054410 Plant Design

LECTURE 7: HEAT EXCHANGER DESIGN

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Ref: Seider, Seader and Lewin (2004), Chapter 13

Lecture Objectives

After this lecture, you should:

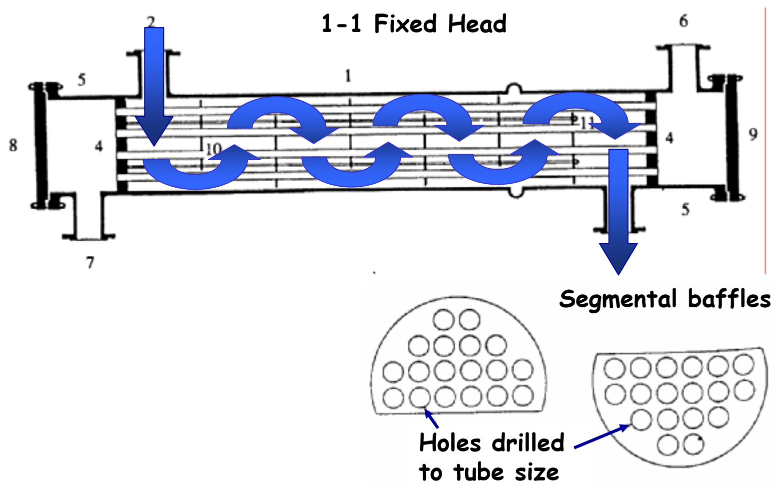
❶ Be familiar with the major types of available heat-exchange equipment, with particular emphasis on shell-and-tube heat exchangers.

❷ Know how to estimate overall heat transfer coefficients for a shell-and-tube heat exchanger.

❸ Know how to compute pressure drops on both sides of a shell-and-tube heat exchanger.

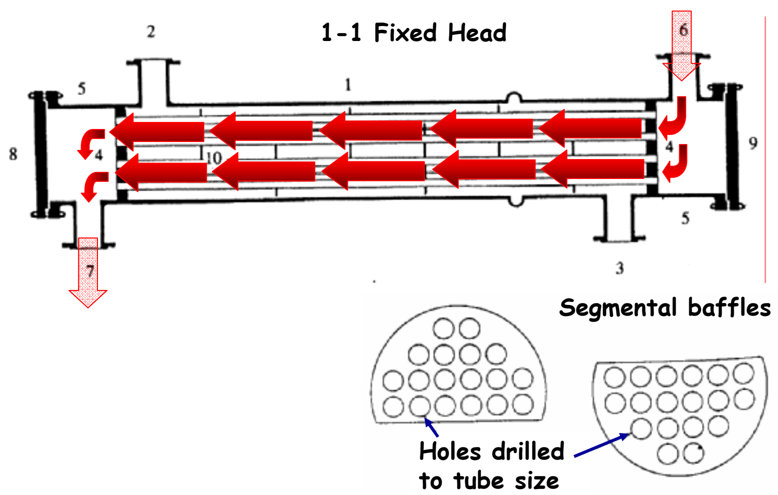
❹ Be able to perform mechanical design of the most appropriate shell-and-tube heat exchanger to meet desired duty and pressure drops.

Shell-and-Tube Heat Exchangers



7 - 3 PLANT DESIGN - Daniel R. Lewin Heat Exchanger Design

Shell-and-Tube Heat Exchangers



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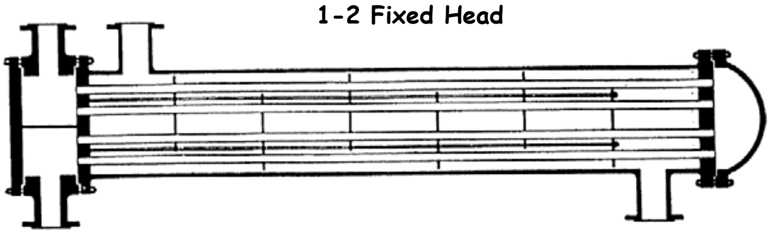
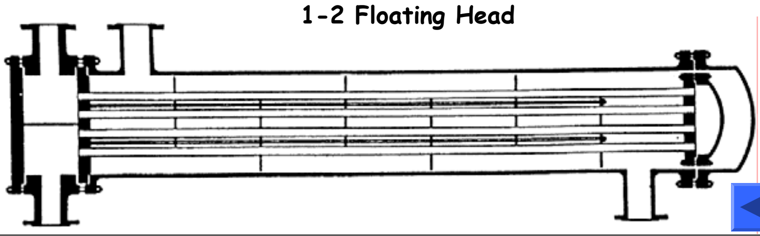
Shell-and-Tube Heat Exchangers

1-2 Fixed Head



1-2 Floating Head



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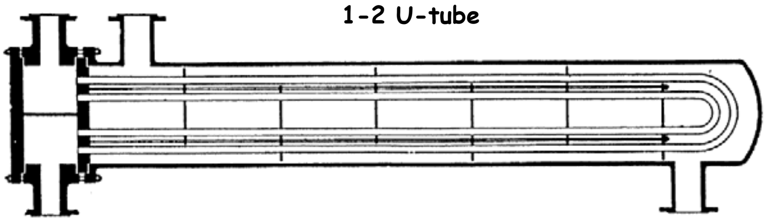
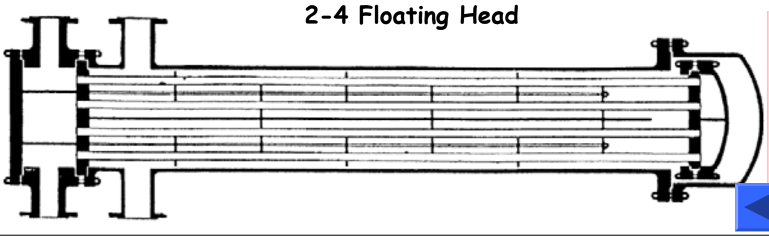
Shell-and-Tube Heat Exchangers

