Equipment Fundamentals: Heat Exchangers

Chapter 3

COLORADO SCHOOL OF MINES

Topics

Equipment – heat exchangers

- Combines information about fluid flow & heat transfer across internal boundaries
- Considerations
    - When do I need to know the specifics of the heat exchange configuration?
    - How is the heat transfer coefficient related to the outlet temperatures?
    - What is an approach temperature?

Fundamentals of heat transfer & exchange

- Heat transfer across boundaries
    - Conduction
    - Convection
    - Radiation
- Coupled with internal energy changes
    - Sensible heat effects
    - Phase change

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Topics

Fundamentals of heat transfer & exchange

- Heat transfer across boundaries
  - Conduction
  - Convection
  - Radiation
- Coupled with internal energy changes
  - Sensible heat effects
  - Phase change
- Area-averaged temperature difference

Equipment – heat exchangers

- Combines information about fluid flow & heat transfer across internal boundaries
- Considerations
  - When do I need to know the specifics of the heat exchange configuration?
  - How is the heat transfer coefficient related to the outlet temperatures?
  - What is an approach temperature?

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Fundamentals

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Heat Transfer – Modes of heat transfer

Conduction

- Flow of heat through material with no bulk movement of the material itself
- Usually thought of through solid, but can also be through a stagnant fluid
- For a flat sold:

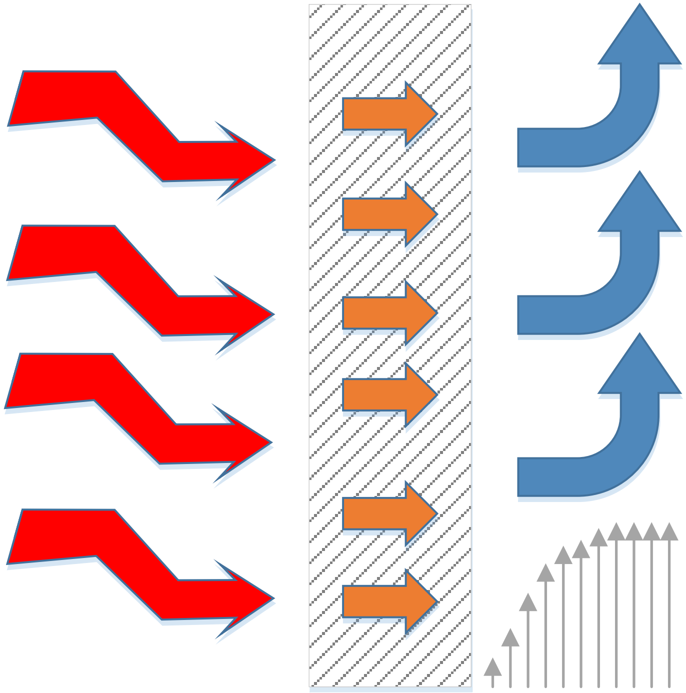
$$\frac{\dot{Q}}{A} = k\,\frac{T_{hot} - T_{cold}}{\Delta x}$$

- Through a circular pipe:

$$\frac{\dot{Q}}{L} = \frac{2\pi}{\ln\left(D_o / D_i\right)}\,k\left(T_{hot} - T_{cold}\right)$$

- Through a sphere:

$$\dot{Q} = \frac{2\pi}{\dfrac{1}{D_i} + \dfrac{1}{D_o}}\,k\left(T_{hot} - T_{cold}\right)$$



Heat Transfer – Modes of heat transfer

Convection

- Flow of heat associated with fluid movement – natural & forced convection

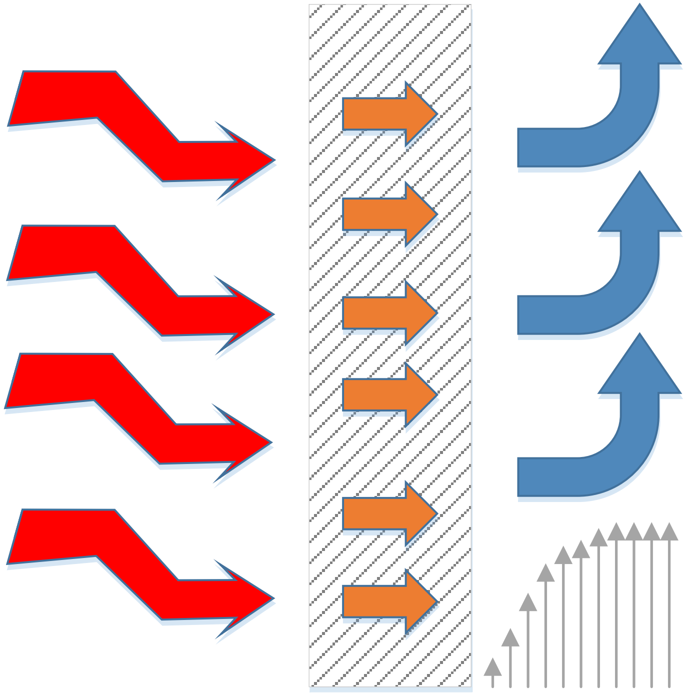
$$\frac{\dot{Q}}{A} = h\left(T_{hot} - T_{cold}\right)$$

Radiation

- Heat transferred via electromagnetic radiation

$$\frac{\dot{Q}}{A} = \varepsilon\sigma\left(T_{hot}^4 - T_{cold}^4\right)$$

$$= \left[\varepsilon\sigma\left(T_{hot}^2 + T_{cold}^2\right)\left(T_{hot} + T_{cold}\right)\right]\left(T_{hot} - T_{cold}\right)$$



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Heat Exchangers – Some Basics

Focus is on the system to have heat flow from the hot fluid(s) to the cold fluid(s) usually without direct contact

- Use bulk flow parameters to relate the heat conduction across the flow barrier to the change in energy of the hot & cold fluids
- Account for the series of resistances to heat transfer between the hot & cold fluids

Heat exchangers

- Heat to & from flowing fluids through impermeable barrier(s)
- Driving force for heat through barriers is the temperature difference between the two fluids on opposite sides of the barrier
- Relate the heat effects in the flowing fluids to the change in enthalpy
  - Often this can be related to the difference in the inlet & outlet temperatures for the fluids

$$\dot{Q}_H = \dot{m}_H\left(\hat{H}_{H,in} - \hat{H}_{H,out}\right) \quad \Rightarrow \quad \dot{Q}_H = \dot{m}_H\hat{C}_{p,H}\left(T_{H,in} - T_{H,out}\right) \quad \text{for constant } \hat{C}_{p,H}$$

$$\dot{Q}_C = \dot{m}_C\left(\hat{H}_{C,out} - \hat{H}_{C,in}\right) \quad \Rightarrow \quad \dot{Q}_C = \dot{m}_C\hat{C}_{p,C}\left(T_{C,out} - T_{C,in}\right) \quad \text{for constant } \hat{C}_{p,C}$$

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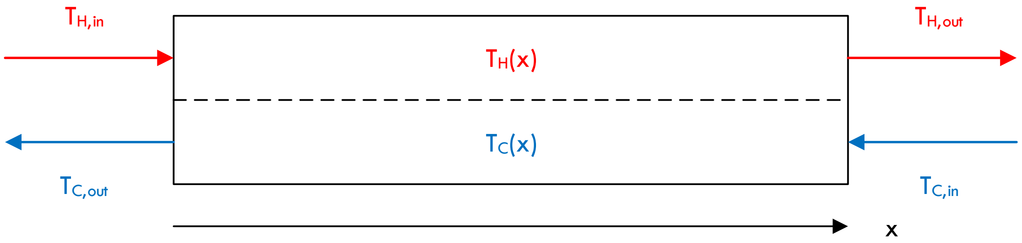
Heat Exchangers – Some Basics

- Relate the heat across the barrier to the temperature difference across the barrier



$$\frac{d\left(\dot{Q}/L\right)}{dx} = U\left(T_h - T_c\right) \Rightarrow \dot{Q} = (UA)\left[\overline{T_h - T_c}\right]_{\text{AREA AVERAGED}}$$

- It can be shown that for many typical configurations the AREA AVERAGED temperature difference is the LMTD (Log Mean Temperature Difference)

$$\dot{Q} = (UA)\overline{(\Delta T)}_{LM} \quad \text{where} \quad \overline{(\Delta T)}_{LM} \equiv \frac{\left(T_{H,0} - T_{C,0}\right) - \left(T_{H,1} - T_{C,1}\right)}{\ln\left(\dfrac{T_{H,0} - T_{C,0}}{T_{H,1} - T_{C,1}}\right)}$$

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Heat Exchangers – Some Basics

LMTD is a prescribed calculation – calculating the LMTD from the procedure is always correct.

$$LMTD=\frac{(\Delta T)_1-(\Delta T)_2}{\ln\left[\dfrac{(\Delta T)_1}{(\Delta T)_2}\right]} \quad \text{and} \quad \lim_{(\Delta T)_1\to(\Delta T)_2}(LMTD)=\frac{(\Delta T)_1+(\Delta T)_2}{2}$$

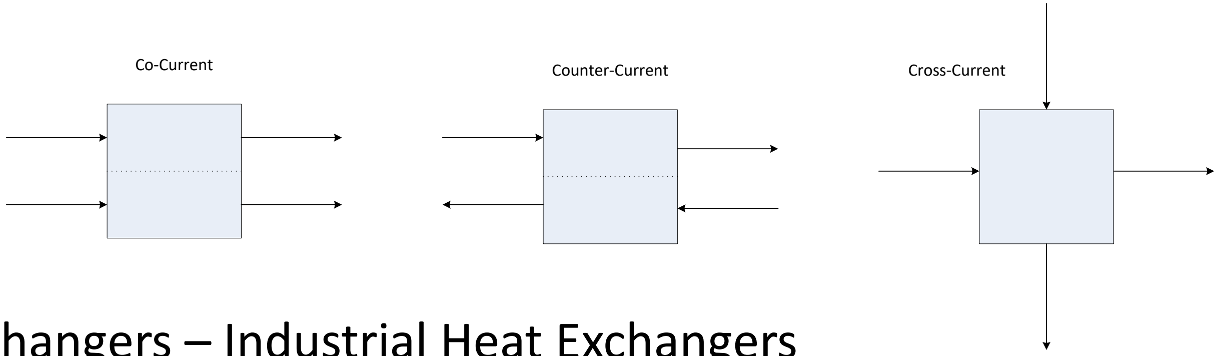
LMTD is appropriate for use as the area averaged temperature difference when temperature vs. heat released/absorbed is a straight line

- 1-1 co-current & counter-current flow and …
- Both hot & cold sides have a constant heat or …
- Only pure component phase change on one side or the other (no subcooling and/or superheating)

Heat Transfer – Some Basics

Heat exchangers – Co-Current vs. Counter-Current vs. Cross-Current flows

- Counter-current flow allows the outlet temperatures to approach more closely to the inlet temperature of the other fluid
- Cross-current flow is complicated & requires knowledge of the actual flow patterns



Heat exchangers – Industrial Heat Exchangers

- Industrial heat exchangers have a combination of heat transfer through multiple barriers and a combination of counter-current & co-current flow
  - LMTD must be "corrected" to give the actual area-averaged temperature difference (i.e., driving force) – this is the source for one type of "F" factor

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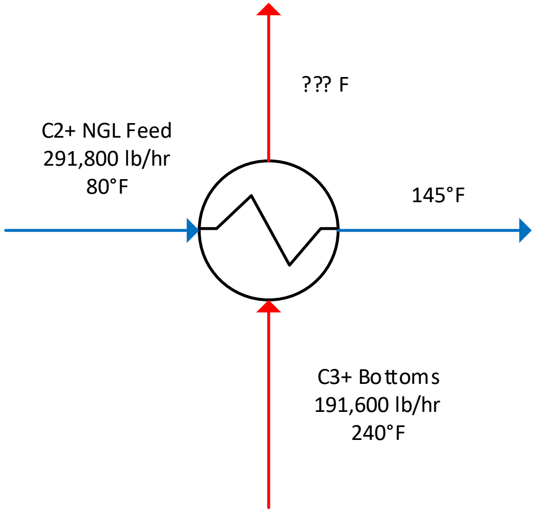
Heat Exchanger – Example 1

Heat 291,800 lb/hr cold C2+ NGL feed from 80°F to 105°F using 191,600 lb/hr hot C3+ bottoms @ 240°F

- Assume only sensible heat effects
  - C2+ NGL feed heat capacity – 0.704 Btu/lb F
  - C3+ Bottoms heat capacity – 0.830 Btu/lb F

Determine

- C3+ Bottoms outlet temperature
- Exchanger duty
- (UA) for the exchanger



Heat Exchanger – Example 1

Exchanger duty & C3+ Bottoms outlet temperature determined from energy balance around exchanger

$$\dot{Q} = \dot{m}_c \hat{C}_{p,c} \left(T_{c,out} - T_{c,in} \right) = (291800)(0.704)(145-80) = 13{,}353{,}000 \text{ Btu/hr}$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{\dot{m}_h \hat{C}_{p,h}} = 240 - \frac{13353000}{(191600)(0.828)} = 155.8^\circ\text{F}$$

Determination of UA requires configuration information

- 1-1 counter-current flow

$$(\Delta T)_{LMTD} = \frac{(240-145)-(155.8-80)}{\ln\left(\dfrac{240-145}{155.8-80}\right)} = 85.1^\circ\text{F}$$

$$UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{13353000}{85.1} = 157{,}000 \frac{\text{Btu}}{\text{hr}^\circ\text{F}}$$

1-1 co-current flow

$$(\Delta T)_{LMTD} = \frac{(240-80)-(155.8-145)}{\ln\left(\dfrac{240-80}{207.7-105}\right)} = 55.4^\circ\text{F}$$

$$UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{13353000}{55.4} = 241{,}000 \frac{\text{Btu}}{\text{hr}^\circ\text{F}}$$

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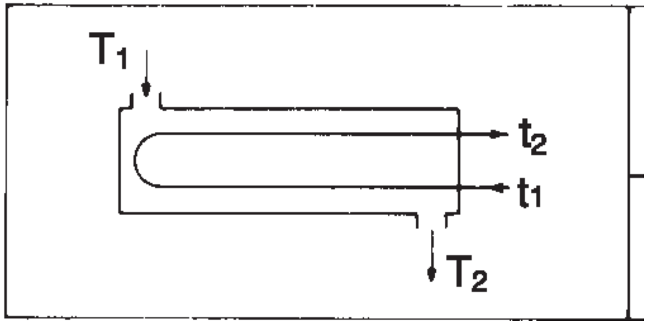
Heat Exchanger – Example 1

Determination of UA requires configuration information

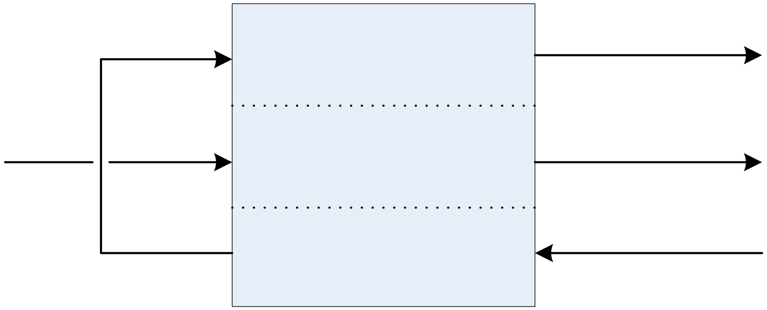
- 1-2 (1 shell & 2 tube passes) combines both counter & co-current flow

  The fluid in the shell pass transfers heat separately to the two tube banks



Ref: *GPSA Data Book*, 13th ed.

1-2 Co & Counter-Flow



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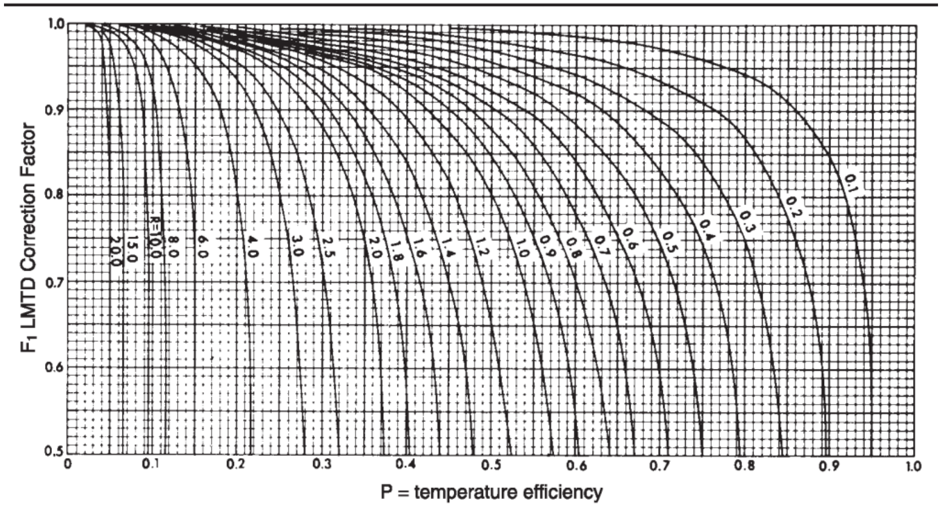
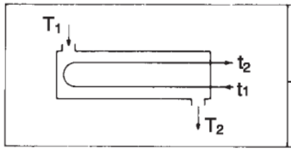
Heat Exchanger – Example 1

- 1-2 exchanger calculations require a configuration correction to relate temperatures to the UA
- Does not include crossflow effects across the tubes

FIG. 9-4

LMTD Correction Factor (1 shell passes; 2 or more tube passes)





LMTD Correction Factor

1 shell pass 2 or more tube passes

$P = \frac{t_2 - t_1}{T_1 - t_1}$ $R = \frac{T_1 - T_2}{t_2 - t_1}$ CMTD = (LMTD)(F_1)

$$F_1 = \frac{\sqrt{R^2+1}\,\ln\left[\frac{1-P}{1-RP}\right]}{(R-1)\ln\left[\frac{2-P\left(R+1-\sqrt{R^2+1}\right)}{2-P\left(R+1+\sqrt{R^2+1}\right)}\right]}$$

Ref: Kern, *Process Heat Transfer,* McGraw-Hill, 1965

Ref: *GPSA Data Book*, 13th ed.

