid has a very low or low fouling characterization or fouling resistance of less than $0.00035 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$);
 - Heat exchanger services where experience exists with this tube size such that a heat transfer limiting condition is reached prior to a pressure drop limit;
 - Water-cooled heat exchangers with the cooling water through the tubes, when the water side fouling resistance is less than $0.00034 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$). Except for closed re-circulating cooling water, when carbon steel tube metallurgy is provided only straight tubes may be used in this diameter. When alloy tubes are provided, or for closed re-circulating cooling water in any tube metallurgy, the use of U-tubes may be considered.
27. Tubes with an outside diameter of 25.4 mm (1 in) or larger shall be used for:
 - a. Heat exchangers where the tube-side fluid has a medium or higher fouling characterization or fouling resistance of $0.00035 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.002 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$) or higher;
 - b. Heat exchangers where it would be economical to use this size because of process design restraints, e.g., pressure drop limitations;
 - c. Properly treated cooling tower water with carbon steel U-tubes, when approved by the Principal;

- d. Vertical thermosiphon reboilers with tube side fouling resistances greater than $0.00026 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.0015 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$)
28. Tubes with an outside diameter of 31.75 mm (1- $\frac{1}{4}$ in), 38.10 mm (1- $\frac{1}{2}$ in) or 50.8 mm (2 in) should be used for:
- Special types of shell and tube heat exchangers in Sulphur Recovery Units (SRU) and SCOT units (such as sulphur condensers, and steam re-heaters). The tube wall thickness may be increased for these special applications;
 - Heat exchangers where it would be economical to use these sizes because of process design restraints, e.g., pressure drop limitations.
29. The heat exchanger designer shall verify that the selected tube wall thickness, including corrosion allowance, is adequate for the internal and external pressure. Consideration shall be given to both the minimum tube wall thickness and bend thinning in U-tubes. Inner rows of U-tubes, up to a total of 10% of the total number of tubes, may be specified as a heavier wall gauge when required for pressure.
30. Unless otherwise specified in the material selection report, carbon steel and low alloy steel tubes shall have a corrosion allowance of 0.8 mm (1/32 in). High alloy steel tubes shall have a corrosion allowance of 0.4 mm (1/64 in).
31. The tubes shall be arranged in standard triangular and square patterns with layout angles of 30, 60, 45 or 90 degrees as defined by TEMA.
32. Ninety or 45 degree layouts shall be used for removable bundles. These bundles shall have continuous cleaning lanes with a minimum width of 6.35 mm ($\frac{1}{4}$ in).
33. The 90° layout is generally preferred due to its higher pressure drop to heat transfer conversion efficiency. Forty-five degrees should be specified for low shell side Reynolds numbers (generally less than 500) to improve the heat transfer coefficient. Forty-five degrees may also be considered in services to reduce vibration due to fluid-elastic instability, but shall not be used in gas service which might be subject to acoustic vibration.
34. Ninety degree layouts shall not be used in cross flow (TEMA Type X) exchangers utilizing low finned tubing.
35. Thirty or 60 degree layouts shall be used for non-removable bundles and may be applied for removable bundles in clean shell side service with a fouling resistance of $0.00018 \text{ m}^2\text{-}^\circ\text{K/W}$ ($0.001 \text{ hr-ft}^2\text{-}^\circ\text{F/Btu}$) or less, or a low to very low fouling margin where mechanical cleaning is not required.
36. The 30° triangular layout is generally preferred due to its higher pressure drop to heat transfer conversion efficiency. Sixty degree triangular layouts may be used to reduce shell side pressure drop.
37. The mean radius of U-bends shall not be less than one and one-half times the nominal outside diameter of the tube (i.e., $1.5 \times \text{tube O.D.}$) unless otherwise noted. When mechanical cleaning (high pressure water lancing) of U-bends are required, the mean radius of U-bends shall comply with (2.6 para. 4).

Where titanium U-tubes are specified, regardless of the need for mechanical cleaning, the minimum bend radius shall be two times the nominal outside tube diameter for tube diameters less than 25.4 mm (1 in), and 3.3 times the nominal tube diameter for tube diameters greater than or equal to 25.4 mm (1 in).

Where standard duplex stainless steel (e.g., S31803 or S32205) U-tubes are specified, regardless of the need for mechanical cleaning, the minimum bend radius shall not be less than 3.3 times the nominal outside diameter of the tube. For benign services (e.g., non sour or wet H_2S), consideration may be given to using a minimum bend radius of two times the nominal tube diameter when approved by the Principal.