1-2 U-tube



2-4 Floating Head



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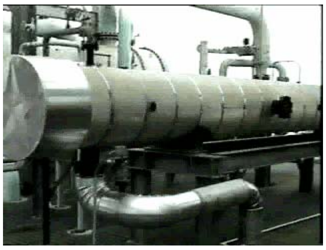
Shell-and-Tube Heat Exchangers

These movies demonstrate the flow of process fluids in a typical shell-and-tube heat exchanger set-up.



Shell-and-Tube Heat Exchangers

The following heuristic (from Seider et al, 2004) is useful to assist in selecting which process fluid should be designed to pass through the tubes, and which should pass through the shell:

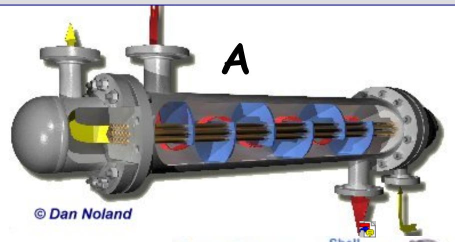
<u>Heuristic 55:</u>

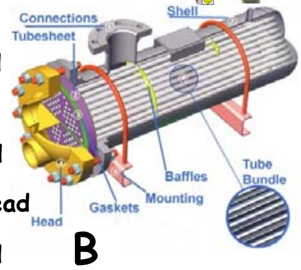
The tube side is for corrosive, fouling, scaling, hazardous, high temperature, high-pressure, and more expensive fluids.

The shell side is for more viscous, cleaner, lower flow-rate, evaporating and condensing fluids.

Quiz: Identify the following…

A





1-1 Fixed Head
1-2 U-tube
1-2 Fixed Head
2-4 Floating Head
3-6 Fixed Head

B

C

http://www.tema.org

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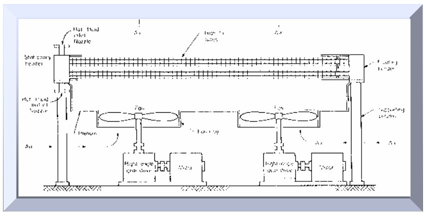
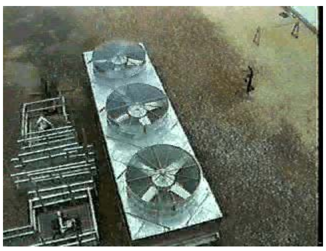
Fin-fan Heat Exchangers

In fin-fan heat exchangers, air is forced in cross-flow across tubes carry-ing process fluid.





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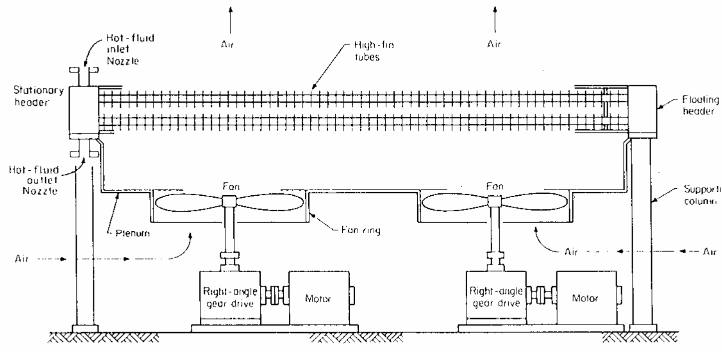
Fin-fan Heat Exchangers



<u>Design issues:</u> (a) Use Heuristic 56 for initial design; (b) Design the tube-banks similarly to a shell-and-tube heat exchanger.

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Fin-fan Heat Exchangers

<u>Heuristic 56:</u>

For an air-cooled exchanger, the tubes are typically ¾-1" in outside diameter. The ratio of fin surface area to tube outside bare area is 15-20. Fan power requirement is the range 2-5 Hp per 10^6 Btu/hr, or 20 Hp per 1,000 ft^2 of tube outside surface (fin-free) area. Minimum approach temperature is about 50 °F (much higher than water-cooled exchangers). Without the fins, the overall heat transfer coefficients would be about 10 Btu/hr ft^2 °F. With the fins, U = 80-100 Btu/hr ft2 oF, based on tube outside, bare surface area.

<u>See also: Ludwig, Vol. 3.</u>

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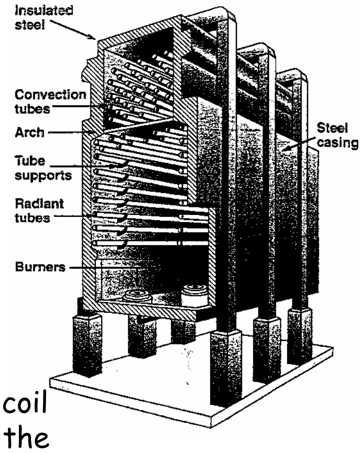
Furnaces -Typical Applications

Example of uses of process furnaces:

- Steam boilers
- Distillation column reboiler
- Heating distillation column feed stream
- Heating reactor feed stream
- Heating a heating stream
- Reactor (reaction inside furnace coil)



Operation principles:

Process stream flows through a coil heated by combustion of fuel in the furnace chamber.