Heat Exchanger – Example 1

- 1-2 exchanger

$$(\Delta T)_{LMTD} = \frac{(240-145)-(155.8-80)}{\ln\left(\dfrac{240-145}{155.8-80}\right)} = 85.1^\circ\text{F}$$

$$P = \frac{145-80}{240-80} = 0.4$$

$$R = \frac{240-155.8}{145-80} = 1.3$$

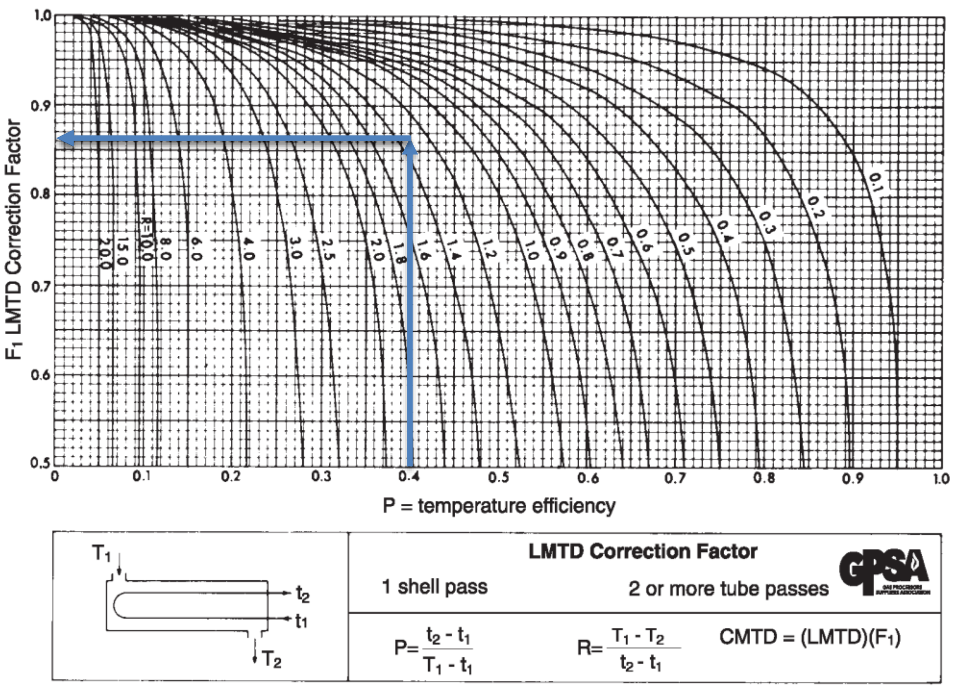
$$F_2 = 0.86 \text{ (from chart)}$$

$$UA = \frac{\dot{Q}}{F_2(\Delta T)_{LMTD}}$$

$$= \frac{13353000}{0.86(85.1)} = 182{,}500 \frac{\text{Btu}}{\text{hr}^\circ\text{F}}$$

FIG. 9-4

LMTD Correction Factor (1 shell passes; 2 or more tube passes)



$P = \frac{t_2 - t_1}{T_1 - t_1}$ $R = \frac{T_1 - T_2}{t_2 - t_1}$ CMTD = (LMTD)(F_1)

Ref: *GPSA Data Book*, 13th ed.

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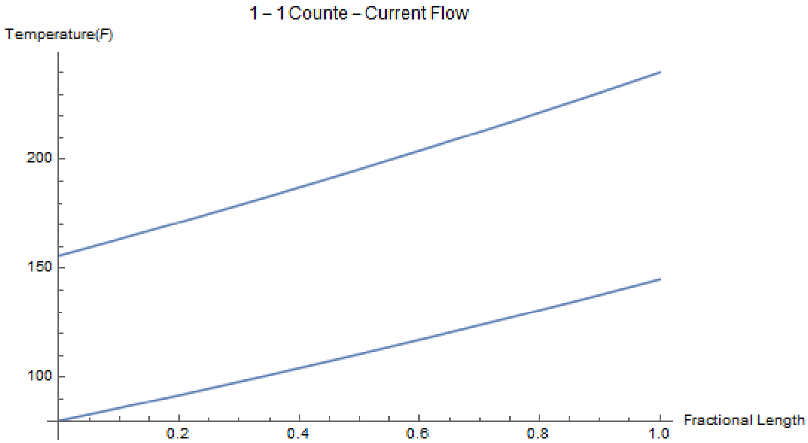
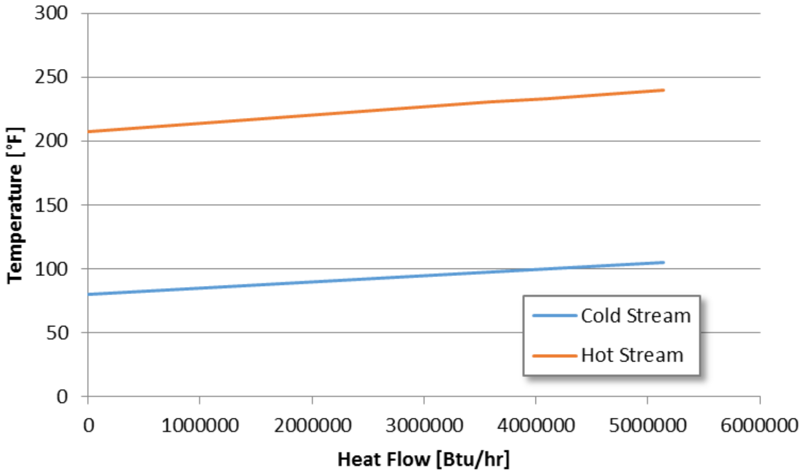
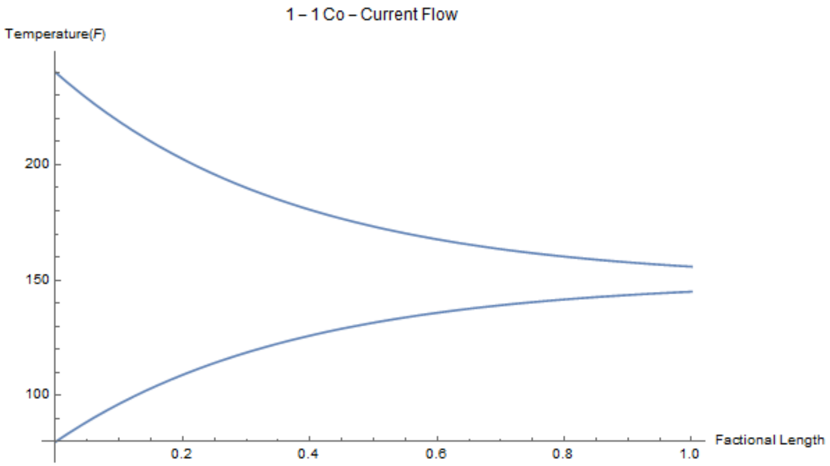
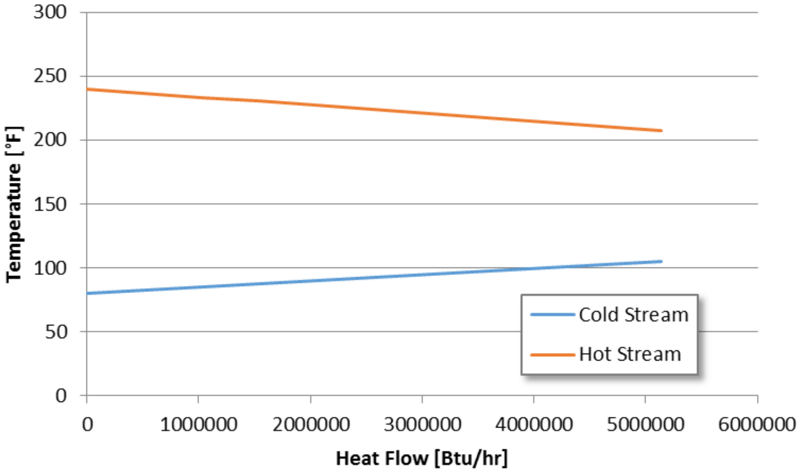
Heat Exchanger – Example 1









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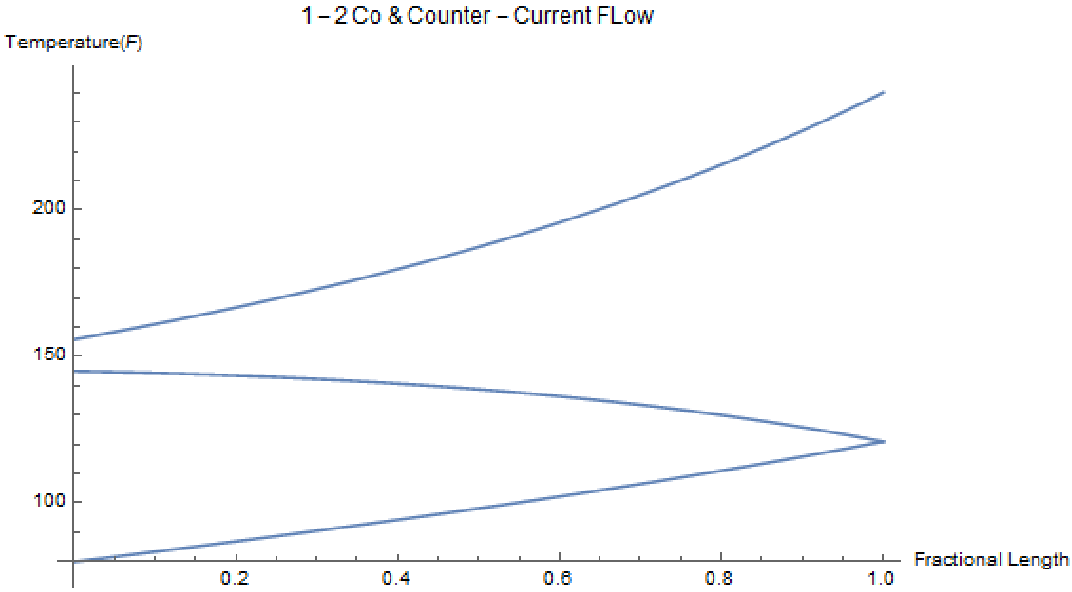
Heat Exchanger – Example 1

Representation of temperature profiles with combined flow becomes more complicated.



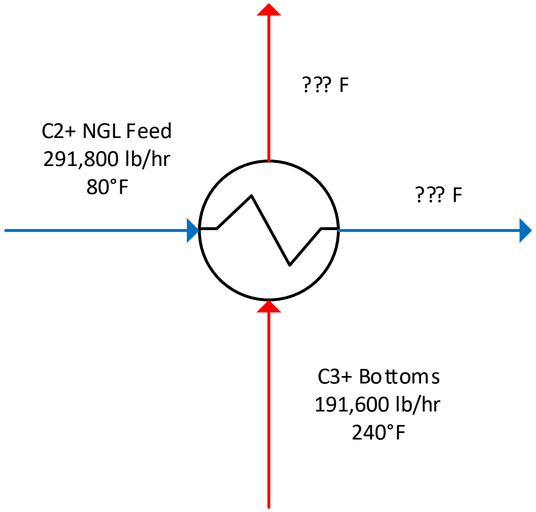
Heat Exchanger – Example 2

Heat 291,800 lb/hr cold C2+ NGL feed starting from 80°F using 191,600 lb/hr hot C3+ bottoms @ 240°F. Drive the exchanger to a 10°F approach temperature

- For 1-1 Counter-Current flow, what are the outlet temperatures?
  - The C2+ NGL Feed is either heated to 230°F (approach on the hot inlet side) or …
  - the C3+ Bottoms is cooled to 90°F (approach on the cold inlet side)
- Assume only sensible heat effects
  - C2+ NGL feed heat capacity – 0.704 Btu/lb F
  - C3+ Bottoms heat capacity – 0.830 Btu/lb F

Determine

- The outlet temperature that is not controlled by the approach
- Exchanger duty
- (UA) for the exchanger



Heat Exchanger – Example 2

Exchanger duty & “other” outlet temperature determined from energy balance around exchanger

- If the hot side inlet has the approach temperature

$$T_{c,out} = T_{h,in} - 10 = 230^{\circ}\text{F}$$

$$Q = \dot{m}_c \hat{C}_{p,c}\left(T_{c,out} - T_{c,in}\right) = (291800)(0.704)(230-80) = 30{,}814{,}000 \text{ Btu/hr}$$

$$T_{h,out} = T_{h,in} - \frac{Q}{\dot{m}_h \hat{C}_{p,h}} = 240 - \frac{30814000}{(191600)(0.828)} = 45.8^{\circ}\text{F}$$

This has a <u>temperature crossover</u> – this is <u>not</u> the controlling side!

- If the cold side inlet has the approach temperature:

$$T_{h,out} = T_{c,in} + 10 = 90^{\circ}\text{F}$$

$$Q = \dot{m}_h \hat{C}_{p,h}\left(T_{h,in} - T_{h,out}\right) = (191600)(0.828)(240-90) = 23{,}797{,}000 \text{ Btu/hr}$$

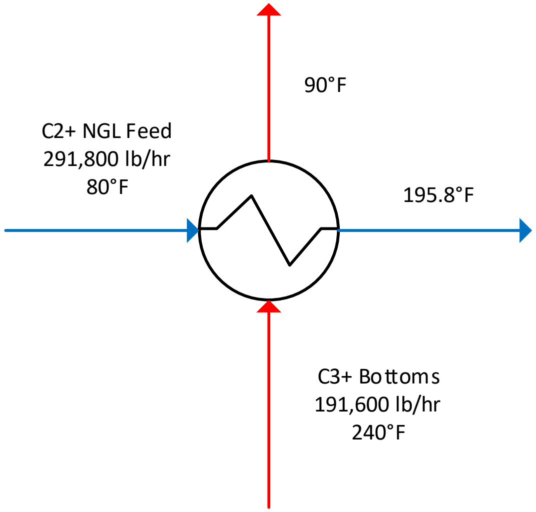
$$T_{c,out} = T_{c,in} + \frac{Q}{\dot{m}_c \hat{C}_{p,c}} = 80 + \frac{23797000}{(291800)(0.704)} = 195.8^{\circ}\text{F}$$

Heat Exchanger – Example 2

Determination of UA for 1-1 Counter-Current flow using the cold & hot outlet temperatures of 195.8°F & 90°F, respectively

$$(\Delta T)_{LMTD} = \frac{(240-195.8)-(90-80)}{\ln\left(\frac{240-195.8}{90-80}\right)} = 23.0^{\circ}\text{F}$$

$$UA = \frac{Q}{(\Delta T)_{LMTD}} = \frac{23797000}{23.0} = 1{,}035{,}000 \frac{\text{Btu}}{\text{hr}^{\circ}\text{F}}$$



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Heat Exchanger – Example 3

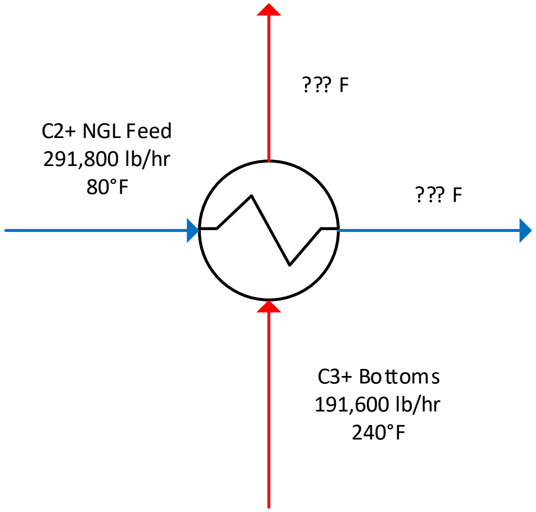
Heat 291,800 lb/hr cold C2+ NGL feed from 80°F using 191,600 lb/hr hot C3+ bottoms @ 240°F. 1-1 Counter-Current heat exchanger from Example #1 designed with 25% excess heat transfer area (UA=196,000 Btu/hr °F)

- Assume only sensible heat effects
  - C2+ NGL feed heat capacity – 0.704 Btu/lb F
  - C3+ Bottoms heat capacity – 0.830 Btu/lb F

Determine

- Both outlet temperatures
- Exchanger duty

Need to couple all three equations relating heat exchanger duty find the three unknowns



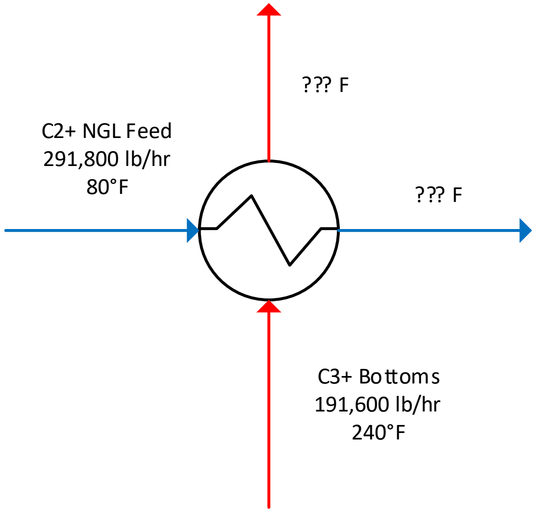
Heat Exchanger – Example 3

Even though it looks like you'll have to solve the three equations in an iterative manner, it can be shown that the heat transfer duty is:

$$\dot{Q} = \frac{(\Lambda - 1)\left(T_{H,in} - T_{C,in}\right)}{\dfrac{\Lambda}{\dot{m}_C C_{p,C}} - \dfrac{1}{\dot{m}_H C_{p,H}}} \quad \text{where } \Lambda \equiv \exp\left[UA\left(\frac{1}{\dot{m}_C C_{p,C}} - \frac{1}{\dot{m}_H C_{p,H}}\right)\right] \quad \text{for } \dot{m}_C C_{p,C} \neq \dot{m}_H C_{p,H}$$

In the limiting case where the mCp terms are equal:

$$\dot{Q} = \frac{T_{H,in} - T_{C,in}}{1 + \dfrac{UA}{\dot{m}C_p}} \quad \text{if } \dot{m}_C C_{p,C} = \dot{m}_H C_{p,H}$$



Heat Exchanger – Example 3

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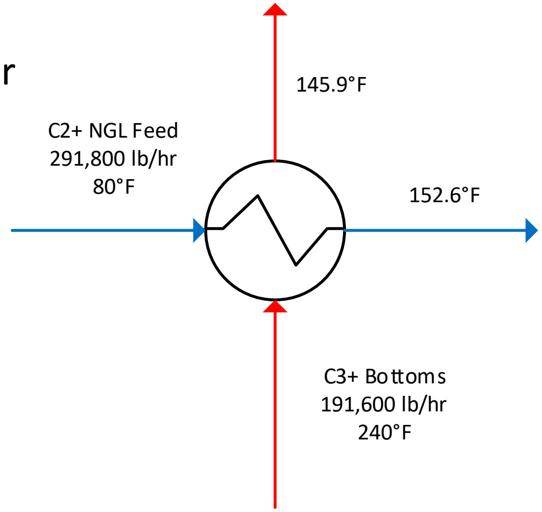
So:

$$\Lambda=\exp\left[(196000)\left(\frac{1}{(291800)(0.704)}-\frac{1}{(191600)(0.828)}\right)\right]=0.7548$$

$$\dot{Q}=\frac{(0.7548-1)(240-80)}{\dfrac{0.7548}{(291800)(0.704)}-\dfrac{1}{(191600)(0.828)}}=14{,}924{,}000\text{ Btu/hr}$$

$$T_{h,out}=T_{h,in}-\frac{\dot{Q}}{\dot{m}_h\hat{C}_{p,h}}=240-\frac{14924000}{(191600)(0.828)}=145.9^{\circ}\text{F}$$

$$T_{c,out}=T_{c,in}+\frac{Q}{\dot{m}_c\hat{C}_{p,c}}=80+\frac{14924000}{(291800)(0.704)}=152.6^{\circ}\text{F}$$



