Class 13

Heat Exchanger Design

Simplified View of Heat Exchangers

used in preliminary stages of design

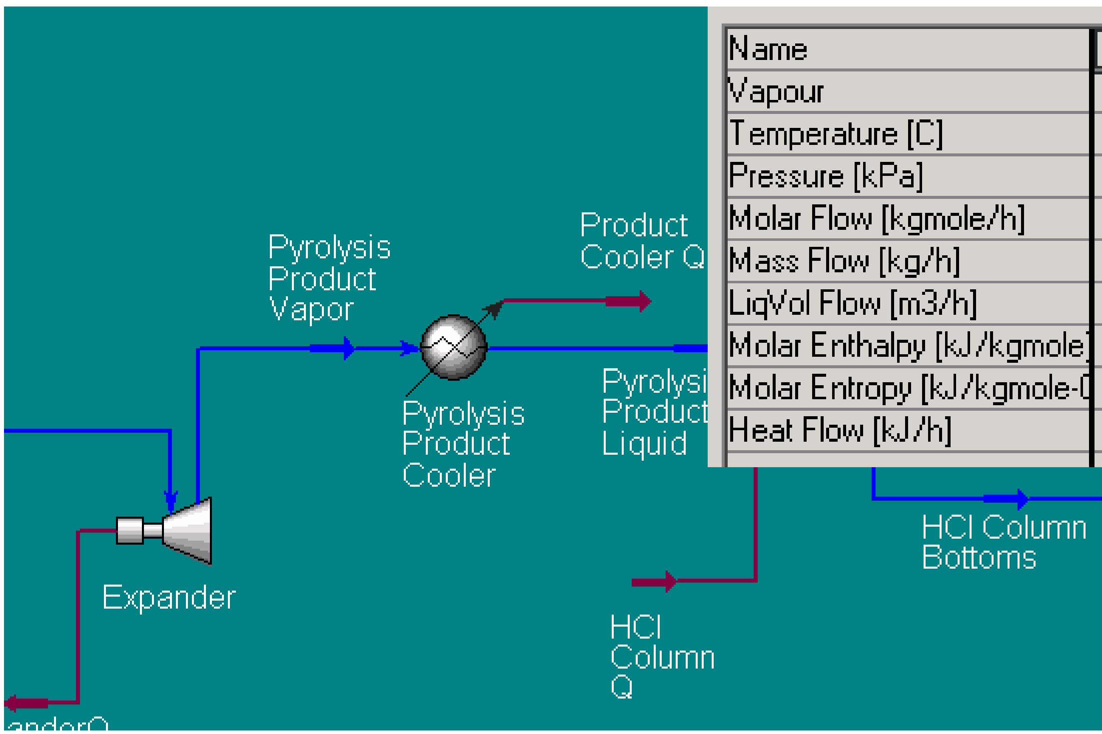
factors ignored:

- source/sink for energy transferred
- rate of energy transfer
- type and size of equipment

only overall heat duty considered

$$Q = \dot{m}\left(H_{out} - H_{in}\right)$$

Example from HYSYS Vinyl Chloride simulation



Name	Pyrolysis Produc	Pyrolysis Produc	Product Cooler
Vapour	1.0000	0.0000	<empty>
Temperature [C]	451.1	11.36	<empty>
Pressure [kPa]	1216	1216	<empty>
Molar Flow [kgmole/h]	1935	1935	<empty>
Mass Flow [kg/h]	1.197e+005	1.197e+005	<empty>
LiqVol Flow [m3/h]	120.2	120.2	<empty>
Molar Enthalpy [kJ/kgmole]	-2.656e+004	-7.526e+004	<empty>
Molar Entropy [kJ/kgmole-C	201.0	81.68	<empty>
Heat Flow [kJ/h]	-5.141e+007	-1.456e+008	9.424e+007

"one-sided" heat exchanger

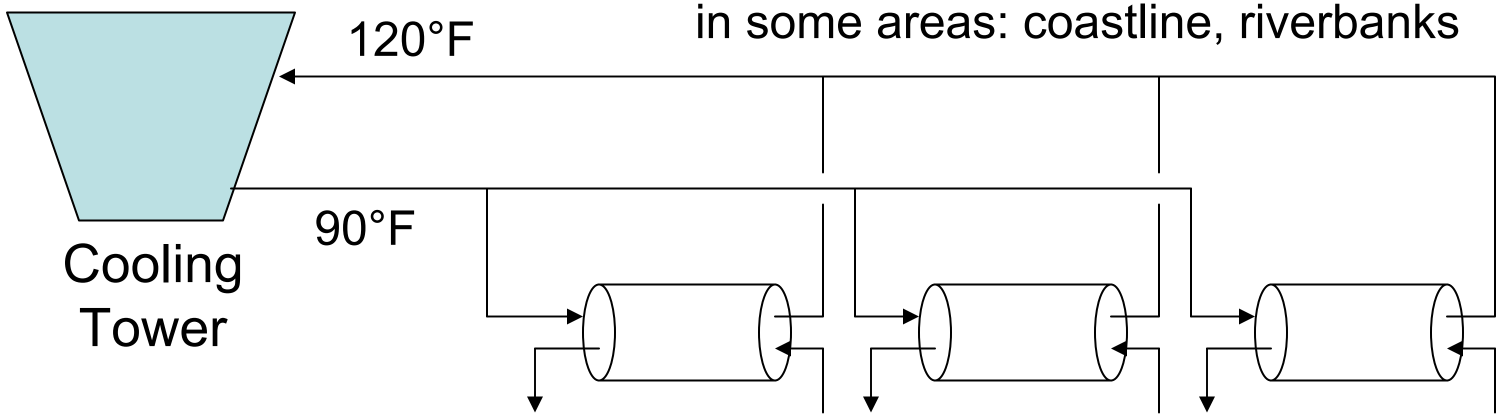
Heat Transfer Media (utilities)

Utility exchangers use media such as cooling water, steam, flue gases, refrigerants, etc.

Table 13.1 provides list of different media covering range -150ºF to 2000ºF

Cooling water (Huerisitic 27, pg 184)

Local abundant source of cool water in some areas: coastline, riverbanks



Direct air cooling to 120°F

If process streams are > 250°F, use waste heat boilers to create process steam.

Heat Transfer Media (utilities)

Cooling below 100°F: refrigerants used, designated by R-number

R-717: ammonia R-134a: tetrafluoroethane

R-290: propane

Refrigeration cycle required: compressor, condensor, expansion valve, + utility exchanger

Many HFC's ("freons") phased out since 1980 because of ozone layer damage.

Chilled water can be used for cooling to 45°F

Chilled brines to 0°F

Heat medium: commonly steam

LP: 50 psig (298°F) MP: 150 psig (366°F) HP: 450 psig (459°F)

Steam condenses in heat exchanger. Steam through-flow trapped.

For heating above 450°F to 750°F: Dowtherm

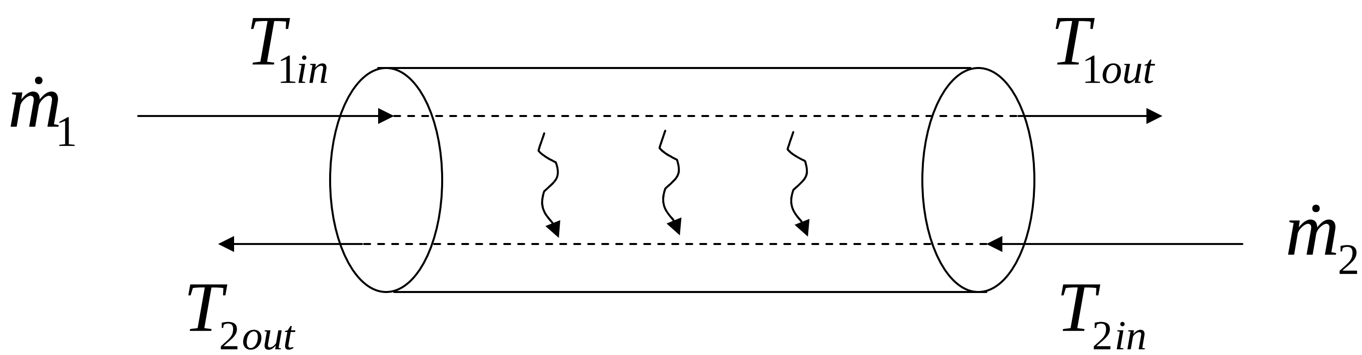
Molten salts (to 1100°F), molten metals (to 1400°F) for higher temperatures

Fired furnaces (to 2000°F) for high temperatures

Temperature Driving Force

When two heat transfer fluids considered, 2-sided heat exchanger model used.

