



Training Services

Shell & Tube Heat Exchangers

IPD-2004/HE-1

Introduction to Heat Transfer

- Introduction to Heat Transfer Mechanisms
- Conduction, Convection, and Radiation
- Overall Heat Transfer Coefficient
- Log Mean Temperature Difference
- Hydraulics
- Example Problems - Tube Side Pressure Drop

IPD-2004/HE-2

Heat Transfer Basics

Three Forms of Heat Transfer

- Conduction
- Convection
- Radiation

IPD-2004/HE-3

Conduction Heat Transfer

Heat transfer within a material.

Fourier's equation governs.

$$Q = A k \frac{dT}{dx}$$

Q is amount of heat transferred

A is the area for heat transfer

k is the thermal conductivity

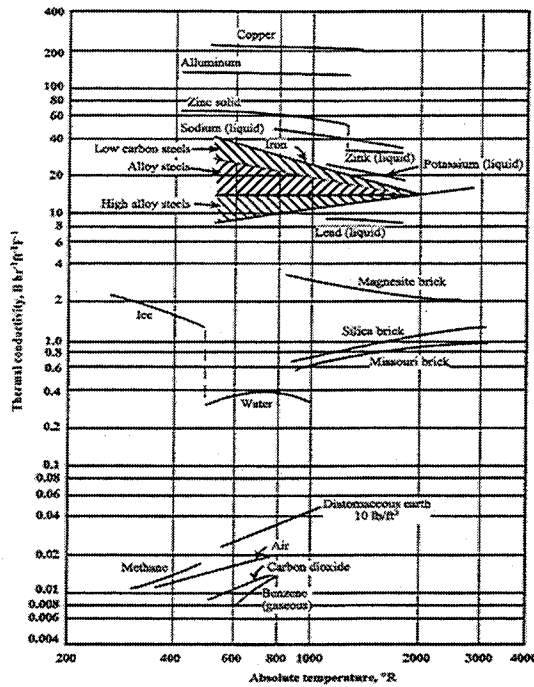
T is the temperature

x is the distance in direction of heat flow

IPD-2004/HE-4

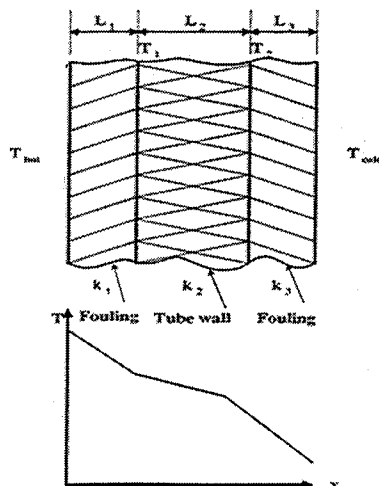
Conduction Heat Transfer

Thermal Conductivity of Various Materials



EXC-R00-01
IPD-2004/HE-5

Resistances in Series



Tube metal has higher conductivity than fouling

EXC-R00-03
IPD-2004/HE-6

Resistances in Series

(continued)

Heat flux is calculated as follows:

$$\frac{Q}{A} = \frac{T_h - T_c}{\left(\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} \right)} = k_{\text{eff}} (T_h - T_c)$$

$$\text{so } k_{\text{eff}} = \left(\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} \right)^{-1}$$

IPD-2004/HE-7

Convection Heat Transfer

Heat transfer between solids and liquids/vapors

$$Q = A h \Delta T$$

where h is a function of physical properties and geometry

IPD-2004/HE-8

Single Phase Convection Heat Transfer

Tubeside heat transfer coefficient

Use Dittus-Boelter Equation:

$$h_i = \frac{k}{D_i} 0.023 \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{D_i v \rho}{\mu} \right)^{0.8}$$

collect variables:

$$h_i = 0.023 \frac{k^{2/3}}{D_i^{0.2}} C_p^{1/3} \mu^{-0.467} (v \rho)^{0.8}$$

Correct for Outside Surface Area:

$$h_{io} = \left(\frac{D_i}{D_o} \right) h_i$$

increase this	hi
k	increases
Di	DECREASES
Cp	increases
μ	DECREASES
ρ	increases
v	increases

k = Thermal conductivity
 C_p = Specific heat
 μ = Viscosity
 ρ = Density
v = Velocity
 D_i = Inside diameter

IPD-2004/HE-9

Single Phase Convection Heat Transfer (continued)

Shellside Coefficient

Equation from Kern:

$$h_o = 0.36 \frac{k}{D_e} \left(\frac{D_e v \rho}{\mu} \right)^{0.55} \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

Collect Variables:

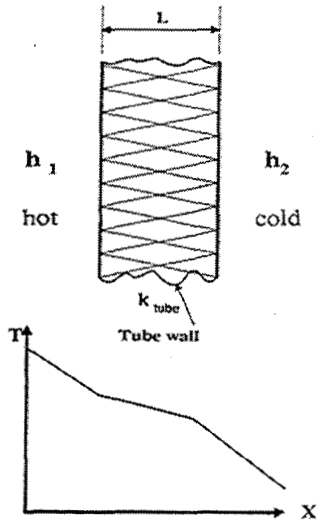
$$h_o = 0.36 \frac{k^{2/3}}{D_e^{0.45} \mu^{0.22}} (v \rho)^{0.55} C_p^{1/3} \left(\frac{\mu}{\mu_w} \right)^{0.14}$$

The last term (viscosity correction) is usually small, and we will ignore it.

increase this	ho
k	increases
De	DECREASES
Cp	increases
μ	DECREASES
ρ	increases
v	increases

IPD-2004/HE-10

Combined Conduction and Convection Coefficients



EXC-R00-04
IPD-2004/HE-11

Combined Conduction and Convection (continued)

$$\frac{Q}{A} = \frac{T_h - T_c}{\left(\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} \right)} = k_{\text{eff}} (T_h - T_c)$$

$\nearrow$ h_1 $\nearrow$ h_2 $\nearrow$ substitute as shown
 U for k_{eff} (convention)

$$\frac{Q}{A} = \frac{T_h - T_c}{\left(\frac{1}{h_1} + \frac{L}{k_{\text{tube}}} + \frac{1}{h_2} \right)} = U (T_h - T_c)$$

There may be a few solid layers present (tube, 2 fouling layers).

IPD-2004/HE-12

Combined Conduction and Convection ***(continued)***

The overall resistance is the sum of the individual resistances.

$$\frac{Q}{A} = \frac{T_h - T_c}{\left(\frac{1}{h_i} + \frac{x_{fi}}{k_{fi}} + \frac{L}{k_{tube}} + \frac{x_{fo}}{k_{fo}} + \frac{1}{h_o} \right)} = U (T_h - T_c)$$

or

$$Q = U A \Delta T$$

IPD-2004/HE-13

Combined Conduction and Convection ***(continued)*** ***Round Geometry***

Tube wall is curved; outside area is different from inside area: tubewall resistance is calculated as:

$$U = \left(\frac{1}{h_i} \left(\frac{r_o}{r_i} \right) + \frac{r_o \cdot \log_e \left(\frac{r_o}{r_i} \right)}{k_{tube}} + \frac{1}{h_o} \right)^{-1}$$

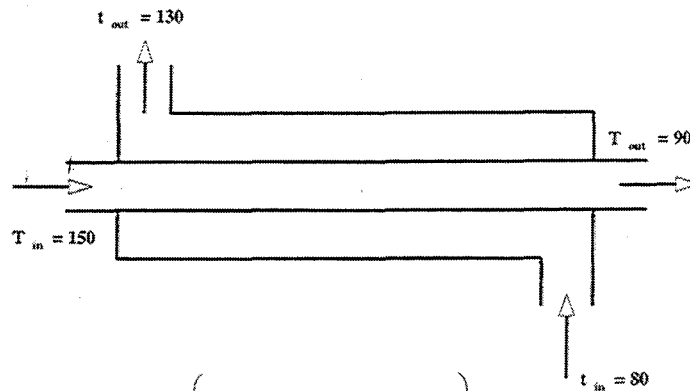
or if you include fouling resistances

$$U = \left(\frac{1}{h_i} \left(\frac{r_o}{r_i} \right) + \frac{r_o \cdot \log_e \left(\frac{r_o}{r_i} \right)}{k_{tube}} + \frac{1}{h_o} + r_{fi} \left(\frac{r_o}{r_i} \right) + r_{fo} \right)^{-1}$$

IPD-2004/HE-14

Temperature Difference

When calculating duty, what ΔT should be used here?



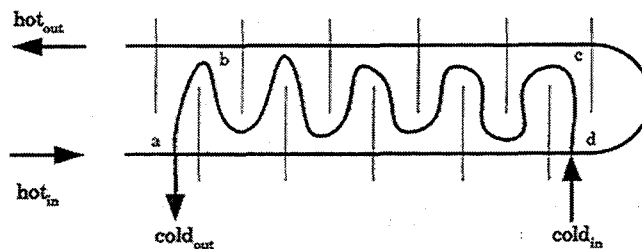
$$\Delta T_{\text{eff}} = \left(\frac{(T_{\text{in}} - t_{\text{out}}) - (T_{\text{out}} - t_{\text{in}})}{\log_e \left(\frac{T_{\text{in}} - t_{\text{out}}}{T_{\text{out}} - t_{\text{in}}} \right)} \right) = \Delta T_{\text{log mean}}$$

EXC-R00-05
IPD-2004/HE-15

Temperature Difference (continued)

In practice, pure countercurrent flow situations often do not exist.

For example: Most exchangers have more than one tube-pass.

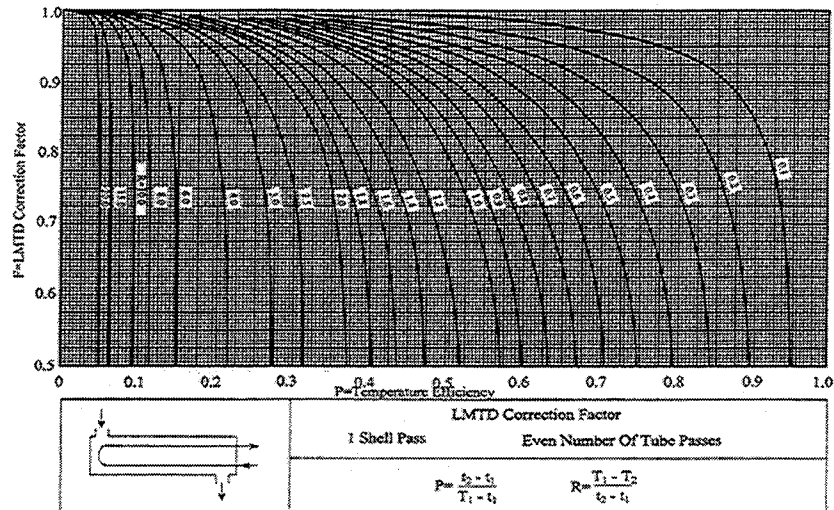


What is the **EFFECTIVE TEMPERATURE** difference in the exchanger?

EXC-R00-06
IPD-2004/HE-16

Correction Factors

E Shell - 1 Shell Pass



EXC-R00-47
IPD-2004/HE-17

Hydraulics

- Tubeside ΔP is easy to calculate.
- Calculations similar to pipe flow calculations.

$$\Delta P_t = \frac{2f_t v^2 L \rho n}{D_i}$$

$$\text{where } f_t \approx 0.046 \left(\frac{D_i v \rho}{\mu} \right)^{-0.19}$$

L = Length of tubes
n = Number of passes
 f_t = Friction factor

IPD-2004/HE-18

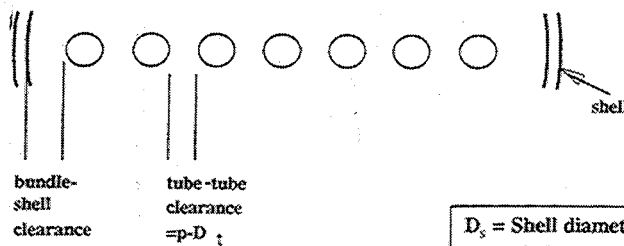
Hydraulics (continued)

Shellside ΔP

Need to know what area to use.

Use flow area between tubes at center of shell:

$$\# \text{ tubes} = D_s / p$$



$$A_{fs} = \frac{D_s}{p} (p - D_t) B_s$$

D_s = Shell diameter
 p = pitch
 B_s = Baffle spacing

EXC-R00-08
 IPD-2004/HE-19

pitch = centre to centre tube spacing

Hydraulics (continued)

- For small bundles or accurate calculation, allow for bundle-shell clearance.

$$\Delta P_s = \frac{f_s D_s (N_b + 1) \rho v^2}{2 D_e}$$

$$v = \frac{W_s}{A_{fs}}$$

$$f_s = 1.79 \left(\frac{D_e v \rho}{\mu} \right)^{-0.19}$$

$$D_e = \frac{4 \left(p^2 - \frac{\pi D_t^2}{4} \right)}{\pi D_t}$$

N_b = Number of baffles
 W_s = Volumetric flow rate
 D_e = Equivalent diameter

IPD-2004/HE-20

Hydraulics in General

ΔP varies with flow rate, length, and diameter scales:

$$\Delta P \propto f(\text{geometry, fluid, flow}) \cdot v^2 \cdot \frac{\text{LengthScale}}{\text{DiamScale}} \cdot \rho$$
$$\Rightarrow \frac{\Delta P_2}{\Delta P_1} = \left(\frac{v_2}{v_1} \right)^2 \left(\frac{L_2}{L_1} \right) \left(\frac{D_1}{D_2} \right)$$

Length Scale (L) is length measured in direction of flow (tube length, or shell diameter).

Diameter Scale is a length measured perpendicular to flow (tube ID or equivalent diameter).

This general form is valid for both shell and tubeside.

For highly turbulent flow, f does not change much with increase of flow rate.

IPD-2004/HE-21

Quick Estimations

Example 1

If existing exchanger has tube $\Delta P = 70$ kPa, and revamp needs extra 10% flow:

$$\Delta P_{\text{revamp}} \approx 70 \text{ kPa} (1.1)^2 = 70 (1.21) = 85 \text{ kPa}$$

Example 2

A heat exchanger design has 2 tube passes, $\Delta P_{\text{tube}} = 15$ kPa (too low).

- Someone proposes changing to 4 tube passes
velocity doubles, number of passes doubles:

$$\Delta P = 15 \cdot 2^2 \cdot 2 = 15 \cdot 8 = 120 \text{ kPa} \quad (\text{too high}).$$

IPD-2004/HE-22

Quick Estimations (continued)

Example 3

Retube existing exchanger from 1" OD 16 BWG (0.065" wall) tubes to 1" OD 14 BWG (0.083" wall) tubes (same flow rate)

Old Tube ID is = 1" - 2 (0.065") = 0.870"

New Tube ID is = 1" - 2 (0.083") = 0.834"

$$\frac{\Delta P_{14}}{\Delta P_{16}} = \left(\frac{v_{14}}{v_{16}} \right)^2 \frac{0.870}{0.834}$$

$$= \left(\left(\frac{0.870}{0.834} \right)^2 \right)^2 \frac{0.870}{0.834} = \left(\frac{0.870}{0.834} \right)^5 = 1.24$$

Therefore, this retubing will increase the pressure drop by 24%.

IPD-2004/HE-23

Introduction to Shell and Tube Exchangers

- | | |
|----------------------------------|-----------------------------------|
| ■ Design Requirements | ■ Baffle Types |
| ■ TEMA Nomenclature | ■ Tube to Tubesheet Joints |
| ■ Selection of TEMA Type Layouts | ■ Materials of Construction |
| ■ Tube Pass Geometry | ■ Protective Coatings |
| ■ Minimizing Leakage Streams | ■ Design Temperature and Pressure |
| ■ Pass Partitions | |

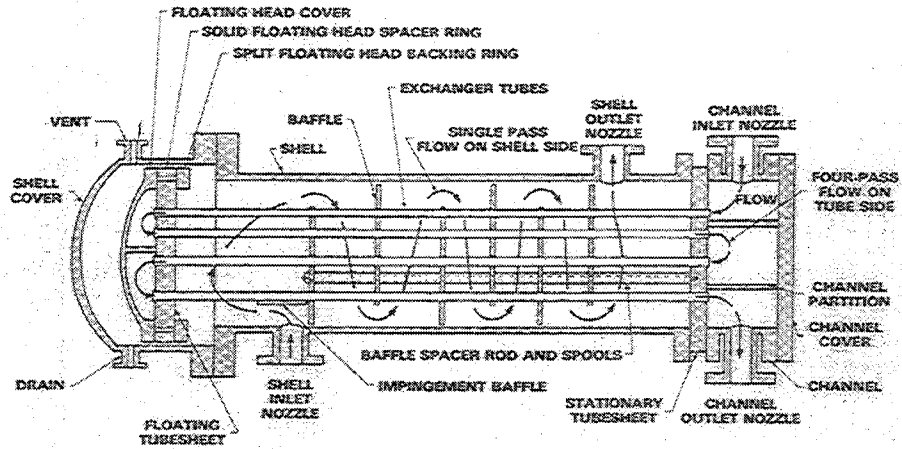
IPD-2004/HE-24

EXC-R00-09
IPD-2004/HE-29

IPD-2004/HE-26

Shell and Tube Heat Exchangers

CROSS-VIEW OF FLOATING TUBE SHEET EXCHANGER



IPD-2004/HE-27

Design Requirements

What are the key parameters that are used to develop a design for a heat exchanger?