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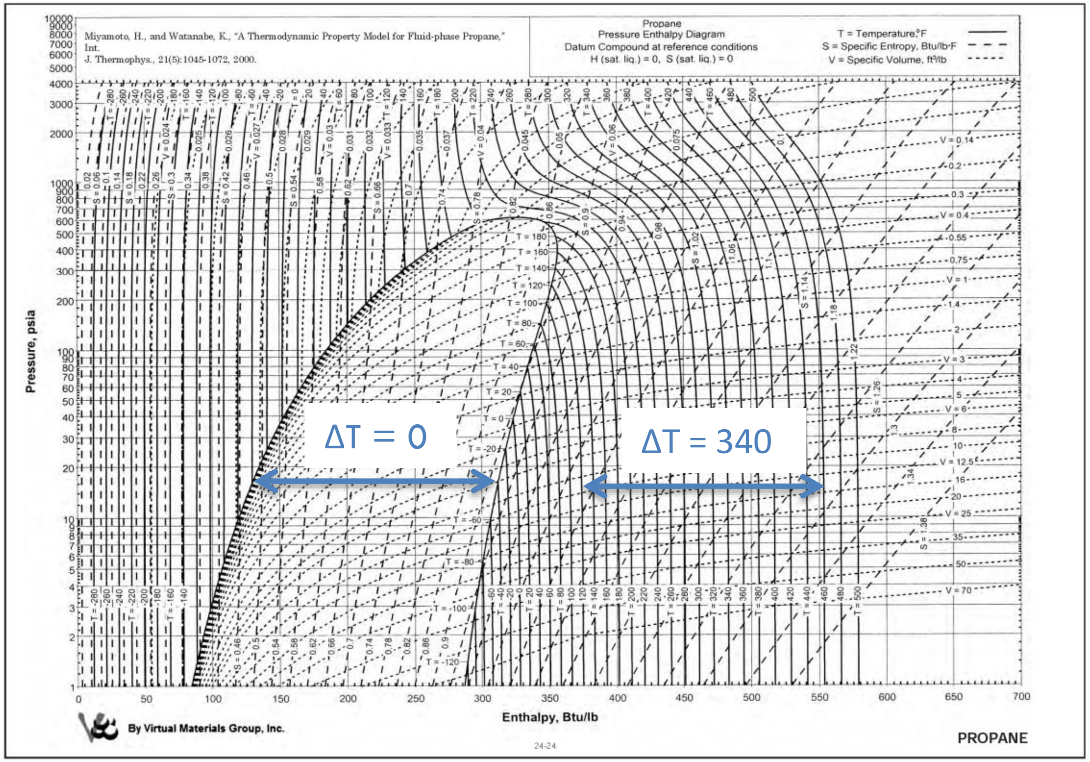
Heat Exchange with Phase Change

Can get significant heat exchange with little to no change in temperature

FIG. 24-20 Propane P-H Diagram



Ref: *GPSA Data Book*, 13th ed.

Heat Exchanger – Example 4

Condense 10,000 lb/hr saturated vapor propane at 120°F using 95°F air. Figure a 10°F approach temperature, so heat the air up to 110°F.

- Needed physical properties
  - Air heat capacity – 0.24 Btu/lb F
  - Propane heat of vaporization @ 120°F – 1,236 Btu/lb

Determine

- Exchanger duty
- Flow rate of air needed
- Exchanger UA

Heat Exchanger – Example 4

Duty determined from the propane energy balance. No sensible heat effect.

$$\dot{Q} = \dot{m}_h \lambda_{vap} = (10000)(1236) = 12{,}360{,}000 \text{ Btu/hr}$$

Air flowrate from its energy balance:

$$\dot{m}_c = \frac{\dot{Q}}{\hat{C}_{p,c}\left(T_{c,out} - T_{c,in}\right)} = \frac{12{,}360{,}000}{(0.24)(110-95)} = 3{,}430{,}000 \text{ lb/hr}$$

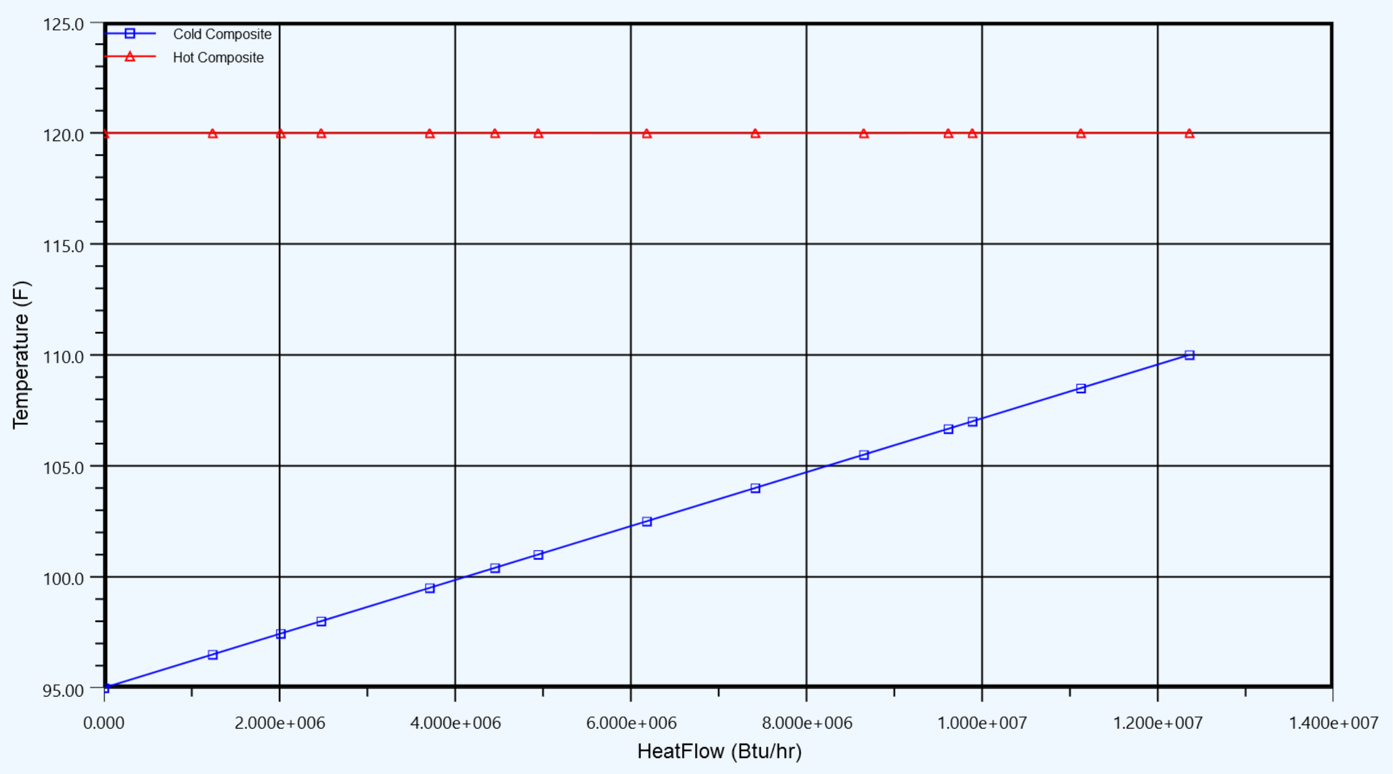
Calculate UA knowing the terminal temperatures

$$(\Delta T)_{LMTD} = \frac{(120-110)-(120-95)}{\ln\left(\frac{120-110}{120-95}\right)} = 16.4°\text{F}$$

$$UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{12360000}{16.4} = 755{,}000 \frac{\text{Btu}}{\text{hr}°\text{F}}$$

Heat Exchanger – Example 4

Calculating the UA is essentially the same as when there is no phase change since the temperature profiles with heat release are still straight lines. The hot stream's profile just happens to be a constant temperature.



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Heat Exchanger – Example 5

Condense 10,000 lb/hr propane vapor that is superheated to 160°F (but still with 120°F vapor pressure) using 95°F air heated up to 110°F.

- Needed physical properties
  - Air heat capacity – 0.24 Btu/lb F
  - Propane vapor heat capacity – 0.52 Btu/lb F
  - Propane heat of vaporization @ 120°F – 1236 Btu/lb

Determine

- Exchanger duty
- Flow rate of air needed
- Exchanger UA

Heat Exchanger – Example 5

Duty determined from the propane balance. Combine sensible heat & latent heat effects

$$\dot{Q} = \dot{m}_h \left[\lambda_{vap} + \hat{C}_{p,c} \left(T_{h,in} - T_{h,BP} \right) \right] = (10000)\left[(1236) + 0.52(160 - 120)\right]$$
$$= (10000)\left[1236 + 20.8\right]$$
$$= 125{,}680{,}000 \text{ Btu/hr}$$

Air flowrate from its energy balance:

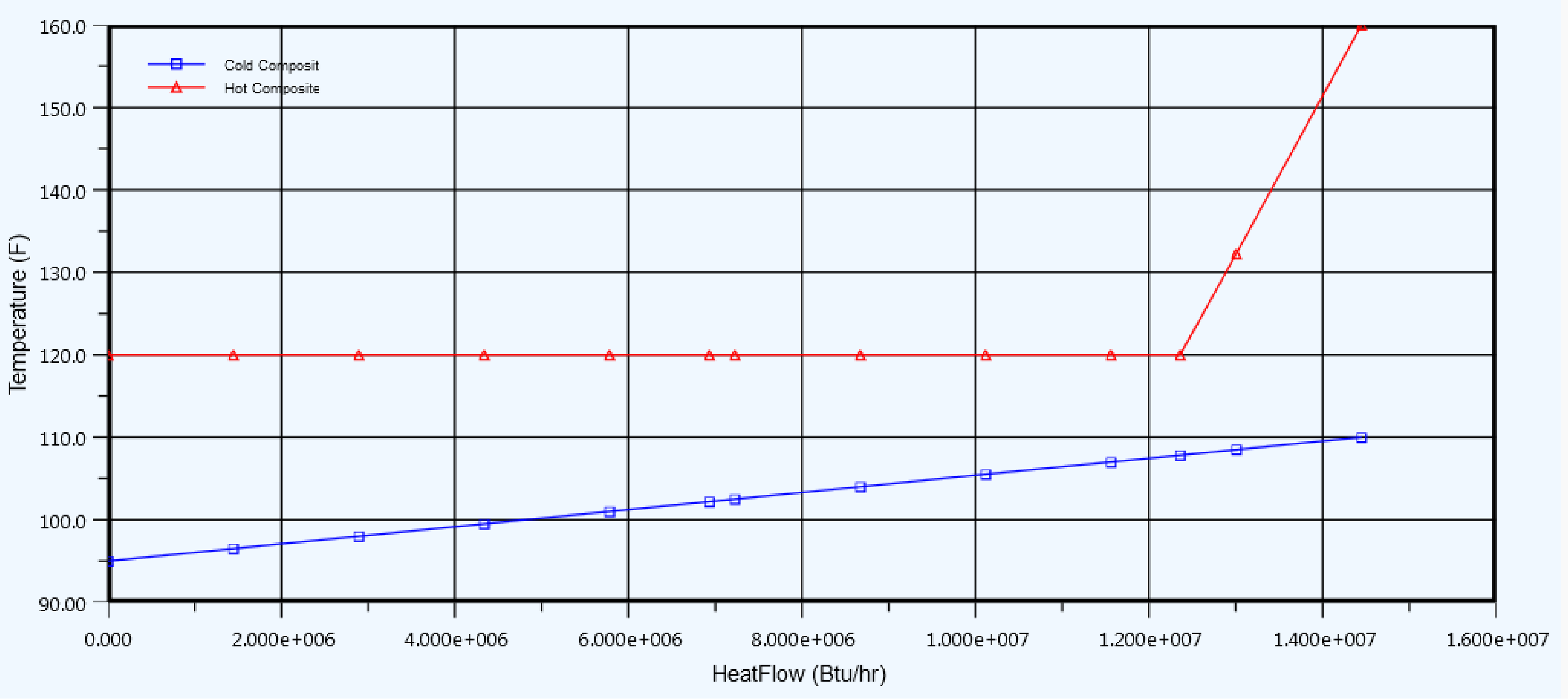
$$\dot{m}_c = \frac{\dot{Q}}{\hat{C}_{p,c}\left(T_{c,out} - T_{c,in}\right)} = \frac{125{,}680{,}000}{(0.24)(110 - 95)} = 34{,}900{,}000 \text{ lb/hr}$$

Just using terminal temperatures gives an incorrect result!

$$(\Delta T)_{LMTD} = \frac{(160 - 110) - (120 - 95)}{\ln\left(\frac{160 - 110}{120 - 95}\right)} = 36.1°\text{F} \quad \Rightarrow \quad UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{125680000}{36.1} = 3{,}480{,}000 \frac{\text{Btu}}{\text{hr°F}}$$

Heat Exchanger – Example 5

Calculating the UA is more complicated than just using the terminal temperatures since there is a drastic break in the temperature profile for the condensing propane



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Heat Exchanger – Example 5

Determine the intermediate air temperature between the sensible & latent heat zones

$$\dot{Q}_{cond} = \dot{m}_h \lambda_{vap} = 123{,}600{,}000 \text{ Btu/hr}$$

$$T_{c,mid} = T_{c,in} + \frac{\dot{Q}_{cond}}{\dot{m}_c \hat{C}_{p,c}} = 95 + \frac{123{,}600{,}000}{(34{,}900{,}000)(0.24)} = 109.8°\text{F}$$

Calculate the UA values for the two zones

$$(\Delta T)_{LMTD} = \frac{(120-109.8)-(120-95)}{\ln\left(\dfrac{120-109.8}{120-95}\right)} = 16.5°\text{F}$$

$$(\Delta T)_{LMTD} = \frac{(160-110)-(120-109.8)}{\ln\left(\dfrac{160-110}{120-109.8}\right)} = 25.0°\text{F}$$

$$UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{123600000}{36.1} = 3{,}420{,}000 \frac{\text{Btu}}{\text{hr°F}}$$

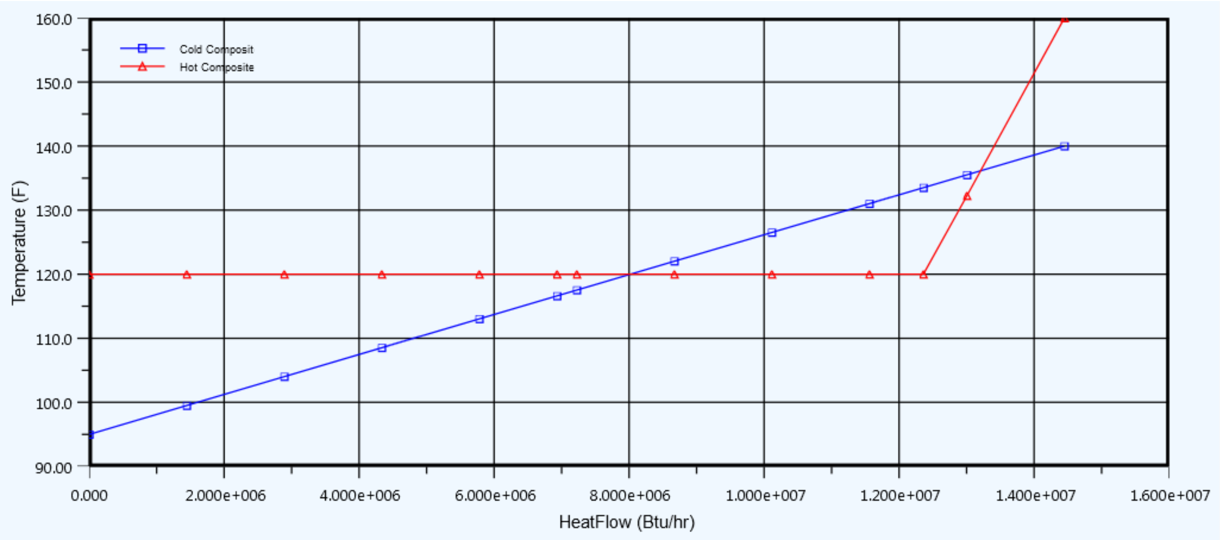
$$UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{2080000}{25.0} = 83{,}000 \frac{\text{Btu}}{\text{hr°F}}$$

The total UA is the sum of these two contributions.

Heat Exchanger – Example 6

Can we condense 10,000 lb/hr propane vapor that is superheated to 160°F (but still with 120°F vapor pressure) using 95°F air heated up to 140°F? This still gives an apparent 20°F approach temperature?

The answer is NO! It is not apparent from the terminal temperatures but there is an internal pinch point to the temperature profiles & there would be a temperature crossover. The air's outlet temperature is constrained by this <u>internal pinch point</u>.