With negligible heat losses,



$$Q = \dot{m}_1 C_{P1} \left(T_{1out} - T_{1in} \right) = \dot{m}_2 C_{P2} \left(T_{2in} - T_{2out} \right) = UA\Delta T_m$$

energy balances

transport equation

ΔT_m : mean temperature driving force

determines required heat exchange area

smallest ΔT at ends: minimum temperature approach

Temperature Driving Force

Common design strategy: specify

inlet conditions of each stream
pressure drops
a minimum approach temperature

Compute end for minimum, exit temperatures, heat duty

Optimal minimum approach temperature depends on conditions:

Heuristic 26

- cryogenic: 1-to-2°F; 10°F or less for temperatures < ambient
- ambient: 10°F; 20°F for ambient to 300°F
- high temperature: ~100°F

When one fluid is boiled:

Heuristic 28

- natural convection: ΔT's < 10°F
- nucleate boiling: ΔT's 20°F to 45°F   high transfer rates
- film boiling: ΔT > 100°F
- transition region: 50°F < ΔT < 100°F
- rule of thumb:
  maintain ΔT = 45°F

Temperature Driving Force

Example 13.4

Fluid 1: 100 lbmol/hr ethyl chloride
10 lbmol/hr ethanol
200°F, 35 psia

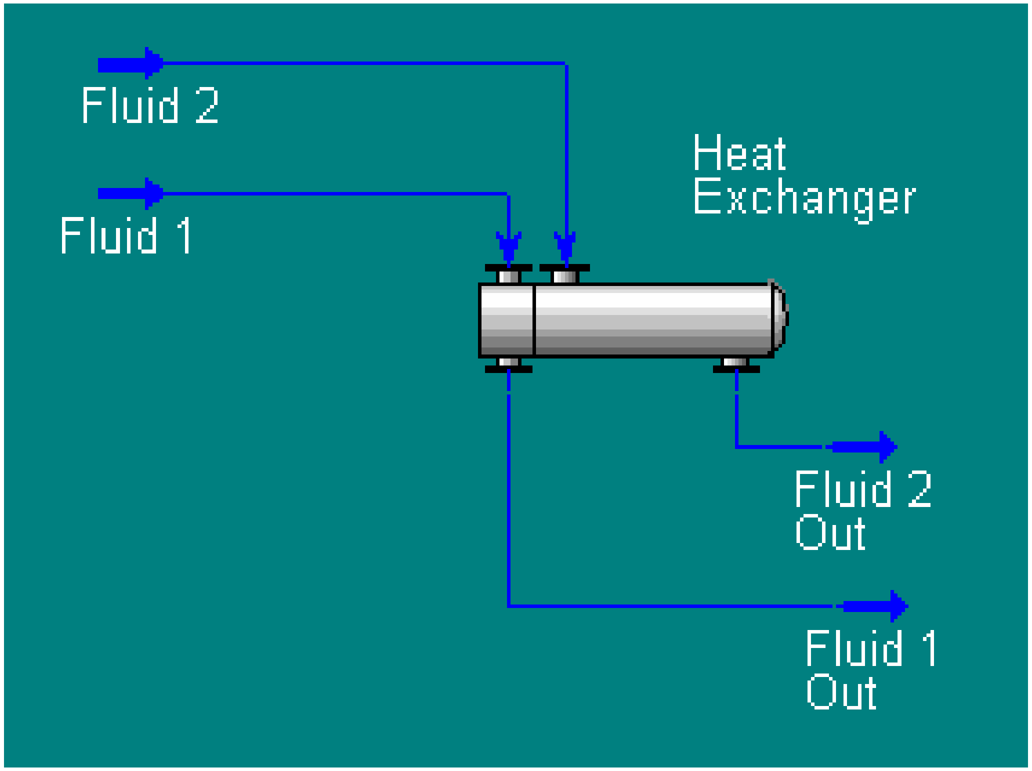
ΔP = 5 psi

Fluid 2: 90 lbmol/hr ethanol
90°F, 100 psia

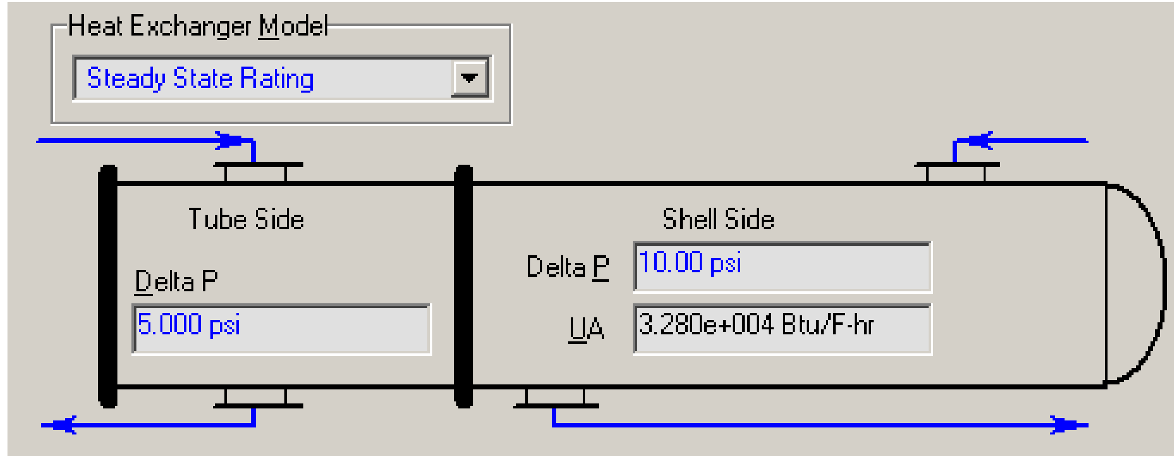
ΔP = 10 psi

Minimum approach temperature: 10°F

HYSYS Solution



HYSYS Solution



Sizing Data

(•) Overall ( ) Shell ( ) Tube

[ ] Accept any input data

Configuration

Number of Shell Passes	1
Number of Shells in Series	1
Number of Shells in Parallel	1
Tube Passes per Shell	1
Exchanger Orientation	Horizontal
First Tube Pass Flow Direction	Counter
Elevation (Base)	0.0000

TEMA Type: A E L

Calculated Information

Shell HT Coeff [Btu/hr-ft2-F]	1.051
Tube HT Coeff [Btu/hr-ft2-F]	31.15
Overall U [Btu/hr-ft2-F]	50.51
Overall UA [Btu/F-hr]	3.280e+004
Shell DP [psi]	10.00
Tube DP [psi]	5.000
Heat Trans. Area per Shell [ft2]	649.3
Tube Volume per Shell [ft3]	6.816
Shell Volume per Shell [ft3]	80.24

Specifications

	Specified Value	Current Value	Relative Error	Active	Estim.
E-100 Heat Balance	0.00 Btu/hr	-1.1e-009	-3.1e-015	☑	☐
E-100 UA	<empty>	3.3e+004	<empty>	☐	☑
E-100 ExchSpec	10 F	10.	-7.8e-005	☑	☑