- Fluid Streams
 - Flow rate
 - Thermophysical properties of fluid
 - Temperature
 - Pressure
 - Pressure drop limitations
 - Phase (liquid/vapor/mixture)
- Fluid Streams (cont.)
 - Boiling/condensing
 - Fouling potential
 - Corrosion potential
- Geometric Concerns
 - Space constraints
 - Weight constraints
 - Manufacturing limits

IPD-2004/HE-28

Design Requirements (continued)

■ Operational Concerns

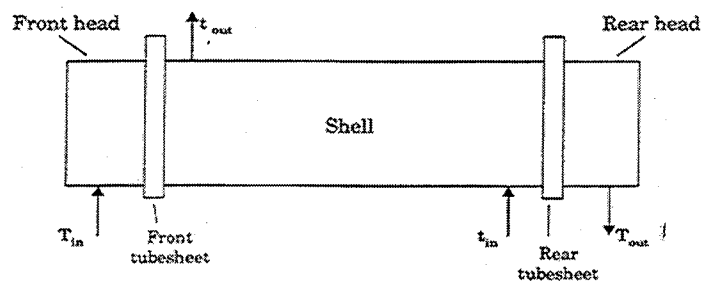
- Spare capacity
- Equipment failure
- Control

■ Overriding Concerns

- Economics
- Economics
- Economics

IPD-2004/HE-29

Tubular Exchanger Layout



Fixed Tubesheet Exchanger

Advantages:

- Lowest cost.
- No shell side gaskets.
- Tubes are replaceable and cleanable.
- Maximum number of tubes for given diameter.

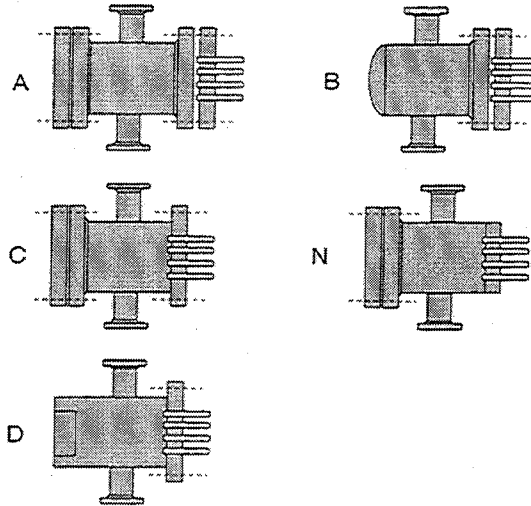
Disadvantages:

- Sensitive to temperature excursions. Differential expansion is a problem.
- Shell side cannot be cleaned other than by solvents.
- Bundle entrance velocity often too high.
- Difficult baffle construction.

EXC-R00-10
IPD-2004/HE-30

TEMA - Tubular Exchanger Manufacturers Association

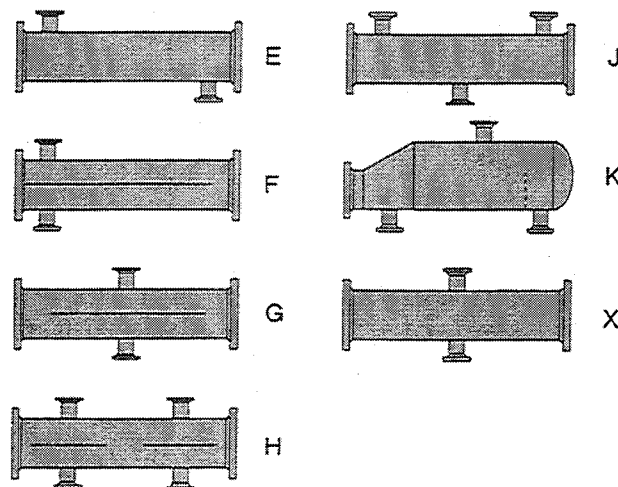
Heat Exchanger Nomenclature - Front Heads



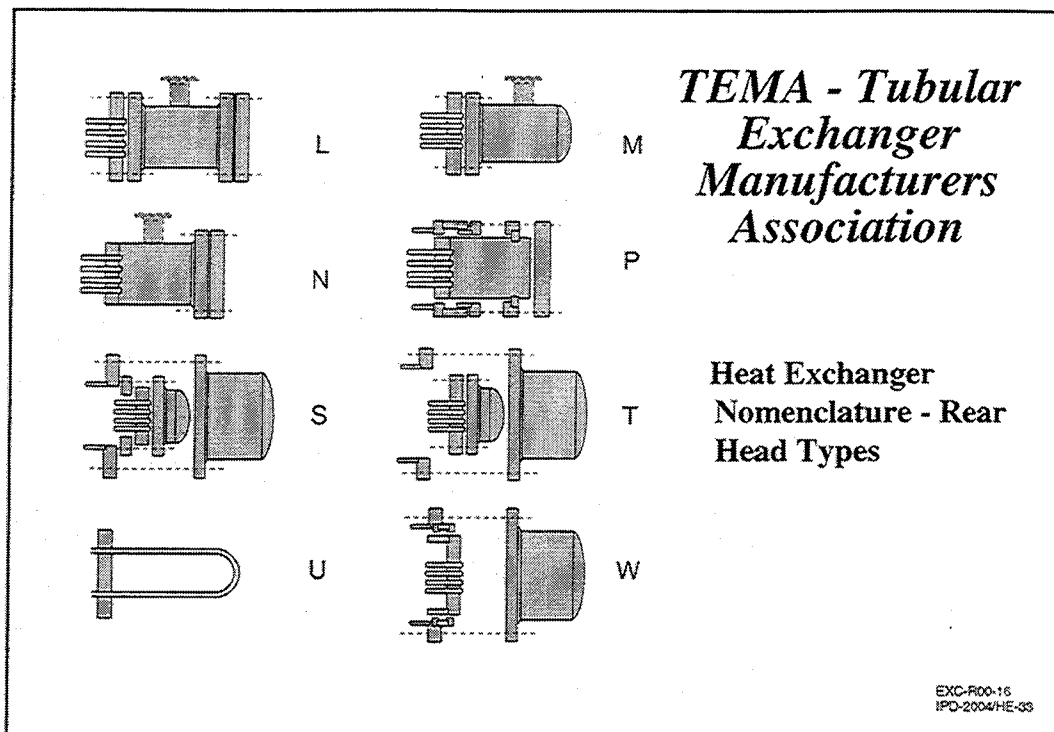
EXC-R00-14
IPD-2004/HE-31

TEMA - Tubular Exchanger Manufacturers Association

■ Heat Exchanger Nomenclature - Shell Types



EXC-R00-15
IPD-2004/HE-32



Selection of Exchanger Type

- 1) Feed preheater
Low pressure
Tubeside - Steam
Shellside - Light naphtha

Selection of Exchanger Type
(continued)

- 1) Feed preheater
 - Low pressure
 - Tubeside - Steam
 - Shellside - Light naphtha
- BEU

IPD-2004/HE-35

Selection of Exchanger Type
(continued)

- 2) Feed preheater
 - Low pressure
 - Tubeside - Coker fractionator bottoms
 - Shellside - Light coker naphtha

IPD-2004/HE-36

Selection of Exchanger Type ***(continued)***

2) Feed preheater

Low pressure

Tubeside - Coker fractionator bottoms

Shellside - Light coker naphtha

AES or AET

IPD-2004/HE-37

Selection of Exchanger Type ***(continued)***

3) Xylene column reboiler

Low pressure

Tubeside - Steam

Shellside - Mixed aromatics

IPD-2004/HE-38

Selection of Exchanger Type (continued)

3) Xylene column reboiler

Low pressure

Tubeside - Steam

Shellside - Mixed aromatics

A or B, E or H or J or X, U

A or B, K, T or U probably BHU

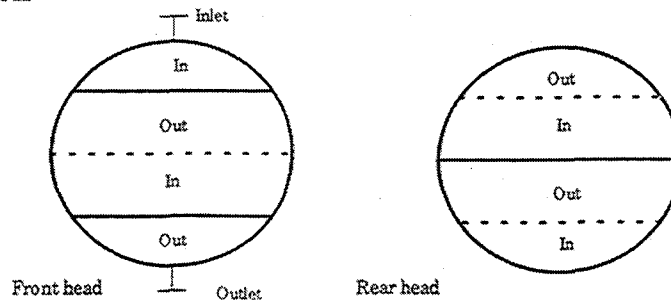
IPD-2004/HE-39

Tube Pass Geometry and Pass Partitions

Separation of fluids in each pass must be maintained since leakage would short circuit the exchanger.

The term "Pass" used in exchangers is not the same as the term pass is used in fired heaters. The number of passes in an exchanger is the number of times one fluid passes through the other fluids compartment.

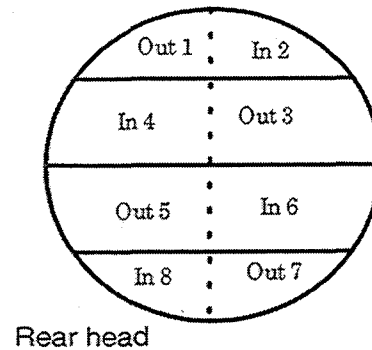
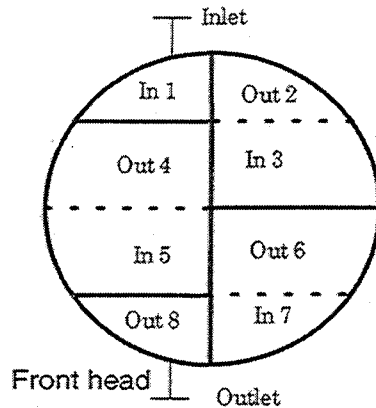
Ribbon



EXC-R00-19
IPD-2004/HE-40

Tube Pass Geometry and Pass Partitions (continued)

Quadrant

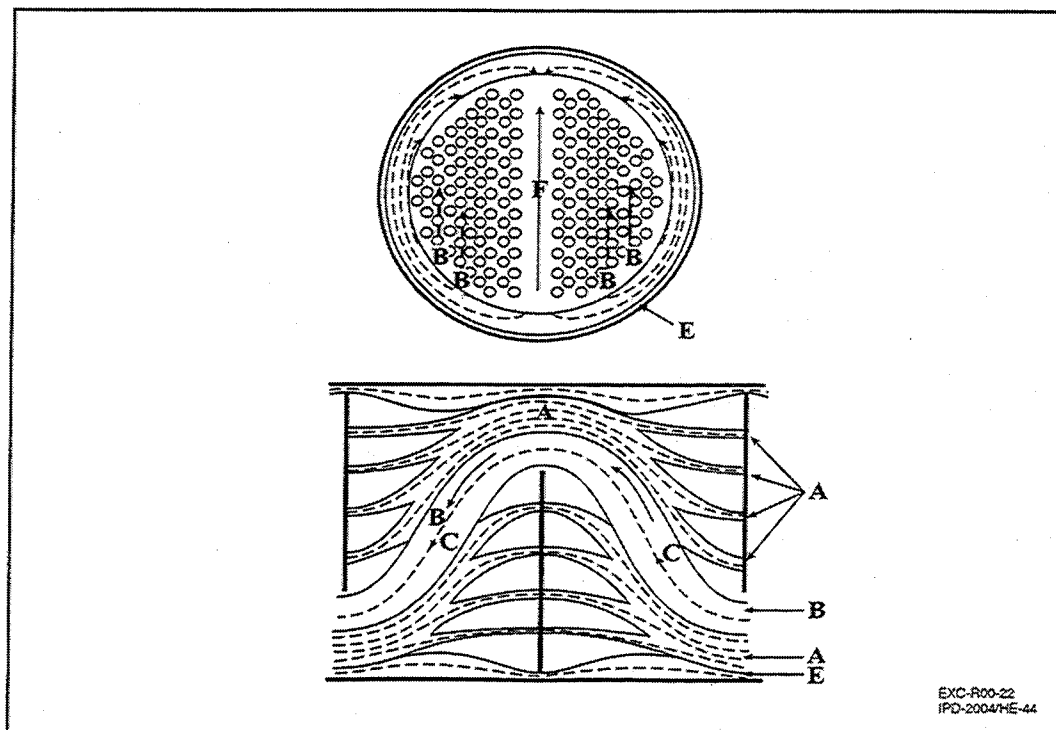
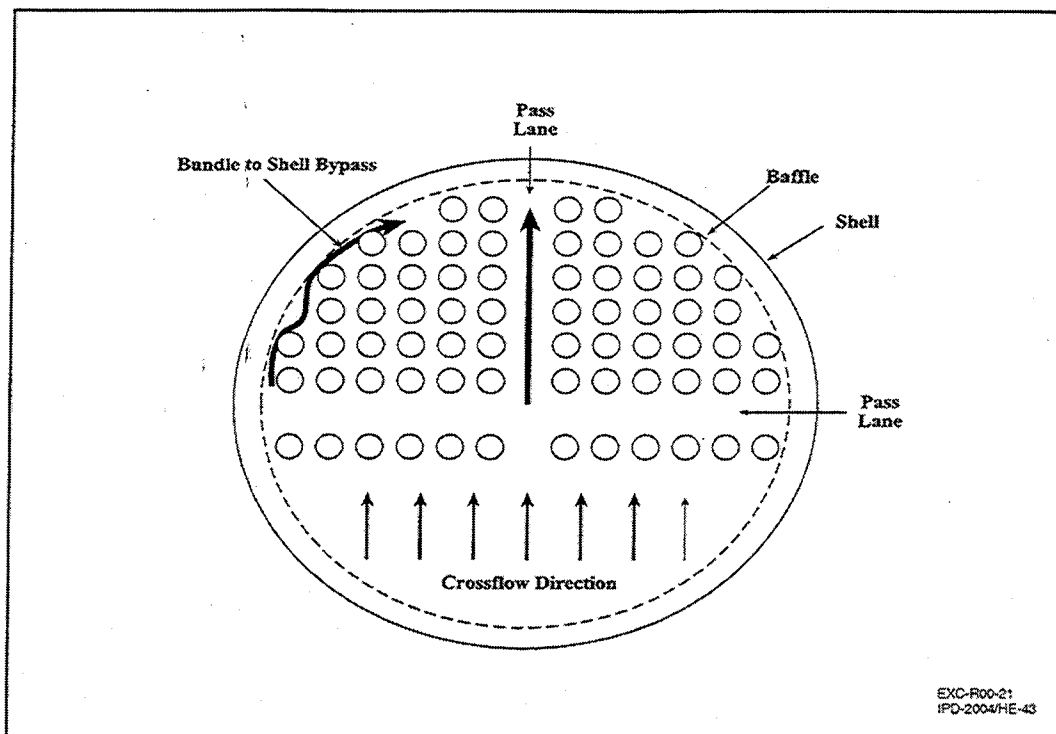


EXC-R00-20
IPD-2004/HE-41

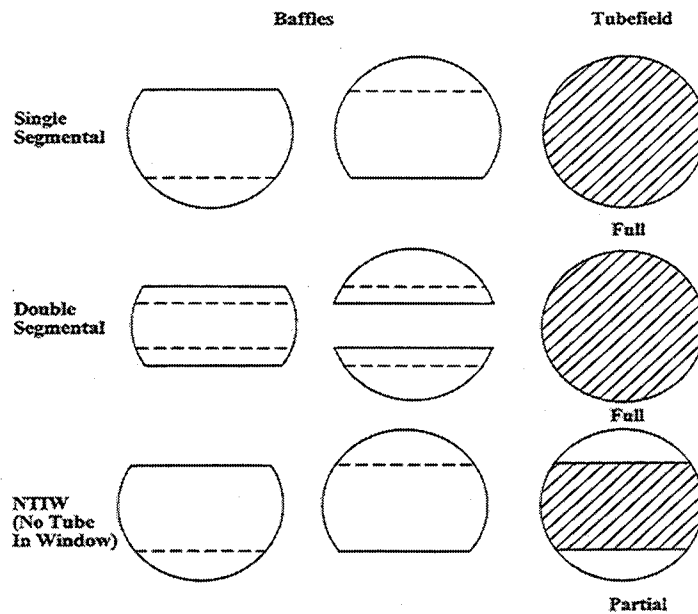
Shellside Flow Distribution

- Fluid flows through the shell in five paths
 - A Stream - Through gap between tube and baffle
 - B Stream - Between tubes across the bundle
 - C Stream - Through gap between bundle and shell
 - E Stream - Through gap between baffle and shell
 - F Stream - Through pass partition lane

