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Revision Memo

12/01 Reaffirmation of previous revision of DP IX-D, December, 1995.

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SCOPE

The following subsection presents the recommended procedures for sizing shell and tube heat exchangers in which there is no change of phase; i.e., vapor/vapor, vapor/liquid, or liquid/liquid exchangers.

REFERENCES

- Standards of Tubular Exchanger Manufacturer's Association (TEMA).
- Heat Transfer Research, Inc. (HTRI) Design Manual.
- Heat Transfer Research, Inc. (HTRI) Report S-SS-3-1, *Stream Analysis Method for Prediction of Shellside Heat Transfer and Pressure Drop in Segmentally-Baffled Exchangers*, January 1967.
- Heat Transfer Research, Inc. (HTRI) Report S-SS-3-2, *Parametric Response Study of Segmentally Baffled Shell and Tube Exchanger Performance*, September 1967.
- Users' Manual for the Hextran computer program.

GENERAL

COMPUTER PROGRAMS

ExxonMobil's heat transfer programs (see [Section IX-K](#)) will perform the following calculations for shell and tube heat exchangers in which there is no change of phase:

- Rating of a dimensionally specified exchanger with or without a given heat duty.
- Design of all essential dimensions of an exchanger to specified process and geometric parameters.

The programs employ the HTRI-developed "Stream Analysis Method" for shell side calculations and refinement of existing correlations for tube side calculations.

The HTRI Stream Analysis Method is the most rigorous means available for calculating shell side pressure drop and heat transfer. Although the method is based on theoretically deduced equations, the correlations are empirical because the constants in the equations were obtained from experimental data. The most notable features of the Stream Analysis Method are:

- The ability to predict with reasonable accuracy the amount of flow leaking through baffle clearances and bypassing the tube bundle.
- The ability to predict accurately the effects of variations in exchanger geometry on heat transfer and pressure drop.
- The recognition of a temperature profile distortion effect because leakage streams undergo different temperature changes than the cross flow stream.
- The demonstrated ability to predict shell side heat transfer and pressure drop over a wide range of geometry and Reynolds Number variations.

SHELL SIDE MANUAL METHOD

A detailed manual calculation method is presented in the HTRI Design Manual. The procedure is very time-consuming for day-to-day use and therefore is not included. The detailed procedure should be referred to only for a more complete understanding of the leakage streams and interpretation of the HTRI program outputs.

ESTIMATION METHOD

The estimation procedure provides a quick but still fairly accurate method to size an exchanger to meet given process requirements within the constraints of the specified pressure drops.

The estimation method will produce results that are reasonably accurate when all of the following conditions are met:

- The tube side is in turbulent flow (Reynolds Number > 10,000) at both the inlet and the outlet. For 10,000 > Reynolds Number > 2,000 at either inlet or outlet the HTRI computer program using the ten-step option is recommended. For Reynolds Number < 2,000 at either inlet or outlet, the HTRI computer program is strongly recommended.
- Baffle cuts of 20% to 30% (h/D) for segmental baffles or 35% to 45% (area) for double segmental baffles.
- Baffle spacing/shell diameter ratio between 0.25 and 0.8.

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- Number of sealing strips: The exchanger has at least one pair of sealing strips for every five to seven tube rows crossed. Sealing strips are not required for typically designed fixed tube sheet or U-tube exchangers with horizontal cut baffles. F-stream seal rods/bars - The exchanger has F-stream seal rods/bars for every five to seven rows crossed. This is especially important for U-tube exchangers when the shell side cross flow passes through the lane formed by the innermost U-bends. In this case crisscrossing the inner row of U-bends is highly recommended. For crisscrossed U-bends, assume that F-stream seals are used, and note on the specification sheet that the innermost rows should be crisscrossed.
- Tube pitch ratio (PR) between 1.25 and 1.33.

Table 1 shows the calculation procedure for the estimation method. A sample calculation is given in **Table 2**.

EFFECTIVE DESIGN SPECIFICATIONS

DESIGN PARAMETERS

The following general specifications give the most efficient conversion of pressure drop to heat transfer for no-change-of-phase heat exchangers.

- Tube Length: Various exchanger geometries, that is, various combinations of shell diameters and tube lengths, can be found to satisfy given process requirements. The exchanger having the smallest amount of heat transfer surface area is not necessarily the most economical solution. The cost of a typical exchanger having a fixed tube diameter, metallurgy, and tube pitch can be approximately represented by the following relations as a function of shell diameter and tube length.

$$\text{Cost of Exchanger} \propto (\text{Shell Diameter})^{1.9}$$

$$\text{Cost of Exchanger} \propto (\text{Tube Length})^{0.55}$$

Based on the equations, for most cases the economical solution would be an exchanger with long tubes and small shell diameter. There are, however, some restrictions on the tube length such as bundle pulling facilities, availability of tubes, and ease of maintenance. In the absence of specific guidelines, a standard tube length of 20 ft (6090 mm) should be used, unless the design is to retrofit an existing plant where a different tube length is standard.