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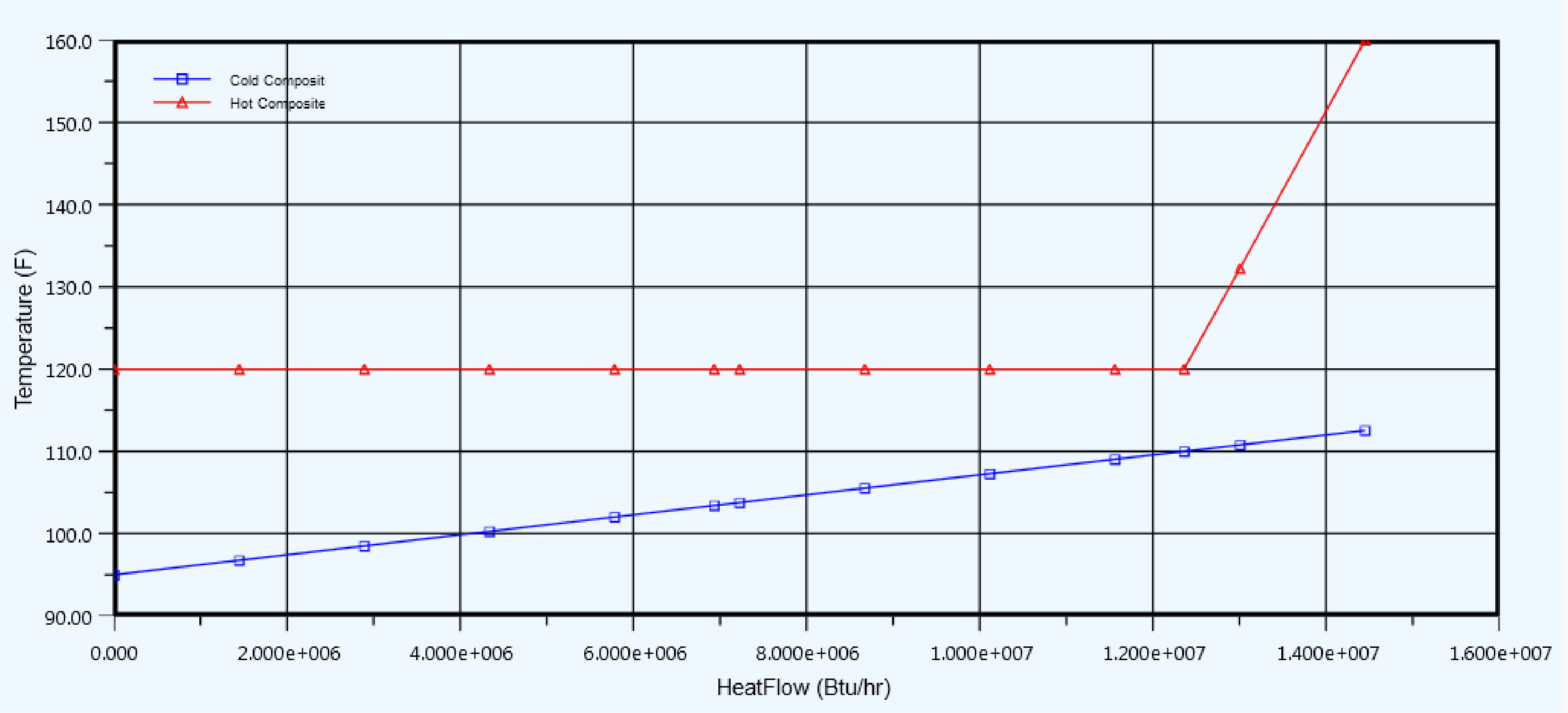
Heat Exchanger – Example 6

More air flow is needed to accomplish this cooling. Using a 10°F internal approach temperature shows that the air's outlet temperature is constrained to 112.5°F. The required air mass flow would be calculated accordingly.



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Heat Transfer – Some Basics

Thermal resistances are added when in series

- Can be combined into an overall heat transfer coefficient
- Across a flat plate (i.e., constant cross sectional area)

$$\frac{1}{U} = \frac{1}{h_i} + \frac{L}{k} + \frac{1}{h_o}$$

- For radial heat transfer (e.g., through the wall of a tube) must also take into account the change is area with respect to radius
  - Overall heat transfer coefficient must also be related to a reference area / diameter

$$\frac{1}{U_o A_o} = \frac{1}{h_i A_i} + \frac{L}{k A_{ave}} + \frac{1}{h_o A_o}$$

$$\frac{1}{U_o} = \frac{1}{h_i}\frac{A_o}{A_i} + \frac{L}{k}\frac{A_o}{A_{ave}} + \frac{1}{h_o} = \frac{1}{h_i}\frac{D_o}{D_i} + \frac{2D_o}{k}\ln\left(\frac{D_o}{D_i}\right) + \frac{1}{h_o}$$

Heat Transfer – Correlations for Film Coefficients

Flow in tubes with no phase change

$$N_{Nu} = 0.023\, N_{Re}^{0.8}\; N_{Pr}^{0.4} \quad \Rightarrow \quad \left(\frac{hD}{k}\right) = 0.023 \left(\frac{D\, v\, \rho}{\mu}\right)^{0.8} \left(\frac{C_p\, \mu}{k}\right)^{0.4}$$

When there is a significant difference between wall & bulk fluid

$$N_{Nu} = 0.023\, N_{Re}^{0.8}\; N_{Pr}^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad \Rightarrow \quad \left(\frac{hD}{k}\right) = 0.023 \left(\frac{D\, v\, \rho}{\mu}\right)^{0.8} \left(\frac{C_p\, \mu}{k}\right)^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

Stirred liquids, heat transfer from coil …

$$N_{Nu} = 0.9\, N_{Re,i}^{0.62}\; N_{Pr}^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad \Rightarrow \quad \left(\frac{hD}{k}\right) = 0.9 \left(\frac{N_i\, D_i^2\, \rho}{\mu}\right)^{0.62} \left(\frac{C_p\, \mu}{k}\right)^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

… from tank jacket

$$N_{Nu} = 0.36\, N_{Re,i}^{0.66}\; N_{Pr}^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \quad \Rightarrow \quad \left(\frac{hD}{k}\right) = 0.36 \left(\frac{N_i\, D_i^2\, \rho}{\mu}\right)^{0.67} \left(\frac{C_p\, \mu}{k}\right)^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14}$$

Typical Overall Heat Transfer Coefficients

Heat transfer coefficients are drastically different for conditions of boiling and/or condensation versus when there is sensible heat change.

- Bubbles break up the films along the wall
- Also dependent upon the temperature difference across the wall

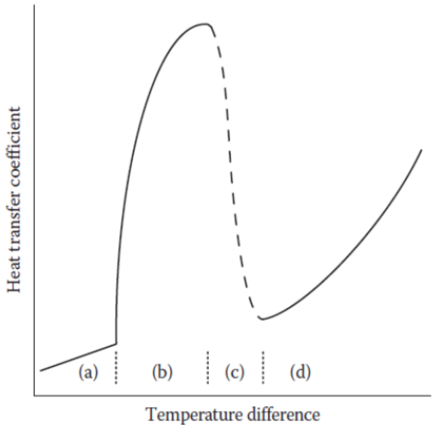
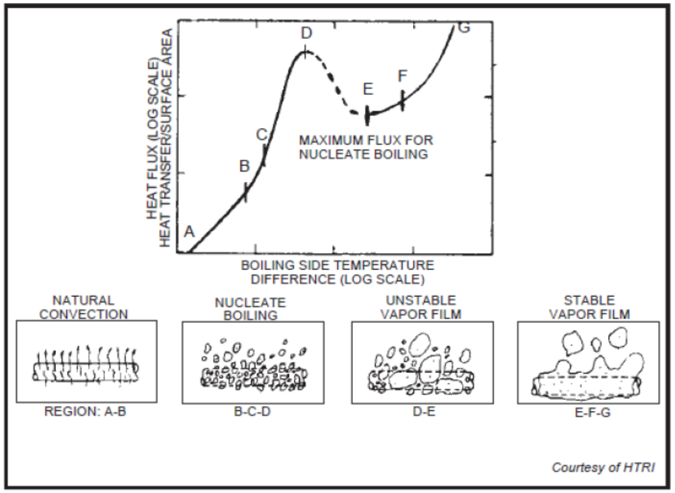


FIGURE 3.5 Heat transfer coefficient as function of temperature difference between liquid and hot surface. Region (a) no boiling, (b) nucleate boiling, (c) transition region, and (d) film boiling.

Fundamentals of Natural Gas Processing, 2nd ed., Kidnay, Parrish, & McCartney, 2011

FIG. 9-15

A Typical Pool Boiling Curve



Courtesy of HTRI

Ref: *GPSA Data Book*, 13th ed.

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Typical Overall Heat Transfer Coefficients

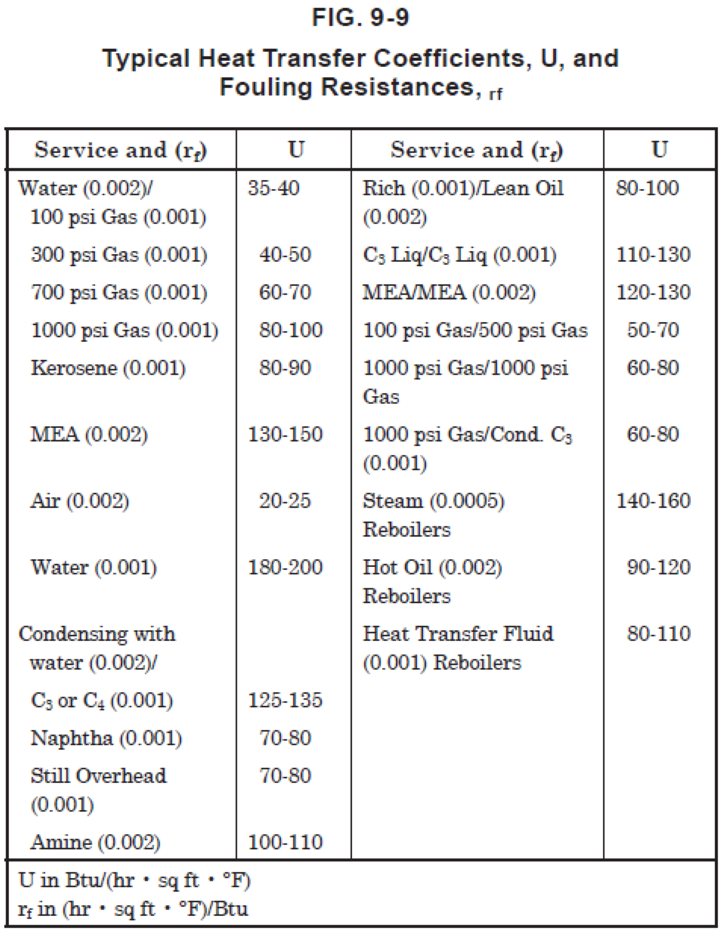


FIG. 9-9

Typical Heat Transfer Coefficients, U, and Fouling Resistances, r_f

Service and (r_f)	U	Service and (r_f)	U
Water (0.002)/ 100 psi Gas (0.001)	35-40	Rich (0.001)/Lean Oil (0.002)	80-100
300 psi Gas (0.001)	40-50	C_3 Liq/C_3 Liq (0.001)	110-130
700 psi Gas (0.001)	60-70	MEA/MEA (0.002)	120-130
1000 psi Gas (0.001)	80-100	100 psi Gas/500 psi Gas	50-70
Kerosene (0.001)	80-90	1000 psi Gas/1000 psi Gas	60-80
MEA (0.002)	130-150	1000 psi Gas/Cond. C_3 (0.001)	60-80
Air (0.002)	20-25	Steam (0.0005) Reboilers	140-160
Water (0.001)	180-200	Hot Oil (0.002) Reboilers	90-120
Condensing with water (0.002)/		Heat Transfer Fluid (0.001) Reboilers	80-110
C_3 or C_4 (0.001)	125-135		
Naphtha (0.001)	70-80		
Still Overhead (0.001)	70-80		
Amine (0.002)	100-110		

U in Btu/(hr • sq ft • °F)
r_f in (hr • sq ft • °F)/Btu

Ref: *GPSA Data Book*, 13th ed.

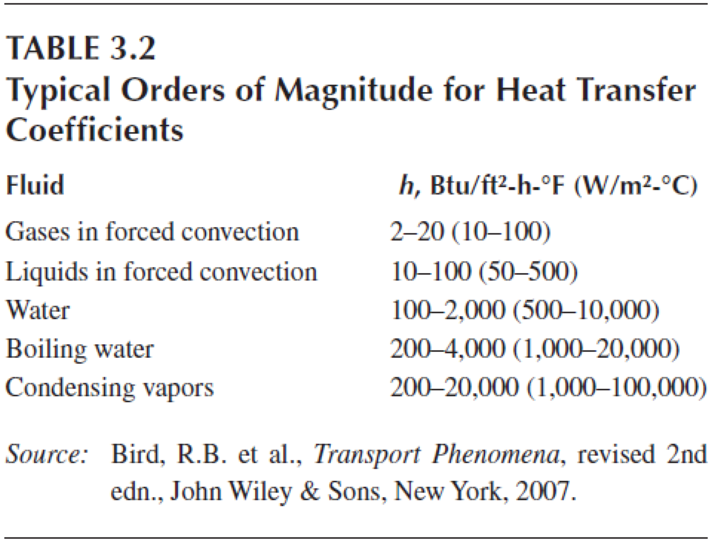


TABLE 3.2
Typical Orders of Magnitude for Heat Transfer Coefficients

Fluid	*h*, Btu/ft²-h-°F (W/m²-°C)
Gases in forced convection	2–20 (10–100)
Liquids in forced convection	10–100 (50–500)
Water	100–2,000 (500–10,000)
Boiling water	200–4,000 (1,000–20,000)
Condensing vapors	200–20,000 (1,000–100,000)

Source: Bird, R.B. et al., *Transport Phenomena*, revised 2nd edn., John Wiley & Sons, New York, 2007.

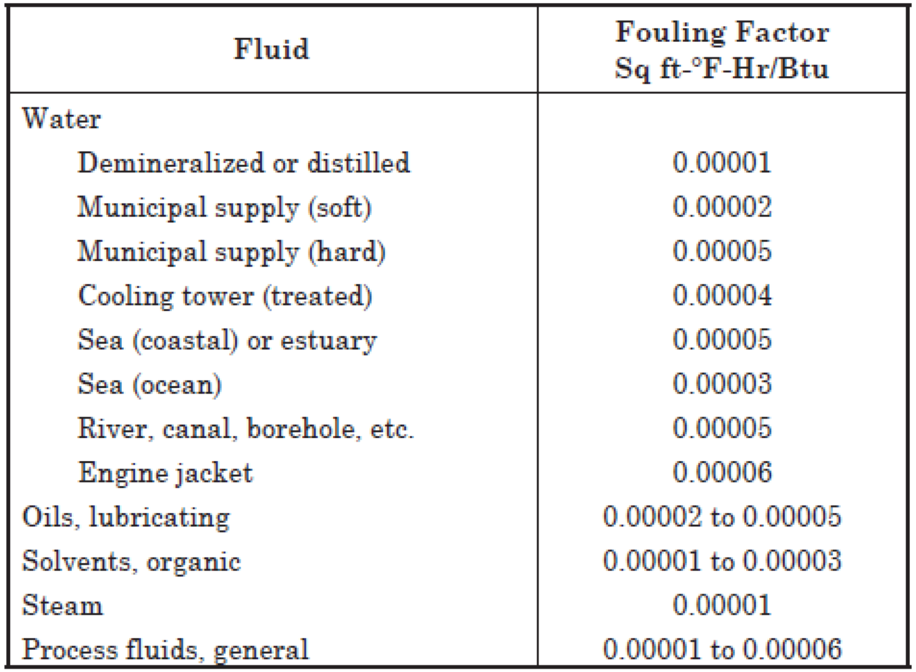
Fundamentals of Natural Gas Processing, 2nd ed., Kidnay, Parrish, & McCartney, 2011

Typical Fouling Factors

Add these resistances to the reciprocal of the “clean” overall heat transfer coefficient

FIG. 9-45

Typical Fouling Factors for PHEs



Fluid	Fouling Factor Sq ft-°F-Hr/Btu
Water	
Demineralized or distilled	0.00001
Municipal supply (soft)	0.00002
Municipal supply (hard)	0.00005
Cooling tower (treated)	0.00004
Sea (coastal) or estuary	0.00005
Sea (ocean)	0.00003
River, canal, borehole, etc.	0.00005
Engine jacket	0.00006
Oils, lubricating	0.00002 to 0.00005
Solvents, organic	0.00001 to 0.00003
Steam	0.00001
Process fluids, general	0.00001 to 0.00006

Ref: *GPSA Data Book*, 13th ed.

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Equipment

Updated: January 29, 2019

Gas Processing Applications

Common heat exchangers

- Shell and Tube
- Kettle reboiler
- Aerial coolers
- Plate Frame
- Plate-Fin (Brazed Aluminum)
- Hairpin
- Tank Heaters

http://www.alfalaval.com/globalassets/images/media/stories/crude-oil-refinery/ppi00393_compabloc-brazil_640x360.jpg

Shell & Tube Heat Exchangers

Workhorses of the gas processing industry

Shell side

- Baffles used in the shell side to minimize channeling

Tube side

- Manifolds allow for even distribution of fluids into the tubes & collection/mixing of fluids out of the tubes
- Multiple tube passes make it easier to pull the tube bundle for maintenance/cleaning and…
- … have better allowance for thermal expansion effects

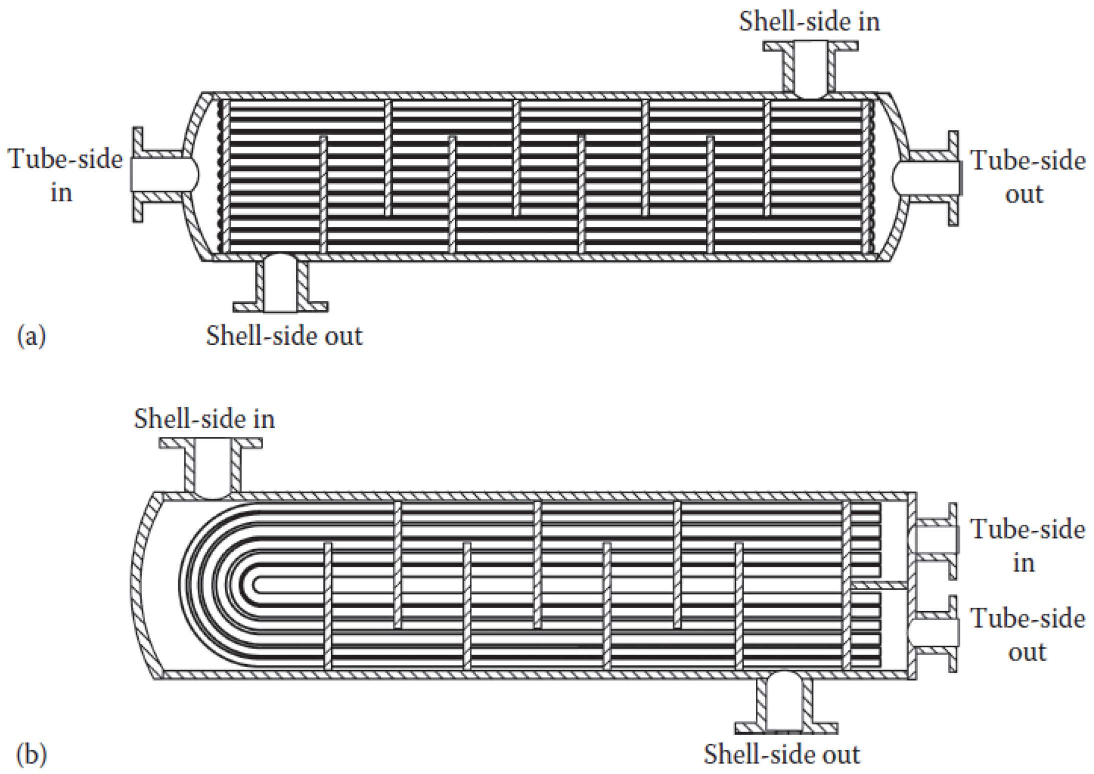

Fig. 3.6, *Fundamentals of Natural Gas Processing*, 2nd ed., Kidnay, Parrish, & McCartney, 2011