IPD-2004/HE-42

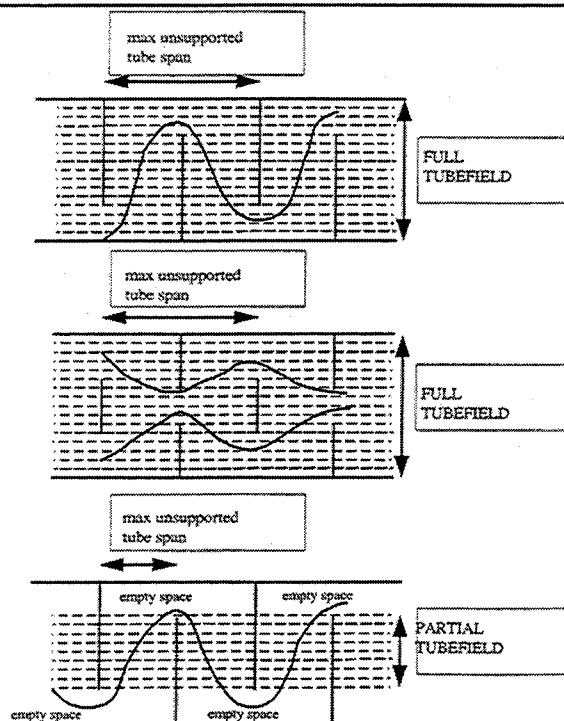


Baffle Types



EXC-R00-24
IPD-2004/HE-45

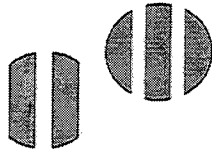
Flow Patterns



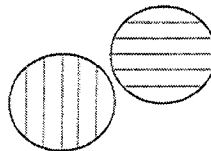
EXC-R00-25
IPD-2004/HE-46

Baffle Types

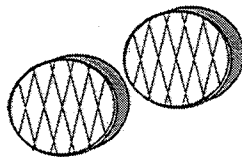
Triple Segmental Baffles



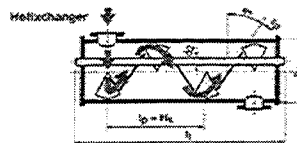
Phillips Rod Baffles



Holtec NEST Baffles



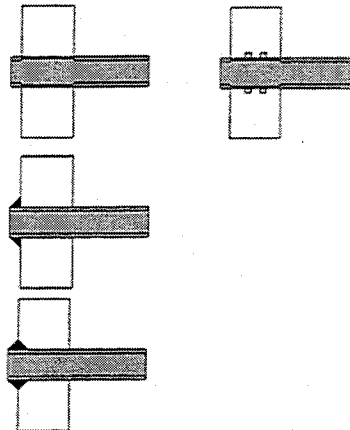
Spiral Baffles



EXC-R00-25
IPD-2004/HE-47

Tube to Tubesheet Designs

- Most difficult mechanical design component
- Large potential for internal leakage
 - Three main types
 - Expanded
 - Seal Welded
 - Strength Welded



EXC-R00-29
IPD-2004/HE-48

Materials of Construction

- Materials must suit temperature and fluids
- Example of selection criteria
- High H₂ partial pressure
 - Killed carbon steel
- High temperature, high H₂ partial pressure
 - 410 or Cr-Mo
- Sea-water cooling
 - Admiralty brass or 70-30 Cu-Ni
- Wet H₂S service
 - Killed carbon steel (Special heat treatment)
- Hot HF service
 - Monel
- Hot acids
 - Alloy 20, 904L, 316 SS, Titanium

IPD-2004/HE-49

Protective Coatings

- Lower alloy requirements can be used if protected from process with a "coating". Coatings can take various forms.
 - Cladding: Cladding is welding a high alloy material on a lower alloy material. The low alloy provides the mechanical strength while the cladding provides corrosion protection.
 - Refractory: Refractory is a high temperature insulating material. By attaching it to a low alloy material, the temperature the material is exposed to is reduced allowing the use of a lower alloy.
 - Paints: Besides external painting for environmental protection, paint and similar coatings are used occasionally. Cooling water is a common use of this process.

IPD-2004/HE-50

Design Temperature and Pressure

- Dictated by worst case conditions for hot and cold use.
- Worst case conditions of both temperature and pressure. This may not be the highest temperature, but a combination of high temperature and higher pressure.
- Minimum design metal temperature (MDMT) is also required. High pressure at low temperatures may result in failure of brittle materials. Lowest ambient or auto-refrigeration temperatures normally used.

IPD-2004/HE-51

Protection of Low Pressure Side

- What happens to low pressure side if a tube ruptures.
- API 521 is reference standard.
- 10/13 Rule used for ASME Code Designs.
- Relief valve option.

IPD-2004/HE-52

Special Types of Heat Exchangers

- Thermosiphon Reboilers
- Kettle Reboilers
- Vertical Combined Feed Exchangers
- Hairpin Heat Exchangers
- Special Tubing and Tube Inserts
- Welded Plate Heat Exchangers

IPD-2004/HE-53

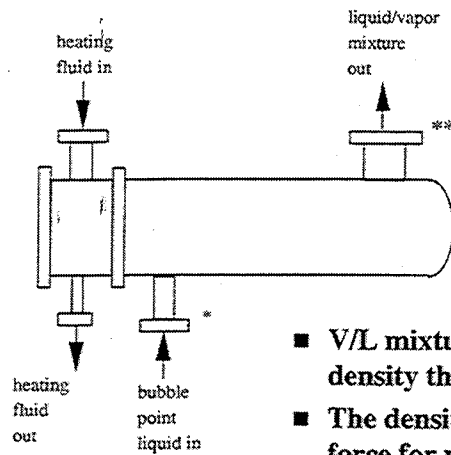
Special Exchanger Types Reboilers

What is the best way to boil column bottoms? Depends:

- Temperature sensitivity
 - Usually an issue in chemicals/pharmaceuticals production
- Allowable pressure drop/operating pressure of column
- Liquid and vapor properties
- Size considerations
 - Number of reboilers on column
 - Liquid-vapor line-sizing
- What sort of column bottoms arrangement is required?

IPD-2004/HE-54

Horizontal Thermosiphon

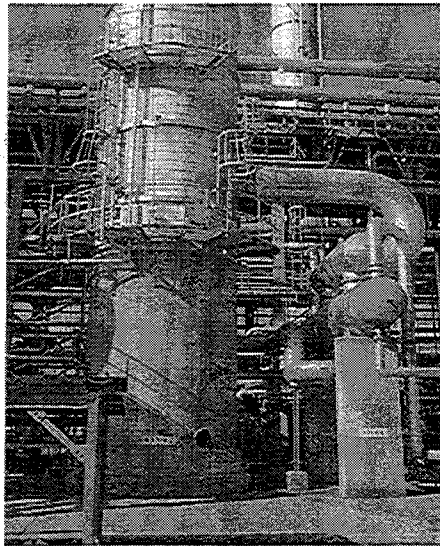


- V/L mixture at ** has much lower density than at *.
- The density difference is the driving force for process flow through the reboiler.

EXC-R00-32
IPD-2004/HE-55

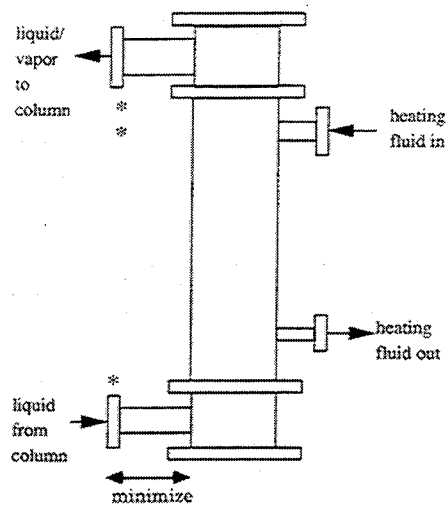
Horizontal Thermosiphon (continued)

- Horizontal Thermosiphon Reboiler on Benzene Column
- BHU Type Reboiler



IPD-2004/HE-56

Vertical Thermosiphon - Tubeside

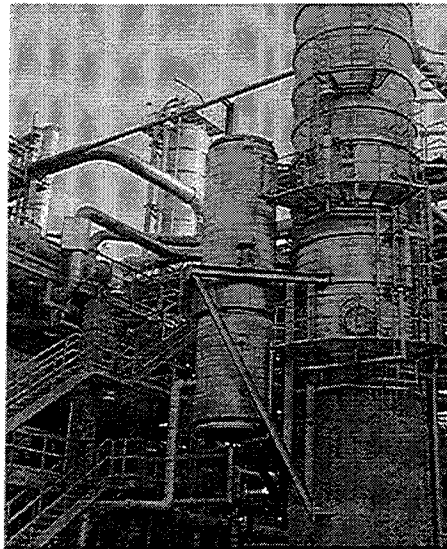


- V/L mixture at ** has much lower density than at *

EXC-R00-33
IPD-2004/HE-57

Vertical Thermosiphon (continued)

- Vertical Thermosiphon Reboiler on Sulfolane
- Dirty Fluid in Tubes



IPD-2004/HE-58

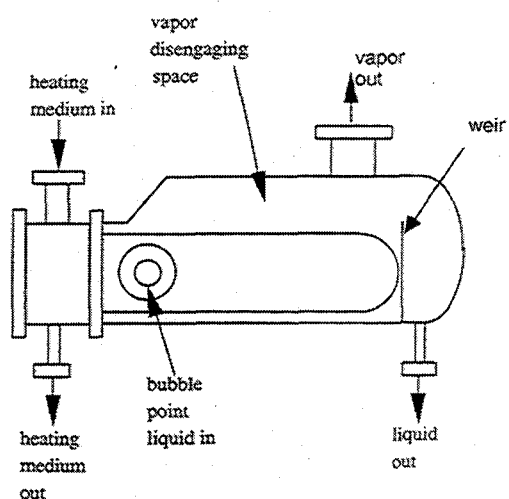
Vertical Thermosiphon (continued)

■ Characteristics

- Usually more expensive than equivalent duty horizontal thermosiphon
 - Tube lengths usually 16 feet maximum
 - Diameter larger, two larger tubesheets ⇒ \$\$\$
 - Usually more flanges than BEU ⇒ \$\$\$
 - TEMA type is usually NE(M or N)
- Uses very little plot space

IPD-2004/HE-59

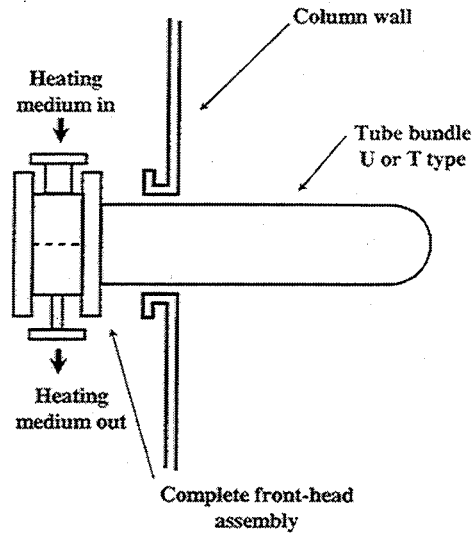
Kettle



- Size process lines carefully to limit pressure drops

EXC-R00-34
IPD-2004/HE-60

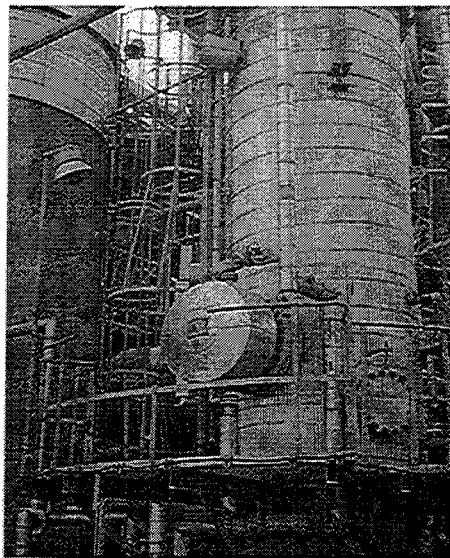
Stab-In Reboilers



EXC-R02-35
IPD-2004/HE-61

Stab-In Reboilers (continued)

■ Stab-In Reboiler and Column

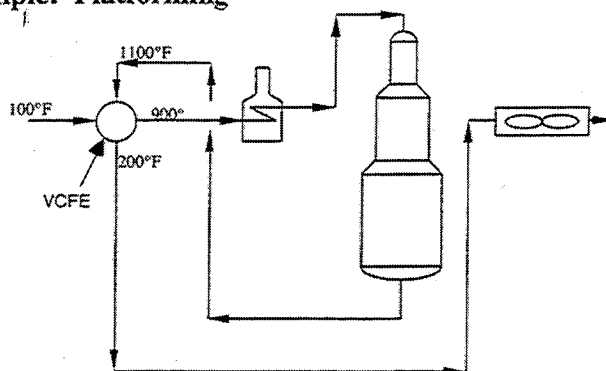


IPD-2004/HE-62

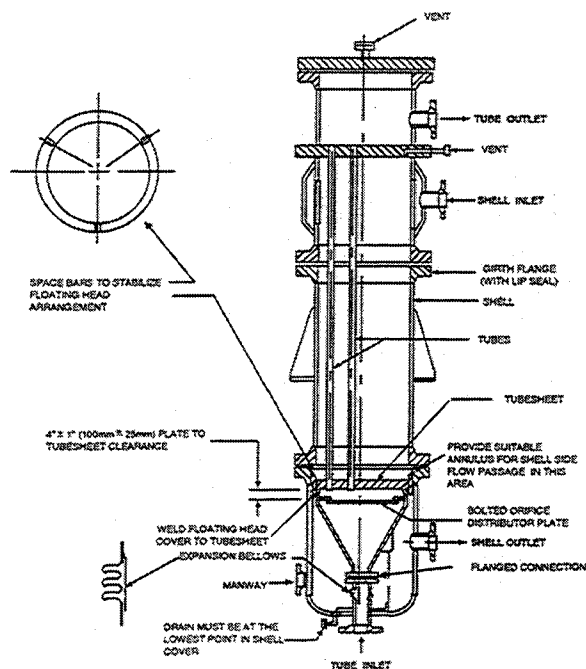
Vertical Combined Feed-Effluent Exchangers (VCFE's)

- High temperature processes need large feed heating duty
- Efficiency optimization requires that as much heat as possible be recovered from products rather than fired heater heat input

Example: Platforming



EXC-R00-38
IPD-2004/HE-63

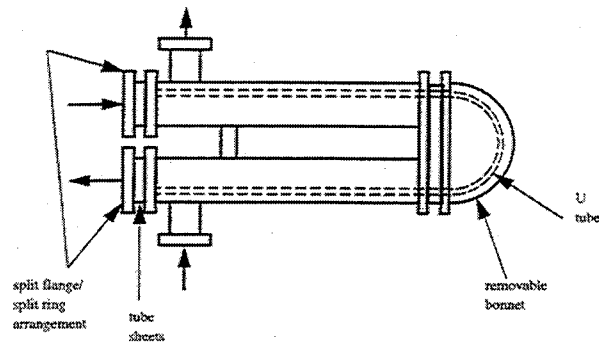


Vertical Combined Feed-Effluent Exchangers (continued)

EXC-R00-37
IPD-2004/HE-64

Hairpin Exchangers

- When small duties are required, hairpin exchangers are specified:
 - Cheaper than very small shell and tube
 - Highly effective (1-pass and/or true countercurrent)



This design is used for double-pipe and multi-tube exchangers.

