Foundation of Technical Education

College of Technical/ Basrah

4 Lectures

FUNDAMENTALS & DESIGN OF HEAT EXCHANGER

1. Classification of Heat Exchangers
2. Calculations of Heat Exchanger
3. Heat Transfer Applications
4. Construction of Shell-And-Tube Heat Exchangers

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HEAT EXCHANGER

1 PREFACE

A Heat Exchanger: heat energy is transferred from one body or fluid stream to another. heat transfer equations are applied to calculate this transfer of energy so as to carry it out efficiently and under controlled conditions. The equipment goes under many names, such as boilers, pasteurizers, jacketed pans, freezers, air heaters, cookers, ovens and so on. The purpose of the heat exchanger is :

1. To heat or cool a stream flowering from equipment to another.
2. To vaporize a liquid stream
3. To condensate a vapor stream.

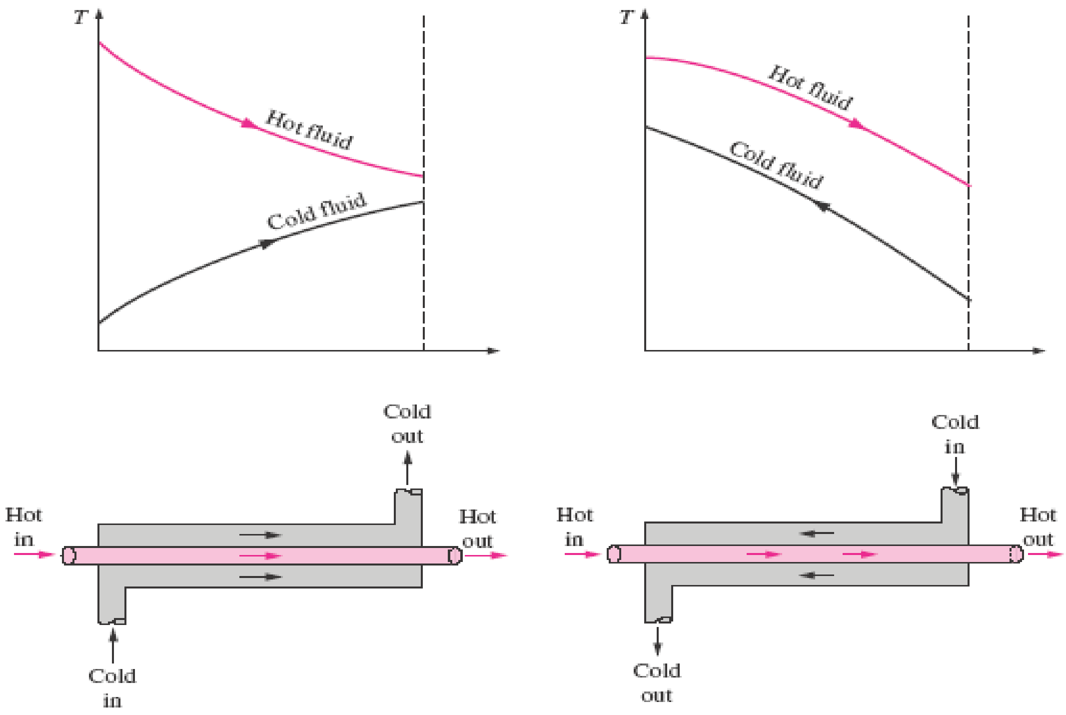
2 CLASSIFICATION OF HEAT EXCHANGERS:

2.1. TYPES OF APPLICATION

a. Boilers and Steam Generators
b. Condensers
c. Radiators
d. Evaporators
e. Cooling towers (direct contact)
f. Regenerators (periodic heat flow hot and cold fluid alternately occupy the space of the heat exchanger)
g. Recuperates (continuous heat flow hot and cold fluid are separated by a wall–shell and tube heat exchangers)

2.2 FLUID FLOW ARRANGEMENT

a) Co-current or parallel flow : The fluids can flow in the same direction through the equipment
b) Countercurrent flow: The fluids can flow in the opposite directions through the equipment
c) Cross flow: they can flow at right angles to each other.



(a) Parallel flow (b) Counter flow

Fig.(1) *Double Pipe Heat Exchanger*

2.3 Mixed and Unmixed fluid:

a. Both fluids unmixed
b. One fluid mixed another unmixed
c. Both fluids mixed

Fig.(2) *Mixed and Unmixed fluid*

3 CONTINUOUS-FLOW HEAT EXCHANGERS

It is very often convenient to use heat exchangers in which one or both of the materials that are exchanging heat are fluids, One of the fluids is usually passed through pipes or tubes, and the other fluid stream is passed round or across these. Most actual heat exchangers of this type have a mixed flow pattern, but it is often possible to treat them from the point of view of the predominant flow pattern.

3.1 DOUBLE-PIPE HEAT EXCHANGER

A double-pipe heat exchanger is constructed from two pipes, one inside the other. **First fluid** flows inside the inner pipe while the **second fluid** flows in the annular space between the two pipes. To obtain larger heat exchange area, several pipes are arranged side-by-side and fittings are attached to allow the fluids to contact the pipes in series.

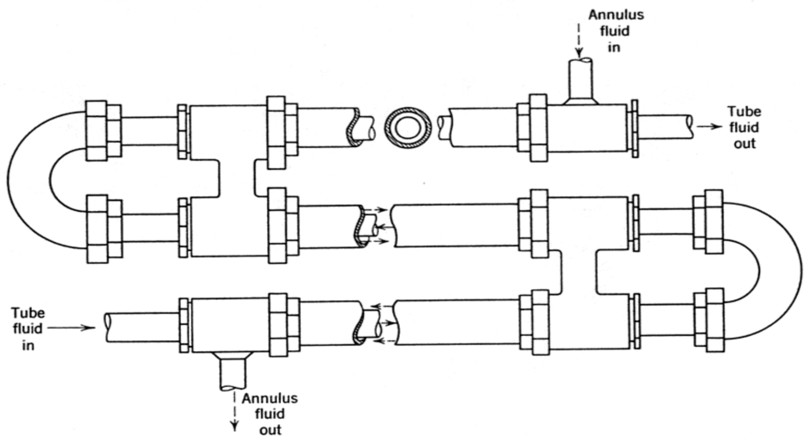


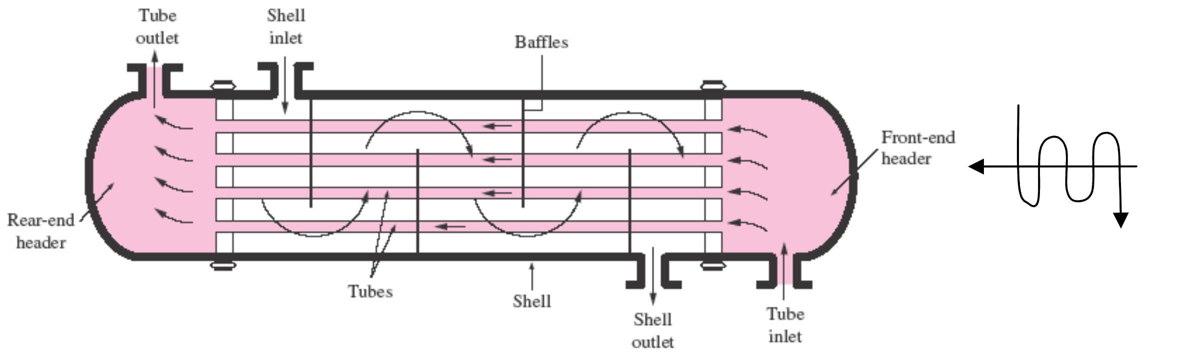
Fig.(3) *Double-Pipe Heat exchanger*

3.2 SHELL-AND-TUBE HEAT EXCHANGER

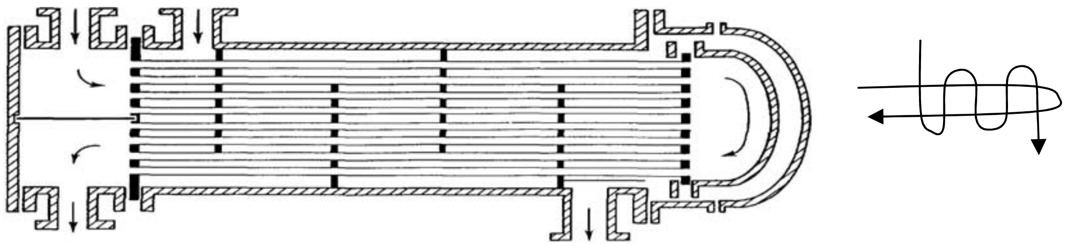
If very large heat exchange areas are required. Shown below is a bundle of small-diameter tubes which are arranged parallel to each other and reside inside a much larger-diameter tube called the "shell", much like strands of uncooked *Spagetti* come in a tube-shaped container. The tubes are all manifolded together at either end so

1-1 Heat Exchanger that the "tube fluid" enters the left side and is distributed equally among all the tubes. At the right side, the fluid exits from each tube, is mixed together in a second manifold, then leaves as a single stream. The second fluid flows in the space in between the outside of tubes. Baffle plates inside the shell force the shell fluid to flow across the tubes repeatedly as the fluid moves along the length of the shell.

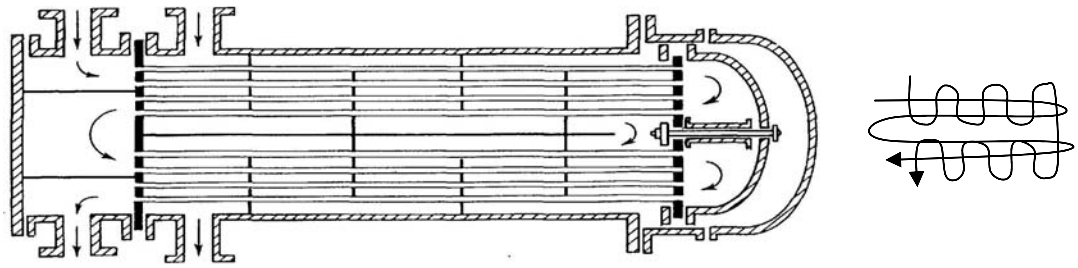
1-2 Heat Exchanger Half of the tubes have flow from left to right while the other half have flow in the opposite direction.



1 pass of shell-1 pass of tubes



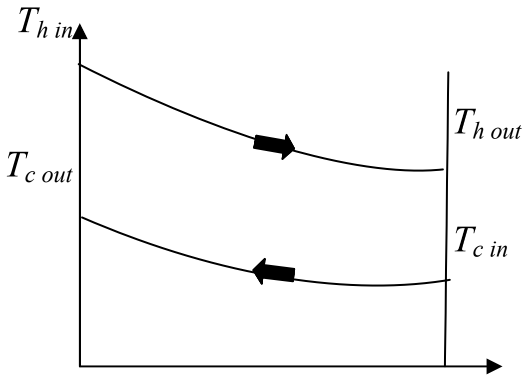
1 pass of shell-2 pass of tubes

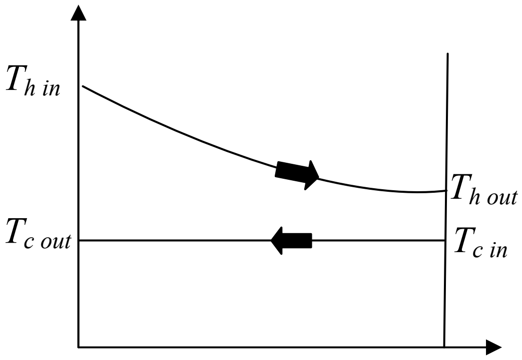


2 pass of shell-4 pass of tubes

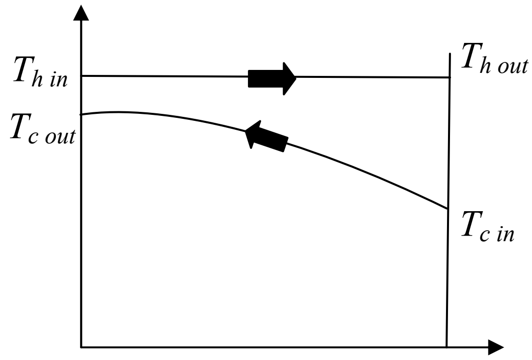
Fig.(4) *Different type of shell and tube Heat exchanger*