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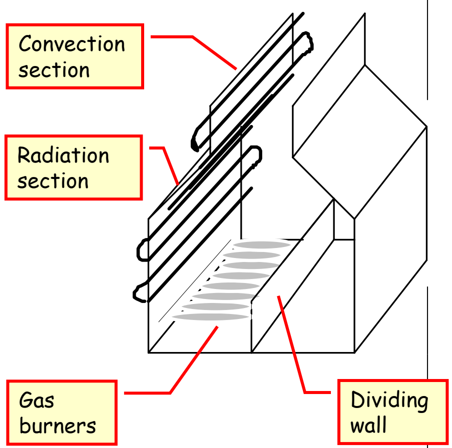
Typical Furnace Coil Arrangements

- Vertical tube bank along the walls of a circular furnace
- As above, with the addition of horizontal tube bank in a convection section
- A single horizontal tube bank in the center of the furnace, with burners along each side wall
- Parallel horizontal tube banks along each wall, with a central wall.



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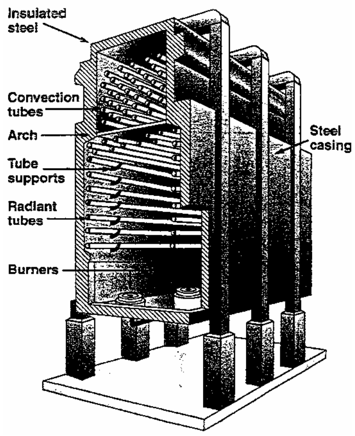
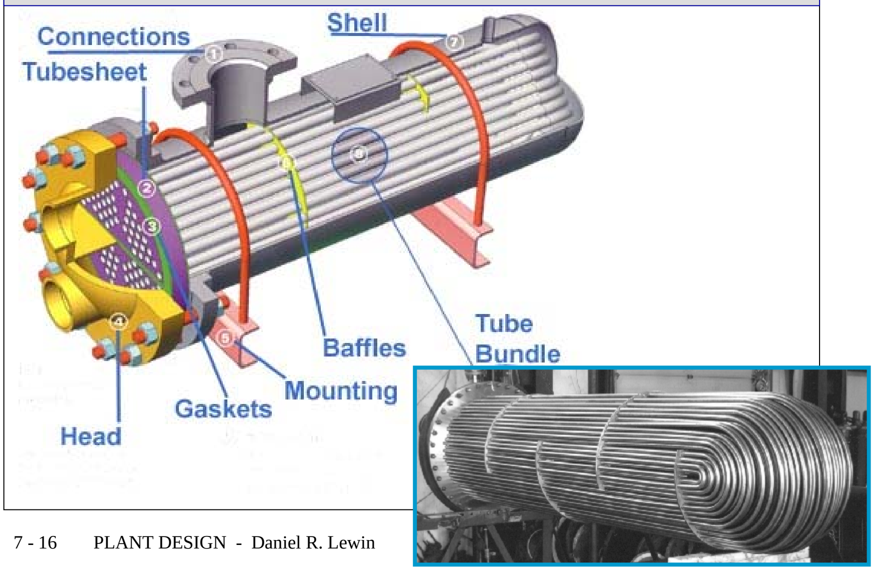
Furnaces – Getting Started

Heuristic 57:

Typical fluxes in fired heaters are 12,000 Btu/hr-ft^2 in the radiation section and 4,000 Btu/hr-ft^2 in the convection section, with approximately equal duties in the two sections. Typical process liquid velocity in the tubes is 6 ft/s. Thermal efficiencies for modern fired furnaces is 80-90%.



Shell-and-Tube Heat Exchangers



Temperature-Driving Forces

The rate of heat transfer in an shell-and-tube exchanger is computed as:

$$Q = m_c \cdot (H_{C,out} - H_{C,in}) = m_h \cdot (H_{H,out} - H_{H,in})$$

Assuming (1) steady-state; (2) counter- or co-current flow; (3) constant overall heat transfer coefficient; (4) no phase changes on either side; and (5) negligible heat losses:

$$Q = U \cdot A \cdot \Delta T_{LM}$$

where: $$\Delta T_{LM} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left(\dfrac{T_{h,in} - T_{c,out}}{T_{h,out} - T_{c,in}}\right)}$$

Temperature-Driving Forces

For multiple-pass shell-and-tube exchangers, the flow directions of the two fluids are combinations of countercurrent and co-current flow, reducing the effective value of ΔT_{LM} . For a **1-2 exchanger**, with assumptions 1, 3, 4 and 5:

F_T for 1-2

$$\boxed{Q = U \cdot A \cdot \Delta T_{LM} \cdot F_T}$$

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

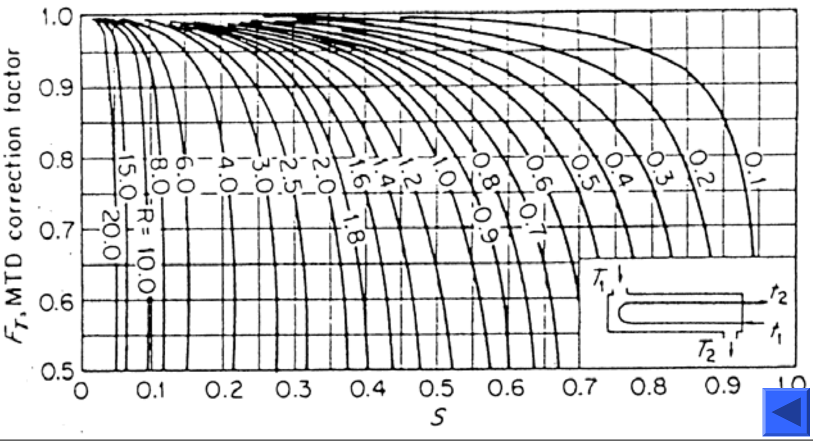
Nagle (1933)

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}}, \text{ and } S = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}}$$

It is desirable to have a value of F_T of 0.85 or higher. Values below 0.75 are unacceptable. **F_T = 1 for phase change in duty fluid** When $F_T < 0.75$, increase the shell passes.

Temperature-Driving Forces

F_T **for 1-2 shell-and-tube exchanger.**



Example 13.5

A hot stream is cooled from 200 to 140 °F by a cold stream entering at 100 °F and exiting at 190 °F. Determine the true ΔT_{LM} and select the appropriate shell-and-tube configuration.

<u>Solution:</u>

For counter-current flow: $\Delta T_{LM} = \dfrac{40 - 10}{\ln\left(\dfrac{40}{10}\right)} = 21.6\ ^\circ\text{F}$

For multiple-pass exchangers:

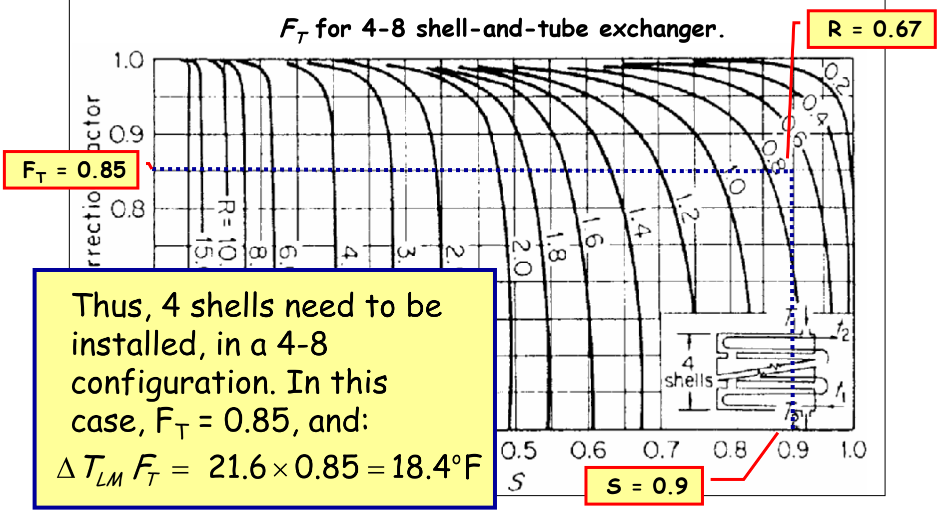
$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}} = \frac{200 - 140}{190 - 100} = 0.667$$

F_T **for 1-2**

$$S = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} = \frac{190 - 100}{200 - 100} = 0.9$$

F_T **for 4-8**

Example 13.5 (Cont'd)



Thus, 4 shells need to be installed, in a 4-8 configuration. In this case, F_T = 0.85, and:

$$\Delta T_{LM} F_T = 21.6 \times 0.85 = 18.4°F$$

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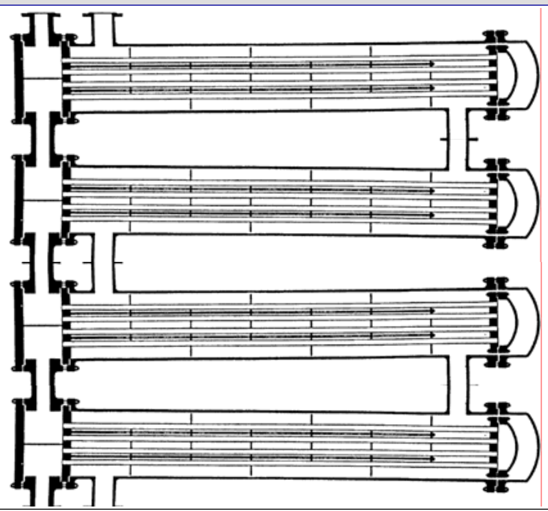
Example 13.5 (Cont'd)



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Class Exercise 1

A hot stream is cooled from 1,150 to 560 °F by a cold stream entering at 400 °F and exiting at 1,000 °F. Determine the true ΔT_{LM} and select the appropriate shell-and-tube configuration.