EXC-R00-39
IPD-2004/HE-65

Special Tubes

- Special tube shapes and surfaces can improve heat transfer performance

BOILING

- Heat transfer is affected by the rate of bubble formation
- Rate of bubble formation affected by:
 - Fluid properties, system pressure
 - $\Delta T = (T_{\text{hot}} - T_{\text{boil}})$
 - Properties of tube surface

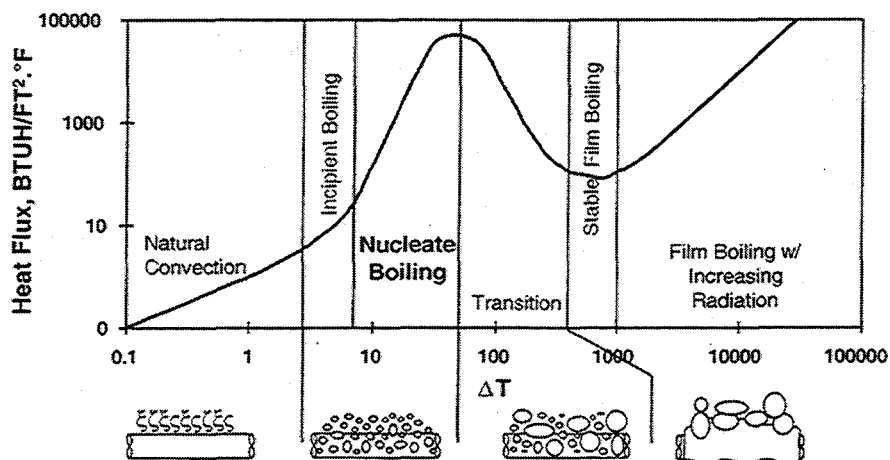
IPD-2004/HE-66

Special Tubes (continued)

- If you cannot change the process pressure or heating medium temperature, then change tube surface
- Specially treated tube surfaces can improve boiling heat transfer by 300%
- Roughen tube surface on boiling side
- This improves bubble formation rate which improves boiling

IPD-2004/HE-67

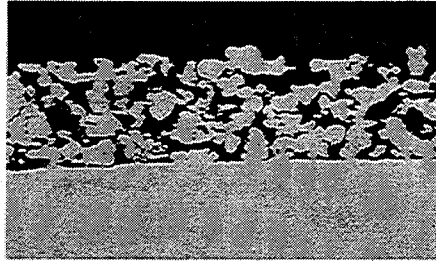
Typical Pool Boiling Curve



IPD-2004/HE-68

Special Tubes (continued)

- Porous Boiling Surface
- Strong Metallurgical Bond
- Coating Thickness 0.005-0.015 inch
- Interconnecting Channels or "Re-entrant Sites"
- Boiling Coefficient 10 Times Bare Tube



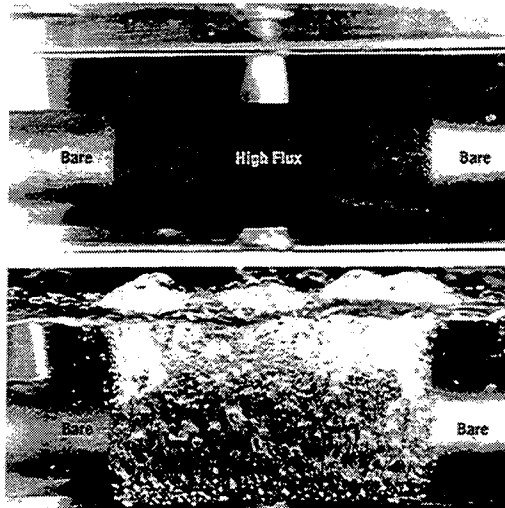
HIGH FLUX
Coating

Tubular
Substrate

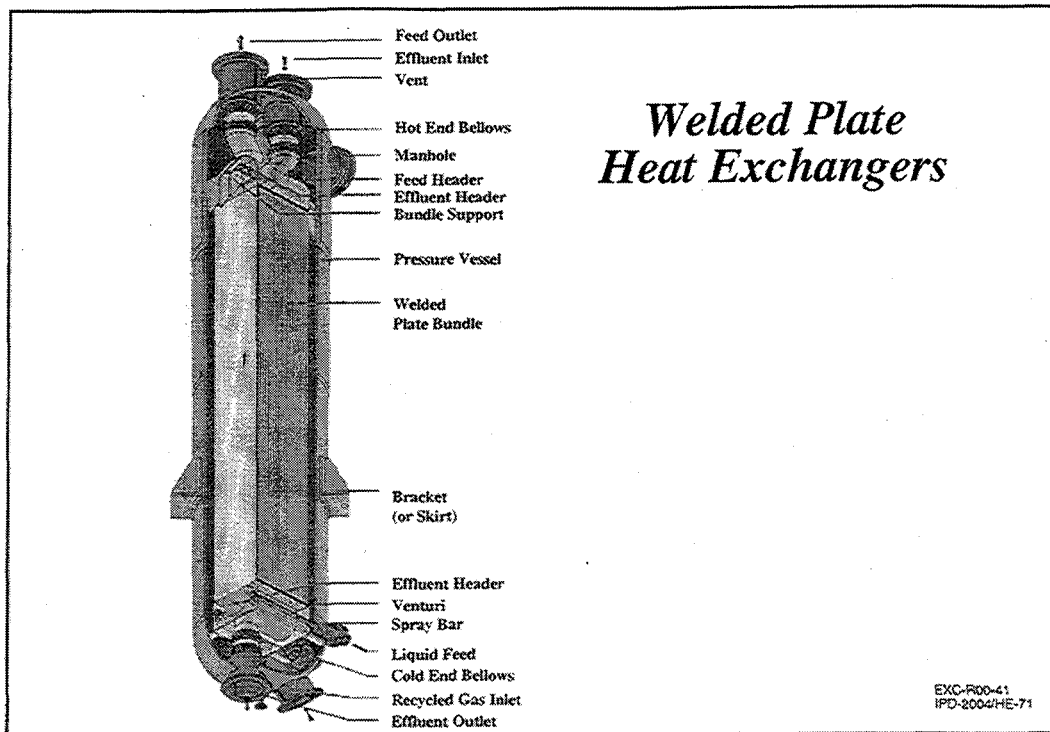
IPD-2004/HE-69

Special Tubes (continued)

UOP High Flux Tubes



IPD-2004/HE-70
EXC-R00-39



Fluid Changing Phase

- Boiling Heat Transfer
- Types of Boiling
- Pool Boiling
- Flow Boiling
- Flow Regimes
- Condensation
- Total Condensers
- Partial Condensers
- Operational Considerations
- Typical Overall U Values

Boiling Heat Transfer

Need to understand boiling:

- Design, Revamp, Troubleshooting of
 - Reboilers
 - Vaporizers
 - 2-phase combined feed-effluent exchangers (Platforming, Pacol, hydrotreaters, etc.)

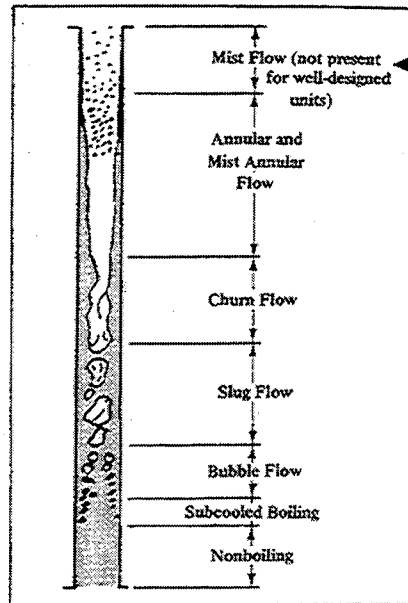
IPD-2004/HE-73

Types of Boiling

- POOL boiling
 - Agitation caused ONLY by bubbles and natural convection
 - Fluid velocity slow or negligible (e.g. kettle reboiler)
- FLOW boiling
 - Agitation is caused by flow of fluid
 - Fluid velocity large (vertical thermosiphon, CFE, VCFE)

IPD-2004/HE-74

Flow Regimes

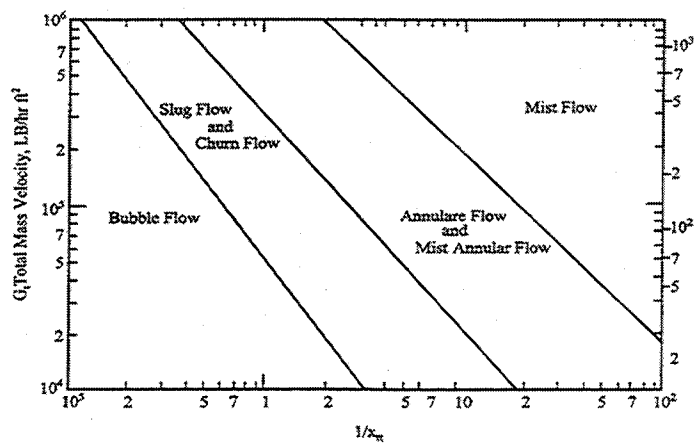


IF % vaporization is flexible

EXC-R00-43
IPD-2004/HE-75

Flow Regimes (continued)

For vertical upflow only



(reference: Fair J.R., Petr. Refiner, vol 39 (2) p105. 1960)

EXC-R00-44
IPD-2004/HE-76

Flow Regimes (continued)

$$X_{tt} = \left(\frac{1-y}{y} \right)^{0.9} \left(\frac{\rho_{vap}}{\rho_{liq}} \right)^{0.5} \left(\frac{\mu_{liq}}{\mu_{vap}} \right)^{0.1}$$

y is weight fraction of vapor

X_{tt} is the "Martinelli Parameter"

larger	y
larger	ρ_{liq}
smaller	ρ_{vap}
larger	μ_{vap}
smaller	μ_{liq}
give larger	$1/X_{tt}$

IPD-2004/HE-77

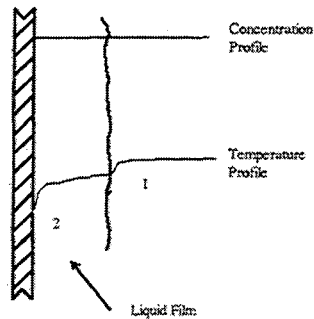
Condensation

- Boiling fluid is agitated by bubble movement
 - Gives very large film coefficients ($\gg 700\text{-}1000 \text{ W/m}^2/^{\circ}\text{C}$)
($\gg 600\text{-}860 \text{ kcal/hr/m}^2/^{\circ}\text{C}$) ($\gg 2300\text{-}3400 \text{ Btu/hr/ft}^2/\text{F}$)
- There is no such mechanism to aid condensation
- Condensers rely on
 - Turbulence induced by flow
 - Diffusion (important for partial condensers)
- Two separate types
 - Total Condensers
 - Partial Condensers

IPD-2004/HE-78

Total Condensers

Generally have higher coefficients, no diffusion involved:

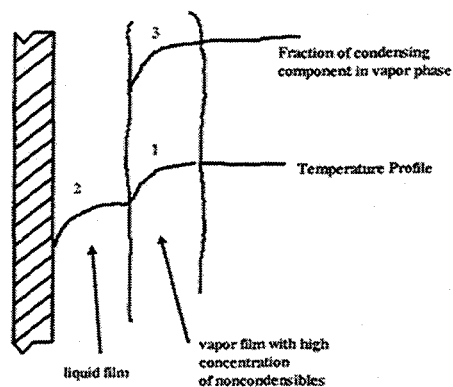


Film coefficient consists of:

- 1 - bulk vapor - liquid layer resistance
- 2 - liquid layer resistance

EXC-R00-45
IPD-2004/HE-79

Partial Condensers



Film condensing coefficient consists of:

- 1 - vapor - liquid layer thermal resistance
- 2 - liquid layer thermal resistance
- 3 - gas-film thermal and diffusion resistances

EXC-R00-46
IPD-2004/HE-80

Partial Condensers (continued)

- Condensing molecules (e.g. naphtha) have to diffuse through noncondensables (e.g. hydrogen)
- Diffusion resistance **DECREASES** coefficient
 - ⇒ Big difference to size of exchanger
- Calculation of condensing coefficients:
 - Complex, time consuming
 - Empirical
 - Use computerized methods
- Correlations are programmed in:
 - HTRI (proprietary correlations - general + private data)
 - HTFS
 - B-JAC

IPD-2004/HE-81

Operational Considerations

- Even “total condensers” (e.g. naphtha splitter condenser) get small amounts of noncondensables.
- Noncondensables (hydrogen, inerts) can build up in the exchanger
 - ⇒ Blanket upper portion of bundle with noncondensables
 - ⇒ Retard diffusion of condensable components to the cold tube surface
 - ⇒ Impair performance of exchanger
- Total condensers must have a vent for noncondensables,
 - At the TOP
 - At OUTLET end of exchanger

IPD-2004/HE-82

Typical Condenser Overall U Values

(fouled, conservative values)

C ₃ -C ₄ water-cooled total condensers	U ≈ 560 W/m ² /C ≈ 480 kcal/hr/m ² /C ≈ 100 Btu/hr/ft ² /F
Kero/naphtha splitter water-cooled total condensers	U ≈ 480 W/m ² /C ≈ 410 kcal/hr/m ² /C ≈ 85 Btu/hr/ft ² /F
Water cooled steam surface condenser	U ≈ 850 W/m ² /C ≈ 730 kcal/hr/m ² /C ≈ 150 Btu/hr/ft ² /F
FCC main column overhead condenser (partial)	U ≈ 420 W/m ² /C ≈ 360 kcal/hr/m ² /C ≈ 75 Btu/hr/ft ² /F

IPD-2004/HE-83

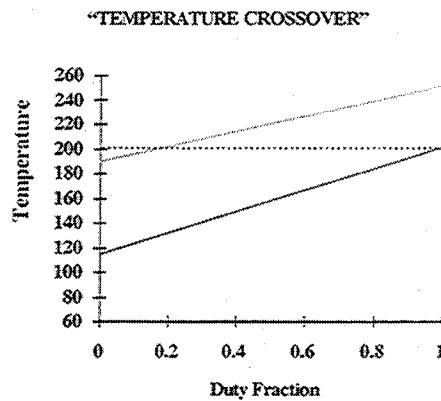
Geometric Influences

- | | |
|---------------------------------|----------------------------|
| ■ Temperature Cross | ■ Other Correction Factors |
| ■ Multiple Shells in Series | ■ Tubes |
| ■ Weighted Log Mean Temperature | ■ Tube Pitch |
| | ■ Baffles |
| | ■ Support Plates |

IPD-2004/HE-84

Temperature Cross

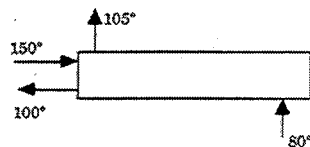
If hot outlet $T < \text{cold outlet } T \Rightarrow$ called "temperature cross"



EXC-R00-60
IPD-2004/HE-85

Temperature Cross (continued)

- If the cross ($T_{\text{cold outlet}} - T_{\text{hot outlet}}$) is GREATER 5-10% T_{lm} , then more than one exchanger is required.
- Guideline: if $F < 0.8$ consider more exchangers.
- Generally $T_{\text{h,out}} > t_{\text{c,out}} \Rightarrow F \leq 0.8$

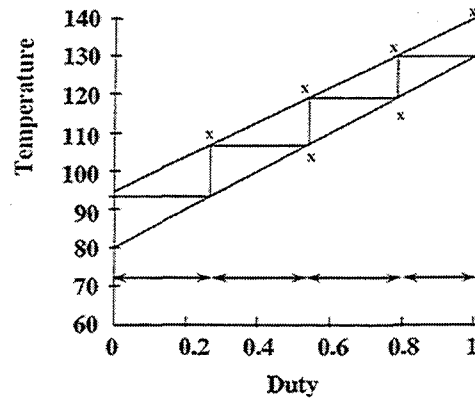


- Calculate F . Is one exchanger enough?
- If more than one exchanger, what is overall F , and how is it calculated?

EXC-R00-61
IPD-2004/HE-86

Multiple Shells in Series

(e.g. Hydrotreater Combined Feed Effluent Exchanger)



x marks inlet and outlet temperatures of each exchanger.

Note: Duties not necessarily equal

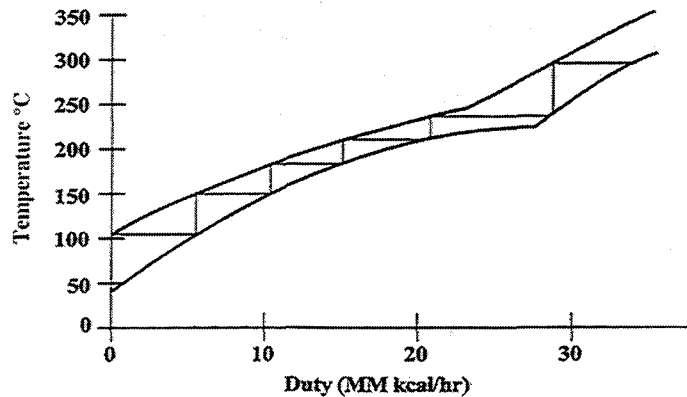
ΔT_{lm} not necessarily equal

F not necessarily equal.

EXC-R00-83
IPD-2004/HE-87

Multiple Shells in Series (continued)

What about realistic heating curves (Hydrotreater CFE)



This needs 6 exchangers in series. For intermediate temperatures shown, are all exchangers the same size?