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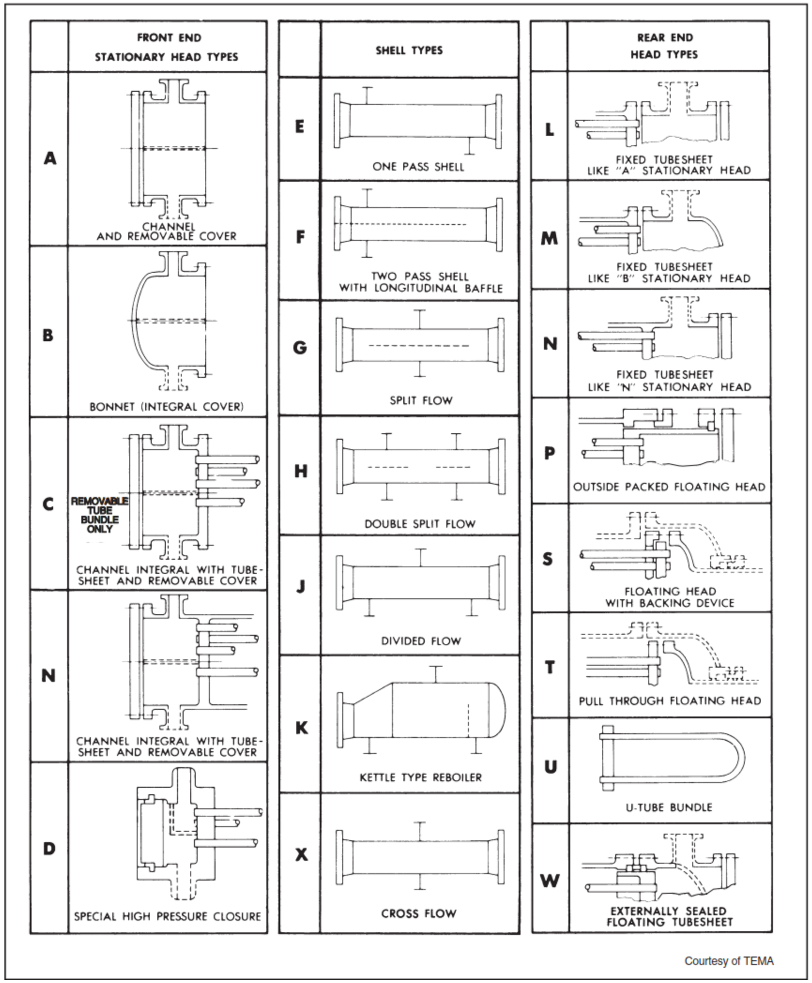
Shell and Tube Heat Exchangers (Types)

FIG. 9-23

TEMA Shell and Tube Exchanger Nomenclature



Courtesy of TEMA

Ref: *GPSA Data Book*, 13th ed.

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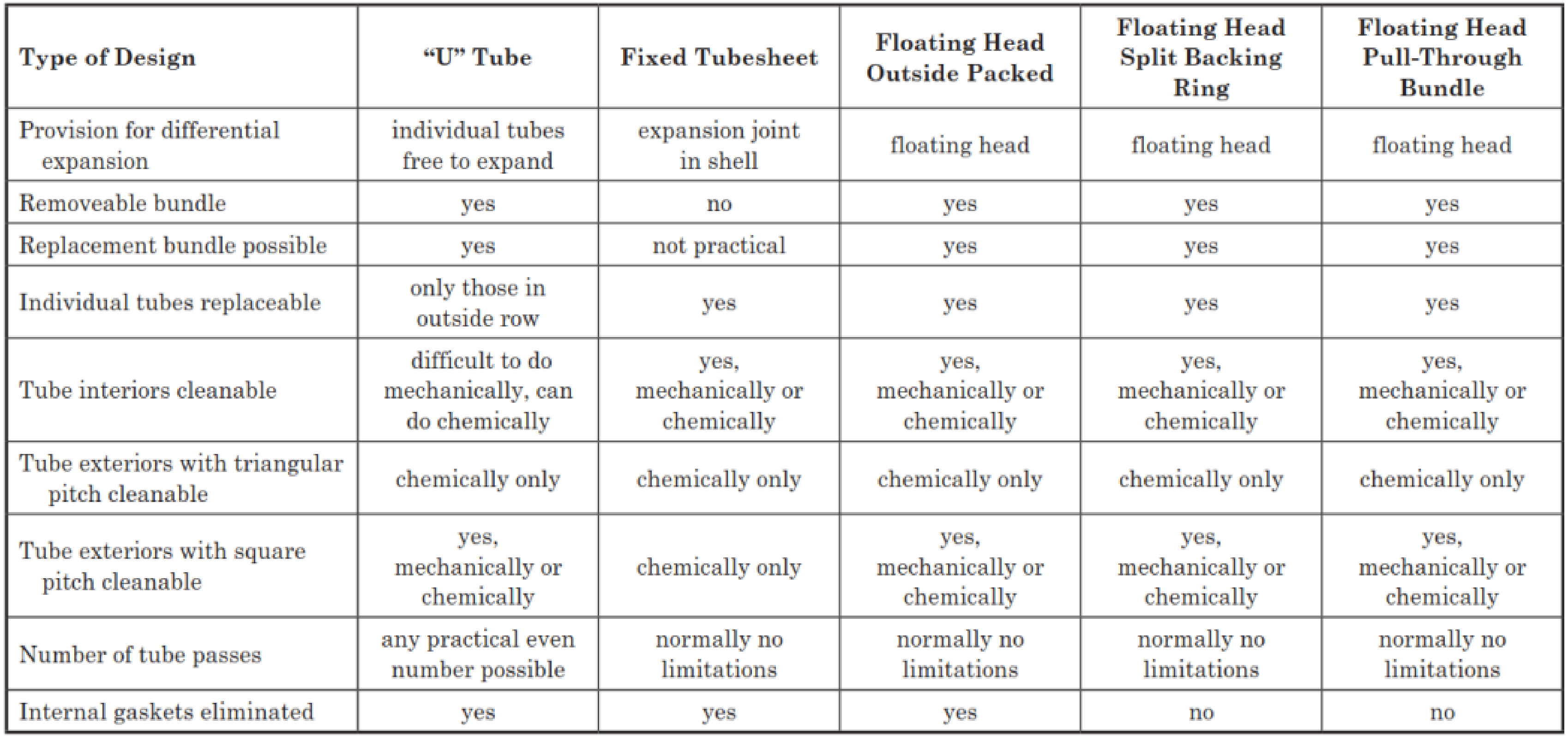
Shell and Tube Heat Exchangers (Selection)

FIG. 9-24

Shell and Tube Exchanger Selection Guide (Cost Increases from Left to Right)



Type of Design	"U" Tube	Fixed Tubesheet	Floating Head Outside Packed	Floating Head Split Backing Ring	Floating Head Pull-Through Bundle
Provision for differential expansion	individual tubes free to expand	expansion joint in shell	floating head	floating head	floating head
Removeable bundle	yes	no	yes	yes	yes
Replacement bundle possible	yes	not practical	yes	yes	yes
Individual tubes replaceable	only those in outside row	yes	yes	yes	yes
Tube interiors cleanable	difficult to do mechanically, can do chemically	yes, mechanically or chemically	yes, mechanically or chemically	yes, mechanically or chemically	yes, mechanically or chemically
Tube exteriors with triangular pitch cleanable	chemically only	chemically only	chemically only	chemically only	chemically only
Tube exteriors with square pitch cleanable	yes, mechanically or chemically	chemically only	yes, mechanically or chemically	yes, mechanically or chemically	yes, mechanically or chemically
Number of tube passes	any practical even number possible	normally no limitations	normally no limitations	normally no limitations	normally no limitations
Internal gaskets eliminated	yes	yes	yes	no	no

Ref: *GPSA Data Book*, 13th ed.

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Kettle Reboiler

Shell & tube heat exchanger with the tubes submerged in boiling liquid on the shell side

- Main resistance to heat transfer is on the tube side since boiling is occurring on the shell side

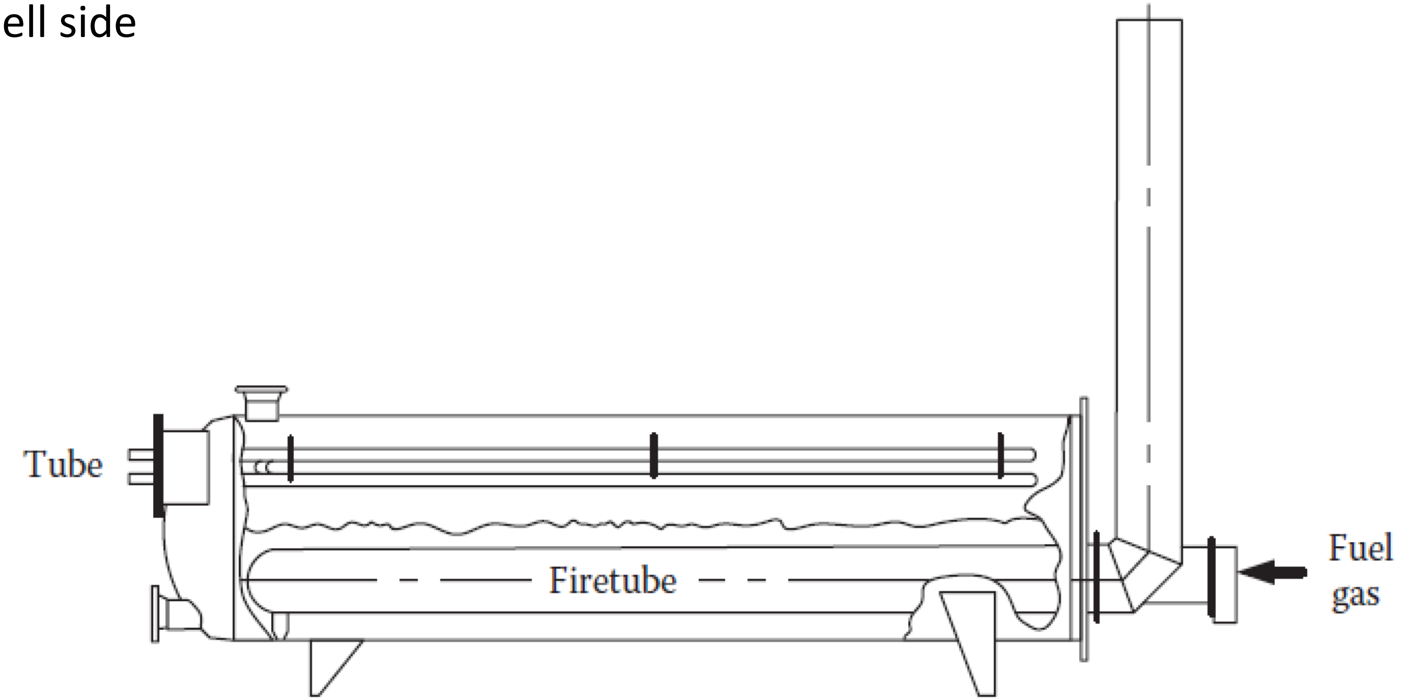

Fig. 3.7, *Fundamentals of Natural Gas Processing*, 2nd ed., Kidnay, Parrish, & McCartney, 2011

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Plate Frame Heat Exchangers

Positives

- Low cost
- Compact – high area per weight & volume
- Can get very close approach temperatures (5°F or lower)
- Can be disassembled to clean

Negative considerations

- Limited maximum allowable working pressure
- Susceptible to plugging

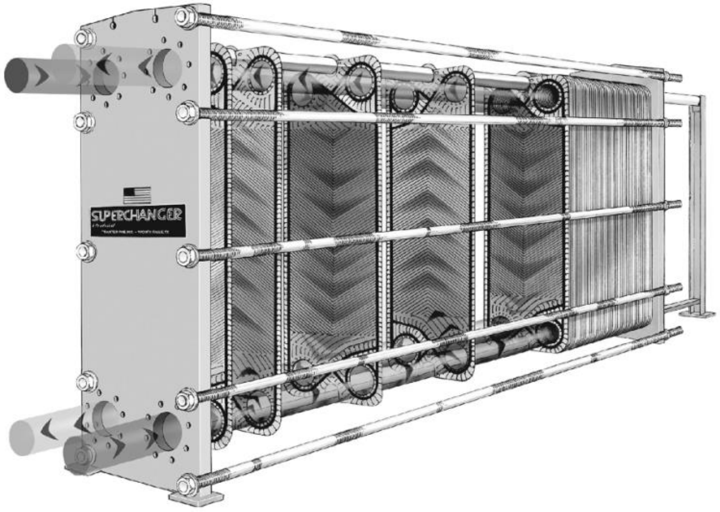
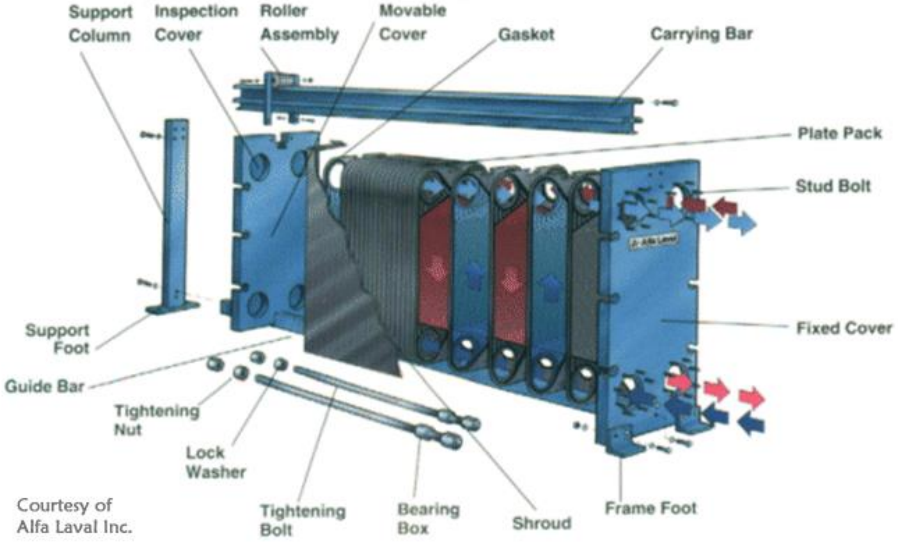


Fig. 3.9, *Fundamentals of Natural Gas Processing*, 2nd ed., Kidnay, Parrish, & McCartney, 2011



Courtesy of Alfa Laval Inc.

http://www.cheresources.com/content/articles/heat-transfer/plate-heat-exchangers-preliminary-design

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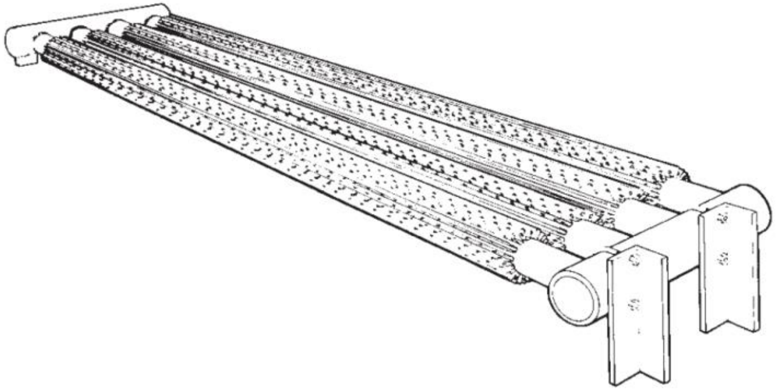
Tank Heaters

Integrated into existing equipment (i.e., tanks or vessels)

FIG. 9-32

Prefabricated Tank Heater



Courtesy of Brown Fintube Co.

https://www.chromalox.com/en/global/case-studies/pocket-heater-reduces-costs-and-downtime

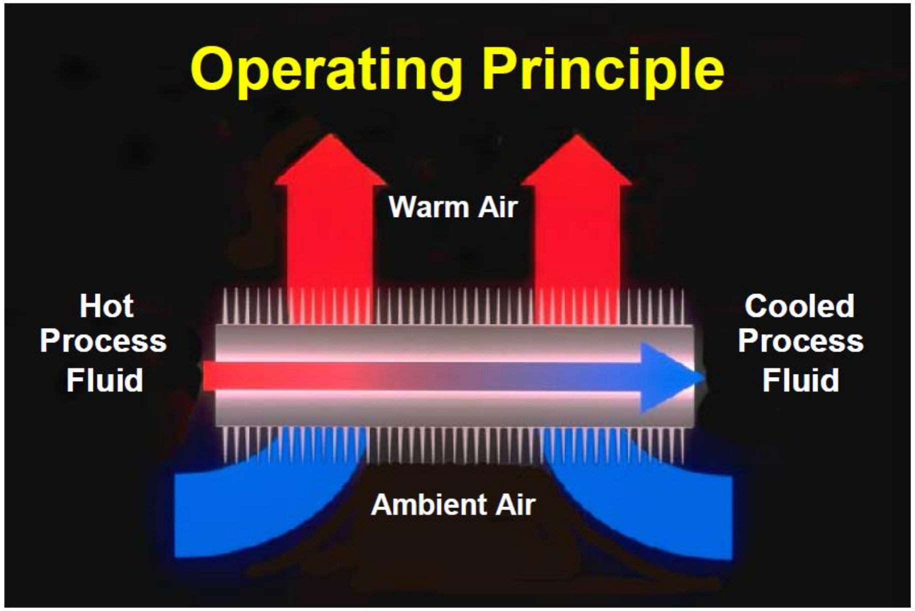
Ref: *GPSA Data Book*, 13th ed.