Solution:

For counter-current flow: $\Delta T_{LM} =$

For multiple-pass exchangers:

$$R = \frac{T_{h,in} - T_{h,out}}{T_{c,out} - T_{c,in}} =$$

$$S = \frac{T_{c,out} - T_{c,in}}{T_{h,in} - T_{c,in}} =$$

Configuration	F_T

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Heat Transfer Coefficients

Sieder-Tate (1936) equations:

(a) for tube-side, dimensionless heat transfer coefficient:

$$J_H = \frac{h_T D_I}{k}\left(\frac{C_p \mu}{k}\right)^{-1/3}\left(\frac{\mu}{\mu_W}\right)^{-0.14} \quad \text{vs.} \quad \left(\frac{D_I G_T}{\mu}\right)$$

Tube-side mass flux: $G_T = \frac{W_T}{A_T}, A_T = \frac{\pi D_I^2}{4}$

(b) for shell-side, dimensionless heat transfer coefficient:

$$J_H = \frac{h_O D_O}{k}\left(\frac{C_p \mu}{k}\right)^{-1/3}\left(\frac{\mu}{\mu_W}\right)^{-0.14} \quad \text{vs.} \quad \left(\frac{d_e G_S}{\mu}\right)$$

Shell-side mass flux: $G_S = \frac{W_S}{A_S}, A_S = \frac{d_S C' B}{144 P_T}$

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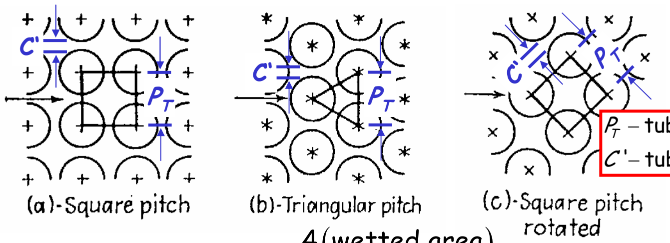
Shell-side Equivalent Diameter



P_T – tube pitch (in)
C' – tube clearance (in)

$$d_e = \frac{4\,(\text{wetted area})}{\text{wetted perimeter}}$$

□ pitch: $d_e = \dfrac{4P_T^2 - \pi D_O^2}{\pi D_O}$ △ pitch: $d_e = \dfrac{\sqrt{12}P_T^2 - \pi D_O^2}{\pi D_O}$

Shell-side mass flux: $G_S = \dfrac{W_S}{A_S},\ A_S = \dfrac{d_S C' B}{144 P_T}$

W_S – total shell side mass flow (lb/hr), A_S – shell crossflow area (ft^2)
d_S – shell diameter (in), B – baffle spacing (in)

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Heat Transfer Coefficients

$$U_{dirty} = \frac{1}{R_{F,O} + \left(\frac{1}{h_O}\right) + \cancel{\left(\frac{t_W A_O}{k_W A_M}\right)} + \left(\frac{A_O}{h_I A_I}\right) + R_{F,I}\left(\frac{A_O}{A_I}\right)}$$

Outside fouling

External film resistance

Wall resistance

Internal film resistance

Internal fouling

$$A_O = \pi D_O L \qquad A_I = \pi D_I L \qquad A_M = \frac{\pi L (D_O - D_I)}{\log(D_O / D_I)}$$

$$U_{dirty} = \frac{1}{R_{F,O} + \left(\frac{1}{h_O}\right) + \left(\frac{D_O}{h_I D_I}\right) + R_{F,I}\left(\frac{D_O}{D_I}\right)} \qquad U_{clean} = \frac{1}{\left(\frac{1}{h_O}\right) + \left(\frac{D_O}{h_I D_I}\right)}$$

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Tube Pressure Drop

Pressure drop of the fluid flowing in the tube-side of a heat exchanger is given by the Darcy formula:

$$\Delta P_t = \frac{f G_T^2 L N_T}{5.22 \times 10^{10} D_I S \left(\mu/\mu_w\right)^{0.14}} \text{ [psi]}$$

f = friction factor [ft²/in²], G_T = tube mass velocity [lb/ft² hr], L = tube length [ft], N_T = total number of tube passes, S = specific gravity, D_I = tube I.D. [ft].

In addition, the repeated changes in direction caused by the numerous passes in the tubes adds additional pressure loss, called the "return loss":

$$\Delta P_r = \frac{4 N_T v_T^2}{2g} \text{ [psi]}$$

The total pressure drop is: $\Delta P = \Delta P_t + \Delta P_r$ [psi]

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Shell Pressure Drop

Pressure drop of the fluid flowing on the shell side of a heat exchanger is given by the Darcy formula:

$$\Delta P_s = \frac{f G_S^2 D_S (N_B + 1)}{5.22 \times 10^{10} D_e S \left(\mu/\mu_w\right)^{0.14}} \text{ [psi]}$$

f – friction factor [ft²/in²]

G_S – mass velocity in shell [lb/ft² hr]

D_S – I.D. of shell [ft]

N_B – number of baffles

D_e – equivalent diameter [ft]

S – specific gravity

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Main Steps Involved

Objective: To design a shell-and-tube exchanger to perform heat transfer from a hot stream to a cold stream.

Specifications: Given stream physical properties, mass flow rate, process stream source and target temperatures, and the mass flow rate and source temperature of the duty stream.

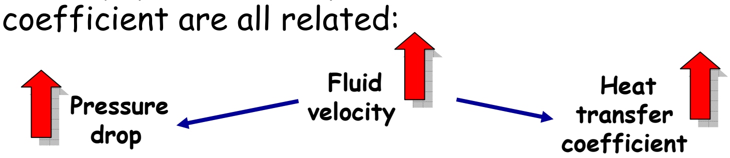
The mechanical design of a shell-and-tube heat exchanger involves two main steps:

❶ Computation of the heat duty

❷ Shell and tube configuration

Iterative Design Procedure

- ❑ It is of interest to reduce the heat transfer surface area to a minimum, since this will lead to the cheapest design.
- ❑ Must satisfy the pressure drop specification (usually pre-defined), which affects the overall heat transfer coefficient.
- ❑ An iterative design is called for since fluid velocity, pressure drop and heat transfer coefficient are all related:



Iterative Design Procedure

Computation of shell-and-tube exchangers involves iteration, since the heat transfer coefficients, pressure drops and heat transfer area all depend on the design's geometric configuration, which needs to be determined.