EXC-R00-84
IPD-2004/HE-88

Tubes

■ Diameter

- Tubes range from 1/4" to 2" OD generally.
- Refinery exchangers use: 5/8", 3/4", 1", 1-3/4" commonly
- General: smaller D $\Rightarrow$ better heat transfer but higher ΔP and harder to clean

■ Wall Thickness

- Commonly ranges from 0.065" to 0.109" (1.65 mm to 2.77 mm)
- FCC Flue Gas Coolers (firedtube) often uses 1.75" OD x 0.2" thk (44.45 mm OD x 5.1 mm thick)

IPD-2004/HE-89

Tubes (continued)

BWG	Thickness (in.)	Thickness (mm)
8	0.165	4.19
10	0.134	3.40
12	0.109	2.77
14	0.083	2.11
16	0.065	1.65
18	0.049	1.24
20	0.035	0.89

- BWG (Birmingham Wire Gage) is a tubewall thickness standard
- HIGHER BWG $\Rightarrow$ THINNER tubewall
- Wall thickness specified as MINIMUM or AVERAGE
- For example: For an FCC Flue Gas Cooler: 1.75" OD x 0.2" min. wall thick.

IPD-2004/HE-90

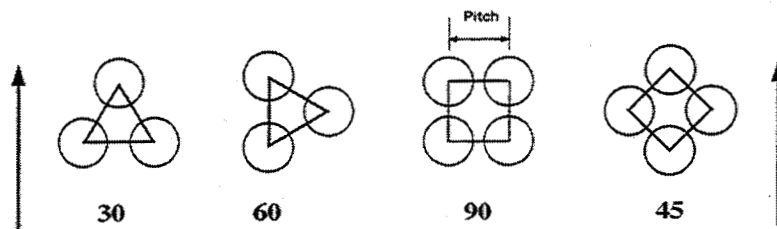
Tubes (continued)

Tube Length

- Tube length varies with suppliers
- TEMA suggests, 8, 10, 12, 16, 20 foot tubes are common
- 16 foot (4.9 m) and 20 foot (6.1 m) are probably the most common
- VCFE's use 30, 45, 52, 60, 65, 70 ft tubes
- A U-tube exchanger uses many different tube lengths

IPD-2004/HE-91

Tube Pitch



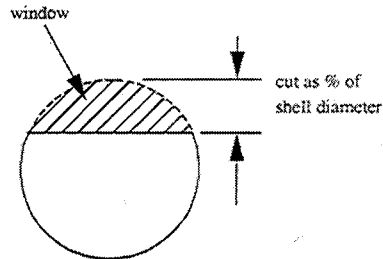
Triangular (2 kinds) and square (2 kinds)

FEATURE	USE PATTERN:
lower ΔP on shellside	square (effective only at low Re number)
shellside fouling	square - easier cleaning
horizontal shellside boiling	square - prevent vapor blanketing
decrease shell size	fit 15% more tubes if triangular pitch used

EXC-R00-88
IPD-2004/HE-92

Baffles

Single Segmental Baffle Cut:



Selecting baffle cut:

- Larger cut $\Rightarrow$ lower shellside ΔP**
 $\Rightarrow$ smaller cross flow worse heat transfer
 $\Rightarrow$ larger shell and tubesheet diameter
 $\Rightarrow$ more expensive exchanger

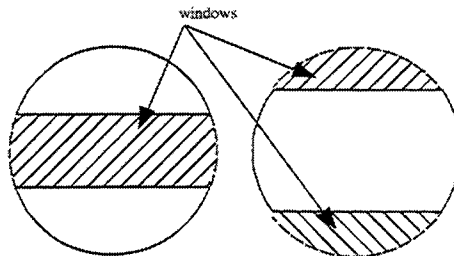
EXC-R00-69
IPD-2004/HE-93

Baffles (continued)

- Smaller cut $\Rightarrow$ higher shellside ΔP**
 $\Rightarrow$ higher acceleration loss through window

Select baffle cut so that ΔP all windows $\approx 20\text{-}25\%$ ΔP total

Double segmental baffles:

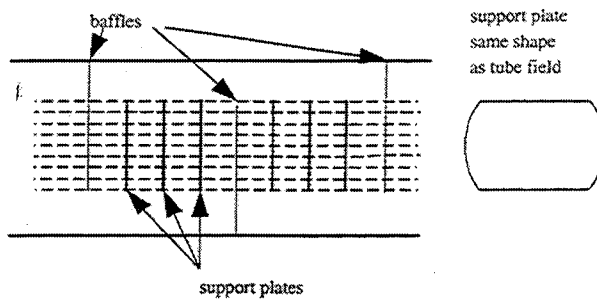


Center window area $\approx$ combined edge window area

EXC-R00-70
IPD-2004/HE-94

Support Plates

- ΔP limitations on shellside force large baffle spacing
- Decrease tube vibration add support plates between baffles (usually NTIW):



- Support plates hold tubes together in a "cage"
- Vibration is less likely, almost no ΔP cost
- ANOTHER KIND of support plate: at the END of U-tube bundles. A full diameter support plate is used. WHY?

EXC-R00-72
IPD-2004/HE-96

Design Practices

- Design
- Information Requirements
- Fouling Factors
- Estimated Heat Transfer Coefficients
- Vibration

IPD-2004/HE-96

Design

- Design is more than just calculations
 - Judgment and understanding
 - Flexibility
 - Experience
 - Understanding of user's wants/needs
 - Current AND future operation
 - Fouling factor requirements
 - Allowable ΔP
 - Ranges of normal operation
 - Maximum weight and size limitation (cranes, etc.)
 - Preferred geometries
 - TEMA type
 - Tube length, diameter, wall thickness
 - Plot-space limitations

IPD-2004/HE-97

Information Required to Design a Heat Exchanger

- Process Specification
 - All temperatures and pressures
 - ΔP allowable for both streams
 - Dew points, bubble points
 - Heating/cooling curves
 - Physical properties
 - Required fouling factors
- Limit Oversizing

IPD-2004/HE-98

Fouling Factors

Fouling Resistances for Oil Refinery Streams

Crude and Vacuum Unit Gases and Vapors:						
Atmospheric Tower Overhead Vapors	0.001					
Light Naphthas	0.001					
Vacuum Overhead Vapors	0.002					
Crude and Vacuum Unit Liquids:						
Crude Oil						
	0 to 250 °F Velocity (ft/sec)			250 to 350 °F Velocity (ft/sec)		
	< 2	2-4	> 4	< 2	2-4	> 4
Dry	0.003	0.002	0.002	0.003	0.002	0.002
Salt *	0.003	0.002	0.002	0.005	0.004	0.004
	350 to 450 °F Velocity (ft/sec)			450 °F and higher Velocity (ft/sec)		
	< 2	2-4	> 4	< 2	2-4	> 4
Dry	0.004	0.003	0.003	0.005	0.004	0.004
Salt *	0.006	0.005	0.005	0.007	0.006	0.006
* Assumes desalting at approximately 250 °F.						
Gasoline	0.002					
Naphtha	0.002 - 0.003					
Light Distillates	0.002 - 0.003					
Kerosene	0.002 - 0.003					
Light Gas Oil	0.002 - 0.003					
Heavy Gas Oil	0.003 - 0.005					
Heavy Fuel Oils	0.005 - 0.007					
Asphalts and Residuum:						
Atmospheric Tower Bottoms	0.007					
Vacuum Tower Bottoms	0.010					

Units for fouling factors:
hr ft² °F/Btu

IPD-2004/HE-99

Approximate Heat Transfer Coefficients

Liquids	Shell Side Cooling	Shell Side Heating	Tube Side Cooling	Tube Side Heating
Oils, 20 °API				
200 °F Average Temperature	40	50	15	25
300 °F Average Temperature	70	85	20	35
400 °F Average Temperature	80	100	65	75
Oils, 30 °API				
150 °F Average Temperature	70	85	20	35
200 °F Average Temperature	80	100	50	60
300 °F Average Temperature	110	130	95	115
400 °F Average Temperature	130	155	120	140
Oils, 40 °API				
150 °F Average Temperature	80	100	50	60
200 °F Average Temperature	120	140	115	135
300 °F Average Temperature	150	170	140	160
400 °F Average Temperature	180	200	175	195
Heavy Oils, 8-14 °API				
300 °F Average Temperature	20	30	10	20
400 °F Average Temperature	40	50	20	30
Diesel Oil	115	130	95	115
Kerosene	145	155	140	150
Heavy Naphtha	145	155	130	140
Light Naphtha	180	190	180	180
Gasoline	200	200	200	200
Light Hydrocarbons	250	250	250	250
Alcohols, Most Organic Solvents	200	200	200	200
Brine, 75% water	500	500	500	500

Note: Coefficients are based on 3/4 inch diameter tubes. For Tube side flows correct By multiplying by 0.75/Actual OD.
Estimated accuracy is 25%. Units Btu/hr-ft²-°F.

IPD-2004/HE-100

Vibration

- Fluid flow in heat exchangers can cause tubes to vibrate.
- This phenomenon is highly complex to the point that it is difficult to predict the potential for vibration. Damage to tubes can occur for certain shell side flow conditions relative to baffle configuration and unsupported tube spans.

IPD-2004/HE-101

Vibration ***(continued)***

- Mechanical failure of tubes may result from flow induced vibration. Damage can result from any combination of the following conditions:
 - Collision damage
 - Baffle damage
 - Tubesheet clamping effect
 - Acoustic vibration

IPD-2004/HE-102

Vibration (continued)

■ Failure Regions

- Vibration can occur in nearly any region of the heat exchanger
- Locations with highly flexible spans or high shell side velocities are of primary concern for failure

IPD-2004/HE-103

Troubleshooting Problems

■ Failures and Troubleshooting

■ Sample Problem

IPD-2004/HE-104

Failures and Troubleshooting

■ THE KEY TO EXCHANGER PERFORMANCE MONITORING

- TAKE MANY MEASUREMENTS
 - WATCH TRENDS
 - THINK ABOUT WHAT YOU SEE
- Who has panel instruments accurate to 1%?
- Who wants to do single-gauge surveys every week?

IPD-2004/HE-105

Failures and Troubleshooting (continued)

The \$1,000,000 answer is:

1) When any (large or expensive) exchanger is new:

- Do single-gauge pressure and temperature survey
- Personally supervise sampling of process stream(s)
- Do best possible job of energy and material balance
- Perform detailed rating of exchanger
- Keep records on file

IPD-2004/HE-106

Failures and Troubleshooting ***(continued)***

- 2) Track ΔP and hot or cold end approach temperatures of exchanger on DCS trending, or graph paper if you have to.
- Mark major changes in flow rate, feed temperatures, or feedstock on graph. Adjust measurements for changes.
 - Any gradual trends not explained by changes above mean that something is steadily getting worse.
 - If the cause is not clear, get new single-gauge survey, double check results.

IPD-2004/HE-107

Failures and Troubleshooting ***(continued)***

■ Examples

- A downward trend in hot approach indicates fouling progression
- An upward trend in ΔP - fouling
- Drastic changes in approach temperatures or pressures can indicate leaks or blockages

IPD-2004/HE-108

Failures and Troubleshooting (continued)

PARALLEL EXCHANGERS

- For exchangers in parallel, use the above procedure but also watch trends in the differences
- Example -- watch trends in:
 - Hot end approach (A) - Hot end approach (B),
or $T_{h,out}(A) - T_{h,out}(B)$
- If differences change at all, one exchanger has a problem
 - Both exchangers should foul at the same rate
 - Performance of both exchangers should change equally under all circumstances

IPD-2004/HE-109

Failures and Troubleshooting (continued)

SUMMARY

- If exchangers are regularly monitored in this way
 - You will not get many unpleasant surprises
 - You will have information for follow-up investigations

IPD-2004/HE-110

Sample Problem

Reactor (Gas) Effluent Coolers

Calculate the design and operating U values from real plant and design data:

Design

Q	MW	4.76
G	kg/hr	70056
T _{in}	°C	147
T _{out}	°C	41
t _{in}	°C	32
t _{out}	°C	40
ΔT _{lm}	°C	?
F	-	?
UA	W/°C	?
A	m ²	480
U	W/m ² /°C	?

Operating

Q	MW	5.27
G	kg/hr	73899
T _{in}	°C	172
T _{out}	°C	65
t _{in}	°C	27.2
t _{out}	°C	35
ΔT _{lm}	°C	?
F	-	?
UA	W/°C	?
A	m ²	480
U	W/m ² /°C	?
U/U _{design}		?

IPD-2004/HE-111

Sample Problem (continued)

Solutions

Design

Q	MW	4.76
G	kg/hr	70056
T _{in}	°C	147
T _{out}	°C	41
t _{in}	°C	32
t _{out}	°C	40
ΔT _{lm}	°C	39.6
F	-	0.845
UA = Q/(FΔT _{lm})	W/°C	142250
A	m ²	480
U	W/m ² /°C	296

IPD-2004/HE-112

Sample Problem (continued)

Solutions

Operating		
Q	MW	5.27
G	kg/hr	73899
T _{in}	°C	172
T _{out}	°C	65
t _{in}	°C	27.2
t _{out}	°C	35
ΔT _{lm}	°C	77
F	-	0.97
UA = Q/(FΔT _{lm})	W/°C	70558
A	m ²	480
U	W/m ² °C	147
U/U _{design}		0.49

U is ½ of design value even though shell and tube flows are higher than design! The problem is severe fouling.

IPD-2004/HE-113

Cleaning

- Chemical or Mechanical Methods depend on service
- Chemical Methods
 - Circulate hot wash oil or light distillate at high velocity – removes sludge
 - Circulate hot fresh water – removes salts
 - Circulate chemical compounds – removes almost anything (caustic for organics, acids for scales)
- Mechanical Methods
 - High pressure water jet – many deposits
 - Scrapers, rotating wire brushes – hard scale, coke
 - Sliding Brushes/Balls (on-line)

IPD-2004/HE-114

Cleaning (continued)

■ **Precautions**

- Verify chemicals effectiveness/inhibition prior to cleaning
- Monitor cleaning carefully
- Chemical/mechanical methods may also remove exchanger metal

IPD-2004/HE-115

Air Cooled Heat Exchangers

IPD-2004/HE-116

Why Use Air-Cooled Exchangers?

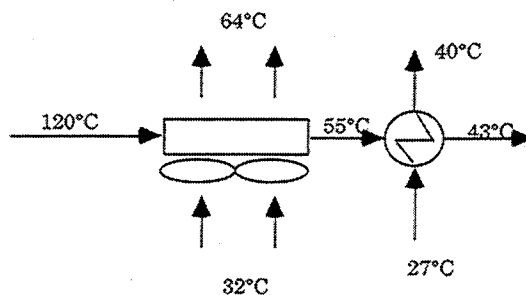
- High process temperatures in water-cooled exchangers cause:
 - Overheating of cooling water at tube wall - mineral scaling
 - High differential temperatures - mechanical problems

CHARACTERISTICS OF WATER COOLING VS. AIR COOLING	
COOLING WATER	AIR COOLING
Utility piping and system maintenance	No utility piping or maintenance
Cooling tower (or expensive metallurgy for seawater)	Cool air is free
Pumps	Only fans
Capacity is limited (piping, cooling tower, pumps)	Unlimited capacity (can add extra coolers easily)
Small plot space	Large plot space
Chemical treatment required	Overcooling

IPD-2004/HE-117

Split of Cooling Range With Water Cooling

Example: Reactor effluent from combined feed-effluent exchanger is typically at about 120°C (248°F). (Hydrotreater, Platformer, etc.)



This arrangement

- Decreases cooling water usage
- Prevents excessive fouling on tube side of water-cooled exchanger

EXC-R00-75
IPD-2004/HE-118

Heat Transfer

Basic equation based on finned area:

$$\frac{1}{U} = \left(\frac{1}{h_{\text{process}} \frac{A_b}{A_f}} + \frac{1}{h_{\text{air}}} + r_w + r_{\text{di}} \frac{A_f}{A_b} + r_{\text{do}} \right)$$

- Where:
- h_{process} = tube-side film coefficient
 - h_{air} = air-side film coefficient
 - r_w = wall and fin metal resistance
 - r_{di} = inside fouling resistance
 - r_{do} = outside fouling resistance
 - A_f = finned area
 - A_b = bare tube area
 - h_{process} = between 100 (gas/low ΔP mixed phase) and 500 (liquid or mixed-phase or condensation).
 - h_{air} = 6 - 10 Btu/(hr-ft² °F)

IPD-2004/HE-119

Heat Transfer (continued)

Why low air-side film coefficient?