- Baffle cut: 20% to 30% for segmental baffles; 35% to 45% for double segmental baffles.
- Baffle spacing/shell diameter: 0.3 to 0.5.
- Number of sealing strips: The number of pairs of sealing strips per tube row crossed between baffle tips should be about 0.14 to 0.20, or about 5 to 7 tube rows per pair of sealing strips for floating head exchangers. Sealing strips are needed for typically designed fixed tube sheets or U-tube exchangers only when vertical cut baffles are specified.
- Tube layout patterns: If inline arrangements are required for cleaning, use a 90° layout for turbulent flow and a 45° layout for transition and laminar flow. If mechanical cleaning is not required, use 30° layout for both laminar and turbulent flow. Do not use 60° layout since it exhibits inferior heat transfer or pressure drop characteristics when compared to a 30° layout.

RELATIONSHIPS BETWEEN HEAT TRANSFER COEFFICIENTS AND PRESSURE DROPS

The heat transfer mechanism in the shell and tube heat exchangers is convective heat transfer, whereby a fluid receives or supplies heat to a heat transfer surface through a film. The film is perceived as a static fluid (viscous) membrane that coats the heat transfer surface; heat is transferred through the film by conduction. Therefore, thicker films provide lower heat transfer coefficients, and thinner films provide higher heat transfer coefficients.

The velocity of the fluid passing over the surface determines the thickness of the film. The higher the velocity, the thinner the film, resulting in a larger heat transfer coefficient. The fluid experiences some pressure drop in order to maintain this velocity. The pressure drop increases with increased velocity and tube length.

This description holds for tube side fluids. For shell side fluids, leakage streams complicate the simple model. On the shell side of an exchanger, all of the fluid does not pass over the tubes because there are other paths available. Extensive work on these leakage streams has been done by HTRI and in the Delaware research projects. An exact evaluation is impossible without a computer program because of the various streams present. **Figure 3** shows the leakage streams. The C, E and F streams bypass the heat transfer surface (tubes) completely. The tube to baffle leakage stream (A) passes over some tubes but bypasses others. All of these streams are hydraulically in balance; i.e., the pressure drop from any one point in the shell to another is the same. The amount of each leakage stream is determined by using such a hydraulic balance. To complicate matters further, a leakage stream mixes with the part of the fluid flowing over the tube bundle at some point. The complexity of the flows make complete manual analysis impractical without simplifying assumptions based on test data and experience. The estimation method presented here is based on such factors.

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EFFECTIVE DESIGN SPECIFICATIONS (Cont)

Table 3 provides satisfactory results in most instances in prorating shell side and tube side heat transfer and pressure drops.

A well designed exchanger utilizes the pressure drop in a way that maximizes the overall heat transfer coefficient. Arriving at a practical and economic solution is an iterative procedure. As a starting point, use the overall coefficients given **Table 1 of Section IX-B**. Calculate the MTD as discussed in the estimation method calculation sheets (As a starting point one may approximate $MTD = \sqrt{(T_1 - t_2) \times (T_2 - t_1)}$) to arrive at the approximate required surface for the given service, and then go through the estimation procedure, calculating heat transfer and pressure drops. Compare the shell side coefficient, tube side coefficient, and (fouling + wall) coefficient. If the shell side coefficient is much smaller than the tube side coefficient, then the shell side coefficient is controlling. If the tube side coefficient is much smaller than the shell side coefficient, then the tube side coefficient is controlling. If (fouling + wall) coefficient is controlling, then fouling is controlling.

When fouling is controlling, it generally does not help to utilize all of the allowable pressure drops unless shell size can be reduced, or the pressure drop is very much underutilized. (Refer to **Section IX-A** for a discussion of online cleaning devices which can be used to reduce fouling.) When the shell side coefficient is controlling, it helps to maximize shell side pressure drop to reduce shell size. Maximizing tube side pressure drop normally does not help. When the tube side coefficient is controlling, it helps to maximize tube side pressure drop but not shell side pressure drop. When all the coefficients are in the same range, it helps to maximize both shell side and tube side pressure drop.

MEANS OF INCREASING HEAT TRANSFER WHEN PRESSURE DROP IS LESS THAN THE MAXIMUM ALLOWED

A design with the optimum parameters that satisfies heat transfer requirements but does not use up the total allowable pressure drop is considered inefficient. For such cases, one can always make the exchanger smaller by taking one or more of the following steps to increase the pressure drop and heat transfer.

Tube Side

1. Increase the tube length.
2. Decrease the tube diameter.
3. Increase the number of tube passes, up to a maximum as specified in **Table 4 of Section IX-C** and within the restriction of tube side velocities.
4. Try integral fin tubes, which has the effect of decreased tube diameter on pressure drop only.

Shell Side

1. Decrease the shell diameter. This can be done by reducing tube pitch to the minimum allowed by construction practice and refinery limitations. Reducing shell diameter without changing tube pitch will result in fewer tubes and thus lower heat transfer area. In cases where the exchanger has enough excess surface, this could be the best solution.
2. Decrease the baffle pitch, but to a point that is never less than 20% shell diameter or 2 in. (50 mm), whichever is smaller.
3. Try TEMA F type shell if the process parameters and refinery practice permit the use of an F shell. In general, F shells should not be used when the shellside pressure drop is greater than 10 psi or the shellside temperature change is greater than 350°F (195°C), due to the thermal and hydraulic leakage across the shell partition. A new design for certain U-tube exchangers with a welded-in shell partition plate is available but lacks a proven performance record. Consult plant maintenance and the Heat Transfer Equipment Section of ER&E regarding the acceptability of an F shell for a given application.