The geometric configuration (to be determined) includes the following:

1. Shell diameter
2. Tube diameter
3. Tube length
4. Tube packing configuration (pitch) and spacing
5. Number of tube and shell passes

Class Exercise 2

Design a shell-and-tube heat exchanger to preheat a stream of 30 T/hr containing ethylbenzene and styrene from 10 to 97 °C.

Additional data:

Density – 856 $kg \cdot m^{-3}$, Viscosity – 0.4765 cP,
Specific heat – 0.428 $kcal \cdot kg^{-1} \cdot {}^{o}C^{-1}$,
Thermal conductivity – 0.133 $kcal \cdot hr^{-1} \cdot m^{-1} \cdot {}^{o}C^{-1}$
Heat supply medium – Saturated steam at 10 barg.

Notes: (a) For this application, the process fluid is fed to the tubes.
(b) Maximum ΔP in the process side is 0.8 bar.
(c) Fouling - process: 0.0002, steam: 0.0001 hr m^2/kcal

Class Exercise 2 - Solution

A. Stream Data

Parameter	Units	Cold side Tube side	Hot Side Shell side	Notes
Fluid		EB/Styrene	Sat. steam	$Q = m_{tube} \cdot Cp_{tube} \cdot \Delta T$
Mass Flow	kg/hr	30,000	2,113	$=30{,}000 \times 0.428 \times (97-10)$
Inlet Temp	°C	10	115	= 1,117,080 kcal/hr $m_{steam} = Q/\lambda$
Outlet Temp.	°C	97	115	$\lambda = 528.7$ kcal/kg
Density	kg/m^3	856	0.9712	
Viscosity	cP	0.4765	0.1262	
Cp	Kcal/kg °C	0.428		
K	Kcal/hr cm	0.133		
Fouling factor	hr m^2/kcal	0.0002	0.0001	

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Class Exercise 2 - Solution

B. LMTD Calculation : T- Shell t - tube

Variable	Units	Value	Notes
$\Delta T_1 = T_i - t_o$	°C	18	
$\Delta T_2 = T_o - t_i$	°C	105	
LMTD	°C	49.33	$LMTD = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} = \frac{105-18}{\ln(105/18)} = 49.33$
c-LMTD	°C	49.33	c-LMTD = $\Delta T_{LM} \times F_T$ $F_T = 1$ (phase change)

C. Heat Duty

Q	Kcal/hr	1.117×10^6	See previous table
$U_{Estimated}$	kcal/(hr °C m^2)	490-980	Item 5: For light organics, U = 100-200 BTU/(hr °F ft^2).
$A_{Estimated}$	m^2	23-46	$A = Q/(U_{Estimated} \times \text{c-LMTD})$

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Class Exercise 2 - Solution

D. Heat Exchanger Configuration

Variable	Units	Value	Notes
Tube passes, N_T		4	Assumed
Shell passes, N_S		1	Assumed
Tubing O.D., D_O	m	0.0254	Taking 1" (I.D.) 12 BWG tubing as basis. Thus D_I = 0.782"
Tubing I.D., D_I	m	0.0198	
Tube velocity, V_T	m/sec	1.4	Allowed range: 1.2-3 m/sec 1.4 m/sec = 4.59 ft/sec
Tube c-section (I.D.), A_T	m^2	3.079×10^{-4}	$A_T = (\pi D_I^2)/4$
q_T in each tube	m^3/hr	1.55	$q_T = A_T \times V_T = 4.31\times10^{-4}$ m^3/sec
No. tubes per pass, N		23	$N = m/(q_T \times \rho)$ $= 30{,}000/(1.55\times856)$
Total no. tubes, N_{Total}		92	$N \times N_T/N_S$
Tube length, L	m	6	Accepted industry standard

Class Exercise 2 - Solution

D. Heat Exchanger Configuration (Cont'd)

Variable	Units	Value	Notes
Heat exchanger area, A	m^2	44	$A = N_S \times N_{Total} \times L \times \pi \times D_O$ = 44 m^2
Pitch		$\triangle$-$1\frac{1}{4}$"	$\triangle$-pitch selected (why?)
Shell I.D., d_s	m	0.4382	$17\frac{1}{4}$" shell holds 106 tubes.
$A_{Available}$	m^2	51	$A = N_S \times 106 \times L \times \pi \times D_O$ = 51 m^2

Note that the available heat transfer area, 51 m^2, is larger than the value estimated previously, 23-46 m^2, so can be reduced!

We shall now compute the heat transfer coefficient and the pressure drops in the tube and shell, and compare with our targets.