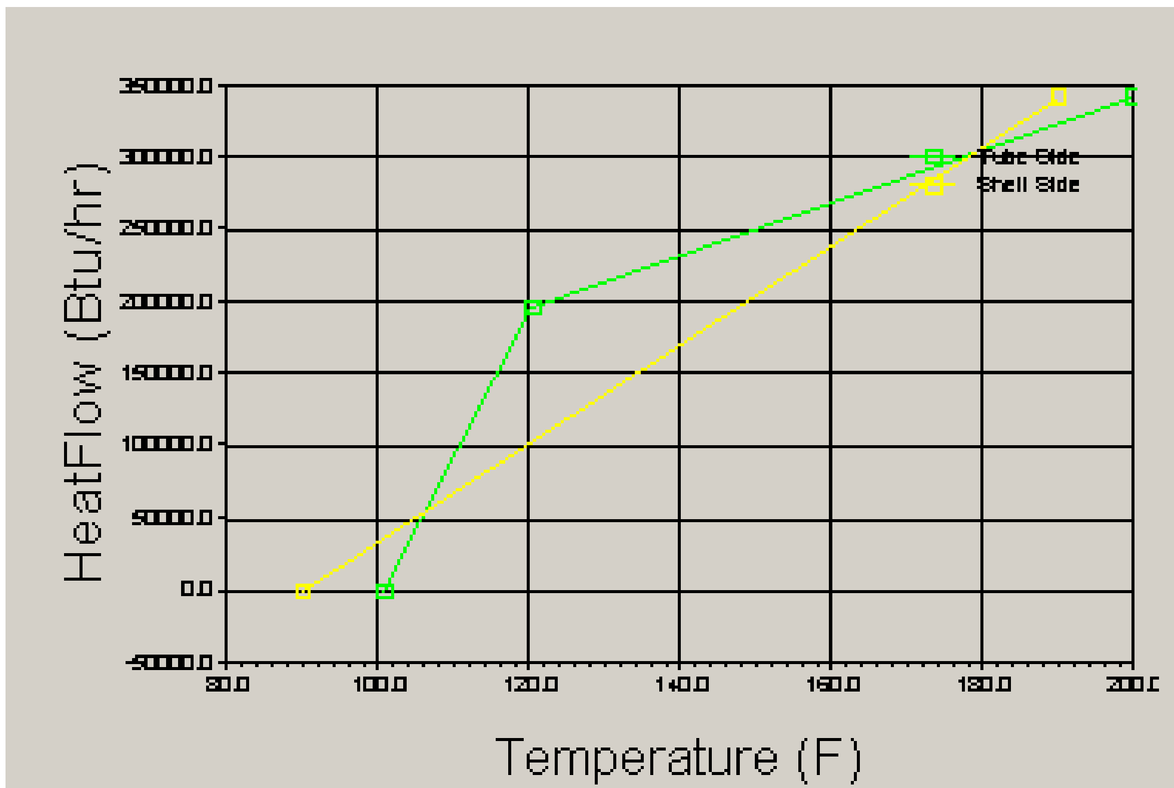
HYSYS Solution

Name	Fluid 1	Fluid 1 Out	Fluid 2	Fluid 2 Out
Vapour	1.0000	0.8922	0.0000	0.0000
Temperature [F]	200.0	100.9	90.00	190.0
Pressure [psia]	35.00	30.00	100.0	90.00
Molar Flow [lbmole/hr]	110.0	110.0	90.00	90.00
Mass Flow [lb/hr]	6912	6912	4146	4146
Std Ideal Liq Vol Flow [barrel/day]	529.1	529.1	356.7	356.7
Molar Enthalpy [Btu/lbmole]	-5.088e+004	-5.399e+004	-1.191e+005	-1.153e+005
Molar Entropy [Btu/lbmole-F]	34.18	29.35	8.568	19.56
Heat Flow [Btu/hr]	-5.596e+006	-5.939e+006	-1.072e+007	-1.038e+007

Approach ΔT spec met

but . . . temperature crossover

Note: condensation of hot stream

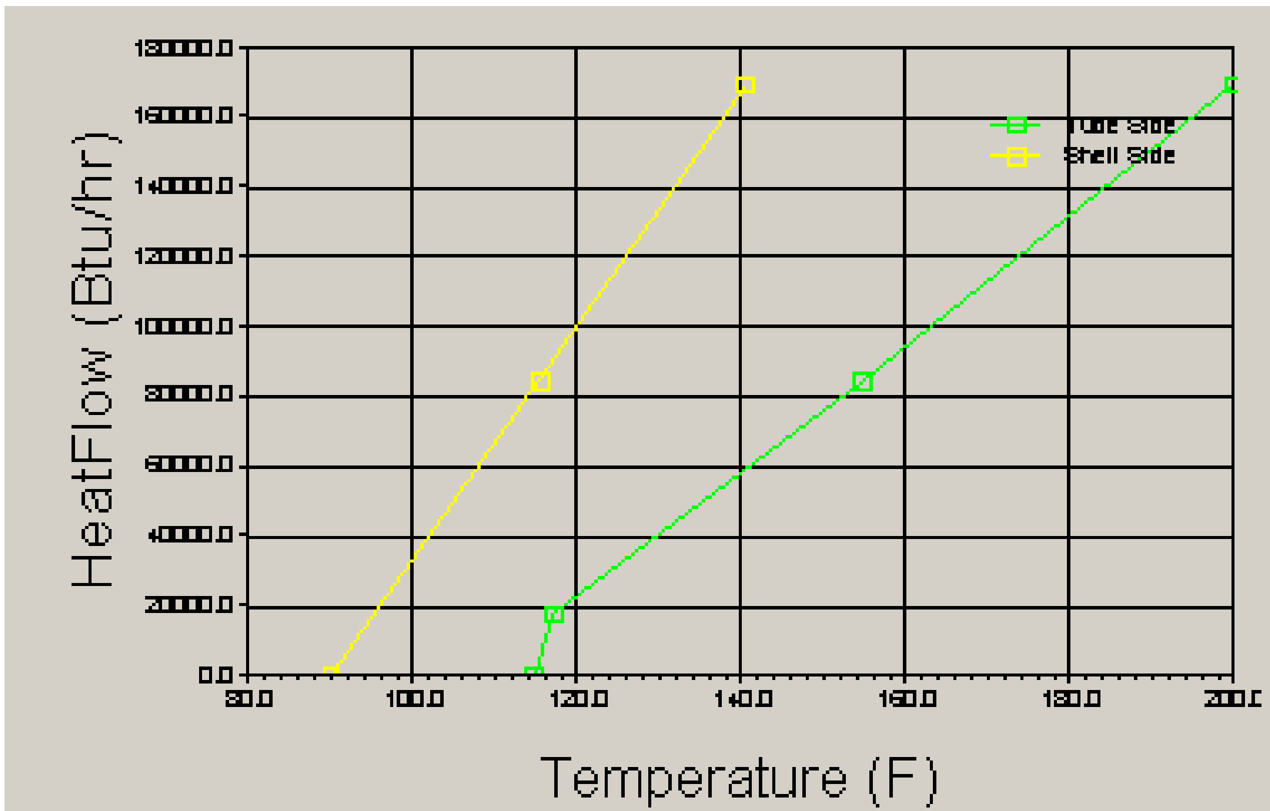


HYSYS Solution

Adjust approach ΔT to eliminate crossover

Specifications

	Specified Value	Current Value	Relative Error	Active	Estim.
E-100 Heat Balance	0.00 Btu/hr	2.1e-009	1.2e-014	☑	☐
E-100 UA	<empty>	4.3e+003	<empty>	☐	☑
E-100 ExchSpec	25 F	25	-3.8e-005	☑	☑



Typical Heat Exchanger Pressure Drops

Liquid streams with no phase change:	7 psi	Heuristic 31
Vapor streams with no phase change:	3 psi	
Condensing streams	1.5 psi	
Boiling streams	1.5 psi	
Process streams through furnaces	20 psi	