MEANS OF MOST EFFICIENTLY DECREASING PRESSURE DROP

Sometimes a design requires more surface than is necessary for heat transfer in order to provide sufficient flow area to prevent exceeding the allowable pressure drop. Two courses of correction are possible.

1. Adjust the "allowable" pressure drop. Possibly the pressure drop to which the unit is designed is arbitrarily low and can be increased.
2. Adjust the geometry so as to provide the smallest decrease in heat transfer per unit decrease in pressure drop, by making one or more of the following changes:

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EFFECTIVE DESIGN SPECIFICATIONS (Cont)

Tube Side

- Decrease the number of tube passes within the limits of minimum tube side velocities. This option should always be considered when the number of tube passes is 10 or more or when shell side heat transfer is controlling and the number of tube passes is 6 or more. Never decrease the number of tube passes if the result will change the tube side flow regime from turbulent to laminar flow, or if the velocities will fall below the minimum required due to fouling or salt deposition consideration.
- Increase the tube diameter, within practical limits. Normally tubes up to 1-1/2 in. (38.1 mm) diameter (2 in. (50.8 mm) for carbon steel) are readily available in the market.
- Decrease the tube length. This is an attractive option when an exchanger has excess surface. Normally, decrease tube length by 2 ft (610 mm) increments for tube lengths over 10 ft (3048 mm), or 1 ft (305 mm) increments for smaller tube lengths.
- Increase tube I.D. by using a thinner tube wall and a metallurgy with better corrosion resistance. This may be an expensive option, but it should be considered when an existing exchanger is retrofitted to avoid even more expensive piping changes.
- If an exchanger does not have enough excess surface, then increase the shell diameter, within allowable limits set by the refinery, normally 60 in. (1525 mm).

Shell Side

- If an exchanger has enough excess surface, consider the following options.
 - Decrease the tube length.
 - Increase the baffle pitch, but to not more than 80% of the shell diameter.
 - Use double segmental baffles.
 - Increase the tube pitch. Note that increasing the tube pitch will reduce the number of tubes for a given shell size and will affect tube side heat transfer and pressure drop.
 - Use a TEMA J shell.
- If an exchanger does not have enough excess surface then increase the shell diameter within allowable limits set by the refinery, normally 60 in. (1525 mm). Note that shell diameter could also be limited by the maximum permissible weight.
- Other ways of reducing shell side pressure drop, for which calculation methods are not provided in this section because they require more complex design procedures include:
 - Use a rod baffle design (useful for large flow rates on shell side).
 - Use a low finned tube design. (Overhead condensers, lube oil product coolers). Refer to [Section IX-G](#).
 - Use a no-tubes-in-the-window designs. (Usually very useful for gas-plant feed effluent exchangers).
 - Use shell side longitudinal flow with or without longitudinal fins.
 - Use a TEMA X shell. (Usually most useful for very large flow rates on the shell side with nozzle sizes > 50% shell diameter.).

If none of the actions listed here will give the desired results, consider the use of units in parallel.

ESTIMATION METHOD CALCULATION PROCEDURE

- Based on the hot and cold fluid conditions select tube metallurgy.
- Get fouling factors for the hot and cold fluids from [Table 3](#) of [Section IX-B](#).
- Select tube side fluid per page 10 of [Section IX-C](#). Place the other fluid on the shell side.
- Determine the heat duty "Q" and/or the missing fluid condition based on a duty balance. The heat received by the cold fluid should equal the heat lost by the hot fluid.
- Calculate the minimum number of shell passes using the graphical method in [Figure 2D](#) of [Section IX-B](#). As a first approximation use this number as the number of shells in series, N_s . Assume number of shell is parallel, $N_p = 1$.
- Based on the tube metallurgy and tube side fouling, determine the tube diameter and the tube wall thickness per pages 4 and 5 of [Section IX-C](#).
- Based on shell side fouling, select the tube pitch and layout, per page 7 of [Section IX-C](#). Check that the selected tube wall thickness is acceptable.

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ESTIMATION METHOD CALCULATION PROCEDURE (Cont)

8. Select the channel and tube bundle types per pages 6 and 7 of [Section IX-C](#). For the trial use an "E" shell with two tube passes.
9. Select the initial overall heat transfer coefficient " U_o " from [Table 1](#) of [Section IX-B](#).
10. Get the single tube external surface per unit length " A_T " from [Table 1](#) of [Section IX-C](#).
11. Per step 1 of [Table 1A](#) (1B metric) calculate the corrected log mean temperature difference "MTD".
12. Assume effective tube length " L_e " to be 19.5 ft (5.95 m) and calculate required number of tubes per bundle "NTT" as follows