4 TYPES OF HEATING AND COOLING CURVES:

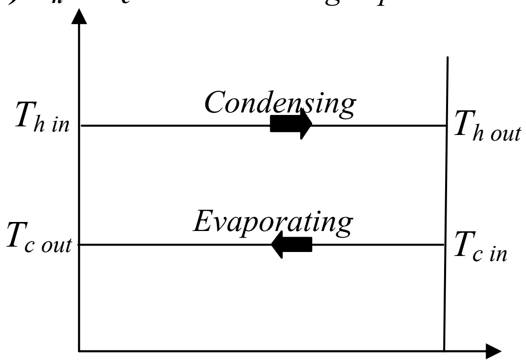




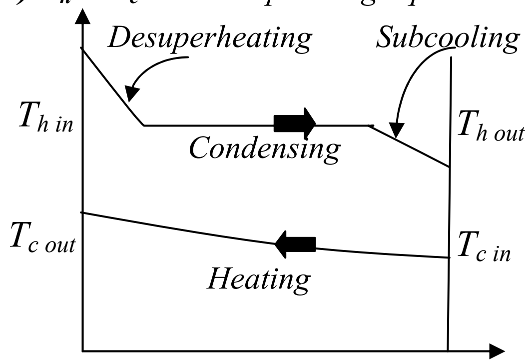
(c) ***C_h>>C_c*** *or a condensing vapor*



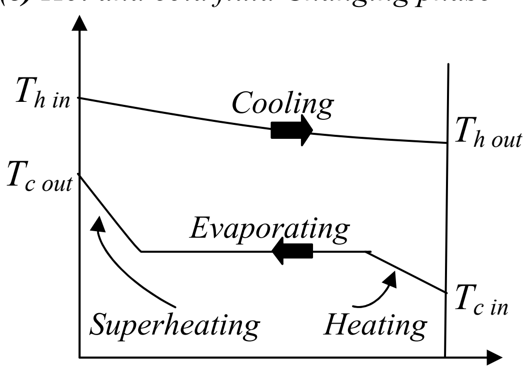
(d) ***C_h<<C_c*** *or an evaporating liquid*



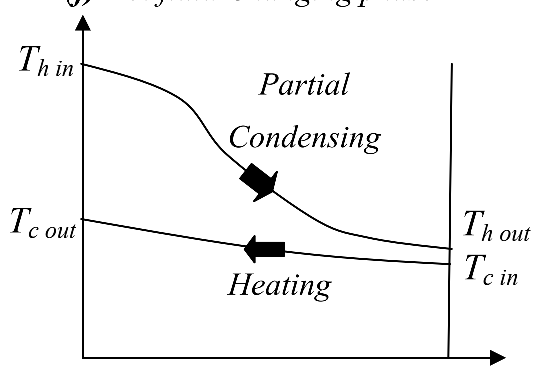
(e) *Hot and cold fluid Changing phase*



(f) *Hot fluid Changing phase*



(g) *cold fluid Changing phase*



(h) *condensable and noncondensable components*

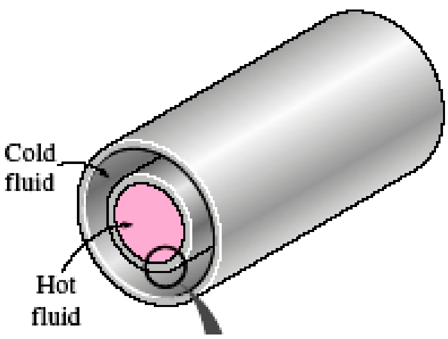
Fig(11) *Temperature distribution for a different* ***counter current*** *heat exchanger.*

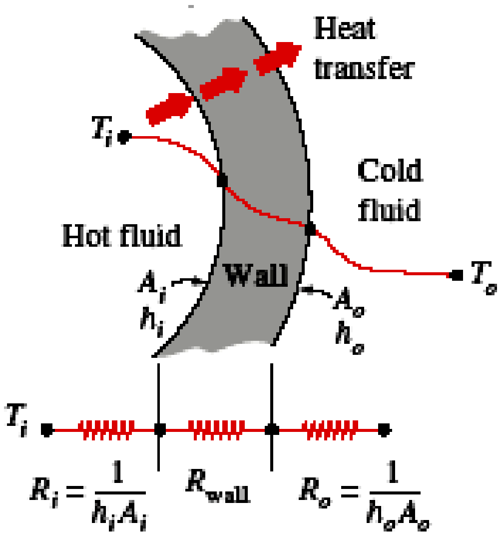
4 Heat Transfer Calculations

The primary objective in the thermal design of heat exchangers is to determine the necessary surface area required to transfer heat at a given rate for given fluid temperatures and flow rates. the fundamental heat transfer relation

$$q = UA\Delta T \qquad (1)$$

the overall heat transfer coefficient U is proportional to the reciprocal of the sum of the thermal resistances. For the common configurations which we shall encounter; cylindrical wall:





$$U_o = \frac{1}{\frac{1}{h_o} + \frac{r_o}{k}\ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_o}{r_i}\right)\frac{1}{h_i}} \qquad (2)$$

$$U_i = \frac{1}{\left(\frac{r_i}{r_o}\right)\frac{1}{h_o} + \frac{r_i}{k}\ln\left(\frac{r_o}{r_i}\right) + \frac{1}{h_i}} \qquad (3)$$

$$UA = U_oA_o = U_iA_i$$

where subscripts ***i*** and o represent the inside and outside surfaces of the wall, respectively, the overall heat transfer coefficient and the surface area must be compatible, i.e.,

$$q = U_oA_o\Delta T = U_iA_i\Delta T \qquad (1)$$

Table(1) gives approximate values of ***U*** for some commonly encountered fluids. The wide range of values cited results:

1. A diversity of heat exchanger materials (of different ***k** value)*
2. A flow conditions (influencing the film coefficients, h),
3. Geometric configuration.

Table(1) *Overall Heat Transfer Coefficient*

Hot fluid	Cold fluid	U (W/m^2°C)
Heat exchangers		
Water	Water	800–1500
Organic solvents	Organic solvents	100–300
Light oils	Light oils	100–400
Heavy oils	Heavy oils	50–300
Gases	Gases	10–50
Coolers		
Organic solvents	Water	250–750
Light oils	Water	350–900
Heavy oils	Water	60–300
Gases	Water	20–300
Organic solvents	Brine	150–500
Water	Brine	600–1200
Gases	Brine	15–250
Heaters		
Steam	Water	1500–4000
Steam	Organic solvents	500–1000
Steam	Light oils	300–900
Steam	Heavy oils	60–450
Steam	Gases	30–300
Dowtherm	Heavy oils	50–300
Dowtherm	Gases	20–200
Flue gases	Steam	30–100
Flue	Hydrocarbon vapours	30–100
Condensers		
Aqueous vapours	Water	1000–1500
Organic vapours	Water	700–1000
Organics (some non-condensables)	Water	500–700
Vacuum condensers	Water	200–500
Vaporisers		
Steam	Aqueous solutions	1000–1500
Steam	Light organics	900–1200
Steam	Heavy organics	600–900

Fouling Resistance

The performance of heat exchangers depends upon the heat transfer surfaces being clean and uncorrected. Should surface deposits be present, thermal resistance increases, resulting in decreased performance. This added resistance is usually accounted for by a ***fouling factor*** (*Fouling Resistance, R_f*) which must be included along with other thermal resistances when calculating the overall heat transfer coefficient.

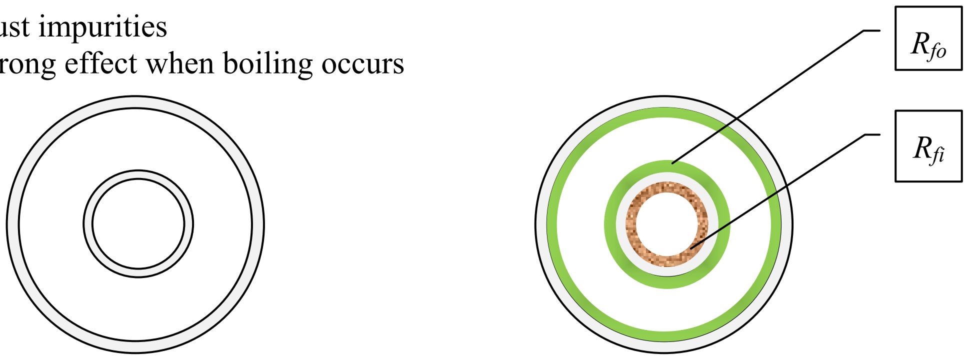
$$U_o = \frac{1}{\frac{1}{h_o} + R_{fo} + \frac{r_o}{k}\ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_o}{r_i}\right)R_{fi} + \left(\frac{r_o}{r_i}\right)\frac{1}{h_i}} \qquad (4)$$

$$U_i = \frac{1}{\left(\frac{r_i}{r_o}\right)\frac{1}{h_o} + \left(\frac{r_i}{r_o}\right)R_{fo} + \frac{r_i}{k}\ln\left(\frac{r_o}{r_i}\right) + R_{fi} + \frac{1}{h_i}} \qquad (5)$$

Typical values of R, ($m^2.K/W$) range from *0.00009* for clean vapors to *0.0002* for hot river water.

1. Material deposits on hot surfaces

2. Rust impurities
3. Strong effect when boiling occurs



Clean *After some time*

Fig.(5) *fouling resistance in double pipe heat exchanger*

5 Log-Mean Temperature Difference(LMTD Method)

5.1 Double-Pipe Heat Exchanger

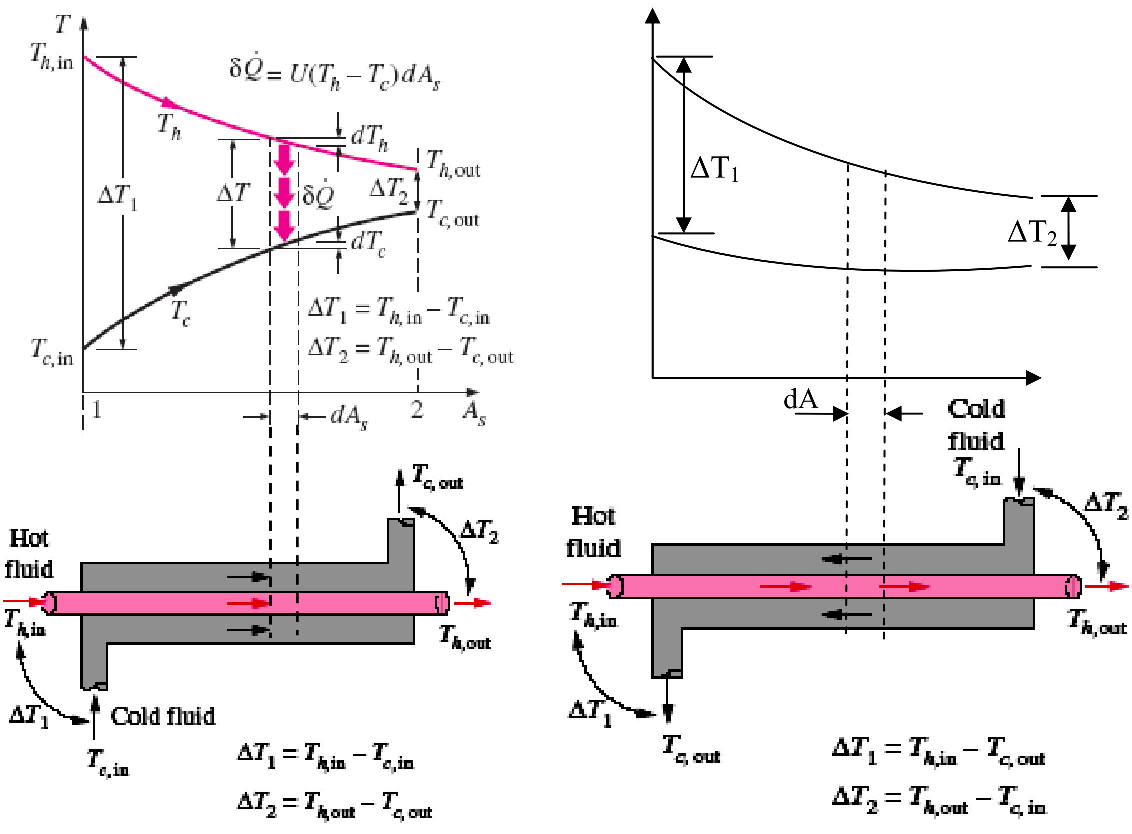
A parallel-flow flat-plate exchanger, whose temperature profiles are shown in *Fig.(2)* We shall assume that:

1. ***U*** is constant

2. heat exchange takes place only between the two fluids

3. the temperatures of both fluids are constant over a given cross-section

4. the specific heats of the fluids are constant



(*a*) Parallel-flow heat exchangers (*b*) Counter-flow heat exchangers

Fig.(6) *The ΔT_1 and ΔT_2 expressions in parallel-flow and counter-flow heat exchanger* the heat transfer between the hot and cold fluids for a differential length *dx* is

$$dq = UdA(T_h - T_c) \qquad (6)$$

since *dA* is the product of length *dx* and a constant width. The energy gained by the cold fluid is equal to that given up by the hot fluid,

$$dq = m_c C_c dT_c = -m_h C_h dT_h \qquad (7)$$

where *m* is the mass flow rate and *C* is the specific heat. Solving for the temperature differentials from equation (7),

$$dT_c = \frac{dq}{m_c C_c} \qquad dT_h = \frac{dq}{m_h C_h}$$

Taking Their difference, we get;

$$dT_h - dT_c = -dq\left(\frac{1}{m_h C_h} + \frac{1}{m_c C_c}\right) \qquad (8)$$

Eliminating dq between (6) and (8) yields

$$d(T_h - T_c) = -UA(T_h - T_c)\left(\frac{1}{m_h C_h} + \frac{1}{m_c C_c}\right) \qquad (9)$$

$$\int_1^2 \frac{d(T_h - T_c)}{(T_h - T_c)} = -UA\left(\frac{1}{m_h C_h} + \frac{1}{m_c C_c}\right)$$

which integrates to give

$$\ln\frac{\Delta T_2}{\Delta T_1} = -UA\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right) \qquad (10)$$

where the *ΔT* terms are as shown in *Fig. (5)*. And From an energy balance on each fluid,

$$\dot{m}_h c_h = \frac{q}{(T_{hi} - T_{ho})} \qquad \dot{m}_c c_c = \frac{q}{(T_{co} - T_{ci})}$$

and substitution of these expressions into (10) gives

$$\ln\frac{\Delta T_2}{\Delta T_1} = -UA\frac{(T_{hi} - T_{ho}) + (T_{co} - T_{ci})}{q}$$

or, in terms of the differences in end temperatures,

$$q = UA\frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2/\Delta T_1)} \qquad (11)$$

Upon comparing this result with eq.(1) , we see that

$$\Delta T = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} = \Delta T_{lm}$$

This average effective temperature difference is called the log-mean temperature difference (***LMTD***). It can easily be shown that the subscripts *1* and *2* may be interchanged without changing the value of ΔT_{lm}.

5.2 MULTIPASS AND CROSS-FLOW HEAT EXCHANGERS

For more complex heat exchangers, such as those involving **multiple tubes**, several **shell passes**, or **cross flow**, determination of the average effective temperature difference is so difficult that the usual practice is to modify (*1*) by a correction factor, giving

$$q = UAF\Delta T_{lm} \qquad (1.3)$$

Correction factors ***F*** for several common configurations are given in Fig.(6). In these figures the notation (T_1, T_2 , t_1, t_2) to denote the temperatures of the two fluid streams has been introduced, since it is immaterial whether the hot fluid flows through the shell or the tubes.

It is normally correlated as a function of two dimensionless temperature ratios:

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)} \qquad S = \frac{(t_2 - t_1)}{(T_1 - t_1)}$$

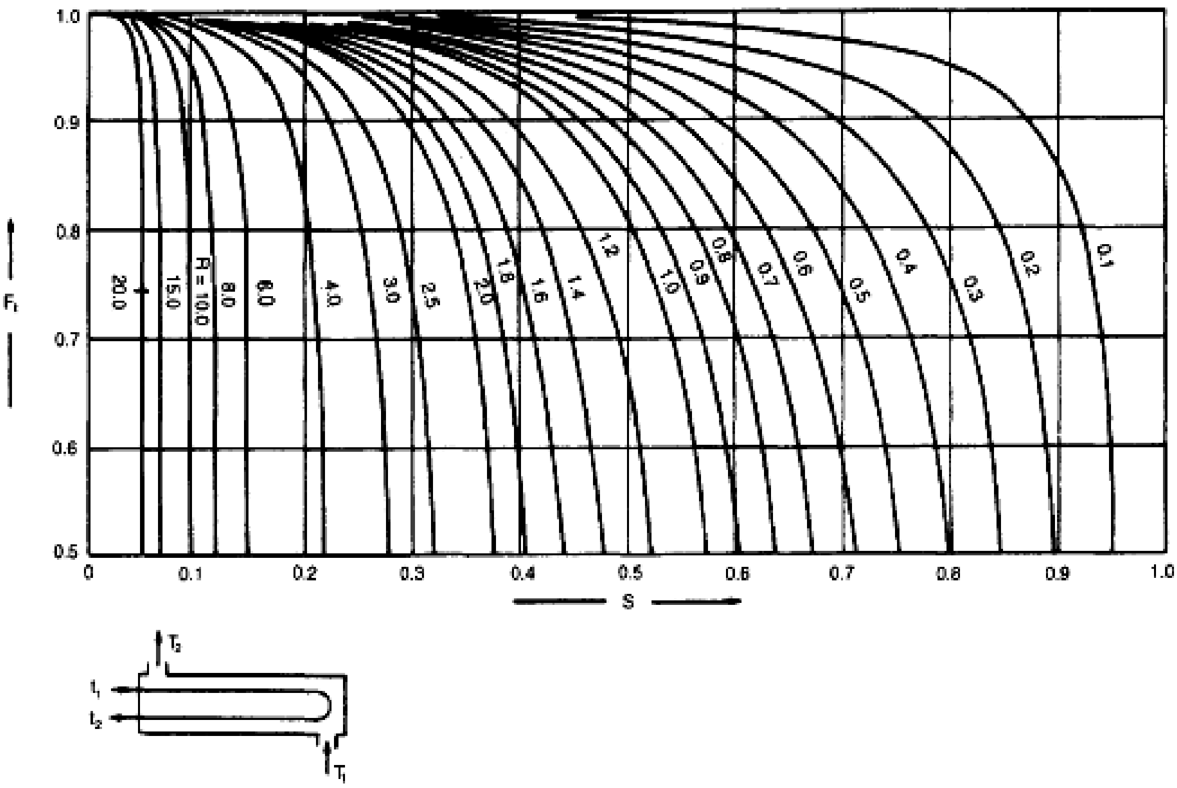


Figure 12.19. Temperature correction factor: one shell pass; two or more even tube 'passes

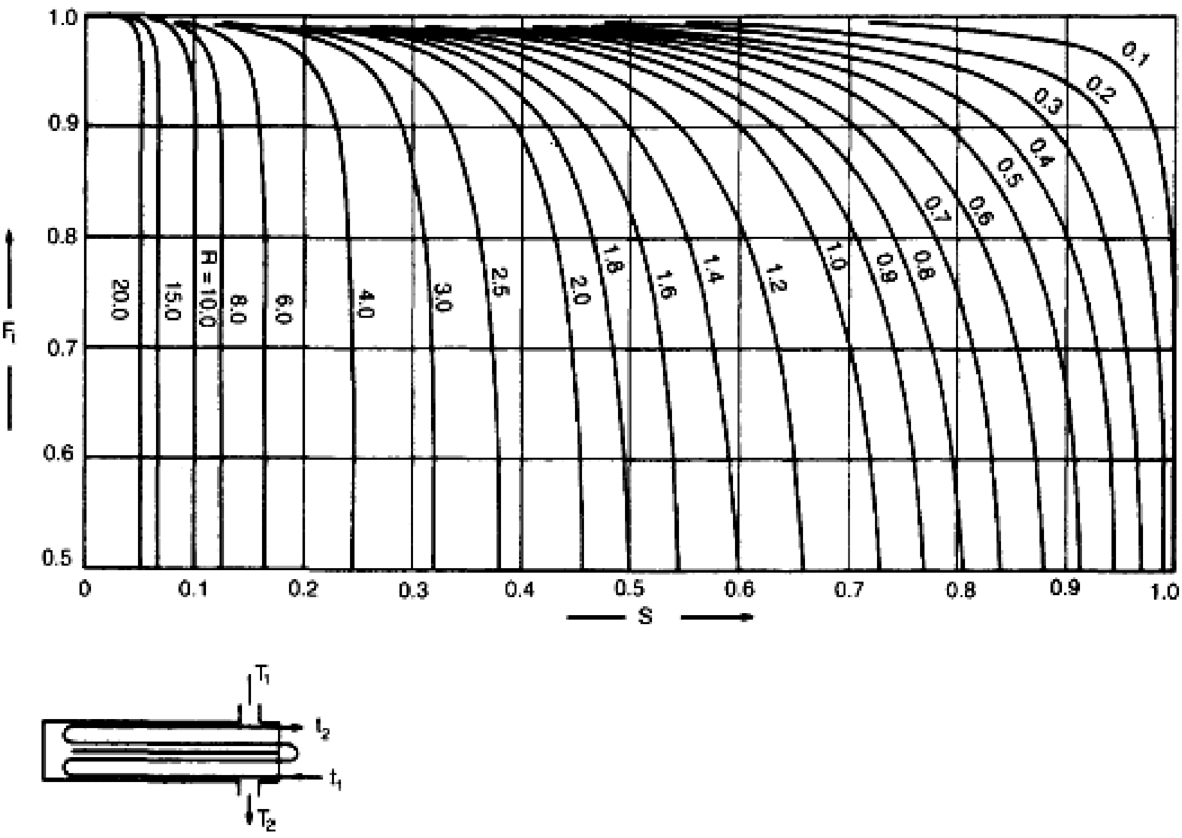


Figure 12.20. Temperature correction factor: two shell passes; four or multiples of four tube passes

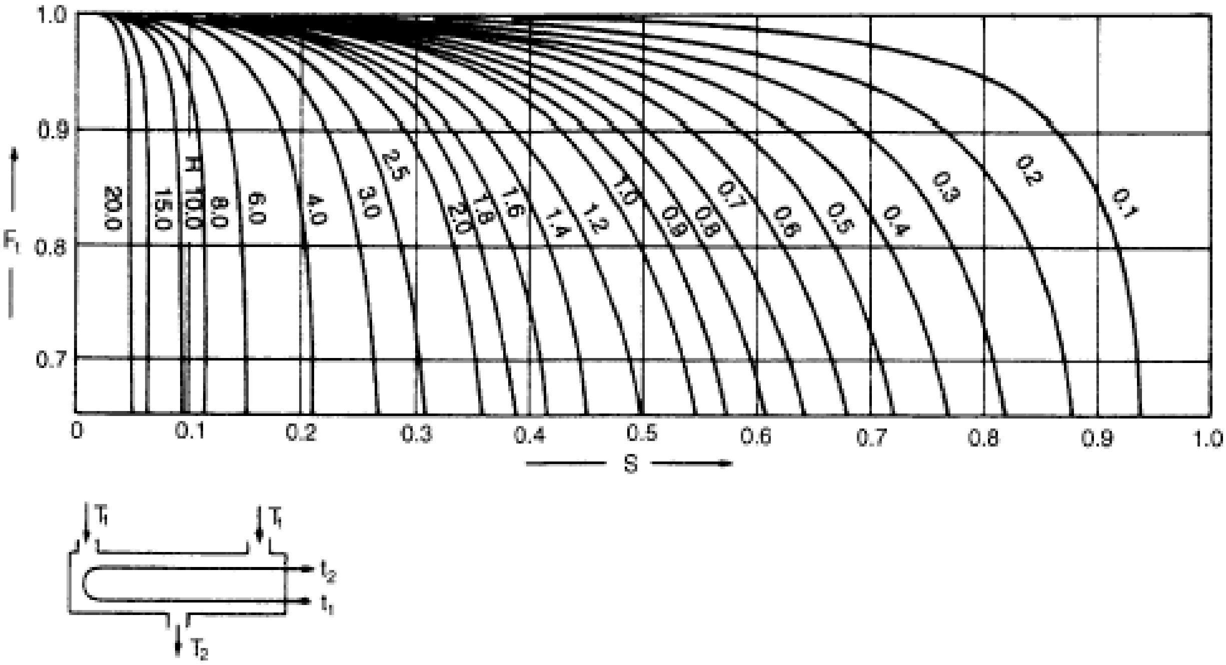


Figure 12.21. Temperature correction factor: divided-flow shell; two or more even-tube passes