Catalog of Heat Exchange Equipment Types

pipe-in-pipe (double-pipe) small, < 1ft^2 transfer area

shell-and-tube 50 to 12,000 ft^2

countercurrent vs cocurrent flow vs crossflow
1-2, 1-4, 1-6, 1-8, 2-4, 2-8, 3-6, 4-8 (# shell pass - # tube pass)

air-cooled (fin-fan) area as much as 20,000 ft^2

compact

plate-and-frame
spiral-plate
spiral-tube
plate-fin

Double-pipe Heat Exchangers

Simplest form: single, straight run

Example: inner: 12-ft long, 1-1/4-in, Sch 40 pipe Area: 5.22 ft^2

More area req'd? use return bends, hairpin units

High temperature, high pressure, corrosive fluid in inside pipe

Generally not used when phase change occurs

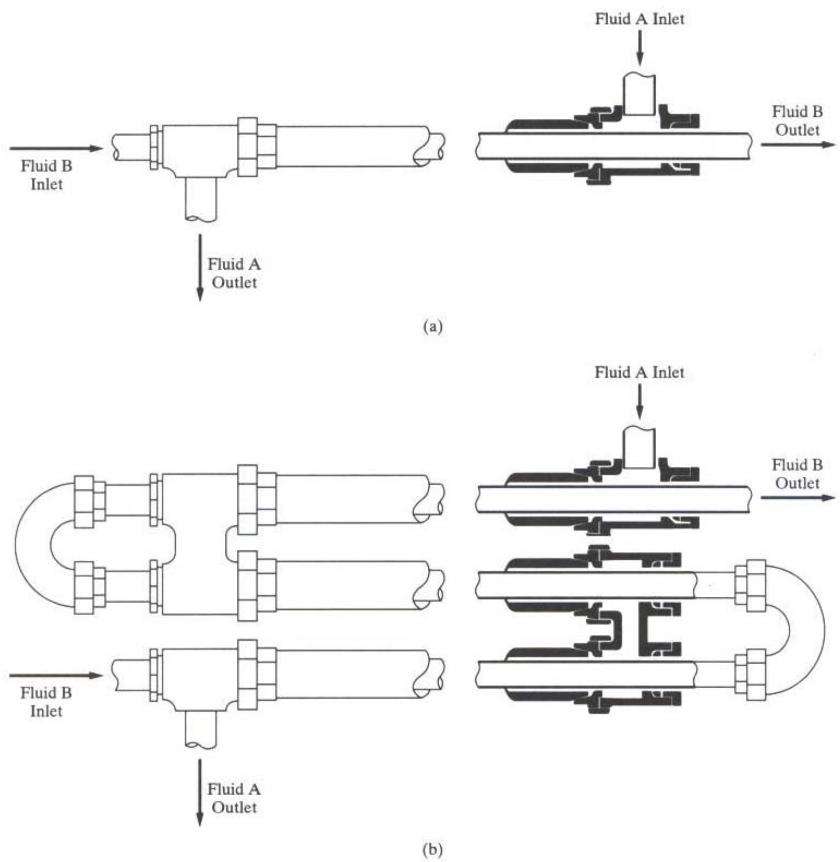


Figure 13.7 Double-pipe heat exchangers: (a) single unit; (b) hairpin unit.

Shell-and-tube Heat Exchangers

Designs standardized by Tubular Exchanger Manufacturers Association (TEMA)

Many configurations available

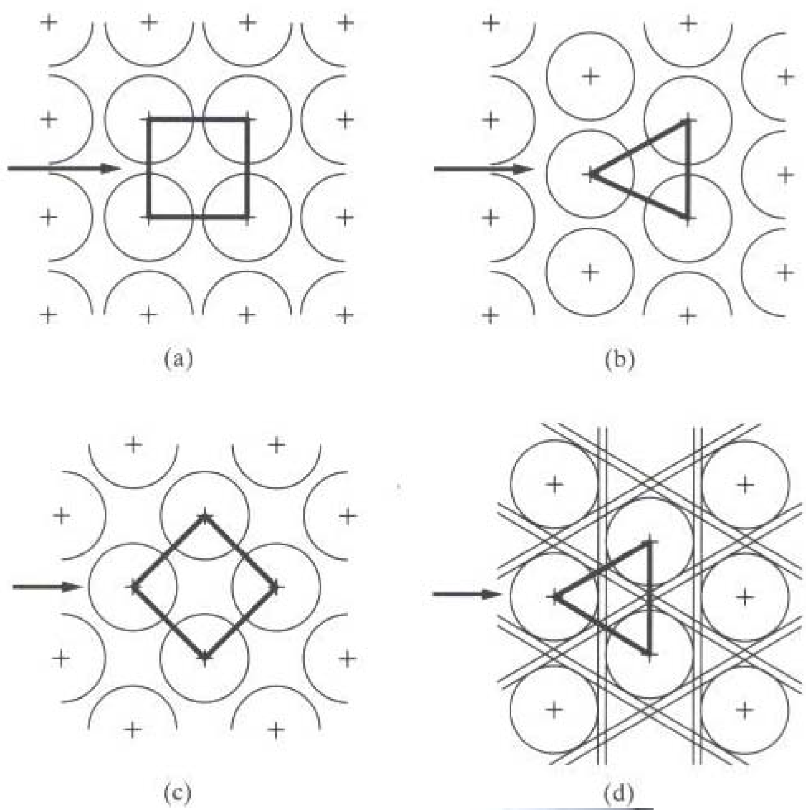


Figure 13.9 Tube layout patterns: (a) square pitch; (b) triangular pitch; (c) square pitch rotated; (d) triangular pitch with cleaning lanes.

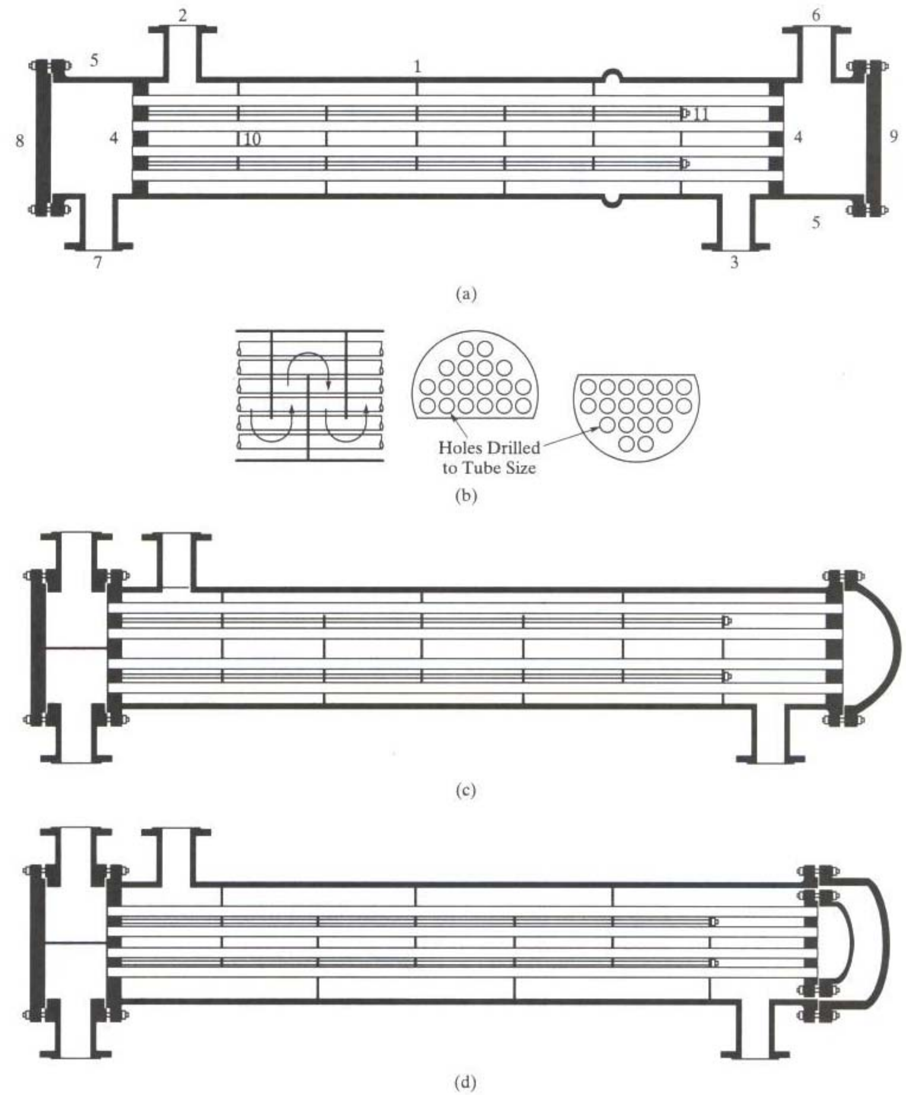


Figure 13.8 Shell-and-tube heat exchangers: (a) 1-1 fixed head; (b) segmental baffles; (c) 1-2 fixed head; (d) 1-2 floating head.