$$NTT = \frac{Q}{U_o \cdot MTD \cdot N_s \cdot N_p \cdot A_T \cdot L_e}$$

13. Start with the maximum permissible shell diameter. Then using [Table 5](#) of [Section IX-C](#) get the maximum number of tubes per bundle. If this number of tubes calculated is greater than the number calculated in step 12, proceed to step 14. Otherwise, increase the number of shells in series, and go back to step 11.
14. For the initial guess use line size nozzles, and use an impingement plate.
15. Use proration factors given in [Table 3](#), calculate the shell size, and confirm the number of tubes by calculations per [Table 5](#) of [Section IX-C](#). Repeat the procedure if required till calculated number of tubes is within $\pm 5\%$ of the number of tubes required. This may not be possible for pipe size shells, in which case use the shell size that gives the number of tubes closest to the number calculated in step 12. For rolled and welded shells select shell I.D. as an integer number. Use the shell diameter, bundle diameter and number of tubes calculated in this step for further calculation of initial trial.
16. For the initial guess use 25% cut single segmental baffles at a pitch equal to 40% of the bundle diameter.
17. Using this initial guess go through the complete calculation procedure per [Table 1A](#) (1B for metric).
18. Check nozzle pressure drops, and if excessive, increase nozzle sizes. The limits for maximum nozzle pressure drops are as follows:
 - a. Shellside Gases - Use 35% of the allowable pressure drop for the inlet & outlet nozzles, total.
 - b. Shellside Liquids - Use 15% of the allowable pressure drop for inlet and outlet nozzles, total.
 - c. Shellside Nozzle pVn_2 value should never exceed TEMA maximum limits.
 - d. Tube side inlet and outlet nozzle pressure drop should be less than 40% of the calculated tube side pressure drop for a single tube pass unit, or, 35% of the calculated tube side pressure drop for a multiple tube pass unit.
19. Check the results for overdesign and pressure drop utilization. Follow the guidelines described earlier in this DP and the proration factors given in [Table 3](#) to arrive at the optimum design. It may take a few more iterations. While selecting a new exchanger during optimization, note the following limitations.
 - a. Cooling water velocity inside tubes must meet the requirements of [Table 4](#).
 - b. Special requirements for intube velocity, may be applicable, e.g., slurry service.
 - c. Maintain Baffle Pitch/Bundle Diameter ratio between 0.25 and 0.8.
 - d. Try to maintain turbulent flow on the tube side.
 - e. Maintain the tubewall temperature below or above criterion set by fouling or pour point. For example, cooling water or waxy services.



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NOMENCLATURE

a	=	Tube row spacing factor, dimensionless
A	=	Total effective heat transfer area, ft ² (m ²)
A _s	=	Effective heat transfer area per shell, ft ² (m ²)
A _T	=	Single tube outside surface area per unit length ft ² /ft (m ² /m)
C _p	=	Heat capacity, Btu/lb-°F (kJ/kg-°C)
DI	=	Tube I.D., in. (mm)
DO	=	Tube O.D., in. (mm)
DN	=	Nominal nozzle I.D., in. (mm)
DOTL	=	Diameter of bundle outer tube limit, in. (mm)
DS	=	Shell I.D., in. (mm)
DSNI	=	Shell side inlet nozzle I.D., in. (mm)
DSNO	=	Shell side outlet nozzle I.D., in. (mm)
DTNI	=	Tube side inlet nozzle I.D., in. (mm)
DTNO	=	Tube side outlet nozzle I.D., in. (mm)
f	=	Non-isothermal friction factor, dimensionless
f _{is}	=	Isothermal friction factor, dimensionless
F _n	=	Correction factor for log mean temperature difference, dimensionless
F _s	=	Shell side pressure drop correction factor, dimensionless
Gr	=	Grashof Number, dimensionless
G _{xt}	=	Total cross flow mass velocity, lb/hr-ft ² (kg/s-m ²)
h _{io}	=	Inside film coefficient based on tube outside surface area, Btu/hr-ft ² -°F (W/m ² -°C)
h _s	=	Outside film coefficient, Btu/hr-ft ² -°F (W/m ² -°C)
HF	=	Shell side friction term, dimensionless
HM	=	Shell side momentum term, dimensionless
j	=	Stanton Number type heat transfer factor, dimensionless
k	=	Thermal conductivity of fluid, Btu/hr-ft ² -°F/ft (W/m-°C)
k _w	=	Thermal conductivity of tube wall at average tube temperature, Btu/hr-ft ² -°F (W/m-°C)
K _e	=	Tube side pressure drop coefficient, dimensionless
ℓ	=	Tube wall thickness, in. (mm)
L	=	Tube length, ft (m)
L _e	=	Effective tube length, ft (m)
LBCC	=	Central baffle pitch, in. (mm)
LI	=	Tube flow length, in. (mm)
LMTD	=	Log mean temperature difference for true counter current flow, °F (°C)
m	=	Shell side flow factor, dimensionless
MTD	=	Corrected log mean temperature difference, °F (°C)
n	=	Baffle spacing to bundle diameter ratio, dimensionless
N	=	Number of tube passes
N _p	=	Number of shells in parallel
N _s	=	Number of shells in series
N _T	=	Total number of shells



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NOMENCLATURE (Cont)