Air-Cooled Exchangers – Fundamentals

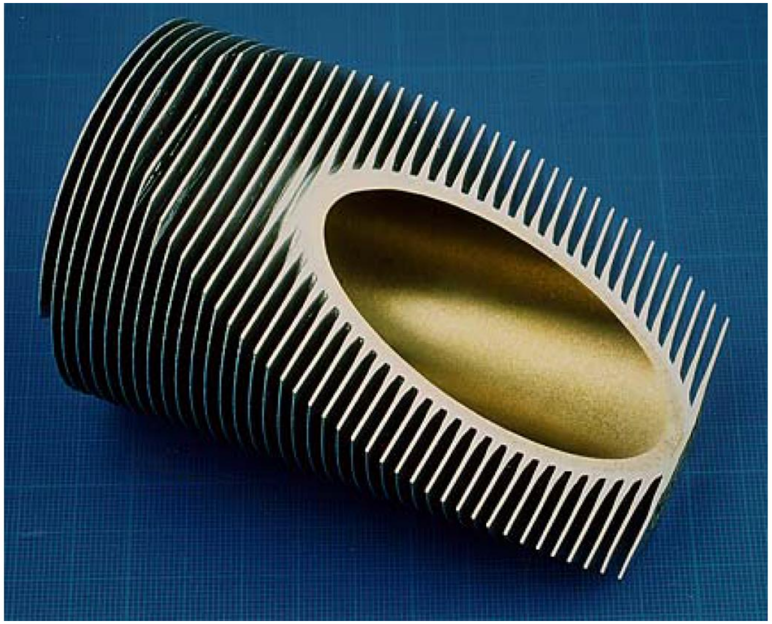
Air cooled exchangers cool fluids with ambient air

- Seasonal variation can greatly impact performance

Utilize finned tube in increase heat transfer surface area



www.hudsonproducts.com



www.hudsonproducts.com

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Aerial Coolers

Fans either push air through (forced draft) or pull air through (induced draft) tube bundle

- Can control the air flow rate either with a variable speed motor or with louviers

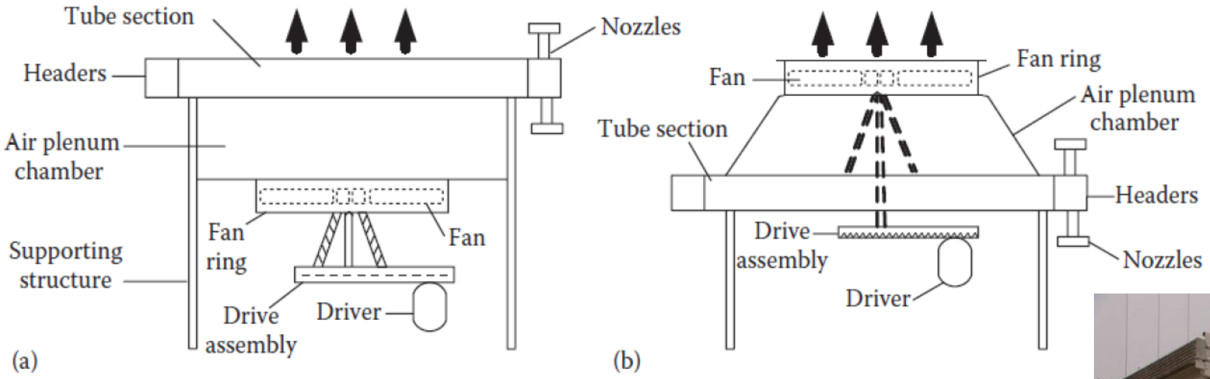

Fig. 3.8, *Fundamentals of Natural Gas Processing*, 2nd ed., Kidnay, Parrish, & McCartney, 2011

http://spxcooling.com/products/detail/air-cooled-heat-exchangers

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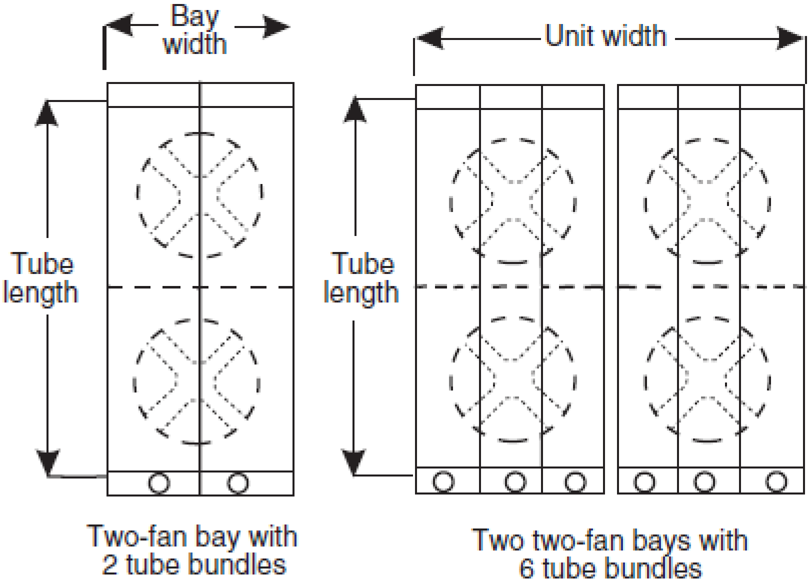
Aerial Cooler Design Considerations

Typically a small number of diameter tubes (e.g., 1in OD) with fins on the air side (e.g., 1/2 or 5/8 in)

Design considerations

- Process side – pressure drop for flow inside of tubes
- Air side
  - Required air flow
    - May need high air flow to prevent temperature crossover, but…
    - High air flow gives higher pressure drop & fan power
- Mechanical considerations
  - Total number of tubes
  - Tube layout: number of passes, number of rows, pitch
  - Bay size: typically 45 ft X 15 ft max



Ref: *GPSA Data Book*, 14th ed.

Air-Cooled Exchangers – Types

Forced Draft:

Advantages:

- Slightly lower horsepower
- Better maintenance accessibility
- Easily adaptable for warm air recirculation
- Most common in gas industry

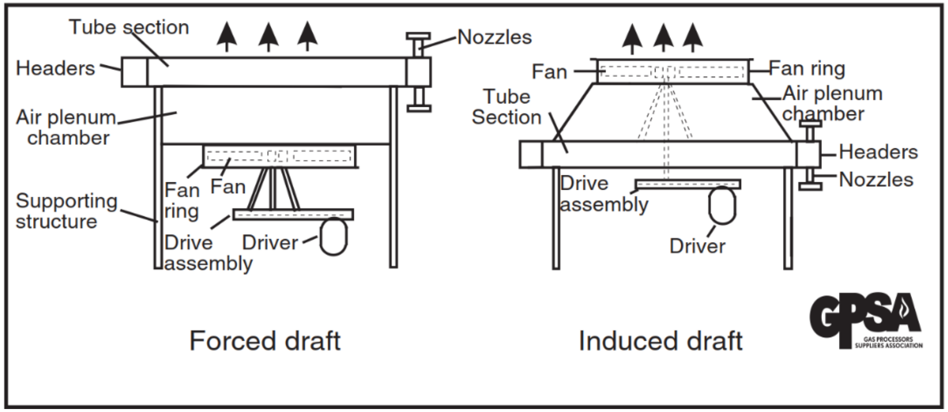
Induced Draft:

Advantages

- Better distribution of air
- Less possibility of air recirculation
- Less effect of sun, rain, or hail
- Increased capacity in the event of fan failure

FIG. 10-2

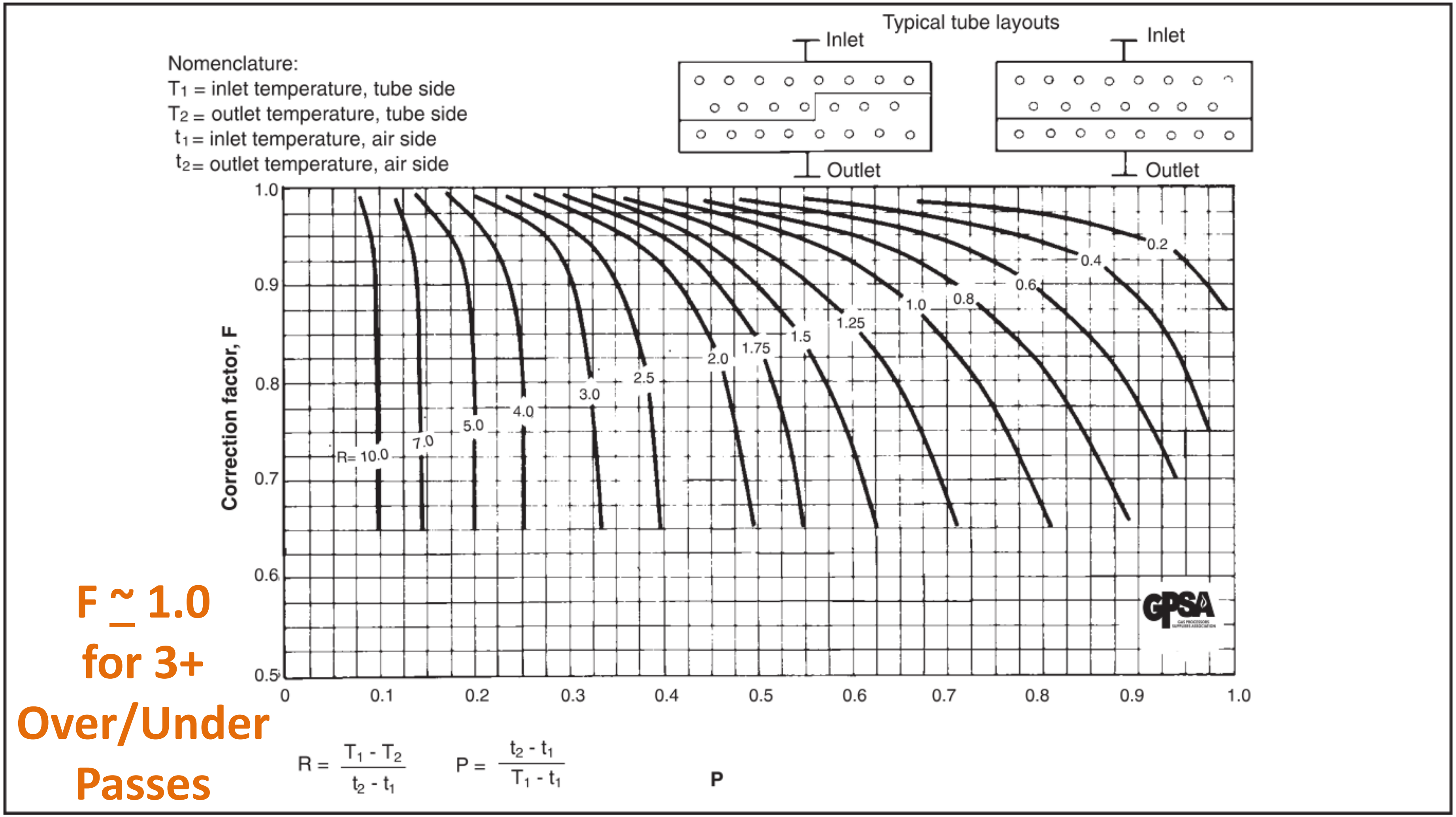
Typical Side Elevations of Air Coolers



Air-Cooled Exchanger – Thermal Design (Δ Temperature – CMTD Figs 10-8 & 9)

FIG. 10-9

MTD Correction Factors (2 Pass — Cross Flow, Both Fluids Unmixed)



F ~ 1.0
for 3+
Over/Under
Passes

Summary

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Summary

Common types of heat exchangers used in the gas processing industry

- Shell & tube
- Kettle reboiler
- Air cooled exchangers
- Plate Frame
- Plate-Fin (Brazed Aluminum)
- Hairpin
- Tank Heaters

Heat exchange basics

- Coupling of fluid energy balances with heat transfer across barrier
- Common heat exchanger configurations
- Typical heat transfer coefficients
- Example process calculations involving heat exchangers

Supplemental Slides

Updated: January 29, 2019

LMTD as Area-Averaged Temperature Difference

If the temperature curves are linearly related to the duty then the temperature difference will also be linearly related to duty

$$\Delta T = (\Delta T)_0 + \frac{\left[(\Delta T)_1 - (\Delta T)_0\right]}{\dot{Q}}\dot{q} \quad \Rightarrow \quad d(\Delta T) = \frac{\left[(\Delta T)_1 - (\Delta T)_0\right]}{\dot{Q}}d\dot{q}$$

Can put into differential form of heat transfer equation & integrate

$$d\dot{q} = U(\Delta T)da \quad \Rightarrow \quad \frac{\dot{Q}}{(\Delta T)_1 - (\Delta T)_0}d(\Delta T) = U(\Delta T)da$$

$$\frac{d(\Delta T)}{\Delta T} = U\frac{(\Delta T)_1 - (\Delta T)_0}{\dot{Q}}da$$

$$\int_{(\Delta T)_0}^{(\Delta T)_1}\frac{d(\Delta T)}{\Delta T} = U\frac{(\Delta T)_1 - (\Delta T)_0}{\dot{Q}}\int_0^A da$$

$$\ln\left[\frac{(\Delta T)_1}{(\Delta T)_0}\right] = U\frac{(\Delta T)_1 - (\Delta T)_0}{\dot{Q}}A \quad \Rightarrow \quad \dot{Q} = UA\frac{(\Delta T)_1 - (\Delta T)_0}{\ln\left[\frac{(\Delta T)_1}{(\Delta T)_0}\right]} = UA\overline{(\Delta T)}_{LM}$$

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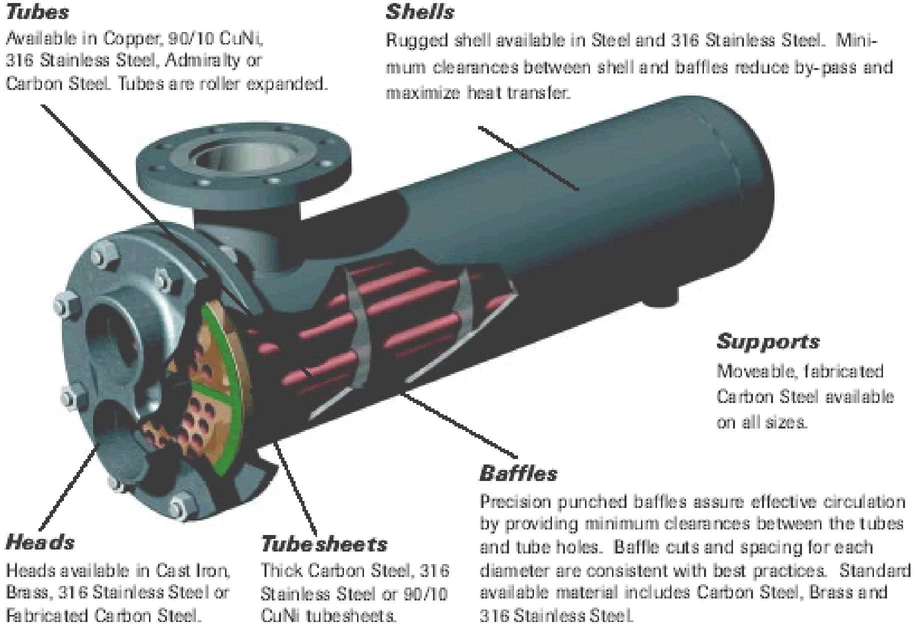
Shell & Tube Heat Exchangers



Tubes
Available in Copper, 90/10 CuNi, 316 Stainless Steel, Admiralty or Carbon Steel. Tubes are roller expanded.

Shells
Rugged shell available in Steel and 316 Stainless Steel. Minimum clearances between shell and baffles reduce by-pass and maximize heat transfer.

Supports
Moveable, fabricated Carbon Steel available on all sizes.

Heads
Heads available in Cast Iron, Brass, 316 Stainless Steel or Fabricated Carbon Steel.

Tubesheets
Thick Carbon Steel, 316 Stainless Steel or 90/10 CuNi tubesheets.

Baffles
Precision punched baffles assure effective circulation by providing minimum clearances between the tubes and tube holes. Baffle cuts and spacing for each diameter are consistent with best practices. Standard available material includes Carbon Steel, Brass and 316 Stainless Steel.

http://www.apiheattransfer.com/Product/54/Type-ST-U-Tube-Shell-Tube-Heat-Exchangers

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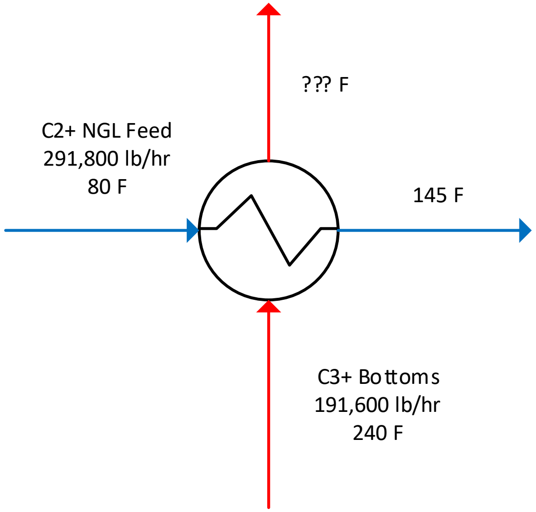
Heat Exchanger – Example 7

Heat 291,800 lb/hr cold C2+ NGL feed from 80ºF to 160ºF using 191,600 lb/hr hot C3+ bottoms @ 240ºF

- Assume only sensible heat effects
    - C2+ NGL feed heat capacity – 0.704 Btu/lb F
    - C3+ Bottoms heat capacity – 0.828 Btu/lb F

Determine

- C3+ Bottoms outlet temperature
- Exchanger duty
- (UA) for the exchanger



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Heat Exchanger – Example 7

Exchanger duty & C3+ Bottoms outlet temperature determined from energy balance around exchanger

$$\dot{Q} = \dot{m}_c \hat{C}_{p,c}\left(T_{c,out} - T_{c,in}\right) = (291800)(0.704)(160 - 80) = 16{,}434{,}000 \text{ Btu/hr}$$

$$T_{h,out} = T_{h,in} - \frac{\dot{Q}}{\dot{m}_h \hat{C}_{p,h}} = 240 - \frac{16434000}{(191600)(0.828)} = 136.4°\text{F}$$

Determination of UA requires configuration information

- 1-1 counter-current flow

$$(\Delta T)_{LMTD} = \frac{(240-160)-(136.4-80)}{\ln\left(\dfrac{240-160}{136.4-80}\right)} = 67.5°\text{F}$$

$$UA = \frac{\dot{Q}}{(\Delta T)_{LMTD}} = \frac{16434000}{67.5} = 243{,}000 \frac{\text{Btu}}{\text{hr°F}}$$

1-1 co-current flow

cannot be done – crossover!