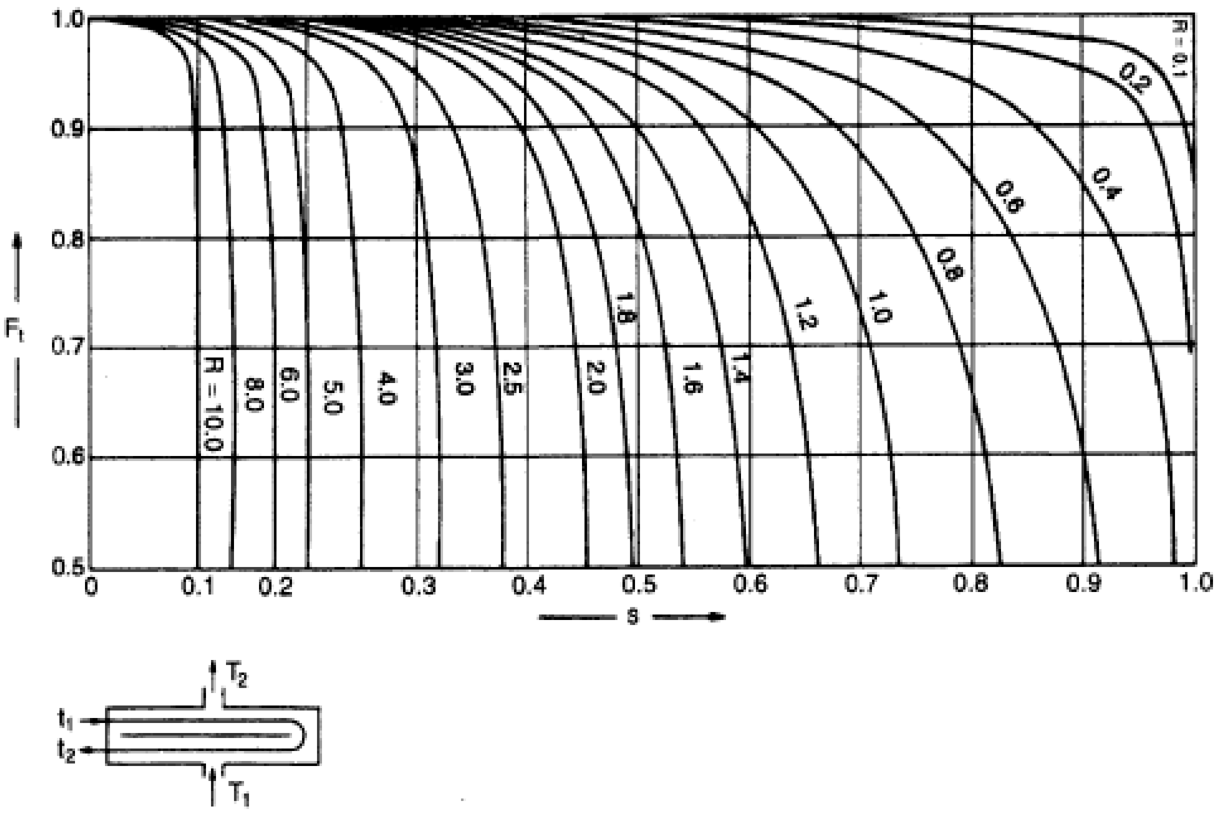


Figure 12.22. Temperature correction factor, split flow shell, 2 tube pass

EXAMPLE 1

In a counter flow heat exchanger, water is being chilled by a sodium chloride brine. If the rate of flow of the brine is 1.8 kg/s and that of the water is *1.05 kg/s* , estimate the temperature to which the water is cooled if the brine enters at *-8°C* and leaves at *10°C*, and if the water enters the exchanger at *32°C*. If the area of the heat-transfer surface of this exchanger is $55\ m^2$, what is the overall heat-transfer coefficient? Take the specific heats to be *3.38* and *4.18 kJ/kg °C* for the brine and the water respectively.

Solution

By heat balance, heat loss in brine = heat gain in water

$$1.8 \times 3.38 \times [10 - (-8)] = 1.05 \times 4.18 \times (32 - T_{w2})$$

$$T_{w2} = 7°C.$$

for counterflow

$$\Delta T_1 = [32 - 10] = 22°C \text{ and } \Delta T_2 = [7 - (-8)] = 15°C.$$

$$\Delta T_m = (22 - 15)/\ln(22/15) = 18.3°C.$$

$$q = UA\Delta T_m$$

Therefore $3.38 \times 1.8 \times 18 = U \times 55 \times 18.3$

$$\underline{U = 0.11\ kJ/m^2\ °C}$$

EXAMPLE 2

Determine the heat transfer surface area and length required for a heat exchanger constructed from a 0.0254m OD tube to cool 6.93 kg/s of a 95% ethyl alcohol solution (c_p = 3810 J/kg K) from 65.6°C to 39.4°C, using 6.30 kg/s of water available at 10°C (c_p = 4187 J/kg K). Assume that the overall coefficient of heat transfer based on the outer-tube area is 568 W/m^2 K and consider each of the following arrangements:

(a) Parallel-flow tube and shell

(b) Counterflow tube and shell

(c) Counterflow exchanger with 2 shell passes and 72 tube passes, the alcohol flowing through the shell and the water flowing through the tubes

(d) Cross-flow, with one tube pass and one shell pass, shell-side fluid mixed

SOLUTION

(a) Writing the energy balance as

$$m_h c_{ph} (T_{h,in} - T_{h\ out}) = m_c c_{pc} (T_{c,out} - T_{c,\ in})$$

we obtain

$$(6.93)(3810)(65.6 - 39.4) = (6.30)(4187)(T_{c,out} - 10)$$

$$T_{c,\ out} = 36.2°C$$

The rate of heat flow from the alcohol to the water is

$$q = m_h c_{ph} (T_{h,in} - T_{h,out})$$

$$q = (6.93\ kg/s)(3810\ J/kg\ K)(65.6 - 39.4)(K)$$

$$q = 691{,}800\ W$$

$$\Delta T_{lm} = \frac{\Delta T_2 - \Delta T_1}{\ln(\Delta T_2 / \Delta T_1)} = \frac{55.6 - 3.2}{\ln(55.6/3.2)} = 18.4°C$$

From the heat transfer surface area is

$$A = \frac{q}{(U)(\text{LMTD})} = \frac{(691{,}800\ \text{W})}{(568\ \text{W/m}^2\ \text{K})(18.4\ \text{K})} = 66.2\ \text{m}^2$$

0.0254m OD tube
L=A/πD = 830 m

(b) For the counterflow arrangement, LMTD=65.6-36.2 = 29.4°C, because $m_c c_{pc} = m_h c_{ph}$. The required area is

$$A = \frac{q}{(U)(\text{LMTD})} = \frac{691{,}800}{(568)(29.4)} = 41.4\ \text{m}^2$$

which is about *40%* less than the area necessary for parallel flow.

(c) For the counterflow arrangement, we determine the appropriate mean temperature difference by applying the correction factor found from ***Fig.12.20*** to the mean temperature for counterflow:

$$S = \frac{(t_2 - t_1)}{(T_1 - t_1)}$$

and the heat capacity rate ratio is

$$R = \frac{(T_1 - T_2)}{(t_2 - t_1)}$$

From the chart of Fig. 14, $F = 0.97$ and the heat transfer area is

$$A = \frac{41.4}{0.97} = 42.7\ \text{m}^2$$

The length of the exchanger for seventy-two 0.0254 m OD tubes in parallel would be

$$L = \frac{A/72}{\pi D} = \frac{42.7/72}{\pi(0.0254)} = 7.4\ \text{m}$$

(d) For the cross-flow arrangement ***Fig.(2)b*** , the correction factor is found from the chart of *Fig. 8.15* to be *0.88*. The required surface area is thus *47.0* m^2, about *10%* larger than that for the reversed-current exchanger.

6 HEAT EXCHANGER EFFECTIVENESS (NTU METHOD)

Another approach introduces a definition of heat exchanger effectiveness ε :

$$\varepsilon = \frac{q}{q_{\max}} \qquad (13)$$

Where: $0 \le \varepsilon \le 1$ and $\varepsilon = 0$ (evaporation & condensation)

q : Actual heat transfer

q_{max}: maximum possible heat transfer is that which would result if one fluid underwent a temperature change equal to the maximum temperature difference (T_{hi} - T_{ci})

This method uses the effectiveness ε to eliminate the unknown ***discharge temperature*** and gives a solution for effectiveness in terms of other known parameters (*m, C, A, and U*). Letting $C = mc$,

$$q = C_h(T_{hi} - T_{ho}) = C_c(T_{co} - T_{ci}) \qquad (15)$$

The maximum possible heat transfer occurs when the fluid of smaller C undergoes the maximum temperature difference available,

$$q_{max} = C_{min}(T_{hi} - T_{ci}) \qquad (16)$$

This transfer would be attained in a counterflow exchanger of infinite area. Combining (14) and (16),we get the basic equation for determining the heat transfer in heat exchangers with unknown discharge temperatures:

$$q_{actual} = \varepsilon\, C_{mim}(T_{hi} - T_{ci}) \qquad (17)$$

6.1 PARALLEL-FLOW HEAT EXCHANGER

Consider the simple parallel-flow heat exchanger of *Fig.(5)* under the same assumptions used in Section 5.1 to determine the log-mean temperature difference. Combining (13), (14) and (I5) , we get two expressions for effectiveness,

$$\varepsilon = \frac{C_h(T_{hi} - T_{ho})}{C_{min}(T_{hi} - T_{ci})} = \frac{C_c(T_{co} - T_{ci})}{C_{min}(T_{hi} - T_{ci})} \qquad (18)$$

Since either the hot or the cold fluid may have the minimum value of C, there are two possible values of effectiveness:

$C_h < C_c$ evaporation $\qquad \varepsilon_h = \frac{T_{hi} - T_{ho}}{T_{hi} - T_{ci}} \qquad (19)$

$C_h > C_c$ condensation $\qquad \varepsilon_c = \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}} \qquad (20)$

where subscripts on ε designate the fluid which has the minimum C. Returning to (9), it may be written in terms of the Cs to give

$$\ln \frac{T_{ho} - T_{co}}{T_{hi} - T_{ci}} = -UA\left(\frac{1}{C_h} + \frac{1}{C_c}\right)$$

$$\frac{T_{ho} - T_{co}}{T_{hi} - T_{ci}} = \exp\left(-\frac{UA}{C_h}\left(1 + \frac{C_h}{C_c}\right)\right)$$

From the energy balance equation (15),

$$T_{co} = T_{ci} + C_h / C_c\,(T_{hi} - T_{ho}) \qquad (22)$$

Substituting this relation in to eq. (21) after adding and subtracting T_{hi} gives

$$\frac{T_{ho}-T_{hi}+T_{hi}-(T_{ci})+C_h/C_c\ (T_{hi}\text{-}T_{ho}))}{T_{hi}\text{-}T_{ci}}=\exp\left[-\frac{UA}{C_h}\left(1+\frac{C_h}{C_c}\right)\right]$$

Which simplified to:

$$\frac{T_{hi}\text{-}T_{ci}}{T_{hi}\text{-}T_{ci}}-\frac{T_{hi}-T_{ho}+C_h/C_c\ (T_{hi}\text{-}T_{ho})}{T_{hi}\text{-}T_{ci}}=\exp\left[-\frac{UA}{C_h}\left(1+\frac{C_h}{C_c}\right)\right]$$

$$\frac{(T_{hi}-T_{ho})+C_h/C_c\ (T_{hi}\text{-}T_{ho})}{T_{hi}\text{-}T_{ci}}=1-\exp\left[-\frac{UA}{C_h}\left(1+\frac{C_h}{C_c}\right)\right]$$

$$\frac{(T_{hi}-T_{ho})}{T_{hi}\text{-}T_{ci}}=\frac{1-\exp\left[-\frac{UA}{C_h}\left(1+\frac{C_h}{C_c}\right)\right]}{(1+C_h/C_c)}$$