NTP	=	Number of tube passes per shell
NTT	=	Number of tubes in a bundle
p	=	Baffle flow factor, dimensionless
P _r	=	Prandtl Number, dimensionless
PR	=	Tube pitch ratio, dimensionless
PT	=	Tube pitch, in. (mm)
ΔP _e	=	Tube entrance, expansion, and turnaround pressure drop, psi (kPa)
ΔP _{exch}	=	Total nozzle to nozzle shell side pressure drop, psi (kPa)
ΔP _n	=	Tube side nozzle pressure drop, psi (kPa)
ΔP _s	=	Shell side pressure drop (excluding nozzles), psi (kPa)
ΔP _{sn}	=	Shell side nozzle pressure drop, psi (kPa)
ΔP _t	=	Tube side frictional pressure drop, psi (kPa)
(ΔP _t) _{nn}	=	Total tube side nozzle pressure drop, psi (kPa)
Q	=	Rate of heat transfer, Btu/hr (W)
r _i	=	Inside fouling factor referred to inside surface area, hr-ft ² -°F/Btu (m ² -°C/W)
r _{io}	=	Inside fouling factor referred to outside surface area, hr-ft ² -°F/Btu (m ² -°C/W)
r _o	=	Shell side fouling factor, hr-ft ² -°F/Btu (m ² -°C/W)
r _w	=	Resistance of tube wall at average wall temperature, hr-ft ² -°F/Btu (m ² -°C/W)
R _c	=	Total resistance (clean) to heat transfer, hr-ft ² -°F/Btu (m ² -°C/W)
R _{io}	=	Inside resistance to heat transfer corrected to outside area, hr-ft ² -°F/Btu (m ² -°C/W)
R _o	=	Outside film resistance to heat transfer, hr-ft ² -°F/Btu (m ² -°C/W)
R _t	=	Total resistance (duty) to heat transfer, hr-ft ² -°F/Btu (m ² -°C/W)
Re	=	Reynolds Number, dimensionless
Re _{xt}	=	Shell side total flow Reynolds Number, dimensionless
Re _{xh}	=	Cross flow Reynolds number for heat transfer, dimensionless
Re _{xp}	=	Cross flow Reynolds Number for pressure drop, dimensionless
RC	=	Reynolds number correction factor, dimensionless
SC	=	Baffle spacing correction factor, dimensionless
S _{TT}	=	Tube sheet material allowable stress at design temperature, lb/sq in. (kPa)
t	=	Tube wall thickness, in. (mm)
t ₁	=	Inlet temperature of fluid being heated, °F (°C)
t ₂	=	Outlet temperature of fluid being heated, °F (°C)
T _w	=	Average wall temperature, °F (°C)
T _{DS}	=	Shell side design temperature, °F (°C)
T _{DT}	=	Tube side design temperature, °F (°C)
T _M	=	Tube sheet design temperature, °F (°C)
T ₁	=	Inlet temperature of fluid being cooled, °F (°C)
T ₂	=	Outlet temperature of fluid being cooled, °F (°C)
TS _b	=	Bulk temperature of shell side fluid, °F (°C)
TS _{in}	=	Inlet temperature of shell side fluid, °F (°C)
TS _{out}	=	Outlet temperature of shell side fluid, °F (°C)
TT _b	=	Bulk temperature of tube side fluid, °F (°C)



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NOMENCLATURE (Cont)

TT_{in}	=	Inlet temperature of tube side fluid, °F (°C)
TT_{out}	=	Outlet temperature of tube side fluid, °F (°C)
TTT	=	Total tube sheet thickness, ft (m)
ΔT_a	=	Terminal temperature difference, °F (°C)
ΔT_s	=	Shell side temperature difference, °F (°C)
U_c	=	Overall clean coefficient of heat transfer, Btu/hr-ft ² -°F (W/m ² -°C)
U_d	=	Calculated overall fouled coefficient of heat transfer, Btu/hr-ft ² -°F (W/m ² -°C)
U_o	=	Overall duty coefficient of heat transfer, Btu/hr-ft ² -°F (W/m ² -°C)
V_n	=	Tube side average nozzle fluid velocity, ft/sec (m/s)
V_t	=	Fluid velocity in tubes, ft/sec (m/s)
W	=	Tube side mass rate per shell, lb/hr (kg/s)
W_s	=	Shell side mass rate per shell, lb/hr (kg/s)
β'	=	Coefficient of thermal expansion used in Grashof Number, °F ⁻¹ (C ⁻¹)
γ^*	=	Natural convection factor, dimensionless
ϵ	=	Short tube laminar flow correction factor, dimensionless
η	=	Transitional flow heat transfer proration factor, dimensionless
λ	=	Tube length geometry factor, dimensionless
μ_b	=	Viscosity at bulk temperature, centipoise (Pa-S)
μ_w	=	Viscosity at wall temperature, centipoise (Pa-S)
ξ	=	Baffle correction factor, dimensionless
ρ	=	Density, lb/ft ³ (kg/m ³)
ψ	=	Low Prandtl Number correction factor, dimensionless
ψ_p	=	Natural convection correction factor, dimensionless
ϕ	=	Viscosity correction for wall temperature



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TABLE 1A
ESTIMATION METHOD CALCULATION FORM (CUSTOMARY UNITS)
SHELL AND TUBE HEAT EXCHANGERS (NO CHANGE OF PHASE)

PROCESS _____ REFINERY AND LOCATION _____ CALC. BY _____
EXCHANGER NO. E _____ DATE _____

1. Terminal Conditions and MTD

Fluid Being Cooled $T_1 =$ _____, $T_2 =$ _____, $T_1 - T_2 =$ _____, $R = \frac{T_1 - T_2}{t_2 - t_1} =$ _____

Fluid Being Heated $t_2 =$ _____, $t_1 =$ _____, $t_2 - t_1 =$ _____, $P = \frac{t_2 - t_1}{T_1 - t_1} =$ _____