Air-cooled Heat Exchangers

used where cooling water is scarce

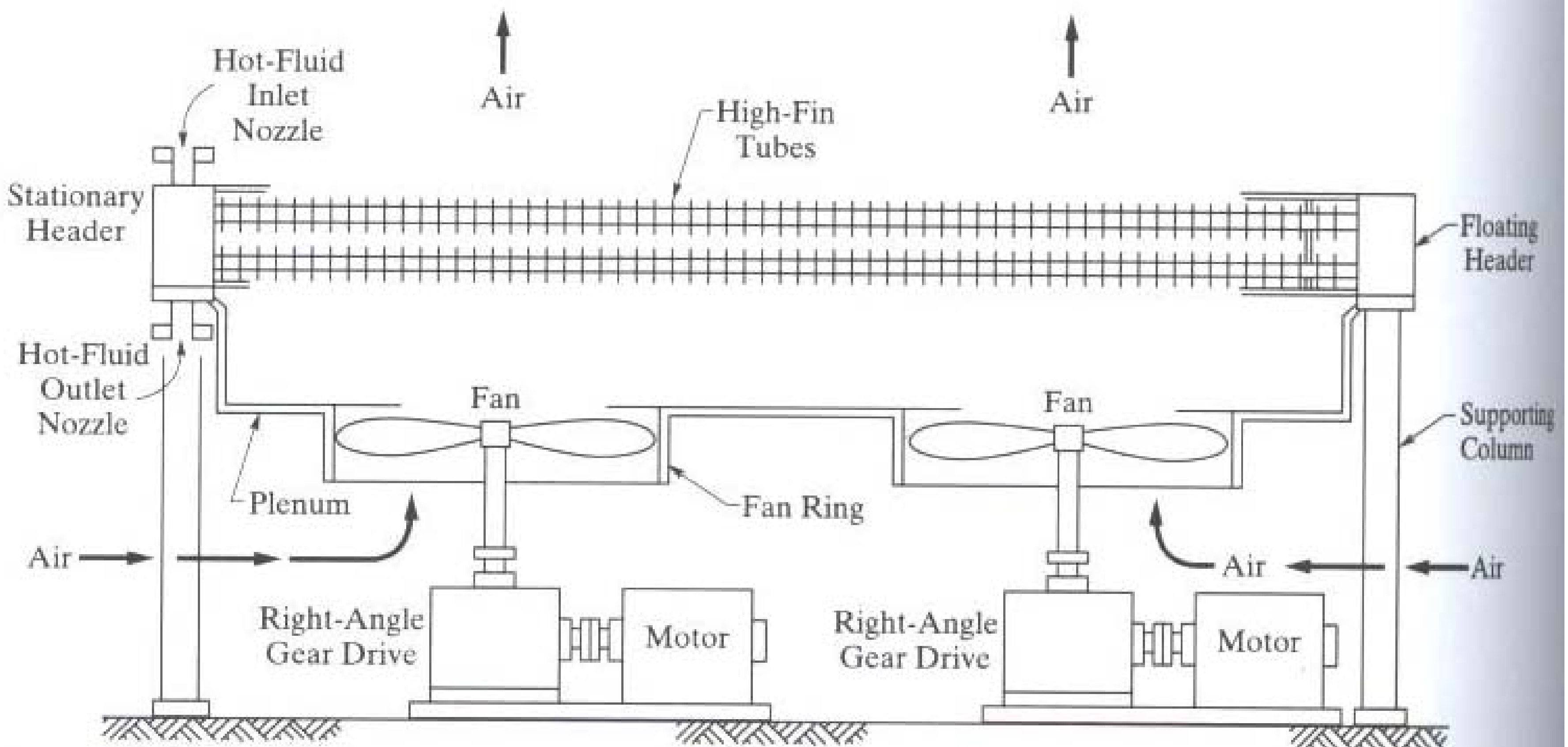


Figure 13.11 Fin-fan heat exchanger.

Compact Heat Exchangers

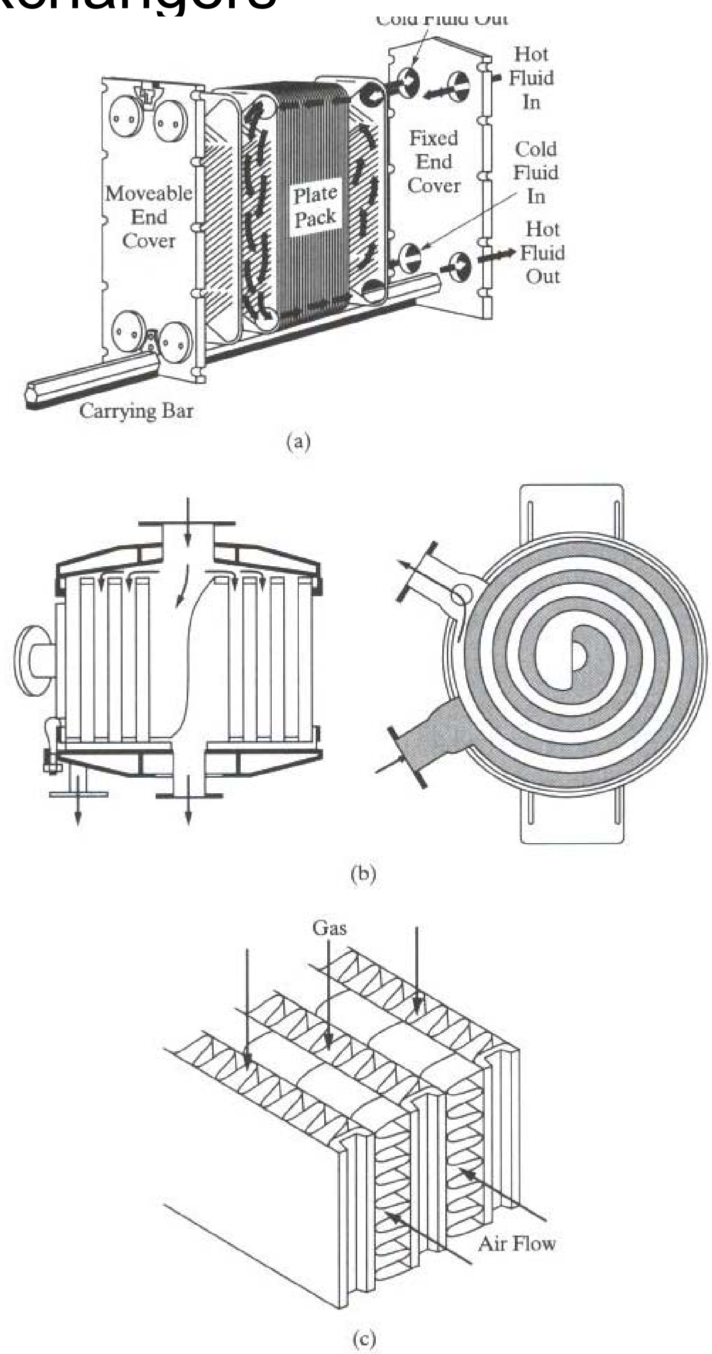


Figure 13.12 Compact heat exchangers: (a) plate-and-frame; (b) spiral plate; (c) plate fin.