Substituting this relation in to equation (*19*)

$$\epsilon_h=\frac{1-\exp[-(UA/C_h)(1+C_h/C_c)]}{1+C_h/C_c} \qquad (23)$$

similarity in to eq.(*20*) we get,

$$\epsilon_c=\frac{1-\exp[-(UA/C_c)(1+C_c/C_h)]}{1+C_c/C_h} \qquad (24)$$

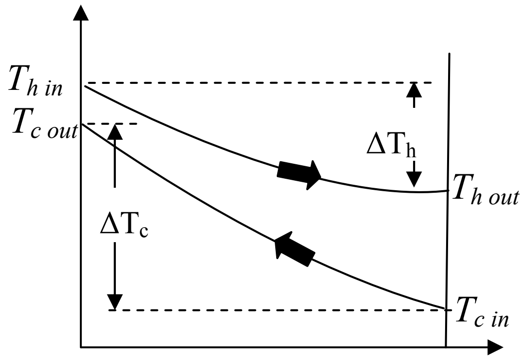
Equations (23) and (24) may both be expressed as

$$\epsilon=\frac{1-\exp[-(UA/C_{\min})(1+C_{\min}/C_{\max})]}{1+C_{\min}/C_{\max}} \qquad (25)$$

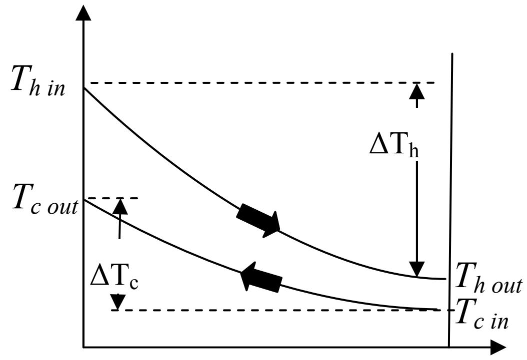
It should be noted that (25) contains only the overall heat transfer coefficient, area, fluid properties, and flow rates.

Giving the effectiveness for a parallel-flow heat exchanger in terms of two dimensionless ratios *(UA/C_{min}) and (C_{min}/C_{max})*, a *UA/C_{min}*, is called the ***Number of Transfer Units*** may be considered as a heat exchanger size factor,

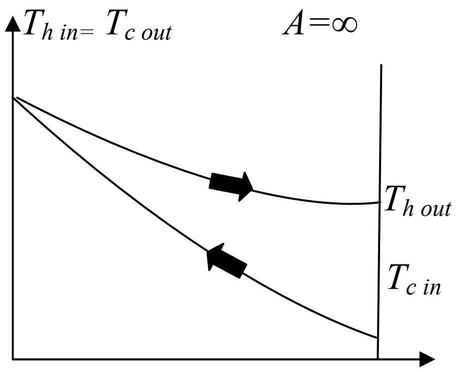
$$NTU = UA/C_{min} \qquad (26)$$



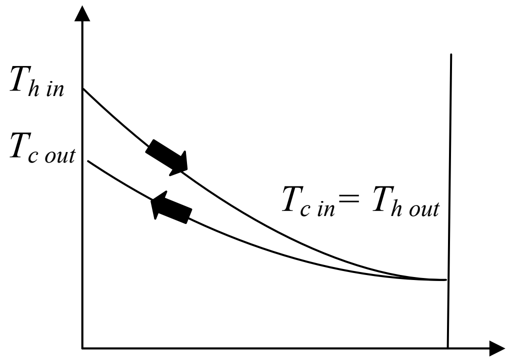
$(mC_p)_{hot} > (mC_p)_{cold}$
$C_{min} = (mC_p)_{cold}$
$\Delta T_c > \Delta T_h$



$(mC_p)_{cold} > (mC_p)_{hot}$
$C_{min} = (mC_p)_{hot}$
$\Delta T_h > \Delta T_c$



$A=\infty$
$q=(mC_p)_{cold}\ (T_{hi}-T_{ci})$



$A=\infty$
$q=(mC_p)_{hot}\ (T_{hi}-T_{ci})$

Table (2) *Expressions for the effectiveness of other configurations* where ***C=C_{min}/C_{max}***

Heat exchanger type	Effectiveness relation
1 *Double pipe:*	
Parallel-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 + c)]}{1 + c}$
Counter-flow	$\varepsilon = \dfrac{1 - \exp[-NTU(1 - c)]}{1 - c\exp[-NTU(1 - c)]}$
2 *Shell and tube:*	
One-shell pass 2, 4, . . . tube passes	$\varepsilon = 2\left\{1 + c + \sqrt{1 + c^2}\dfrac{1 + \exp[-NTU\sqrt{1 + c^2}]}{1 - \exp[-NTU\sqrt{1 + c^2}]}\right\}^{-1}$
3 *Cross-flow (single-pass)*	
Both fluids unmixed	$\varepsilon = 1 - \exp\left\{\dfrac{NTU^{0.22}}{c}[\exp(-c\ NTU^{0.78}) - 1]\right\}$
C_{max} mixed, C_{min} unmixed	$\varepsilon = \dfrac{1}{c}(1 - \exp\{1 - c[1 - \exp(-NTU)]\})$
C_{min} mixed, C_{max} unmixed	$\varepsilon = 1 - \exp\left\{-\dfrac{1}{c}[1 - \exp(-c\ NTU)]\right\}$
4 *All heat exchangers with c = 0*	$\varepsilon = 1 - \exp(-NTU)$

Note that for an evaporator or condenser $C = 0$, because one fluid remains at a constant temperature, making its effective specific heat infinite.

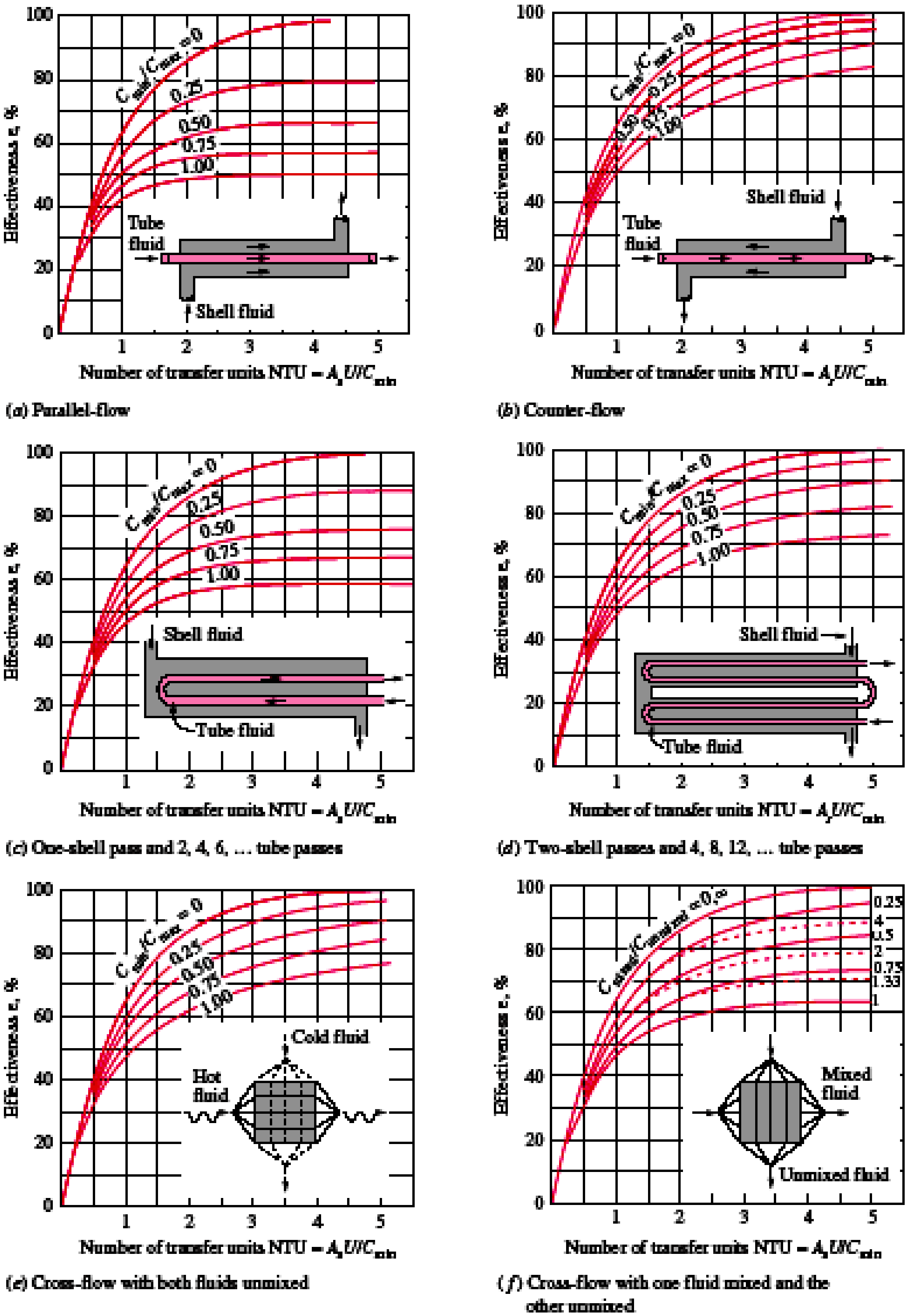

Effectiveness ε, %
Number of transfer units NTU = $A_s U/C_{min}$
C_{min}/C_{max} = 0
0.25
0.50
0.75
1.00
Tube fluid
Shell fluid
(a) Parallel-flow
(b) Counter-flow
(c) One-shell pass and 2, 4, 6, … tube passes
(d) Two-shell passes and 4, 8, 12, … tube passes
Cold fluid
Hot fluid
(e) Cross-flow with both fluids unmixed
$C_{mixed}/C_{unmixed}$ = 0,∞
Mixed fluid
Unmixed fluid
(f) Cross-flow with one fluid mixed and the other unmixed

Fig.(7)

Example 4

From a performance test on a well-baffled single-shell, two-tube-pass heat exchanger, the following data are available: oil *(c_p = 2100 J/kg K)* in turbulent flow inside the tubes entered at 340 K. at the rate of 1.00 kg/s and left at 310 K; water flowing on the shell side entered at 290 K and left at 300 K.

A change in service conditions requires the cooling of a similar oil from an initial temperature of 370 K but at three fourths of the flow rate used in the performance test. Estimate the outlet temperature of the oil for the same water flow rate and inlet temperature as before.

Solution

$$q_h = q_c$$
$$C_h(T_{hi} - T_{ho}) = C_c(T_{co} - T_{ci})$$
$$C_c = 6300 \ W/K$$

and the temperature ratio *P* is, from Eq. (8.19),

S=0.6 R=0.33 From Fig. **8.13.** F = 0.94

$$\Delta T_{mean} = (F)(\text{LMTD}) = (0.94)\frac{(340 - 300) - (310 - 290)}{\ln[(340 - 300)/(310 - 290)]} = 27.1 \text{ K}$$

the overall conductance is

$$UA = \frac{q}{\Delta T_{mean}} = \frac{(1.00 \text{ kg/s})(2100 \text{ J/kg K})(340 - 310)(\text{K})}{(27.1 \text{ K})} = 2325 \text{ W/K}$$

Since the thermal resistant on the oil side is controlling, a decrease in velocity to 75% of the original value will increase (he thermal resistance by roughly the velocity ratio raised to the 0.8 power.

Under the new conditions, the conductance, the NTU, and the heat capacity rate ratio will therefore be approximately

$$UA = (2325)(0.75)^{0.8} = 1850 \ W/K$$

$$NTU = UA/C_{oil} = 1.17$$

$$C_{min} = C_{oil} = 0.75 \times 1.00 \ kg/s)(2100 \ J/kg \ K)$$
$$C_{max} = C_{water} = 6300 \ W/K$$

$$C_{min}/ C_{max} = 0.25$$

from Fig. 8.19 the effectiveness is equal to 0.61. Hence

$$T_{oil\ out} = T_{oil\ in} - \varepsilon\, \Delta T_{max}. = 370 - [0.61(370 - 290)] = 321.2 \ K.$$

7 OTHER HEAT TRANSFER APPLICATIONS

7.1 JACKETED PANS

In a jacketed pan, the liquid to be heated is contained in a vessel, which may also be provided with an agitator to keep the liquid on the move across the heat-transfer surface, as shown in *Fig.3(a)*.