- Because
 - Air Properties
 - Low thermal conductivity
 - Low density
- Plus
 - Fans Sizing
 - Limits in pressure drop (air velocity)
- Air-side resistance usually larger than process-side resistance
- Therefore: finned area: bare tube area $\Rightarrow$ 18-20:1

IPD-2004/HE-120

Sample Problem

Calculate U for a range of h_{process} :

■ We will use

$$H_{\text{air}} = 7.0 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$$

$$R_w = 0.005 \text{ (hr}\cdot\text{ft}^2\cdot^\circ\text{F)/Btu}$$

$$R_{\text{di}} = 0.001 \quad "$$

$$R_{\text{do}} = 0.001 \quad "$$

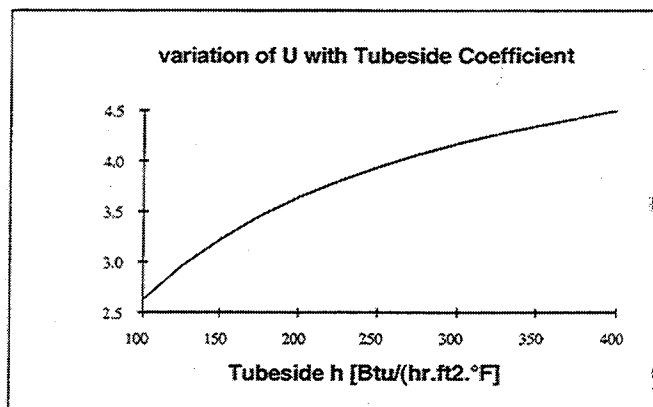
■ E.G. $H_{\text{process}} = 100$

$$\frac{1}{U} = \left(\frac{1}{100 \left(\frac{1}{20} \right)} + \frac{1}{7} + 0.005 + 0.001 \left(\frac{20}{1} \right) + 0.001 \right) =$$

gives $U = 2.6 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$

IPD-2004/HE-121

Typical Values



U ranges between 2.5 to 4.5

EXC-R00-76
IPD-2004/HE-122

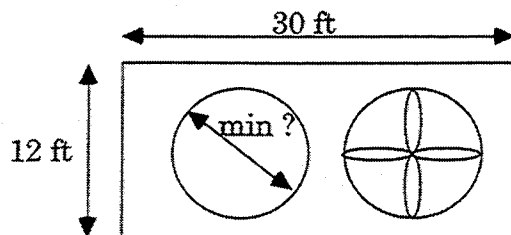
Typical Values

TYPE	U [Btu/(hr·ft ² ·°F)]
Condenser	3.8 (column overhead) 3.0-3.5 (NHT, Platformer product condensers) 6.0 (steam)
Liquid Cooler	2.6 (HVGO) 3.0 (naphtha, kero, etc.) 3.5 (light aromatics) 4.0 (C ₃ -C ₅) 5.0 (water)
Gas Cooler	3.0 - 3.5

IPD-2004/HE-123

Air-Side Design

- Fan limitations
 - Low developed head
 - ⇒ Low air-side allowable ΔP (0.6" water)
 - Air velocities (based on projected bundle area) 550 ft/min (9.2 ft/sec)
- Fan area: Projected bundle area
 - At least 40% - proper air distribution



EXC-R00-77
IPD-2004/HE-124

Air-Side Design (continued)

- Projected bundle area = (width • height)
= 12 ft • 30 ft = 360 ft²
- Total area of fan-circle = (bundle area × 0.40)
= 360 ft² × 0.40
= 144 ft² (minimum)
- Area per fan = 72 ft² (min) ⇒ 2 fans assumed
- Fan diameter = $\left[\frac{(\text{area per fan})(4)}{\pi} \right]^{\frac{1}{2}}$

= $\left[\frac{(72 \text{ ft}^2)(4)}{\pi} \right]^{\frac{1}{2}} = 10 \text{ ft diameter (min)}$

IPD-2004/HE-125

Air-Side Design (continued)

- Fan Power
Depends on
 - Required air-flow, air density
 - Air-side pressure drop
 - Fan efficiency (55 - 70%)

$$\text{Fan Power} = \left[\frac{(\text{ACFM})(\Delta P_{\text{air}})}{(6387)(\eta)} \right]$$

Where: fan power - horsepower
 ACFM - actual ft³/min of air
 ΔP_{air} - in inches of water
 η - fan efficiency 55 - 70%
 6387 - conversion factor

IPD-2004/HE-126

Hydraulics

- Pressure drop calculations - same as for piping, shell and tube exchangers

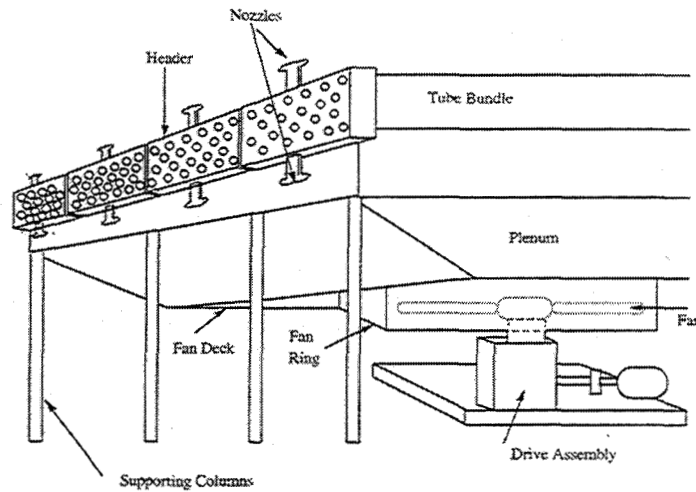
IPD-2004/HE-127

Mechanical Considerations

- Typical Arrangement
- Header Types
- Standards
- Materials of Construction
- Finned Tubing Types
- Fan Arrangements
- Fans
- Driver Types

IPD-2004/HE-128

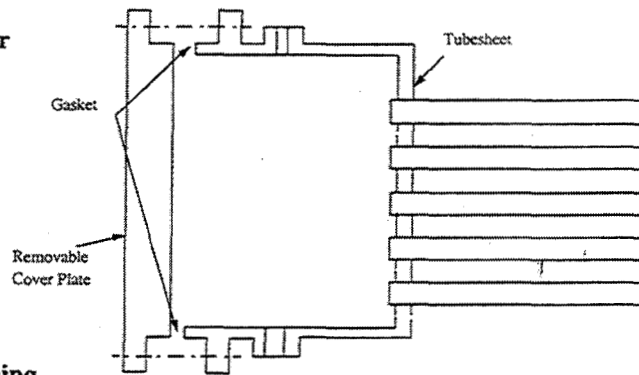
Typical Arrangement



EXC-R00-78
IPD-2004/HE-129

Header Types

Removable cover



■ Advantages

- Easy cleaning
- Easy access to tubes and tubesheet (easy inspection and repair)

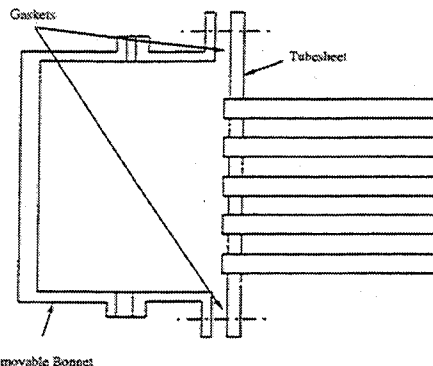
■ Disadvantages

- Gasket requirements
- Mechanical design of long flanges is difficult for high pressure and temperature (>300 psig and $>350^{\circ}\text{F}$)
- Header and tubesheet all one piece (repairs difficult)

EXC-R00-79
IPD-2004/HE-130

Header Types (continued)

Removable bonnet

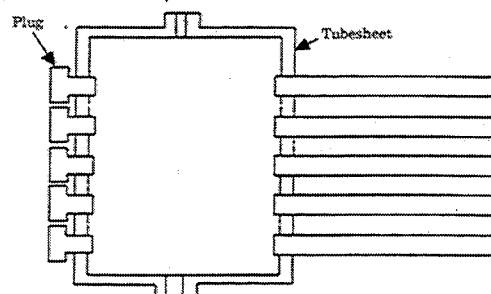


- **Advantages**
 - Good cleaning of header and tubesheet
 - Header only or tubesheet only can be replaced
- **Disadvantages**
 - Removal of piping necessary for tubesheet inspection
 - Gasket requirements
 - Mechanical design of bonnet is difficult for high pressure and high temperature (>300 psig and >350°F)

EXC-R00-80
IPD-2004/HE-131

Header Types (continued)

Plug headers



- **Advantages**
 - Easy cleaning of tubes (brush, hydroblasting)
 - No gaskets
 - No long-flange design
 - Fewer bolts (many plugs)
 - Good for high pressure design (up to 3000 psi)
 - Good for hydrogen service
- **Disadvantages**
 - headers have to be chemically cleaned (water, steam, light hydrocarbon)

EXC-R00-81
IPD-2004/HE-132

Header Types ***(continued)***

Examples

- #1 Hydrotreater products condenser - ?
 - H₂ service
 - High pressure - 850 psig
- #2 Bottoms cooler - ?
 - Low pressure - 160 psig
 - Low temperature - 330°F

IPD-2004/HE-133

Header Types ***(continued)***

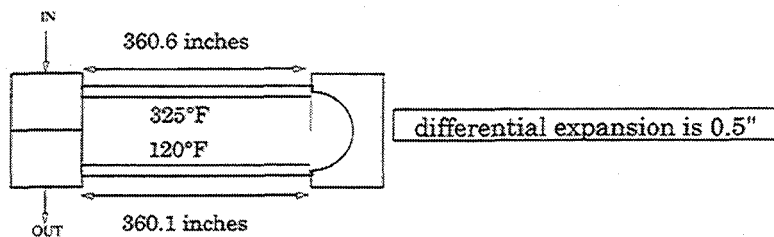
Examples

- #1 Hydrotreater products condenser - Plug
 - H₂ service
 - High pressure - 850 psig
- #2 Bottoms cooler - Removable cover or bonnet
 - Low pressure - 160 psig
 - Low temperature - 330°F

IPD-2004/HE-134

Header Types (continued)

- For large temperature differences $> 200^{\circ}\text{F}$, one must deal with differential expansion of top and bottom tubes
- Example
 - First pass at 325°F , second pass at 120°F , Carbon Steel, 30 ft cold length (360 inches at 70°F) α is about $6.3 \times 10^{-6} \text{ in}/^{\circ}\text{F}$ for carbon steel (is the thermal expansion coefficient)



- Separate inlet and outlet headers

EXC-R00-82
IPD-2004/HE-135

Standards

- API 661 Air-Cooled Exchangers For General Refinery Service
- Winterization
- Noise Measurement
- ISO 13706 Petroleum and Natural Gas Industries
 - Air-Cooled Heat Exchangers

IPD-2004/HE-136

Materials of Construction

- Tubes and Headers (Set by Process Conditions)
 - Carbon steel
 - Killed carbon steel (H_2 , wet H_2S service)
 - Chrome Moly steels
 - Stainless steel
- Fins
 - Aluminum
- Plenums, Fan Rings and Supports
 - Structural carbon steel (often galvanized)

IPD-2004/HE-137

Finned Tubing Types

- Process temperature constrictions are dependent
 - Fin material
 - Fin construction - thermal contact of fin to tube
- Problems
 - Separation of fin and tube
 - Thermal expansion
 - Poor heat transfer

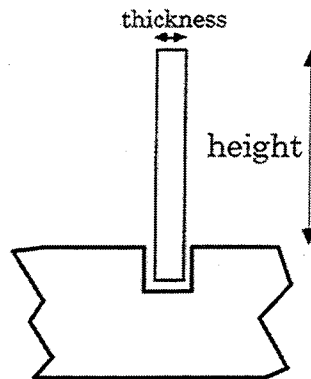
IPD-2004/HE-138

Finned Tubing Types (continued)

Embedded Fins (< 750°F process temp)

■ Description

- Helical groove is cut into the outside of the tube wall.
- Metal is displaced on each side of the groove.
- Fins are wound into groove.
- Displaced metal forced against each side of the fin.



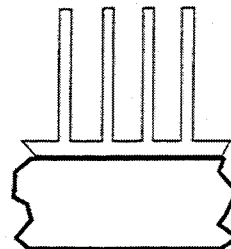
EXC-R00-84
IPD-2004/HE-139

Finned Tubing Types (continued)

Extruded (< 550°F process temp)

■ Description

- Fins are formed by extrusion from an aluminum outlet tube
- Mechanically bonded to the tube



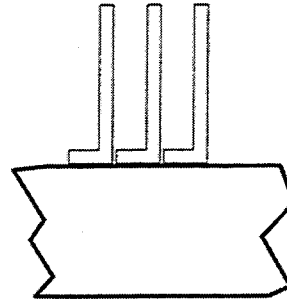
EXC-R00-85
IPD-2004/HE-140

Finned Tubing Types (continued)

Footed (L-foot) (< 350°F process temp)

■ **Description:**

- Footed fins are tension wrapped around tube
- Each fin butts up against the adjacent fin to cover the complete tube
- Shields tube against atmospheric corrosion



Ⓒ

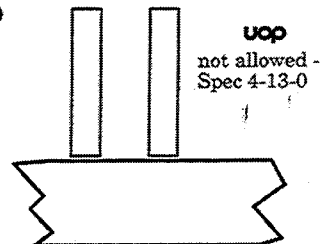
EXC-R00-86
IPD-2004/HE-141

Finned Tubing Types (continued)

Tension wrapped (<250°F process temp)

■ **Description**

- Fin is tension wound around tube.



Ⓒ

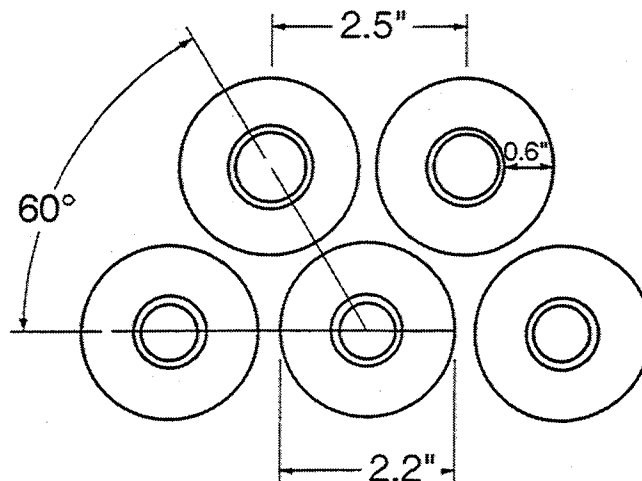
EXC-R00-87
IPD-2004/HE-142

Finned Tubing Types (continued)

- | | |
|--|---|
| <ul style="list-style-type: none"> ■ Cost scale from least expensive to most expensive <ul style="list-style-type: none"> - Tension wrapped - Footed - Embedded - Extruded | <ul style="list-style-type: none"> ■ Typical Tube Sizing <ul style="list-style-type: none"> - Tubes <ul style="list-style-type: none"> — 1" OD tubes — 12 BWG (0.109" thick) — 2-1/2" triangular pitch - Fins <ul style="list-style-type: none"> — Height 0.6" — Thickness 0.12", usually aluminum — About 9 fins per inch — Finned area: bare tube area 18-20 : 1 |
|--|---|

IPD-2004/HE-143

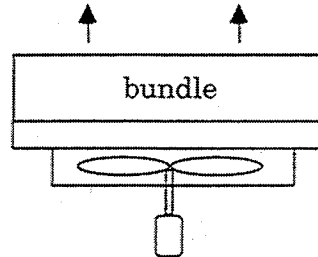
Finned Tubing Types (continued)



EXC-R00-88
IPD-2004/HE-144

Fan Arrangements and Types

- **Forced Draft** - air is forced or pushed across tubes. Fan is placed below the bundle.

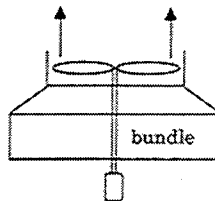


ADVANTAGES	DISADVANTAGES
Lower horsepower (cool air)	Air distribution not as good as induced.
Better accessibility of motor and fan	Hot air velocity away from bundle is small.
Easily adaptable to recirculation for winterization	Possibility of recirculation of hot air.
	Low natural draft capability, unless skirt is specified.

EXC-R00-89
IPD-2004/HE-145

Fan Arrangements and Types (continued)

- **Induced Draft** - air is pulled across tubes. Fan is placed above the bundle.