Class Exercise 2 - Solution

E. Heat Transfer Coefficient Calculations

1.1 Tube-side heat exchange.			
Tube c-section (I.D.), A_T	m^2	3.08×10^{-4}	$A_T = (\pi D_I^2)/4$
c-section area/pass, A_T'	m^2	8.21×10^{-4}	$A_T' = A_T\times(106/4)$
G_T	Kg/(hr m^2)	3,653,462	$G_T = m_{tube}/A_T'$
Re_T		42,304	$Re = G_T\times D_I/\mu$
J_H		130	See Item 10
Pr		5.52	$Pr = Cp\times\mu/K$
$\Phi = (\mu/\mu_W)^{-0.14}$		1	Assume $\mu = \mu_W$
$J_H = (h_I D_I/K)\cdot Pr^{-1/3}(\mu/\mu_W)^{-0.14} \Rightarrow h_I = 1{,}538$ Kcal/(hr m^2 °C)			
h_I	Kcal/(hr m^2 °C)	1,538	
1.2 Shell-side heat exchange.			
h_O	Kcal/(hr m^2 °C)	7,342	For steam, use accepted value: 1,500 Btu/(hr ft^2 °F

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Class Exercise 2 - Solution

E. Heat Transfer Coefficient Calculations (Cont'd)

1.3 Overall heat transfer coefficients.			
U_{Clean}	Kcal/(hr m^2 °C)	1,033	
U_{Dirty}	Kcal/(hr m^2 °C)	756	
$U_{Service}$	Kcal/(hr m^2 °C)	444	$U_{Service} = Q/(A_{Available}\times c\text{-LMTD})$
$U_{Estimated}$	Kcal/(hr m^2 °C)	490-980	

$$U_{clean} = \frac{1}{\left(\frac{1}{h_O}\right)+\left(\frac{D_O}{h_I D_I}\right)}$$

$$U_{dirty} = \frac{1}{R_{F,O}+\left(\frac{1}{h_O}\right)+\left(\frac{D_O}{h_I D_I}\right)+R_{F,I}\left(\frac{D_O}{D_I}\right)}$$

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Class Exercise 2 - Solution

F. Pressure Drop Calculations

2.1 Tube-side pressure drop.

Re_T		42,304	$Re = G_T \times D_I/\mu$
f	ft^2/in^2	0.000185	See Item 11
ΔP_t, friction	psi	2.80	
ΔP_r, return	psi	5.30	$\Delta P_r = 4N_T V_t^2/2g$ $= 4 \times 4 \times 4.59^2/2*g$ $=$ 5.3 psi
ΔP	bar	0.55	$\Delta P_{TOT} = \Delta P_t + \Delta P_r$ = 8.1 psi

Class Exercise 2 - Solution

F. Pressure Drop Calculations (Cont'd)

2.2 Shell-side pressure drop.

B	in	17.25	Assume baffle spacing = shell I.D.
$N_B + 1$		14	N_B = no. of baffles = L/B – 1 = 12.7
d_e	in	0.72	Computed as d_e = 0.72"
C'	in	0.25	$C' = P_T - D_O$
A_S	ft^2	0.413	$A_S = d_S C'B/(P_T \times 144)$
G_S	Kg/(hr m^2)	55,030	$G_S = m_{steam}/A_S$
Re_s		22,152	$Re = G_S \times d_e/\mu_s$
f	ft^2/in^2	0.0015	See Item 13:
ΔP_S	bar	0.08	ΔP_S = 1.16 psi

Note that the shell ΔP is usually much lower than the tube value.

Class Exercise 2 - Solution

F. Summary of Step 1.			
Variable	**Units**	**Target**	**Actual**
U	Kcal/(hr m^2 °C)	490-970	444-1,033
A	m^2	23-46	51
ΔP_{tubes}	bar	0.8	0.55

The heat transfer surface is larger than necessary. In contrast, the pressure drop is much lower than its permitted value. Possible next steps include:

❶ Increase/decrease the shell diameter
❷ Increase/decrease the number of tubes
❸ Increase/decrease the number of passes
❹ Increase/decrease tube diameter

Summary

After reviewing the materials in this lecture, you should:

❶ Be familiar with the major types of available heat-exchange equipment, with particular emphasis on shell-and-tube heat exchangers.
❷ Know how to estimate overall heat transfer coefficients, including the effect of fouling.
❸ Be able to perform mechanical design of the most appropriate shell-and-tube heat exchanger to meet desired duty and pressure drops.

Standard Tube-sheet Layouts

Item 4. Standard Tube Sheet Layouts (Kern, 1950, pp. 841-842)

Shell I.D. (in)	One-Pass □ Pitch	One-Pass △ Pitch	Two-Pass □ Pitch	Two-Pass △ Pitch	Four-Pass □ Pitch	Four-Pass △ Pitch	Six-Pass □ Pitch	Six-Pass △ Pitch
	1-in O.D. Tubes on 1¼-in Pitch							
8	21	21	16	16	14	16		14
12	48	55	45	52	40	48	38	46
15¼	81	91	76	86	68	80	68	74
17¼	112	131	112	118	96	106	90	104
19¼	138	163	132	152	128	140	122	136
21¼	177	199	166	188	158	170	152	164
25	260	294	252	282	238	256	226	252
31	406	472	394	454	380	430	368	424
37	596	674	574	664	562	632	544	614