Temperature Driving Forces

$$Q = UA\Delta T_m \qquad \Delta T_m : \text{ mean temperature driving force}$$

Describing ΔT_m is complex combination of factors:
flow configuration, fluid properties, phase change

Assumptions for use of ΔT_{LM}

steady-state flows
countercurrent or cocurrent
constant overall U
sensible enthalpy changes with constant C_p
negligible heat losses

Stream temperature profiles are linear

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\Delta T_1 / \Delta T_2\right)}$$

For isothermal condensation/boiling on one or both sides,
ΔT_{LM} applies to all heat exchanger configurations

Temperature Driving Forces

Correction factor for multiple-pass heat exchangers -- F_T

$$\Delta T_m = F_T \Delta T_{LM}$$

R and S Factors

$$R = \frac{T_{Hin} - T_{Hout}}{T_{Cout} - T_{Cin}} \qquad S = \frac{T_{Cout} - T_{Cin}}{T_{Hin} - T_{Cin}}$$

1-2 heat exchanger

$$F_T = \frac{\sqrt{R^2+1} \cdot ln\left[\dfrac{1-S}{1-RS}\right]}{(R-1) \cdot ln\left[\dfrac{2-S\left(R+1-\sqrt{R^2+1}\right)}{2-S\left(R+1+\sqrt{R^2+1}\right)}\right]}$$

$0.85 \le F_T \le 1$ desirable

$F_T \le 0.75$ unacceptable

Temperature Driving Forces

1-2 heat exchanger

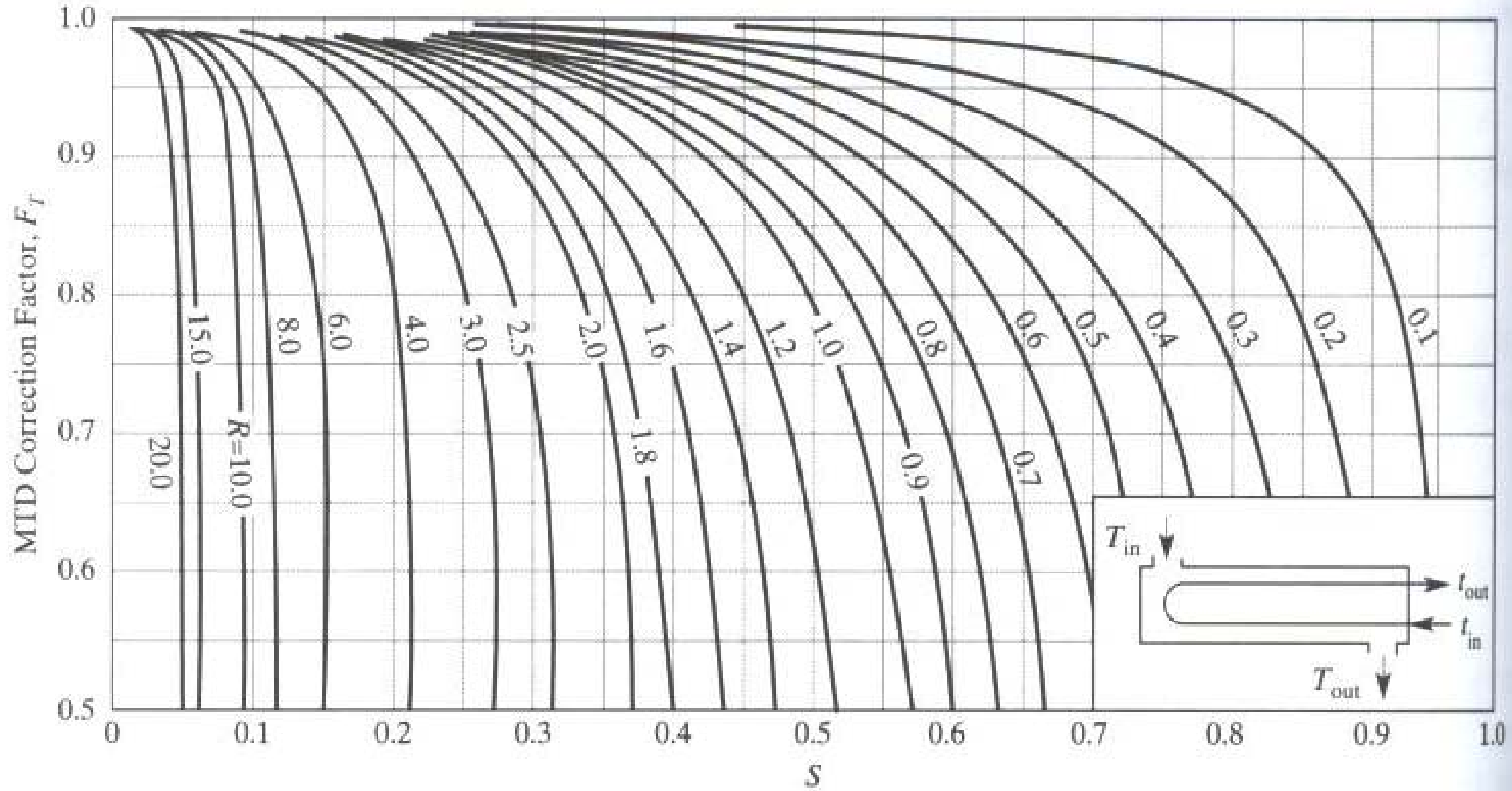


Figure 13.14 Temperature-driving-force correction factor for 1-2 shell-and-tube exchanger. [Adapted from Bowman et al., *Trans. ASME*, **62**, 283 (1940).]

F_T not changed drastically from 1shell pass-2 tube pass (1-2) when additional tube passes are added (i.e. 1-4, 1-6, or 1-8

F_T charts available for multiple-shell-pass and crossflow exchangers

F_T summary

When shell and tube heat exchangers with multiple-tube passes, or multiple shell and tube passes are used, the flow direction of the two fluids are combinations of countercurrent and cocurrent flow. The resulting ΔT_m for given values of ΔT_1 and ΔT_2, based on countercurrent flow, is less than standard ΔT_{LM}. Hence, $F_T < 1$.

When F_T is unsatisfactory, a multiple-shell-pass heat exchanger is used. The more shell passes, the higher is the value of F_T. For a given number of shell passes, the number of tube passes has very little effect on F_T

Heat Transfer Coefficients

Overall heat transfer coefficient, U, based on area of inner wall or outer wall.

Sum-of-thermal-resistances formulas

$$U_o = \frac{1}{R_{fo} + \frac{1}{h_o} + \frac{t_w A_o}{k_w A_m} + \frac{A_o}{h_i A_i} + \frac{A_o}{A_i} R_{fi}}$$

conduction; film; fouling

$$A_m = \frac{\pi L \left(D_o - D_i \right)}{ln\left(D_o / D_i \right)}$$

$$U_i = \frac{1}{\frac{A_i}{A_o} R_{fo} + \frac{A_i}{A_o h_o} + \frac{t_w A_i}{k_w A_m} + \frac{1}{h_i} + R_{fi}}$$

Heat Transfer Coefficients

For preliminary design, use reasonable estimates for U, see Table 13.5

For detailed equipment design, use published correlations

Categories for Correlations

Turbulent Flow, Straight, Smooth Ducts, Pipes, Tubes, Circular Xsection

Turbulent Flow, Annular Region Between Concentric Pipes

Turbulent Flow on Shell Side

Laminar Flow

Condensation

Boiling

Compact Exchangers

Heat Transfer Coefficients

Example of correlation

Turbulent Flow on Shell Side

$$Nu = \frac{h_o D}{k_b} = C \cdot Re^n \cdot Pr^{1/3} \cdot \left(\frac{\mu_b}{\mu_w} \right)^{0.14}$$

viscosity correction

Donohue (1949)

D : tube o.d.