The source of heat is commonly steam condensing in the vessel jacket. Practical considerations of importance are:

1. There is the minimum of air with the steam in the jacket.

*2. The steam is not superheated as part of the surface must then be used as a de-*superheater over which low gas heat-transfer coefficients apply rather than high condensing coefficients.

3. Steam trapping to remove condensate and air is adequate.

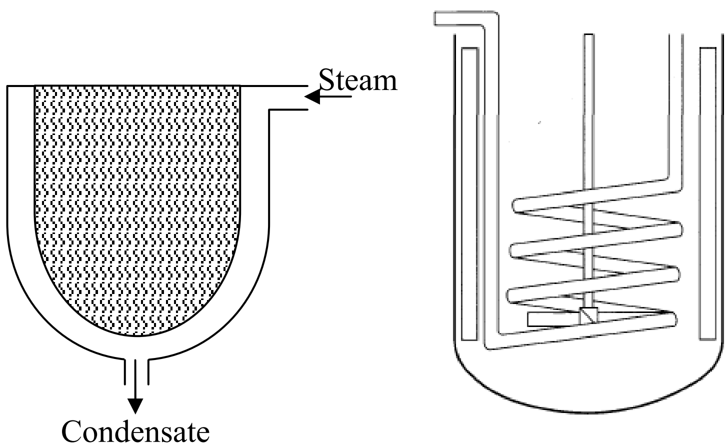
Some overall heat transfer coefficients are shown in *Table 3*. Save for boiling water, which agitates itself, mechanical agitation is assumed. Where there is no agitation, coefficients may be halved.

Table(3) Some Overall Heat Transfer Coefficients In Jacketed Pans

Condensing fluid	Heated fluid	Pan material	U (J/m^2 s °C)
Steam	Thin liquid	Cast-iron	1800
Steam	Thick liquid	Cast-iron	900
Steam	Paste	Stainless steel	300
Steam	Water, boiling	Copper	1800

7.2 HEATING COILS IMMERSED IN LIQUIDS

In some processes, quick heating is required in the pan, a helical coil may be fitted inside the pan and steam admitted to the coil as shown in *Fig.3(b)*. This can give greater heat transfer rates than jacketed pans, because there can be a greater heat transfer surface and also the heat transfer coefficients are higher for coils than for the pan walls.



***(a)** Jacketed pans* ***(b)** Heating Coil*

***Fig.(8)** Heat exchange equipment*

Example (3)

Milk is flowing into a pipe cooler and passes through a tube of *2.5 cm* internal diameter at a rate of *0.4 kg/s*. Its initial temperature is 49°C and it is wished to cool it to 18°C using a stirred bath of constant *10°C* water round the pipe. What length of pipe would be required? Assume an overall coefficient of heat transfer from the bath to the milk of $900\ J/m^2s\ °C$, and that the specific heat of milk is *3890 J/kg °C*.

Solution

$$q = m\ C_p\ (T_1 - T_2)$$

$$q = 3890 \times 0.4 \times (49 - 18) = 48240\ J/s$$

Also $q = UA\Delta T_m$

$$\Delta T_m = [(49 - 10) - (18 \quad 10)] / \ln[(49 - 10)1(18 - 10)]$$

$$\Delta T_m = 19.6°C.$$

Therefore $48,240 = 900 \times A \times 19.6$

$$A = 2.73\ m^2$$

but $A = \pi DL$

Now $D = 0.025$ m.

$L = 2.73/(\pi \times 0.025) =$ **34.8 m**

Example (4)

Steam required to heat soup in jacketed pan Estimate the steam requirement as you start to heat *50 kg* of soup in a jacketed pan, if the initial temperature of the soup is *18°C* and the steam used is at *100 kPa gauge*. The pan has a heating surface of 1 m^2 and the overall heat transfer coefficient is assumed to be $300\ J/m^2s°C$.

Solution

From steam tables, saturation temperature of steam at 100 kPa gauge = 120°C and latent heat = $\lambda = 2202\ kJ/kg$.

$$q = UA\ \Delta T$$

$$= 300 \times 1 \times (120 - 18)$$

$$= 3.06 \times 10^4\ J/s$$

Therefore amount of steam

$$= q/\lambda = (3.06 \times 10^4)/(2.202 \times 10^6)$$

$$= 1.4 \times 10^{-2}\ kg/s$$

$$= 1.4 \times 10^{-2} \times 3.6 \times 10^3$$

$$= 50\ kg/h.$$

7.3 Scraped Surface Heat Exchangers

The processing industry particularly for products of higher viscosity, consists of a jacketed cylinder with an internal cylinder concentric to the first and fitted with scraper blades, as illustrated in *Fig.(9).* The blades rotate, causing the fluid to flow through the annular space between the cylinders with the outer heat transfer surface constantly scraped. Coefficients of heat transfer vary with speeds of rotation but they are of the order of *900-4000* $J/m^2s°C$. These machines are used in the freezing of ice cream and in the cooling of fats during margarine manufacture.

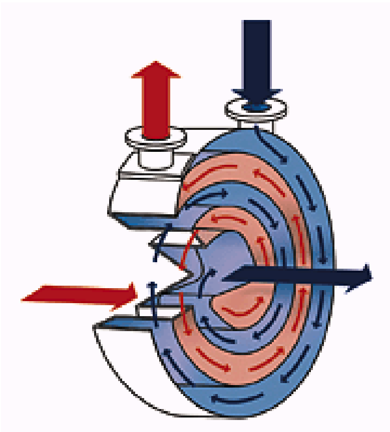
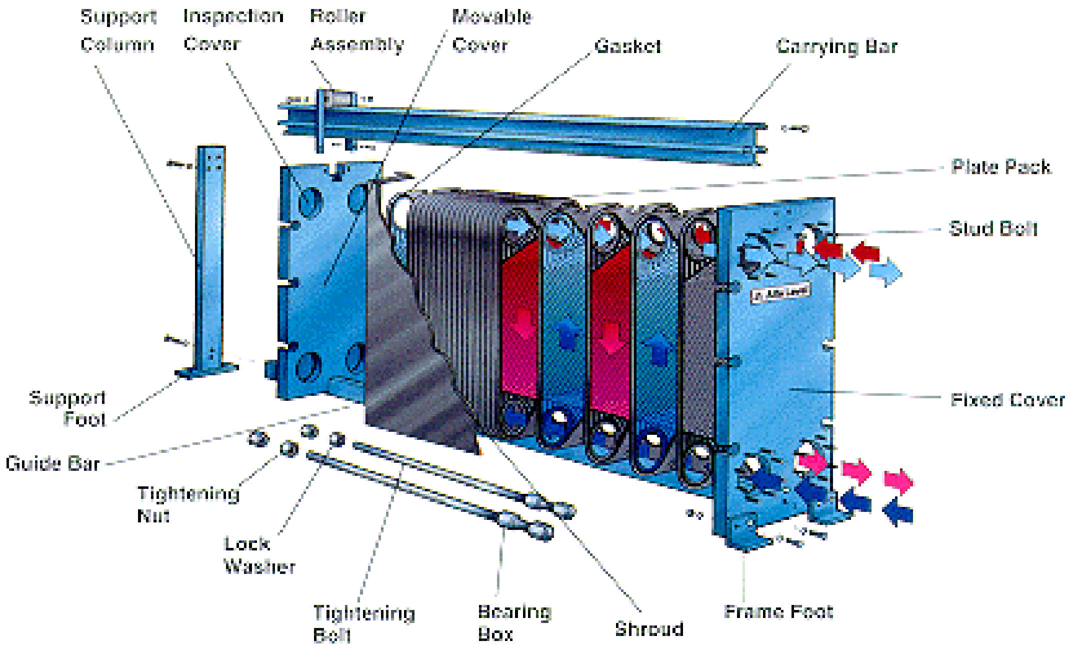



Fig.(9) *Heat exchange equipment*

7.4 Plate Heat Exchangers

A popular heat exchanger for fluids of low viscosity, is the plate heat exchanger, where heating and cooling fluids flow through alternate tortuous passages between vertical plates as illustrated in Fig.(10). The plates are clamped together, separated by spacing gaskets, and the heating and cooling fluids are arranged so that they flow between alternate plates. Suitable gaskets and channels control the flow and allow parallel or counter current flow in any desired number of passes. A substantial advantage of this type of heat exchanger is that it offers a large transfer surface that is readily accessible for cleaning. The banks of plates are arranged so that they may be taken apart easily. Overall heat transfer coefficients are of the order of *2400-6000* $J/m^2s°C$.



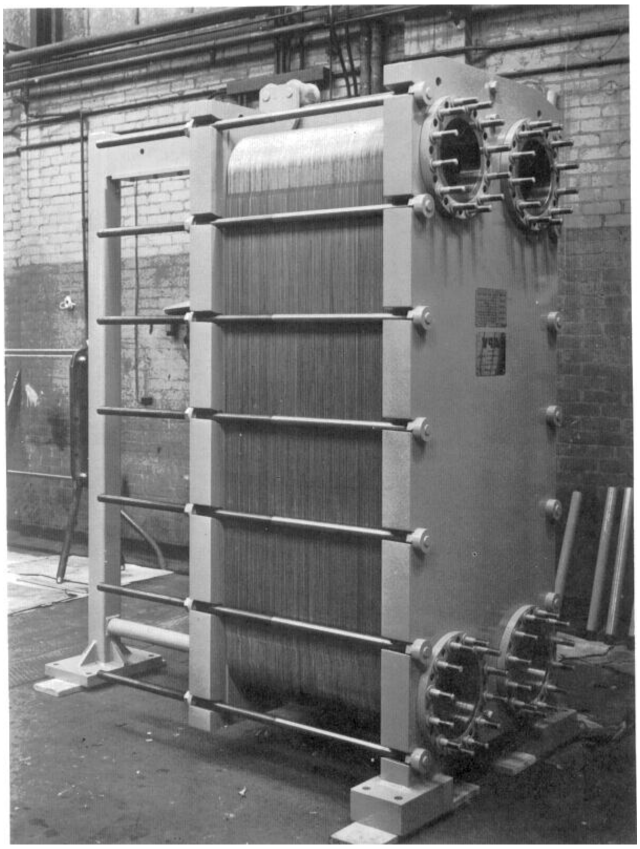


Fig.(10) *Plate Heat exchange*

8 A DESIGN METHODOLOGY

1. **Heat Exchanger Sizing** Choose a typical value for U based on the type of service, then determined the outlet temperatures based on the performance specifications (number of tube, baffle spacing, …etc) and the energy balance Q also calculate heat transfer area from $Q=UAF\Delta T_{lm}$.

2. **Heat Exchanger Rating** is the computational process in which the inlet flow rate and temperatures, the fluid properties and the heat exchanger parameter are taken as input. And the outlet temperatures and thermal duty **Q** (if the heat exchanger length is specified) on the required length is calculated as output else pressure drop of each stream. (U and A are known)

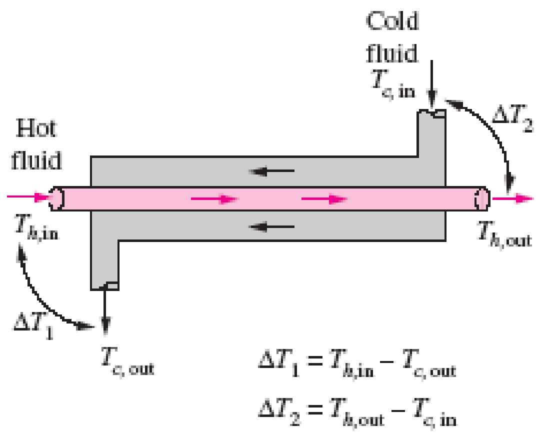
3. **Heat Exchanger Design** Is the process that determine the heat exchanger specifications such as (length and diameter of tube, shell thickness, spacing and cut baffle.)

- Determine heat transfer area based on sizing calculation
- Determine number of tube and pass
- Check the velocity if below acceptable range, choose suitable number of pass.
- Calculate overall heat transfer coefficient and estimate the fouling resistance.
- Check $|U_{assume} - U_{calculate}| < 0.001$
- Check the pressure drop for shell and tube sides , then determine the pumping power requirements for shell and tubes sides.