ADVANTAGES	DISADVANTAGES
Good air distribution across bundle	Higher fan-power required (air is hot)
Hot air is blown far away from bundle	Air outlet temperatures must be limited (prevent fan damage)
Bundle is protected from above by plenum and fan	Fan maintenance - damage to bundle
Has 'stack' to improve natural draft (fan failure)	Fan maintenance difficult (hot air, accessibility)

IPD-2004/HE-146
EXC-R00-90

Fan Types

- **Fixed-Pitch Fans (2 types)**
 - Blade pitch is permanent
 - Blade pitch can be adjusted manually when fan is stopped
- **Variable Pitch Blades**
 - Automatic or operator controlled, can vary fan capacity during operation AV fans": AutoVariable pitch
 - Operational flexibility greater than fixed-pitch fans.
 - Used for power conservation during turndown, or decrease in ambient air temperature
 - Control of process outlet temperature

UOP Practice: Specify 1/2 of fans in 'large (>50 hp)' duties as AV fans. This is an ENERGY CONSERVATION measure.

IPD-2004/HE-147

Driver Types

- **Standard Motors**
 - **Single Speed**
 - Limited turndown/winterization capability
 - Poor temperature control
 - Lowest cost
 - **Two Speed Motors**
 - Two forward operating speeds
 - Better turndown/winterization capability
 - Better temperature control
 - More costly

IPD-2004/HE-148

Driver Types (continued)

■ Variable-Speed Motors

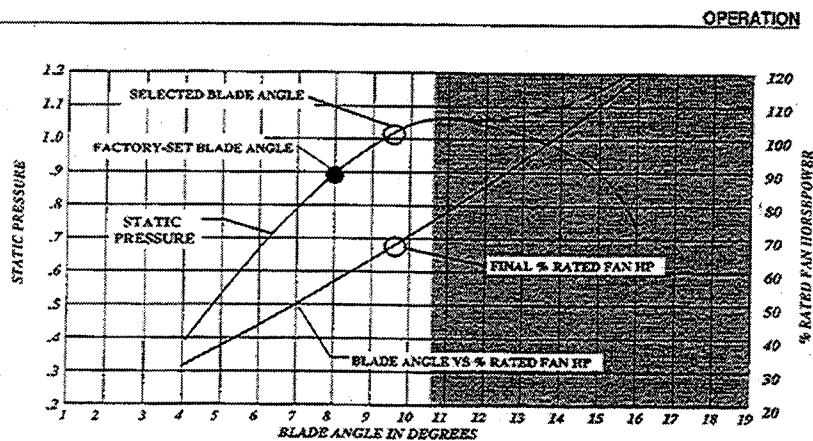
- Vary electrical frequency to motor; motor speed varies
 - Used to vary air flow rate
 - Best energy conservation
 - Best process temperature control
 - Very high cost
 - Used instead of AV fans

■ Steam Turbines

- UOP Practice: Specify 50% of motors as variable speed motors in large duties (>50 HP motor)

IPD-2004/HE-149

Fan Blade Angle Inspection



Note to the chart above that static pressure (and air flow) has reached its maximum at an 11 degree blade setting and blade overload is beginning. With further increase in blade angle, anything may happen, as indicated by the dotted extensions into the shaded overload area. Note

that the final selected blade angle is 5% below the point where the static pressure curve becomes level. The horsepower curve has been added to illustrate the point that in an overload condition, horsepower will increase without increased performance.

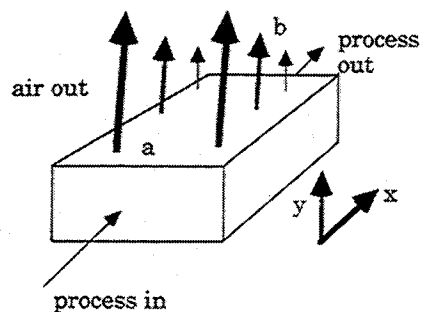
IPD-2004/HE-150

Geometric Effects

- Effects of Geometry
- Compensating for F-Factor Losses
- Combined Services
- Header Piping Layout

IPD-2004/HE-151

Effect of Geometry on Overall ΔT

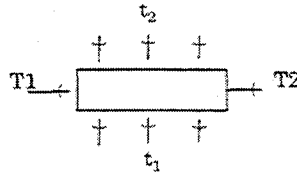


- Process fluid flows in “x” direction - cools down
- Air flows in “y” direction - heats up
- Air leaving bundle-top at “a” is hotter than air leaving at “b”
 $\Delta T (=T_{\text{process}} - T_{\text{air}})$ changes both *along* and *up* the bundle

EXC-R00-91
IPD-2004/HE-152

Effect of Geometry (continued)

- What is the effective temperature difference?



- t_2 is average air outlet temperatures, calculated as follows:

$$t_2 = \frac{Q}{W_{\text{air}} C_{p,\text{air}}} + t_1$$

- Where: Q is the duty [MW, kcal/hr or Btu/hr]
 W_{air} is the mass flow rate of air [kg/hr or lb/hr]
 $C_{p,\text{air}}$ is heat capacity of air [J/(kg·°C),
 kcal/(kg·°C) or Btu/(lb·°F)]

IPD-2004/HE-153

Effect of Geometry (continued)

- Effective overall temperature driving force depends on:
 - Log-mean temperature difference (T_{lm} , LMTD)

$$\Delta T_{\text{lm}} = \frac{(T_1 - t_2) - (T_2 - t_1)}{\log_e \left[\frac{(T_1 - t_2)}{(T_2 - t_1)} \right]}$$

- Cross flow correction factor, F :
- F (calculated for a range of inlet and outlet temperatures) is readily available from charts
- Charts plot F vs. R and P

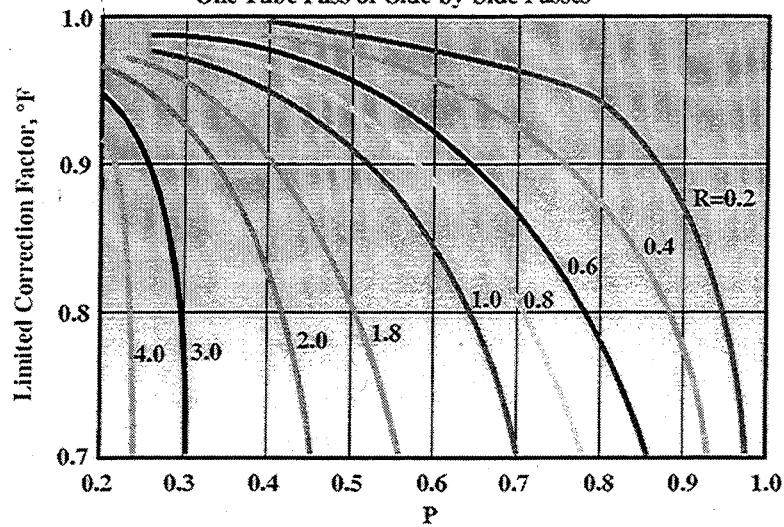
$$R = \frac{T_1 - T_2}{t_2 - t_1} \quad P = \frac{t_2 - t_1}{T_1 - t_1}$$

$$\Delta T_{\text{corr}} = F \cdot \Delta T_{\text{lm}}$$

IPD-2004/HE-154

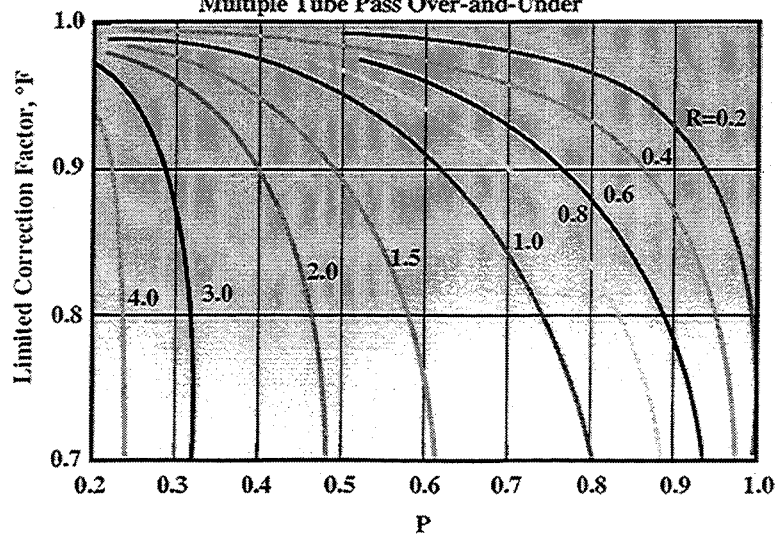
Correction Factors for LMTD Air Crossflow

One Tube Pass or Side-by-Side Passes



Correction Factors for LMTD Air Crossflow

Multiple Tube Pass Over-and-Under



Effect of Geometry (continued)

■ Example, one tube pass:

Process inlet = 250°F outlet = 120°F

Air inlet = 80°F outlet = 150°F

$R = 1.85$ $P = 0.41$ $F = 0.85$

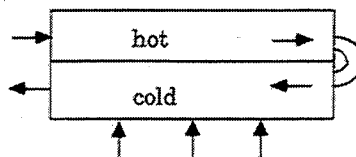
$\Delta T_{lm} = 65^\circ\text{F}$ $\Delta T_{corr} = 55^\circ\text{F}$

i.e. 15% loss of effectiveness

IPD-2004/HE-157

Compensating for F-Factor Losses

Consider the following two-pass arrangement



- Process fluid in lower pass “sees” cooler air
- Hotter process fluid sees hotter air
- temperature differences are “evened” out
- F-factor for two-pass flow (same temperatures as above) is 0.925 (compare to 0.85 for single-pass)
- MULTI-PASS gives better efficiency

EXC-R00-92
IPD-2004/HE-158

Compensating for F-Factor Losses (continued)

- Limit air temperature rise
 - Use fewer layers of tubes - depending on process type and process temperature difference
 - ⇒ Prevent F from getting too low

General rule for selection of number of tube layers

TYPE OF AIR COOLER	CONDENSER	COOLER
4 layers	$\Delta T_{\text{process}} < 50^{\circ}\text{F}$	$\Delta T_{\text{process}} < 80^{\circ}\text{F}$
6 layers	$\Delta T_{\text{process}} > 50^{\circ}\text{F}$	$\Delta T_{\text{process}} > 80^{\circ}\text{F}$

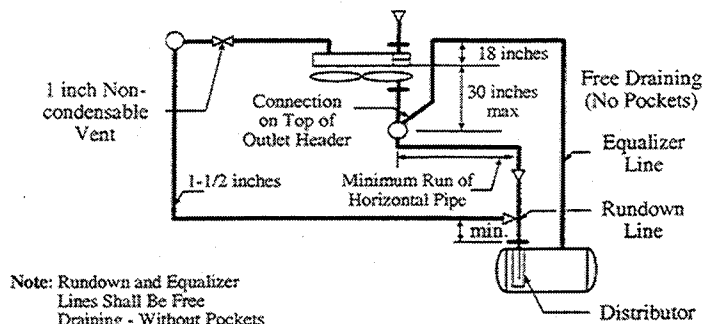
IPD-2004/HE-159

Combined Services

- Combine small services into same structure
- Cheaper (fewer motors, less structural cost)
 - Cheaper (fewer motors, less structural cost)
 - Less plot space
 - Avoid multiple length units
- Install louvers over portions of air-cooler, for control of less demanding service - prevent overcooling

IPD-2004/HE-160

Design Considerations - Piping Air Cooler on the P&I Diagram



Equalizer Line - Size 1/4 to 1/3 of Rundown Line Diameter

Rundown Line - Size Large

Distributor - Use Pipe with Vertical Slot

CRI-R00-24
IPD-2004/HE-161

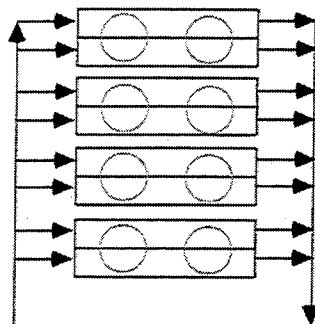
Header Arrangements

- Large duties require multiple bundles and/or bays
- Consider the following
 - Single-phase stream (liquid or gas)
 - Mixed-phase stream (Platformer effluent, NHT effluent)
- Following header arrangements are used:
 - D-type
 - E-type
 - C type

IPD-2004/HE-162

Header Arrangements (continued)

D Type



Any number of bundles

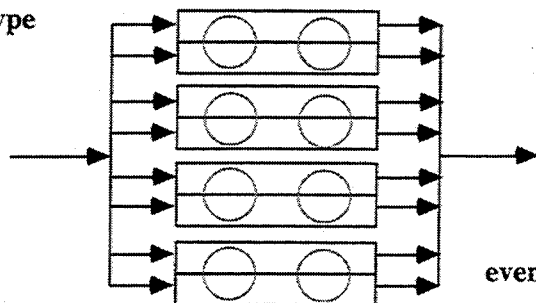
■ This gives

- Even distribution of single-phase
- Uneven distribution of two-phase flow
- No piping symmetry
- Low pressure drop

EXC-R00-93
IPD-2004/HE-163

Header Arrangements (continued)

E Type



even bundles count

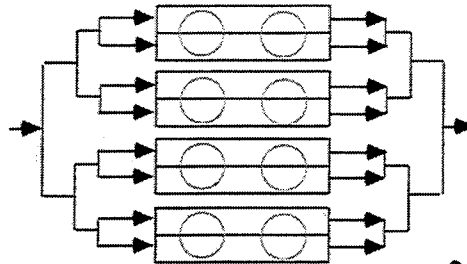
■ This gives

- Better distribution of 2-phase than D-type
- Symmetry around main header only
- Slightly higher pressure drop than D-type but less than C-type. (Additional elbows in piping)

EXC-R00-94
IPD-2004/HE-164

Header Arrangements (continued)

C Type



2, 4, 8, 16 bundles, 2ⁿ

- This gives
 - Best possible distribution of 2-phase mixture
 - Symmetry at every level
 - Highest pressure drop of the three (additional elbows in piping)
 - Most expensive to fabricate

EXC-R00-95
IPD-2004/HE-165

Header Arrangements (continued)

■ GENERAL RULE

- Single-phase - use D-type
- Two-phase -
 - low allowable ΔP - use E-type
 - large allowable ΔP - use C-type

IPD-2004/HE-165

Header Arrangements ***(continued)***

Examples

- | | |
|---|---|
| <ul style="list-style-type: none">■ #1 Hydrotreater product condenser?<ul style="list-style-type: none">- Large allowable ΔP- Two-phase in header- Water injection■ #2 Compressor intercooler?<ul style="list-style-type: none">- Single phase only | <ul style="list-style-type: none">■ #3 Column overheads condenser?<ul style="list-style-type: none">- Vapor-phase only in header■ #4 Isomar products condenser?<ul style="list-style-type: none">- Small allowable ΔP- Two-phase in header |
|---|---|

IPD-2004/HE-167

Header Arrangements ***(continued)***

Examples

- | | |
|---|---|
| <ul style="list-style-type: none">■ #1 Hydrotreater product condenser - C-type<ul style="list-style-type: none">- Large allowable ΔP- Two-phase in header- Water injection■ #2 Compressor intercooler - D-type<ul style="list-style-type: none">- Single phase only | <ul style="list-style-type: none">■ #3 Column overheads condenser - D-type<ul style="list-style-type: none">- Vapor-phase only in header■ #4 Isomar products condenser - E-type<ul style="list-style-type: none">- Small allowable ΔP- Two-phase in header |
|---|---|

IPD-2004/HE-168

Design Procedures

- Design Data Requirements
- Design Calculations
- Problems of Overdesign
- Temperature Control

IPD-2004/HE-169

Design Data Requirements

- Required data for hot fluid
 - Physical properties at inlet and outlet
 - Specific heat
 - Viscosity
 - Thermal conductivity
 - Heat load (duty)
 - Flow rate (reduced flow, turndown operations)
 - Inlet and outlet temperatures

IPD-2004/HE-170

Design Data Requirements (continued)

■ Required data for air

- Ambient temperature
 - Dry bulb temperature that is exceeded less than 5% of the summer months
 - Plant data
 - Meteorological survey
 - To use the maximum recorded temperature would produce uneconomically large air coolers.
 - Acceptable to add a few degrees to ambient temperature account for heat given off by surrounding equipment

IPD-2004/HE-171

Design Data Requirements (continued)

■ Design standards (vary by refinery)

- Tube lengths
 - Generally 30 ft (9.15 meters long)
 - Longer tubes $\Rightarrow$ cheaper installed cost per ft²
- Fouling Factors
 - Air side fouling factor
 - Tube side fouling factor (negligible compared to film coefficients for most services).

$$\frac{1}{U} = \left(\frac{1}{h_{\text{process}} \frac{A_b}{A_f}} + \frac{1}{h_{\text{air}}} + r_w + r_{\text{di}} \frac{A_f}{A_b} + r_{\text{do}} \right)$$

IPD-2004/HE-172

Problems of Overdesign

- Low tube-side velocity (laminar flow) produces high fouling rates
- Cooling liquids close to pour-point (high viscosity) results in excessive ΔP
- Excessively low $\Delta P_{\text{process}}$ causes a maldistribution of process fluid

IPD-2004/HE-173

Process Temperature Control

- Control temperature is accomplished by
 - Varying the air flow (vary ΔT_{corr})
- Louvers
 - Controlled manually
 - Controlled automatically
 - Disadvantage:
 - Snow, ice or corrosion affects operation of louvers
- Process Side Bypass

IPD-2004/HE-174

Process Temperature Control ***(continued)***

Auto-Variable Pitch Fans

- Blade pitch adjusted to vary air flow rate as ambient air temperature increases
- Pitch of the blade changes to increase air flow
- **Advantage**
 - Saves electricity when at low pitch
- **Disadvantage**
 - Higher maintenance on mechanical parts

IPD-2004/HE-175

Process Temperature Control ***(continued)***

Two Speed Motors

- Motor can be run at fast and slow speed as required
- **Advantage**
 - Saves electricity when at low speed
- **Disadvantage**
 - Limited to two air flow rates per fan

IPD-2004/HE-176

Process Temperature Control (continued)

Variable Speed Motors

- Fan speed reduced when less air is required
- **Advantage**
 - Reduces power consumption
- **Disadvantage**
 - Variable speed drives are expensive

IPD-2004/HE-177

Process Temperature Control (continued)

Internal Recirculation

- **Winterized Mode**
 - The top exhaust louvers automatically controlled by a temperature controller on process fluid outlet
 - As fluid temperature rises, the louvers open, more hot air out, more cold air in
- **Non-winterized Mode**
 - Louvers are completely opened both fans operate in standard mode

IPD-2004/HE-178

Process Temperature Control (continued)

Internal Recirculation (cont.)