$T_1 - t_2 =$ _____, $T_2 - t_1 =$ _____, $T_1 - t_1 =$ _____

$$LMTD = \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln \left[\frac{T_1 - t_2}{T_2 - t_1} \right]} =$$

Ns

$F_n =$ _____ for _____ shells in series
(Figure 2) _____ for _____ shells in series
_____ for _____ shells in series

Np

_____ shells in parallel
_____ shells in parallel
_____ shells in parallel

$$MTD = F_n (LMTD) = () () =$$

$$= () () =$$

$$= () () =$$

2. Bulk Temperatures

Tubeside Fluid (Heated) $TT_{in} =$ _____, $TT_{out} =$ _____
(Cooled)

$$TT_b = \frac{TT_{in} + TT_{out}}{2} =$$

Shellside Fluids (Heated) $TS_{in} =$ _____, $TS_{out} =$ _____
(Cooled)

$$TS_b = \frac{TS_{in} + TS_{out}}{2} =$$

$$\text{Estimated Wall temperature } T_w = TT_b + 0.6 (TS_b - TT_b) =$$

3. Properties of Fluids

	Tubes	Shell
$\rho =$	_____ lb/ft ³	_____ lb/ft ³
$\mu_b =$	_____ cP	_____ cP
$C_p =$	_____ Btu/lb-°F	_____ Btu/lb-°F
$k =$	_____ Btu/hr-ft ² -°F/ft	_____ Btu/hr-ft ² -°F/ft
$\mu_w =$	_____ cP	_____ cP



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TABLE 1A (Cont)
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4. Flow Rates and Names of Fluids

	Tube	Shell
Fluid		
Name/Phase =	_____/____	_____/____
Total Mass rate	_____ lb/hr	_____ lb/hr
W = Total mass rate/Np =	_____ lb/hr	W _s = Total mass rate/Np = _____ lb/hr

5. Fouling Factors

r_i = _____ hr-ft²-°F/Btu
r_o = _____ hr-ft²-°F/Btu

6. Mechanical Design Features

	Tubes	Shell
TEMA Type		
Design Temperature	= _____ °F	_____ °F
Design Pressure	= _____ psig	_____ psig
Nozzle Size	DTNI = _____ in.	DSNI = _____ in.
	DTNO = _____ in.	DSNO = _____ in.

7. Exchanger Geometry

	First Trial	Second Trial	Third Trial
Tube O.D. (DO), in.	_____	_____	_____
Tube I.D. (DI), in.	_____	_____	_____
Tube pitch (PT), in.	_____	_____	_____
Tube wall thickness (t) = $\frac{DO - DI}{2}$	_____	_____	_____
Tube length (L), ft	_____	_____	_____
Tube flow length (LI), ft	_____	_____	_____
LI = (24) for U tubes			
LI = (12) for all others			
TEMA exchanger type	_____	_____	_____
Single tube area/length, A _T , ft ²	_____	_____	_____
A _T = $\frac{(\pi)(DO)}{12}$	_____	_____	_____
Baffle spacing (LBCC), in.	_____	_____	_____
Baffle type (Segmental or Double Segmental)	_____	_____	_____
Bundle Diameter (DOTL), in.	_____	_____	_____
Shell Diameter (DS), in.	_____	_____	_____
Number of tube side passes (NTP) (for U tubes minimum number is 2)	_____	_____	_____



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TABLE 1A (Cont)
ESTIMATION METHOD CALCULATION FORM (CUSTOMARY UNITS)
SHELL AND TUBE HEAT EXCHANGERS (NO CHANGE OF PHASE)

8. Iteration, Tube Side		First Trial	Second Trial	Third Trial
a. Heat Duty = Q	=	_____	_____	_____
b. Assumed value of U_o	=	_____	_____	_____
c. $A = Q/U_o$ (MTD)	=	_____	_____	_____
d. $A_s = A/N_T$	=	_____	_____	_____
e. Tube Metal = _____; $K_w =$ _____, DO = _____ in., DI = _____ in., $\ell =$ _____ in., L = _____ ft				
f. Tube Pitch (PT) and Layout	=	_____	_____	_____
g. $r_{io} = (DO/DI) r_i$	=	_____	_____	_____
h. $r_w = \ell / 12 k_w$	=	_____	_____	_____
i. NTT	=	_____	_____	_____
j. $N = NTT/NTP$	=	_____	_____	_____
k. $V_t = \frac{W}{19.625 \rho N (DI)^2}$	=	_____	_____	_____
l. Heat Transfer Coefficient				
(1) $Re = \frac{6.32 W}{\mu_b (DI) N} = \frac{124 \rho V_t (DI)}{\mu_b}$	=	_____	_____	_____
(2) For Water		_____	_____	_____
$h_{io} = \frac{368}{DO} (V_t DI)^{0.7} \left(\frac{TT_b}{100} \right)^{0.26}$	=	_____	_____	_____
(3) Fluids other than water				
(a) $Pr = \frac{2.42 c_p \mu_b}{k}$	=	_____	_____	_____
(b) If $Re \geq 10,000$	=	_____	_____	_____
$h_{io} = \frac{0.2642 k}{DO} Re^{0.8} Pr^{0.4} \phi P$				
$\phi P = (\mu_b / \mu_w)^{0.167}$ for liquids				
$= \left(\frac{TT_b + 460}{T_w + 460} \right)^{0.5}$ for heating gases				
$= 1.0$ for cooling gases				
(c) If $Re < 10,000$				
(1) $\Delta T = \left \frac{T_w - TT_b}{1} - \frac{1}{1} \right $	=	_____	_____	_____
(2) $\beta' = \frac{\rho_x}{(t_x - t_y)(1/\rho_{av})}$	=	_____	_____	_____
(3) $Gr = 413(10^4) \left(\frac{DI^3 \rho^2 \beta' \Delta T}{\mu_b^2} \right)$	=	_____	_____	_____