1. Kern Method

Kern Method is used to estimate the size of the heat exchanger for a given specification (***Sizing Method***), its restricted to a fixed baffecut (25%) and cannet adequately account for baffle-shell and tube-to-baffle leakages

Input (know)	**Output (unknown)**
T_{ci}	T_{ho}
T_{hi}	Q
T_{co}	d_i
m_h	d_o
m_c	



2. Bell Delaware's Method

It's a rating analysis and give more satisfactory predications of the heat transfer coefficient and pressure drop than ***Kern Method***, and it is takes into account the effects of leakage and passing.

Data of Kern Method → Data of Bell Method

3. NTU Method

It the heat duty is not known because only the inlet temperatures are given while the outlet temperatures are not. On the other hand the heat exchanger length is fixed and the outlet temperatures and pressure drops are to be calculated.

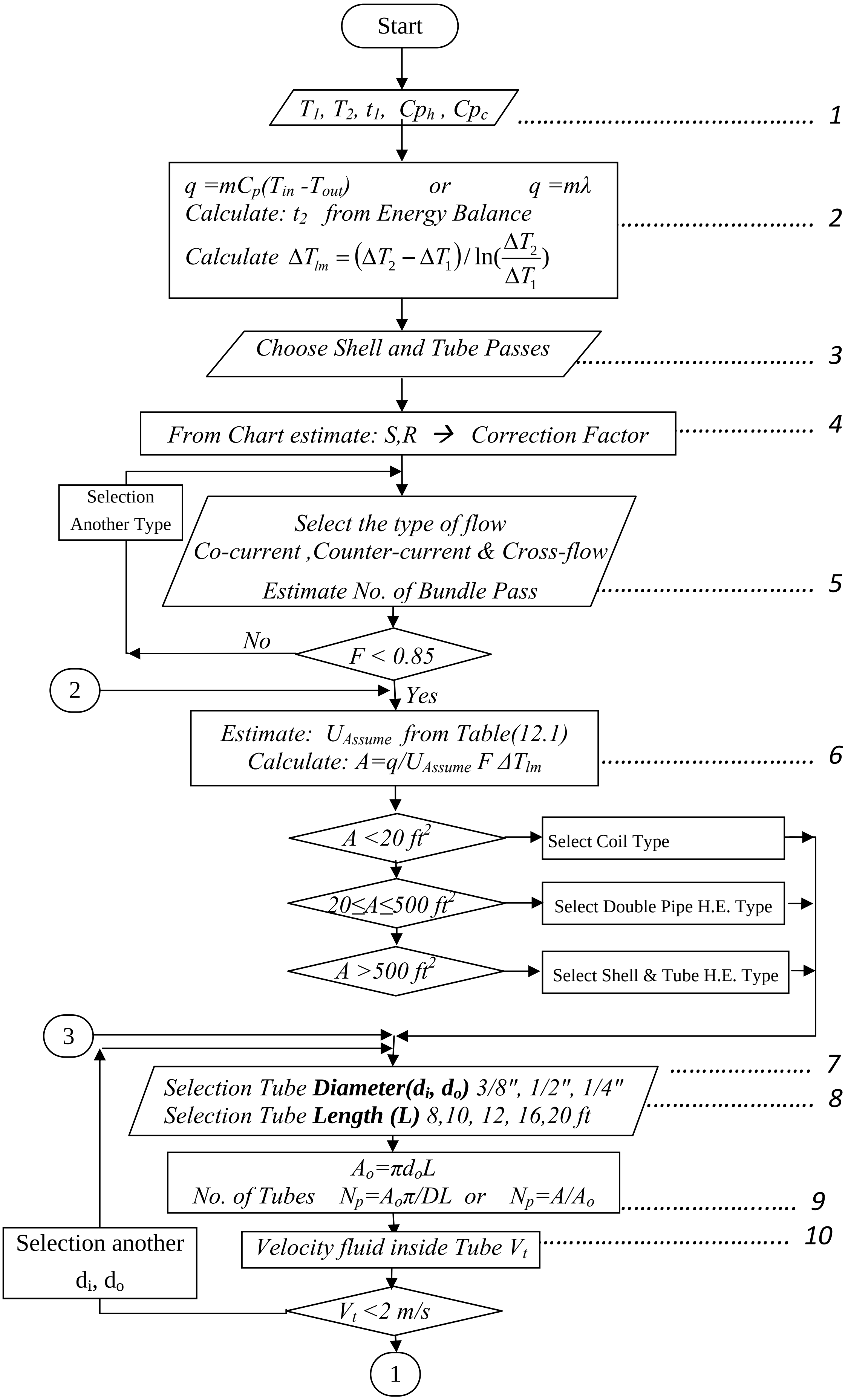
Input (know)	Output (unknown)
T_{ci}	T_{ho}
T_{hi}	T_{co}
L	d_i
m_h	d_o
m_c	ΔP_{tubes}
	ΔP_{shell}
	Q

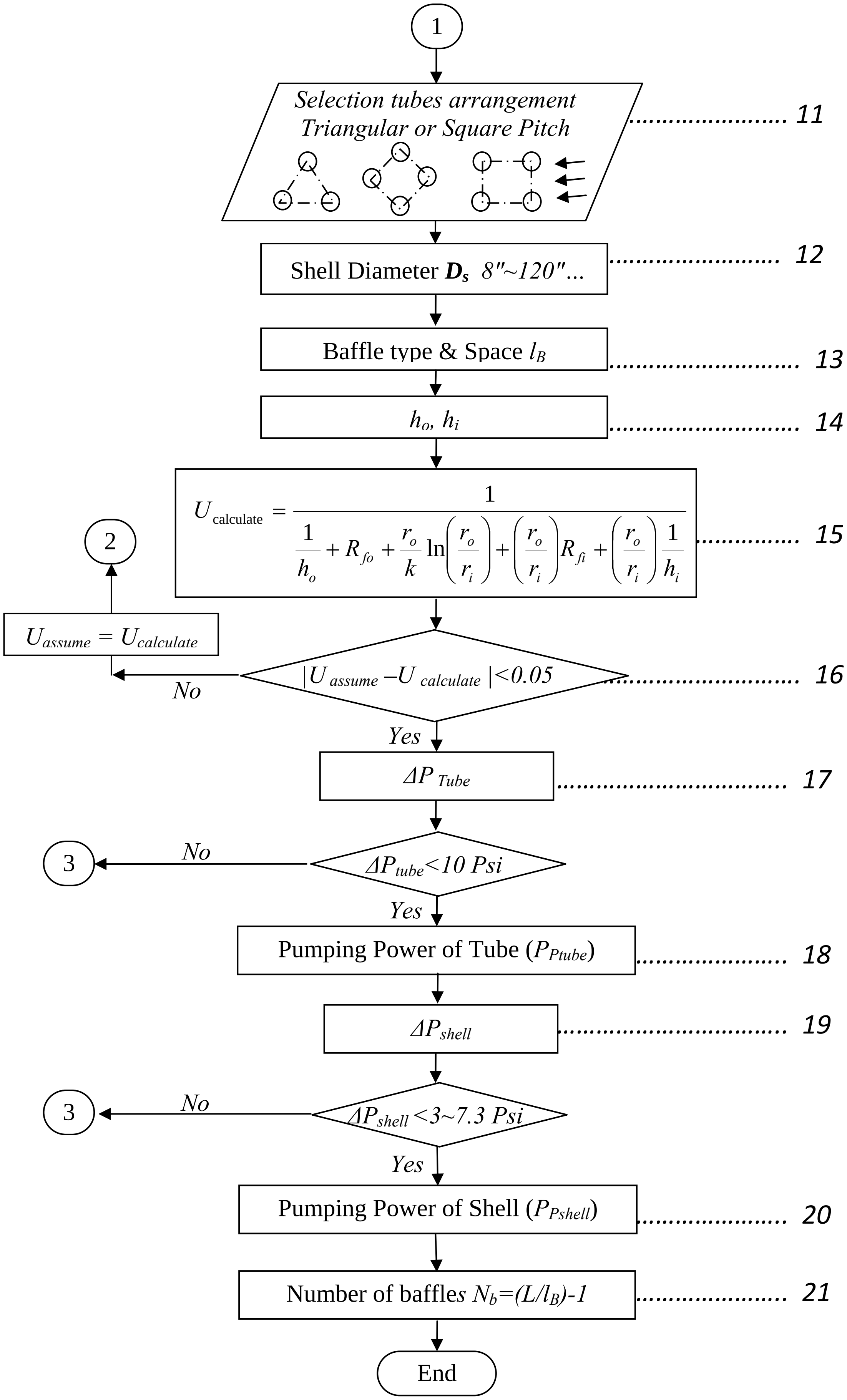
4. F-Method

When all of the terminal temperatures are known, than;

$$Q = UAF\Delta T_{lm}$$

9. Heat Exchanger Design by Kern Method




1
Selection tubes arrangement
Triangular or Square Pitch
11
Shell Diameter D_s 8"~120"...
12
Baffle type & Space l_B
13
h_o, h_i
14
$U_{calculate} = \frac{1}{\frac{1}{h_o} + R_{fo} + \frac{r_o}{k}\ln\left(\frac{r_o}{r_i}\right) + \left(\frac{r_o}{r_i}\right)R_{fi} + \left(\frac{r_o}{r_i}\right)\frac{1}{h_i}}$
15
2
$U_{assume} = U_{calculate}$
$|U_{assume} - U_{calculate}| < 0.05$
No
16
Yes
ΔP_{Tube}
17
3
No
$\Delta P_{tube} < 10$ Psi
Yes
Pumping Power of Tube (P_{Ptube})
18
ΔP_{shell}
19
3
No
$\Delta P_{shell} < 3\sim7.3$ Psi
Yes
Pumping Power of Shell (P_{Pshell})
20
Number of baffles $N_b = (L/l_B) - 1$
21
End

Step 7. Choose tube type d_i and d_o

Table 12.3. Standard dimensions for steel tubes

Outside diameter (mm)	Wall thickness (mm)				
16	1.2	1.6	2.0	—	—
20	—	1.6	2.0	2.6	—
25	—	1.6	2.0	2.6	3.2
30	—	1.6	2.0	2.6	3.2
38	—	—	2.0	2.6	3.2
50	—	—	2.0	2.6	3.2

Assuming tube length (L) (*6, 8, 12, 16*) *ft*

$A_o = \pi d_o L$

Number of tube $N_t = A/A_o$

Step 12. Calculate the shell diameter D_s

a. $$Ds = 0.637\sqrt{\frac{K}{CT}\left[\pi D_O^2 N_t (PR)^2\right]}$$

where K is the Tube layout constant

K=1 for 90° and 45°

K=0.87 for 30° and 60°

CT is the tube count constant

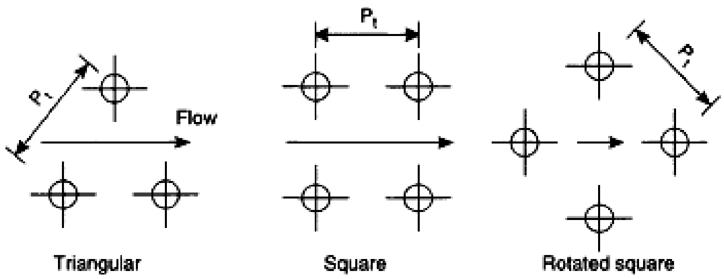
CT = 0.93 one tube pass

CT = 0.90 Two tube pass

CT = 0.85 Three tube pass

PR is the tube Pitch ratio = P_t/d_o

P_t is the tube Pitch



Fig(13) *Tube patterns*

b. *Bundle Diameter D_s*

$D_s = D_b + Clearance$

$D_b = d_o(N_t/K_1)^{(1/n1)}$

Where K_1 and n_1 are constant from Table 12.4

Table 12.4. Constants for use in equation 12.3

Triangular pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.319	0.249	0.175	0.0743	0.0365
n_1	2.142	2.207	2.285	2.499	2.675
Square pitch, $p_t = 1.25d_o$					
No. passes	1	2	4	6	8
K_1	0.215	0.156	0.158	0.0402	0.0331
n_1	2.207	2.291	2.263	2.617	2.643

and clearance between bundle and the shell estimate from fig. (12.10)