C : 0.2

$$Re = \frac{D \cdot G}{\mu_b}$$

$$Pr = \frac{C_{Pb} \mu_b}{k_b}$$

C_{Pb} : heat capacity of bulk fluid

k_b : thermal conductivity of bulk fluid

G : mass velocity, average of two:
1) parallel in baffle window
2) normal to centerline tubes

n : 0.6

μ_b : viscosity of bulk fluid

μ_w : viscosity of fluid at wall surface temperature

Heat Exchanger Design Using HYSYS

Example 13.7

Existing 2-8 heat exchanger, carbon steel

shell ID: 39 in

1024 3/4-in 14 BWG, 16-ft-long tubes, 1-inch square pitch

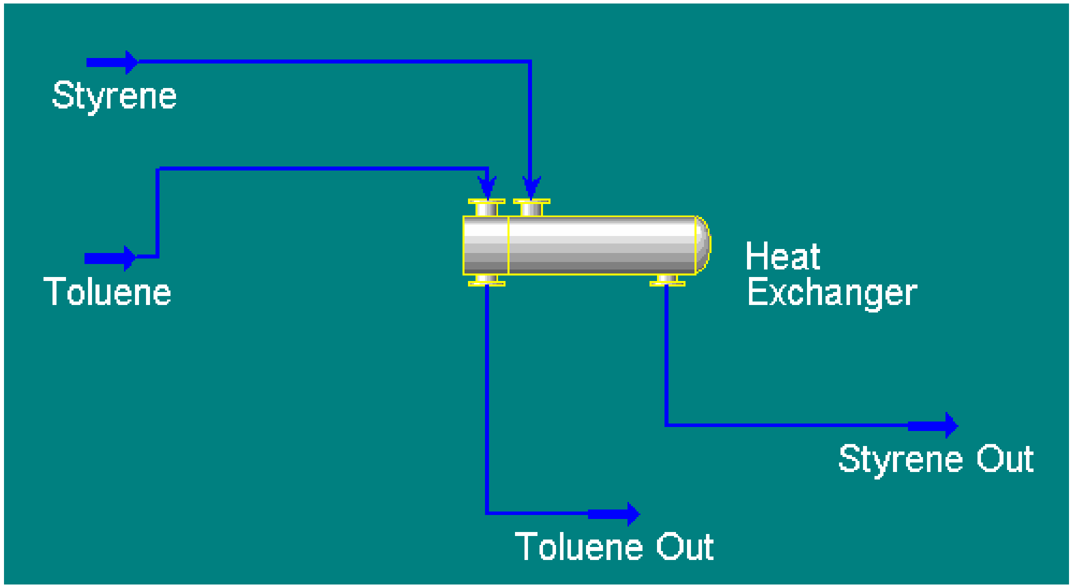
38 segmental baffles, baffle cut: 25%

Tube inlet/outlet nozzles: 4-in, Sch 40 pipe; Shell 2.5 in, Sch 40 pipe

fouling factors: 0.002 (hr-ft2-°F)/BTU on each side

tube side: Toluene feed stream, 125000 lb/hr, 100°F, 90 psia

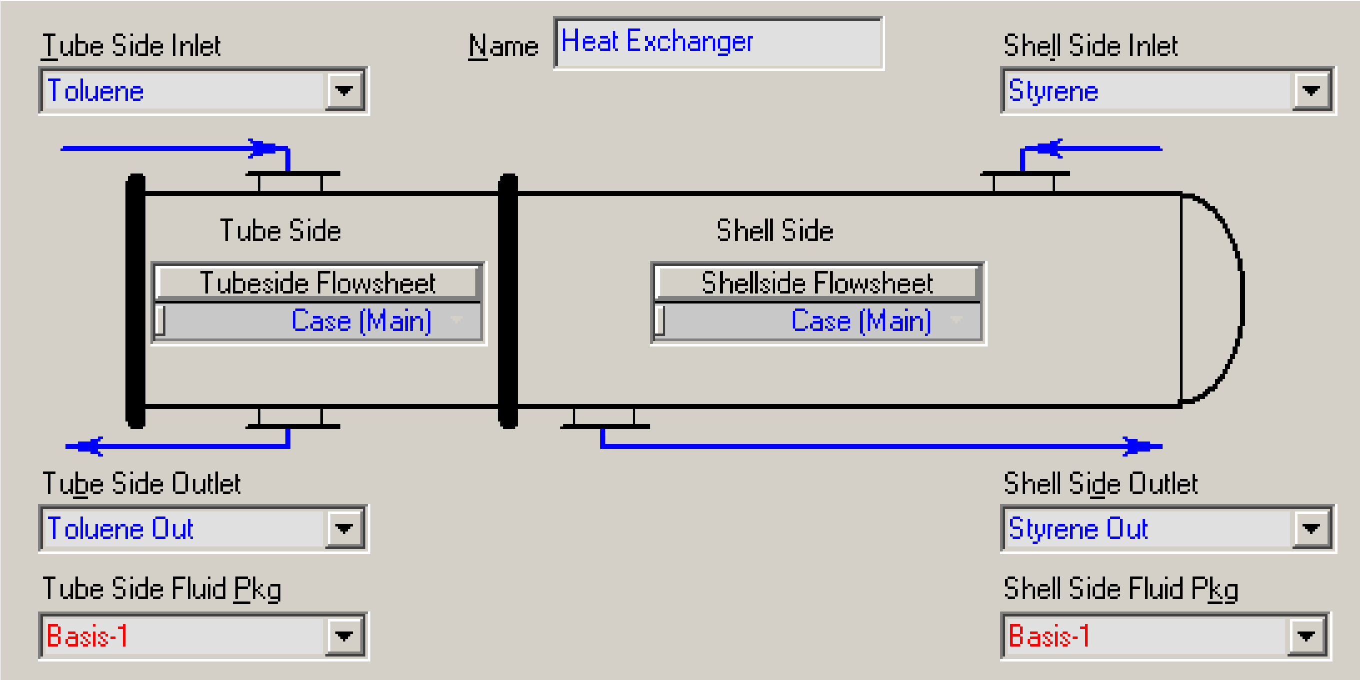
shell side: Styrene product stream, 150000 lb/hr, 300°F, 50 psia

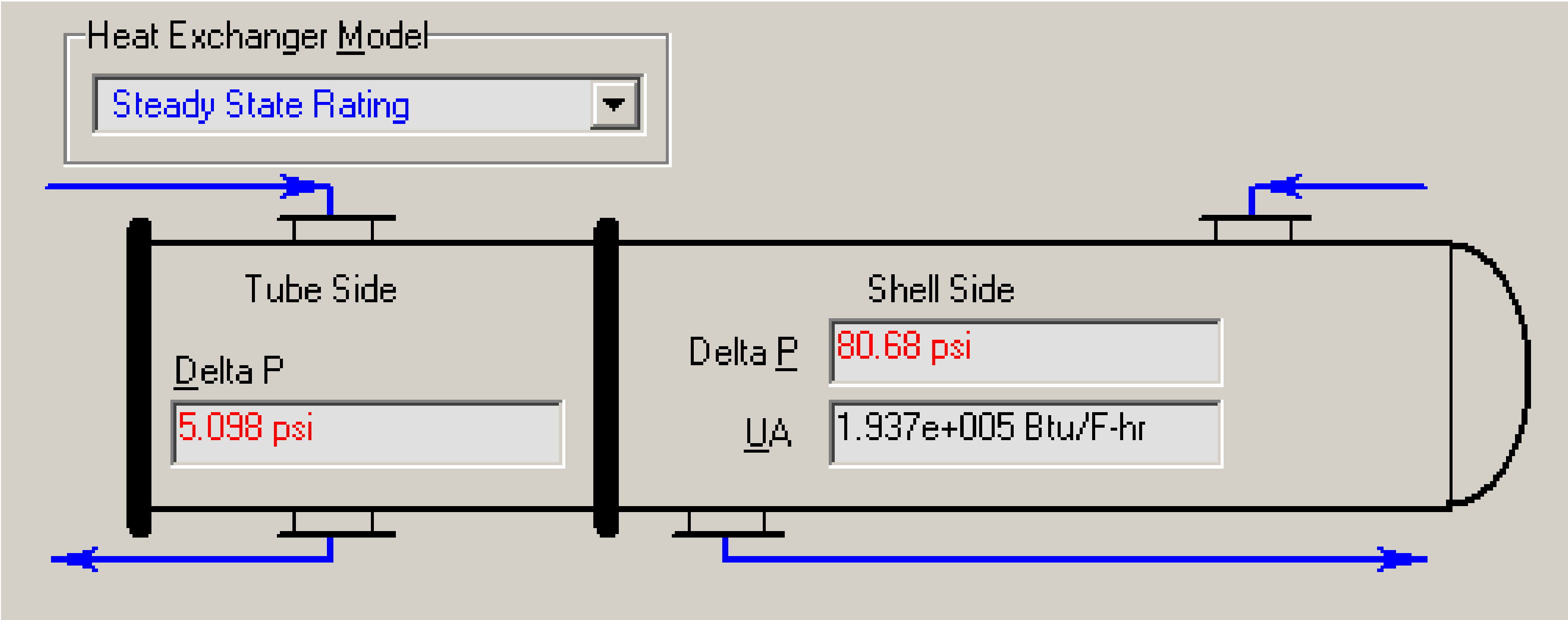


Determine:
(1) Exit Ts
(2) Q
(3) ΔP

Heat Exchanger Design Using HYSYS

Example 13.7





Heat Exchanger Design Using HYSYS

Example 13.7

Configuration

Number of Shell Passes	2
Number of Shells in Series	1
Number of Shells in Parallel	1
Tube Passes per Shell	8
Exchanger Orientation	Horizontal
First Tube Pass Flow Direction	Counter
Elevation (Base)	0.0000