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TABLE 1A (Cont)
ESTIMATION METHOD CALCULATION FORM (CUSTOMARY UNITS)
SHELL AND TUBE HEAT EXCHANGERS (NO CHANGE OF PHASE)

8. Iteration, Tube Side (Cont)	First Trial	Second Trial	Third Trial
(3) $Gr = 413(10^4) \left(\frac{DI^3 \rho^2 \beta' \Delta T}{\mu_b^2} \right)$	=		
(d) If $Re \leq 2000$			
(1) From Figure 1.5 determine γ^*	=		
(2) From Figure 1.6 determine ϵ	=		
(3) $\lambda = DI/LI + \epsilon$	=		
(4) From Figure 1.7 determine ψ	=		
(5) $h_{io} = \frac{12k}{DO} \left\{ 2.5 + 4.5 [(Re + \gamma^*) \lambda]^{0.37} Pr_b^{0.17} \left(\frac{\mu_b}{\mu_w} \right)^{0.14} \right\} \psi$			
(e) If $2000 < Re < 10,000$			
(1) $(h_{io})_{turb} @ Re = 10,000$ (from b))	=		
(2) $(h_{io})_{lam} @ Re = 2,000$ (from d))	=		
(3) $\eta = 1.25 - Re/8000$	=		
(4) $h_{io} = \eta (h_{io})_{lam} + (1 - \eta)(h_{io})_{turb}$	=		
(f) $T_w = TT_b + U_o (R_{io})(TS_b - TT_b)$	=		
(g) Evaluate $(\mu_b / \mu_w)^{0.14}$ for laminar flow or ϕP for turbulent flow and make necessary corrections to heat transfer coefficients	=		
m. Pressure Drop			
(1) Nozzle Pressure Drop,			
(a) $V_n = \frac{0.051 W}{(DTNI)(DTNO)\rho}$	=		
(b) $\Delta P_n = \frac{\rho V_n^2}{5152}$	=		
(2) Entrance, Expansion, and Turn-around			
(a) From Table 4 determine K_e	=		
(b) $\Delta P_e = \frac{K_e \rho V_t^2}{9274}$	=		
(3) Frictional Pressure Drop			
(a) From Figure 1.8 evaluate f_{is}	=		
(b) From Figure 1.9 evaluate ϕ	=		
(c) From Figure 1.10 evaluate ψ_p	=		
(d) $f = f_{is} \phi \psi_p$	=		
(e) $\Delta P_t = \frac{5.19}{10^3} f \left[\frac{\rho V_t^2 (NTP)L}{DI} \right]$	=		
(4) $(\Delta P_t)_{nn} = N_s [\Delta P_n + \Delta P_e + (F_t) (\Delta P_t)]$ get value of F_t from Table 4	=		

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TABLE 1A (Cont)
ESTIMATION METHOD CALCULATION FORM (CUSTOMARY UNITS)
SHELL AND TUBE HEAT EXCHANGERS (NO CHANGE OF PHASE)

9. Iteration, Shell Side	First Trial	Second Trial	Third Trial
Tube row spacing factor a = 1.0 for Square Pitch a = 0.867 for all others	_____	_____	_____
Pitch Ratio (PR) = PT/DO	_____	_____	_____
Baff. Space to Bundle Dia. Ratio (n) n = LBCC/DOTL	_____	_____	_____
Shell side flow factor (m) m = 0.5 for J shell m = 2.0 for F shell m = 1.0 for E shell	_____	_____	_____
Baffle flow factor (p) p = 1.0 for segmental baffles p = 0.5 for double-segmental baffles	_____	_____	_____
Baffle Correction Factor (ξ) ξ = 1.0 for segmental baffles ξ = 0.8 for double-segmental baffles	_____	_____	_____
Estimated tube wall temperature (T_w), °F $T_w = TS_b - 0.3 (TS_b - TT_b)$	_____	_____	_____
Wall viscosity correction (ϕ) $\phi = (\mu_b / \mu_w)^{0.14}$	_____	_____	_____
Total cross flow mass velocity, (G_{xt}), lb / hr – ft ² $G_{xt} = 144 \left[\frac{PR}{PR - 1} \right] \left[\frac{(m)(p)}{n} \right] \left[\frac{W_s}{(DOTL)^2} \right]$	_____	_____	_____
Total flow Reynolds Number (Re_{xt}) $Re_{xt} = \frac{(DO) (G_{xt})}{(29) (\mu_b)}$	_____	_____	_____
Nominal Crossflow fraction (FFB_n) Figure 1.1	_____	_____	_____
Baffle Spacing Correction (SC) Figure 1.2	_____	_____	_____
Reynolds Number Correction (RC) Figure 1.3	_____	_____	_____
Cross flow fraction for pressure drop (FFB_p) $FFB_p = (FFB_n) (SC) (RC)$ (Maximum = 1.0)	_____	_____	_____
Cross flow Reynolds Number for ΔP (Re_{xp}) ($Re_{xp} = (Re_{xt}) (FFB_p)$)	_____	_____	_____
Shell side friction for factor (f) Figure 1.4	_____	_____	_____
Cross flow fraction for heat transfer (FFB_h) $FFB_h = FFB_p + 0.125$ (Maximum = 1.0)	_____	_____	_____
Cross flow Reynolds Number for heat transfer (Re_{xh}) $Re_{xh} = (Re_{xt}) (FFB_h)$	_____	_____	_____
Shell side heat transfer factor (j) Figure 1.4	_____	_____	_____