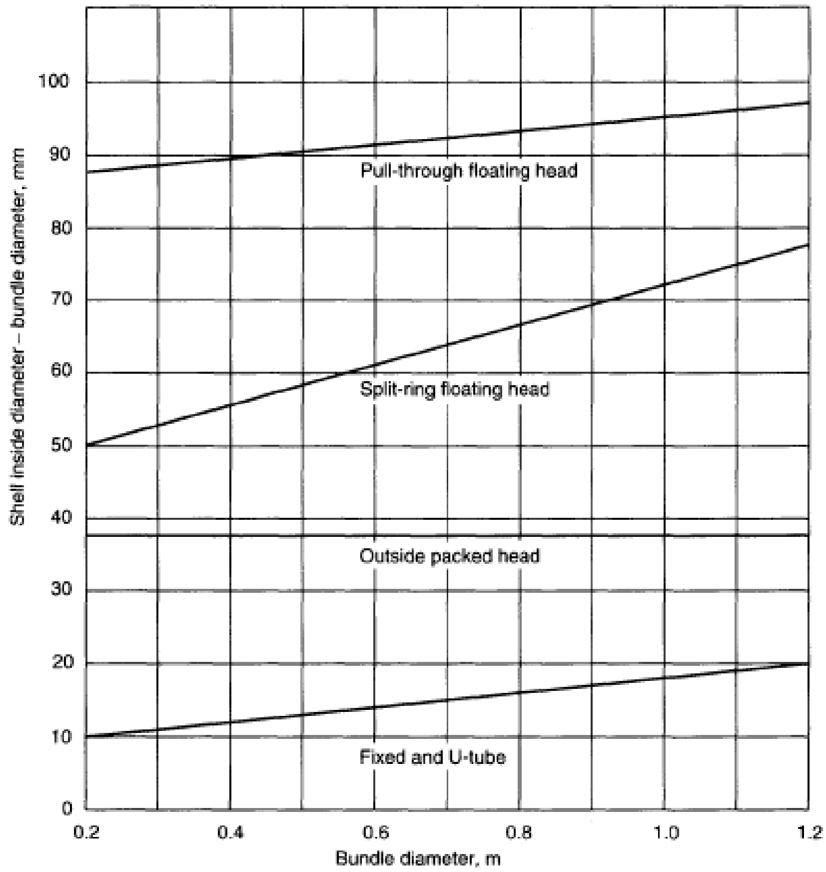


Figure 12.10. Shell-bundle clearance

Step 13. Baffle type & Space l_B

$$l_B = \frac{Ds}{5}$$

or

$$l_B = (0.3 - 0.5)Ds$$

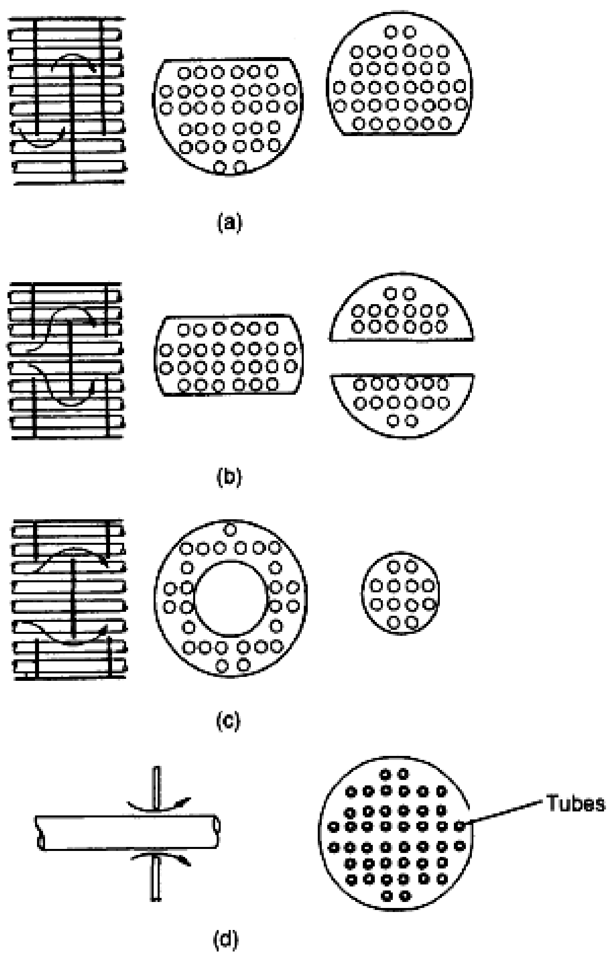


Figure 12.13. Types of baffle used in shell and tube heat exchangers. (*a*) Segmental (*b*) Segmental and strip (*c*) Disc and doughnut (*d*) Orifice

Step *14. Calculate heat transfer coefficient for inside tube h_i*

$$Nu = \frac{hidi}{K_f} = j_h \operatorname{Re} \Pr^{0.33} \left(\frac{\mu}{\mu_w} \right)^{0.34}$$

j_h : heat transfer factor from *fig.(12.23)*

K_f Thermal conductivity of the tube side fluid

μ_w *:* Viscosity of the fluid at wall temperature T_w

$$T_w = \frac{1}{2}\left[\frac{T_{ci} + T_{co}}{2} + \frac{T_{hi} + T_{ho}}{2} \right]$$

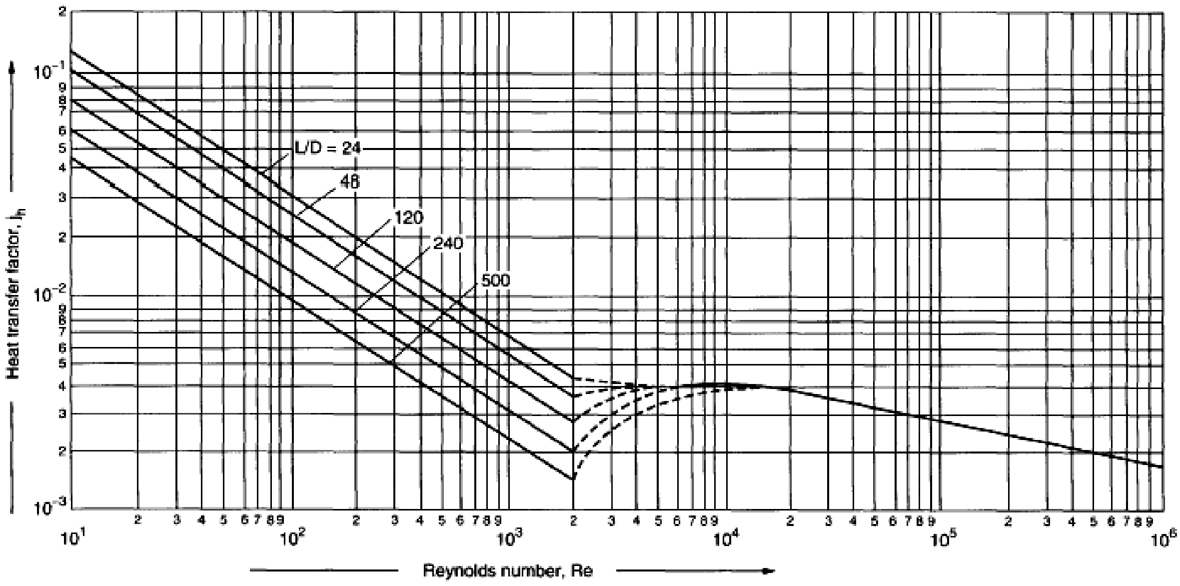


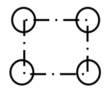
Figure 12.23. Tube-side heat-transfer factor

Step *14. Calculate heat transfer coefficient for outside tube h_o*

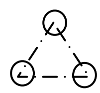
$$h_o = \frac{0.36 k_h}{D_e} \mathrm{Re}_{shell}^{0.55} \mathrm{Pr}^{1/3}$$

Where D_e is the equivalent diameter

$$D_e = \frac{1.27}{d_o}(P_t^2 - 0.785 d_o^{\,2}) \quad \textit{for square Pitch}$$



$$D_e = \frac{1.10}{d_o}(P_t^2 - 0.917 d_o^{\,2}) \quad \textit{for Triangular Pitch}$$



$$\mathrm{Re}_{Shell} = \frac{m_h D_e}{A_s \mu_h}$$

A_s is the shell cross flow area

$$A_s = \frac{(Pt - d_o) D_s}{Pt} l_B$$

Step *17. Calculate tube side pressure drop*

$$\Delta P_{tube} = \left[\frac{4 f_i L N_p}{di} + 4 N_p\right] \rho_c \frac{V_c^2}{2}$$

$$fi = (1.58 \ln(\mathrm{Re}_{tube}) - 3.28)^{-2}$$

Step 18. The pumping power (P_{Pt}) for tube side

$$P_{Pt} = \frac{m_c \Delta P_t}{\rho_c \eta_p}$$

Where η_p is the pump efficiency

$$\eta_p \approx 0.8$$

Step ***19. Calculate shell side pressure drop***

$$\Delta P_{shell} = 8 j_f \left(\frac{Ds}{de}\right)\left(\frac{L}{l_B}\right)\left(\frac{\rho V^2}{2}\right)\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

Where j_f is the fraction factor from *fig.(12.30)*

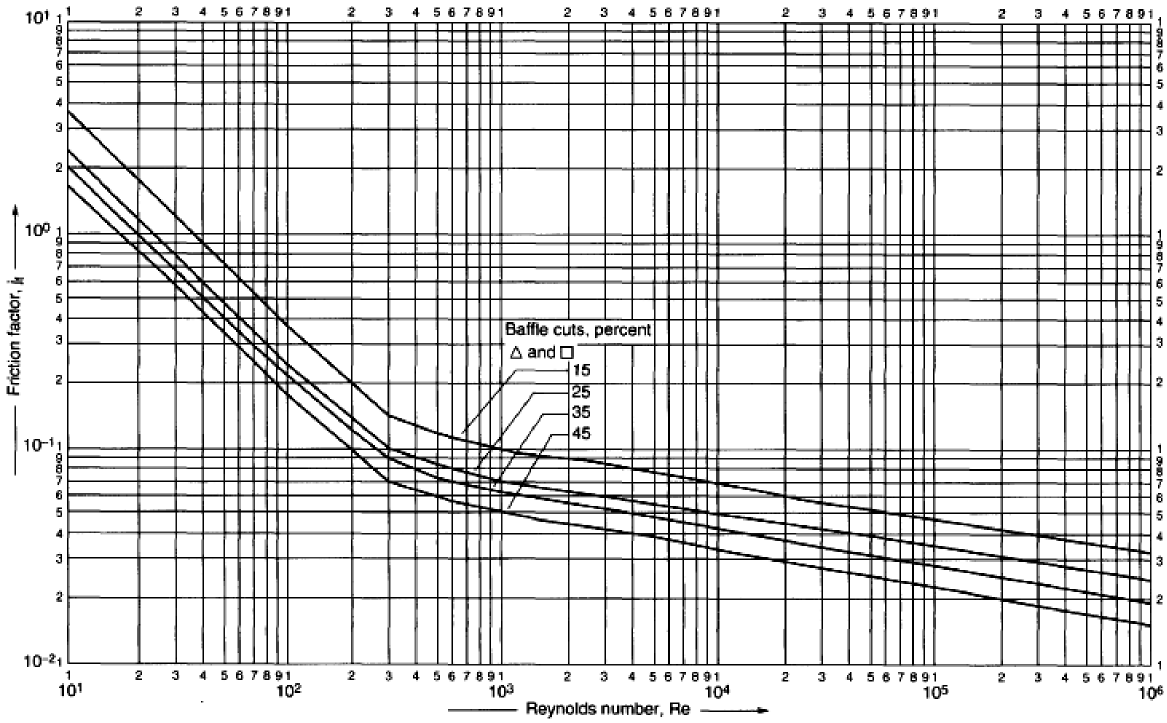


Figure 12.30. Shell-side friction factors, segmental baffles

Step ***20. The pumping power (P_{Pshell}) for shell side***

$$P_{Pshell} = \frac{m_h \Delta P_{shell}}{\rho_h \eta_p}$$

Step 21. Calculate the number of baffles

$$N_b = \frac{L}{l_B} - 1$$