Heat Exchanger – Example 7

1-2 exchanger

$$(\Delta T)_{LMTD} = \frac{(240-160)-(136.4-80)}{\ln\left(\dfrac{240-160}{136.4-80}\right)} = 67.5^\circ\text{F}$$

$$P = \frac{160-80}{240-80} = 0.5$$

$$R = \frac{240-136.4}{160-80} = 1.3$$

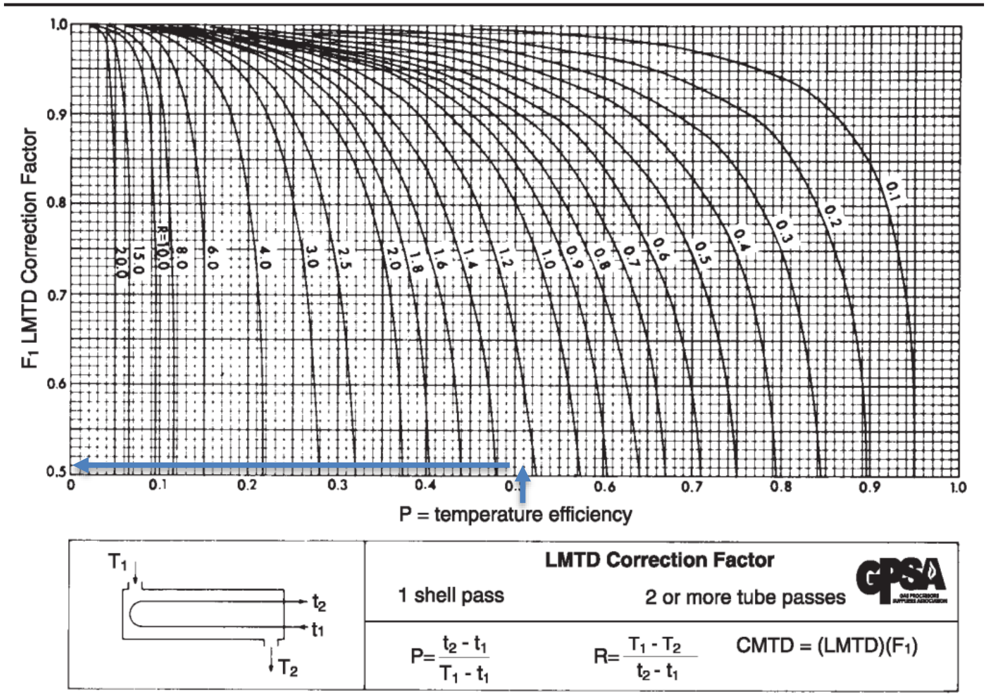
$$F_2 = 0.51 \text{ (from chart)}$$

$$UA = \frac{\dot{Q}}{F_2 (\Delta T)_{LMTD}}$$

$$= \frac{16434000}{0.51(67.5)} = 477{,}400 \frac{\text{Btu}}{\text{hr}^\circ\text{F}}$$

FIG. 9-4

LMTD Correction Factor (1 shell passes; 2 or more tube passes)



Ref: *GPSA Data Book*, 13th ed.

Heat Exchanger – Example 7

Correction factor below 0.8. Try 2-4 exchanger

$$(\Delta T)_{LMTD} = \frac{(240-160)-(136.4-80)}{\ln\left(\dfrac{240-160}{136.4-80}\right)} = 67.5°\text{F}$$

$$P = \frac{160-80}{240-80} = 0.5$$

$$R = \frac{240-136.4}{160-80} = 1.3$$

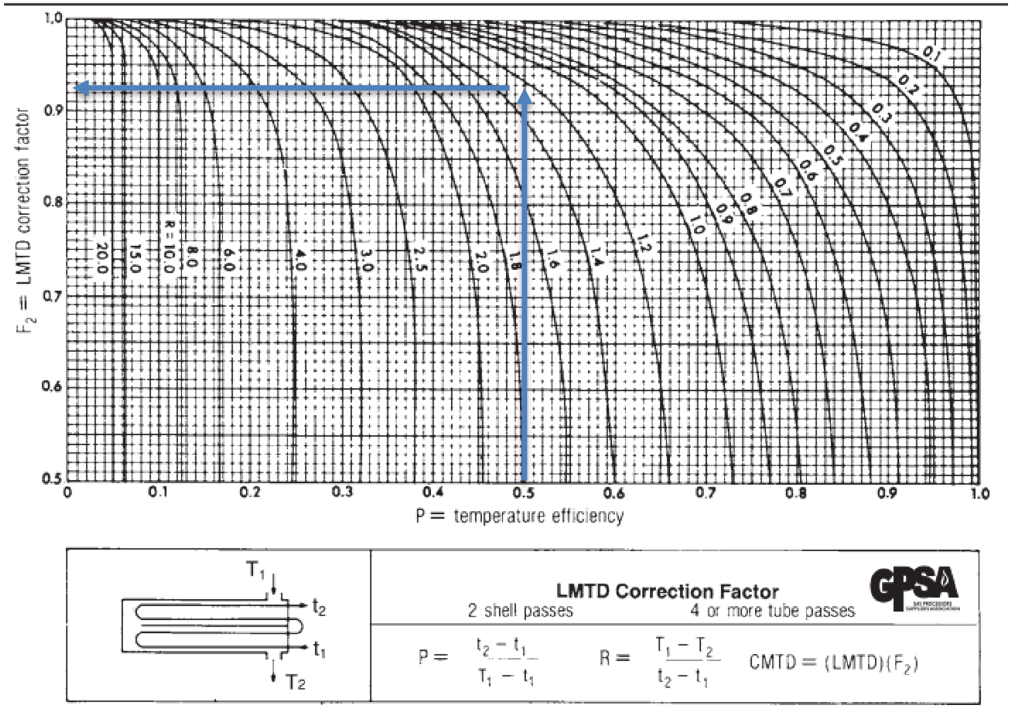
$$F_2 = 0.925 \text{ (from chart)}$$

$$UA = \frac{\dot{Q}}{F_2 (\Delta T)_{LMTD}}$$

$$= \frac{16434000}{0.925(67.5)} = 263{,}200 \frac{\text{Btu}}{\text{hr°F}}$$

FIG. 9-5

LMTD Correction Factor (2 shell passes; 4 or more tube passes)



Ref: *GPSA Data Book*, 13th ed.

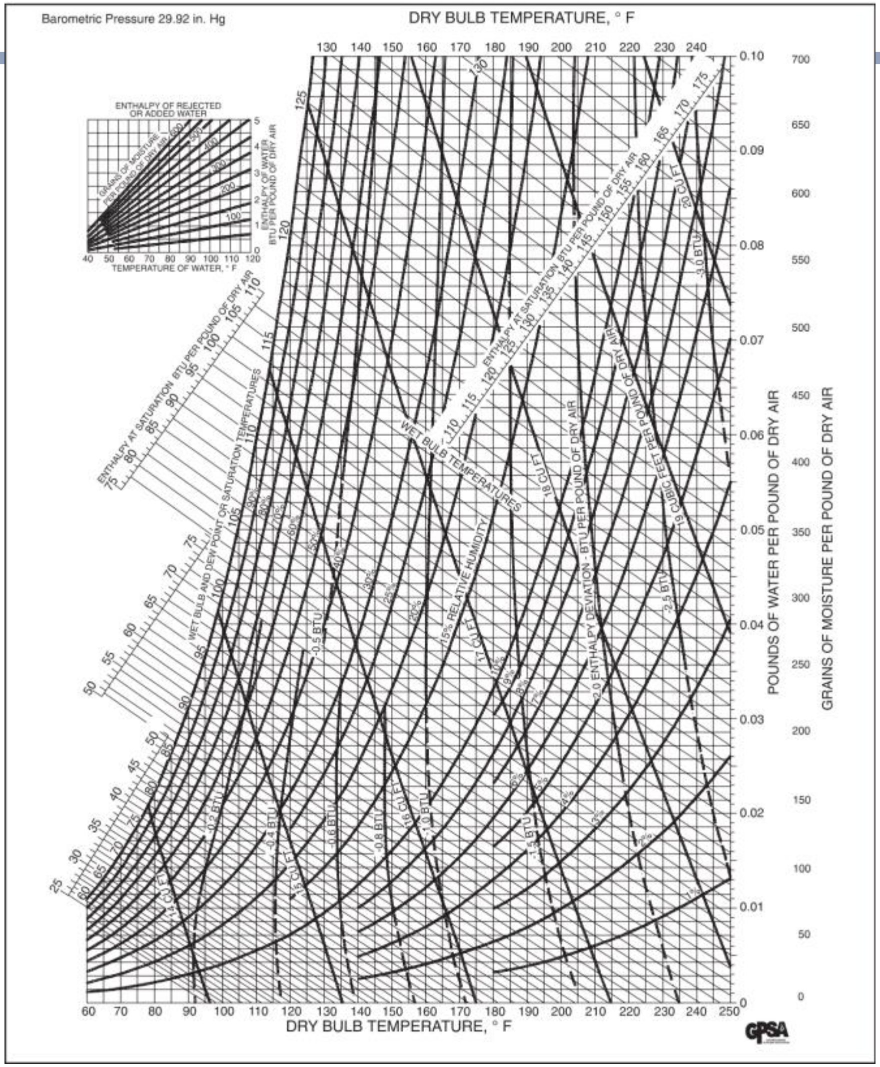
Cooling Tower Principles

Evaporative cooling (Psychrometry)

- Dry Bulb versus Wet Bulb Temperature
  - Contact dry air with water
  - Saturation of air (vaporization of some water) takes energy
  - Air is cooled below ambient – to "Wet Bulb" temperature
- Takes advantage of air below 100% humidity
  - Wet Bulb MUST be lower than Dry Bulb temperature

FIG. 11-2

Psychrometric Chart



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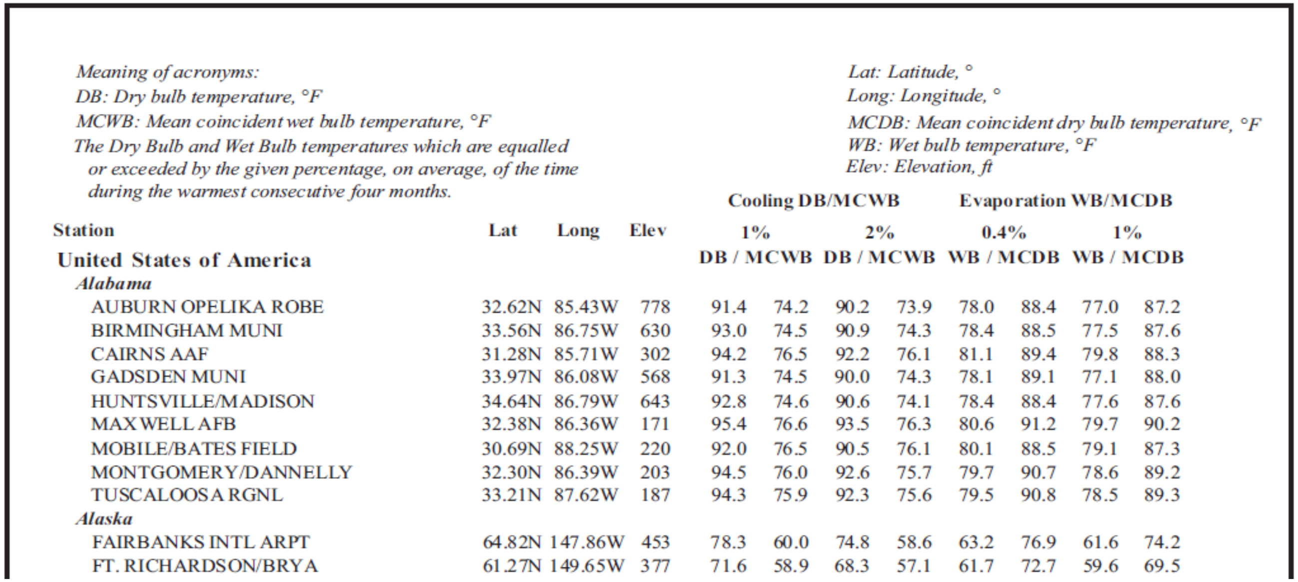
Cooling Tower Principles

Evaporative cooling (Psychrometry)

- Wet bulb and dry bulb data for various locations around the world Fig 11-3

FIG. 11-3

Dry Bulb/Wet Bulb Temperature Data for Selected Locations^2



Meaning of acronyms:
DB: Dry bulb temperature, °F
MCWB: Mean coincident wet bulb temperature, °F
The Dry Bulb and Wet Bulb temperatures which are equalled or exceeded by the given percentage, on average, of the time during the warmest consecutive four months.

Lat: Latitude, °
Long: Longitude, °
MCDB: Mean coincident dry bulb temperature, °F
WB: Wet bulb temperature, °F
Elev: Elevation, ft

				Cooling DB/MCWB				Evaporation WB/MCDB			
Station	Lat	Long	Elev	1%		2%		0.4%		1%	
United States of America				DB	MCWB	DB	MCWB	WB	MCDB	WB	MCDB
Alabama											
AUBURN OPELIKA ROBE	32.62N	85.43W	778	91.4	74.2	90.2	73.9	78.0	88.4	77.0	87.2
BIRMINGHAM MUNI	33.56N	86.75W	630	93.0	74.5	90.9	74.3	78.4	88.5	77.5	87.6
CAIRNS AAF	31.28N	85.71W	302	94.2	76.5	92.2	76.1	81.1	89.4	79.8	88.3
GADSDEN MUNI	33.97N	86.08W	568	91.3	74.5	90.0	74.3	78.1	89.1	77.1	88.0
HUNTSVILLE/MADISON	34.64N	86.79W	643	92.8	74.6	90.6	74.1	78.4	88.4	77.6	87.6
MAXWELL AFB	32.38N	86.36W	171	95.4	76.6	93.5	76.3	80.6	91.2	79.7	90.2
MOBILE/BATES FIELD	30.69N	88.25W	220	92.0	76.5	90.5	76.1	80.1	88.5	79.1	87.3
MONTGOMERY/DANNELLY	32.30N	86.39W	203	94.5	76.0	92.6	75.7	79.7	90.7	78.6	89.2
TUSCALOOSA RGNL	33.21N	87.62W	187	94.3	75.9	92.3	75.6	79.5	90.8	78.5	89.3
Alaska											
FAIRBANKS INTL ARPT	64.82N	147.86W	453	78.3	60.0	74.8	58.6	63.2	76.9	61.6	74.2
FT. RICHARDSON/BRYA	61.27N	149.65W	377	71.6	58.9	68.3	57.1	61.7	72.7	59.6	69.5

Example:

How cold can you get?

Air temperature: 95° F

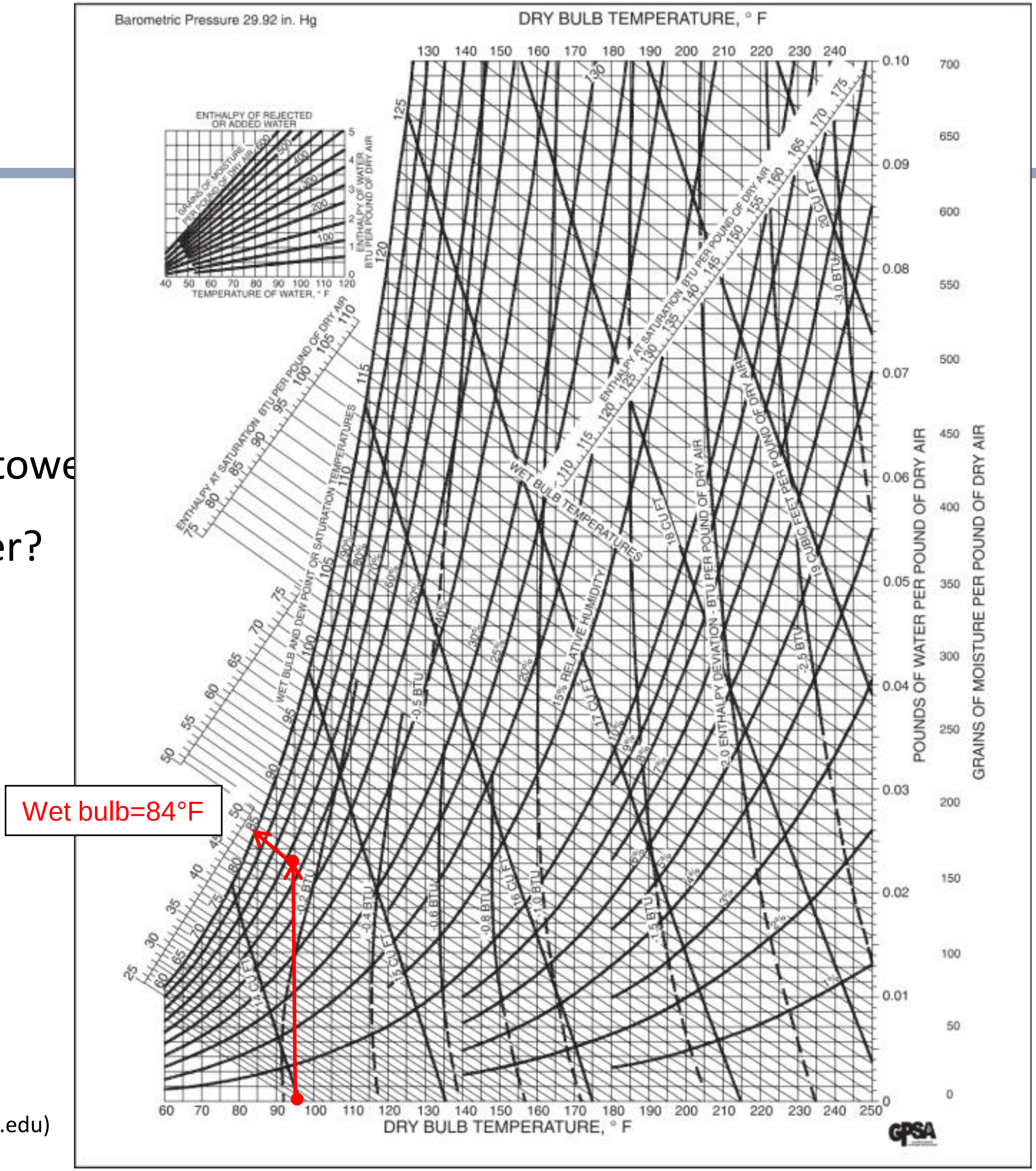
RH = 65%

Temperature with cooling towe

Temperature with air cooler?