■ **Advantage**

- Cheaper than full external recirculation
- Requires less plot area

■ **Disadvantage**

- Cannot provide complete freeze protection
- Air intake not controlled
- Subject to uneven air mixing
- High wind velocities affect air intake/outlet

IPD-2004/HE-179

Process Temperature Control (continued)

External Recirculation

■ **Winterized Mode**

- Top exhaust louvers controlled by process fluid temperature
- Provides control of air temperature into bundle
- When the inlet air louver closes, the bypass louver opens
- TIC measures air temperature at fan, and adjusts louvers

■ **Non-winterized Mode**

- Side manual louvers are opened
- Exhaust louvers are adjusted

IPD-2004/HE-180

Process Temperature Control (continued)

External Recirculation (cont.)

■ Advantage

- Maximum winterization protection
- Control of the inlet air temperature
- Requires little attention
- Reduces fan power (automatically controlled variable-pitch fans)

■ Disadvantage

- High cost (more controls, structural steel, louvers)
- Requires field assembly
- More moving parts

IPD-2004/HE-181

Inspection

■ Scheduled preventive maintenance and inspection - semi-annually

- To provide trouble free operation

■ Fans and Motors

- ✓ Check fans for vibration regularly
- ✓ Check and replace worn or cracked belts
- ✓ Inspect fan blades for deflection
- ✓ Check for cracks near hubs
- ✓ Grease all bearings
- ✓ Change oil in gear drives

IPD-2004/HE-182

Inspection (continued)

■ Tube Section

- √ Check inside tube section for accumulation of grease, dirt, bugs, leaves, etc.
- √ Clean tubes
 - Water washing or injection to clean salts
 - Hydroblasting

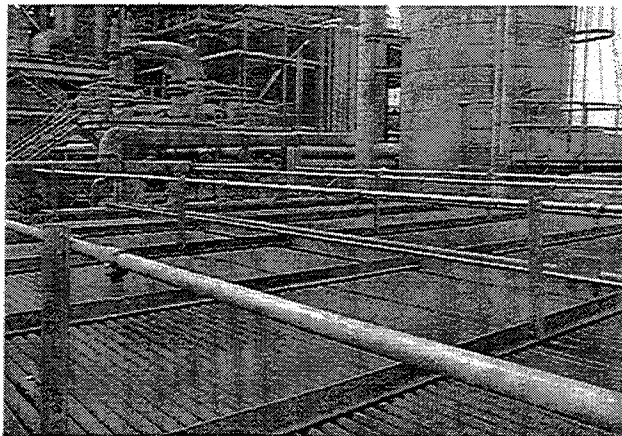
■ Current Temp > Ambient Design Temp

- Shut down plant
- Spray air cooler with water (not recommended)

IPD-2004/HE-183

Water Sprays

- Water Spray for Additional Cooling
- Fouling on Tubes
- Short Belt Life



IPD-2004/HE-184

Revamp Calculations

- Process flow rate is changed (inlet temperatures same)

$U, \Delta P_{\text{process}}$ change

- **OBJECTIVE:** Find new temperatures and duty

- **Method**

- Calculate $U_{\text{old}} = Q_{\text{old}} / (A \cdot \Delta T_{\text{corr,old}})$
- Calculate tube and fin resistance
- Calculate h_{process} (existing)
- (alternatively, get film coefficient from data sheet)
- Calculate h_{new} using scale-up formula:

$$h_{\text{new}} = h_{\text{old}} \left(\frac{\text{new mass flow}}{\text{old mass flow}} \right)^{0.8}$$

IPD-2004/HE-185

Revamp Calculations (continued)

- Calculate U_{new} by recombining h_{air} , tube and fin resistance, and h_{new}
- Estimate new process outlet temperature
- Air flow rate is same (same fan)
- Calculate new air outlet temperature

$$\frac{Q_{\text{new}}}{Q_{\text{old}}} = \left(\frac{T_{\text{proc,in}} - T_{\text{guess}}}{T_{\text{proc,in}} - T_{\text{proc,out,old}}} \right) \cdot \frac{\text{Mass Flow}_{\text{new}}}{\text{Mass Flow}_{\text{old}}}$$

$$\Delta t_{\text{air,new}} = (t_{\text{air,out,old}} - t_{\text{air,in}}) \cdot (Q_{\text{new}}/Q_{\text{old}})$$

- Calculate new $R, P, \Delta T_{\text{lm}}, \Delta T_{\text{corr}}$
- Calculate $A_{\text{req}} = Q_{\text{new}} / (U_{\text{new}} \Delta T_{\text{corr}})$

if $A_{\text{req}} \neq A$, adjust process outlet temperature, repeat procedure.

IPD-2004/HE-186

Revamp Calculations (continued)

- **EXAMPLE:** An existing cooler has 50 000 ft² area
 - Air inlet is 82°F
 - Air outlet is 135°F
 - Process inlet is 150°F
 - Process outlet is 110°F
 - Air cooler is 6 layer, two-pass
 - Duty is 3 MM Btu/hr
- Calculate existing U value

IPD-2004/HE-187

Revamp Calculations (continued)

$\Delta T_{lm} =$	20.8
$R =$	0.75
$P =$	0.78
$F =$	0.85
$\Delta T_{corr} =$	17.7

$$U_{old} = \frac{Q}{A \Delta T_{corr}} = \frac{3 \times 10^6}{50000 \times 17.7}$$

$$U_{old} = 3.39 \text{ Btu/(hr-ft}^2\text{°F)}$$

- Process flow rate is increased by 10%
Existing process side film coefficient ($h_{process}$) is 168 Btu/(hr-ft²°F) from the data sheet.
 $h_{new} = 168(1.1)^{0.8} = 181.3 \text{ Btu/(hr-ft}^2\text{°F)}$

IPD-2004/HE-188

Revamp Calculations (continued)

- $U_{\text{new}} = 3.5 \text{ Btu}/(\text{hr}\cdot\text{ft}^2\cdot^\circ\text{F})$
(0.005 for metal resistance, 7 for h_{air})

- Guess $T_{\text{process,new}} = 112^\circ\text{F}$

- New duty and air outlet temperature

$$Q_{\text{new}} = Q_{\text{old}} \left(\frac{T_{\text{proc,in}} - T_{\text{guess}}}{T_{\text{proc,in}} - T_{\text{proc,out,old}}} \right) \cdot 1.1 =$$

$$= 3 (150-112) / (150-110) 1.1 = 3.135 \text{ MM Btu/hr}$$

$$t_{\text{air,out,new}} = 82 + (135 - 82) \cdot 3.135 / 3 = 137.4^\circ\text{F}$$

$$\Rightarrow R = 0.69, P = 0.81, F = 0.85$$

$$\Rightarrow \Delta T_{\text{lm}} = 20.1, \Delta T_{\text{corr}} = 17.1$$

$$\Rightarrow A_{\text{required}} = (3.135 \times 10^6) / (3.5 \times 17.1) = 52\,400 \text{ ft}^2.$$

IPD-2004/HE-189

UOP Process Design Parameters

- Pressure Drop
- Approach Temperature
- Temperature to Storage
- Selection of Sides
- Temperature Cross

IPD-2004/HE-190

Shell & Tube Pressure Drop

- Normal - 5 psi on shell side & 10 psi on tube side
- Shells in series - Extra 5 psi for each extra shell
- Horizontal Reboilers - Estimated 0.35 psi for bundle only
- Vertical Reboilers - Function of tube length
- Combined Feed Exchangers - 7 - 15 psi total for both sides
- Vacuum Condensers - 1 - 1.5 psi
- High Viscosity Fluids (>20 cp) - Evaluate it using HTRI

IPD-2004/HE-191

Air Cooler Pressure Drop

- Normal - 5 psi
- Vacuum Condensers - 3 psi
- High Viscosity Fluids (>10 cP) - Evaluate it using HTRI
- Greater than 5 cP @ inlet or 100 cP @ outlet -consider using tempered water instead

IPD-2004/HE-192

Shell & Tube Approach Temperature

- Water Condensers - 7 to 15°F
- Water Coolers - Typically < 120°F
- Steam Reboilers - 25°F approach
- Hot Oil Reboilers - 50°F approach
- Combined Feed - 85°F
- Packinox Combined Feed - 65°F

IPD-2004/HE-193

Air Cooler Approach Temperature

- Target - 20°F above dry bulb

IPD-2004/HE-194

Temperature to Storage

- Light Hydrocarbons - $< 100^{\circ}\text{F}$
- Kerosene - 125°F
- Middle Distillate - 125°F
- Vacuum Gas Oil - 150°F
- Reduced Crude / Resid - 175 to 200°F

IPD-2004/HE-195

Selection of Sides

- Fouling Fluids - Tube side
- Viscous Fluids (≥ 2 cs) - Shell side
- Suspended Solids - Tube side
- Highest Temperature - Tube side
- Highest Pressure - Tube side
- Cooling Water - Tube side
- Corrosive Fluid - Tube side
- Much Larger Flow ($> 2:1$) - Shell Side
- Often Creates Conflicts - Compromise

IPD-2004/HE-196

Temperature Cross

- Requires Multiple Shells for Multiple Tube Pass Exchangers
- Increases Pressure Drop
- Increases Exchanger Capital Cost
- Often Done to Save Utilities
- Not Required in Hairpin or Plate Exchangers



Training Services

Heat Exchanger Appendices

IPD-2004/HE-APP-1

Winterization

- Freezing and Overcooling
- Problems Caused by Overcooling
- Calculations
- Methods of Winterizing

IPD-2004/HE-APP-2

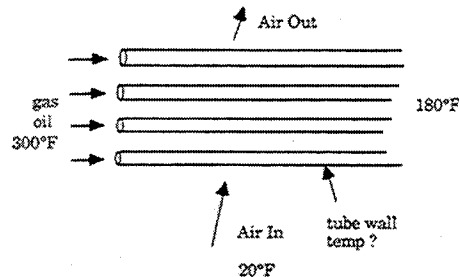
Freezing and Overcooling

- Design must give stable operation ALL YEAR (summer and winter)
- Prevent: Freezing/Overcooling
 - Excessively cold air can cause overcooling or freezing
 - Worst for bottom layer of tubes
- “Overcooling”
 - Is a relative concept:
 - Light naphtha at 25°F is not overcooled
 - Light naphtha (*with free water*) at 25°F is overcooled
 - HVGO at 25°F is overcooled - (HIGH μ)

IPD-2004/HE-APP-3

Freezing and Overcooling (continued)

- “Critical” process temperatures
 - Freezing point
 - Pour point
- Bottom layer of tubes most likely to be overcooled
- Air is coldest here, therefore
 - ⇒ Tube wall is colder here
 - ⇒ Tube side fluid will be prone to overcooling



IPD-2004/HE-APP-4

Problems Caused by Overcooling

- FREEZING may occur at tube wall
 - Results in constriction of flow
 - ⇒ Limits flow of hot fluid to blocked tubes
 - ⇒ Freezing is intensified
 - Tube-side pressure drop increases flow maldistribution

IPD-2004/HE-APP-5

Calculations

- Calculation of inner tube wall temperature
$$T_w = T_b - r (T_b - t_b)$$
- r : Overall thermal resistance including inside and outside surface areas of the finned tube (dimensionless)

Dependent on tube and fin geometry, process conditions

- T_w : Inner tube wall temperature
- T_b : Bulk temperature of tube-side fluid
- t_b : Bulk temperature of air
- for: 1" OD, 0.1" wall thickness CS tube
2-1/4" triangular pitch
10 fins per inch, 0.625" high, 0.016" thick
 $r = 0.774$

IPD-2004/HE-APP-6

Calculations

(continued)

■ Example:

- Suppose the pour point of a gas oil is 60°F
- Recommended wall temperature
= pour point + *safety margin*
15-25°F depending on process fluid (API standard), $T_w = 60^\circ + 25^\circ = 85^\circ\text{F}$
- for this case, $T_w = 180 - 0.774 (180 - 20)$
 $T_w = 56^\circ\text{F}$
(require 85°F for safe operation)

■ Therefore - WINTERIZE THIS EXCHANGER

IPD-2004/HE-APP-7

Methods of Winterizing

- API 661 Appendix G – Winterization of Air-Cooled Heat Exchangers
- Control overcooling by **LIMITING AIR-FLOW**
 - WINTER airflow < SUMMER airflow
- Louvers
 - Contain and control air flow when process fluid is in danger of being overcooled
- Example
 - Directly after shutdown, NO FLOW, FANS OFF
 - During start-up, FLOW SMALL, FANS ON

IPD-2004/HE-APP-8

Methods of Winterizing (continued)

■ Steam Coils

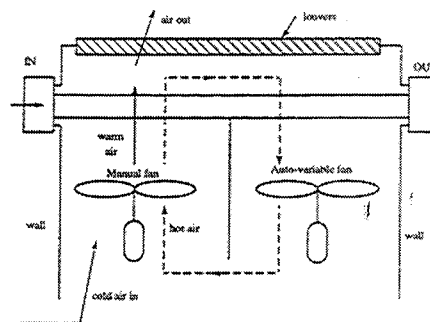
- HEAT air entering tube bundle
- Alternative: Prevent overcooling by increasing air inlet temperature
- Condensate can freeze

IPD-2004/HE-APP-9

Methods of Winterizing (continued)

■ Internal air recirculation

Air circulated
inside main
enclosure



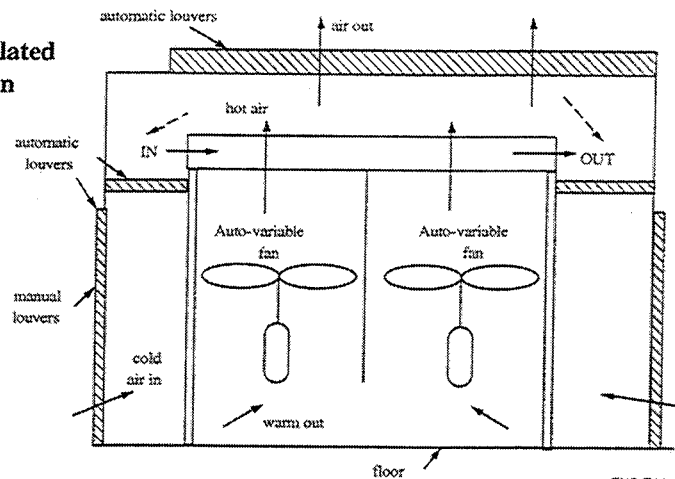
- Manual fan - at process inlet end
 - Brings cold air into enclosure
- Auto-variable fan at process outlet end - pulls hot air DOWN through bundle
 - Manual fan mixes some hot air with cold air

EXC-R00-93
IPD-2004/HE-APP-10

Methods of Winterizing (continued)

■ External air recirculation

Air recirculated
outside main
enclosure



EXC-R00-94
IPD-2004/HE-APP-11

Methods of Winterizing (continued)

- Uses auto-variable fans at low pitch
 - Sides fitted with manual louvers
 - Top fitted with automatically controlled louvers
 - Recirculation chamber
- Over both sides of unit
 - Provides a duct for mixing cold air with hot (recirculated) air
 - Floor included - prevent cold air leaks

IPD-2004/HE-APP-12