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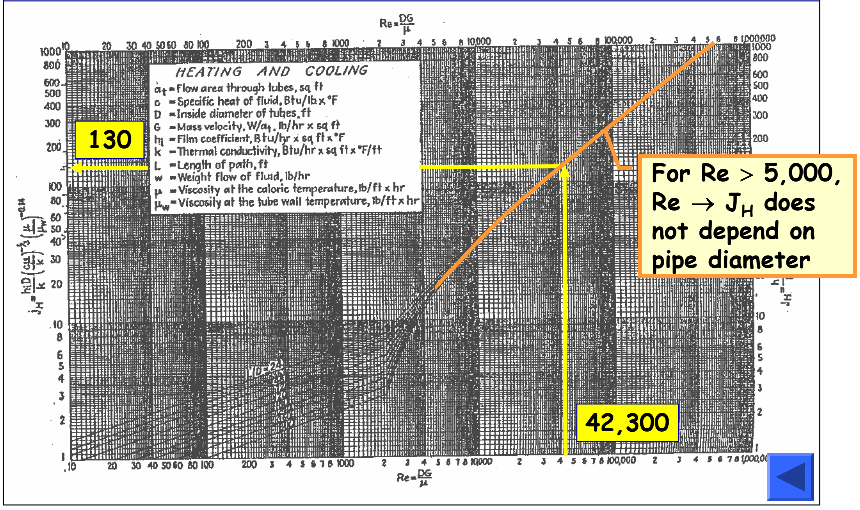
Item 10. Tube-side Heat Transfer



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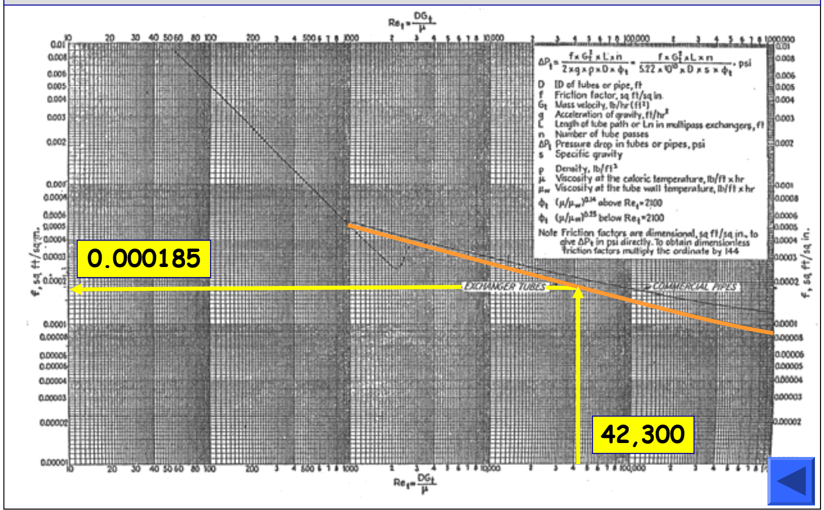
Item 11. Tube-side Friction Factor



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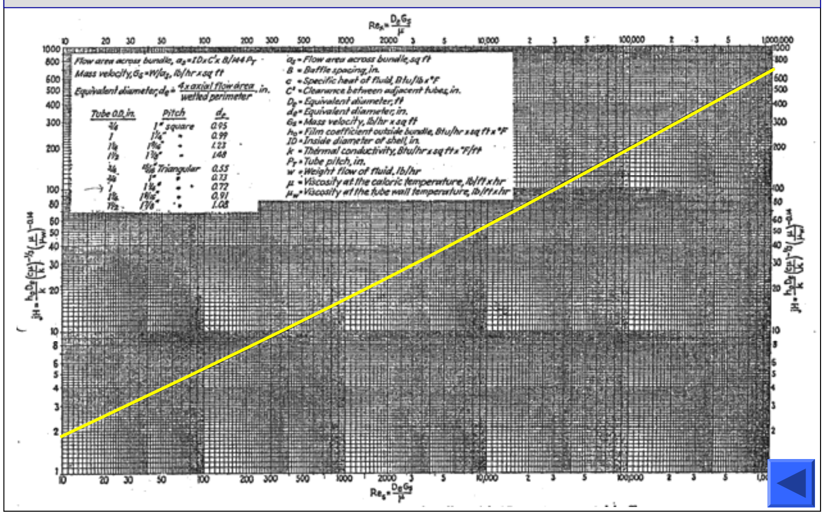
Item 12. Shell-side Heat Transfer



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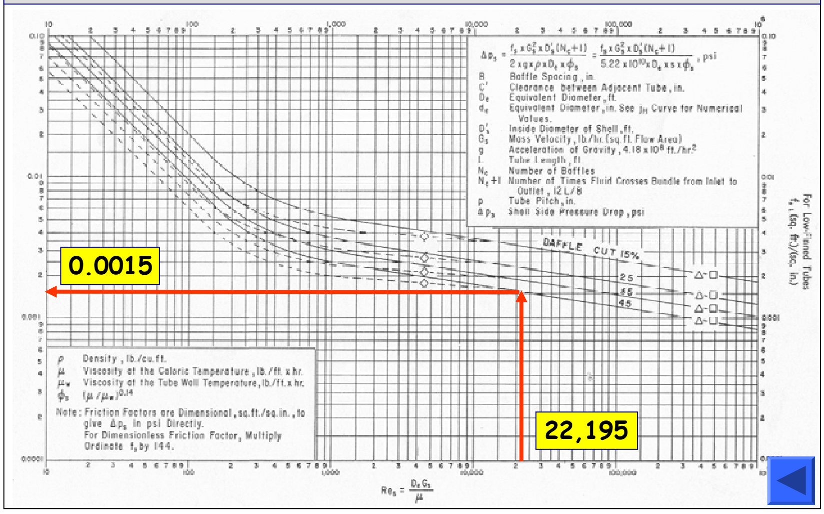
Item 13. Shell-side Friction Factor



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