FIG. 11-2

Psychrometric Chart



Wet bulb=84°F

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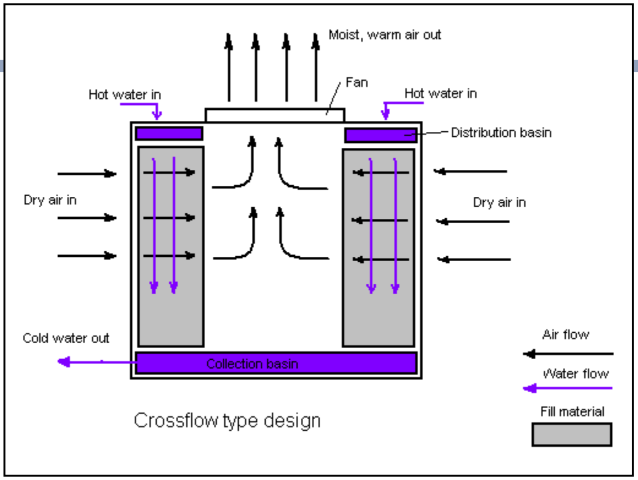
Cooling Towers – Mechanical Induced Draft

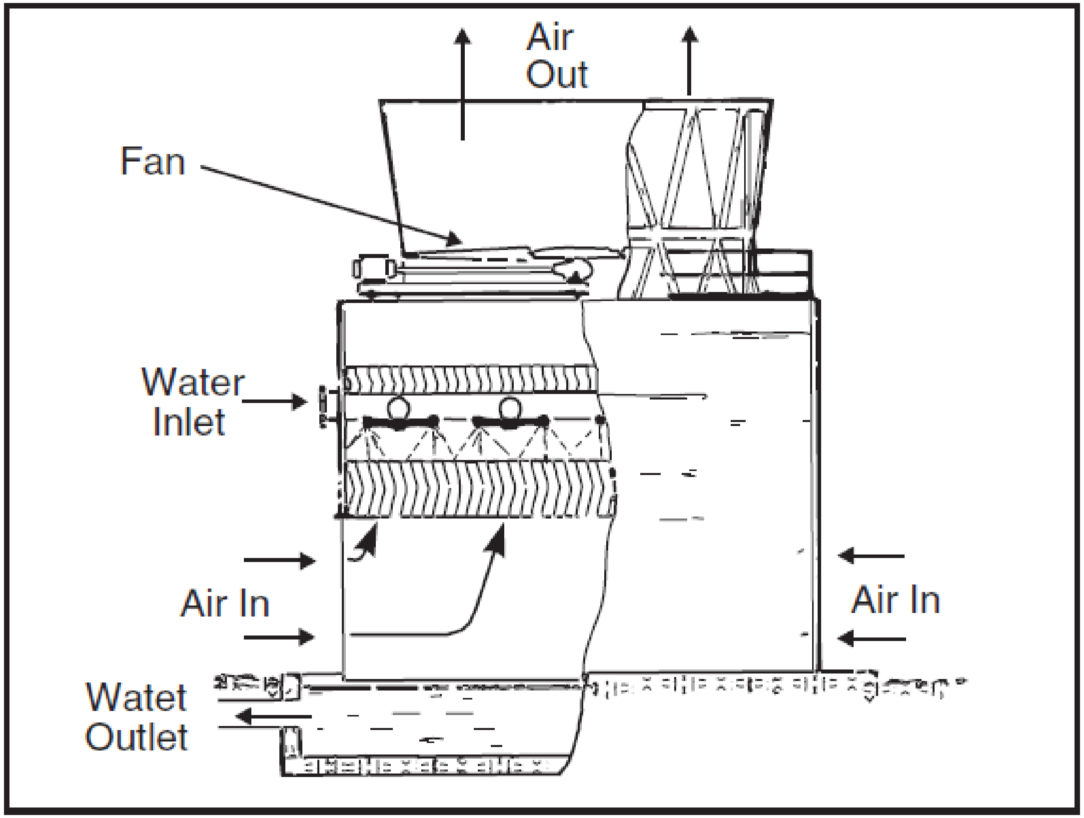


www.iklimnet.com

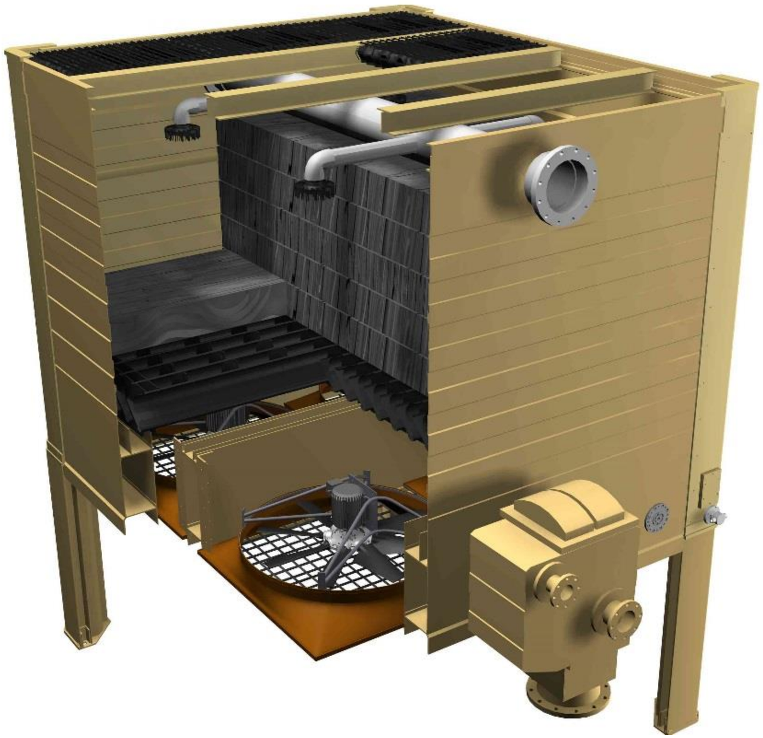
www.rjdesjardins.com

FIG. 11-7

Mechanical Induced Draft Counterflow Tower



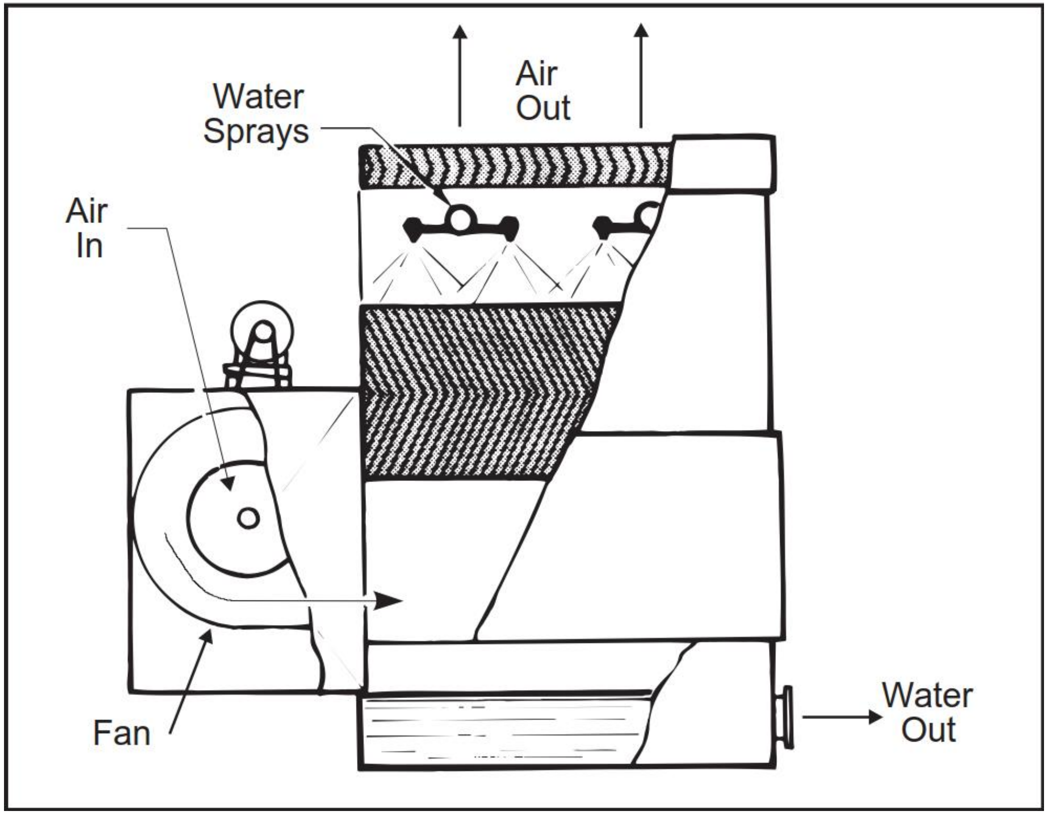
Cooling Towers – Mechanical Forced Draft



Towertechinc.com

FIG. 11-6

Mechanical Forced Draft Counterflow Tower



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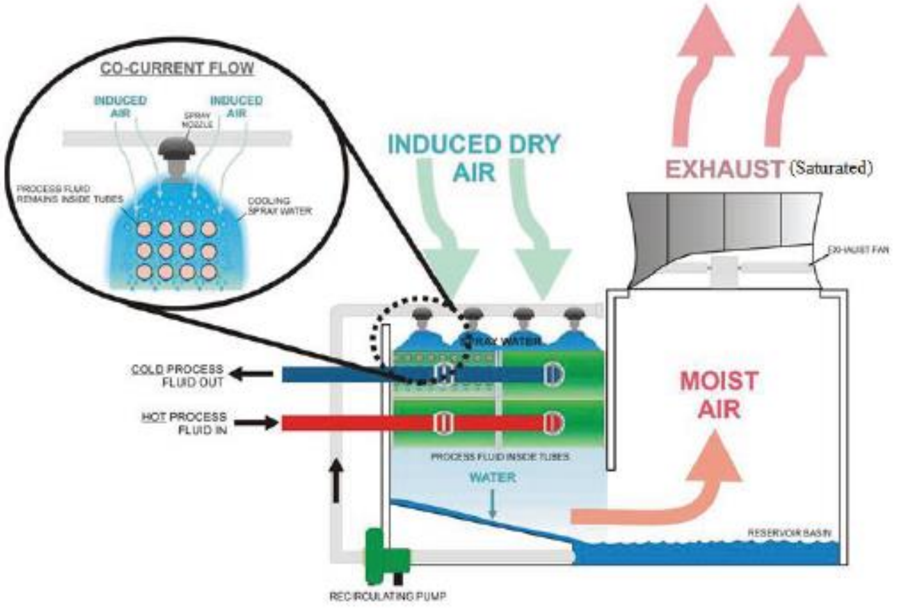
Cooling Towers – Wet Surface Air Cooler

www.niagarablower.com

FIG. 11-12

Wet Surface Air Cooler



Courtesy of Niagara Blower

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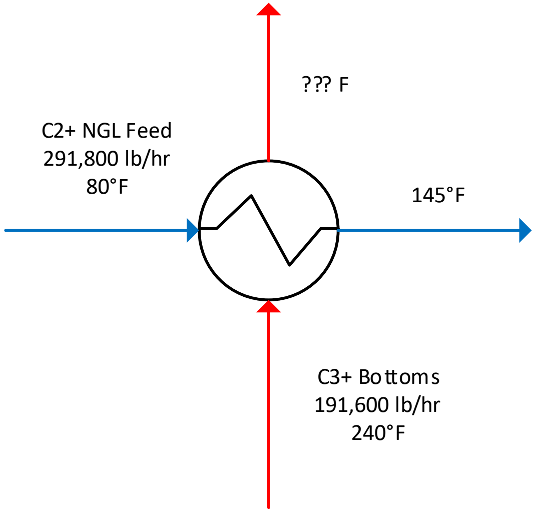
Heat Exchanger – Example 1

Heat 291,800 lb/hr cold C2+ NGL feed from 80°F to 105°F using 191,600 lb/hr hot C3+ bottoms @ 240°F

- Assume only sensible heat effects
  - C2+ NGL feed heat capacity – 0.704 Btu/lb F
  - C3+ Bottoms heat capacity – 0.830 Btu/lb F

Determine

- C3+ Bottoms outlet temperature
- Exchanger duty
- (UA) for the exchanger
- **Does the flow configuration in a 1-2 exchanger make a difference?**



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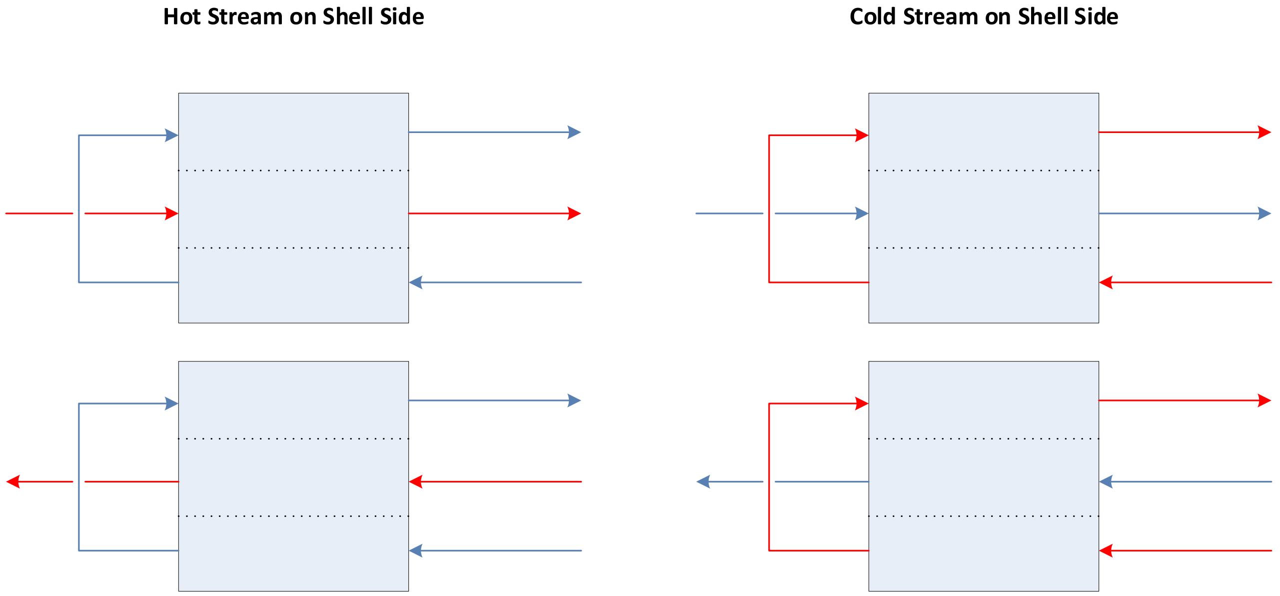
Heat Exchanger – Example 1

Determination of UA requires configuration information

- 1-2 (1 shell & 2 tube passes) combines both counter & co-current flow

Four possible flow configurations



Hot Stream on Shell Side

Cold Stream on Shell Side

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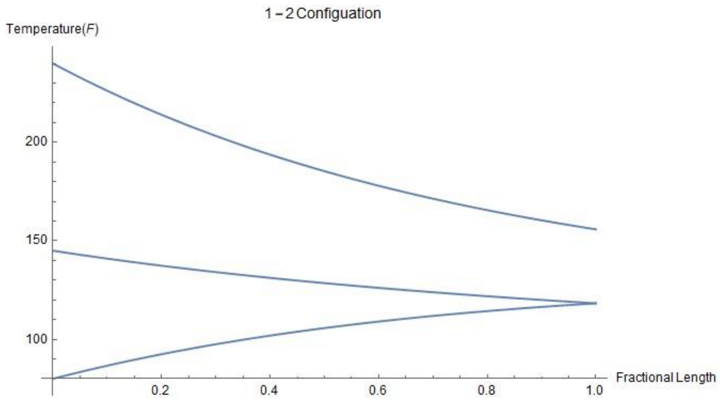
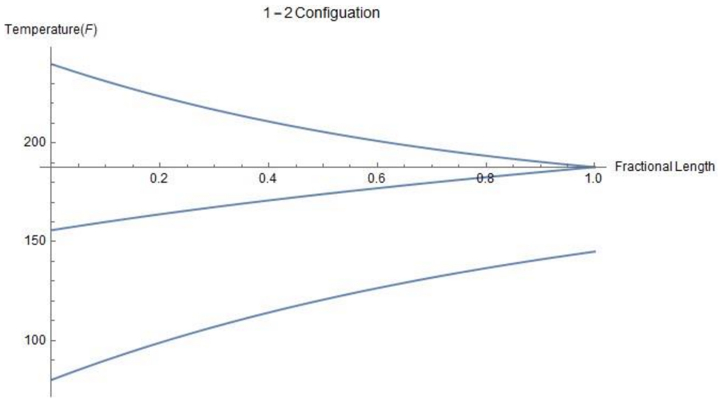
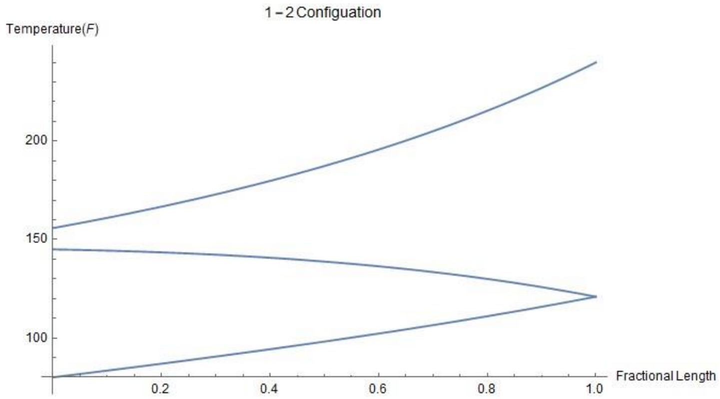
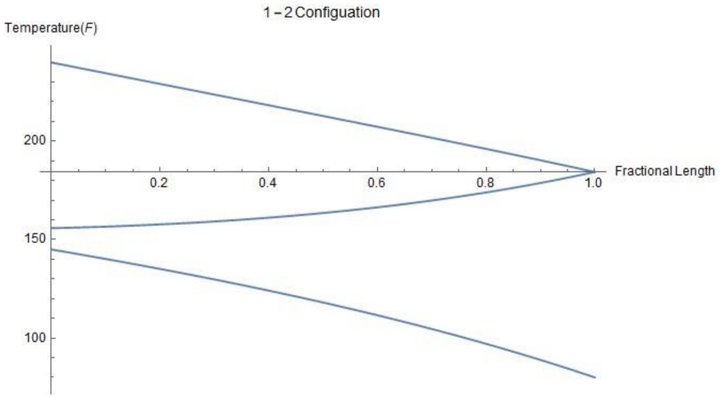
Heat Exchanger – Example 1

Four possible flow configurations – all have the same exit temperatures but different internal profiles









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Heat Exchanger – Example 7

I thought you said temperature crossovers weren't possible????