TEMA Type: A F L

Calculated Information

Shell HT Coeff [Btu/hr-ft2-F]	165.5
Tube HT Coeff [Btu/hr-ft2-F]	224.0
Overall U [Btu/hr-ft2-F]	60.22
Overall UA [Btu/F-hr]	1.937e+005
Shell DP [psi]	80.68
Tube DP [psi]	5.098
Heat Trans. Area per Shell [ft2]	3217
Tube Volume per Shell [ft3]	30.48
Shell Volume per Shell [ft3]	82.47

Shell and Tube Bundle Data

Shell Diameter [in]	39.00
Number of Tubes per Shell	1024
Tube Pitch [in]	1.000
Tube Layout Angle	Square (90 degrees)
Shell Fouling [F-hr-ft2/Btu]	0.002000

Shell Baffles

Shell Baffle Type	Single
Shell Baffle Orientation	Vertical
Baffle Cut (%Area) [%]	25.00
Baffle Spacing [in]	5.00

Dimensions

OD [in]	0.750
ID [in]	0.584
Tube Thickness [in]	0.083
Tube Length [ft]	16.000

Tube Properties

Tube Fouling [F-hr-ft2/Btu]	0.002000
Thermal Cond. [Btu/hr-ft-F]	30.00
Wall Cp [Btu/lb-F]	0.107
Wall Density [lb/ft3]	490.7

Heat Exchanger Design Using HYSYS

Example 13.7

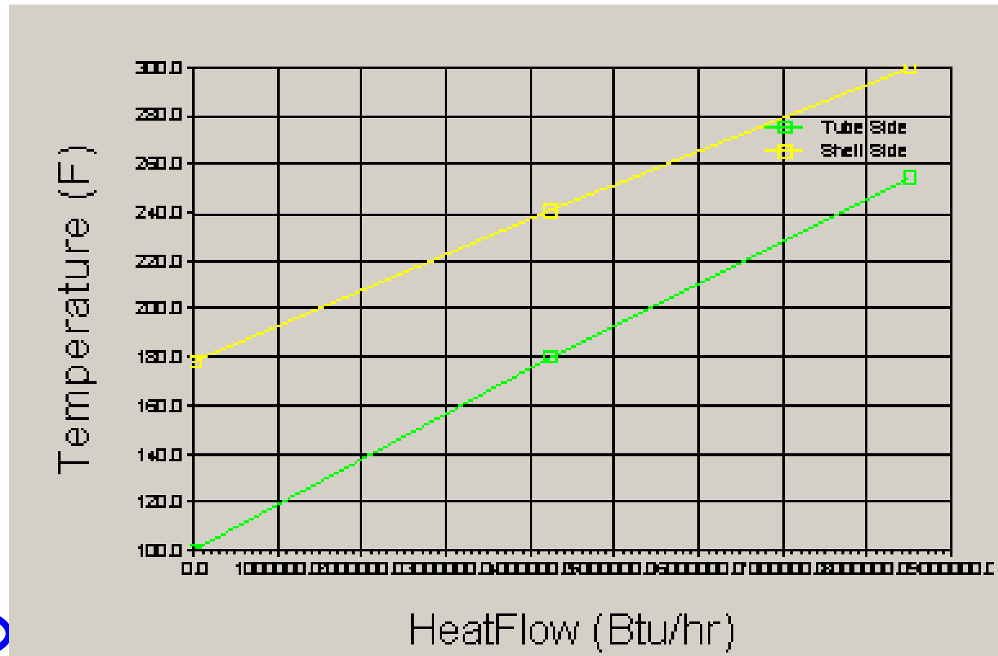
Name	Fluid 1	Fluid 1 Out	Fluid 2	Fluid 2 Out
Vapour	1.0000	0.8922	0.0000	0.0000
Temperature [F]	200.0	100.9	90.00	190.0
Pressure [psia]	35.00	30.00	100.0	90.00
Molar Flow [lbmole/hr]	110.0	110.0	90.00	90.00
Mass Flow [lb/hr]	6912	6912	4146	4146
Std Ideal Liq Vol Flow [barrel/day]	529.1	529.1	356.7	356.7
Molar Enthalpy [Btu/lbmole]	-5.088e+004	-5.399e+004	-1.191e+005	-1.153e+005
Molar Entropy [Btu/lbmole-F]	34.18	29.35	8.568	19.56
Heat Flow [Btu/hr]	-5.596e+006	-5.939e+006	-1.072e+007	-1.038e+007

Overall Performance

Duty	8.512e+06 Btu/hr
Heat Leak	0.000e-01 Btu/hr
Heat Loss	0.000e-01 Btu/hr
UA	1.94e+05 Btu/F-hr
Min. Approach	45.703 F
LMTD	43.94 F

Detailed Performance

UA Curvature Error	0.0000 Btu/F-hr
Hot Pinch Temp	300.0000 F
Cold Pinch Temp	254.2971 F
Ft Factor	0.7259
Uncorrected LMTD	60.524 F



Heat Exchanger Design Procedure

(assuming inlet T, P, composition, phase, and flow rate are known for both streams and an exit T or equivalent spec is given for one of two streams)

1.) If utility stream is needed, select it from Table 13.1, together with its entering and leaving T
2.) Decide which stream is tube side/shell side
3.) Estimate shell & tube pressure drops (Heuristic 31)
4.) Calculate heat duty and remaining exiting conditions of streams from overall energy balance using eqn 13.1
5.) If a heating/cooling utility is used, calculate its flow rate from an overall energy balance (eqn 13.1)
6.) Assume a 1-1 heat exchanger (1 shell pass/1 tube pass)
7.) Check for valid solution and that a reasonable ΔT exists on both sides; if a phase change occurs, a heating/cooling curve is calculated to make sure that no temperature crossover is computed
8.) Make a preliminary estimation of the heat exchanger area, A, using an assumed overall heat transfer coefficient, U, from Table 13.5
9.) Compute the mean driving force for heat transfer using heating/cooling curves or eqn (13.3)

10.) Use eqn 13.7 to estimate A with $F_T = 1$; if A > 8,000 ft^2, multiple exchangers of the same area are used in parallel

11.) From the estimated A, preliminary estimates are made of the exchanger geometry

12.) A tube side velocity in the range of 1 to 10 ft/s is selected (4 ft/s typical)

13.) Calculate total tube inside cross sectional area

14.) Select a tube size (e.g. ¾ in. O.D. 14 BWG)

15.) Calculate number of tubes per pass per exchanger

16.) Select a tube length (e.g. 16 ft typical) and then calculate the number of tube passes per exchanger

17.) Adjust the tube side velocity and tube length to obtain an integer number of tube passes

18.) If more than one tube pass is necessary, use Figures 13.14 – 13.16 to correct the log-mean temperature driving force; this may require more than one shell pass (see Example 13.5)

19.) Select a tube sheet layout from Table 13.6 and a baffle design & spacing for the shell side

20.) This completes the preliminary design of the heat exchanger

21.) A revised design is made by using the geometry of the preliminary design to estimate an overall heat transfer coefficient from the calculated individual coefficients and estimated fouling factors, as well as ΔP; the entire procedure is iterated to achieve satisfactory tolerance