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TABLE 1A (Cont)
ESTIMATION METHOD CALCULATION FORM (CUSTOMARY UNITS)
SHELL AND TUBE HEAT EXCHANGERS (NO CHANGE OF PHASE)

9. Iteration, Shell Side (Cont)	First Trial	Second Trial	Third Trial
Shell side film coefficient (h_s) Btu/hr-ft ² -°F			
$h_s = (0.415)(C_p)(G_{xt})(FFB_h)(i) \left[\frac{k}{(C_p)(\mu)} \right]^{2/3} (\phi)(\xi)$			
Shell side Fraction Term (HF)			
$HF = 0.00875 \left[\frac{(f)(L)(m)(12)}{(a)(PR)(DO)(n)} \right] \left(\frac{1}{\phi} \right) F_s$			
(F _s from Table 4)			
Shell side momentum term (HM)			
$HM = 0.0168 (n)^{2.6} \left[\frac{(L)(m)(12)}{(n)(DOTL)} - 1.0 \right]$			
Shell side pressure drop without Nozzle (ΔP_s), psi			
$\Delta P_s = \frac{(0.3)[H_F + H_M]}{\rho} \left[\frac{(G_{xt})(FFB_p)}{10000} \right]^2 \left(\frac{1}{\xi} \right)$			
Nominal shell nozzle I.D. (DN), in.			
$DN = \sqrt{(DSNI)(DSNO)}$			
Shell side nozzle pressure drop (ΔP_{SN}), psi			
$\Delta P_{SN} = \frac{0.84}{\rho} \left[\frac{W_s}{(1000)(DN)^2} \right]^2$			
Total shell pressure drop (ΔP_{exch}), psi			
$\Delta P_{exch} = N_S [\Delta P_s + \Delta P_{SN}]$			
Calculated Tube wall temperature (T _w), °F			
$T_w = TS_b - U_o \left(\frac{1.0}{h_s} \right) (TS_b - TT_b)$			
Calculated viscosity correction (ϕ)			
$\phi = (\mu_b / \mu_w)^{0.14}$			
Using calculated value of ϕ correct h_s & ΔP_s			
10. Overall calculations			
Inside resistance referred to outside area (R _{io}), hr-ft ² -°F/Btu			
$R_{io} = 1/h_{io}$			
Outside resistance (R _o), hr-ft ² -°F/Btu			
$R_o = 1/h_s$			
Total clean resistance (R _c), hr-ft ² -°F/Btu			
$R_c = R_{io} + R_o + r_w$			
Calculated overall clean coefficient (U _c), hr-ft ² -°F/Btu			
$U_c = 1/R_c$			



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TABLE 1A (Cont)
ESTIMATION METHOD CALCULATION FORM (CUSTOMARY UNITS)
SHELL AND TUBE HEAT EXCHANGERS (NO CHANGE OF PHASE)

10. Overall calculations (Cont)	First Trial	Second Trial	Third Trial
Calculated overall fouled coefficient (U_D), Btu/hr-ft ² -°F			
$U_D = \frac{1}{R_{i0} + R_o + r_w + r_o + r_{i0}}$			
Tubesheet thickness (TTT), ft			
$TTT = \frac{(C_1)(DS + 2.5)(P / S_{TT})^{0.5} + 1.0}{12.0}$			
$C_1 = 0.625$ for U-tubes, 10 for all others			
P = higher of shell side or tube side design pressure.			
Effective tube length (L_e), ft			
$L_e = L - TTT$			
Effective heat transfer area per shell (A_s), ft ²			
$A_s = (NTT)(L_e)(A_T)$			
Total effective heat transfer area (A), ft ²			
$A = (A_s)(N_T)$			
Overall duty coefficient (U_o), Btu/hr-ft ² -°F			
$U_o = \frac{Q}{(A)(MTD)}$			
$\% \text{ Overdesign} = \frac{U_D - U_o}{U_o} \times 100$			
11. Tubesheet design temperature (Normally required for Copper alloys)			
a. For coolers. (Cooling Water on tube side)			
(1) Clean conditions, °F			
$T_M = T_{DT} + \frac{R_{i0}}{R_c}(T_{DS} - T_{DT})$			
(2) Fouled conditions, °F			
$T_M = T_{DT} + U_o(R_{i0} + r_{i0})(T_{DS} - T_{DT})$			
(3) Controlling case, °F			
Larger of the clean or fouled			
b. For exchanges			
(1) Tube side fluid being cooled			
$T_M = T_{DT} - 0.1(T_{DT} - T_{DS})$			
(2) Shell side fluid being cooled			
$T_M = T_{DT} + 0.3(T_{DS} - T_{DT})$			

